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The potential of phase change energy storage for office cooling load shifting

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Abstract. An analysis of the performance of four cooling approaches to meet daytime office cooling requirements for an open plan office was undertaken using a finite-volume model implemented in Matlab. The four cooling approaches simulated were: a conventional HVAC system (A), using extra ventilation on demand (B), using cool night time air to charge a store with RT15 (C) and charging a store with RT8HC using night time off peak electricity (D). Charging the thermal store with cool night time air reduced the yearly cooling demand by 39% compared to the conventional HVAC cooling system. Mechanical ventilation on demand achieved a reduction of 34% compared to the same conventional system. Using off peak electricity to charge the PCM store achieved a reduction of 15%, and successfully shifted daily cooling demands into night time operation during spring and autumns months. Due to the high capital cost of the thermal stores, the lowest cost of energy was obtained using extra ventilation on demand, with a cost of 84.45£/MWh for an assumed 20-years of operation.

Keywords:

Phase change materials, Thermal energy storage, Lumped capacitance numerical model, Night time cooling, Encapsulated latent heat storage, Office load shifting, HVAC.

1. Introduction

The demand for space cooling is rapidly growing worldwide and will be exacerbated by global warming [1]. Currently, it is estimated that 40 to 45% of the EU's total energy demand [1] is used in buildings. According to the European Union approved directive 91 in 2002 [2] and ratified in 2010, all new buildings in the EU are expected to reduce or consume “nearly zero” energy after 2020 [3]. Although currently marginal, the energy used for cooling and humidification is expected to grow and efforts should be made to reduce its associated emissions [1].

2. The developed simulation model

To assess the potential reduction in CO₂ emissions achievable by the integration of a compact latent heat storage container into a building space cooling system, a typical open plan office was chosen, based on that used in a study by Korolija et al [4]. The office building is a three-storey building schematically shown in Figure 1A with a 32 by 16 meter footprint, and floor-to-ceiling height of 3.5 meters. The proposed heating and cooling solution is the installation of a centralized air distribution system with a compact thermal store, heat pump and the centrifugal blower installed on the building rooftop, as seen in **Error! Reference source not found.** For the conventional approach, the thermal store and centrifugal fan are removed, retaining the heat pump to supply the required heating and cooling needed.



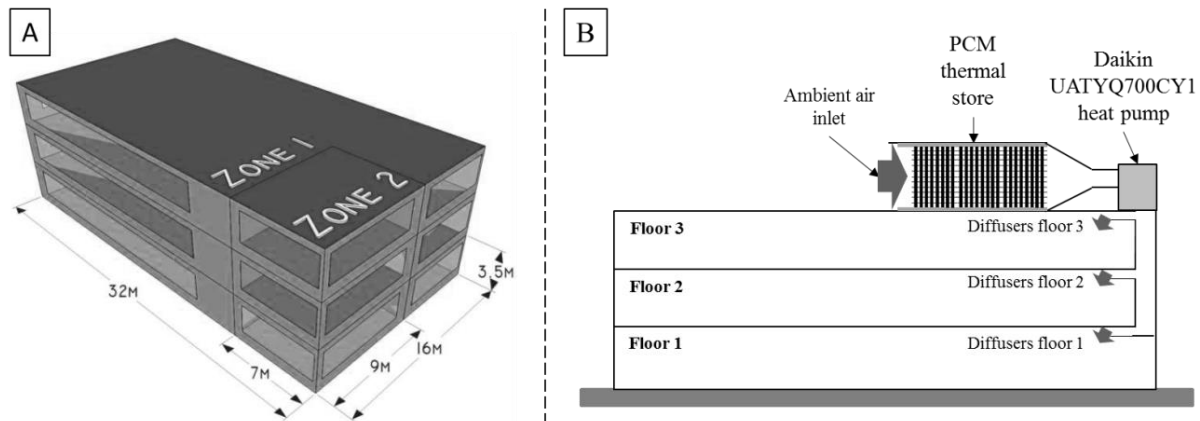


Figure 1 - Schematic drawing of the simulated 3-storey building (A), from [4] and a representation of the thermal store installed on the rooftop (B).

