MEE5220 -Convection Heat Transfer

Cost-Driven Preliminary Optimization Approach Using Genetic Algorithm: Design a Cooling System Integrated with PCM-Based Thermal Energy Storage

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Date Submitted: 11/29/2023

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Abstract

Using latent thermal energy storage for active building cooling has caught the attention in recent years due to their proven efficiency to save energy for a reasonable added initial cost. This paper presents a preliminary framework to optimize the sizing of an office building cooling system integrated with a PCM-based thermal energy storage. The optimization was based on minimizing the capital cost of the cooling system components, carbon-dioxide emissions penalty, and electricity. Genetic Algorithm optimization techniques is utilized for this framework, and two type of phase change materials are assessed, including ice and inorganic hydrated salt with melting point of 5 C. The preliminary optimization framework concludes that using the hydrated salt with a melting point of 5 C for the cooling system reduces total cost by 32 % in comparison to using the conventional latent thermal energy storage medium (ice-based).

Introduction / Literature Review

1: Background on Space Cooling in buildings

The energy consumed by buildings accounts for one third of the energy consumption worldwide which makes residential and commercial buildings a major player in the ongoing climate change crisis. [1] states that about 25 % of global carbon dioxide emissions is directly coming out from the buildings' consumption of energy. Moreover, Heating Ventilation and Air Conditioning systems represents 38 % of the overall energy consumption of the buildings according to the same paper [1]. Narrowing down the scope of research to the Unites States, the national renewable energy laboratory (NREL) reveals that buildings in the US are consuming 40% of the total energy produces, and 75 % of these building's usage of energy is in the shape of electricity [2]. The department of energy in the US states that the energy consumption of the HVAC systems in the commercial and residential buildings are 40 % and 50 % respectively. The above-mentioned statistics give an indication of the importance of revolutionizing the energy efficiency in buildings, especially HVAC systems, and invest in innovative and decarbonizing technologies to reduce and optimize the energy consumptions. Space cooling, especially, in the hot and hot humid regions in the united states, is an important and vital thing in buildings that needs to be improved in terms of energy consumption. The interior cooling of residential and commercial buildings is reported to require 409 billion kilowatt-hours of power, which is estimated to be equivalent to 10 % of the overall electricity consumption in the United States up until 2022 [3]. Adrian R. Katili, Rabhab Boukhanouf , and Robin Wilson discusses in [4] different active and passive technologies and the applicability of the utilizing them for hot and humid climates . They reviewed different cooling load profiles for sensible and latent cooling loads and highlighted the limitations of the commonly used thermal comfort models and discussed their alternatives, such as effective temperature approach. A total of six active cooling technologies were explored by the Nottingham University research team. The first technology is the most dominant A/C system in the world which is the Mechanical Vapor Compression system. This system dominates residential and commercial buildings for their reliability, great cooling controllability and their easy installation. It’s concluded that this type of active cooling technology tends to have low COP in hot and humid areas. Split systems are the second active technology reviewed. This type of cooling systems uses air as the cooling medium for the condenser. The research team highlights that 8 % of energy consumption reduction could be achieved if water is used to cool the condenser. The third reviewed technology is the centralized systems. This type is very common for large scale cooling loads such that it facilities and circulates the cooling around the entry building / facility using ventilation ductwork. One of the most promising centralized cooling systems is using seawater-cooled plants which can ensure an efficiency range between 4 to 5. However, this technology is very expensive to implement as it involves high capital cost. The fourth technology reviewed is a promising replacement for the conventional mechanical compressor system, which is Absorption cooling system. This system replaces the electric-powered compressor with a thermal one which can use in many heat recycling applications and solar power-based cooling system. This technology is very promising for small to medium scale cooling load. It can save about 35 % when implanting this technology in combination with solar power in office buildings. The fifth technology is Solar-assisted desiccant system. This is an evolving technology and can be integrated with compressor vapors A/C and it can save up 24 % of energy by the regeneration of desiccants using solar power. The system can deal with the humidification process which is incredibly significant concern in humid regions. Furthermore, the researchers discussed many passive cooling technologies, including Direct and Indirect Evaporative Cooling, Radiant cooling, Ground cooling and Night Ventilation. Nevertheless, the authors state that theses passive technologies are more applicable in dry hot climates and less practical in hot humid climates due to the possibility of increasing the humidity rates in the building above the comfort rate. The authors acknowledged that even though passive cooling system technologies can have potential for low saving of energy, but they are less promising in hot humid regains and the latest of these technologies cannot replace the conventional cooling systems [4][5].

One of the most promising active cooling technologies that have caught the attention of many academics and engineers is using ice thermal energy storage tanks because of its high potential for saving energy.

2: An Overview of Thermal Storage Tank

Thermal storage tank (TES) is a thermal system that stores thermal energy in form of cooling or heating load for later use. The tank, depending on the application, could be charged with thermal energy by active processes such as by the compressor of the HVAC system, or it could be charged passively from the excessive heating and cooling industrial processes. The energy stored in the tank is kept during the off-peak energy demands and released during peak demand of energy. Thermal energy systems play important rule in thermal balancing, especially in applications where energy demand mismatches with the excessive supply. Implementing a TES system in such an application would save a good amount of energy that would have gone wasted. Thermal energy storages are classified based on the application, but a more in-depth way to classify the thermal energy storage is based on the state of the material used to store the energy with as shown in figure (1).

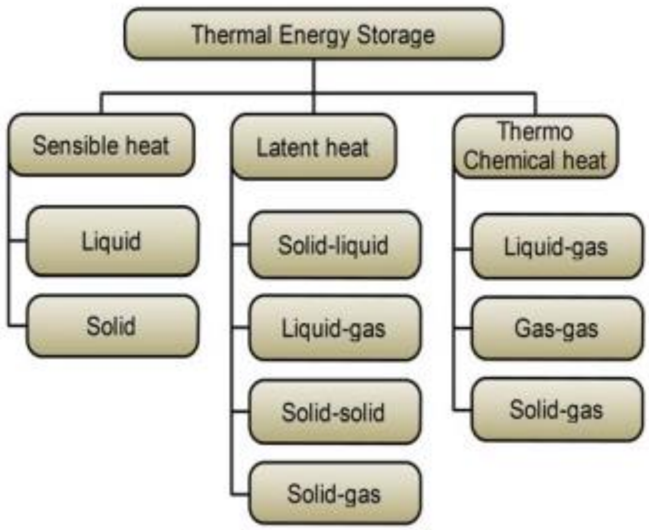


Figure (1): Classification of thermal energy storage based on the state of the thermal storage material [6]

This book by Ibrahim Dincer and Marc A. Rosen [7] discuses in a comprehensive way the various methods to store the cooling and heating thermal energy. Ignoring the chemical thermal storage tank, thermal energy storages, in general, store thermal energy in two forms: sensible and latent. The sensible energy storage is facilitating the storing process in a change in the material temperature. On other words, the storing mechanism is by increasing or decreasing the storing medium (i.g water) to a certain temperature required by the application. The specific heat capacity is an especially important aspect when it comes to choose storing medium for sensible TES's. specific heat capacity is the amount of energy is needed to raise the temperature of the of 1 kg of a given material 1 degree Celsius. Therefore, having a larger heat capacity means that more energy can be stored for each degree in temperature change because the heat capacity is proportional to the amount of energy stored in the tank according to the following relation: $Q = m \cdot c_P \cdot \Delta T$

Moreover, the fact that the temperature of charging and discharging of energy is variable, this allows the sensible TES's to be used in a large range of applications. Nevertheless, one of the disadvantages of sensible TES is that it requires large size of storing capacity. This is why the attention in developing TESs has shifted towards latent TES's.

