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Research Article

Incorporation of Numerical Analysis for Improving the Performance of a Condenser Unit in an Air Condition by Employment of Phase Change Material as a Latent Heat Storage Medium

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Abstract

In this study, mathematical models based on equations for air conditioning condenser units will be improved and numerically solved using phase change materials for heat dissipation capacity. The acquired findings are utilised to analyse the thermal performance of both charging and discharging processes. The effects of the temperature of the inlet heat transfer fluid, mass flow rate and phase change temperature range on the thermal performance have been studied. As a result of the findings, the phase change temperature range of the PCM must be correctly recognised and taken into account while modelling the system's performance. The current study suggests an unique hybrid space-conditioning system that combines PCMs and employs surfaces as sources of heating or cooling to improve temperature distribution and comfort while using less energy. The concept is to employ PCM to deliver cool air to discharge or dissipate heat, hence overcoming the majority of the drawbacks of both techniques. However, the PCM simulation results demonstrate that adding thermal inertia to the system improves heat transfer from the condenser and allows for a greater condensing temperature, increasing the system's energy efficiency. The energy stored in the PCM is transferred to the air conditioning cell for the primary goal of improving performance, cooling time period, storage capacity, and maintaining a steady cooling effect for longer periods of time during power outages when phase change material is used.

Keywords- Phase change Material (PCM), Energy storage, Heat Transfer.

Nomenclatures

A	area of material [m^2]
C	specific heat capacity of material (kJ/kg °C)
C_p	specific heat at constant pressure [$Jkg^{-1} K^{-1}$]
C_e	mass flow factor
D, d	diameter [m]
F	liquid fraction
G	gravity [$m\ s^{-2}$]

h specific enthalpy [kJ kg^{-1}]
H volumetric enthalpy [kJ m^{-3}]
K thermal conductivity [W $m^{-1}K^{-1}$]
l tube length [m]
L thickness of material (m)
$\dot{m}$ mass flow rate [kg s^{-1}]
m mass of material (kg)
N wetted perimeter [m]
Nu Nusselt number
P pressure [kPa]
Pr Prandtl number
q heat energy [kWh]
Q quantity of energy input/output [kW]
R radius [m]
Ra Rayleigh number
Re Reynolds number
S stroke of piston [m]
t time [s]
T temperature [C°]
U total heat transfer coefficient [W $m^{-2}K^{-1}$]
V volume displacement of the compressor [m^3]
w width [m]
W compressor power [W]

Greek Symbols

α thermal diffusivity [m^2 s^{-1}]
β expansion coefficient [K^{-1}]
δ melt layer thickness [m]
θ the difference of temperature: $T\text{-}T_m$ [K]
η_{is} isentropic efficiency
η_v volumetric efficiency
μ dynamic viscosity [Pa s^{-1}]
ρ density of material [m^3 kg^{-1}]
ϑ kinematic viscosity [m^2 s^{-1}]
ΔH latent heat of fusion [J kg^{-1}]
Δx axial space step [m]
Δr radial space step [m]
Δt time step [s]
ΔT temperature difference (°C)

Subscripts

is Isentropic
Sol Solid
l Liquid
g Gas
M Melt
sh Superheat
tp two-phase

sc sub-cooled
f fluid in the tube (refrigerant)
Air ambient air
i Inner
o Outer
e expansion valve
ev Evaporator
con Condenser
C Compressor
Eff Efficient
PCM the phase change material
COP the coefficient of performance
Inl Inlet
R Refrigerator
w,e, n, s west east, north and south faces of control volumes
1 inlet to compressor or outlet of the evaporator
2 the outlet of compressor or inlet to the condenser
3 the outlet of condenser or inlet to the expansion valve
4 the outlet of expansion valve or inlet to the evaporator

1. Introduction

1.1 Importance of Phase Change Material

New renewable energy sources, such as refrigeration and air conditioning, solar, marine and wind energy, are inherently stochastic and intermittent, necessitating the development of energy storage devices. Thermal energy storage via sensible and latent heat has garnered a lot of press in recent years, and it now has a variety of practical applications. Conversion of solid–liquid phase-change materials (PCMs) for melting latent heat is a successful technology that has gotten a lot of interest because of its high heat storage density and low temperature fluctuation behaviour compared to other visible solutions.

1.2 PCM as a tool for solving energy crises

Refrigeration and air conditioning systems are directly or indirectly contributing to the current energy crisis, as their use in the home, business, and transportation sectors is fast expanding. Nowadays, power outages are frequently caused by accidents, or by the implementation of demand side management schemes (DSM) by the electricity supplier to shift power usage to avoid high loads, or by the user to shift their electricity usage to off-peak pricing periods (electrical load shifting), and it is critical to maintain consistent temperatures inside cold storage facilities and cold transport vehicles [3]. The majority of frozen and refrigerated foods are temperature sensitive. Heat infiltrating the walls contributes significantly to the heat loadings in a cold store. The refrigerated system removes the heat burden, however the stored product is not cooled if there is a power outage [1]. Phase change materials (PCM) will be used in thermal energy storage systems (TES) to store heat and cold at a shifted time. While in the transition state, phase change material (PCM) melts within a restricted temperature range and absorbs a considerable amount of energy, reducing the temperature rise in the environment. In the event of a power outage, PCM with an appropriate melting temperature can be employed to supply thermal capacity and maintain a safe internal temperature. [2] PCM can also be used in load shedding to move electricity usage to a more efficient time.

1.3 Commendation of PCM Application

In recent years, many Cold Thermal Energy Storage (CTES) devices have gotten a lot of attention. Cold storage is a unique type of space in which the temperature is kept extremely low using equipment and accurate sensors. Because of its unique geographical location and diverse soil, India produces a vast assortment of fruits and vegetables, including apples, grapes, oranges, potatoes, chillies, ginger, and other spices. Because of the huge coastline areas, marine items are also produced in great amounts. [4]. Fruit and vegetable output is currently over 100 million MT, and given the rate of population increase and demand, the production of risible commodities is increasing every year. For perishable goods, cold storage facilities are the most important infrastructure component.