The indoor temperature of the building was considered uniform with no thermal stratification, heat exchange with the environment was calculated based on the recommended building fabric U-values from the CIBSE guide A [5] (0.18W/m².K for Cavity wall, 0.15 W/m².K for flat roof and ground floor and 1.98 W/m².K for the glazing). The predicted building yearly energy consumption was then compared with that from an EnergyPlus [6] simulation of the same building presented by Korolija et al. [4]. The heat balance calculation for each floor was based on equation 1. A fresh air ventilation rate of 10 l/s per person is required to meet the building ventilation standards. The auxiliary heat input (Q_{aux}) will be positive or negative according the required auxiliary energy to meet the set temperature.

$$\dot{Q}_{indoor} = \rho_{air} c_{p,air} (F_{air} + V_{person} \cdot N_{person}) \cdot (T_{amb} - T_{indoor}) + \dot{Q}_{solar} + \dot{Q}_{gain} + \dot{Q}_{LHS} + \dot{Q}_{aux} + \dot{Q}_{fabric} \quad 1$$

Adopting the recommended values from the CIBSE guides [5], the building indoor temperature was either heated to 20°C or cooled to 24°C during the occupied hours (from 07.00 to 19.00) [4], with the indoor temperature maintained between 12 and 28°C during unoccupied hours. Casual gains from office appliances and lighting were assumed to be 27 W/m², for the occupants casual gains were assumed to be 108W/person and the number of occupants per floor were based on an occupant density of 9 m²/person, in line with the CIBSE guide recommendations [5]. The building fabric heat transfer with the indoor air (Q_{fabric}) was calculated using equation 2.

$$\dot{Q}_{fabric} = \left[3 * Area_1 * (T_{wallN}(k) + T_{walls}(k) + T_{glassN}(k) + T_{glassS}(k)) + 3 * Area_2 * (T_{wallE}(k) + T_{wallW}(k) + T_{glassE}(k) + T_{glassW}(k)) + A_{space} * (T_{roof}(k) + T_{floor}(k) + 4 * T_{deck}(k) + 3 * k_{solid} * T_{solid}(k)) - T_{indoor}(k) * (12 * (Area_1 + Area_2) + (6 + k_{solid}) * A_{space}) \right] * \alpha_{indoor} \quad 2$$

All the exterior building surface temperatures were calculated assuming an instantaneous heat balance between the absorbed shortwave solar radiation, long wave radiation and convective heat transfer to the external environment. The external convective heat transfer correlation used was that from the EnergyPlus reference handbook [7], presented in equation 3. Coefficients D, E and F varied according to the external surface characteristics ([7]) and V_z is the local wind speed.

$$h_{cv} = D + EV_z + FV_z^2 \quad 3$$

To calculate the respective thermal mass of each building component exchanging heat with the indoor zone temperature (as expressed in equation 2), the materials used were aerated concrete for the inner wall leaf, concrete slabs for the floor and roof and a single layer of glass for the double glazing. The internal convective heat transfer was assumed to be 4 W/m².K for all vertical and 2 W/m².K for all horizontal surfaces, based on the EnergyPlus engineering reference [7]. To account for the extra thermal mass summed by office appliances (tables, desks, chairs, etc) a solid mass was included on each floor, occupying 2% of each floor air volume. The total building thermal mass (the thermal mass of the building fabric exchanging heat by convection with the indoor air) was 420675kJ/K, the cavity walls capacitance was 25128kJ/K, the glass capacitance was 19044kJ/K, the floors and roof combined capacitance was 290487kJ/K and the extra thermal mass 28672kJ/K per floor.

Passive solar gains were determined for each wall and then summed for each floor. Each wall had a specific angle of incidence with the sun calculated using the correlations detailed in [8]. To obtain each wall azimuth angle, each wall was oriented to a cardinal direction; the longer walls to north and south and shorter walls east and west. Their surface azimuth angle was their relative azimuth angle towards the south.

The predicted yearly cooling loads with the conventional HVAC system were considerably higher than the heating loads, as presented in Table 1 and in accordance to the values obtained in the EnergyPlus simulation by Korolija et al [4]. The highest cooling loads were obtained during the summer and autumn months and the majority of the yearly heating demand during the winter months.

Table 1 presents the average daily heating and cooling demand for each month. The cooling demand, peaks in August and is almost non-existent during winter months. To meet the base cooling load during the cooling season (when the daily cooling load surpasses its heating demand), from April to October, a daily average of 349 kWh/day cooling is required.

Table 1 – Predicted office daily heating and cooling loads for each month.