Latent TES facilitates the thermal energy storing by phase change of the storing medium from liquid too solid (freezing) or from solid to liquid (melting). As the storing medium is undergoing a phase change transition, thermal energy is absorbed or released. There are many advantages of using latent TESs instead of the sensible ones, and storage size is the most important advantages. For instances, storing 1 kg of ice requires only 334 kJ (latent heat of water) of energy to be released, whereas approximately 80 kg of water is required to store same amount of energy. Another advantage is that thermal energy is absorbed and release at a constant temperature which makes the storing process easier to control. However, the phase change temperature is different from a material to a material which makes it problematic to find the proper phase change material (PCM) for a certain application. The latter created an interesting field for academics to explore further and research in the properties of the phase change materials which is discussed in the next section.

3: Phase Change Materials

Phase change materials can be used in variety of applications, especially increasing the efficiency of the thermal envelop. [8] conducted an extensive state of art review for cooling and heating methods in which utilizing PCM (with a phase changer temperature range from 20-32 C) is feasible, such as PCM trombe wall, PCM wallboards, PCM shutters, PCM building blocks, ceiling boards etc. The paper concludes that the PCM-integrated systems have a good potential to save cooling and heating cost. In addition to that, [9] discusses comprehensively the implementation PCM particularly for passive and active cooling applications in buildings. Many insightful findings and conclusions are drawn from this highly important paper. They highlighted that using PCM in the buildings are effective in general because of four important factors: 1) decreasing energy consumption. 2) shifting cooling demand's peak loads, providing more thermal comfortable environments inside the building by decreasing the fluctuations of temperature. Moreover, the paper concludes that using PCMs for passive cooling applications are dynamically efficient when temperature difference during the say is relatively large (up to 15 C). This finding somehow aligns with what is concluded in the paper discussed earlier [4] that passive cooling systems are, in general more effective in dry and hot climates. Furthermore, the paper went to discuss the disadvantages of PCM-based cooling technologies, including the low heat transfer coefficients, PCMs' incomplete solidification during the night. Also, the research team published this paper came up with a topology diagram that comprehensibly summarizes the different applications of PCM in buildings (See Figure (2)).

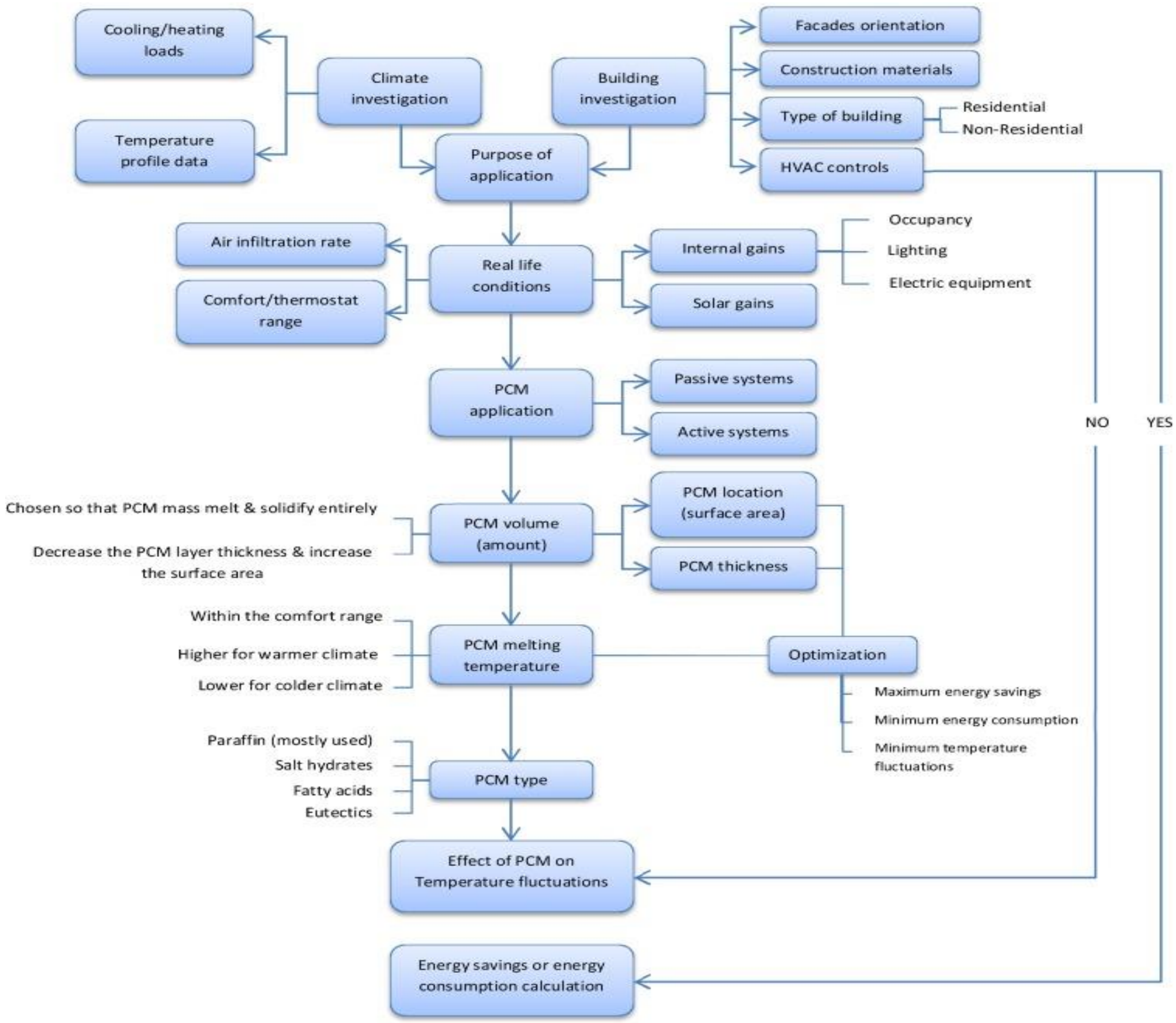


Figure (2): A topology diagram for PCMs cooling applications in buildings [11].

3-1: PCM-Based thermal Storage.

The selection of the feasible PCM for thermal energy storage is dependent upon the type of application, but there exists a general approach to choose the feasible PCM based on its thermal and material properties. A feasible PCM would have a large latent heat, high conductivity, melting temperature range should be within the practical range of the application, melting homogeneously with minimal subcooling, chemically stable, nontoxic, or corrosive and low cost. As mentioned previously, each application gives a range of the allowable melting temperature of the chosen PCM. For example, the PCM selected for A/C systems should have a melting point below 20 C. Moreover, storing the excessive heat from solar energy and rejected heat from thermal systems require a melting temperature range between 15 C to 90 C. PCMs with melting point above 90 C are suitable for absorption refrigeration application. Many papers introduced stochastic selection method to choose the feasible PCM. For example, [9] implemented Muti-Attribute Decision Making (MADM) ranking and Multiple Objective Decision Making (MODM) method to select PCM from a wide range and attributes.