Aside from stabilizing market pricing and equitably distributing goods based on demand and timing, other advantages and benefits are provided by the cold storage sector to both farmers and consumers. Although, if we focus on household electric tariffs, we can see that refrigerators and air conditioners take a significant portion of it. According to recent study, PCM can be utilized to reduce power consumption.

1.4 Refrigeration cycle Vapour Compression Refrigeration System (VCRS)

A circulating liquid refrigerant is used as the medium in vapour compression, which collects and removes heat from the place to be cooled before rejecting it elsewhere. A typical single-stage vapour compression system is shown in Figure 1. A compressor, a condenser, a thermal expansion valve (also known as a throttle valve), and an evaporator are all part of such a system. Circulating refrigerant enters the compressor as a saturated vapour and is compressed to a higher pressure, resulting in a higher temperature.

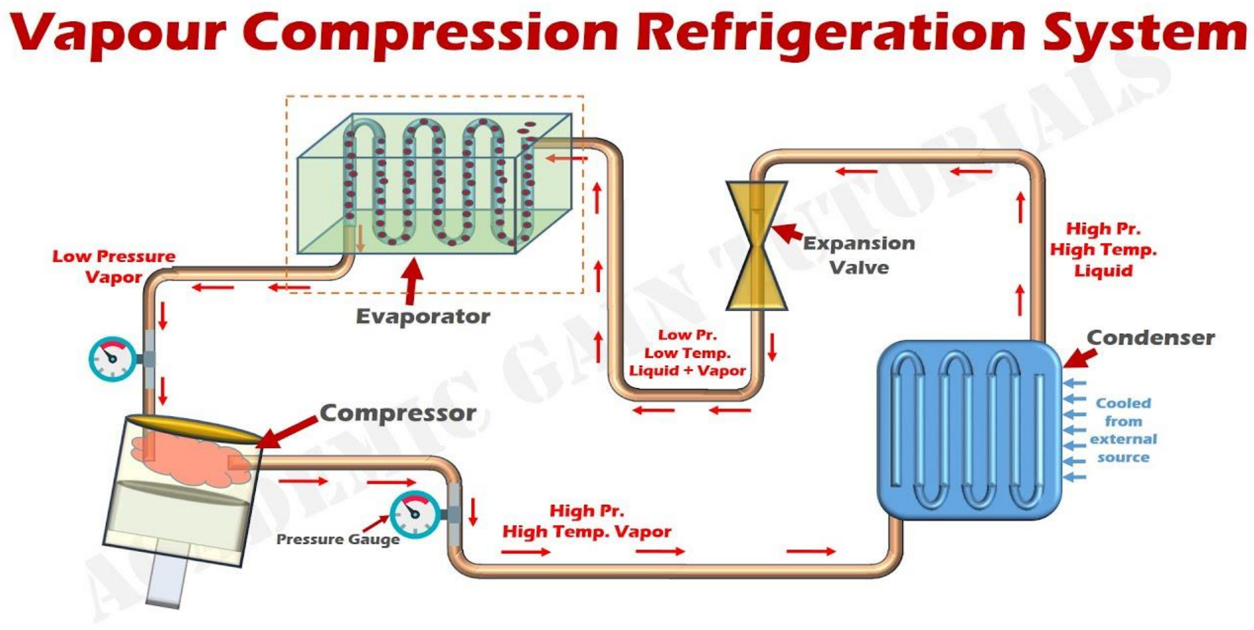

Figure 1. Schematic Diagram of a VCRS.

The hot, compressed vapour is now a superheated vapour, and it is at a temperature and pressure where it can be condensed with either cooling water or cooling air. The hot vapour is directed to a condenser, where it is cooled and condensed into a liquid by passing through a coil or tubes with cool water or cool air flowing over them. This is when the circulating refrigerant rejects heat from the system, which is then transported away by either water or air (whichever may be

the case). The condensed liquid refrigerant, which is in the thermodynamic state of a saturated liquid, is then passed via an expansion valve, where the pressure is abruptly reduced.
The adiabatic flash evaporation of a portion of the liquid refrigerant occurs as a result of the pressure drop. The adiabatic flash evaporation's auto-refrigeration effect decreases the temperature of the liquid and vapour refrigerant combination to below the temperature of the enclosed space to be refrigerated. The cold mixture is subsequently passed through the evaporator's coils or tubes. Warm air in the enclosed space is circulated by a fan through the coil or tubes carrying the cold refrigerant liquid and vapour mixture. The liquid fraction of the cold refrigerant mixture evaporates as a result of the heated air. The circulating air is cooled at the same time, lowering the temperature of the enclosed space to the appropriate level.
The circulating refrigerant absorbs and eliminates heat in the evaporator, which is then rejected in the condenser and moved elsewhere by the water or air that is employed in the condenser. The refrigerant vapour from the evaporator is directed back into the compressor to finish the refrigeration cycle as a saturated vapour.

1.5 Classification of Phase Change Material (PCM)

A phase change material (PCM) is a substance that releases/absorbs enough energy to generate useful heat/cooling at phase transition. In most cases, the transition will be between one of the first two fundamental states of matter, solid and liquid. The phase transition may also occur between non-classical states of matter, such as crystal conformance, in which the material transitions from one crystalline structure to another, which may have a higher or lower energy state.
In comparison to sensible heat storage, a PCM may store and release huge amounts of energy by melting and solidifying at the phase change temperature (PCT). When a material transitions from solid to liquid or vice versa, or when the internal structure of the material changes, heat is absorbed or released; PCMs are hence referred to as latent heat storage (LHS) materials.
Organic (carbon-containing) compounds originating from petroleum, plants, or animals, and salt hydrates, which often utilise natural salts from the sea or mineral deposits or are by-products of other operations, are the two main types of phase transition material. Solid to solid phase change is a third type. Figure 2 depicts more investigation on PCM.