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Cooling load [kWh/day]	6	0	11	91	309	403	577	594	340	110	13	0
Heating load [kWh/day]	223	249	140	62	10	2	0	1	4	35	106	236

The average daily cooling required to maintain comfort values suggested by CIBSE [6], varies drastically between spring and autumn and summer months. The rooftop heat pump chosen provided 72.6kW of cooling and 69.8kW of heating [9]. To charge a thermal store in economy 7 tariff times, between 23.00 and 06.00, the total charging time available is 7 hours, consequently limiting the maximum storage capacity to 508kWh with the selected heat pump.

Two thermal energy storage system capacities were designed and simulated with the aim of reducing the cooling load of the building by: 1) using cold night time air temperatures (with a thermal storage capacity of 349kWh between 8 and 24°C using PCM RT15), and 2) using cold night time air temperatures with off-peak electrical cooling (with a thermal storage capacity of 508kWh between 1 and 24°C using PCM RT8HC) charged during economy 7 electricity tariff [10] times. Both thermal stores were designed to be fully charged in 7h and discharged in 10h, leading to charging/discharging cooling rates of 50/35kW for the store using RT15 and 73/51kW for the store using RT8HC.

2.1. Thermal storage model developed

A 2D finite volume model was developed in Matlab to simulate the performance of a store comprised of parallel slabs containing PCM in an office cooling system, similar to the geometry developed by Osterman et al. [11]. It assumes simple thermal diffusion, with no volume changes during the melting process and isotropic heat propagation within the PCM. Although the duct contained many PCM slabs it was assumed that flow between each pair of slabs would be similar allowing the simulation domain to be simplified. The model thus simulated air flow through a rectangular duct containing PCM slabs on its outer walls. The ducts side walls contained half of a rectangular PCM slab, the duct it was insulated on the top and bottom walls.

The air flow Reynolds number was calculated using the hydraulic diameter of the air gap between the slabs. The laminar Nusselt number was calculated using the expression for constant heat flux detailed in the VDI heat atlas [12]. For turbulent flow regimes, the Gnielinski's expression for turbulent pipe flow was used, also detailed in [12], using the hydraulic diameter of the flow passage .

2.2. Storage model validation

To determine the accuracy of the numerical models developed, an open-loop air flow experimental rig was assembled to test a developed thermal store. The temperature of the air supply was controlled using a shell and tube heat exchanger with a silicon oil circulating on the shell side, with temperature controlled by a Huber Tango temperature control unit. A centrifugal fan mounted on the air inlet circulated air through the shell and tube heat exchanger, delivering the required air flow to the thermal

store, as shown in Figure 2B. A PCM thermal store was then constructed to verify the accuracy of the algorithm developed. It was constructed from 1mm thick HDPE sheet and wooden battens. Figure 2A presents a photograph of the parallel slab store constructed. It consisted of 5 PCM slabs 9 mm thick of 1000x1000 mm dimensions with 12 mm air spaces between them. It contained 40 L of PCM. Wooden battens were inserted between the PCM slabs to ensure a 12mm air gap was maintained and enabled the 1m long air flow path to be extended to 5m as seen in Figure 2B. Using a Huber Tango the store inlet air temperature could be set to temperatures between 5 and 30°C to fully charge and discharge the PCM store.

