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Stasis PCM Test Results

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STASIS GROUP

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Executive Summary

A phase change material product distributed by Stasis Group, designed to reduce the cooling energy requirements of multi-story commercial buildings, was tested in a side-by-side comparison at FLEXLAB® (LBNL, 2018), a calibrated test facility for low energy building technology solutions. The product was installed according to Stasis recommendations in the ceiling plenum of a 600 ft^2 cell representative of an office space, while an identical cell was operated in the same conditions, except for the night-time pre-cooling sequence and without the product, to gather baseline data. The cells were operated to replicate summer conditions of a candidate office building for retrofit. For the tests conducted at FLEXLAB, two Phase Change Material (PCM) compositions were tested, with the second product having an upgraded composition for thermal performance and upgraded packaging that was used to meet with fire safety requirements for a plenum application.

Two rounds of tests were performed, which were designed to collect data for both core (designated as Round 1) and perimeter (Round 2) office space conditions. Following the analysis of the data collected during Round 1, the experimental design was modified to better represent the thermal conditions of a typical office space in relation to the PCM performance. A new product was also brought in for testing in Round 2, which had an improved operational temperature range and packaging properties. In Round 1, multiple tests were conducted to evaluate the performance of the product under different conditions of internal loads, temperature setpoints, pre-cooling strategies and temperature control strategies. In Round 2, the internal conditions were kept constant, while the test conditions only differ in the amount of PCM installed in the test cell plenum and the length of the nighttime pre-cooling sequence.

The performance benefits of PCM are generally two-fold: the material may reduce peak cooling energy use during daytime hours (a peak demand energy use benefit) and may allow for energy savings with pre-cooling at night through night flush HVAC controls when night time outside air temperatures are lower, and fan energy use for cooling is more efficient than the use of daytime compressor-based cooling. The focus of this research project was mainly on quantifying the energy savings potential of the latter case although some effects on peak reduction were noted. Between the cell with PCM and nighttime precooling and the reference cell without PCM or precooling, a reduction in cooling load during occupied hours was noted between 7.6 kWh (or 22% of occupied hours cooling load) and 12.5 kWh (or 33% of occupied

hours cooling load). Future work can further evaluate the peak cooling load reduction benefits of this product in more detail.

In Round 1, the results for the original product tested showed limited potential for shifting cooling load between occupied hours and nighttime, when compared to a baseline office that uses the same pre-cooling strategy without PCM. A significant shift in cooling load can be observed when the baseline cell does not use any pre-cooling strategies. That shift could be attributed to the thermal mass of the envelope of the cell or from the sensible and latent heat of the PCM, but the sources of the shift are not easily distinguishable.

The test conditions that were developed during the project proposal stage were investigated and we discovered by going in the test cell during different stages of the cycle and physically testing the material, that the PCM product was not performing at the freeze and melt conditions expected. Further tests were conducted to investigate the discrepancies between performance and expectations. A combination of factors was identified as the source of the limited energy savings and those factors were helpful in determining the potential issues that could hinder the performance of the PCM when installed in real buildings. This work was used to design product improvements and the re-design of the second round of experiments. Overall, the results of the Round 1 testing provided valuable insights into experimental design conditions to help discern the PCM's performance, as well as the desired thermal performance of the PCM product, which informed the development of the improved PCM product.

In Round 2, the combination of PCM and nighttime pre-cooling was shown to provide load shifting, which can in part be attributed to the PCM. This load shifting from daytime to nighttime can be used to reduce the HVAC energy use, by replacing mechanically cooled return air with outside air at night, and the daily cost of energy, assuming lower rates at night (not evaluated in this report). Compared to the baseline case, which did not include a pre-cooling strategy, the PCM plus pre-cooling strategy resulted in 12-18% daily HVAC energy savings.

This report highlights the results of Round 2, which reflect the performance of the modified PCM product when operated in suitable conditions. The results of Round 1, along with an analysis of the experimental data to identify the factors that prevented the product from performing as desired are presented in Appendix B.

The following table gives an overview of the results obtained during the Round 2 experiments. The results are given as the daily cooling load (including pre-cooling) and as the HVAC energy, which includes fans and cooling energy, assuming a cooling efficiency of 1 kW/RT (i.e. 1 kW of electricity is needed to produce 1 refrigerant ton – or 3.51 kW of cooling) and assuming that pre-cooling is done with 100% outside air, which corresponds to the optimal condition for cooling energy savings. In practice, the chiller plant efficiency will vary depending on several factors, including outside air temperature, however the use of a varying efficiency was outside of the scope of this study, and should be the focus of future work. In Table 1 'Ref' and 'Test' refers to the reference cell (without PCM) and the test cell (with PCM). Positive savings means that the test cell has a lower HVAC consumption than the reference cell. Overall, the PCM product used in Round 2 demonstrated a savings ranging from 12% to 18% depending on the amount of installed PCM, and the length of time the test cell used night time outside air-based pre-cooling.

TABLE 1 : ROUND 2 RESULTS SUMMARY

Experiment conditions				Daily Cooling load [kWh]			Daily HVAC Energy [kWh]			
Date	ID	PCM Qty	Pre-Cooling in Ref. Cell	Ref. Cell	Test Cell	Diff.	Ref. Cell	Test Cell	Savings	Savings [%]
9-Dec	T1	100%	No	36.0	43.5	-7.5	15.3	13.4	**1.9**	**12%**
10-Dec	T1	100%	No	36.5	41.3	-4.8	15.4	13.1	**2.3**	**15%**
11-Dec	T1	100%	No	38.4	41.1	-2.7	15.8	13.1	**2.7**	**17%**
14-Dec	T2	83%	No	38.0	41.6	-3.5	15.7	13.2	**2.5**	**16%**
15-Dec	T2	83%	No	36.9	39.7	-2.8	15.3	12.6	**2.7**	**18%**
16-Dec	T3	83%	No	36.4	38.1	-1.8	15.2	12.8	**2.4**	**16%**
19-Dec	T4	65%	No	35.3	37.1	-1.8	14.6	12.8	**1.8**	**12%**
20-Dec	T4	65%	No	34.5	38.0	-3.4	14.8	12.9	**1.8**	**12%**

Introduction

'Phase Change Material' (PCM) is the term used to define chemical components designed to freeze and melt at different temperatures that is suitable for a given application. The heat released or stored by the material during freezing and melting (respectively) is significantly higher than the heat used to change the component temperature. PCMs are used in a variety of applications where tight temperature control is desired, such as pharmaceuticals, food storage (Oro, de Gracia, Castell, Farid, & Cabeza, 2012), solar panels (photovoltaic (Huang, Eames, & Norton, 2004) and thermal (Tian & Zhao, 2013)) and building envelope (Kuznik, David, Johannes, & Roux, 2011).

Stasis Group is an integrator of a PCM product aimed at reducing the mechanical energy used for building conditioning. In a typical office building operating in Southern California, the energy required for space cooling represents about 21% of the total energy use (source: California Commercial End-Use Survey) and corresponds to the second most important end use of electricity close behind lighting. The ventilation end-use, which grows with the cooling load in systems where the fans are controlled to meet the cooling demand and participates in the overall HVAC energy consumption, represents about 18% of the total energy use. Technologies to reduce cooling energy can help a building owner save on their energy bills and reduce the carbon footprint of the building stock.

Stasis Group contracted Lawrence Berkeley National Laboratory (LBNL) to conduct tests of two versions of their PCM product (Rounds 1 and 2 respectively). LBNL has the capability to perform side-by-side evaluations of building technologies using FLEXLAB, a test facility of multiple testbeds that each have two identical cells.

Objective and Purpose

The purpose of this project was to evaluate the efficacy of a new PCM product to shift the cooling load from daytime to nighttime, when chiller efficiency is potentially higher, or when the use of an economizer can reduce the need for compressor-based cooling and electricity is typically cheaper. It should be noted that some utilities with deeper renewables penetration are exploring different rate structures with decreased daytime energy use rates. Analysis of this condition was outside the scope of this study.

The role of the PCM is to absorb heat during the day through the melting of the PCM and release heat at night by freezing the PCM. This experiment compared a FLEXLAB test cell equipped with a PCM with an identical cell, operated in the same conditions without the product and the associated nighttime cooling sequence. A comparison of the hourly cooling load and the daily HVAC energy between those two cells provides an estimate of the cooling thermal load and energy savings. The change in thermal load can be converted into site energy savings by assuming the HVAC efficiency.

The objective is to quantify the potential reduction of daytime cooling load enabled by the PCM and to translate that metric into electricity savings by assuming that nighttime pre-cooling is operated with unconditioned outside air. The temperature profile of the air along its return path and of the surface of the PCM was also studied to provide input on the optimal conditions in which the product operates. This allows us to validate the freezing and melting point of the product and quantify its heat storage capacity. The tests were performed in conditions made to replicate the loads (internal and external) seen both a typical core office space (Round 1) and

south-facing perimeter office space (Round 2) during a hot summer day in Southern California, Burbank climate (maximum dry bulb of 31.6°C and minimum of 12.3°C).

Round 1 Testing Summary

Following from the Objective and Purpose of this work, Round 1 testing aimed to determine the cooling load reduction enabled by using the StasisPCM solution in the plenum of a multi-story building core office space. Compared to an office space that does not use night air pre-cooling (also known as 'purge'), the PCM cell with the outside air pre-cooling strategy had an estimated daily savings of 20% to 25%. The Round 1 test data indicated however that the PCM product tested did not have the physical performance that Stasis had expected. In fact, the product tested had a fabrication error resulting in the wrong chemical composition. Consequently, these tests and results are not indicative of their manufactured product. The tests were of interest however in providing insight into the expected temperature conditions in the return plenum during daytime and nighttime pre-cooling, in evaluating the impact of the exposed thermal mass elements in the cell and validating the energy savings potential of night-time pre-cooling strategy using outside air. Energy savings were observed to come mainly from the pre-cooling strategy, storing much of the cooling in the thermal mass of the test facility envelope, which might have prevented the ceiling plenum from reaching warmer temperatures, and consequently lessened the potential for the PCM product to store and release heat.

The test data was analyzed to determine the factors that explained the differences between this and the expected test behavior. The following factors were identified, in order of assumed importance from greatest to least:

- The floor slab in the test cell acted as a thermal heat sink, preceding the PCM in the path of the supplied air, and by absorbing available cooling first thus reduced the potential for coolth storage by the PCM during the night and heat storage in the day.
- The phase change material composition had a higher freezing and melting point than had been claimed. The freezing and melting necessary for the product to work to its full potential did not happen in the ranges of temperature used in the experiments, but partial phase transition may have occurred during testing.
- The film used for packaging the product had a higher thermal resistance than expected, resulting in lower heat transfer between the product and the return air. This film was required, however, to meet fire safety requirements.
- A few days should be used at the start of each test to make sure that both cells are in a stable condition, including the temperatures of any thermal mass elements, especially when transitioning from cases where pre-cooling went from being used to not used.

- The plenum space in FLEXLAB includes wood beams run at regular spacing. There was some speculation that air velocities in the return air plenum might have been lower than had the plenum height been restricted to a space only below the beams. This would have the effect of a reduced air velocity around the PCM, resulting once again in a lower heat transfer between the product and the return air. In Round 2 this was investigated further however and was determined not to be a significant factor.

Those factors were corrected in Round 2 with the experiments focusing on a new improved PCM product with a different composition. The Round 1 results also influenced Round 2 testing by identifying the need for added insulation installed on top of the floor to reduce the thermal mass impacts. In practice, other design strategies may be used in a ceiling plenum PCM application to ensure that supplied outside or cooled air to charge the PCM is applied more directly to this location – such as using automated supply dampers discharging to the ceiling plenum. A complete description of the Round 1 test methodology, results and analysis is presented in Appendix B.

Round 2 Methodology and Experimental Conditions

Methodology - Variables of Interest

The thermal cooling load is the first main variable of interest, which is the cooling supplied by the air to maintain the cell setpoint. This variable is measured at the inlet and outlet of the hydronic (waterside) cooling coil and at the supply and return on the airside system. Part of the waterside cooling load is used to compensate for the heat released by the fan. Therefore, the airside heat energy measured is equal to the waterside energy plus a fraction of the fan energy. The fraction of fan energy released into the airstream was determined to be ~80% during calibration, by balancing the airside and waterside energy.

To lessen the effects of potential measurement error, the variable of interest for this experiment is taken as the average between the airside energy and the computed waterside plus fan energy. This waterside/airside average heat rate corresponds to the amount of heat removed from the space by the chilled air.

HVAC total energy use is also a key metric of this study. In this report, “energy” represents electrical energy since there is no gas-based source of equipment in our cooling system, and no heating was used. The HVAC total energy is computed for the chilled water system by assuming a 1 kW/RT (or 285 $W_{elec}/kW_{cooling}$ or COP of 3.51) chilled water production efficiency on the energy transferred at the cooling coil, plus the addition of the fan electricity consumption. In practice, the chilled water plant energy will vary based on outside air temperature, equipment

efficiency and pump controls strategies, however this analysis was not part of the scope of this study.

The hourly cooling load, HVAC energy and the total daily energy use was used to evaluate the performance of the PCM and validate test conditions.

This research also measured the temperatures at key plenum and PCM locations to document their test conditions. Temperatures at the surface of the PCM were measured during the benchmarking test where the product was subject to a higher temperature gradient than during the regular test. We also measured the temperature of the air at varying elevations around the PCM, as mounted in the plenum on vertical rods, to determine the temperature gradient around the product.

Finally, we measured the temperature of the air along the path from the air supply and return. To determine which areas had the most heat gain and loss, we examined the supply air temperature, the vertical air temperatures in the cell and the plenum and the return air temperature to determine those areas where the air lost or gained most of its heat.

The data were collected across multiple days, for a total of 8 days of experiments.

Conditions

In Round 2, the PCM was tested in conditions replicating a south facing perimeter zone in an office building in Southern California. A model was developed using Carrier Hourly Analysis Program (HAP) (Carrier, 2018) to determine the heat loads (internal and external) experienced in a typical office in this location during a typical summer day. In our experimental conditions, the testbed was insulated from the outside and the windows of the test facility were covered and insulated to reduce heat gains or losses through the envelope of the test cell. All the heat loads determined by simulation, which mostly occur via conduction through the opaque surfaces and via solar radiation through the windows, were replicated by internally mounted and controlled light fixtures and heaters. The adjustment load had a peak intensity of 3516 W, as quantified by the Carrier HAP analysis.

The internal load density was chosen to represent an open-space in a recently-constructed office building with efficient equipment. The values for these internal loads were provided by the product manufacturer to best represent the conditions of the market they are targeting:

- Lights with a power density of 0.5 W/ft² (5.38 W/m²)
- Plug loads with a power density of 0.75 W/ft² (8.07 W/m²)
- Occupant density was 10 people for 1000 ft² (10.8 people for 100 m²) exerting 68 W/person

For a 600 ft² spaces (which is the area of a cell in FLEXLAB), the peak internal load is therefore 1158 W. A detail of the internal and external load schedules and intensity is described in a following section.

Testbed Description

The tests were performed in FLEXLAB, a state-of-the-art experimentation facility designed to evaluate the performance of energy solutions applied to commercial building spaces. FLEXLAB is composed of four life sized testbeds that are each composed of two side-to-side identical cells that represent a typical commercial building space. Different elements of the envelope and interior layouts are customizable depending on the need of the product tested.

For this experiment, the tests were conducted in the rotational testbed, which was built on a rotating platform to allow testing with different solar orientations. In our case, windows were covered and insulated to prevent solar gain in the space.

Dimensions and envelope

Each test cell is 20 ft (6.1 m) wide (from East to West) by 30 ft (9.1 m) long (from North to South), for a total floor area of 600 ft² (55.7 m²). The ceiling height was 9 ft 1 in (2.8 m) with a drop to 8 ft (2.4 m) between the North wall and 5 ft (1.5 m) away from it. The ceiling had a small opening on the South end, with a gap that was 5 in (0.13 m) wide away from the South wall, where it was expected that most of the air supplied to the cell was returned to the plenum. There were no other return grilles located in the ceiling. This return air pathway was chosen as to maximize the length of the return air pathway through the ceiling plenum to the main air handler. The plenum had an average height of 5 ft 2 in (1.57 m), with the upper 1 ft 7 in (0.48 m) crossed by structural wooden beams organized in a large grid. All Round 1 testing was done with this plenum condition. For Round 2 tests, a plastic tarp was hung across the bottom of the beams to prevent any airflow in the upper part of the plenum resulting in an average ceiling plenum height of 3 ft 7 in (1.09 m). See Figure 1 and Figure 2 for more details on the test cell geometry.

The test cells' envelope was insulated to minimize the heat transfer through the exterior walls. Each test cell had a concrete topping slab of 3in height poured onto a metal deck with an average of 7" of rigid insulation. For Round 1 tests, the slab, only covered by a low resistivity carpet, had high thermal coupling with the interior space air temperature. As part of the findings from Round 1 testing, in Round 2 the floor was covered by 2in of polyiso (R-Value of 13.1 h·ft2·°F/Btu) to reduce the thermal mass impacts of the slab on experimental results. The R-value for each element of the test cell is given in Table 2.

TABLE 2 : ROUND 2 WALL, ROOF AND FLOOR R-VALUES

Component	North Wall	South Wall (Opaque + Insulated Window)	Exterior East/West Walls	Roof	Floor
R-Value ($h \cdot ft^2 \cdot °F/Btu$)	4.39 / 4.47	3.4	13.9	3.52	17.41
R-Value ($K \cdot m^2/W$)	24.9 / 25.4	19.0	78.8	20.0	98.9

Considering the area of each surface, this corresponded to a total heat rate through conduction of about 13 W/K. In addition, the infiltration rate in a preliminary test estimated the heat losses to be at about 7 W/K. This total of 20 W/K was balanced by an addition to the internal load calculated hourly in the HAP model.

Internal Loads

The lighting heat loads were produced using four light fixtures that are recessed in the ceiling, with a power of 75 W each, for a total of 300 W. In each cell, the occupant and plug load equipment heat loads, and the external heat load were replicated using nine electric resistive heaters with a thermal output of 135 W each, and two radiant quartz heaters with a thermal output of 1450 W each. The heaters that represented occupant loads were mocked up as cylindrical devices with heat mats mounted in their interior to represent the radiative and convective aspects of occupant heat loads. The total heaters nominal capacity in each cell was 4115 W. The radiant quartz heaters were positioned to deliver heat towards the south façade of the test cell to replicate the effect of sun patches, which would be closer to the glazed surfaces of the replicated space. The heater positions and schedules are shown in Figure 3.