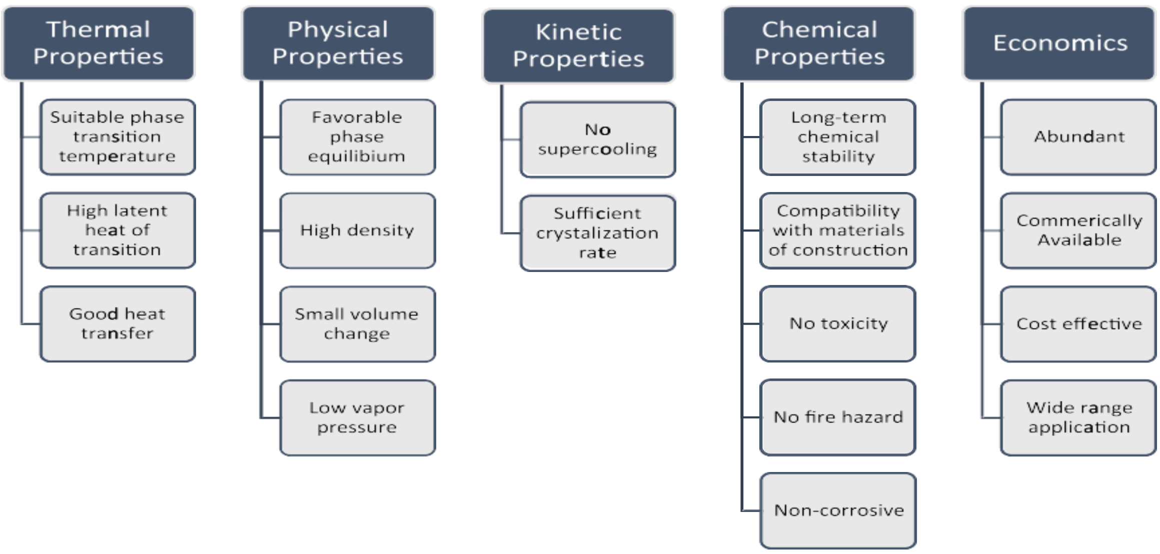

Figure 2. Characteristics properties of a latent heat storage material (PCM).

2. Literature Survey

2.1 Introductory of Survey

Because of the extreme summer temperatures in many northern Indian states, where the outdoor ambient temperature can exceed 45°C during the summer, air conditioning consumes 75 percent of total power plants during the summer (AC). The performance of the air conditioner is directly proportional to the temperature outside. The performance of the condensers degrades as the ambient temperature rises, lowering air conditioner efficiency. When the ambient temperature exceeds 35°C, AC performance might be reduced by 20%, necessitating an increase in power to provide the same amount of cooling to an area. Another disadvantage is that the compressor's life expectancy is reduced. As a result, the possibility of lowering peak-period electrical demand by modifying HVAC controls has been confirmed [5]. Pre-cooling of the refrigerant entering the condensers could be a feasible solution for better AC system performance during the hot summer months while also saving energy [6].

In the literature, various types of pre-cooling procedures have been proposed. In Alabama, a three-ton split AC unit was used to test evaporative cooling by spraying water straight onto the condenser. A maximum improvement of 19% and 9% in energy efficiency and electrical conservation, respectively, has been obtained [7].

Mineral deposits and scaling on the condenser coils, on the other hand, have been cited as major issues. To prevent such issues, a 2.5-ton household AC system with an indirect evaporative pre-cooling approach employing a media pad evaporative cooler has been developed. The technology has improved its energy efficiency by up to 22 percent [8]. On a 7.5-ton DX unit, more research on evaporative cooling utilising a media pad evaporative cooler was done, controlling the water flow as a function of the ambient dry bulb temperature. A yearly water consumption rate of 99 000 gal/yr was used, resulting in a reduction in energy consumption from 14 334 to 2930 kW h/yr [9]. Using an evaporative pre-cooling condenser system and an average ambient temperature of 43.8°C, an experimental investigation on a proposed system of package unit and two air cooled chillers was conducted. For the packaged unit and the air-cooled chiller, effective power savings of 1.42kW and up to 4.6kW were realised, respectively [10]. Although, evaporative cooling (EC) is a cost-effective and environmentally beneficial technique that employs water evaporation to absorb heat from the environment and create cooling air. Because of its modest cooling capacity, EC has a limited application in hot and humid climates [11]. The effect of various parameters such as building location, mass level, and time-of-use utility rate on the potential of pre-cooling techniques on reducing energy cost savings strategies has been studied to develop general guidelines for operators and designers to assess the potential of pre-cooling techniques on reducing energy cost savings strategies [12].

2.2 The Use of Phase Change Material (PCM)

Scientists are primarily interested in new energy uses, such as thermal energy storage (TES). PCM (phase change material) has been widely used in a variety of industries as a typical ecologically acceptable energy-saving material [13–14]. Thermally activated building systems, as well as Refrigeration and Air Conditions for temperature control (TABS) [15], suspended ceilings [16], external facades [17], and the ventilation system (AC system) [18], have all used PCM-based-TES for cooling.

2.3 PCM stores Sensible as well as latent heat

Due to its high energy density, thermostatic effect, and low cost as compared to traditional cooling approaches, PCM is known as an effective technology that retains larger amounts of thermal energy per unit mass. The latent heating/cooling of the environment is provided by

PCM's absorption/release of heat during its physical phase transition [19-20]. The use of PCM storage systems in air conditioning, heating, and ventilation systems to store thermal energy from the evaporator or condenser can improve interior thermal comfort and improve building energy efficiency. Paraffin waxes (n-alkanes) are the most widely utilised and recognised PCMs [21-22].

2.4 PCM for Refrigeration and Air Conditions for temperature control

AC systems paired with PCM-TES have been shown to improve operating performance [23]. During the summer in Tunisia, a numerical research on a solar-driven AC system with integrated PCM cold storage was conducted using the Transient System Simulation programme (TRNSYS). Solar loop, ejector cycle, PCM cold storage, and air conditioned space make up the proposed system. The non-cold system was found to surpass the comfort temperature more than 26% of the time, whereas the ideal storage volume of 1000 L PCM-cold storage resulted in great indoor comfort 95% of the time with a room temperature below 26°C [24].

