Performance Analysis and Control Strategies to Enhance Free Cooling with an Active PCM-Heat Exchanger

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Abstract

This paper presents a simulation case study of an active thermal energy storage device with phase-change materials (PCM) for free-cooling purposes. The charging/discharging of the PCM, arranged in layers in a heat exchanger, is done through a controlled air stream passing through these layers. First, the effect of airflow rate on charging/discharging time is investigated. Then, different control strategies targeting reduction in peak demand and HVAC system sizing are evaluated. Results show that for the present case-study 14% reduction in energy consumption and 10% enhanced energy flexibility could be achieved using the PCM heat exchanger (PCM-HX) for free cooling.

Introduction

The role of buildings as flexible loads is becoming more important; they can act as energy generators, energy storage, or/and controllers of demand.

In this regard, the Annex 67 of the IEA Energy in Buildings and Communities Programme (IEA-EBC) defined energy flexible buildings as those with “the ability to manage [their] demand and generation according to local climate conditions, user needs and grid requirements”.

Increasing energy flexibility for the design of smart energy system and buildings is influenced by four important factors (Reynders 2015):

- physical characteristics of the building
- HVAC systems and storage equipment
- adequate control systems and strategies
- comfort requirements

In this context, thermal energy storage (TES) along with appropriate control strategies is a key factor to increase the energy flexibility of a building by reducing the mismatch between supply and demand for heating or cooling (Zhu, Ma et al. 2009, Tabares-Velasco, Christensen et al. 2012, Klein, Herkel et al. 2017).

PCM applications

Zhu, Ma et al. (2009) presented an overview of research conducted on PCMs concerning their dynamic characteristics and energy performance in buildings. This overview classified the possible building applications of PCM into four categories:

- free cooling
- peak load shifting
- active building systems
- passive building systems

In passive applications, PCMs are integrated into the building envelopes to increase the effective thermal storage capacity in order to better energy saving and temperature regulation. The main problem with incorporating PCMs in the building envelope is the difficulty of exchanging a high rate of heat between the air and the PCM. Passive ways for cooling may also not be as effective in extreme climate conditions. This is mainly because the outdoor ambient air temperature does not decrease sufficiently at night. Stetiu and Feustel (1998) found that for climates with relatively high ambient temperatures (above 17 °C) during the night, it would be beneficial to force the supply air along the storage surfaces to facilitate good heat exchange.

On the other hand, active thermal energy storage systems with PCM using room air or outdoor air (or a mix of the two) can provide the following advantages:

- flexibility in heat transfer area
- steady conditions for charging/discharging energy
- adjustment of the melting point of the PCM
- integration with mechanical systems
- applications in retrofit projects

Free cooling

Free cooling has been defined as “cool[ing] the building interior with or without minimum electricity usage”. (Givoni 1984, Santamouris and Asimakopoulos 1996, Szokolay 2012).

In free cooling, a TES is used to store the “ambient cold” to be used later during hot daytime periods (Stritih 2003, Zalba, Marín et al. 2004). Thermal energy for free cooling can be stored by using: sensible heat storage, latent heat storage or the combination of both (Dincer and Rosen 2002, Sharma, Tyagi et al. 2009). In this application, Latent Heat Thermal Energy Storage (LHTES) by PCMs is often preferred over other storage techniques due to its high energy storage density and isothermal storage process (Waqas and Din 2013). Cool air during the night is used to solidify the PCM and the accumulated cooling capacity is then extracted when required. For free cooling systems, PCMs have to be selected so that the cooled air temperature is within the range of human comfort, a zone

which can vary significantly according to a number of factor (Heier, Bales et al. 2015).

Overview of the paper

The present study investigates a free cooling system with active PCM heat exchanger (PCM-HX), in order to control the cooling load of an office space during summer. By shifting the peak load away from the peak hours of electrical demand using PCMs, the peak load may be “divided” throughout the day thus reducing the highest peaks and enhance energy flexibility. This system (PCM-HX) also allows full control of the process during the charging/ discharging of the PCM, thus optimizing the performance of the whole system.