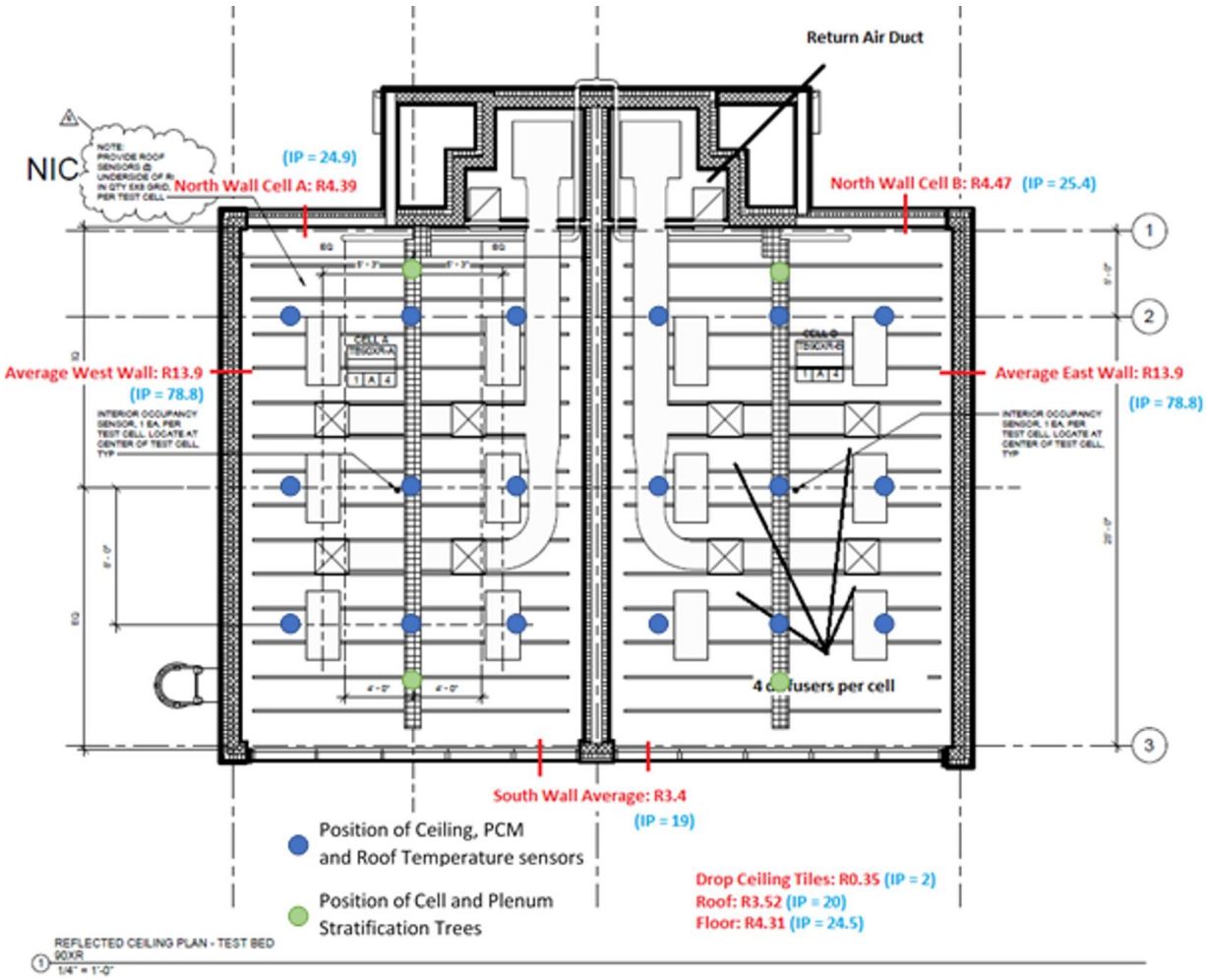


FIGURE 1 : ROUND 2 CEILING PLAN

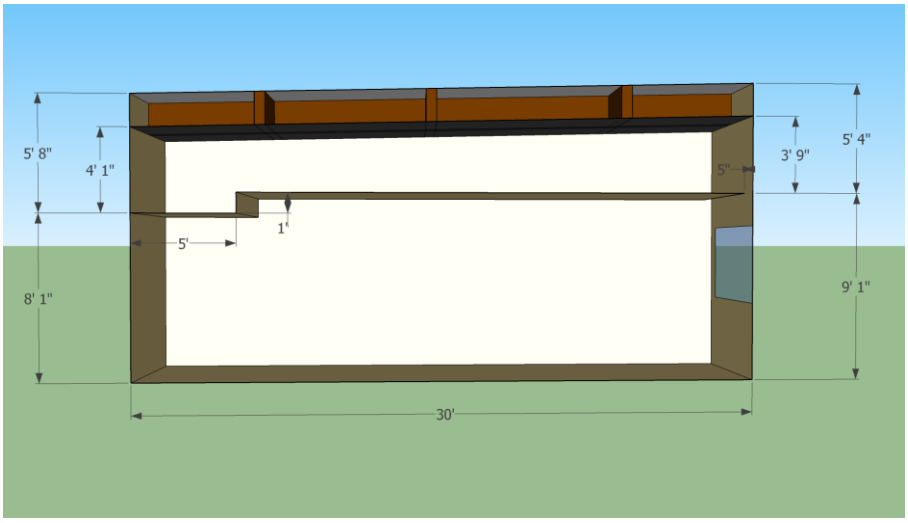
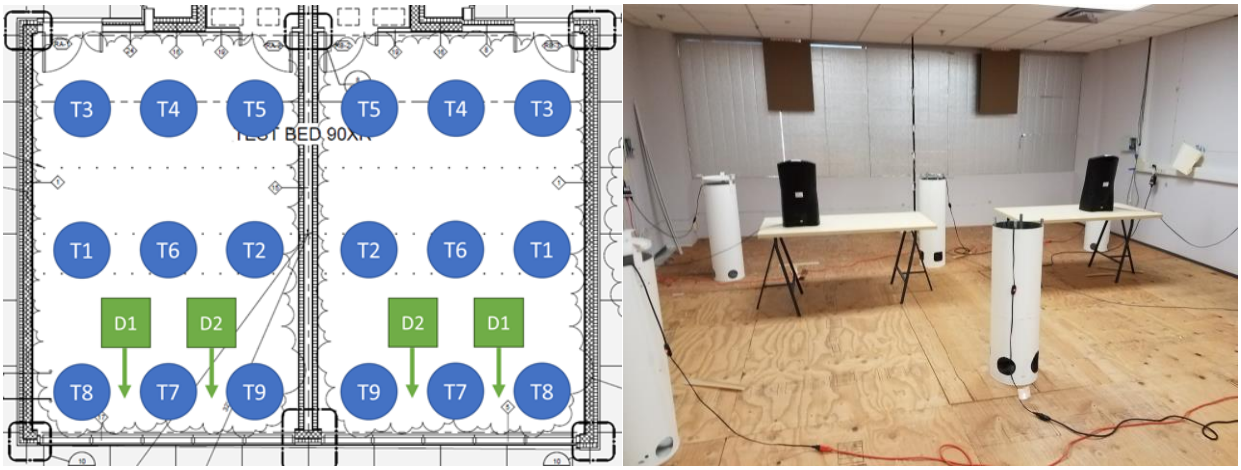


FIGURE 2 : CELL SECTION SHOWING PLENUM DIMENSIONS



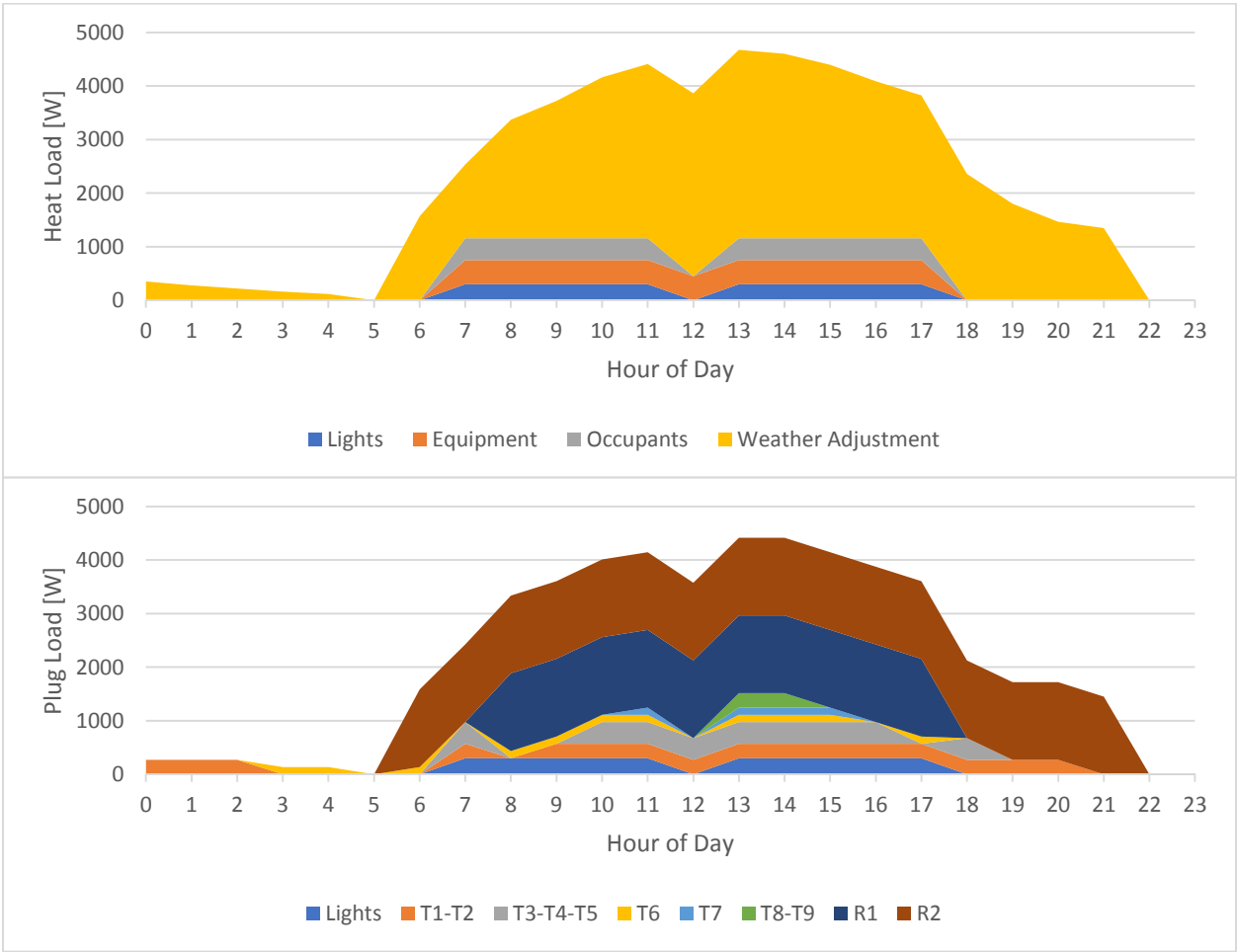


FIGURE 3 : ROUND 2 HEATERS MAP (TOP LEFT) AND PICTURES (TOP RIGHT). REPLICATED LOAD SCHEDULE (MIDDLE) AND HEATERS SCHEDULES (BOTTOM)

HVAC System

For Round 2 FLEXLAB's HVAC system in each cell was controlled as a packaged CAV air handler system with water sourced cooling and heating coils. For this test, during occupied hours, the HVAC was controlled as follows: the air flow rate was constant at 600 cfm (353 cmh), or about 1

cfm/ft² (6.33 cmh/m²), and the cooling coil valve was modulated between open and closed to maintain the zone temperature around the 74°F (23.3°C) setpoint with a fixed 2°F (1.1°C) dead band. When the valve was open, the chilled water was supplied at a constant flowrate of 0.53 cfm (15 lpm) and at a temperature between 50.0 and 53.6°F (10 and 12°C). In this experiment, the economizer was off, and there was no outside air supplied in the cell during occupied hours. This allows for a better control over the supply air temperature.

The introduction of outside air during nighttime pre-cooling was represented by 55°F (12.78°C) supply air temperature, to simulate a typical night time free cooling condition. During unoccupied conditions, after pre-cooling, the fan was off unless the temperature in the space exceeded the setpoint of 82°F (27.78°C), although in no test was additional cooling needed after pre-cooling to maintain the space under this setpoint.

The conditioned air was brought into the cell through four supply diffusers mounted in the ceiling and was returned to the air handler through a return grille mounted in the plenum on the North wall (Figure 1). The expected air path goes from the supply diffuser to the occupied zone in the cell and was returned to the plenum through the South-end gap in the ceiling and travelled through the entire plenum back to the return grille. The supply diffusers in the southern most row had their distribution set to supply air in the east, west and north directions only to prevent supply air from being bypassed directly into the plenum, which was shown to reduce the plenum air temperature during calibration runs prior to the experiment.

Test Schedules

There were three distinct phases during a test that determined the internal load and the HVAC control:

- Occupied hours: from 7am to 6pm PST, where the internal loads are high, and the cell indoor temperature was controlled to maintain comfort for the occupants within a given dead band
- Unoccupied hours: from 6pm to 5am PST, where the internal loads are lower and the cell indoor temperature setpoint was setback to reduce unnecessary energy consumption for cooling
- Pre-cooling: from 12am to 3am or 4am PST depending on the test, where the fan was operated at a high flow rate and the supply air was maintained at 55°F (12.8°C) to simulate a night flush using outside air to remove the heat stored in the PCM.

Sensors and Measurements

More than a hundred sensors were used for this experiment. Each sensor recorded data every minute. There were three types of sensors used in this experiment. Numerous power sensors monitored each electrical load (space heaters and lights) and the AHU fan power. Flow sensors

measured the air flow rate at the supply and return ducts of the air system. Hydronic flow sensors measured the supply of the water to the heating and cooling coil. HVAC and space temperatures were monitored at the air supply and return of the air handler, at the inlet and outlet of the water loops, and at various locations in the cell's occupied zone and in the ceiling plenum (to a total of 104 temperature sensors).

There were 14 temperature sensors total between two vertical rods (7 sensor per rod) located on the south and north end of both cells. Those temperature sensors were mounted below the ceiling at different heights from the floor to determine the temperature stratification of the air in the cell. The average value of two sensors on the walls and one sensor on each rod, at desk level, was used to accurately control the temperature of the cell. While a typical office building would use a single thermostat sensor to control an office space the size of the test facility, using an average of four points of measurements allowed for a better comparison between the two adjacent cells as it reduced the impact of any local thermal anomaly, such as an air draft on the sensor that would occur in one cell and not the other. This method also insured that the test cell air average temperature at desk level was closer to the thermostat reading. At the same location as the cell rods, above the ceiling were 2 vertical rods with 4 sensors each, that were dispersed along the rod evenly. The positions of the vertical rods are marked on Figure 1.

The temperature was monitored on the underside of the dropped ceiling in 9 positions in both cells. Each layer of PCM product tested was monitored using 9 temperature sensors that were installed on the PCM's packaging surface.

The roof interior surface temperature was monitored using 9 temperature sensors in both cells. The position of those 9 locations is marked on Figure 1.

The return plenum entry temperature was monitored by 3 temperatures sensors located in the drop ceiling gap at the south wall, where we measured that about 72% of the air supplied to the room is returning to the plenum, based on air velocity measurements at that location. The rest of the air is assumed to return to the plenum through infiltration around the ceiling tiles.

Product Tested

The product tested in Round 2 was a Phase Change Material product manufactured and distributed by Stasis, sealed in a black-colored packaging and grouped in mats. Two products were tested together in Round 2, which differed in the density of PCM in mats (the composition of the PCM was not disclosed further to LBNL). The higher density mats (0.67 lb/ft²) were installed on a metal grid at about 12in (0.30m) above the ceiling tiles, except where it is not possible (ducts, light troffers…). The lower density mats (0.5 lb/ft²) were installed directly on top of the dropped ceiling panels. Between tests, some of the PCM mats were removed, starting from the North side going towards the middle of the plenum. The quantity of PCM in each test is presented in Table 3.

The product had a rated melting point of 23.2°C and a freezing point of 21.0 °C.

FIGURE 4 : ROUND 2 PCM INSTALLED IN FLEXLAB PLENUM (COURTESY OF ROB MORTON)

Calibration

Before the experiment, the two cells were calibrated by using the same internal conditions and HVAC control as the test cell during testing and comparing the thermal load of the two cells.

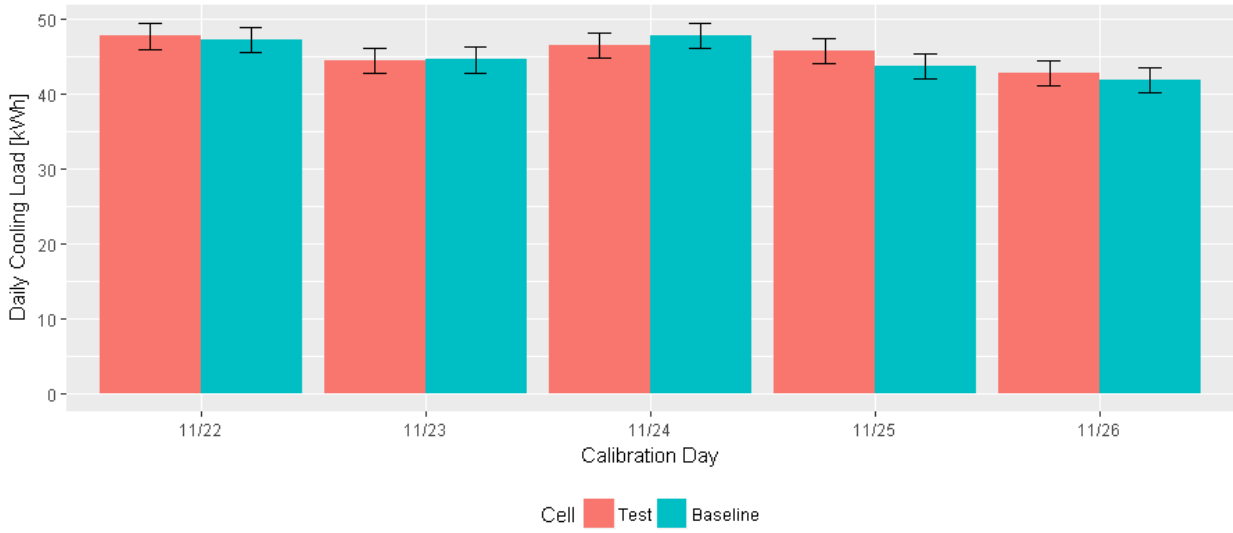


FIGURE 5 : CALIBRATION RESULTS

The purpose of this test run is to determine with the given sets of measurement systems and minor thermal variances between the two test cells the degree to which they match each other under the same conditions, i.e. for comparative test purposes the relative accuracy between the two cells. The plot on Figure 5 shows the total daily cooling load for the test and baseline cell during the calibration runs.

For those 5 days of calibration, the maximum error (i.e. the difference between the daily cumulative cooling load in the test and baseline cell, with the baseline cell load used as reference) was 4.82% (on 11/25), and the minimum was 0.3% (on 11/23). If we only compare the load during occupied hours (i.e. removing the pre-cooling load), the difference between the two cells cooling load range between 0.7% and 5.2%. In terms of hourly load, Figure 6 shows how close the cell loads are to one another if we ignore that the on/off HVAC control can be triggered at a slightly different time in each cell, which makes it difficult to compute metrics on hourly load differences. For this reason, cumulative daily load comparisons between the two cells is the preferred approach to assess relative accuracy between the two cells.

A degree of measurement error will be present in the daily load calculations, based on the inherent accuracies of the measurement devices used. High precision devices included airside temperature sensors with an accuracy of +/- 0.05°C, water temperature sensors with +/- 0.03°C, airflow measurement with +/- 3% of reading, and waterflow measurement with +/- 2% of reading. For all these experiments, this propagates to a maximum airside load calculation

uncertainty of 162 W and a maximum waterside load calculation uncertainty of 115 W. In daily loads, those uncertainties can accumulate up to +/- 1.7 kWh of error or about 4%, which corresponds to the conservative extremities of uncertainty due to sensors error.

The calculation of the uncertainty for the load was made using a first-order Taylor series expansion on the load calculation, using the sensors uncertainty. The uncertainty varies with the temperature and flow rate and the maximum value is shown in this section.

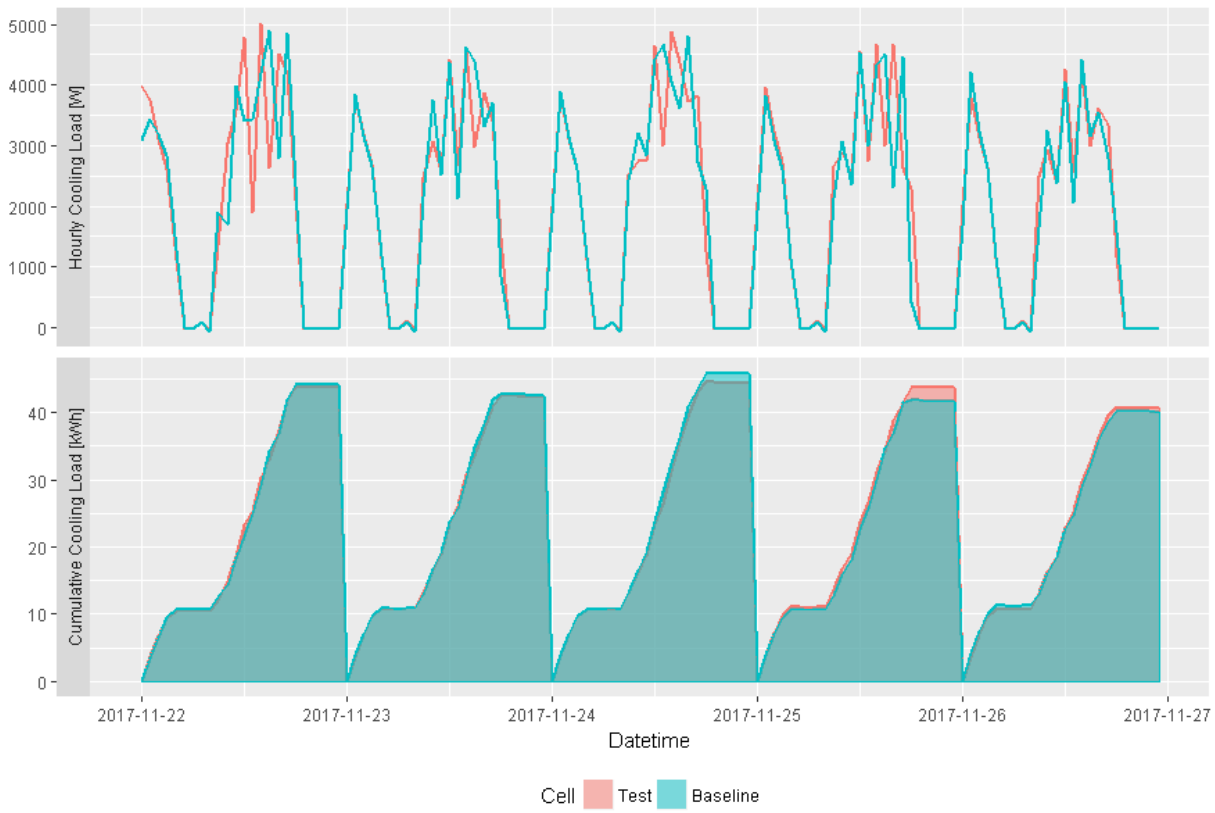


FIGURE 6 : HOURLY COOLING LOAD AND DAILY CUMULATIVE COOLING LOAD DURING CALIBRATION

Round 2 Testing Schedule

The tests described in Table 3 were conducted to determine the performance of the PCM. For those tests, both cells were operated with identical conditions, apart from the pre-cooling period that was only used in the reference cell. In both cells, the economizer was not used, and no outside air was introduced in the cells (except for potential infiltration).