2.5 Numerical Studies on PCM

Another numerical analysis using PCM in a desiccant AC found average increases in electrical energy savings of up to 55 percent [25]. The charging and discharging temperatures of the PCM, as well as the air outlet temperature of an air-PCM unit, have been calculated using a numerical model. The thermal performance enhancement caused by different unit sizes and charging/discharging air mass flow rates has been documented, with the best design of RT20 and RT25-based PCM being 0.03* 1.5 m2 unit with 0.25 kg/s air mass flow rate [26]. The influence of multi-PCMs with low PCM melting points of 5.3 °C, 6.5 °C, and 10 °C has been investigated numerically in traditional AC systems using a 3-D ANSYS FLUENT model to explore the effect of multi-PCMs with low PCM melting points of 5.3 °C, 6.5 °C, and 10 °C, The effect of fluid inlet temperature and flow rate on the TES unit's charging process. According to the simulation results, the total charging capacity of the multi-PCM TES unit increased by 32.22 percent when compared to the single-PCM unit. For a flow rate of 0.3 kg/s, lowering the input temperature increased charging capacity and significantly reduced the TES unit's overall charging time. The entire charging capacity, however, was unaffected by the HTF inlet temperature [27]. The goal of this study is to investigate the pre-cooling based PCM of air delivered to split AC units in the UAE during hot weather. Due to high outdoor ambient temperatures of up to 55°C during the summer daytime, power plants operate under peak electricity demand conditions. As a result, the development of more power plants will be required. In an attempt to lower peak demand, a PCM-based air pre-cooling approach is suggested, which absorbs ambient cooling during the night and delivers it to the fresh air supply stream during the day.

3. Theory Methodology

3.1 Numerical Modeling

Grald and MacArthur [33,34], He et al. [35,36], and Willatzen et al. [37] have also done a lot of work on refrigeration system simulation. The following are the basic assumptions of the transient model based on these studies:

i. In heat exchangers, liquid and vapour refrigerants are in thermal equilibrium.
ii. Pressure wave dynamics have negligible effects.
Expansion is isenthalpic in nature.
iv. There is no adiabatic compression.
v. Metallic elements in the system have low thermal resistances in contrast to other thermal resistances, but their capacitance is significant.

This section describes the overall system and makes assumptions for mathematical modelling for each component independently. The compressor, evaporator, expansion valve, PCM heat exchanger, and condenser are the five components of the system. The PCM heat exchanger under investigation is placed between the compressor and the condenser. Figure 1 shows a schematic view of the simulation system.

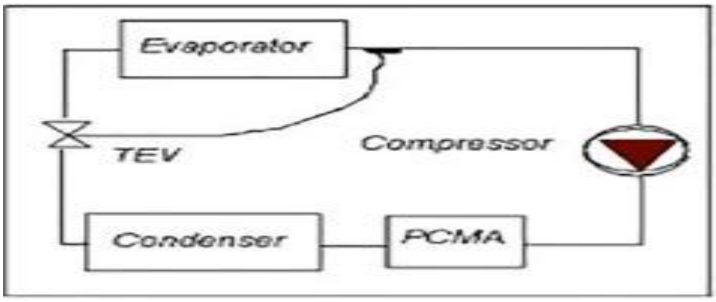

Figure 3. Schematic view of system compartments.

Equation 1 can be used to calculate the amount of energy (cooling or heating) that the system produced when compared to a room with a conventional thermocouple using a mathematical model.

The amount of energy required to create a temperature change.

$$Q = mc\Delta T \qquad (1)$$

Mass of the material can be found by

$$m = \rho V \qquad (2)$$

Volume of the material can be found by

$$V = AL \qquad (3)$$

3.2 Compressor modeling

Compressors draw gases into themselves quickly and compress them by expending mechanical energy. The pressure and temperature of the gas rise as a result of this process.

At high pressure, the vapour refrigerant transforms from a saturated condition to a superheated state.

For modelling the compressor, the following assumptions are taken into account [28]:

- Both the volumetric and isentropic efficiencies are maintained.
- Heat dissipation in the compressor shell and the refrigerant charge are regarded as insignificant.
- An adiabatic compression process is expected.

As a result, bellows [28] can be used to determine isentropic efficiency:

$$\eta_{\text{isen}} = \frac{h_{2,\text{ is}} - h_1}{h_2 - h_1} \qquad (4)$$

[28] determines the refrigerant mass flow rate:

$$\dot{m}_R = \rho V_R \eta_v \times \frac{RPM}{60} \tag{5}$$

where v is the compressor's volumetric efficiency and com- pressor displacement is defined by [28]:

$$\eta_v = 0.851 - 0.0241\frac{p_2}{p_1} \tag{6}$$

$$V = \pi\frac{D^2}{4} \times S \tag{7}$$

The following equation [28] is used to compute compressor power:

$$\dot{w}_c = \dot{m}_R(h_1 - h_2) \tag{8}$$

3.3 Condenser modeling

The condenser receives high-pressure, high-temperature refrigerant vapour, which subsequently condenses. Figure 2 depicts the situation.

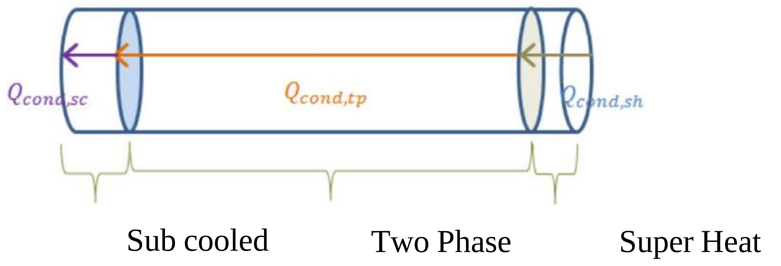

Figure 4. Different regions in the condenser.

The condenser is separated into three zones: a superheated zone, a two-phase zone, and a sub-cooled zone.

Equations must be extracted individually in one phase and two phase states for modelling condenser heat transfer [28]. The condenser is modelled using the following consumptions:

• It's a one-dimensional flow.

• Refrigerant pressure drop is ignored, as is heat transmission between refrigerant and pipes in the axial orientation.