The phase change materials used in TESs can be classified into three main categories which are organic, non-organic, Eutectics of organic and non-organic compounds [17][11][10]. Paraffin waxes and fatty acids are examples of organic PCMs, and Hydrate salts would be an example of inorganic PCMs. The spotlight will be focused on paraffin waxes and salt hydrates since they are the most used PCMs on TESs. Paraffin waxes are heavy organic hydrocarbons which are basically one the crude oil refining's byproducts. Paraffin waxes are widely used PCMs for their adequate features which are having acceptable moderate storage density of about 200 kJ/kg, wide range of melting point, featuring negligible subcooling, chemically stable without phase separation and cheap in prices. One of the sounding disadvantages of paraffin waxes is the low thermal conductivity of about 0.2 W/K-m^2, so strategies for heat transfer enhancements must be implemented. Hydrated Salts are inorganic chemical salts solution dissolved in water at different percentages. When heat is absorbed, the salts separate from the water and form a liquid mixture, while the rejection of heat solidifies the mixture again. One of the Hydrated salts' advantages is their high latent heat of approximately 250 kJ/kg. Also, they have relatively larger thermal conductivity (about 0.5 W/m^2-K) compared to Paraffin Waxes. However, this type of inorganic PCMs introduces many challenges, including cooling down below phase change point without solidifying, sometimes unstable chemically and higher in price. Figure (3) shows a summary of PCMs classifications and figure (4) shows the melting temperature range of PCMs types with respect to their enthalpy.

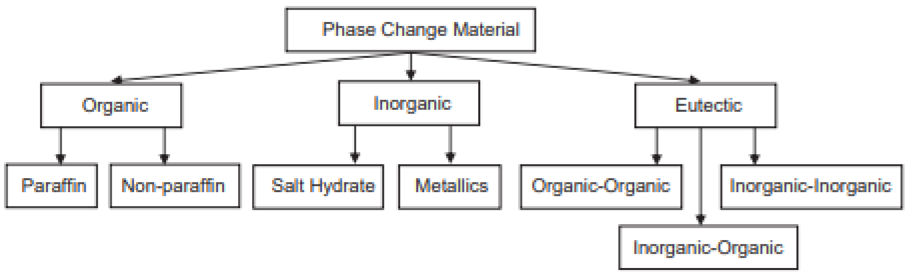


Figure (3): Classification of PCMs [10]

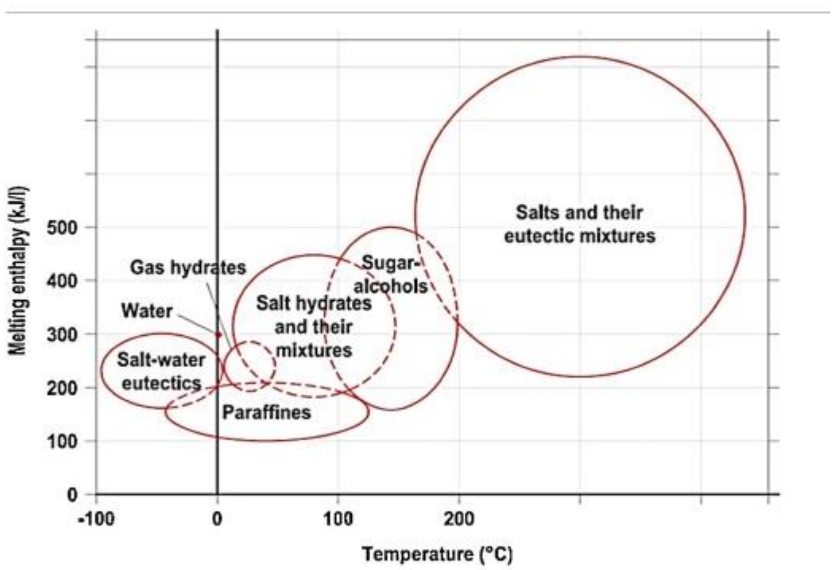


Figure (4): Melting temperature and enthalpy range of PCMs types [11]

3-1-1: Solutions to Common Problems in Phase Change Materials Used in TESs

The challenge associated with the low conductivity in all PCMs can be resolved using heat transfer enhancement strategies in order to enhance the penetration of heat inside the PCM. Some of the examples of these heat transfer enhancements are as follows: metallic fillers, finned tube and meta matrix structure. Another problem facing PCMs, Hydrated Salts in particular, is the subcooling. This problem is usually solved by using nucleating agents, such as Borax, which works to minimize subcooling. Also, another common problem in Hydrated salts is the phase segregation. This can be treated by utilizing some thickening agents (such as Bentonite), but this is come with a trade-off reducing the rate of crystallization and thermal conductivity of the hydrated salt.

3-1-2: Encapsulation of Phase Change Materials Inside TESs

The encapsulation of the PCM is basically keeping the PCM in a container so that the heat transfer fluid in the TES is not affected by it. There are many purposes of encapsulation, including maintain the liquid or solid phase by isolating the PCM from its surroundings, maintaining the PCM’s composition unchanged, minimizing the reaction of the PCM with its surroundings, making the PCM easier to handle and enhancing the PCM’s heat transfer with working fluid during the discharging process.

There are three major types of encapsulation classification based on size. The first type is the Macro-encapsulation in which the PCM is contained in metallic and plastic capsules of sizes greater than 1 mm. The capsules can take many shapes, such as spherical, cylindrical, or rectangular. The second type is the micro-encapsulation in which the capsule has a size range from 1 to 1000 micrometers. This type of encapsulations features higher heat transfer rates due to the increase in the surface area to volume ratio. The temperature range span between the core of the capsule and its surface is negligible which makes the phase change process efficient ad homogenous. The third type is the nano-encapsulation which is a new arising encapsulation technology. The size of the capsules is less1 micrometer. The phase changer process is insanely enhanced, but it comes out of expensive investment.

3-1-3: Design of PCM-Based Thermal Energy Storage

The significant value of using PCM-based TES for cooling purposes is undermined by the poor heat transfer. Therefore, an effective way of designing these heat storage tank must be adopted in order to maximize the PCM that can be utilized. For any cooling system, there is a certain maximum temperature of the cooling storage in which no useful cooling can be accomplished. [12] discuses a thorough review of the PCM-based thermal energy storage system's mathematical model. The mathematical models developed on this paper cab be used to predict behavior of the heat flow within the phase change material. The latter can be utilized to perform simulation studies on TESs and design purposes. Nevertheless, these models didn't catch so much attention for characterizing thermal storage tanks as numerical methods is preferable for their simplicity. According the second law of thermodynamics, an additional irreversible heat transfer process is created when energy transfers from a heat source to a heat sink. This will result in increasing in exergy losses [13], and thus, the stored energy will fail to reach the required temperature of the heat sink. In order to obtain the same heat flow as obtained without having thermal energy storage, the overall temperature difference between the heat source and heat sink must be raised, so the additional exergy loss is minimized. Moreover, the energy is stored directly in the heat transfer fluid for sensible storage tanks, but it's indirectly stored in the PCM for the latent storage tanks. Consequently, the exergy loses in the latent thermal storage tank is more significant comparing to the sensible ones. A Tay et al has developed a computational dynamics model (CFD) for a PCM-thermal storage tank in which the heat transfer fluid is carried using a tube-tank configuration. The author concluded that the heat transfer flow inside the PCM is dominantly one-dimensional. As a result, the researcher developed a mathematical model using

$\varepsilon - NTU$ method to characterize the latent tube in tank thermal energy storage. The model developed enabled to assess the ability of the PCM-storage tank to achieve the required temperature for building's cooling and selecting the correct PCM-based storage tank design specifications accordingly. The heat transfer phenomena due to the phase change transition that occurs through the between PCM and the heat transfer fluid (HTF) can treat the tank as a heat exchanger that facilitate the heat transfer in one dimensional; hence, the usual heat exchanger design and rating techniques can be used.