In addition to those tests, we used 5 days before the experiments to calibrate the cells for identical conditions, and 2 days after the installation of PCM to benchmark the performance of the product.

TABLE 3 : ROUND 2 TESTING CALENDAR AND CONDITIONS

Index	Days	Pre-cooling Sequence	High Density (0.67 lb/ft²)		Low Density (0.5 lb/ft²)		Ratio of total PCM mass
			Area [ft²]	Ratio of total ceiling area	Area [ft²]	Ratio of total ceiling area	
T1	9-Dec 10-Dec 11-Dec	12am to 4am	426	71%	524	87%	100%
T2	14-Dec 15-Dec	12am to 4am	376	63%	408	68%	83%
T3	16-Dec	12am to 3am	376	63%	408	68%	83%
T4	19-Dec 20-Dec	12am to 3am	266	44%	358	60%	65%

Round 2 Results

Benchmarking

During benchmarking, the test cell, with PCM installed in the plenum, was operated with a control sequence meant to determine the preferred conditions for the PCM to be working by identifying the freezing and melting temperatures.

During the two days of benchmarking, the fan was run to provide a constant air flow of 1000 cfm (1699 cmh) and the supply air temperature was controlled to reach various setpoints at the plenum entry to test the product under different temperature gradients, with the setpoint being modified every 6 hours. This schedule is documented in Table 4 with the plenum entry setpoints defined as a few degrees higher or lower than the PMC melting or freezing points, as provided by Stasis.

The same schedule was repeated twice and is broken down in the results for each of the four cycles, with each cycle corresponding to a 6-hour long period of heating followed by a 6-hour long period of cooling. During the first two cycles, circulator fans located in the plenum were turned on to increase the convective coefficients at the surface of the product. The fans were turned off during the third and fourth cycles. When the air was heating up, the heaters in the cell were turned on to assist the heating coil, and those heaters were turned off when the air was cooling down.

TABLE 4 : ROUND 2 BENCHMARKING SETPOINT SCHEDULE

Hours of benchmarking	0 - 6	6 - 12	12 - 18	18 - 24
Plenum Entry Setpoint	Melting Point +5K 82.8°F (28.2°C)	Freezing Point -5K 60.8°F (16.0°C)	Melting Point +3K 79.1°F (26.2°C)	Freezing Point -3K 64.4°F (18°C)

Figure 7 shows the air temperature during benchmarking at different locations along the return air path. The plenum entry temperature was measured at the gap between the dropped ceiling and the South wall, were it is estimated that about 72% of the air is returning. The South end and North end of the plenum corresponds to the temperature of the air measured by 3 temperature sensors mounted on vertical rods close to the South and North wall of the cell, and the HVAC return temperature in the duct where the air that travelled through the plenum is returned to the coils.

The red and blue horizontal lines correspond respectively to the rated melting and freezing point of the product tested.

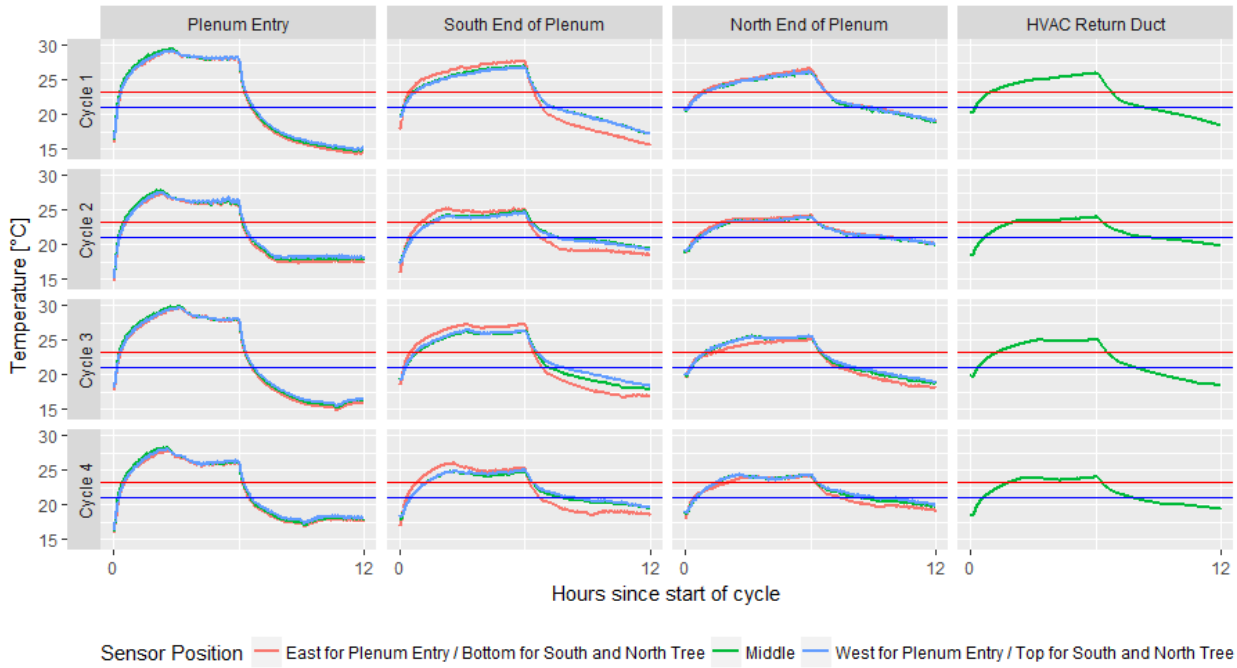


FIGURE 7 : ROUND 2 AIR TEMPERATURE DURING BENCHMARKING

This figure shows that the plenum entry air takes a few hours to reach the setpoint and that the amplitude of the temperature change is dampened as the air crossed the plenum back to the duct. A significant reduction in amplitude is observed between the plenum entry and the South end plenum, which is surprising considering that those sensors were only about 5 ft away from each other. In comparison, a similar reduction in amplitude was observed between the South and North end of the plenum, even though those sensors were about 20 ft away from each other's, and almost no difference was observed between the North end of the plenum and the return air temperature.

No significant differences are observed between the first and second cycles in terms of air temperature.

Figure 8 shows the temperatures measured at the surface of the product during benchmarking. The plot is split into four rows which corresponds to the four temperature cycles of the benchmarking, and in three columns which correspond to different locations along the return air path. The lines' colors correspond to different point of measurements, as documented in Figure 1. The "bottom layer" corresponds to the low density PCM placed on the ceiling tiles and the "top layer" corresponds to the high density PCM placed on the raised metal grid.

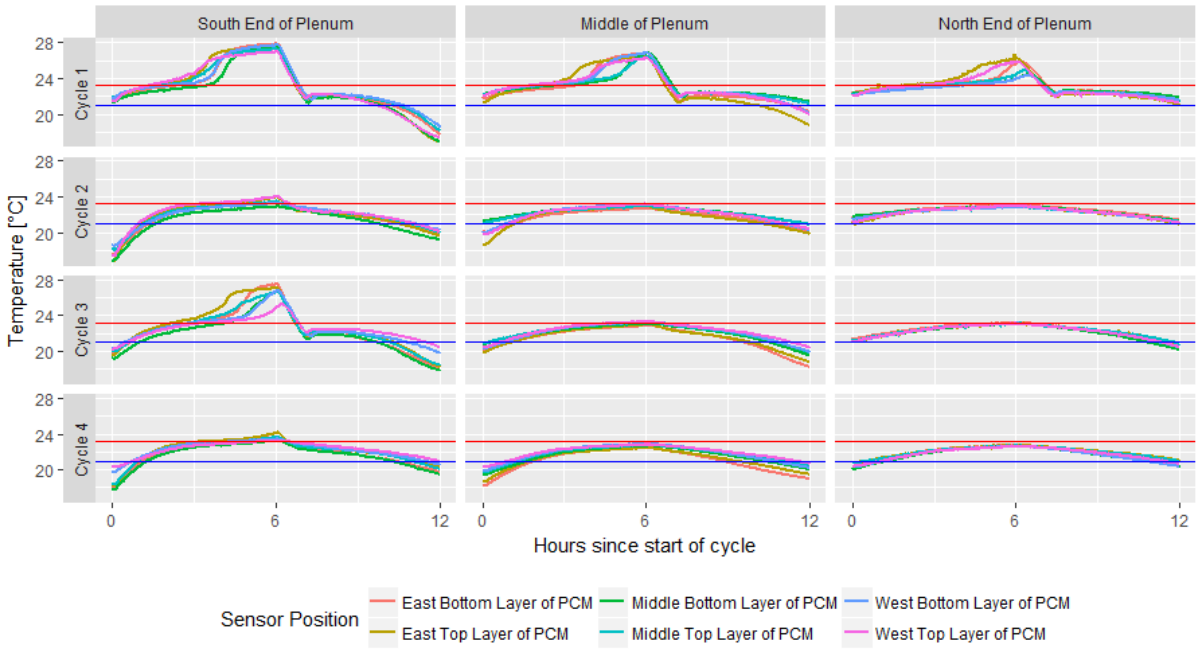


FIGURE 8 : PCM SURFACE TEMPERATURE DURING BENCHMARKING

These plots clearly show the difference of behavior of the PCM when the rising temperature is approaching the melting point (most noticeable on hour 3 of Cycle 1 on the South End of Plenum) and when the falling temperature is approaching the freezing point (most noticeable on hour 7 of Cycle 1 on the South End of Plenum), although it seems that the freezing is happening at a slightly higher temperature than expected. The freezing and melting were not physically confirmed during the tests and are only assumed based on the temperature measurements profile. This behavior happened consistently at each oscillation on the south end of the plenum, but more rarely in the middle and north end of the plenum, since the temperature gradient with the air was lowered as the air loses/gains heat in the early section of the plenum.

The temperature at the surface of the PCM changed very quickly when the product was fully melted or fully frozen (which is most noticeable around hour 6 of Cycle 1 on the South End of Plenum, when it is assumed that the PCM was fully melted), but those temperature changes were a lot slower within the band between the melting point and freezing point, suggesting that the product did not change phase close to those points but rather continuously within those two temperatures, which acts as the extremities of the phase change.

There is little evidence that the plenum fans had a significant effect on the melting or freezing behavior of the product in the first cycles. The difference in behavior between the first and third temperature rise (which had the same setpoint but only the first had fans on), can be explained by the differences in conditions at the start of each phase: the PCM was a lot hotter

during the first rise than during the third one. There were no significant differences between the second and fourth temperature rise and following temperature fall which started in similar conditions, suggesting that the fan had limited impact on heat transfer. Small differences would not justify the installation of fans along with the product in real buildings, and it was decided from that observation that the test would be done with the plenum fans off.

To present the same information differently, the time response of the PCM τ may be considered, which is defined as the ratio between the temperature gradient between the PCM and the air and the PCM temperature time derivative and calculates at each timestep t_i using:

$$\tau(t_i) = \frac{T_{PCM}(t_i) - T_{air}(t_i)}{\frac{dT_{PCM}}{dt}(t_i)}$$

This variable is also representative of the heat capacitance of the PCM by the heat transfer resistance between the air and the PCM but can also be the time it would take for the PCM to reach the air temperature keeping a constant temperature change rate.

For this calculation, the air temperature in the middle of the plenum was assumed to be the average of the temperature measured at the north and south end of the plenum, and the PCM sensors at the east, middle and west locations were averaged for each section along the return path of the air. The time response is plotted against the PCM surface temperature in Figure 9. The plot is separated by location and the color corresponds to whether the PCM was being cooled or heated by the air.

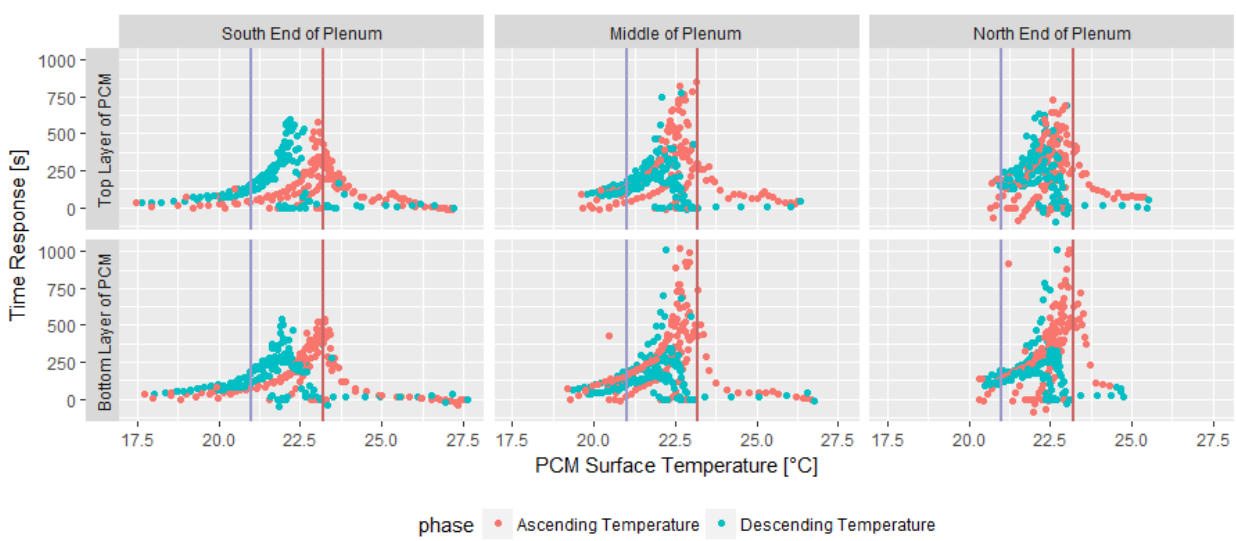


FIGURE 9 : PCM SURFACE TEMPERATURE VS PCM TIME RESPONSE DURING BENCHMARKING

This plot confirms that the rate of temperature change of the PCM was a lot slower (higher time response) between the melting and freezing point, with a peak time response during heating

occurring at a PCM surface temperature close to the melting point, whereas the peak time response during cooling occurred when the PCM's surface temperature was further away from the freezing point. While both heating and cooling peaks were separated by about 1K on the South end of the plenum, they were almost overlapping in the middle and north end of the plenum. This behavior may be explained by the South end plenum PCM fully freezing and melting at each oscillation, whereas the other locations were only partially melted and frozen when cooling and heating alternates. Partially frozen or melted PCM is quicker to return to a previous state when conditions changes.

In conclusion, the benchmarking showed that within a 6-hour period and enough temperature gradient, the PCM may be more thoroughly melted and frozen in the early sections of the plenum, but that the middle and north sections were operating mostly within the band of partial melting and freezing. Nevertheless, even during partial melting and freezing, the heat capacity (proportional to the time response) of the PCM was increased, albeit with a lower total heat released/absorbed. The plenum fans seemed to have negligible effect on the heat transfer or the behavior of the PCM and were therefore were not used again during testing.

Daily Cooling Load Results

The plot on Figure 10 shows the daily total cooling load difference between the two cells. The negative values mean that the test cell had a higher cooling load than the baseline, due to the extra cooling introduced during the nighttime pre-cooling. The plot also shows the ratio of that difference relative to the total daily load in the reference cell. While this is not the main variable of interest for evaluating the product, since the product's main goal is to shift cooling load at night rather than reducing it, this result guides the analysis of the validity of test conditions and of the amount of additional cooling introduced by pre-cooling. The colors of each bars reference the test conditions, which were introduced in Table 3.

In all tests, the test cell had a cooling load higher than the baseline cell by a difference between 4.8% (on 12/16) and 20.8% (on 12/09). While there was not enough data to determine an accurate trend in the daily load profile, we can notice that the highest level of cooling load differences happen with the first test and gets lower with every reduction of the amount of PCM (100% in T1, 83% in T2 and T3, 65% in T4) or the pre-cooling time (4 hours long in T1 and T2, 3 hours long in T3 and T4). Both of those conditions reduced the amount of "coolth" supplied to the space during pre-cooling, the first by reducing the amount than can be stored by reduced PCM, and in the second case by the reduction of the duration of the pre-cooling strategy.

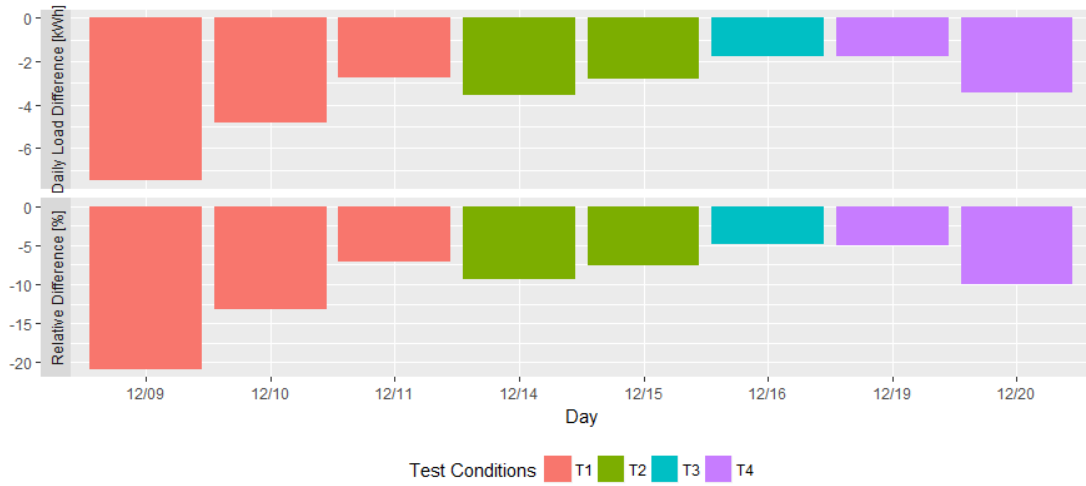


FIGURE 10 : ROUND 2 DAILY CUMULATIVE COOLING LOAD DIFFERENCE

Hourly Cooling Load Results

In this section and the following, the load shift and energy savings provided by the PCM are reviewed. First, the hourly cooling load that was expended in each of the experiments is presented. In the following section, these results are extended to HVAC energy use. The plot in Figure 11 shows the hourly cooling load for each experimental condition by cell, along with the load difference between the two cells (red and green areas).

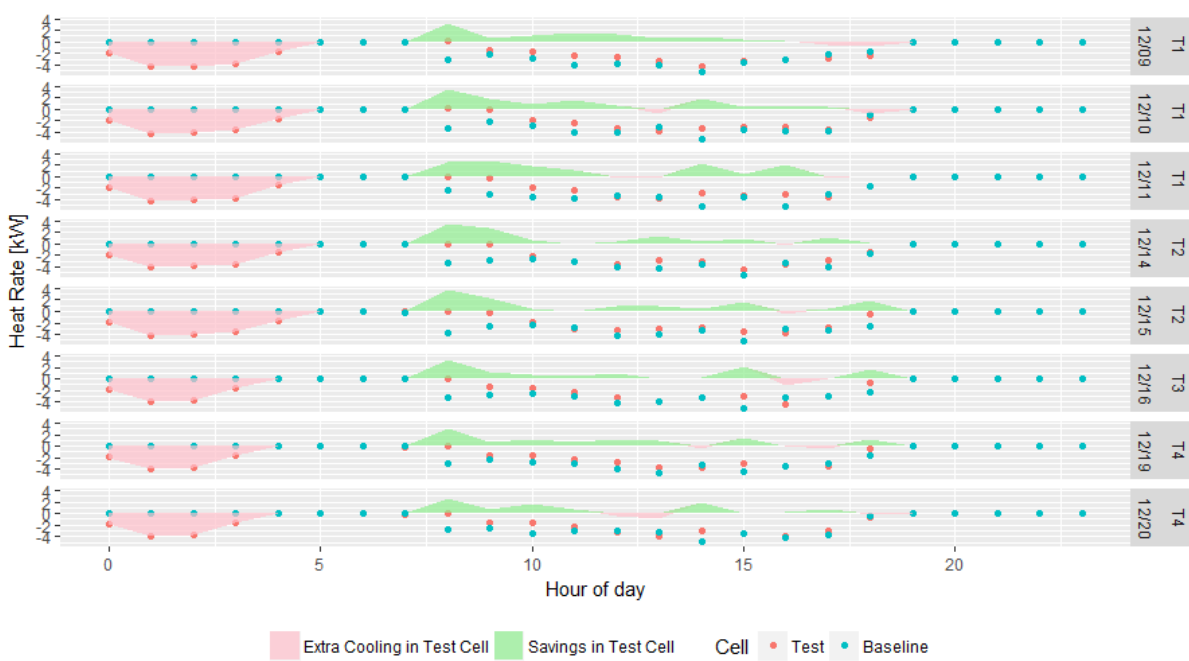


FIGURE 11 : ROUND 2 CELLS' HOURLY COOLING LOAD

If the load difference areas are integrated (the red and green areas), it combines to the daily load difference presented in Figure 10.