Thermodynamic heat transfer in the superheated zone

The condenser's incoming refrigerant is in a superheated state. This area's heat transmission is computed as follows [28]:

$$\dot{Q}_{\text{cond,sh}} = \dot{m}\left(h_2 - h_{2,g}\right) = U_{sh}A_{sh}(T_{sh} - T_{\text{air}}) \tag{9}$$

where A_{sh} is superheat area, T_{sh} is average temperature in superheat area and U_{sh} is total heat transfer coefficient and is calculated as [29]:

$$U_{sh} = \frac{1}{A_{sh}} \left(\frac{1}{\frac{1}{A_i u_i} + \frac{\ln\left(\frac{R_e}{R_i}\right)}{2\pi k L} + \frac{1}{A_0 u_0}} \right) \tag{10}$$

where u_i is heat transfer coefficient in refrigerant tubes in which flow is superheat and in one phase. Also u_o is heat transfer coefficient of air passing over pipes which is obtained from [30]; [31]; [28].

3.4 Heat transfer in two-phase region

Heat transfer in the two-phase region is equal to the heat gained by reaching refrigerant from superheat to saturated liquid at condenser pressure and is calculated from equation 8 [28]:

$$\dot{Q}_{\text{cond,tp}} = \dot{m}\left(h_{2,g} - h_{2,l}\right) \tag{11}$$

Subsequent to calculating $\dot{Q}_{cond,tp}$, the length of superheated area can be calculated as [28]:

$$\dot{Q}_{\text{cond,tp}} = U_{\text{tp}} A_{\text{tp}} \left(T_{\text{sat}} - T_{\text{air}}\right) \tag{12}$$

$$A_{tp} = \pi d_i L_{tp} \tag{13}$$

in which A_{tp} is two phase regions area and U_{tp} is two phase regions heat transfer coefficient and calculated as follows [29]:

$$U_{tp} = \frac{1}{A_{s,tp}} \left(\frac{1}{\frac{1}{A_i u_i} + \frac{\ln\left(\frac{R_e}{R_i}\right)}{2\pi k L} + \frac{1}{A_0 u_0}} \right) \tag{14}$$

3.5 Heat transfer in subcooled region

Heat transfer value and condenser outlet temperature, T_3, are Calculated as below [28]:

$$\dot{Q}_{\text{cond,sc}} = U_{\text{sc}} A_{\text{sc}} \left(T_{\text{sc}} - T_{\text{air}}\right) = \dot{m} C_p \left(T_{\text{sat}} - T_{\text{air}}\right) \tag{15}$$

$$A_{sc} = \pi d_i L_{sc} \tag{16}$$

$$L_{\text{sc}} = L - \left(L_{\text{sh}} + L_{\text{tp}}\right) \tag{17}$$

$$\mathrm{T}_{\mathrm{sc}} = \frac{\mathrm{T}_{\mathrm{sat}} + \mathrm{T}_3}{2} \tag{18}$$

Thus heat transfer in condenser part is the sum of heat transfer in three regions [28]:

$$\dot{Q}_{\text{cond}} = \dot{Q}_{\text{cond,sh}} + \dot{Q}_{\text{cond,tp}} + \dot{Q}_{\text{cond,sc}} \tag{19}$$

3.6 Expansion valve modelling

The expansion valve is a mechanical device that regulates the volume of fluid that enters the evaporator.

A fixed orifice, capillary tube, and thermostatic expansion valve can all be used

to model it.
The expansion valve is modelled as a fixed orifice in this article.
As a result, the following assumptions are taken into account for this model [28]:

• In a one-dimensional flow, gravity force and refrigerant charge are considered negligible and the flow coefficient is fixed.
• The expansion process is described by constant enthalpy.
mass flow rate in the orifice is [28]:

$$\dot{m}_e = C_e d_e^2 \sqrt{\rho_{\text{in}} (p_2 - p_1)} \quad (20)$$

where C_e is the mass flow factor which depends on geometry of orifice tube and d_i is inner tube diameter.
As it is assumed, enthalpy is constant, thus [28]:

$$h_3 = h_4 \quad (21)$$

3.7 Evaporator modeling

The evaporator is a device that uses heat to transform fluid from a liquid to a gas.
The following assumptions are taken into account when modelling an evaporator:
• Flow is a one-dimensional phenomenon.
• Axial heat transfer between the refrigerant and the tube is not taken into account.

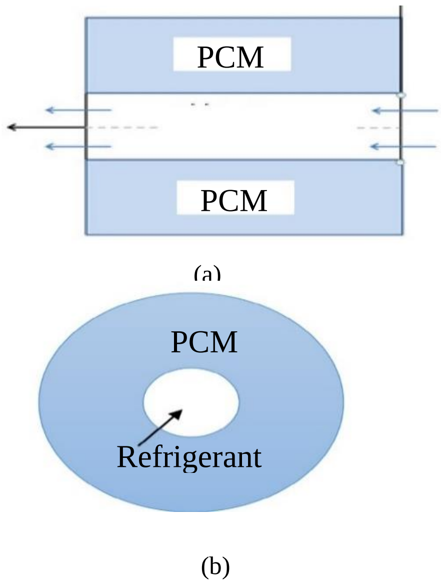

Figure 5. Schematic view of the PCM heat exchanger (a) axial (r,x) view and (b) radial (r,θ) view.

• There is no pressure decrease along the evaporator.
• The evaporator's superheating refrigerant is ignored, and heat transfer includes two phases and is calculated as [28]:

$$\dot{Q}_{ev} = \dot{m}(h_1 - h_4) \quad (22)$$

3.8 Coefficient of performance

The COP is calculated from equation (23) [28]:

$$\mathrm{COP} = \frac{\dot{Q}_{ev}}{\dot{W}} = \frac{\dot{Q}_{ev}}{Q_{cond} - \dot{Q}_{ev}} \tag{23}$$

3.9 Geometry of PCM heat exchanger

Figure 3 shows a schematic depiction of the PCM heat exchanger, which includes a heat storage unit. The inner and outer tubes are 5 and 20 mm in diameter, respectively. The main refrigerant flows through the inner tube after PCM is filled in an annular gap. In PCM, heat is transferred by both conduction and convection methods. This geometry was chosen for greater heat transfer as well as ease of application. Figure 2 depicts the refrigerant's temperature variation trend at three different axial points. It should be emphasized that all outcomes and comparisons for all cases are during the first hour of the process. The refrigerant temperature rises rapidly with the steep slope in the curve due to the large temperature difference between the refrigerant and the PCM. As time passes and the PCM temperature falls, the thermal potential of the refrigerant falls, resulting in a smaller temperature difference between the two mediums.