The effectiveness is a ratio used to evaluate the heat exchanger's performance. This ratio is basically the actual energy discharged from the latent storage tank to the heat transfer fluid to the maximum possible energy that theoretically can be discharged. The maximum theoretical energy is happening at the maximum temperature difference in the heat exchanger. In this case, the difference between the temperature of the heat transfer fluid going into the tank and the phase changing temperature is at which maximum theoretical energy is defined at. [9] utilizes $\varepsilon - NTU$ method as a building block to size the tubing system that carries heat fluid inside latent TES. Also, the paper presented a novel optimization method based on capital cost of a fully integrated latent TES system used for heating. In this optimization method, the authors where able to assess multiple gematrical variables such as the tube diameter and number of tubes required and optimize them from the prospective of capital cost.

4: Presented Approach Summary

In this paper, an integrated PCM-thermal energy storage cooling system for an office building is designed. LMTD-method is used to size the heat exchangers in this system. Also, a preliminary optimization framework using Genetic Algorithm is presented and initiated to optimize multiple geometrical and thermodynamics variables at given constraints. The objective function presented in the system works to size the different component of the system based on minimizing the cost. the cost function consists of three terms: estimated initial cost, estimated electricity cost over a year of operation, estimated environmental plenty for carbon dioxide penalty over a year. It’s worth mentioning that this is a preliminary optimization framework that can be refine and edited in future for better accuracy, and that’s why “estimated” is emphasized.

- **Methodology**

1: Problem Statement and System Description

A cooling system using latent -thermal energy storage technology is asked to be designed for an office building in Melbourne Fl. The office building has a peak cooling load of 2.2 kW and the operation hours of the cooling system is from 7 am to 6 pm for 5 days in the week. The system is idle during the rest of the day such that TES is charged during night because of the low cost of the off-peak hours. The goal is to design the components of this building’s cooling system shown in figure (5), in particular, the PCM-TES, condenser and evaporator and size these components to minimize cost over a conservative given inputs and specifications of temperature and cooling load. The optimization model developed should be applied for ice and a PCM of a melting point of 5 C. Table (1) shows the main inputs, specifications, and assumptions.

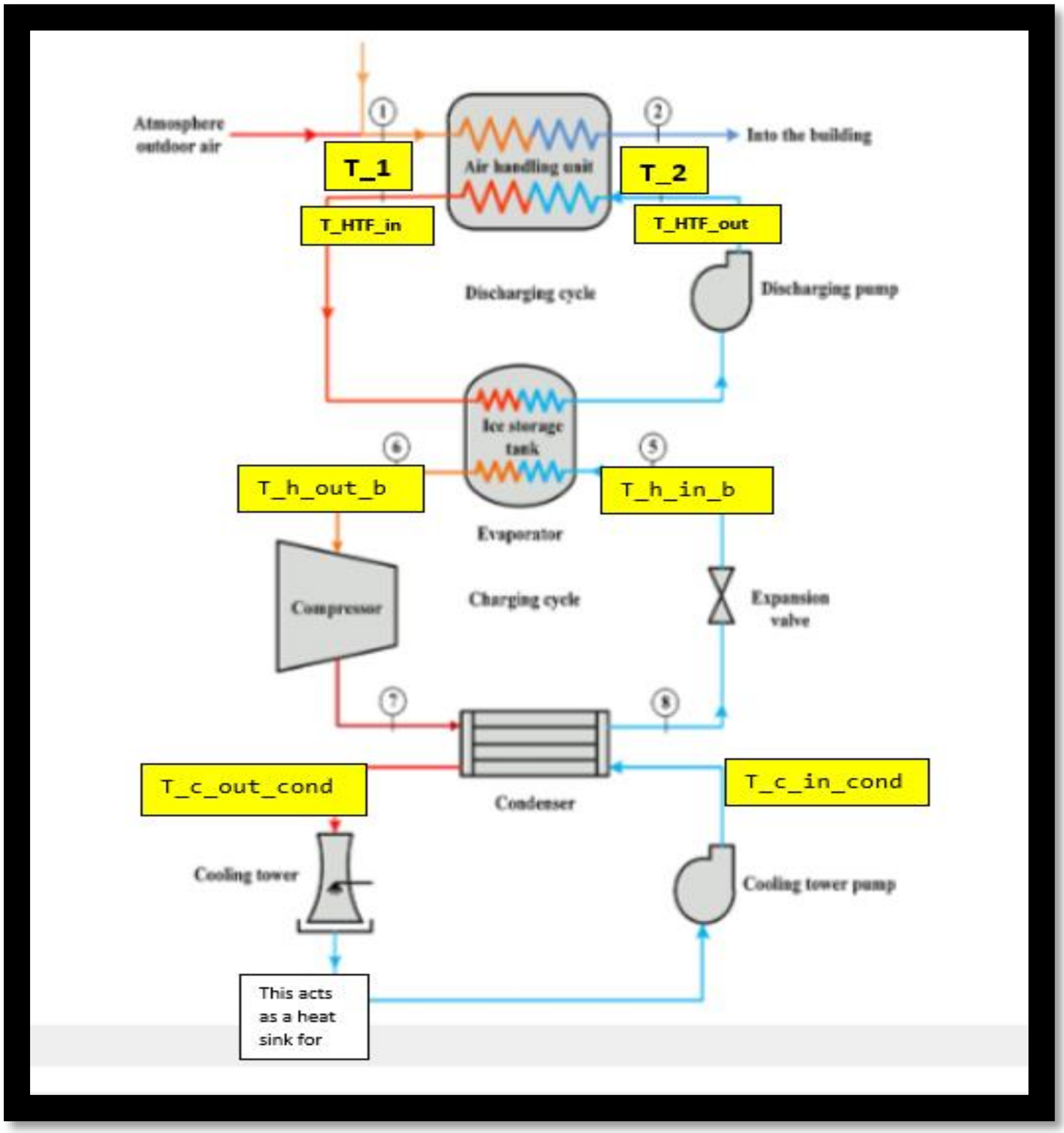


Figure (5): Schematic of the cooling system specified in the problem statement

Table (1): System inputs and specifications

Overall system specifications	
Inputs/ System Specifications	**Notes/ Justifications /Assumptions**
T_1= 32.7 C is the dry bulb ambient temperature which is equal to the maximum temperature in the hottest day of year in Melbourne Fl	Typically, the cooling system's sizing and performance are optimized over a given daily profile of ambient temperature for an entire year or season, but this is ignored for now in this paper [15]
T_2 = 23 C	is the targeted comfort Temp in the office building
Q_c = 2.2 kW is the peak cooling load	Typically, the cooling system's sizing and performance are optimized over a given daily load profile for an entire year, but this is ignored for now in this paper
T_HTF_in is the heat transfer fluid temperature going into the heat storage tank	[14] Typical range for a refrigeration system when using ice for TES is (3 to 5C), and it's (10 to 15) when using higher melting point PCM
T_HTF_out is heat transfer fluid temperature going into the fluid	[14] Typical range for a refrigeration system when using ice for TES is (3 to 5C), and it's (10 to 15) when using higher melting point PCM
T_h_in_b = T_HTF_in	It's assumed that evaporator charges the TES of the PCM's temperature raise up the heat transfer fluid temperature. If we were to apply a control strategy this becomes important to specify when want to charge the TES.
T_h_out_b = T_storing	[14] The goal of the evaporator is to charge PCM-TES to the acceptable storing temperature which ranges from -10 C to the melting point of the PCM used
T_c_in_cond = 35	The temperature of the cooling fluid coming from the cooling tower into the condenser
T_c_out_cond = 40.56	The temperature of the cooling fluid leaving the condenser into the cooling tower. It is recommended that to keep a temperature difference of 5.56 for the cooling tower