In all test conditions, a large strip of negative “savings” is seen at the beginning of the day, with the strips being shorter for the last three days, when the nighttime pre-cooling sequence was reduced by one hour. The test cell used more energy to bring in cool air with the intent to charge (freeze) the PCM, with the idea to enhance the potential for absorbing heat during the day. The stored coolth resulted in a strip of savings during occupied hours, with most of it happening at the beginning of the occupied hours (between 7 and 10am). The strips of savings are lingering until almost the end of the occupied hours, which suggests that the PCM and the test cell envelope are still absorbing heat late in the afternoon.

Another way of looking at the results is by pointing out that the average daily temperature of the test cell, including the PCM, is lower than the baseline cell because of the pre-cooling strategy, and that in combination with an increased overall heat capacity – or “coolth” retention – brought by the PCM, this translated into a lower cooling load seen by the HVAC system.

The reduction of cooling load during occupied hours corresponded to a difference ranging between 7.6 kWh, or 22% of total occupied load (on 12/20, T4) to 12.5 kWh, or 33% of total occupied load (on 12/11, T1) compared to the reference cell without PCM or nighttime precooling. These differences are beyond the threshold of uncertainty determined during calibration (5.2%).

There was no significant difference between T1 (24%, 29% and 33% load reduction) and T2 (30% and 33% load reduction), which only differed by the amount of PCM being used in the plenum. However, the difference between 4- and 3-hour long pre-cooling periods (T2 compared to T3) is noticeable: load reductions dropped from 30% and 33% to 27%, although there was only one day done with T3 conditions. Between T3 and T4, the amount of PCM was further reduced and the load reductions dropped from 27% to 22%.

HVAC Energy Use Results

In this section, the HVAC energy use during the different tests is calculated, assuming a constant chilled water production efficiency of 1 kW/RT (equivalent to a COP of 3.51) is applied to the measured cooling coil load. It should be noted that the results that are presented here are only valid for the assumed chilled water production efficiency and for the actual efficiency of the HVAC fan that was used in this study. Real buildings might have more or less efficient cooling production or ventilation systems.

Figure 12 shows the hourly electricity use for three end-uses: cooling, fans and plug loads (including lights), for both the test cell and reference cell and for all 8 days of test. For plug loads, only the replicated internal loads are shown in this graph, and the portion of heater output that was used for adjusting weather conditions is not shown. The cooling energy used during pre-cooling is presented in a lighter hue to represent the fact that this cooling energy (i.e. chiller energy) might be reduced or entirely removed using outside air. The actual energy

savings will depend on how much of that night pre-cooling is done using unconditioned outside air.

The cooling energy during occupied hours is demonstrated to start later in the test cell than in the reference cell. Pre-cooling had effectively delayed the use of mechanical cooling during occupied hours by 2 to 4 hours.

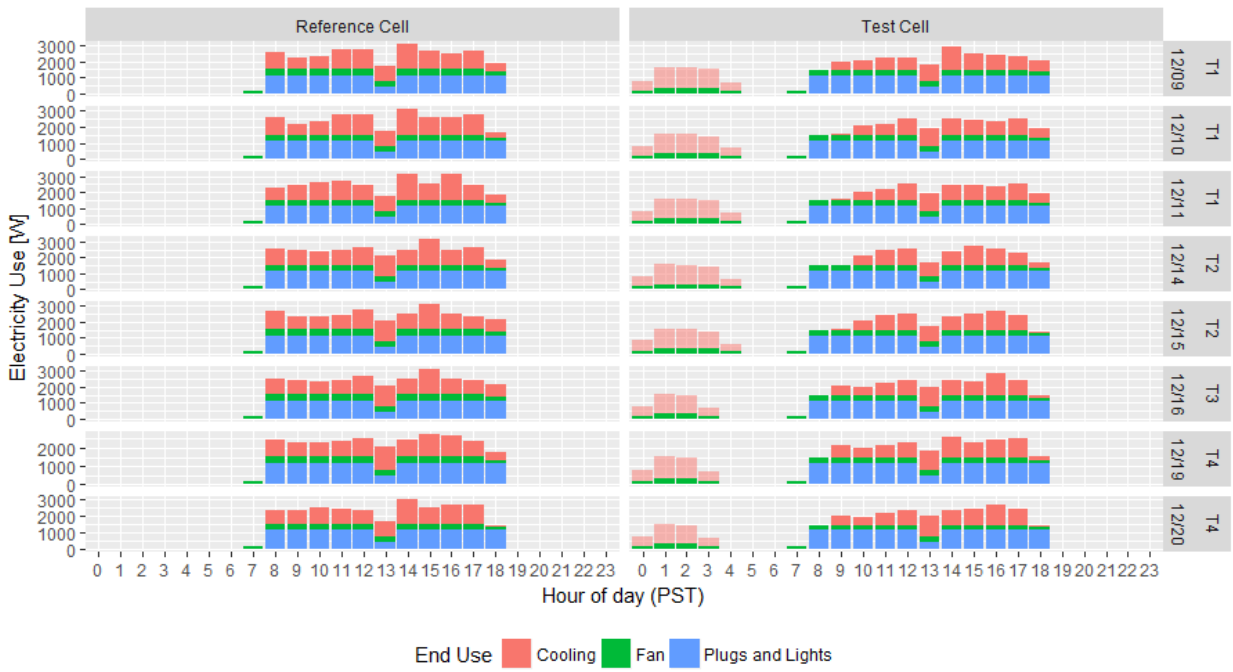


FIGURE 12 : ROUND 2 HOURLY ELECTRICITY USE PER END USE PER CELL

In Figure 13, the difference in hourly energy use between the two cells for the three end uses is shown. This allows for a better comparison of where the test cell differed from the reference cell. Since both cells used the same plug load schedules, the differences were in cooling energy for both pre-cooling and occupied hours and in fan energy.

This plot confirms the observations made previously: the additional cooling absorbed at night in the test cell was mostly used within the first few hours of occupied time, although it is noted that some cooling energy was saved throughout the day in most test conditions, which was not evident on the previous plot. There was little evidence that reducing the amount of PCM in the plenum (between T1 and T2 and between T3 and T4) had any impact on the savings observed, but there is evidence that the length of pre-cooling (between T2 and T3) had an impact, with shorter cooling savings periods at the beginning of the occupied hours.

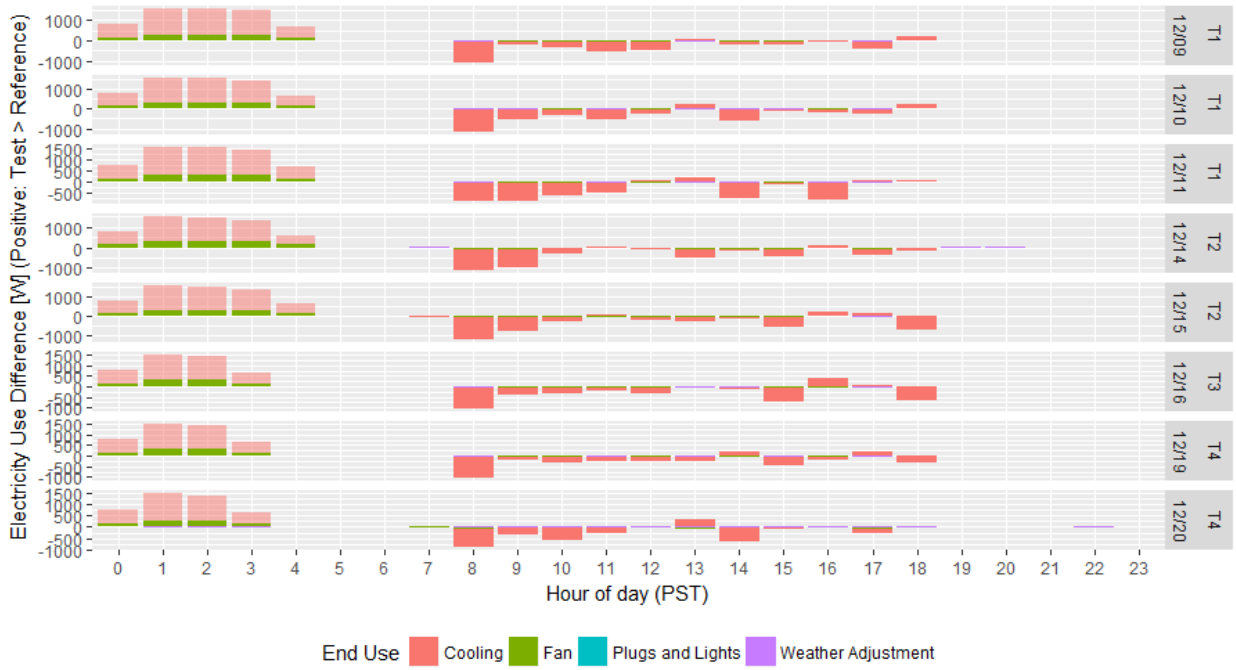


FIGURE 13 : ROUND 2 HOURLY ELECTRICITY USE DIFFERENCE PER END USE

Figure 14 shows the daily electrical energy used by mechanical cooling and fans. Plug loads were removed from the plot for clarity since it was the same in both cells. Like in Figure 12, the cooling energy during pre-cooling is presented in a lighter hue.

The full cooling energy as measured is represented as both mechanical (chiller) based energy consumed during the pre-cooling period (lighter shade of red) and other periods of operation (darker shade of red). This full cooling energy is applicable to conditions where the outside air temperature would be too high to be used for pre-cooling, with the chiller being used for this purpose. Under those conditions, the test cell, used more HVAC energy to cool down the space, as indicated in Figure 12. Comparing the daily load, the maximum difference was on 12/09 with an increase in cooling energy by 20%, and the minimum difference was on 12/16 with an increase of 8%.

However, if the cooling energy used during pre-cooling can be brought with unconditioned outside air, the cooling energy used would only be as shown in the darker shade of red. In this condition all tests showed net HVAC energy savings. The maximum HVAC savings was on 12/15 (T2) with a reduction of energy by 18% (32% of cooling), and the lowest was on 12/20 with a reduction of energy by 12% (23% of cooling).

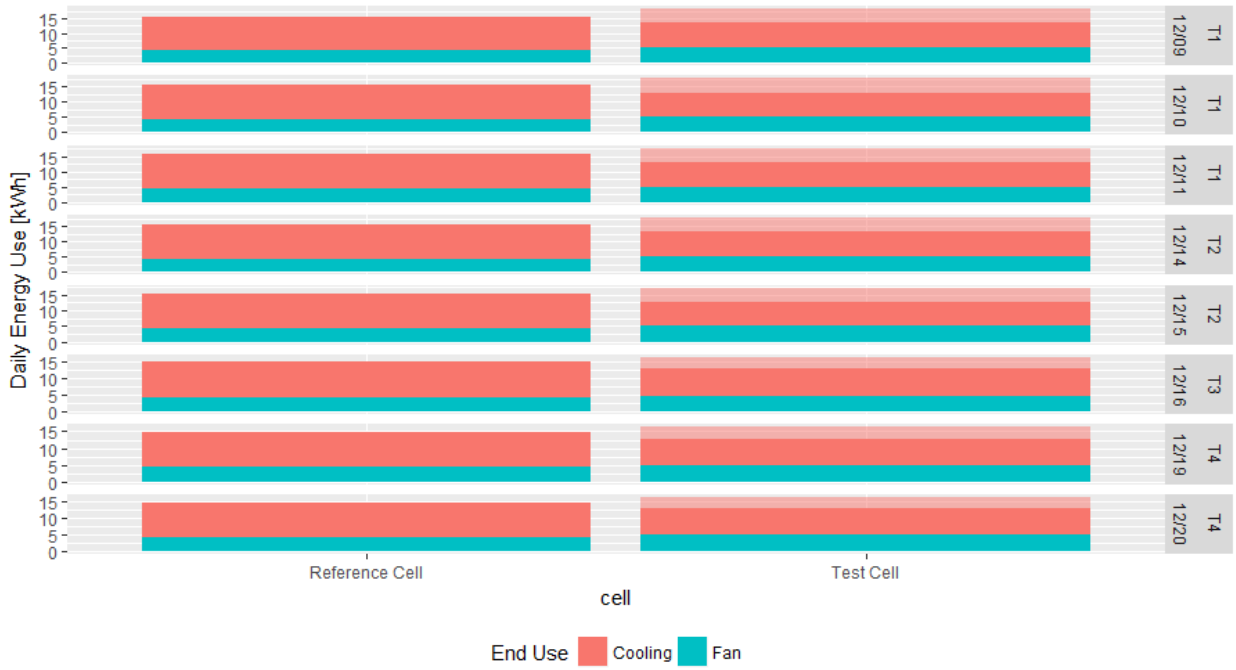


FIGURE 14 : ROUND 2 DAILY ELECTRICITY USE PER END USE PER CELL

The actual amount of daily savings observed will therefore depend on the ratio of unconditioned outside air used during pre-cooling. Maximum savings will be observed when no mechanical cooling is required at night to recharge the PCM. A space equipped with PCM would have a higher daily HVAC consumption if pre-cooling is done without outside air, by using mechanical cooling to lower the return air temperature. Figure 15 presents the range of savings and loss that were observed for each day of experiment, depending on whether outside air was used during pre-cooling. Positive values mean that test cell used more energy than the reference cell.

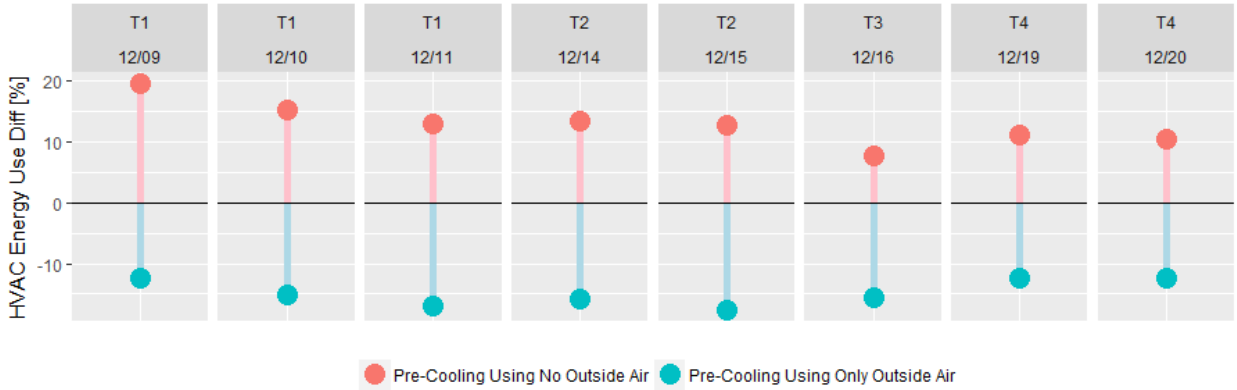


FIGURE 15 : ROUND 2 POTENTIAL HVAC ENERGY SAVINGS BY DAY OF EXPERIMENT

Testbed Thermal Mass Effects Analysis

This section describes what can be deduced within the constraints of the test setup and data available to compare the thermal mass impacts of the test cell's envelope compared to the PCM's thermal mass on impacting the heat loads in the space. It should be noted that there are uncertainties is this analysis that point to possible areas for further investigation. Several factors prevent a more complete understanding of this relationship, as described below.

Every hour, the amount of cooling energy supplied by the mechanical system is equal to the sum of all heat loads in the space. There are three main heat load sources that are investigated in this section: the heat from the cell envelope; the heat from the lighting and plug loads; and the heat from the PCM and plenum envelope. The gross heat between the supply air diffusers and the entrance of the plenum is calculated as:

$$Q_{below\ ceiling,gross} = \rho_{air} * \dot{V} * c_p * \left(T_{supply} - T_{south\ plenum}\right)$$

Where $Q_{below\ ceiling,gross}$ is the load in W, ρ_{air} is the air density in kg/m^3, $\dot{V}$ is the air flow in m^3/s, c_p is the air specific heat in J/kg·K and T_{supply} and $T_{south\ plenum}$ are the boundary temperatures in °C.

This quantity generally represents the heat provided to the envelope located below the dropped ceiling and includes impacts from plug loads and lighting. We deduct the amount of heat coming from the plug loads and lighting (here we assume 80% of lighting heat load is rejected into the space below the ceiling):

$$Q_{below\ ceiling,net} = Q_{below\ ceiling,gross} - Q_{plugs} - 0.8 * Q_{lights}$$

The rest of the lighting heat load (20%) is assumed to be rejected into the ceiling plenum, to account for the heat loss from the ballast.

The resulting net heat is equivalent to what is being gained or lost to the cell envelope below the ceiling. Those gains and losses are the result of either energy storage or release from the thermal mass of the envelope or from heat exchange with the outside environment through conduction or infiltration (marginal, but not insignificant).

When the same calculations are applied between the entrance of the plenum and the return duct air temperature, removing the other 20% of lighting energy use from the gross measurements, we get the net heat equivalent to what is gained or lost to the plenum envelope and PCM.

Figure 16 shows the evolution of each of these energy sources over the day for each test condition and each cell. The heat mass below and above the ceiling is calculated using the equations above, while the plugs' and lights' heat (both below and above the ceiling) is based

on the measured value. The mechanical cooling energy (Q_{total}) is the blue line superimposed on the plot and is equal to the difference in bar height on each side of the x axis.

First, it is noted that in the PCM test cell the pre-cooling sequence was driving heat out of the cell envelope below the ceiling as much as it was out of the plenum. In other words, when the cold supply air arrived in the plenum, it had already warmed up and had lost some of its cooling potential by cooling down the cell envelope. It is also noted that some of the heat that was driven out of the cell envelope during pre-cooling was reabsorbed by the cell envelope below the ceiling mostly within the first hours of the day. The plenum heat can be seen throughout the entire day and seems higher in the test cell than it was in the baseline cell.

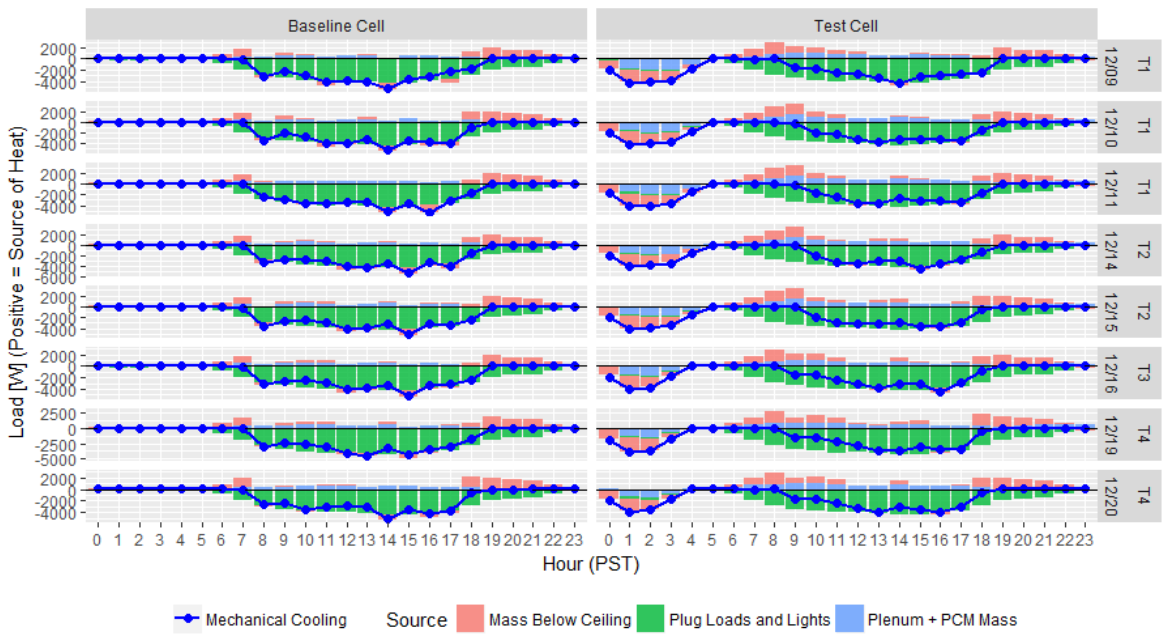


FIGURE 16 : ROUND 2 HEAT SOURCE ANALYSIS

In Figure 17 the difference between those sources heat in the two cells is compared. The sum of those differences is equal to the cooling load reduction. This plot shows that a good portion of the stored heat is in the cell envelope below the ceiling, but there is also a noticeable effect of the conditions above the ceiling, including the envelope present there and the PCM.

According to this analysis, about 29% to 48% of the cooling load reduction during occupied hours can be attributed to heat losses in the plenum, which includes the PCM.