As a result, at the end of the process, the temperature differential between the refrigerant and the PCM is nearly constant, and the temperature of the refrigerant is nearly constant.

In addition, the refrigerant temperature at the exit is lower than at the intake due to heat transfer between the refrigerant and the PCM at the early stages of the PCM heat exchanger. Building a system transient model entails first creating transient models of separate components, which are then combined to simulate the entire system. Each model's specifics are listed below.

3.10 Mathematical description of PCM model in the form of flowchart

In order to mathematically describe the PCM model, a number of assumptions have been made. These are the ones.

(1) In the PCM, axial conduction is low, and the axial temperature gradient is tiny in comparison to the radial temperature gradient.

(2) The Stefan number Ste 1, which indicates that the sensible thermal storage capacity is negligible in comparison to the latent thermal storage capacity and may thus be ignored.

(3) The heat transfer fluid's (refrigerant) capacitance is modest and can be ignored.

(4) In the liquid PCM, natural convective heat transfer is assumed to be part of the equivalent conductive heat transfer, which, according to the test data, enhances thermal conductivity by 50%.

(5) There is no consideration for heat storage in the metal tubes.

The energy balance equation for the PCM is expressed as follows using the aforementioned assumption:

Figure 4 depicts the variation in refrigerant temperature along the axis over time. It can be observed with the help of mathematical model tool.

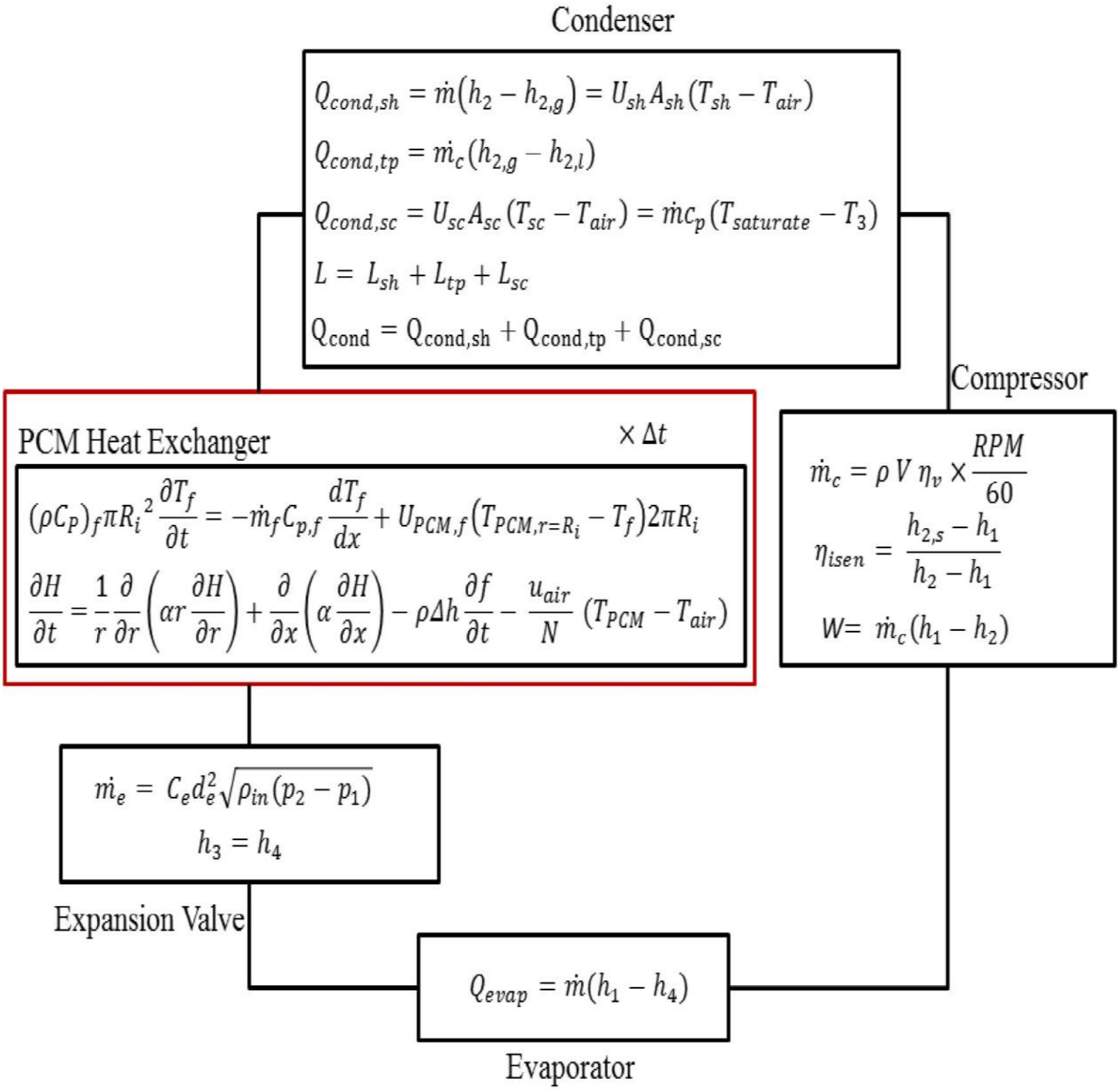

Figure 6. Overview of relations in refrigeration cycle utilising PCM [32].

Because more heat is transferred from the refrigerant to the PCM, the temperature drop at the heat exchanger's outlet is greater than at the heat exchanger's middle. As a result, the refrigerant exits the heat exchanger at a lower temperature than it entered, which is beneficial throughout the cycle.