T_wb = 26.6	The wet bulb temperature obtained from the A/C design sheet for Melbourne Fl [15]
T_ch = 11 hours (charging time) T_dc = 13 hours (discharging time)	Assume the charging time and discharging time are constant for all PCMs tested
Specification of Fluids used in the cooling system	
Ice: First phase change material used	• Melting Temperature = 0 C • Latent Heat= 336 kJ/kg • Conductivity=2.22 W/m-C • Density = 917 kg/m^3
PCM-1: Potassium Fluoride which inorganic PCM (hydrated salt) is the second PCM tested	• Melting Temperature = 0 C • Latent Heat= 231 kJ/kg • Conductivity=0.584 W/m-C • Density = 1445 kg/m^3
R-134a: the refrigerant used in the charging side	• For a typical refrigeration system, the saturated temperature of the evaporator varies from -30 to the storing temperature of the PCM, where the condensing temperature is fixed at T_c_out_cond +5 , where T_c_cond_out_cond is The temperature of the cooling fluid leaving the condenser into the cooling tower. This is an acceptable condensing temperature as it should be kept within the range (T_wb+5 to 60C) [14]
Glycol-water: 20 % Glycol concentration; is the heat transfer fluid in the discharging cycle	• Freezing Temperature = -0.07306 C (this is typical too kept below the freezing point of the PCM to avoid subcooling) • Heat capacity: 4191 J/C-kg • Prandtl Number = 10.7 • Dynamic Viscosity = 0.001398 Pa-s • Density = 1000 kg/m^3
General Assumptions-Design Related	
Assumptions	**Notes / justification**
The tubes in the TES are distributed evenly and rea parallel to each other	For simplicity

The supercooling is neglected during they phase change process and the natural convection heat transfer of the PCM during the melting process is neglected	The phase change process is dominated by conduction according to [9]
Encapsulation is neglected for the hydrated salt	For simplicity
Assuming the cooling load is constant over the 11 hours in a year	This is a preliminary framework such that it can be refined in the fu
The optimization is only based on minimizing cost, Thermodynamics inefficiencies are negated	It can be integrated into the framework in future for a more robust and accurate optimization.
The volume of the tank is 20% larger than volume of the PCM required	This 20% clarence to give place fir the piping.
A cylinder-shaped tank is considered that its height is equal to its diameter	This characterization of the tank shape minimizes heat leakage according to [16] This will create a geometric constraint that will be discussed in later sections

2: Mathematical Model:

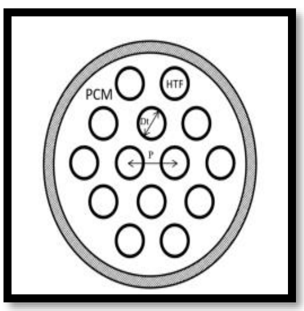
Note that correlations used in this mathematical model are compiled from the following references:[14],[9]and [18]. Refer to the appendices for the full calculations. This section just presents the mathematical model implemented generally:

- **Determining the required storing Volume of PCM:**
  - The cooling load for 11 hours: $Qc = \dot{Q}_c * t_{dh}$ in kWh
  - The required Storage capacity: $\dot{Q}_{st=} \frac{\dot{Q}_c}{eta\_st}$ in kWh, where eta_st is the storage efficiency which is set at 90 %

- Volume of the PCM required: V_pcm = $\dot{Q}_{st} * \frac{3600}{\rho_{pcm} * h\_pcm}$, where ρ_{pcm} is the density of the PCM in kg/m^3 and h_pcm is the latent heat of the PCM in J.kg

➢ **Design for the tube size and number carrying the HTF inside the PCM-TES:**

- Thermal Resistance Circuit:



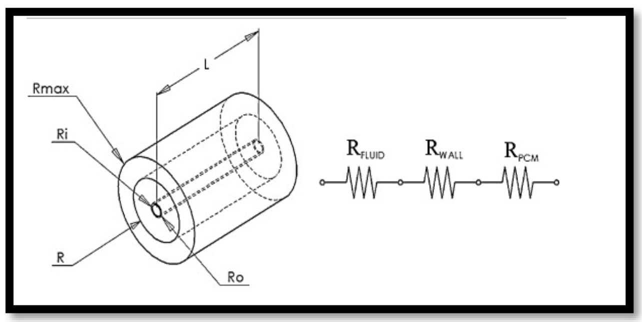


Figure (6): side and top view of the tubing system inside TES along with the thermal resistance Circuit

Where R_fluid is thermal resistance due the forced internal convection in the transfer fluid, R_wall is the conductive thermal resistance of the tube wall. R_PCM is the thermal resistance induced by the conduction and relevant shape factor. The shape factor is normalized by δ which is a fraction of the PCM to be change the phase. This geometrical parameter varies from 0.005 to 0.995, and it's easier to deal since it's not time dependent. The total thermal resistance along with the formulas of each resistance term is given as follows:

$$R_{t,ice} = R_{HTF} + R_{wall} + R_{ice}$$

$$R_{HTF} = \frac{1}{D_i \cdot 3.142 \cdot h_{f,ice} \cdot L_{ice,i}}$$

$$R_{wall} = \frac{\ln\left[\frac{D_o}{D_i}\right]}{2 \cdot 3.142 \cdot k_{wall} \cdot L_{ice,i}}$$

$$R_{ice} = \frac{\ln\left[\frac{(pc_f \cdot ((p_{pitch} \cdot 0.5)^2 - (D_o \cdot 0.5)^2) + (D_o \cdot 0.5)^2)^{0.5}}{D_o \cdot 0.5}\right]}{2 \cdot 3.142 \cdot k_{ice} \cdot L_{ice,i}}$$

Here the pitch is the same as r_max shown in figure (6). pc_f is δ and fixed at 0.9 for parametric studies, but for the optimization thermal resistance induced by the PCM is calculated for all δ range using numerical technic as the following:

$$R_{pcm} = \sum_{\delta=0.005}^{0.995} (R_{pcm}(\delta))$$

The above formula was solved using the for loop in MATLAB with an interval of 0.01.

h_f_pcm is the convection heat transfer coefficient of the HTF side which is calculated using the following formulas:

$$A_{cross,tube} = 0.25 \cdot 3.142 \cdot D_i^2 \cdot N$$

$$Re_f = \dot{m}_{HTF} \cdot \frac{D_i}{A_{cross,tube} \cdot \mu_{HTF}}$$

$$h_{f,ice} = Nusselt_{f,ice} \cdot \frac{k_{HTF}}{D_i}$$

$$Nusselt_{f,ice} = 3.66 + \frac{0.0668 \cdot \frac{D_i}{L_{ice,i}} \cdot Pr_{HTF} \cdot Re_f}{1 + 0.04 \cdot \left[\frac{D_i}{L_{ice,i}} \cdot Re_f \cdot Pr_{HTF}\right]^{(2/3)}}$$

The above Nusselt number correlation given for laminar flow. Even though the parametric studies conducted in EES that flow of the HTF remains laminar for the given range of the mass flow rate, the Nusselt number correlation for turbulent flow in included in the MATLAB code to have a comprehensive framework.

The overall heat transfer coefficient based on the outer surface area of the tube is then calculated:

$$U = \frac{1}{R_{t,ice} \cdot D_o \cdot 3.142 \cdot L_{ice,i}}$$

Using the LMTD method, the length required for the tube is calculated:

$$\delta_{T,1} = T_{HTF,in} - T_{melting,ice}$$

$$\delta_{T,2} = T_{HTF,out} - T_{melting,ice}$$

$$LMTD = \frac{\delta_{T,1} - \delta_{T,2}}{\ln\left[\frac{\delta_{T,1}}{\delta_{T,2}}\right]}$$

$$\dot{Q}_{st} = \frac{Q_{st}}{time_{dc}}$$

$$\dot{Q}_{st} = U \cdot D_o \cdot 3.142 \cdot L_{ice,new} \cdot N \cdot LMTD$$

Note that the length is calculated iteratively as an initial value must be set.