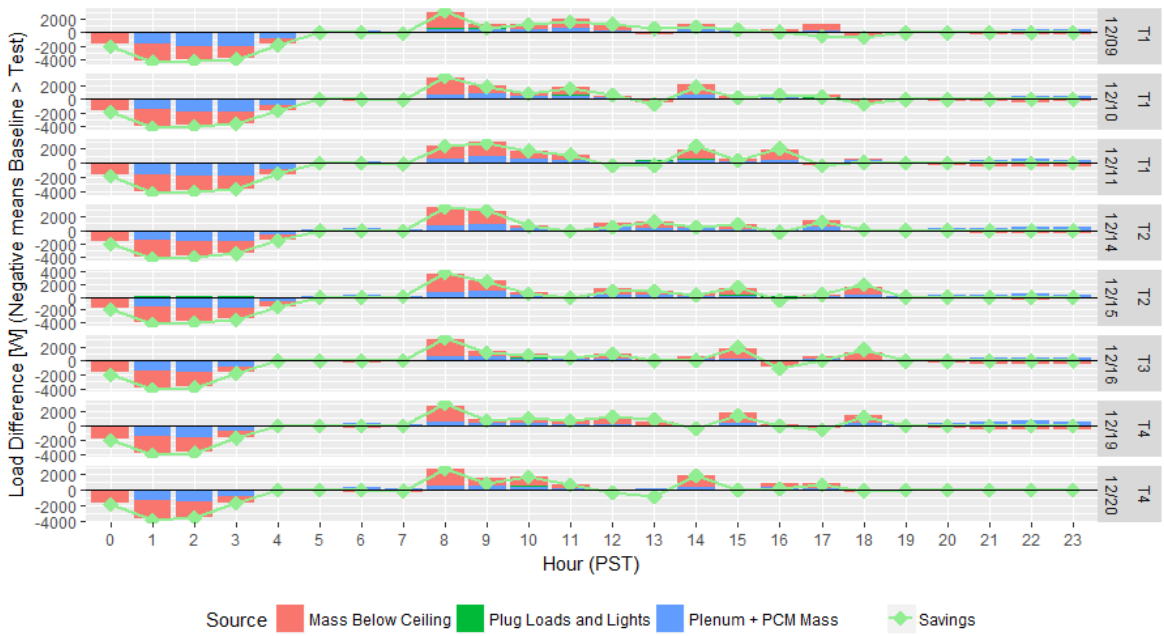


FIGURE 17 : ROUND 2 DIFFERENCE IN HEAT SOURCE LOAD BETWEEN THE CELLS AND LOAD SAVINGS

Round 2 Recommendations

Considering the results of this experiment, LBNL was asked to suggest potential improvements to the product that might enhance its performance and further drive the cooling load shift and cooling energy reduction.

Based on the test results observed, it may be that amount of PCM installed in an application is not as critical as the thermal heat transfer properties between the PCM and the air. In general, the product will work best when the heat exchange rate with the air is maximized. This can be done by making sure that the product is installed in an area with higher air velocity, which should also correspond to an area where the PCM can come in contact with a significant volume of travelling air. In addition, the packaging of the PCM product and its relationship to the enclosed PCM should be carefully reviewed to ensure that thermal transmittance is favorable. In contrast, areas with low air velocity will have a lower heat exchange rate. A challenge with an installation in a return air plenum is that while inherently there is already a low air velocity in this space, there are in addition obstructions that will create areas of air stagnation that may be difficult to assess adequately prior to installation. Related to that point, another solution may be to install the product directly inside the return duct, provided there is adequate area in the return duct and air velocity rates in this location prove to be significant enough to cause good heat transfer. This would likely result in a lower amount of PCM that can be installed, as compared to the plenum application, but could significantly increase the heat exchange rate should air velocities and packaging be favorable for heat transfer. An installation in the return airstream should be balanced to provide favorable PCM heat transfer, while not

establishing a significant fan energy penalty because of increased duct system pressure. This is likely to be achievable with careful attention to the duct system design.

The freezing and melting temperature of the PCM is also seen to be critical to ensuring the success of the peak shifting and energy savings properties of this product. It is recommended that the freezing and melting temperatures should be tailored to the specific application of the product, partnered with a period of measurement of temperature conditions in the proposed installation area using the pre-cooling strategy. The variables that might influence the optimal freeze and melt points include differences in geometry, load distribution, plenum or ductwork design and envelope characteristics. These conditions vary from one building to another, and as such the average plenum temperature may well be significantly different across different buildings and construction. The freezing and melting temperatures should be a few degrees lower and higher (respectively) than the daily average plenum temperature. While the hourly profile of the plenum temperature should change after the installation of the PCM, the daily average temperature should be relatively unaffected by the increased thermal inertia. It would be worth conducting a field evaluation of multiple building and construction types to ascertain the range of variability in the desired installation conditions to help guide product development as well.

Round 2 Conclusions and Future Work

This experiment intended to demonstrate the impact of introducing Stasis PCM in the plenum of an office space in combination with nighttime pre-cooling on the HVAC energy use, particularly as a means to shift the use from daytime to nighttime, where lower outside air temperature can be used. On a typical summer day in Southern California, the combination of PCM with pre-cooling was shown to lower the cooling load during occupied hours between 26% to 36%. While some of that shift can be attributed to heat stored in the cell envelope, between 29% and 48% of the heat during occupied hours is lost to the plenum and PCM combined.

When no mechanical cooling is used during precooling, this reduction in cooling load creates a reduction in HVAC energy consumption between 12% to 18%. If precooling is only done with mechanical cooling, the additional energy use at night increases the daily HVAC energy use by 8% to 20%.

Reducing the PCM to 83% (T1 to T2) of the initial mass did not have an impact on the load reduction but some modest impact could be seen when further reducing the mass of PCM to 65% (T3 to T4), with the cooling load during occupied hours being reduced by 22% compared to 27%, although this effect could have been compounded with the shortening of the pre-cooling period.

By shortening the pre-cooling period from 4 hours to 3 hours (T2 to T3), the load shift during occupied hours was reduced from 30-33% to 27%, which only slightly reduced the potential

HVAC energy savings from 16-18% to 16%, but greatly reduced the cost of running pre-cooling with no outside air from 12.6-13.3 kWh to 7.6 kWh.

While this set of experiments intended to evaluate the HVAC savings brought by a combination of Stasis PCM and nighttime pre-cooling, further tests would be needed to isolate the impact of the PCM alone on the load shifting, compared to the effect of the rest of the building envelope. This can be done in part by having both cells operate with nighttime pre-cooling. Further testing could also evaluate more realistic savings by using – rather than assuming – unconditioned air during pre-cooling, and using chiller efficiency curve data to more accurately reflect baseline energy consumption. Considering the difference in climate between the testbed location and the intended market, these tests would require particular consideration to the outside conditions. Simulation of the test cells could also be used to extrapolate the results to different climate or conditions and calculate annual savings. This would also enable calculating financial savings by coupling the electrical demand reduction with target tariffs used by office buildings.

In addition, different HVAC systems, such as under-floor air distribution, and different building construction, such as building with lower mass, could be tested. The product could also be used differently by introducing mid-day charging, with the objective of reducing peak demand in the afternoon for emerging tariff structures that lower energy use rates during the day.

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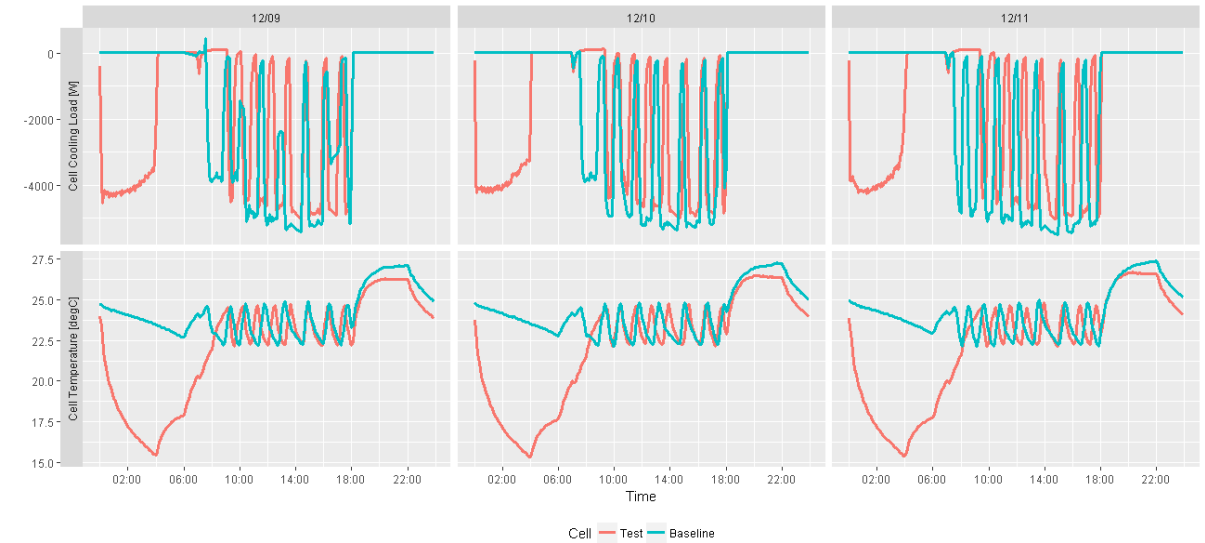
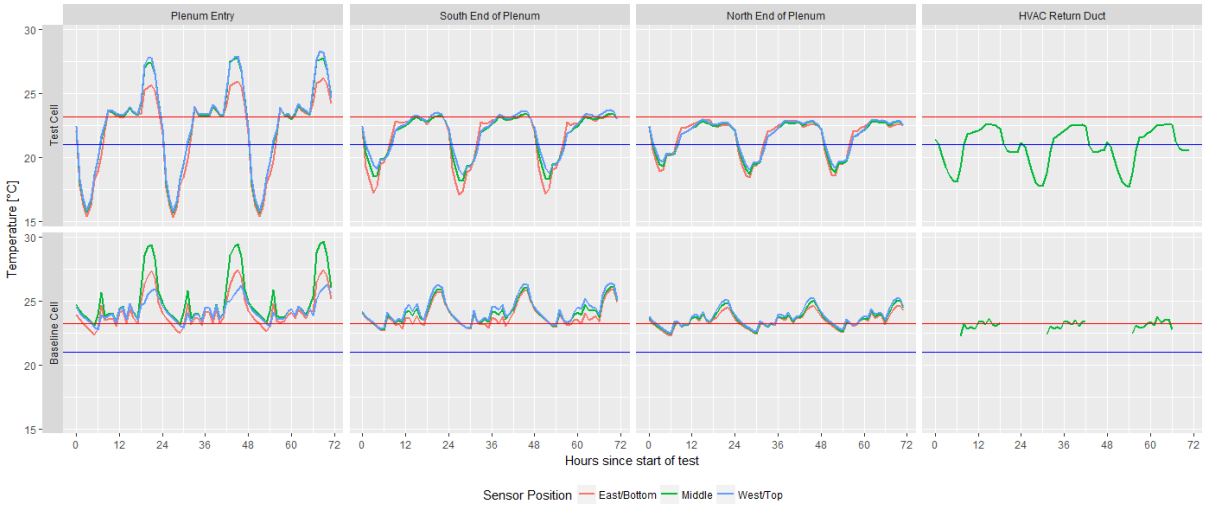
Appendix A – Round 2 Individual tests results

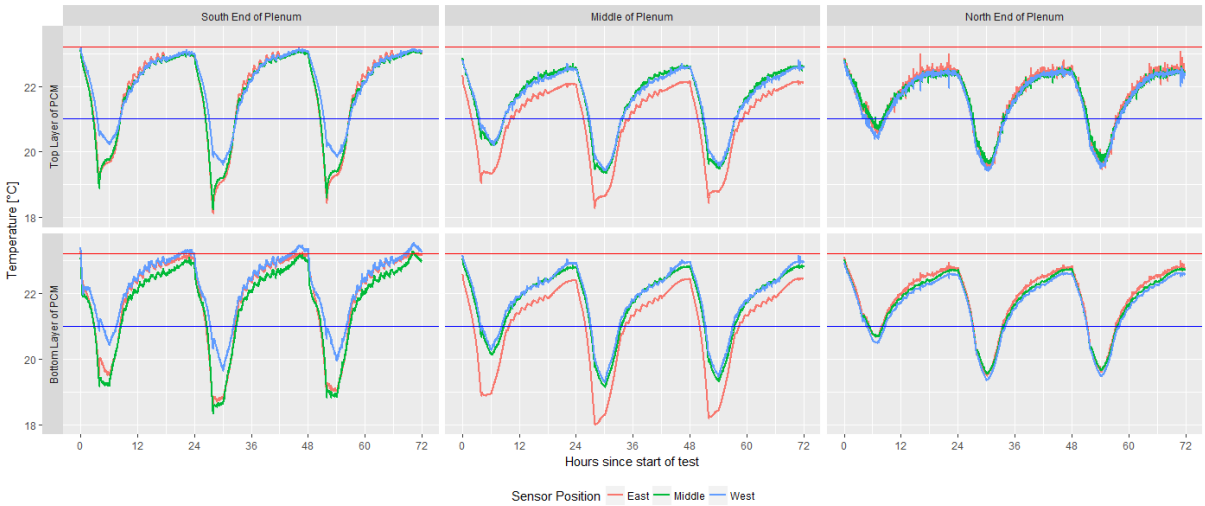
The following charts show the detailed results of each individual day of testing. In each cluster, the top plot shows the cell temperature and cooling load. The middle plot shows the temperature of the air at various locations in the plenum for both cells, and the bottom charts show the temperature measurement at the surface of the PCM in the test cell.

Index	# of days	Pre-cooling Sequence	High Density (0.67 lb/ft²)		Low Density (0.5 lb/ft²)		Ratio of total PCM mass
			Area [ft²]	Ratio of total ceiling area	Area [ft²]	Ratio of total ceiling area	
T1	3	12am to 4am	426	71%	524	87%	100%
T2	2	12am to 4am	376	63%	408	68%	83%
T3	1	12am to 3am	376	63%	408	68%	83%
T4	2	12am to 3am	266	44%	358	60%	65%