Active PCM-HX

Geometry of thermal zone

The test building is 4 m wide by 4 m long by 3.2 m high, and has no interior partitions. In this study, we are using the temperature capacitance multipliers approach for a typical office zone. Hong and Lee (2019) recommended using this multipliers approach to increase the zone air capacity equivalently to represent the effective storage capacity of the zone internal thermal mass. They recommend multiplier of 3 to 6 for lightly furnished offices, 6 to 10 for typical offices, and 10 to 15 for heavily furnished offices. According to this, we are using a capacitance multiplier of 10 to represent a typical office zone. The building has a double glazing window in the middle of its facade facing south (40% window-to-wall ratio (WWR)). The U-value of the window is $1.2 \text{ W}/(\text{m}^2\text{K})$ and its transmittance is 0.8. The bottom opaque section of the facade is at 0.8 m, which is at the work plane height, as shown in Figure 1. Auxiliary mechanical cooling can be provided with a cooling coil inside the room (bottom right of Figure 2).

Figure 1: Schematic of the office zone with PCM-HX

General concept

In the present study, the integration of an active PCM-HX for an office zone is investigated.

The PCM-HX system is part of an insulated HVAC duct system (Figures 2a and 2b). A similar approach (though not identical) has been presented by Stathopoulos, El Mankibi et al. (2016). A circulation fan drives an airflow between the channels in order to charge/discharge the PCM. This circulation fan can be either ON or OFF (in other words, no operation at partial flowrates). If cooling

is needed during on-peak hours: the fan is turned on and dampers are opened, in order to discharge the PCM-HX. Note that in this concept no cooling coil is used to lower the temperature of the PCM-HX.

The following modes of operation are implemented for the PCM-HX:

- **Charging mode:** The circulation fan is turned on to bring outside air through PCM-HX during night to charge the PCM (Figure 2a). In this mode of operation, dampers are close and there is no exchange between the plenum and the room air.
- **Discharging mode:** The dampers are opened and circulation fan is turned on during hot day to discharge the PCM (Figure 2b). Therefore, the discharge mode enables full air exchange with the room during peak load hours.
- **Standby mode:** In this mode, the fan is off and dampers are closed. In this mode of operation, there is no exchange between the plenum and the room air or outside air.

a. PCM-HX in charging mode- during night

b. PCM-HX in discharging mode – during day

Figure 2: PCM-HX in isolated ceiling plenum (note: drawing not at scale)

PCM-heat exchanger (PCM-HX)

Figure 3 presents a schematic of the proposed PCM-HX. This device consists of an arrangement of PCM panels that allows an airflow to pass between them. The length,

Figure 3. Schematic of PCM-HX. Isometric view (example with two PCM panels in series)

width and thickness of each PCM panel are denoted by $L_{PCM} = 1.2$ m, $W = 1$ m and $t_{PCM} = 5.2$ mm respectively. Several PCM panels may be installed “in series” within the PCM-HX; e.g., if n PCM panels are used then: $L_{HX} = n \times L_{PCM}$. For example, Fig. 3 shows two panels connected in series, with a total length $L_{HX} = 2L_{PCM}$. It should take into account, for practical application the length of the PCM-HX would be more than L_{HX} , because there is threshold value to PCM panels connected in series. According to our investigation (Morovat 2019) increasing the length of the PCM-HX leads to higher power output and also higher air outlet temperature, but more time is required to charge/discharge the system. From a building operation standpoint, the charging/discharging time required is generally around 8 h or less, although in some applications (such as solar) one day discharge time could be of interest. Taking this into account a PCM-HX length equal to 2.4 m (used in this paper) is close to the “optimum length” when a discharge time of 8 hours or less is required. Conversely, longer PCM-HX devices imply higher pressure drops through the system, and consequently increasing fan power required.

This geometry is appropriate for applications with high energy storage density and only low temperature differences between air and PCM are available, such as in free-cooling applications. The simplicity of the proposed geometry makes this structure more adaptable to different configurations. The PCM used in this paper is DuPont Energain™. Table 1 provides an overview of thermo-physical properties of the PCM used in the present paper.

Table 1. Properties of the PCM (Dermardiros 2015)

Element	Quantity
Thickness (base case)	5.2 (mm)
Density	850 (kg/m^3)
Average Specific Heat	3500 ($kJ/kg \cdot K$)
Peak Melting Temperature	22.3 ($^{\circ}C$)
Latent Heat of Fusion	70000 (J/kg)
Peak Freezing Temperature	17.8 ($^{\circ}C$)
Melting range	20.8 – 24.5 $^{\circ}C$
Conductivity, solid	0.22 ($W/m \cdot K$)
Conductivity, liquid	0.18 ($W/m \cdot K$)

The melting point of the PCM plays an important role in the design of the storage unit (Arkar, Vidrih et al. 2007).