Finally, the pressure drop due for laminar flow is then calculated:

$$f_{avg,ice} = \frac{4}{Re_f} \cdot \left[\frac{3.44}{\sqrt{L_{plus,ice}}} + \frac{\frac{1.25}{4 \cdot L_{plus,ice}} + \frac{64}{4} - \frac{3.44}{\sqrt{L_{plus,ice}}}}{1 + \frac{0.00021}{L_{plus,ice}^2}}\right]$$

$$P_{drop} = f_{avg,ice} \cdot \frac{L_{ice}}{D_i} \cdot \frac{\dot{m}_{HTF}^2}{2 \cdot \rho_{HTF} \cdot A_{cross,tube}^2}$$

- **Evaluating the pumping power (In discharging cycle):**

W_pumping_inlet = P_drop*V_rate/eta_pump

V_rate is the volumetric rate which is equal to the mass flow rate of the heat transfer fluid divided by the density of HTF. The efficiency of the pump is set at 0.8.

- **Design of condenser and evaporator using LMTD method (Charging Cycle):**

  - Heat Exchanger selected:

A Chevron type heat exchanger that has a herringbone pattern shape is selected for both condenser and evaporator. Figure (7) shows the schematic of this heat exchanger:

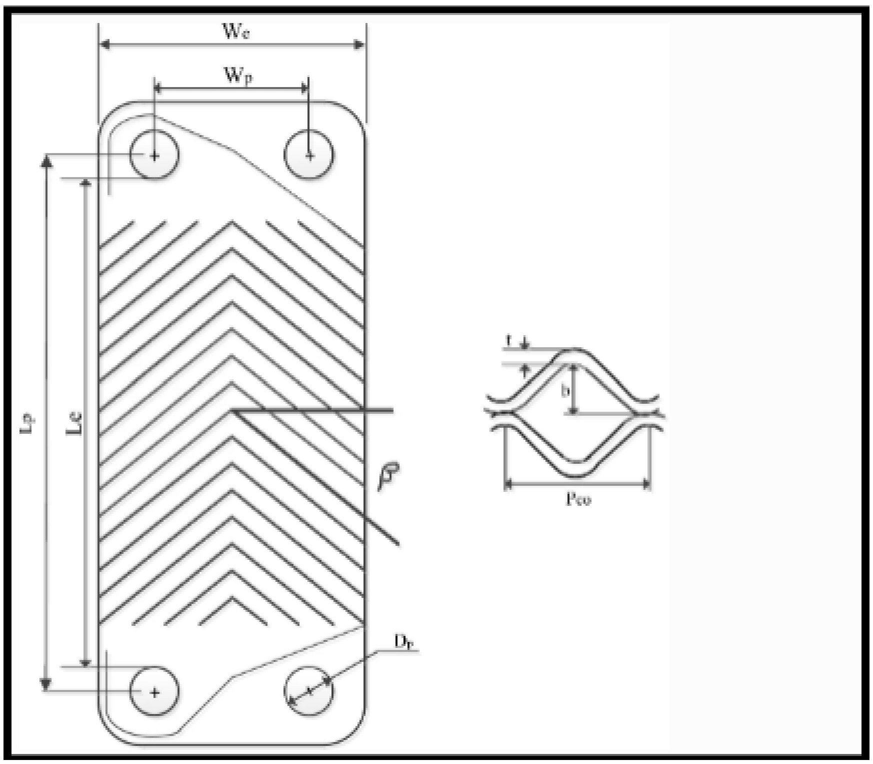


Figure (7): scheme of the heat exchanger type used for condenser and the evaporator in the charging cycle.

W_e , W_p, and L_p are fixed standardized dimensions specified available for commercial use [19] The LMTD method is used to design for the number of the plates in the condenser and evaporator. The correlations used to calculate the convective heat transfer coefficients are obtained from [20]

3: Cost Model

Table (2) shows all the correlations used to calculate the estimated initial installing cost, estimated electricity spending cost over a year cycle, and environment penalty. Note that the initial cost of AHU and the pumps are not included in the model as well as the electric consumption cost of the fans of the cooling tower and the AHU.

Table (2): summary of Cost Correlations

Investment Cost	
System component	**Correlation / Notes**
Storage unit cost based on the capacity of the PCM required	C_st = 8.67 *10^(2.911*exp (0.1416*ln(V_ice)))
The cost of the tubing system inside the TES	C_HEX = [(516.621*A_pcm)+268.45*F_p* F_m *F_L] ; where A_pcm is the heat transfer area of the tubing system inside the TES, F_m is the material cost which is given for the stainless steel as follows : F_m = 2.70 + (10.7639*A_ice)^(0.07) F_p is the cost associated with gauge pressure, which is low, and thus it's given to be equal to F_m. F_L is the cost factor associated with the length of the tube(Check out the MATLAB code for more details)
Cost of the Condenser	C_cond = (516.621*A_cond)+268.45; where A_cond is the heat transfer area of the condenser
Cost of the Evaporator	C_evap = 16648.3 *A_b^(0.6123), where A_b is the heat transfer area of the evaporator
Cost of the cooling tower	C_cooling_tower=[746.749 *(m_dot_cooling_tower^0.79)*(T_coooling_tower_df^0.57)*((T_c_out_cond- T_wb)^(-0.9924))*((0.022*T_wb) +0.39)^(2.447)]
Cost of the Expansion Valve	C_ExpV = 114.5*m_dot_h_cond
Cost of the Compressor	C_comp=[(39*m_dot_h_cond)/(0.9-0.6))*(P_cond/P_evap)*log(P_cond/P_evap)]
Electricity Consumption Cost	

C_elect = [(W_dot_compressor*(c_elc_off_peak/3600)) +(W_pumping_inlet *(c_elc_on_peak/3600))]

Where W_dot_compressor is the power input to the compressor, W_pumping is the input power into the The pump in the charging cycle. c_ele__off_peak is the off-peak electric rating in Melbourne Fl which is 0.09 $/kWh whereas c_ele_ob_peak is the ob-peak electric rating in Melbourne Fl which is 0.14$/kWh.

Environmental Penalty

C_env =((mass_co2/1000)* c_co2)

Where c_co2 is the penalty cost of co_2 per 1 cycle which is given as 90 $ per ton of carbon dioxide emssions. Here the mass_co2 is the mass of annual emission carbon dioxide in kg ,and it is given as follows: mass_co2= emm_factor_co2*(E_cons_A); here = emm_factor_co2 is CO2 emission factor and it is set at 0.968 where is E_cons_A is given using the following formula:

E_cons_A = (W_dot_compressor*N_off)+ (W_pumping_inlet*N_on)

N_off is the number of housr of the charging cycle over a year and N_on is the number of the hours of the discharging cycle over a year.

4: Pre-Optimization Scanning: Parametric studies for TES

Parametric studies were conducted in EES to assess the response of many design-concerning parameters for the thermal storage tank. The trendlines of tube length inside tank, the drop pressure, and the cost of tubing system inside the tank. The behavior of theses parameters as changing the mass flow rate of the heat transfer fluid from 0.05 to 0.2. at three fixed number of tubes varying from (20, 50 and 150) and for ice and the hydrated salt PCM. Figures 9 through 14 show the plots resulted from these parametric studies.