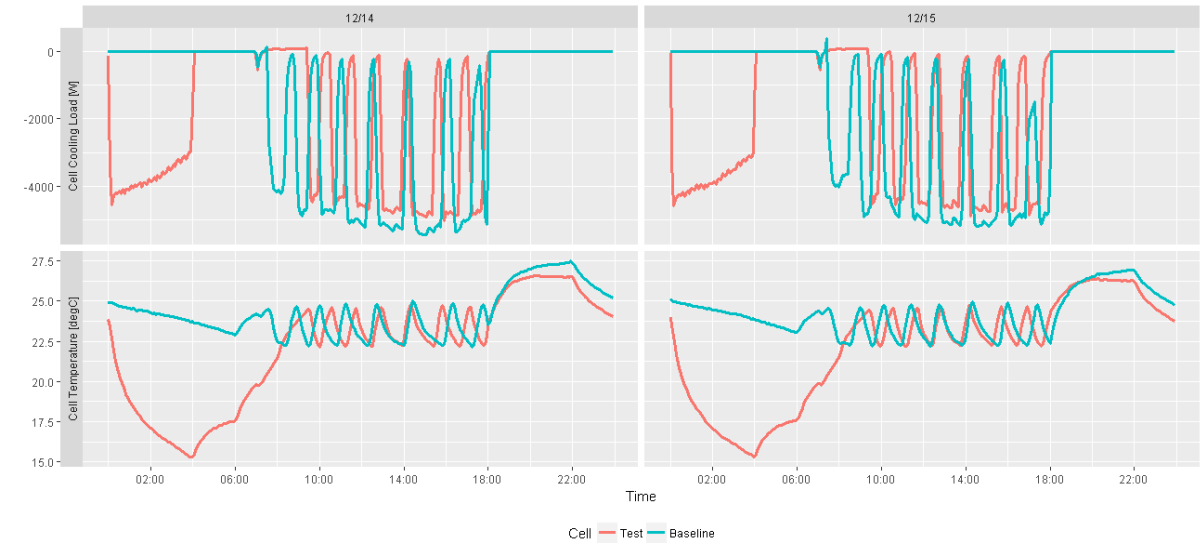
Test Conditions T1

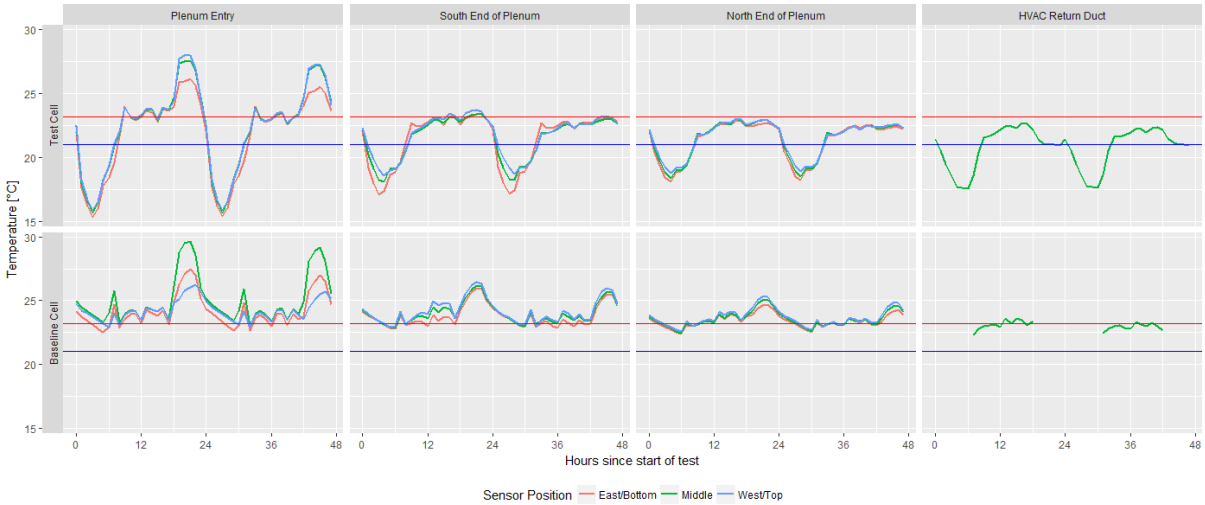


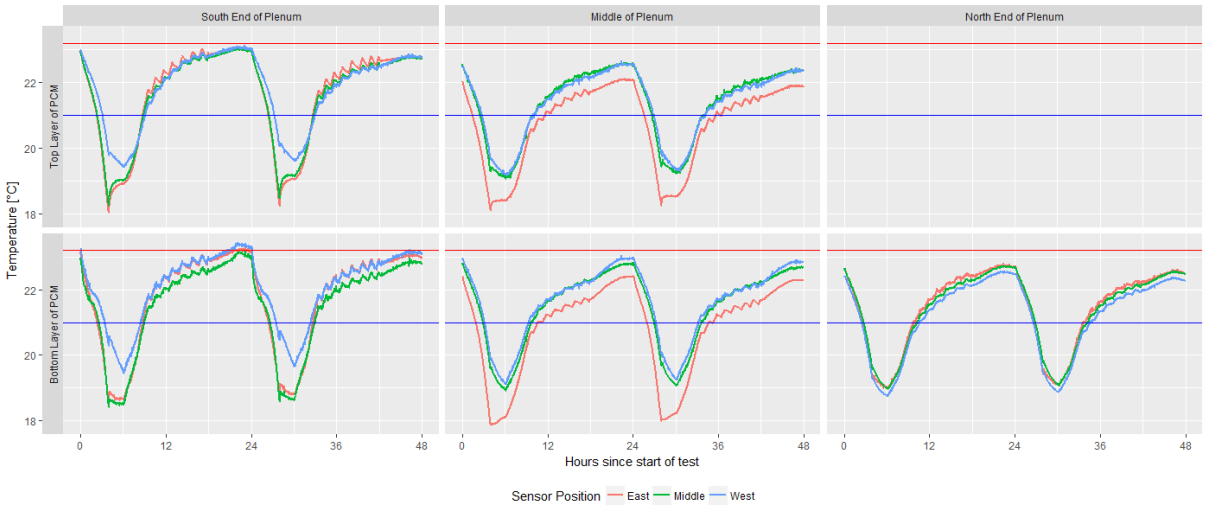




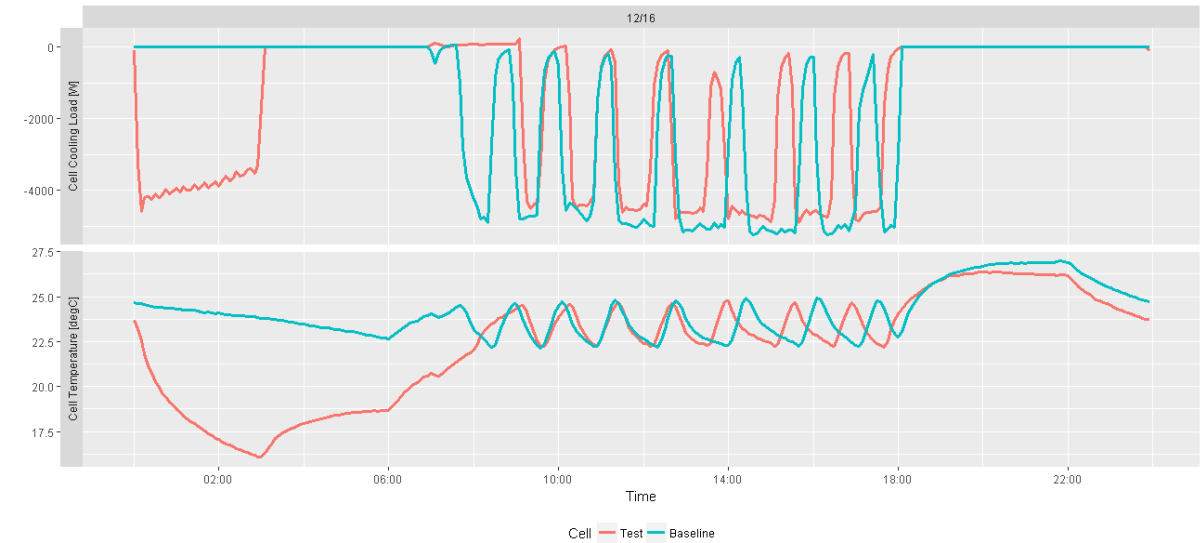
Test Conditions T2

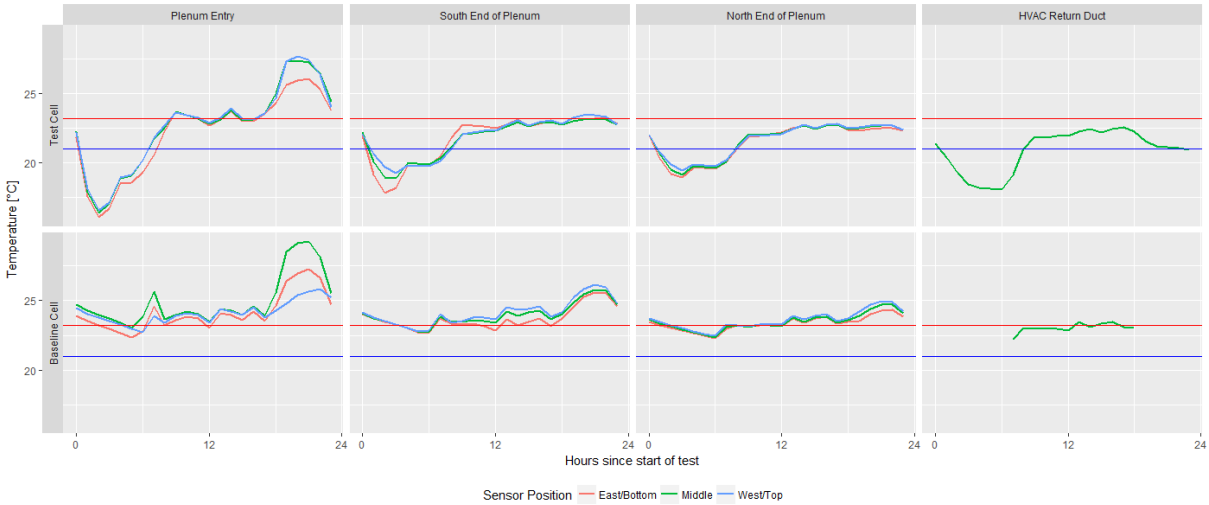


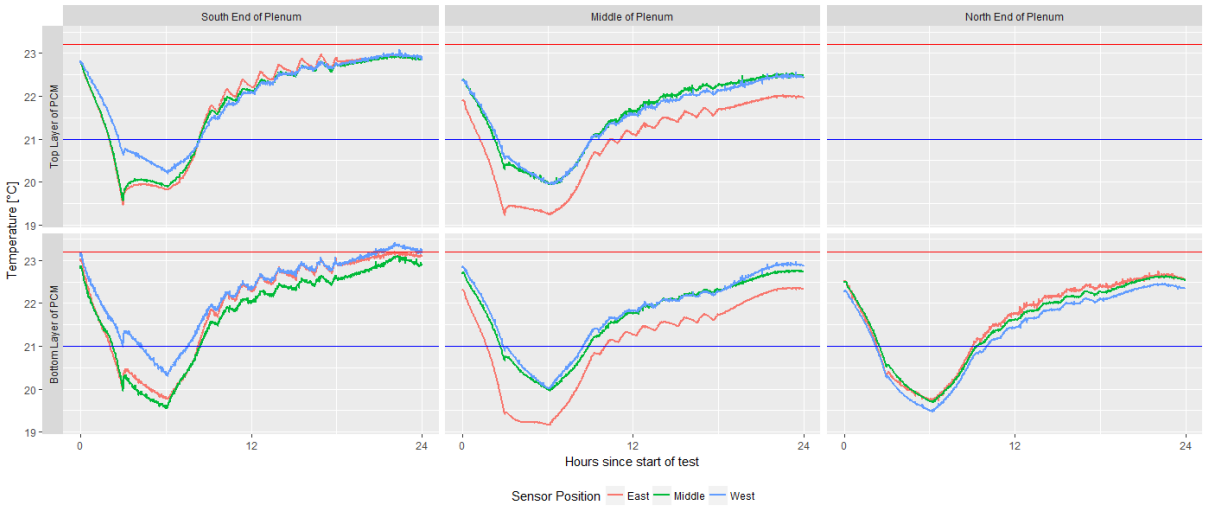




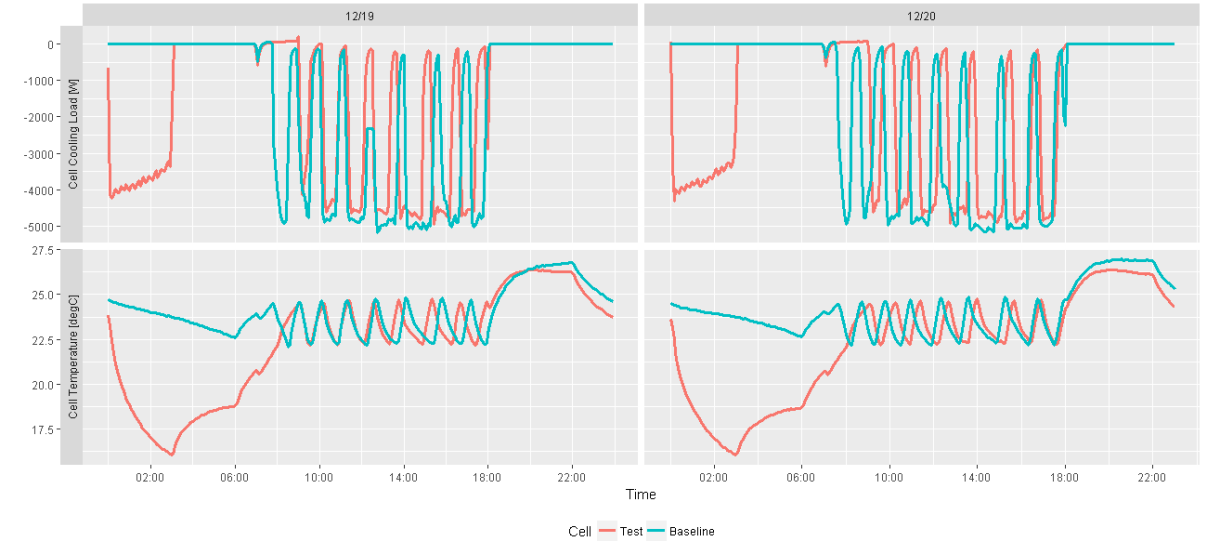
Test Conditions T3

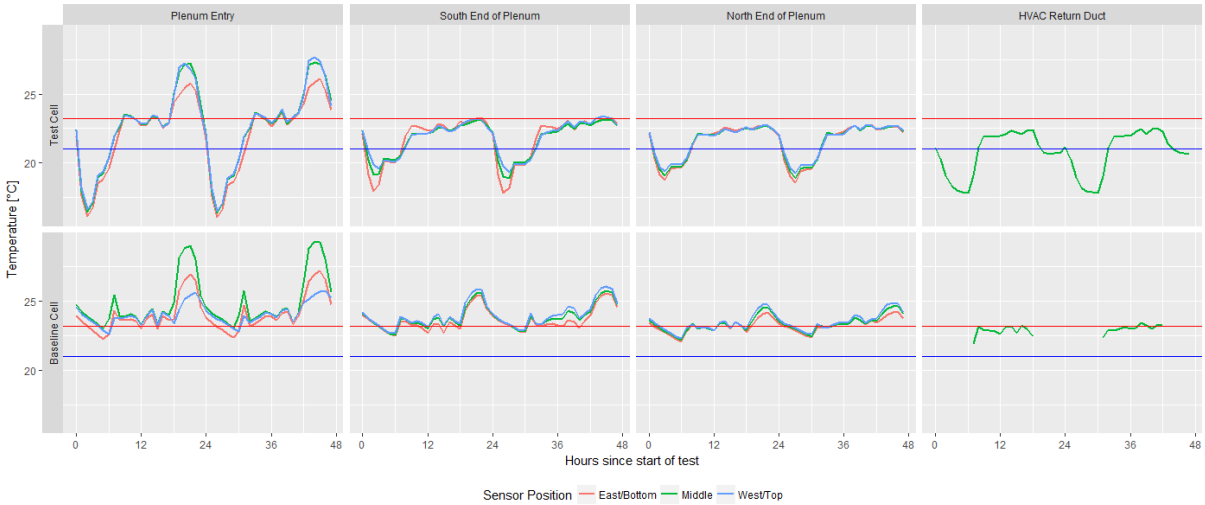


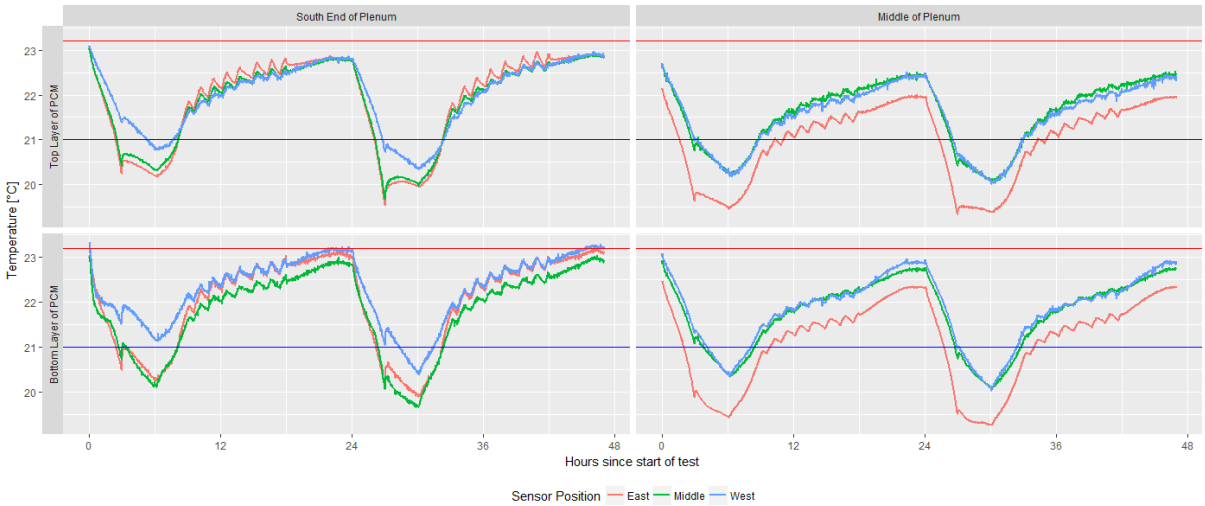




Test Conditions T4







Appendix B – Round 1 Results and Analysis

Summary

The following table gives an overview of the results obtained during Round 1. The test conditions are clarified in subsequent sections. The results are given as the daily cooling load (including pre-cooling energy) and as the HVAC energy, which includes fans and cooling energy, assuming a cooling efficiency of 1 kW/RT (i.e. 1 kW of electricity is needed to produce 1 refrigerant ton – or 3.51 kW of cooling) and assuming that pre-cooling is done with 100% outside air (this condition represents the optimal condition for savings potential). Positive savings means that the test cell has a lower HVAC consumption than the reference cell. 'Ref' and 'Test' refers to the reference cell (without PCM) and the test cell (with PCM).

TABLE 5 : RESULTS SUMMARY

Experiment conditions				Total Cooling load [kWh]			HVAC Energy [kWh]			
Date	Test Run	HVAC Controls	Nighttime Pre-Cooling in Reference Cell	Ref. Cell	Test Cell	Diff.	Ref. Cell	Test Cell	Savings	Savings [%]
28-Jun	T1	CAV_1	Yes	26.6	26.1	0.5	5.0	5.0	0.0	0%
30-Jun	T3	CAV_2	Yes	34.1	34.9	-0.8	8.3	8.0	0.3	4%
2-Jul	T5	CAV_3	Yes	22.9	23.5	-0.6	4.2	4.4	-0.2	-5%
3-Jul	T6	CAV_3	Yes	26.1	27.6	-1.5	5.4	5.1	0.2	4%
4-Jul	T7	CAV_4	No	20.7	27.9	-7.1	7.2	5.4	1.8	25%
6-Jul	T9	CAV_5	No	17.9	25.0	-7.1	6.4	4.0	2.4	38%
9-Jul	T12	CAV_6	No	26.4	27.2	-0.8	9.1	7.0	2.2	24%
11-Jul	T14	CAV_6	No	24.2	25.5	-1.4	8.5	6.4	2.1	25%
12-Jul	T15	CAV_6	No	23.1	25.1	-2.0	8.2	6.3	1.9	23%
14-Jul	T17	CAV_7	No	28.3	31.5	-3.2	9.7	7.6	2.1	22%
15-Jul	T18	CAV_7	No	26.9	30.3	-3.4	9.2	7.6	1.6	17%
16-Jul	T19	CAV_7	No	31.0	32.3	-1.3	10.5	8.3	2.2	21%
17-Jul	T20	CAV_7	No	30.3	32.3	-2.0	10.3	8.0	2.3	22%
18-Jul	T21	CAV_7	No	27.0	29.4	-2.4	9.3	7.2	2.1	23%
19-Jul	T22	CAV_7	No	25.0	29.2	-4.2	8.7	7.1	1.7	20%
20-Jul	T23	CAV_7	No	24.8	28.3	-3.6	8.6	7.0	1.6	19%
22-Jul	T25	CAV_8	Yes	31.4	27.0	4.4	7.6	6.3	1.2	16%
23-Jul	T26	CAV_8	Yes	30.5	30.3	0.2	7.6	7.4	0.2	3%

Round 1 Methodology and Experimental conditions

Methodology - Variables of Interest

The cooling load and total HVAC energy use were the primary variables of interest for this study. The description and measurement of these variables was as described for the Round 2 tests.

Conditions

For this test, the PCM was tested in conditions made to replicate a core zone in an office building. These conditions were replicated with the following parameters:

- The boundary walls of a core zones are connected to other conditioned spaces, and there is therefore little to no heat transfer through the walls
- The only loads for the HVAC are the heat from occupants, the interior equipment and the lights. Those loads are high during working hours and low or null in the evening.

The internal load density was chosen to represent a pre-1980 office building that would be a suitable candidate for retrofit. To that end, the values were picked from the NREL commercial reference building models (*U.S. Department of Energy Commercial Reference Building Models of the National Building Stock*, Deru et al., 2011):

- Lights have a power density of 1.6 W/ft² (16.9 W/m²)
- Plug loads have a power density of 1.0 W/ft² (10.8 W/m²)
- Occupant density is 5 people for 1000 ft² (5.38 people for 100 m²) exerting 68 W/person

In newer construction buildings, the main difference would be the light power density (1.0 W/ft² instead of 1.6 W/ft²).

For a 600 ft² spaces (which is the area of a cell in FLEXLAB), the expected peak internal load is 1764 W.

Testbed Description

The test for Round 1 was performed in the same FLEXLAB testbeds as the test for Round 2, but a few items were changed between the two rounds. Please refer to the Round 2 Testbed Description for details on the dimensions and general description of the testbed condition.

Envelope

The envelope is the same in the two tests, except for the floor that is not insulated in Round 1. The floor construction thermal resistance in Round 1 is equivalent to an R-Value of 4.31 h·ft2·°F/Btu (24.5 K·m2/W). The floor was insulated for Round 2 to prevent the floor from storing heat. In Round 1, no tarp was hung in the ceiling plenum, so that the plenum volume in the return airpath included the structural wood beams and their interstitial spaces between them.

Internal Loads

The lighting heat loads are replicated using four light fixtures that are recessed in the ceiling, with a power of 75 W each, for a total of 300 W, which is lower than the expected 960 W (1.6 W/ft² for 600 ft²) for the target building era of construction. The rest of the expected lighting heat load, the occupant heat and plug loads heating output are replicated using electric resistive heaters that are dispersed in the cell, and with a total power that ranged from one test to another (see test conditions later), between 1300 W and 2200 W. Therefore, the total load in the cell ranged from 1600 W to 2500 W depending on the test, when the replicated load is 1764 W with an addition of 200 W (1864 W) to offset heat losses through the envelope.

HVAC System

For Round 1 FLEXLAB's HVAC system in each cell was controlled as a packaged CAV air handler system with water sourced cooling and heating coils the same as described for Round 2.

Test Schedules

The same periods were used between the two rounds, but with slightly different schedules:

- Occupied hours: from 6am to 5pm PST, where the internal loads are high, and the cell indoor temperature is controlled to maintain comfort for the occupants within a given dead band
- Unoccupied hours: from 5pm to 6am PST, where the internal loads are lower and the cell indoor temperature setpoint is setback to reduce unnecessary energy consumption for cooling
- Pre-cooling: scheduled in the middle of the night for different lengths of time depending on the test, where the fan is operated at a high flow rate and the supply air is maintained at 55°F (12.8°C) to simulate a night flush using outside air to remove the heat stored in the PCM.

Sensors and Measurements

In terms of sensors, Round 1 didn't have sensors installed in the gap between the drop ceiling and the South wall, and the South plenum vertical temperature measurements were used as a proxy to the temperature of the air going in the plenum.

Since there was only one layer of PCM installed in Round 1, the 18 sensors used to measure the PCM temperatures were installed on 9 locations on both sides of the PCM.

Product Tested

The product tested was a phase change material product distributed by Stasis, sealed in a silver-colored packaging and grouped in mats. The mats were installed on a metal grid at about 12in (0.30m) above the ceiling tiles, except where it is not possible (ducts, light troffers…). The product installed covered about 80%, or 480 ft² (44.59 m²), of the ceiling area. Three mats of

the same PCM product in a different, clear packaging were tested and used to compare the performances of the packaging films by looking at the surface temperature.

The properties of the phase change material or the packaging films were not disclosed to the lab. The product was expected to have a melting point of 73.4°F (23°C). The freezing point was not disclosed but was expected to be a few degrees below the melting point.

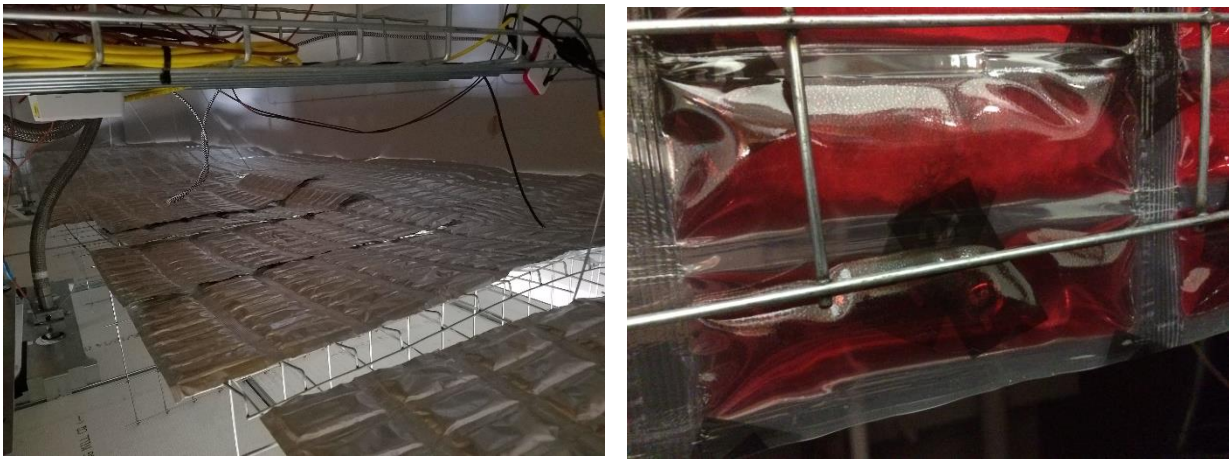


FIGURE 18 : PCM INSTALLED IN ROUND 1. (LEFT) SILVER PACKAGING (RIGHT) CLEAR PACKAGING

Round 1 Testing Schedule

The tests described in Table 6 were conducted to determine the performance of the product tested. For those tests, both cells were operated in the exact same conditions, apart from the pre-cooling period that was only used in the reference cell when specified. When pre-cooling was used in the reference cell, the same sequence as the one used in the test cell was used. In both cells, the economizer was not used, and no outside air was introduced in the cells (except for potential infiltration). The introduction of precooling air during unoccupied conditions was done with 55°F (12.78°C) supply air temperature, to simulate a typical night time free cooling condition using outside air. During unoccupied conditions, after pre-cooling, the fan was off and a setpoint of 82°F (27.78°C) was used to keep the space conditioned, although in no test was additional cooling needed after pre-cooling to maintain the space under this setpoint.