For free cooling applications, PCMs should be selected in such a way that the cooled air temperature of the PCM-HX be within the range of defined comfort levels (Butala and Stritih 2009) which is between 23 $^{\circ}C$ and 27 $^{\circ}C$ for summer season. Therefore, for free cooling systems, the melting temperature of the PCM should be between 19 $^{\circ}C$ and 24 $^{\circ}C$ (Butala and Stritih 2009). In a mechanically conditioned space, the best performance is achieved when PCM melting temperature is around (± 2 $^{\circ}C$) the setpoint temperatures of the room (Heier, Bales et al. 2015, Guarino, Athienitis et al. 2017). In view of these considerations, a PCM with melting temperature of 22.3 $^{\circ}C$ is used in this paper. Such a material is suitable for the cooling season and is commercially available.

In addition, melting range of PCM defines the temperature that can be maintained in the room. Determination of the actual melting temperature range of PCM means to find the starting melting temperature at which PCM starts melting and the ending melting temperature at which the melting process is finished, as mentioned in table 1. In this study, the melting temperature range of the PCM is 20.8–24.5 $^{\circ}C$.

In this paper, the PCM-HX system has 2.4 m length, (corresponding to two PCM panels in series), and the analysis is limited to eight sheets of PCM. Figure 4 provides a schematic of the PCM-HX with 6 air channels between layers.

Figure 4. Schematic of PCM-HX with six air channels.

According to our investigation (Morovat 2019), the charge/discharge time is inversely proportional to the number of channels. The single channel system would take more than 7 hours to discharge, and would not be useful for fast peak reduction measures. In contrast, a 6 channel PCM-HX system provides faster and more easily controllable storage of heat/cool. Therefore, this

configuration is chosen for investigation of free cooling with the PCM-HX.

Methodology

An explicit finite difference formulation, implemented in Python 3.6, is used to numerically solve the heat transfer equations corresponding to the heat transfer phenomena in the office. Hed and Bellander (2006) presented a similar method to simulate a PCM air heat exchanger. In the present study, an enthalpy-temperature function for the PCM accounts for enthalpy changes during the phase change

Governing equations

Equation 1 is a finite-difference formulation of the heat balance equation in the office zone.

$$\sum_j U_{ij}^t (T_j^t - T_i^t) + \sum_k U_{ik}^t (T_k^t - T_i^t) - \frac{C(T)_i}{\Delta t} (T_i^{t+1} - T_i^t) + \dot{Q}_i^t = 0 \quad (1)$$

To assure numerical stability in the solution, the time step must be chosen according to the stability criterion defined in equation 2:

$$\Delta t = \frac{\rho C_p (T) \Delta x^2}{2k} \quad (2)$$

Equation 3 is the air control volume differential equation:

$$\dot{m} C_p dT_{air} = depth \cdot d \cdot [h_{conv,front} (T_{front} - T_{air}) + h_{conv,back} (T_{back} - T_{air})] \quad (3)$$

Equation 4 is solved to find the outlet air temperature. This information is used to calculate the equivalent heat source at the air channel control volume (Equation 5).

$$T_{air,outlet} = T_{air,inlet} \cdot \exp\left(\frac{-h_{conv} A_{PCM}}{\dot{m}_{air} C_{p,air}}\right) + T_{surface} [1 - \exp\left(\frac{-h_{conv} A_{PCM}}{\dot{m}_{air} C_{p,air}}\right)] \quad (4)$$

$$\dot{Q}_{PCM} = \dot{m}_{air} C_{p,air} (T_{air,outlet}^t - T_{air,inlet}^t) \quad (5)$$

Equation 6 calculates the extracted heat by an auxiliary mechanical cooling with proportional control (PI control).

$$\dot{Q}_{cooling}^t = k_p (T_{setpoint}^t - T_{room,air}^t) + k_i \int (T_{setpoint}^t - T_{room,air}^t) \quad (6)$$

where:

k_p , proportional gain of the controller, W/°C

k_i , proportional gain of the controller, W/(°C·s)