The COP of the system employing PCM vs the COP of the system without PCM is obviously high and low, respectively, indicating a slight increase in percentage by installing a PCM heat exchanger in the system. In general, utilising a PCM heat exchanger lowers the condenser outlet temperature, resulting in a higher COP.

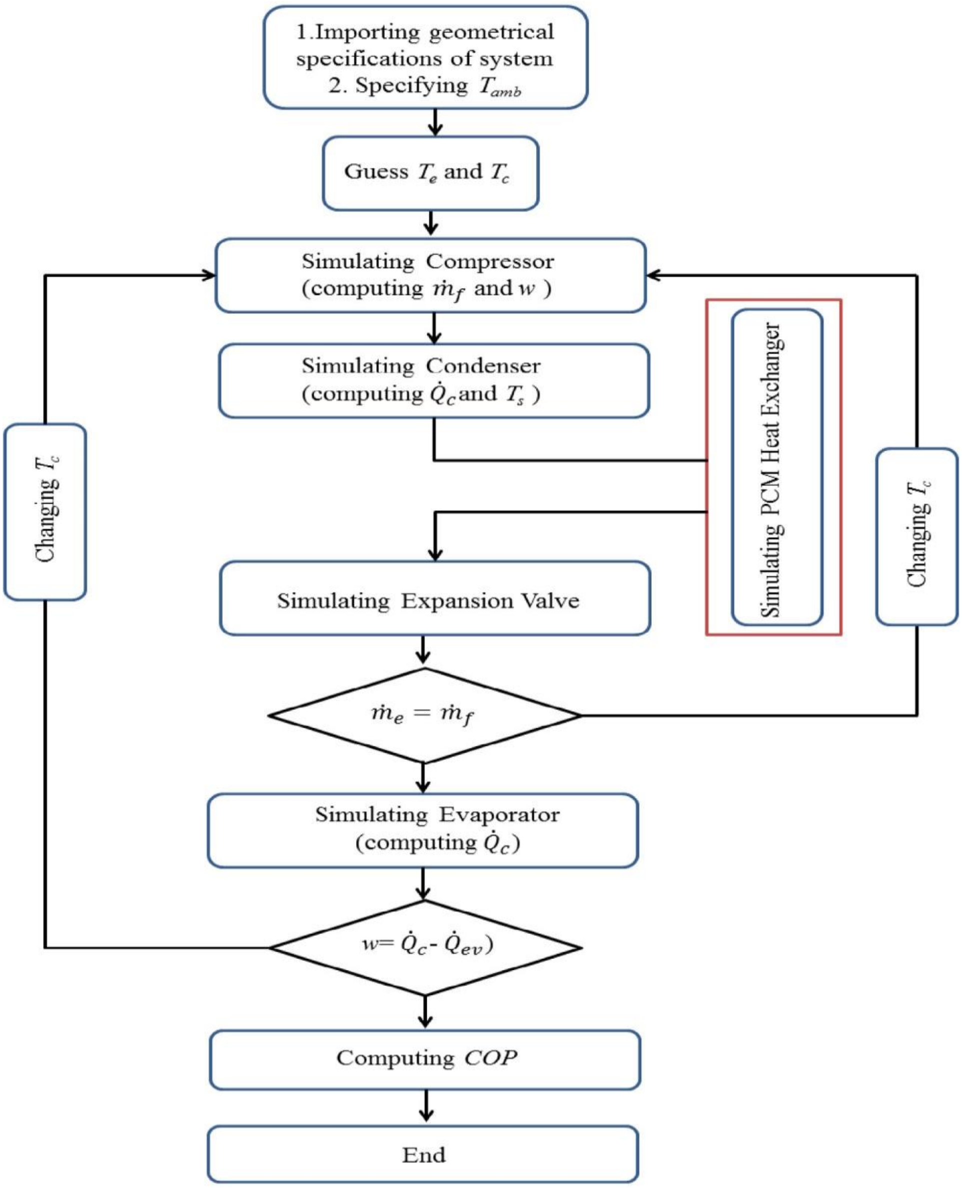

Figure 7. Simulation algorithm in refrigeration cycle utilising PCM heat exchanger.

Figure 8 shows temperature variation of refrigerant via time for different tube lengths It may be deduced that extending the tube length lowers the refrigerant temperature. Also, because of higher heat transfer between the PCM and the refrigerant, the refrigerant temperature drops more at the outlet than in the middle of the tube.

Because the refrigerant temperature remains constant for half an hour and is 37 °C, PCM lowers the temperature of the refrigerant for a short time before melting completely, keeping the refrigerant at 37 °C. Temperature drops 1 °C for every meter added to the length. All of these figures are based on numerical simulation.

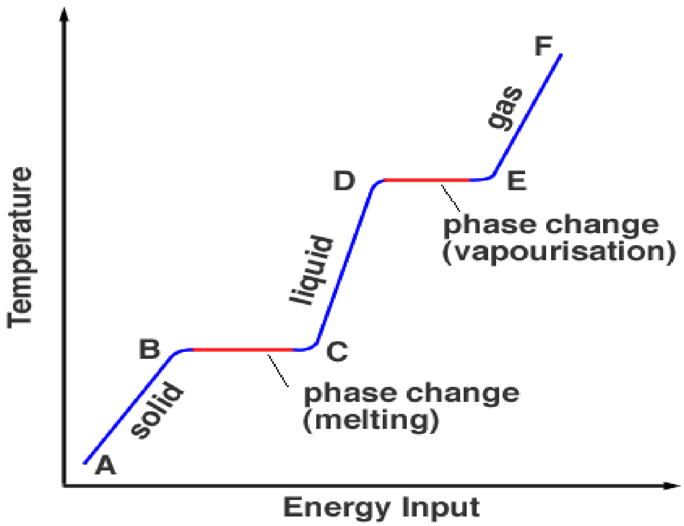

Figure 8. Graph of sensible heat and latent heat of PCM.