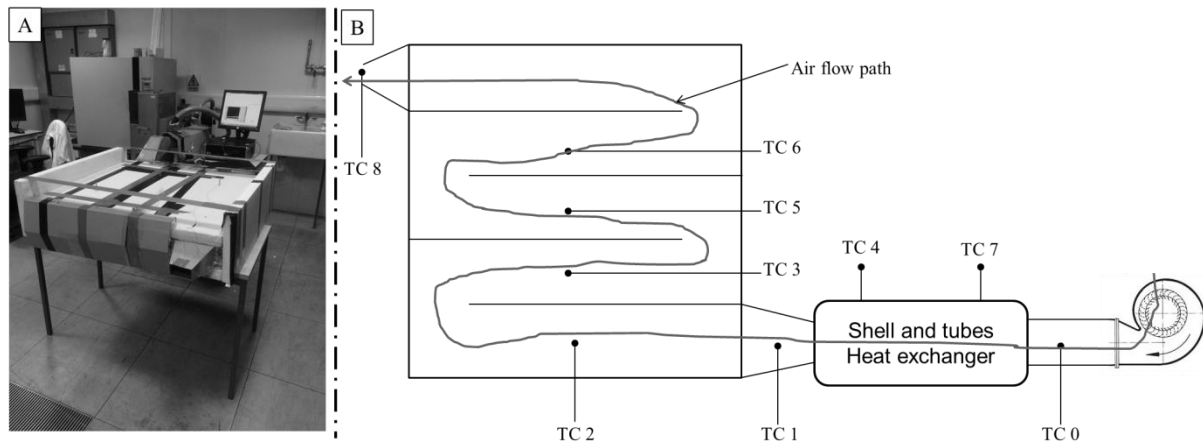


Figure 2 - General view of the assembled parallel slab store (A) and a schematic diagram showing the thermocouples location in the test rig (B).

2.2.1. Shape stabilized PCM used

A shape stabilized PCM composite was prepared in situ by blending Croda 21, an organic ester mixture developed by Crodatherm [13] with a 18°C melting point, with an enhanced polymeric adsorbent and additional materials in collaboration with Phase Energy Ltd [14]. The composite had a higher thermal conductivity and an extra 10 J/g enthalpy compared to the base mixture of Croda21 with the block copolymer only as seen in Table 2.

Table 2 - Material properties of the selected PCMs used in the experiment.

Material (%wt)	$T_{\min} - T_{\max}$	H_{stored}	λ_{solid}	λ_{liquid}	Price	
	°C	kWh/m ³	W/m.K		£/kg	£/kWh
Croda 21		53	0.180	0.150	4.00	69.7
Croda 21/adsorbent (80/20)	12-24	40	0.200	0.180	4.02	91.9
Croda 21/composite adsorbent (77/23)		43	0.470	0.350	3.50	81.1

The shape stabilized PCM produced could store up to 173.2J/g in the temperature range between 6 and 24°C, corresponding to a volumetric storage capacity of 45kWh/m³. The thermal store developed had a storage capacity of 1.89kWh over the same temperature range.

Figure 3 presents the experimental results obtained and the simulation predictions. The obtained simulation results are in good agreement with the experimental results obtained (both charging and discharging enthalpies have a relative error of less than 5%), with a slight variation in the heat transfer rate during the charging process (freezing of the PCM). This might be due to dimensional changes not considered during the freezing of the shape stabilized PCM slab (contraction of the storage material allowing the formation of air gaps within the PCM slabs).

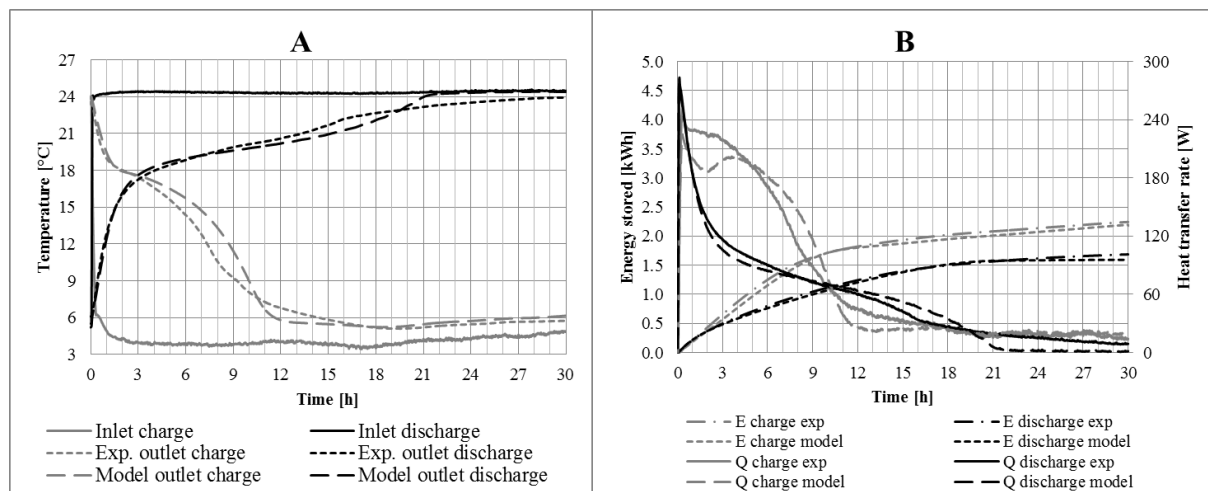


Figure 3 - Experimental results and simulation predictions for the parallel slab store outlet temperature (A), heat transfer rate and storage capacity (B).