TABLE 6 : ROUND 1 TESTING CALENDAR AND CONDITIONS

Index	# of days	Setpoint	Pre-cooling Sequence	Pre-Cooling in Reference Cell	Plug Loads Occupied – Unoccupied	Lights
CAV_1	1	74 +/- 2°F (23.3 +/- 1.1°C)	1000 cfm (0.47 cms) 10pm to 4am	Yes	1300-560 W	300 W
CAV_2	1	74 +/- 2°F	1000 cfm (0.47 cms)	Yes	2200-560 W	300

		(23.3 +/- 1.1°C)	10pm to 4am			W
CAV_3	2	79.5 +/- 2°F (26.4 +/- 1.1°C)	1000 cfm (0.47 cms) 10pm to 4am	Yes	2200-560 W	300 W
CAV_4	1	79.5 +/- 2°F (26.4 +/- 1.1°C)	1000 cfm (0.47 cms) 10pm to 4am	No	2200-560 W	300 W
CAV_5	1	83.1 +/- 2°F (28.4 +/- 1.1°C)	1000 cfm (0.47 cms) 10pm to 4am	No	2200-560 W	300 W
CAV_6	3	83.1 +/- 2°F (28.4 +/- 1.1°C)	600 cfm (0.28cms) 1am to 4am	No	2200-560 W	300 W
CAV_7	7	81.5 +/- 2.7°F (27.5 +/- 1.5°C)	1000 cfm (0.47 cms) 11pm to 3am	No	2200-560 W	300 W
CAV_8	2	81.5 +/- 2.7°F (27.5 +/- 1.5°C)	1000 cfm (0.47 cms) 11pm to 3am	Yes	2200-560 W	300 W

Round 1 Results

Daily Cooling Load Results

The plot on Figure 19 shows the daily cooling load difference between the two cells. Positive values mean that the test cell had a lower cooling load than the baseline for that day. It also shows the ratio of that difference relative to the total daily load in the reference cell.

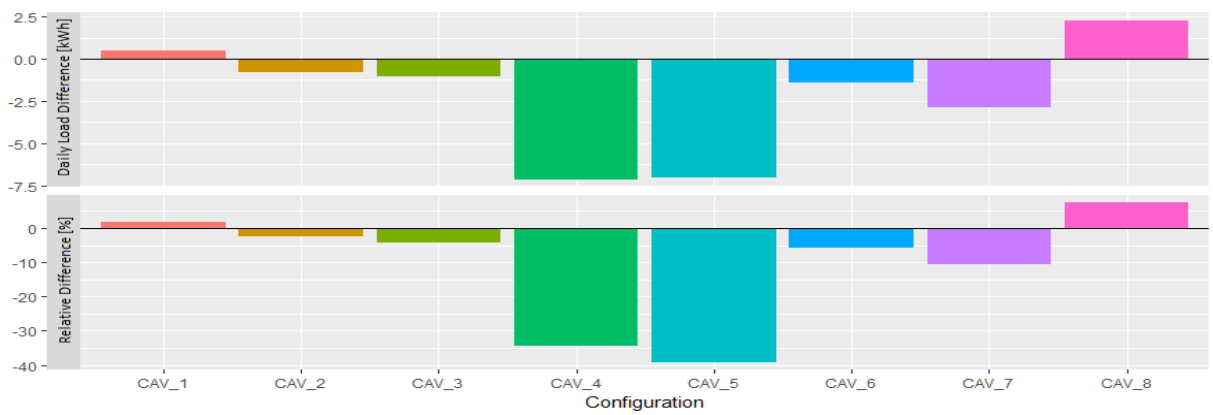


FIGURE 19 : ROUND 1 DAILY CUMULATIVE COOLING LOAD DIFFERENCE

CAV-1, 2, and 3

For CAV-1 and 2 the cells were run with an occupied period setpoint of 74 +/-2F (23.3 +/- 1.1°C), representing a typical office interior condition during a warmer period of the year. The results from these test runs did not illustrate any energy savings for the test cell over the reference cell, where both cells employed a pre-cooling strategy.

For those tests, the cells daily loads are within 5% of each other's, which is lower than the uncertainty determined during calibration. Those tests did therefore not result in a significant daily load difference that could be measured.

CAV 4 and 5

In CAV runs 4 and 5, the conditions of the cells were quickly altered, first by not using pre-cooling in the baseline cell, then by increasing the occupied setpoint in both cells by 3.6°F (2K). The tests were only run for a short time (1 day and 2 days respectively) and there was no time attributed to conditioning the cells before the tests. This results in the cell being operated in an unbalanced way, with the baseline cell envelope temperature warming up continuously over the period of the tests: in CAV 4, because we stop cooling down the cell at night, and in CAV 5 because the interior setpoint during occupied hours is increased. The large difference in daily load between the two cells can be mostly attributed to coolth in the baseline cell envelope being carried over from previous test conditions.

Figure 20 shows the daily average of the slab temperature in both cells during the experiment. We can see that at the end of CAV 3, when we stopped pre-cooling in the reference cell and reduce the time of the pre-cooling sequence in the test cell, along with increasing the cell setpoint, the slab temperature is rising in both cells, but the rise is quicker in the reference cell over CAV 4 and 5. This shows that the slab in the baseline cell was absorbing a lot of heat (i.e. releasing its coolth) during those tests, supplementing the HVAC system, while the slab in the test cell mostly remained at the same temperature. Conditions stabilized after a couple of days in CAV 6. For that reason, the results obtained for CAV 4 and 5 cannot be conclusive.

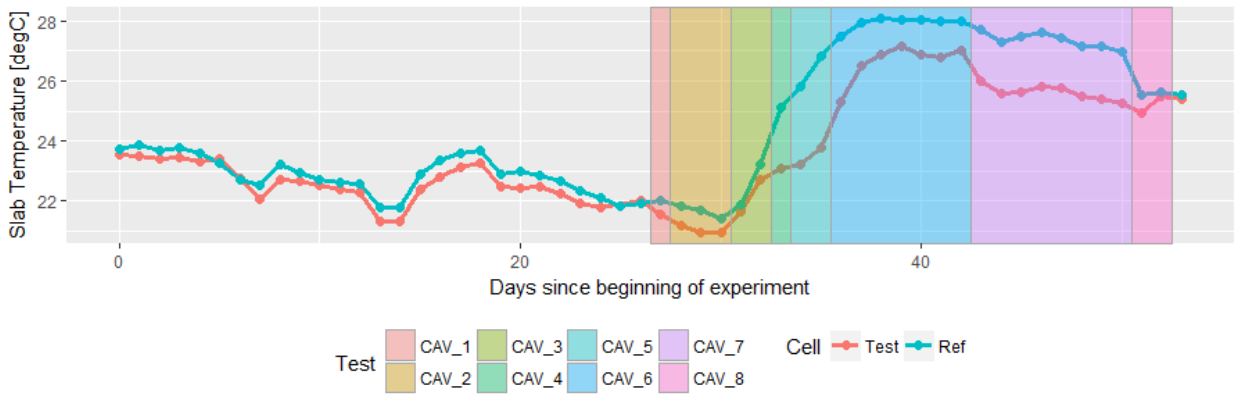


FIGURE 20 : ROUND 1 SLAB TEMPERATURE THROUGH EXPERIMENT

CAV-6 and 7

In CAV 6, the pre-cooling air flow rate was reduced from 1000 cfm (0.47 cms) to 600 cfm (0.28cms) in the test cell (no pre-cooling in the baseline cell). We can see in Figure 20 that this results in an increase of the daily average temperature of the slab in the test cell. The cell envelope temperature (measured by proxy by the slab temperature sensor) is mostly stabilized after a couple of days. We only use the last three days of that experience to analyze the results. Compared to CAV 4 and 5, the test conditions of CAV 6 are a lot closer to a true comparison between two identical buildings, with one building using PCM and pre-cooling. This condition resulted in a low daily cooling load difference (5.6%).

In CAV 7, the cell setpoint was reduced from 83.1 +/- 2°F (28.4 +/- 1.1°C) to 81.5 +/- 2.7°F (27.5 +/- 1.5°C), and the pre-cooling air flow rate was brought back to 1000 cfm (0.47 cms) and shifted by one hour (from 11pm to 3am). The first day for that condition was used to condition the cell envelope, leaving 7 days to be used for the comparison. The daily load difference between the two cells is 10.4%, which can be attributed to the excess cooling brought by pre-cooling in the test chamber and lost to the outside of the cell.

CAV-8

Finally, in CAV 8, the pre-cooling was turned back on in the baseline cell, with the idea to get more data for the comparison of both cells with pre-cooling, but with a higher setpoint than previously evaluated (in CAV 1 to 3). The daily cooling energy for that sequence is positive (meaning that the test cell had a lower daily cooling load than the baseline cell), but close to the uncertainty limit. This result can be explained by the fact that this condition was run for a short amount of time, and we can see in Figure 20 that the baseline cell slab was releasing its heat at the beginning of the test.

Hourly Cooling Load Results

The results are presented for the all experiments and aggregated by test conditions.

The plot on Figure 21 shows the hourly cooling load for each experimental condition and by cell, along with the load difference between the two cells (red and green strips). If we integrate the load difference (area under the curve for red and green strips), we get back the results presented in the daily load difference section (see Figure 19).

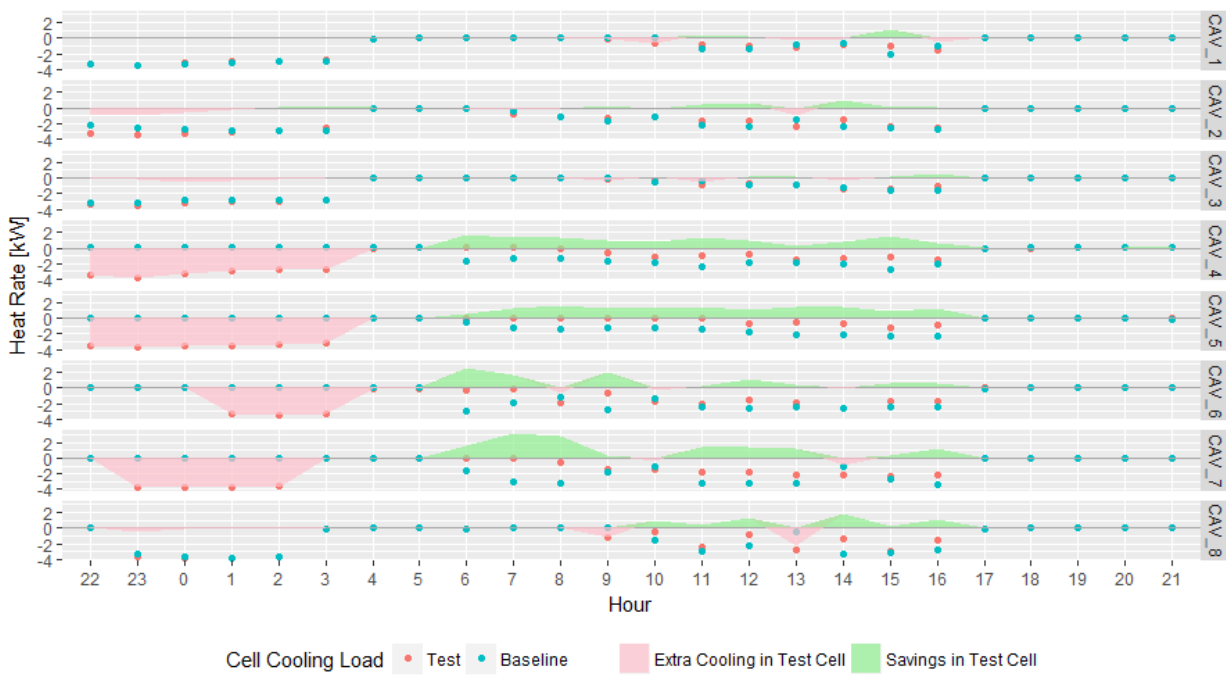


FIGURE 21 : ROUND 1 CELLS' HOURLY COOLING LOAD

The first observation we can make is that for testing conditions where pre-cooling was used in both cells (CAV-1, 2, 3 and 8), the difference between the two cells is minimal. Both cells seem to perform identically with some difference during the day due to the nature of the control sequence: in an on/off temperature control, a slight difference in temperature between the two cells can trigger the cooling in one cell and not the other, which leads to the two cells being operated out-of-sync.

Where the pre-cooling sequence was only applied in the test cell (CAV-4, 5, 6 and 7), we can see a large strip of negative "savings" at the beginning of the day. The test cell uses a lot of energy to bring in cool air with the intent to charge (freeze) the PCM, with the idea to enhance the potential for absorbing heat during the day. We do indeed see some load reduction occurring during the day, when the baseline cell needs more cooling than the test cell to condition the cell during occupied hours. However, upon further investigation, it was discovered that the PCM was likely not freezing or melting fully over the day in these periods, and consequently we can assume that the load reduction seen in these cases are largely due to the internal loads being partially dissipated in the cold thermal mass of the test cell envelope.

Figure 22 shows pictures of the clear packaged PCM that were taken at the end of the day on July 6th (CAV-5), when the plenum air temperature was above 75.2°F (24°C) for 4 to 5 hours prior to the observation. The PCM in the clear packaging is only partially melted, and the PCM in the silver packaging felt entirely frozen, even though the expected melting point was 73.4°F (23°C).

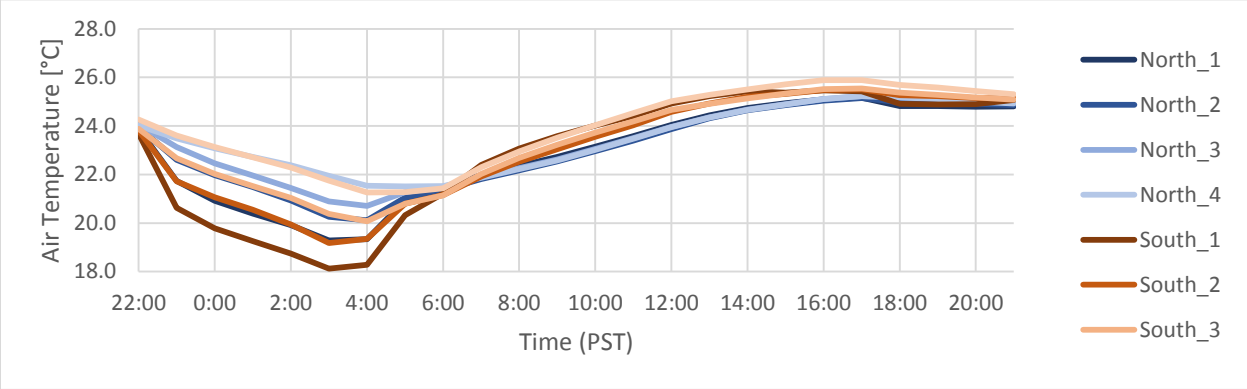


FIGURE 22 : ROUND 1 (TOP) AIR TEMPERATURE IN THE PLENUM ON 07/06, (BOTTOM) PICTURES OF THE CLEAR PACKAGE PCM TAKEN ON 07/06 5PM PST (CAV -5)

In addition to this observation, a subsequent section will discuss how the energy dissipated in the test cell was mostly observed in the cell below the plenum, suggesting that it was the slab and the cell envelope that created most of the observed shift.

HVAC Energy Use Results

In this section, we look at the HVAC energy use during the different tests. To do so, we assume a constant chilled water production efficiency of 1 kW/RT (equivalent to a COP of 3.51), that we apply to the measured cooling coil load.

Figure 23 shows the hourly electricity use for three end-uses: cooling, fans and plug loads (including lights), for both the test cell and reference cell and for all 8 conditions tested. The cooling energy use during pre-cooling is presented in a clearer hue to represent the fact that this cooling energy might be reduced or entirely shaved using outside air. The actual energy savings will depend on how much of that night pre-cooling is done using outside air.

For CAV 1 to 3 and CAV 8, that is when pre-cooling is used identically in both cells, we can see that both cells behave in a comparable way with similar end-uses consumption profiles. For the other test cases, where the reference cell did not use pre-cooling, we can note that the cooling energy during occupied hours starts later in the test cell than in the reference cell. Pre-cooling has effectively delayed the use of mechanical cooling during occupied hours by 2 to 4 hours.

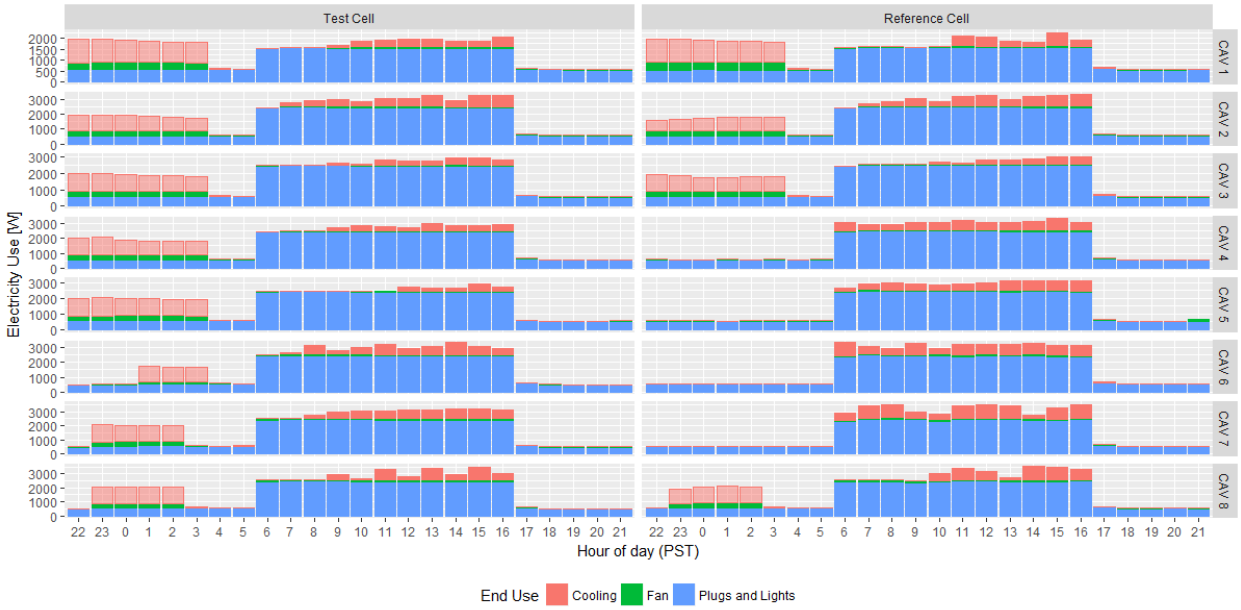


FIGURE 23 : ROUND 1 HOURLY ELECTRICITY USE PER END USE PER CELL

In Figure 24, we look at the difference in hourly energy use between the two cells for the three end uses. This gives us a better look at where the test cell differed from the reference cell.

We can confirm that the plug loads and lights in both cells were kept at a virtually the same level throughout the tests. The remaining differences are in cooling energy (for both pre-cooling and occupied hours) and in fan energy (mostly during pre-cooling when the reference cell did not use it).

This plot confirms the observations made previously: there is almost no difference between the two cells when pre-cooling is used in both cells (CAV 1, 2, 3, 8). When pre-cooling is not used in the baseline cell, the additional cooling dispensed at night in the test cell is mostly lost within the first few hours of occupied time (CAV 6 and 7). CAV 4 and 5 show a continuous cooling savings throughout the occupied hours, which is not as high as it could be, given that the baseline cell envelope was continuously warming up during that time, and therefore absorbing some of the internal loads.

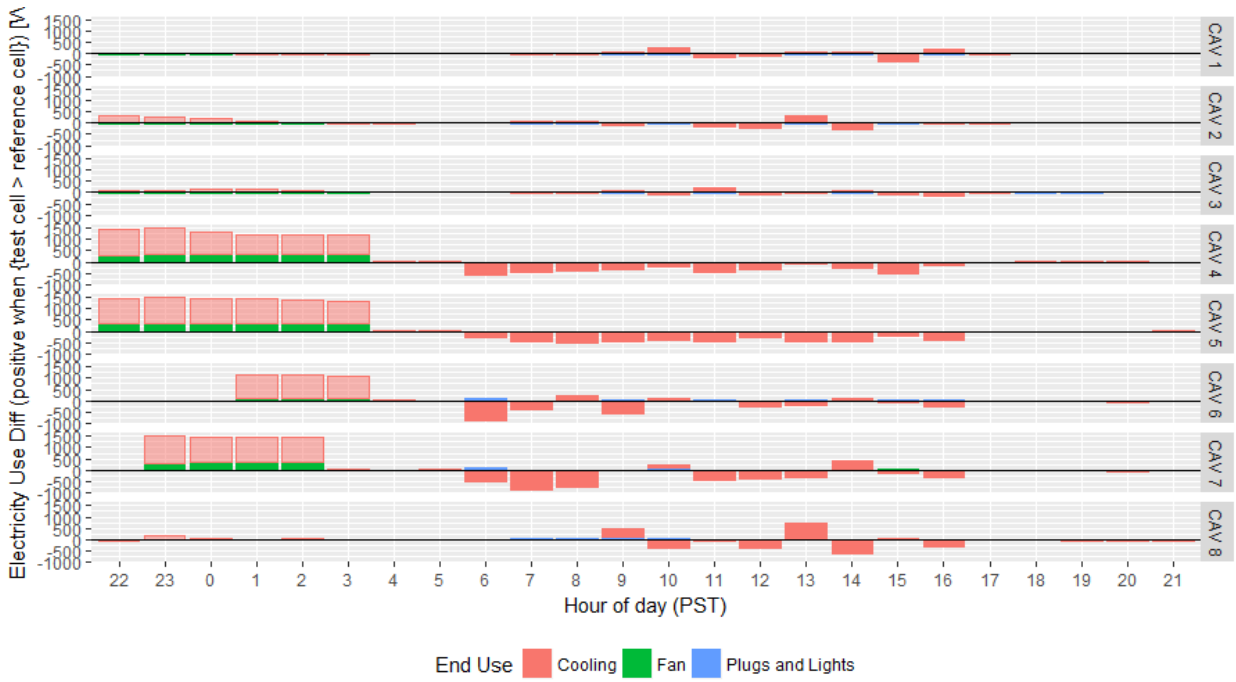


FIGURE 24 : ROUND 1 HOURLY ELECTRICITY USE DIFFERENCE PER END USE

Figure 25 shows the daily electrical energy used by cooling and fans. Plug loads was removed from the plot since it is the same in both cells and made the plot difficult to read. Like in Figure 23, the cooling energy during pre-cooling is presented in a lighter hue. We can note that in cases where pre-cooling is used in both cells (CAV 1 to 3 and 8), the HVAC energy consumption is similar in both cells.