The refrigerant temperature results for varied PCM heat exchanger lengths at different times are presented with curve slope for better comparison. Temperature fluctuation by location at a certain time is shown by the slope of the curve. The values of curve slope in Figure 8 show that refrigerant temperature fluctuation by site is considerable at the start of the operation and diminishes as time passes. It can be argued that as the length of the heat exchanger rises, so does the heat transfer surface, resulting in increased heat transfer between the refrigerant and the PCM, lowering the refrigerant temperature.

4. Results & Discussions

4.1 Effect of Different PCMs on Compressor On-Off Cycling.

The thermostat, which is placed between the compressor and condenser compartment, activates the compressor on-off mode. To keep the evaporator cabinet temperature around -5^{0}C, the thermostat setting point has been modified such that it begins the compressor at - 3.9^{0}C and stops at -5^{0}C. When PCM is utilised, it absorbs too much heat because of its phase shift nature (solid to liquid), which prevents the compartment temperature from rising quickly, resulting in a longer compressor off cycle. During the whole melting phase of PCM, a low enough temperature near the target point inside the cabinet is maintained, and the compressor is not triggered into on mode soon by the thermostat. As a result, the compressor's off mode is extended. For a limited time, this longer off mode of the compressor reduces the number of on-off cycles dramatically.

4.2 Effect of PCM on Temperature Fluctuation inside the Cabin of Air Condition Condenser Coil.

At no load, the influence of PCM on the average air temperature inside the cabin. The average temperature fluctuation for a certain time is greatly reduced for the system with PCM compared to the system without PCM, according to the Mathematical Solution. The numerical solution clearly shows that power consumption is caused by differences in phase change temperature and latent heat of fusion.

4.3 Effect of PCM on Compressor Running Time at Different Thermal Load.

At various thermal loads, the percentage of average compressor running duration each cycle. The important findings are as follows:

• Using PCM, the average compressor running time per cycle was greatly lowered, resulting in lower energy usage.

• When compared to running without PCM, average compressor running time each cycle is reduced by around 5-30%, depending on the thermal load and the type of PCM used.

Because of their greater melting point of around (-100°C), which enhanced higher heat transfer rate and finally lengthened the off cycle of the compressor as compared to the on cycle, the reduction of compressor running time for the better values as compared to other PCMs.
As a result, the percentage of running time was lowered. [38] designed and created a model of an enhanced refrigerator utilising PCM, and discovered that using PCM reduced compressor running time by 25% compared to not using PCM. According to the findings of this study, depending on the PCM and thermal load, the PCM reduced compressor running time by roughly 2-36 percent as compared to when the PCM was not used, which is almost identical to the findings of [38] ,which indicate good consistency.

5. Conclusions

The performance increase of a cold storage with and without PCM panels was investigated using numerical simulation. The amount of phase transition material utilised in various ratios. To assess the performance of the air conditioning unit, numerical simulations were run at various loads. Based on the findings of the experiment, the following conclusions were formed.

1. Using phase change material (PCM) panels in the air conditioner condenser coil can reduce temperature rise and keep the temperature stable for up to 8 hours.
2. A temperature drop of 1^{o}C per hour was reported in the air conditioning system employing phase change material panels.
3. Depending on the PCM and heat load, the phase change material improved COP by 11 to 20% as compared to no phase change material.
4. Depending on the heat load, the average compressor running time per cycle with phase change material is lowered significantly, ranging from 17 to 30% less than without phase change material.
5. The coefficient of performance (COP) of the refrigeration cycle with PCM is significantly higher than that of the refrigeration cycle without PCM panel, according to numerical results. The coefficient of performance (COP) is calculated at various loads, and it is discovered that the COP is optimum at 1.5Kg of thermal load, and that it drops as thermal loads increase.
6. As the PCM melts, it absorbs the thermal load that enters the air conditioning area, limiting the rise in cold storage temperature and maintaining a constant temperature inside the cold storage during power outages, which may occur accidentally or on purpose to meet electrical load sifting requirements.

6. Further Scopes

When selecting PCMs for a certain application, three crucial elements to consider are melting temperature, latent heat of fusion, and PCM thermophysical issues. A high heat of fusion and a precise melting/solidification temperature are two of the most important criteria in the selection process (without sub cooling).

2. Erythritol, Hydroquinone, and Dmannitol are three Heat Transfer Fluids HTFs that have undergone substantial research and are appropriate for absorption refrigeration systems operating at temperatures between 140 and 200 degrees Celsius.

The degree of subcooling (of inorganic materials) is a significant factor that impacts the system's thermal capacity, in addition to melting point temperature. Eutectic mixes, on the other hand, can be used if the problems have been identified and treatments have been tried.

3. The storage tank design is crucial for maximising the heat intake and output of the LHTSS. The shell and tube design has shown to be the most efficient, albeit at the cost of PCM volume capacity.

4. Another proven efficient design is the multitube design, which allows for complete charging and discharging in a small amount of time. There hasn't been any research into a good heat-exchanger that takes into account PCM's thermo-physical behaviour. The temperature behaviour of the PCM while charging and discharging is an important aspect to consider. It is determined by the container design and the PCM's heat transmission rate.

5. The temperature of the absorber and condenser is nearly identical. Despite this, no attempt has been made to combine them into a single unit with PCM. The use of PCM at the condenser should be handled with caution, as the PCM's thermal conductivity is crucial for dissipating energy in the process.

The effect of PCM with condenser in hot weather (> 40 °C) on compressor run-time has never been investigated, which is surprising given how widely refrigeration is used in hot regions.

6. Materials with sub-zero temperatures are identified in cold storage, but their thermal reliability, phase separation, and sub-cooling concerns are not thoroughly investigated. Thermal cold storage phase transition materials are rarely studied and observed at the industrial (large scale) level.

7. However, combining Nano particles with various PCMs opens up an infinite number of options and tests, resulting in a new generation of enhanced PCM. Property customization is an excellent technique to make application-based PCM properly suited to the operational conditions of the system.

8. Although there is less work on phase segregation and high temperature cycling in the literature due to the time required to investigate this phenomena, this attribute determines the product's reliability and should be given greater attention. Many salt hydrates have good thermal properties, but they need better techniques to improve them and avoid the problems they're known for.

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