2.3. Thermal storage optimization

To cool the simulated office building, larger flow rates are required, achieved with thermal stores with no baffles in the air flow. A simulation was carried out testing the day time cooling reduction potential of using free night cooling with the RT15 [15] composite. The possibility of shifting cooling load with night time charging was simulated with RT8HC [15]. The models were used to simulate a scenario in which each latent heat store was fully utilised. In regards to the operational temperature range, the store filled with RT8HC was charged at 1°C and the store filled with RT15 charged at 7°C; both were discharged at 24°C.

Figure 4 presents the simulated results for the parallel slab optimized stores with and without fins. The model simulated a full utilization scenario for each latent heat store. The store using RT8HC was charged at 1°C and at 7°C using RT15; both were discharged at 24°C. Both stores were designed to provide 10 hours of cooling at a constant rate of 50.8kW for the store filled with RT8HC and 34.9kW for the store filled with RT15. Both stores were charged in 7 hours, the predicted charging rates were 72.6kW for the store filled with RT8HC and 49.86kW for the store filled with RT15. The centrifugal fan flow rate was controlled between 50% and 150% of the nominal flow rate, according to the heat demand needed.

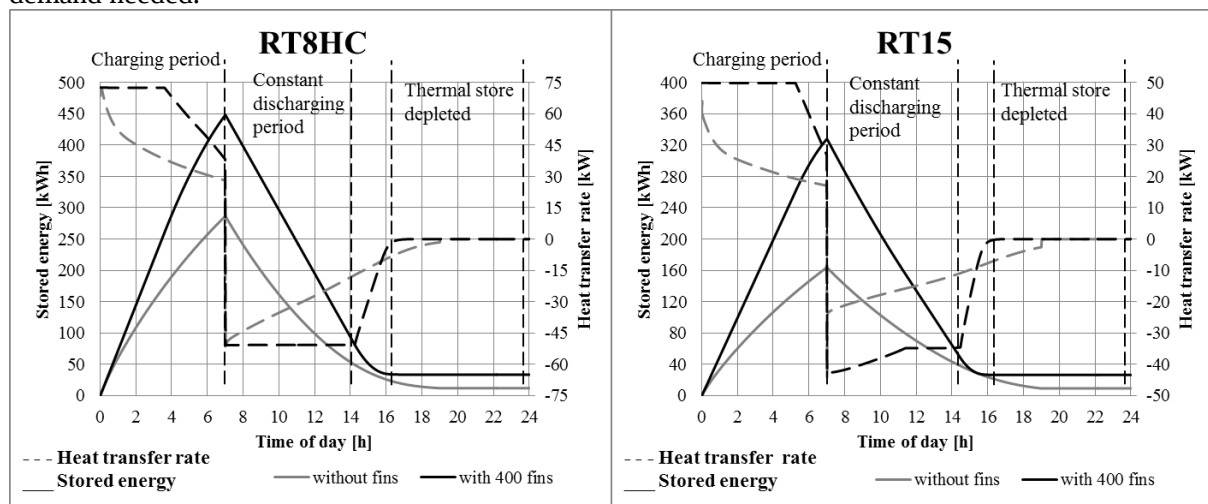


Figure 4 - Simulation predictions for the optimized parallel slab thermal store with RT8HC and for the thermal store with RT15 with and without fins.

3. Yearly simulation results

Using London Gatwick design weather data and the building design parameters, simulations were performed with HVAC systems and thermal stores for a whole year to determine potential enhancements in the daily performance and consequently reductions in yearly energy requirements with each approach. Four cases with extra fresh air provision were compared to the conventional HVAC system (A): on demand (B), using a store filled with RT15 (C) and using a store filled with RT8HC and with electrical night cooling (D). The integration of the thermal store into the space cooling network is schematically illustrated in Figure 5A.