We first compare the full cooling energy, as measured and converted to consumed energy, which assumes that no outside air was used during pre-cooling, which is equivalent to conditions where the outside air temperature would be too high to incorporate into the supply air mix, resulting in the pre-cooling being done entirely with mechanical cooling. Under those conditions, the test cell, when using a pre-cooling strategy while the reference cell doesn't, is using more HVAC energy to cool down the space, since some of the heat removed at night is gained back through the envelope before the beginning of occupied hours. In a perfectly adiabatic situation, we would expect to see both cells having similar daily energy (with some imbalance in ventilation energy). For CAV 4 and 5, we notice a much higher energy consumption in the test cell than in the reference cell (respectively 59% and 66% increase in energy consumption), due to the reference cell being in an unbalanced condition. The other two tests show a higher HVAC energy in the test cell between 11% (CAV 6) and 29% (CAV 7).

If we remove the cooling energy used during pre-cooling from the results (i.e. assuming all of it can be brought with outside air), we get the higher bound of daily energy savings. In that case,

CAV 6 and CAV 7 show savings of 24% (from 8.6 kWh to 6.6 kWh) and 20% (from 9.5 kWh to 7.5 kWh) respectively. Those savings can mostly be attributed to the effect of the thermal mass of the envelope, considering that there is no evidence that the PCM did provide any additional shift in cooling for test conditions where both cells operated with pre-cooling.

FIGURE 25 : ROUND 1 DAILY ELECTRICITY USE PER END USE PER CELL

PCM Performance Evaluation Under Different Temperatures

This section reviews the behavior of the PCM observed during the different test conditions. In this analysis, we are mainly focused on evaluating whether the PCM experienced any freezing and melting, as this is the key condition relating to potential energy savings. The conditions we monitored were the plenum air temperature, the supply flow rate in the cell (assumed to be the flow rate crossing the entire plenum area from South to North) and the duration of the pre-cooling strategy, which regardless of the test condition supplied air at 55°F (12.7°C) at 1000 cfm (0.47 cms) (and 600 cfm (0.28cms) for CAV-6).

Figure 26 shows the temperature measured at different locations on the PCM for each test (see Figure 1 for location). It demonstrates the effect of the different pre-cooling strategies and the cell setpoint on plenum temperatures. Regardless of the temperature at the beginning of the pre-cooling sequence, we observe a drop in the temperature in the ceiling plenum of about 4C in 3 to 6 hours depending on the controls sequence. The change of phase would have manifested in a change of slope in the temperature profile. When the PCM is freezing, the cold air surrounding it would drive the heat out at near-constant temperature. This doesn't

necessarily mean that no freezing or melting happened – the change of phase might be happening at a slow rate which would be difficult to notice in the temperature profile - and, when entering the cell between two tests to modify the conditions, we would notice that in the first tests (with low setpoint) the PCM would be solid to the touch, while in the later tests (with higher setpoint), the PCM would be liquid or mushy to the touch. This complete change of phase happened progressively over multiple days. Normal plenum temperature profiles, without any impacts from the PCM, would expect to see a gradual decrease in temperature during the precooling period, and a less rapid rise in temperature during the following unoccupied hours when the pre-cooling is over, and the fan is off, followed by a cycling of higher and lower temperatures as the space is controlled within a setpoint dead band during occupied hours. This trend is clearly noted in each of the test runs.

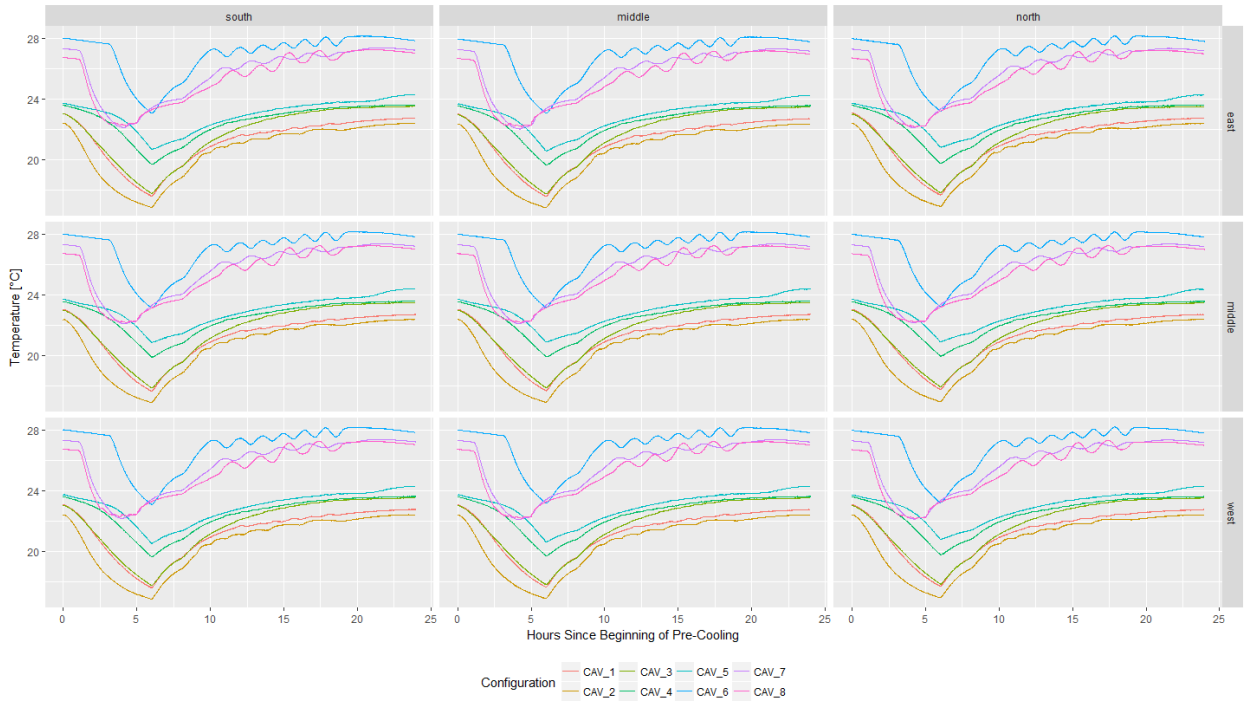


FIGURE 26 : ROUND 1 TEMPERATURE OF THE PCM SURFACE AT DISTINCT LOCATIONS IN THE PLENUM

There seems to be no hard evidence of melting happening in the PCM around 73.4°F (23°C). Additional controls sequences were run with increased setpoints during occupied hours (CAV 3 to 8) to determine if freezing/melting would happen at higher temperatures, however the characteristic changes in temperature slope were not observed. It is possible that there was a set of temperature set points not tested that was the optimal temperature range for the PCM to work. In the temperature graph, we see that the PCM rarely crosses 75.2°F (24°C) in a single test and we might interpret the end of the pre-cooling in CAV 7 and 8 to be hitting a plateau,

and the beginning of CAV 4 and 5 to have a slow start, suggesting some change in phase in that area.

While the occupied setpoints for CAV 4 and 5 were different (79.5°F (26.4°C) and 83.1°F (28.4°C) respectively) and both were higher than the PCM temperature at the end of the day, the slab was still warming up from previous test that were conducted at much lower temperature (75.2°F (24°C)), and was in both cases at slightly above 73.4°F (23°C). When we reduced the pre-cooling time in CAV 6, the slab started heating up quicker (with less heat removed from it at night), which leads to the upper set of lines on the plot (see Figure 20).

From those conclusions, this product might have performed better in an environment where the slab temperature was maintained slightly above 75.2°F (24°C), or in a space where the slab doesn't drive the air temperature. Overall, the PCM did not appear to exhibit the freeze/melting behavior around 73.4°F (23°C) as had been suggested by the supplier. A clear recommendation from this study is to reevaluate the actual freeze/melt conditions of the product and re-engineer the PCM product to exhibit the performance desired.

In Figure 27, we look at the difference of temperature between the air entering the plenum (as measured by the average temperature measurements of the vertical rod on the South end) and the average PCM surface temperature. Positive values mean that the PCM surface temperature is higher than the air temperature.

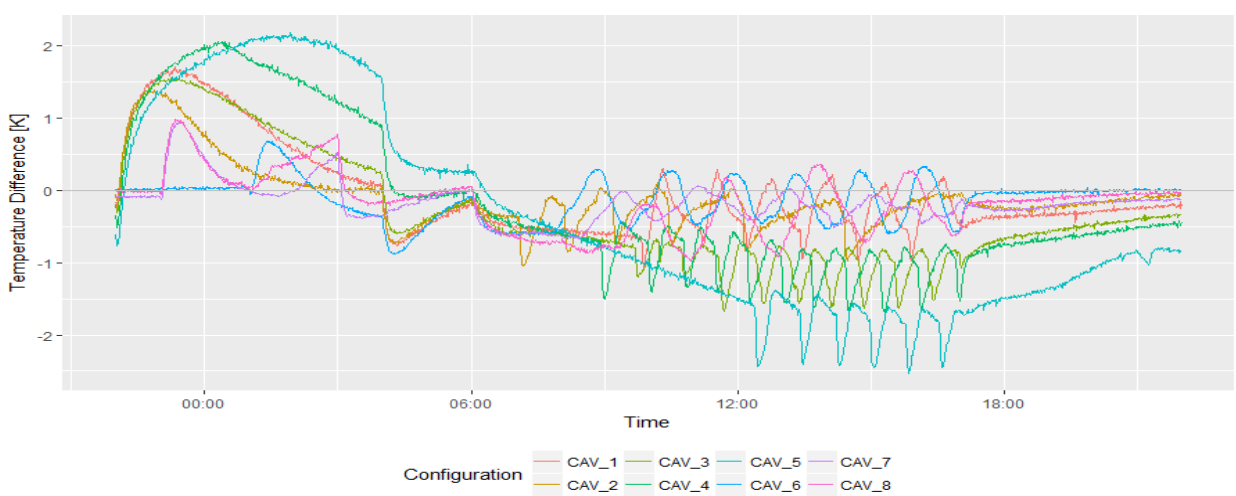


FIGURE 27 : ROUND 1 AVERAGE PCM SURFACE TEMPERATURE AND PLENUM AIR TEMPERATURE

This graph only serves to point out the temperature gradient seen by the PCM during the pre-cooling sequence and during the occupied hours. In all tests, the air temperature is always within 3.6°F (2K) of the PCM temperature. The product should be designed with a very low

thermal resistance to be able to drive heat in and out of the package with temperature gradient under 3.6°F (2K) at the expected setpoint conditions.

Testbed Thermal Mass Effects Analysis

Finally, leading from the discovery of the impacts of thermal mass in the test cell on effecting the absorption of thermal energy during pre-cooling, this section details the effects of the test cell's thermal mass on impacting the heat loads in the space.

Figure 28 shows the evolution of each energy sources over the day for each test condition and each cell. Cell A is the test cell and cell B is the baseline cell. The mechanical cooling energy is overlapped on the plot and is equal or close to being equal to the difference in bar height on each side of the x axis.

First, we notice that in both the test cell and reference cell, the pre-cooling sequence is driving heat out of the cell envelope and not so much out of the plenum. In other words, when the cold supply air arrives in the plenum, it is already hot and as lost most of its cooling potential. We also notice that the heat that is driven out of the cell envelope during pre-cooling is reabsorbed by the slab during the day, which balance at least partially the plug loads. The plenum heat is barely noticeable in the plots, and when it is, it is in the reference cell, where there is no PCM.

In Figure 29 we compare the difference in those sources heat between the two cells. The sum of those differences is equal to the cooling load reduction. This plot is a good evidence that the savings, when they occur are largely due to the cell envelope inertia. Most of the thermal mass of the envelope is assumed to be mostly in the floor slab.

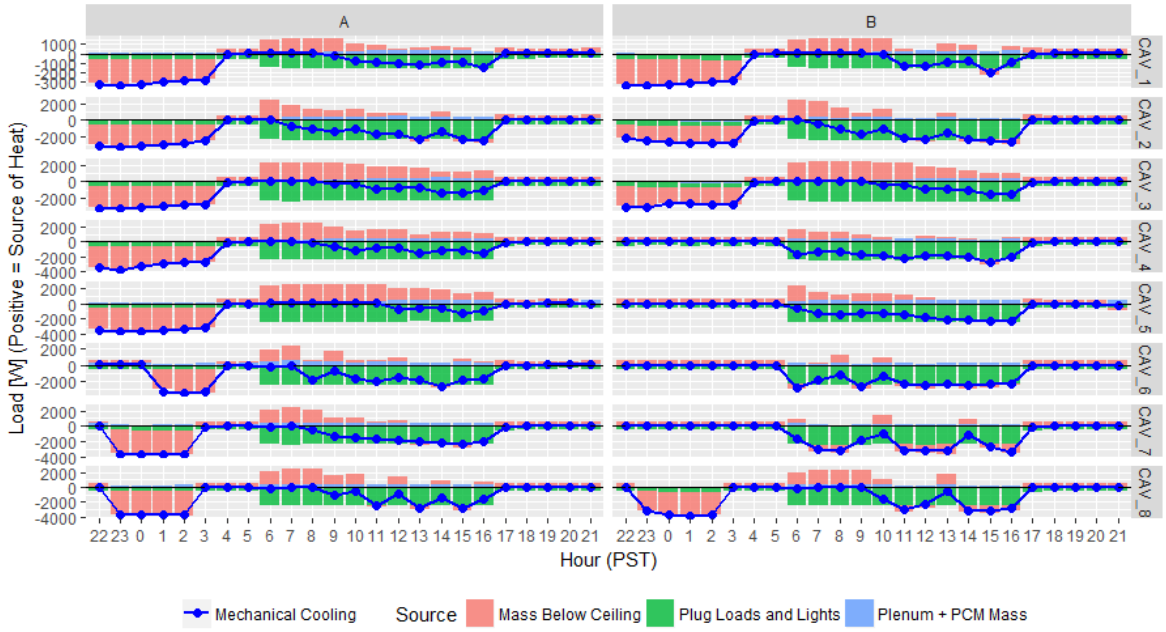


FIGURE 28 : ROUND 1 HEAT SOURCE ANALYSIS

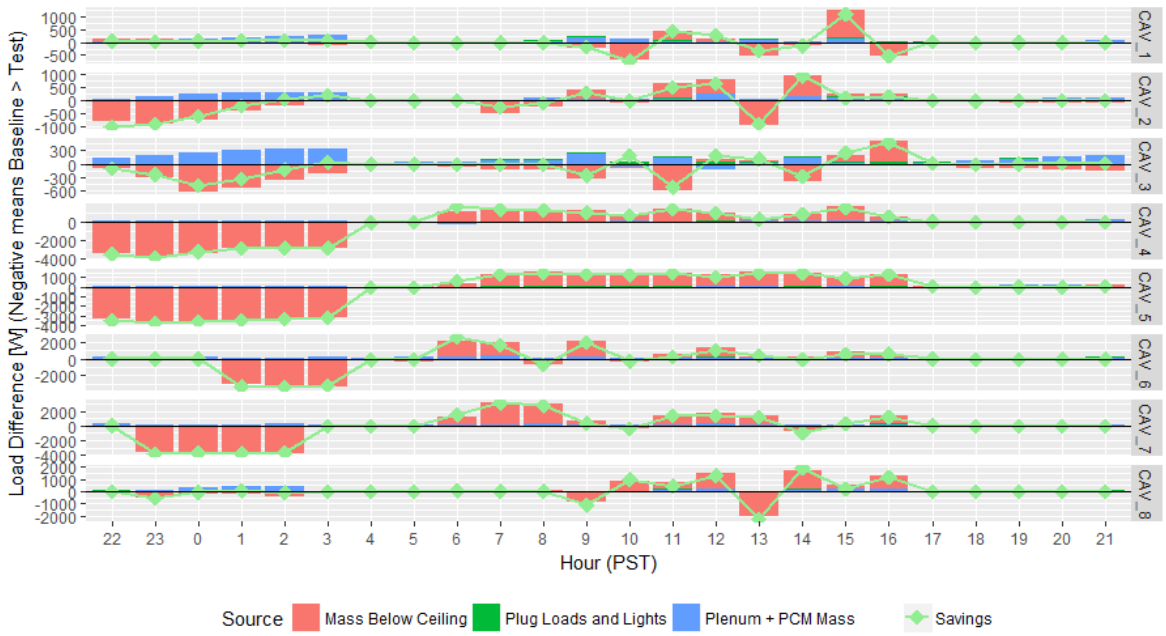


FIGURE 29 : ROUND 1 DIFFERENCE IN HEAT SOURCE LOAD BETWEEN THE CELLS AND LOAD SAVINGS

Round 1 Recommendations

Considering the results of this experiment, several suggested improvements were provided to Stasis to aid in the product development that might enhance its performance and further drive the cooling load shift and cooling energy reduction. These included:

- Evaluation and specifying a more suitable packaging material that was design both for fire protection purposes and heat transfer
- Product chemistry improvements to enable freezing and melting at plenum conditions observed during daytime operation and night time outside air-based pre-cooling.

One recommendation was also discussed for evaluation during the testing done in Round 2:

- Install a tarp at the underside of the structural beams to potentially increase the air velocity inside the return air plenum. This could have the effect of increasing heat transfer on the PCM packages.

Last, additional recommendations were made to improve upon test setup conditions to reduce the influence of the test cell's thermal mass on the pre-cooling strategy:

- Insulate the cell floor with 2 layers of 2" polyisocyanurate, followed by ¾" plywood.

Appendix C – Sensors Specifications

	Measurements	Sensors	Quantity	Uncertainty
Weather	Global and diffuse horizontal irradiance	Delta-T Devices SPN1-A990	1	+/- 5% | +/- 10W/m²
	Outside air dry bulb temperature	BAPI BA/10K-2(XP)-O-BB	1	+/- 0.1°C
HVAC (per cell)	Ducted air temperature (return, mixed and supply)	BAPI BA/10K-2-(XP)-SP	3	Calibrated at +/- 0.05°C
	Ducted air flowrate (supply and return)	Ebtron Gold BTM116-PC	2	+/- 3% (< 5000 fpm)
	Ducted air pressure (supply and return)	TEC DG-700	2	+/- 1% | +/- 5 iwg
	Chilled water temperature (supply and return)	BAPI BA/T1K-DIN-[0 TO 100F]-I-2"-BB	2	+/- 0.055°F
	Chilled water flowrate	Siemens Sitrans FM MAG 1100	1	+/- 0.2% (> 0.3 fps)
	Hot water temperature (supply and return)	BAPI BA/T1K-DIN-[32 TO 212F]-I-2"-BB	2	+/- 0.055°F
	Hot water flowrate	Siemens Sitrans FM MAG 1100	1	+/- 0.25% (> 0.3 fps)
	Fan Power	Circuit breaker measurements	1	+/- 2% (typically +/- 1%)
Loads (per cell)	Cell lights and heaters power	Circuit breaker measurements	6	+/- 2% (typically +/- 1%)
Cell and Plenum Conditions (per cell)	Cell air vertical temperature measurements (sensors mounted on 2 rods, north and south)	BAPI BA/10K-2-(XP)-SP	14	Calibrated at +/- 0.05°C
	Cell air thermostat temperature	BAPI BA/10K-2-(XP)-SP	2	+/- 0.1°C
	Dropped ceiling gap temperature	BAPI BA/10K-2-(XP)-SP	3	+/- 0.1°C
	Plenum air vertical temperature measurements (sensors mounted on 2 rods, north and south)	BAPI BA/10K-2-(XP)-SP	8	Calibrated at +/- 0.05°C
	Slab internal temperature	BAPI BA/10K-2(XXP)-1-8"-SS	3	+/- 0.05°C
PCM Conditions (test cell)	PCM surface temperatures	Littlefuse PR103J	18	+/- 0.05°C
	Plenum surface temperatures	Littlefuse PR103J	9	+/- 0.05°C
	Roof surface temperatures	Littlefuse PR103J	9	+/- 0.05°C