Equation 7 calculates energy consumption of the auxiliary cooling:

$$W_{cooling} = \frac{\dot{Q}_{cooling}}{COP} \quad (7)$$

In this study, the fan energy is considered, but it is shown to be negligible compared to the energy transferred into the PCM-HX system. The fan energy requirement was minimal since the panel surface is flat and smooth. Entrance and exit effects are taken into account while effects such as ducting and flow diverting/adding

transitions are not considered. To estimate the required fan power, first, the friction losses are calculated. Losses through the system vary according to the Darcy-Weisbach equation (ASHRAE, 2009):

$$\Delta P = f \frac{L}{D_h} \cdot \frac{\rho v^2}{2} \quad (8)$$

The Blasius relationship is developed for smooth pipes in turbulent range; it is often used due to its simplicity:

$$f = \frac{0.3164}{Re^{0.25}} \quad (9)$$

Additionally to friction losses, other miscellaneous losses due to the entrance and exit effect is considered:

$$\Delta P_{tot} = \Delta P + \sum K \frac{\rho v^2}{2} \quad (10)$$

The fan power required is:

$$P_{fan} = \frac{(A \cdot v) \Delta P_{tot}}{\eta_{fan}} \quad (11)$$

Modelling of specific heat

A continuous curve based on a skewed normal distribution requiring five parameters is used to obtain the effective heat capacity of the PCM as a function of temperature (Equation 8). This equivalent specific heat is then updated at every time step, and used as input into finite difference model. Kuznik, Virgone and Noel (2008) have characterized the material using the DSC method with a heating and cooling rate close to rates commonly found in buildings – 0.05 K·min⁻¹.

$$C_p(T) = \Delta h \frac{1}{\sqrt{2\pi}} \exp\left(\frac{-(T - T_c)^2}{2\omega^2}\right) [1 + \operatorname{erf}\left(\frac{\text{skew}(T - T_c)}{\sqrt{2}\omega}\right)] + C_{p,average} \quad (11)$$

As can be seen from Figure 5, there is usually a shift between the peak temperature at which melting occurs and at which freezing occurs, which is called hysteresis. More details about the presented PCM-HX evaluation is presented in (Morovat 2019).

Figure 5. Melting and freezing curve

Results

Results of this paper are presented in two parts:

(a) Evaluation of the charging/discharging time of the PCM-HX.

(b) Application of the free cooling with PCM-HX under different control strategies.

Charging /discharging time

A high temperature difference between PCM melting point and the incoming airflow facilitates freezing the PCM the charging process at night. If the temperature difference is small, high airflow rates are needed to solidify the PCM completely in the required period (Turnpenny, Etheridge et al. 2000, Yanbing, Yi et al. 2003, Zalba, Marín et al. 2004, Saman, Bruno et al. 2005).

In free cooling applications, the nighttime period that allows charging the PCM is rather short. Consequently, higher airflow rates can ensure the maximum charging of the PCM storage. For free cooling applications, it has been recommended that the charging process airflow rates be three to four times those used during discharge process (Arkar and Medved 2007, Arkar, Vidrih et al. 2007, Medved and Arkar 2008). Conversely, during the discharge process, lower flow rates are better than high flow rates to maintain the outlet air temperature within the defined comfort levels (Arkar, Vidrih et al. 2007).

In this regard, the total mass flow rate considered is 600 kg/h in charging (during night), and 250 kg/h in discharging (during hot day). Table 2 presents parameters for charging and discharging mode of operation.

Table 2. List of parameters in charging and discharging

Parameter	Charging mode	Discharging mode
Total mass flow (kg/h)	600.0	250.0
Average velocity (m/s)	1.5	0.5
Air channel width (mm)	16.0	16.0
Convection Coefficient ($W/m^2 \cdot K$)	18.0	15.0
Re number	3050.2	1270.9

Figures 6 and 7 present the air outlet temperature, and PCM-HX charge/discharge heat flow in 24 hours period, respectively.

Figure 6. Air outlet temperature for charging/discharging PCM-HX

Figure 7. PCM-HX charge/discharge heat flow

It can be seen from Figures 6 and 7 charging time (90% of total) is around 4 h, while discharging (90% of total) time is around 6 h.