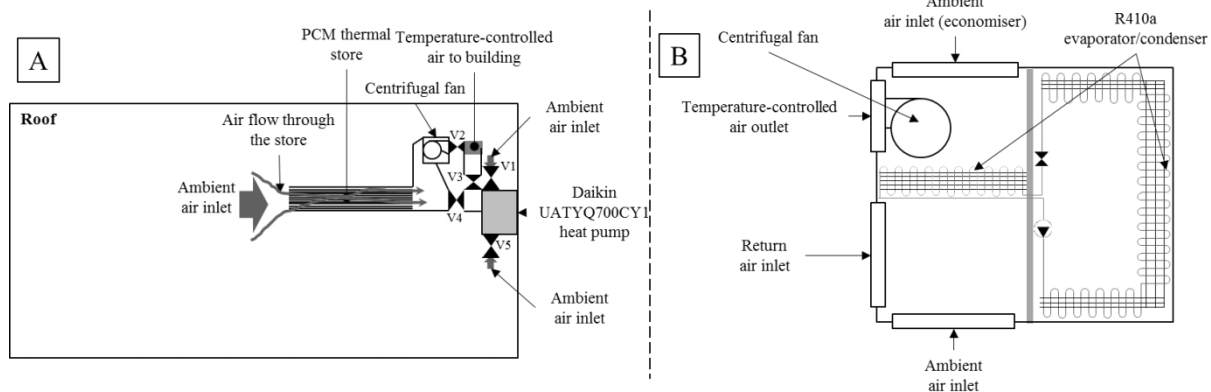


Figure 5 - Schematic representation of the parallel slab latent heat storage integration into the building HVAC system (A) and detailed view of the rooftop heat pump (B)

All cooling systems modelled were activated during daytime office hours when the indoor air temperatures were above 22.5°C. The heat pump was activated only if the indoor air temperature increased above 24°C. The coefficient of performance (COP), used for the heat pump during office working hours was 3 [9]. For the simulation using extra ventilation on demand, valve V2 would open (shown in Figure 5) and the centrifugal fan activated when indoor temperature increased above 22.5°C, allowing colder outdoor air to be introduced reducing the building cooling load during office working hours.

For the simulation with a thermal store filled with RT15, during the night valve V1 and V4 would open, directing the outside ambient air flow to cool/charge the thermal store. During office working hours (from 07.00 to 18.00), if the indoor temperature increased above 22.5°C valve V4 closes, V2 and V3 open and the centrifugal fan is activated, reversing the air flow through the store and consequently discharging it to provide space cooling to the office building. The air flow from the centrifugal fan was allowed to vary between 50% and 150% of its nominal flowrate.

For the simulations using the heat store filled with RT8HC, the heat pump was assumed to charge the latent heat store with a COP of 4.5, based on the Daikin UATYQ-CY1 rooftop air conditioners service manual [9]. During store charging, valve V5 and V4 are open, allowing the ambient air flow to be cooled by the heat pump if the ambient air temperature is above the required cooling temperature of 3°C. To effectively work and improve energy efficiency, all of the specified cooling systems are only operated if their COP calculated according to equation 4, is above 3.5. For the extra ambient fresh air provision on demand system, the ambient temperature is used instead of the thermal store outlet temperature. Using the centrifugal fan specifications from SODECA [16], the 3kW electrical input to the fan had an estimated 65% efficiency reported at 15 000m³/s (varying its flowrate and electrical consumption according to the required cooling demand), meaning the designed cooling systems were only activated for cooling loads above 16kW.

$$COP(t) = \frac{\rho \times \dot{V}_{fan}(t) \times C_p \times (T_{store\ out}(t) - T_{indoor}(t)) \times \eta_{fan}(t)}{\Delta P_{fan}(t)} ;$$

4

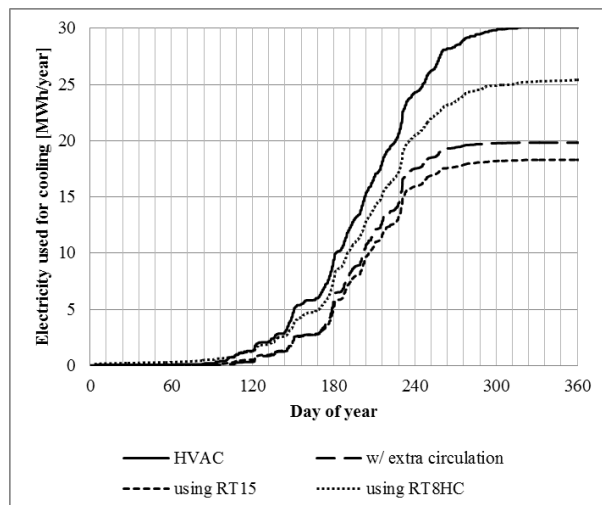


Figure 6 – Yearly total electricity consumed for cooling in the building for all cases studied.