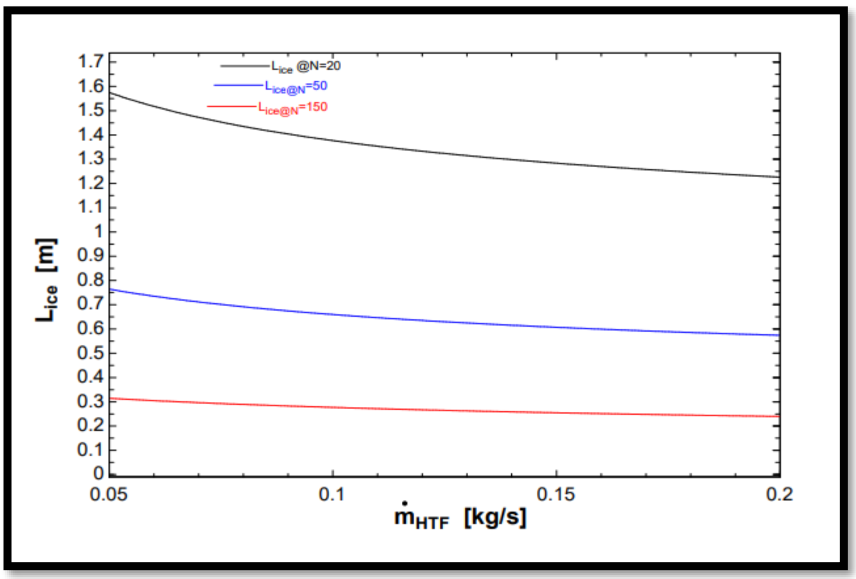


Figure (9): Tube length of the ice-TES as HTF mass flow rate increases

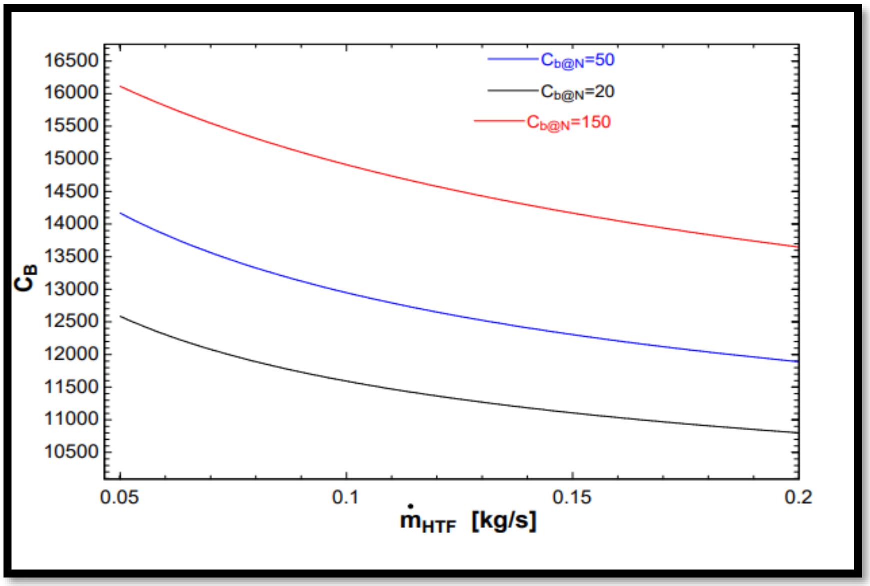


Figure (10): Cost of the tubing system for ICE-TES as HTF mass flow rate increases

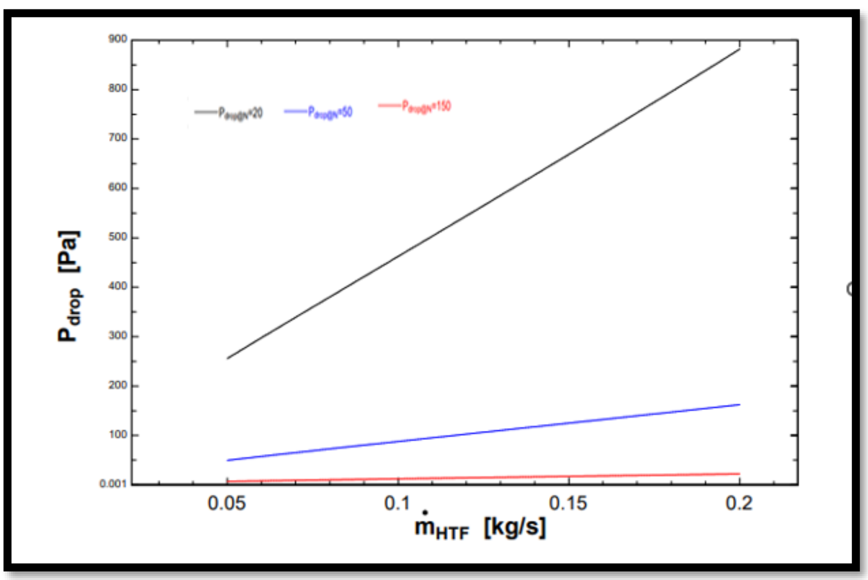


Figure (11): pressure drop for the ice-TES as HTF mass flow rate increases

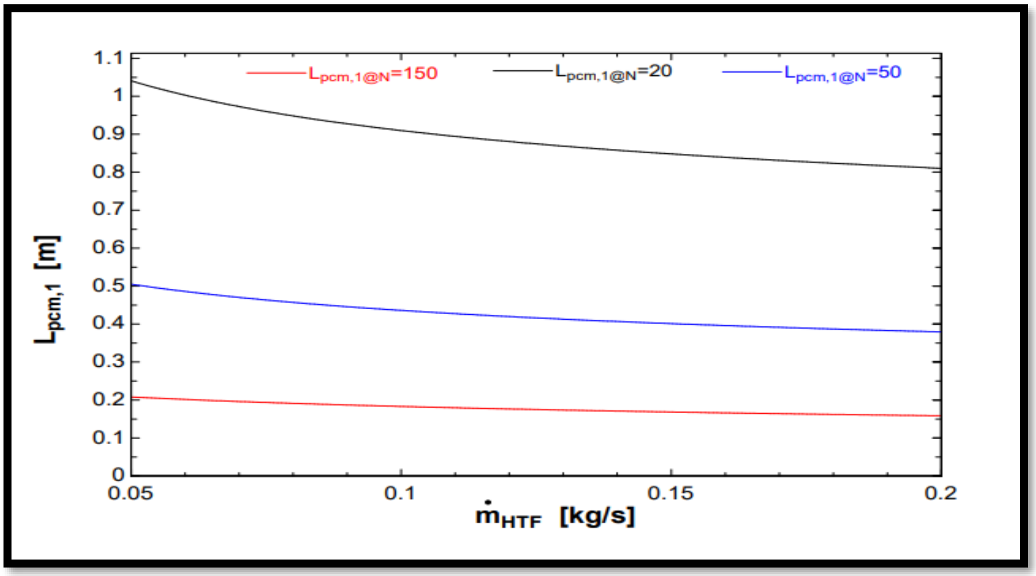


Figure (12): Tube length of the Hydrated salt-TES as HTF mass flow rate increases