Previous work (Morovat 2019) showed that high flow rates provide a higher power output. However, low airflow rates discharge the PCM-HX more slowly. As a result, the average energy transfer over a 24 h period for the four airflow rates is approximately the same.

Control strategies

The appropriate application of control strategies in HVAC systems is a key factor to improve the energy efficiency of buildings (Tabares-Velasco, Christensen et al. 2012, Afroz, Shafiullah et al. 2017). The use of active thermal energy storage systems with PCM has not been widely adopted, mainly due to the lack of a proper integration into the building HVAC system. In this regard, proper integration, including suitable control strategies, has a significant impact on the performance of the system. In addition, if integrated with HVAC system design, the inclusion of the PCM-HX can be used to avoid the traditional oversizing of the HVAC equipment. Thus, both system operating and capital costs can be reduced in an effective manner.

The following sections outline the results obtained from free cooling application of the PCM-HX into an office zone under three different scenarios toward achieving peak load and energy consumption reduction.

- The effect of discharging PCM-HX stored energy in a zone with step setpoints.
- The effect of using a linear ramp of room temperature setpoint.
- The effect of discharging start time of the PCM-HX.

Design day conditions for control studies

Weather conditions similar to a summer day (Design Day) in Montreal are selected as the simulation scenario, since the demand load tends to peak under these conditions. Figure 8 presents the curves used to model outdoor temperature and solar flux on the south facade. In this simulation scenario, it is assumed that the outside temperature follows a perfect sinusoidal curve, fluctuating between 15 °C and 26 °C. This curve was obtained from a simulation carried out using EnergyPlus with a Montreal weather file (Crawley, Lawrie et al. 2000).

Figure 8. Outdoor temperature and solar flux

Numerical experiment

It was assumed that the PCM-HX was well insulated on all sides (i.e., adiabatic conditions). The initial conditions of the experiment were the following:

- The initial temperature of the room, at $t = 0$, was set at 26 °C everywhere.
- It was assumed that the PCM-HX was “fully charged”: the material is fully solid and at a uniform temperature of 17 °C.
- Air from the room is circulated through the PCM-HX whenever heat extraction is required.

The simulation runs for a period of 24 h using the weather conditions presented in Figure 7. The simulation time step was set to 60 seconds (it was previously determined that the critical time step for a finite difference formulation was 87s, at least). A heat pump with COP = 3 is considered as the auxiliary cooling system. Capacity of the cooling system is assumed 3500W. The proportional gain of the controller (k_p) is 4000 W/ °C and the integral gain of the controller (k_i) is 0.1 W/°C·s.

Effect of discharge heat flow rate of the PCM-HX

The setpoint temperature during daytime (between 6:00 to 18:00) is assumed 24 °C and during nighttime (between 18:00 to 6:00), the set point is 26 °C. The charging mode of the PCM-HX device is from 2:00 to 5:00 (nighttime) and discharging mode is from 6:00 to 15:00 (daytime).

In this section, two scenarios regarding effect of heat flow rate of the PCM-HX has been investigated:

- (1) Scenario #1: PCM-HX with laminar air stream through the air channels: airflow rate is 250 kg/h ($Re = 1270.9$, air velocity = 0.5, and $h_c = 10$)
- (2) Scenario #2: PCM-HX with turbulent air stream through the air channels: airflow rate is 600 kg/h ($Re = 3050.2$, air velocity = 1.5, and $h_c = 18$)

In this study, the device charges when the outdoor temperature is around 15 °C, for better freezing of the PCM. Figure 9 presents a comparison of the results for the electric load, and Figure 10 shows temperature setpoint and room air temperature with and without PCM-HX. In the electric load, fan energy is considered. Based on equations 8 to 11, the pressure drop is 16 pa, and the energy used by the circulation fan is about 0.07 kWh per day, which is negligible compared to heat transferred to the PCM-HX (4.4 kWh per day).

Figure 9. Auxiliary cooling power – different airflow through PCM-HX

Figure 10. Temperature setpoint and room air temperature

The results show that the free cooling with PCM-HX could reduce energy consumption by 14% from 7.6 kWh to 6.55 kWh. In peak load perspective, higher the airflow rate, leads to faster discharge of the PCM-HX, without significant effect on peak load during daytime (around 1%). On the other hand, lower airflow rate causes smooth reduction in energy consumption during all hours of the day, and peak load reduction as well. In this case, peak load during hot daytime (2:30 P.M) reduces by 4%.