Figure 6 presents the aggregated yearly results for each studied case. It can be seen that the obtained reduction in electricity use for cooling by using only extra air circulation can reach up to 34%. Using RT15 with ambient night time cooling used 39% less electricity for cooling. Shifting daily cooling loads with RT8HC obtained a yearly electricity reduction of 15%. The poor performance values are due to the heat gains from ambient occurring when the thermal store is not being used.

The thermal storage system with RT8HC and night time cooling provided the greatest reduction in electricity use during peak times due to its ability to effectively shift the building cooling load.

Figure 7 presents the daily cooling load for the heat pump and COP of the simulated cooling approach for each season. The most significant levels of energy reduction with the extra ambient fresh air provision on demand are predicted mainly during the spring and autumn months and the reductions are minimal in summer months due to the higher ambient temperature.

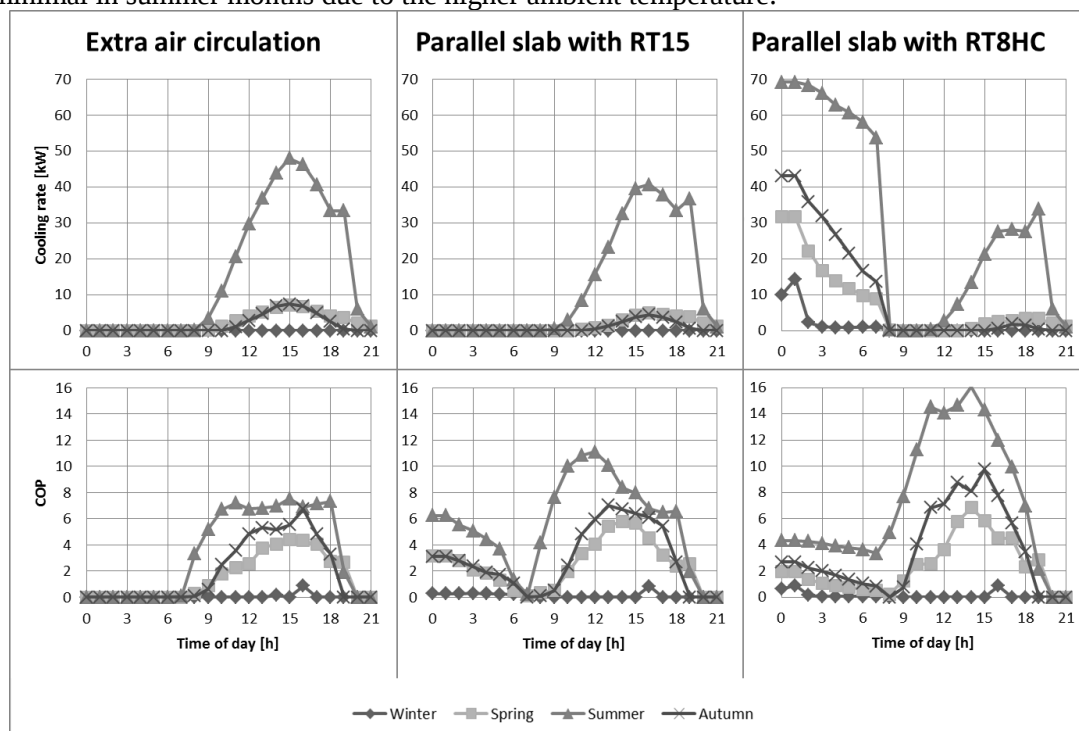


Figure 7 - Seasonal results for the heat pump daily cooling loads and system COP with extra air circulation, for the parallel slab thermal stores with RT15 and with RT8HC

The reduction in the cooling load during office working hours in summer is larger using a store filled with RT15 with free night cooling; however, the overall gain is reduced by the additional energy required for the fan to charge the store during the night. Using RT8HC with electrical night cooling successfully enabled daily cooling loads to be shifted into off peak times in both spring and autumn months and significantly reduced the cooling load during the summer months, shifting the summer peak cooling load to 19.00. Full load shifting during summer months was prevented by the capacity of

the selected heat pump (due to restricted charging period, the maximum energy able to be charged was $7h \times 16.13kW \times 4.5 = 508kWh$, less than the maximum required in summer months, 594kWh).

The increase in predicted COP for the thermal storage system using RT15 compared to the extra ambient fresh air provision on demand system was more pronounced during summer months and only slightly higher during spring and autumn months. The predicted daily COP profile for the storage system with RT8HC was very similar to that of the system with RT15 during spring and autumn months, but considerably higher during summer months.

3.1. Economic comparison of all modelled systems

A general financial assessment was made to assess the three approaches described and compared to the conventional HVAC system. Table 3 presents the initial capital expense (CAPEX) and operational expense (OPEX) costs for the 4 cases studied. For the heat pump system, Daikin UATYQ700MCY1 [9] was selected due to meeting the heating and cooling required. An industrial-scale centrifugal fan [17] with a nominal flow rate of 15 000m³/h was selected from SODECA [16]. For the parallel slab thermal stores, assuming that the shape-stabilized PCM slabs are enclosed in steel sheets and that compressed corrugated sheets are included as fins in the air gaps to increase heat transfer area, a cost of 650£/ton was used [18] as a cost for the metal.

Due to the large size of the designed systems, the whole system installation costs were assumed to be 5% of the initial CAPEX [19]. In the thermal stores, the composite PCM was the largest capital component, representing 53% of the initial CAPEX. Operation and maintenance costs were assumed to be 0.9£/m² of office space, in accordance with [20]. The waxes used were assumed to have an indicative price of 4£/kg [15] and the composite adsorbent material developed by [14] an indicative price of 2.65£/kg, when mixed with a 80/20 mass ratio providing a composite storage material cost of 3.73£/kg. For the standard electricity tariff, Npower provided a price of 0.1664£/kWh with a standing charge of 55.1£/year. Using economy 7 electrical tariffs, Npower provided a peak tariff of 0.2131£/kWh and an off-peak tariff of 0.0765£/kWh, with the same standing charge. The levelized cost of energy (LCOE) for each system was calculated using equation 5, from [21].

$$LCOE = \frac{\left(\frac{CAPEX \times r}{1 - (1+r)^{-n}} + OPEX + E_{used} \times E_{tariff} \right) \times n}{W_{produced} \times n} \left[\frac{\text{£}}{\text{kWh}} \right]; \quad 5$$

Table 3 – CAPEX and OPEX costs comparison for the systems studied.

Component		HVAC (heat pump only)	+extra circulation	Parallel Slab		Ref.
				With RT15	With RT8HC	
Thermal Store cost [£]	Insulation+ structure			1 057	1 089	[22]
	PCM enclosure			6 317	6 527	[18]
	PCM + composite			32 474	33 556	[15]
	Centrifugal fan		3 142			[23]
Heat pump cost [£]		17 305				[24]
Installation costs [£]		865	1 022	3015	3081	[19]
CAPEX [£]		18 170	21 469	63 310	64 700	
Yearly system maintenance [£/year]		1 382				[20]
Yearly electricity used [kWh]		7 206	5 504	5 245	6 087 (econ7)	
OPEX [£/year]		8 588	6 886	6 627	7 469	-
LCOE [£/MWh]		97.17	84.45	120.77	129.89	

For a 20-year life cycle with an annual interest rate of 8%, the LCOE calculated for each system is presented in Table 3. The system with extra air circulation presents the lowest LCOE due to the reduction in the energy required to meet the cooling load achieved with ambient air cooling. Due to the high CAPEX costs of the thermal stores, their yearly reduction in electricity costs compared to the conventional system were not enough to provide a more economical cost of energy.

4. Conclusions

Medium to highly glazed buildings with lower U-value building fabric generally consume more energy for cooling than for heating due to the high solar gains transmitted through the glazing, high

occupancy and electronic devices emitting heat. The use of extra mechanical ventilation gave the lowest price for cooling of 84.45£/MWh. This is a consequence of the typical UK external ambient air temperature during office working hours being mostly below 24°C, (97% for the London Gatwick IWEW weather data).

Using a thermal store with RT8HC with off peak electrical cooling successfully shifted office daily cooling loads during spring and autumn months and in summer months was capped by the capacity of the heat pump used. Night time ambient cooling with RT15 obtained the highest reduction for cooling, but the high capital cost makes it less feasible than using extra ventilation only.

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