Effect of discharging start times of the PCM-HX

It is well known that a sudden setpoint transition –e.g. between a night setback and daytime setpoint– creates a spike in the peak demand. The effect of using ramps to pass from one set-point value to another has been investigated by (Braun and Lee 2006, Lee and Braun 2008, Candanedo, Dehkordi et al. 2015), among others. Therefore, using linear ramp of room temperature set point could be an effective solution to reduce or shift the peak load. In this section, three more scenarios has been investigated:

- (1) Scenario #3: Two-hours linear ramp of temperature setpoint (from 4:00 to 6:00, and 18:00 to 20:00)
- (2) Scenario #4: Discharging PCM-HX from 09:00 A.M.
- (3) Scenario #5: Discharging PCM-HX from 13:00 P.M.

Figure 11 provides a comparison between results for different discharging start times of the PCM-HX.

Figure 11. Auxiliary cooling power – different discharging start times of PCM-HX

It was found that, using a two-hour room temperature set point linear ramp, could shift peak load in an effective manner. Results show, for the case with discharging start time from 09:00, cooling power fluctuation increases by 44 % compared to the reference case; while in the case with discharging start time from 13:00 cooling power fluctuation reduces by 24 % compared to the reference case.

Energy flexibility

Energy flexibility in buildings has been defined as “the possibility to deviate the electricity consumption of a building from the reference scenario at a specific point in time and during a certain time span” (Torres Ruilova 2017). In this section, a flexibility factor has been calculated based on Equation (9).

$$Flexibility\ factor = \frac{\int_{REF} q_{cooling} dt - \int_{ADR} q_{cooling} dt}{\int_{REF} q_{cooling} dt + \int_{ADR} q_{cooling} dt} \quad (9)$$

where:

- Flexibility factor = 0: the cooling use is similar in reference case and active demand response (ADR) case.
- Flexibility factor = 1: no cooling is used in ADR case.
- Flexibility factor = -1: no cooling is used in ref. case.

Table 3 provides a comparison between reference case study (without PCM-HX) and case #5 (with PCM-HX and two-hour linear ramp).

Table 3. Flexibility scenarios of the building

Case study	Ref. case	Case #5	Changes Relative to Ref. case
Mean power load during daytime (W)	629.17	370.90	41 % reduction
Cooling power fluctuation (W) – (7 a.m. to 5 p.m.)	410.22	312.15	24 % reduction
Energy consumption (kWh)	7.6	6.55	14 % reduction
Flexibility factor (–)	-	0.10	10 % increase

The results presented in Table 3 shows that such a PCM-HX along with appropriate control strategies could provide flexibility to the grid, reduce mean power load during daytime and peak power demand, simultaneously.

The simulation studies in this paper shows that airflow rate through air channels and discharging start times of the PCM-HX are two important factors toward achieving best performance of free cooling with PCM-HX. In a future work, the performance of the PCM-HX will be optimized for free-cooling applications by applying a model predictive control (MPC) method to determine optimal flow rates and discharging start times.

Conclusion

In this study, the integration of the proposed PCM-HX for free cooling in an office zone is evaluated along with the effect of enhanced control strategies. The potential of using PCM-HX with different number of air channels together with linear ramps is investigated. The main results are:

- The PCM-TES is charged (90% of its capacity) in 4 h and discharged (90% of its capacity) in 6 h. Having multiple channels, different flow rates and different configurations, these discharging times can change to better match to load profile.
- The combination of the two-hours linear ramp of temperature setpoint and PCM-HX can reduce energy consumption by 14% and achieve 4% peak load reduction (due to thermal load in daytime);
- Appropriate air stream trough air channels and start discharge time of the PCM-HX are two important factors toward achieving optimum performance of free cooling with PCM-HX. Correct implementation of these factors can reduce electric load fluctuation by 24%;
- Using PCM-HX for free cooling along with two hour ramp increase energy flexibility of the building by 10%.

Results shows that the use of active PCM-HX for free cooling applications along with appropriate control strategies could be an effective solution to have better energy flexibility in buildings, reduce the sizing of the HVAC unit at the design stage, and reduce energy bills, thus reducing both operation and initial capital costs.

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