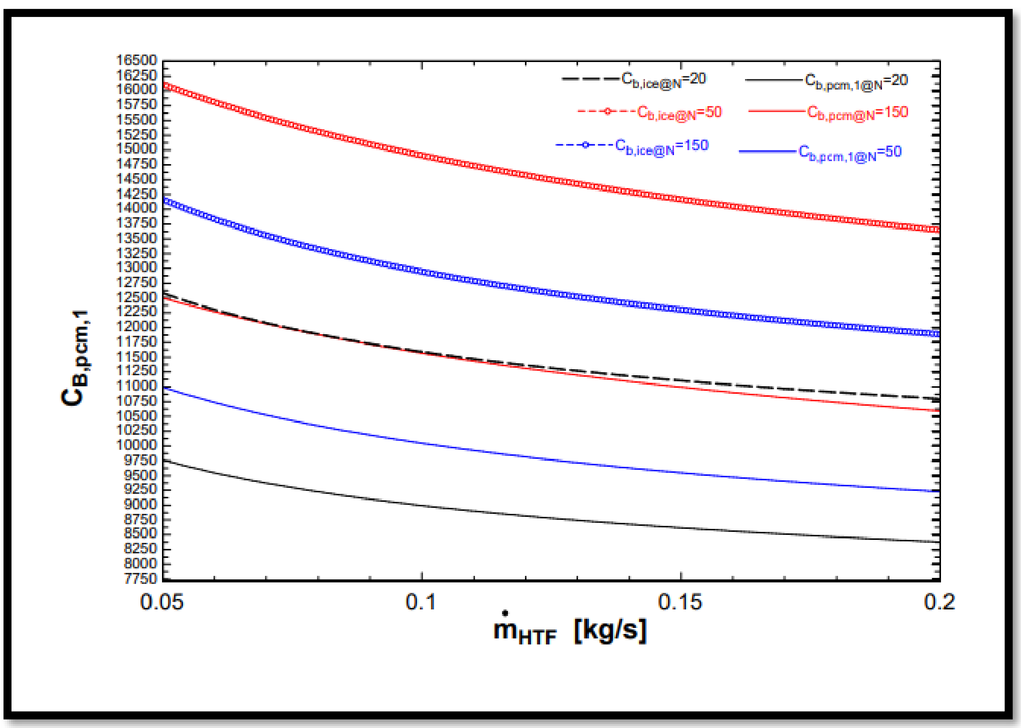


Figure (13): Cost of the tubing system for ICE-TES and the hydrated salt -TES as HTF mass flow rate increases

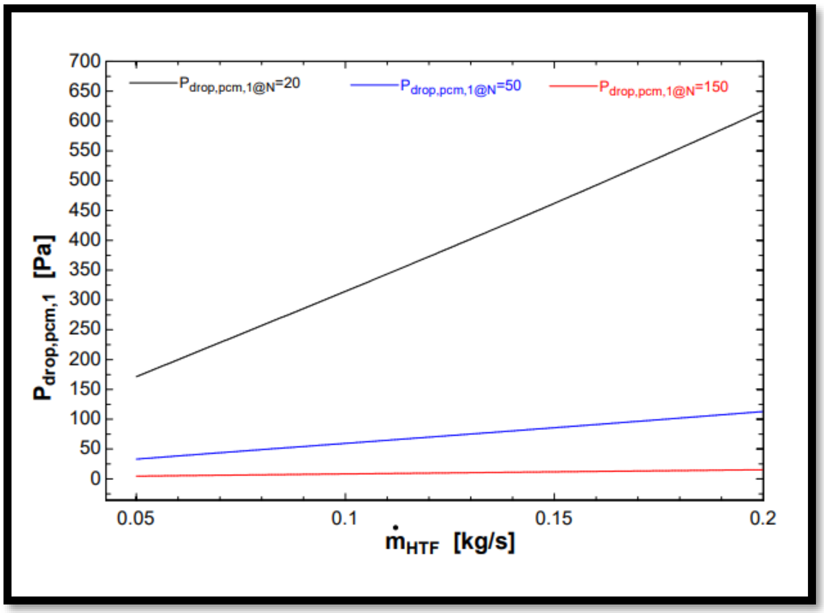


Figure (14): pressure drop for the hydrated salt-TES as HTF mass flow rate increases

Figure (9) shows that as the mass flow rate increases a smaller length size required for the ice-TES which will in return the decrease installment cost (see figure (10)) as it's correlated with the surface area of the tubing system. Figures (10) shows also having while having a greater number of tubes make the system more compact, it results in requiring more installing money. Figure (11) shows that as the mass flow rate increase the pressure drop increases in general, but having a smaller number of tubes increases the pressure drop even further because. Increasing the number of the tubes would result in minimizing the effect of the friction forces induced by increasing the mass flow rate of the heat transfer fluid and its interaction with subrace of the tube. While unceasing the numbed oof tubes minimize the effect of the pressure drop, this comes with a trade-off of increasing the installing cost. Figure (12) shows the relationship between the tube length and increasing the mass flow rate for the hydrated salt with a melting point of 0 C. It shows the same behavior as the ice-based TES but it results in requiring a smaller tube length. The pressure drop increases as well but it is smaller comparing to the ice-based TES. Moreover, increasing the number of the tubes would minimize the effect of pressure drop for the hydrated salt-TES, but with same trade-off of increasing the installing cost that is reported for the ice-TES. Nevertheless, the plot in figure (13) is an interesting one. The concern regarding increasing the number of tubes and keeping an adequate pressure drop may be resolved by using the hydrated salt with melting point of 5 C. Figure (13) shows that the installing cost of the hydrated salt TES using 150 tubes the same as the installation of cost of ice TES when using twenty tubes.

While these parametric studies provide a good insight into the design parameters of the thermals storage tank when just varying one operating variable (mass flow rate of the HTF), there are many decision variables involved in the design process with a set of defined constraints. It's too complex for the parametric studies to account for many decision variables and constraints, hence an optimization method of some sort needs to be used.

5: Optimization Method

- **Genetic Algorithm GA:**

For this paper, the genetic algorithm GA optimization method is used. It is a quite common optimization methods used widely to optimize complex and non-nonlinear systems that has multimodal landscapes. The method is able to solve constrained and non-constraint minimization and maximization problems based on the natural selection concept, which is the process used in the biological evolution in the biology field. The genetic algorithm find solution by randomly applying changes to the current solutions too generate new ones util it gets the best solution. The genetic algorithm works on a population cottoning a certain number of solutions, and this population has a certain size, also known as pop size, which is number of solutions within this population. Each solution in this population is referred as an individual, and each individual solution is defined by a chromosome. A set of features is used to represent the chromosome which in returns defines the individual solution. Each chromosome consists of a set of genes. Moreover, each individual (solution) has a prescribed fitness value. A function called a fitness function is used to select the best individual (solution). The result of this function is the fitness value that indicates the quality of the individual(solution). When we have higher fitness value, that means a higher quality is obtained. Moreover, the criteria of selecting the best individuals (generating an optimal feasible regain) is done according to their quality that is implemented to generate what is known as a ,mating pool, such that the individuals with higher quality have higher probability of being picked for the mating pool.

The individuals(solutions) that get selected for the mating pool are called parents. For every two parents being selected in the mating pool, two children are produced (offsprings). The quality will get better by generating children by mating the best quality parents. Therefore, the quality offspring is getting better than parents. The bad individuals (solutions) prescribed by their bad quality will be prevented from generating bad quality children and elementary them from the mating pool. This process will keep mating and pick high quality individuals, and the chances of the keeping individuals of good features will get higher while the bad ones are left out. Eventually this process will be terminated once an optimal or acceptable solution is obtained [21][22]. Figure (15) shows the steps of GA steps in more technical way.

There are many advantages of using GA, and one of the most important one is that its logic can be applied in lots of theoretical and applied science applications. This is because it can facilitate complex and nonlinear objective define functions with a mix of variables that can discrete, continuous, integers and ect. On the other hands, one of the limitations of using this algorithm is that it requires large amount of computational resource's and does not guarantee to find a global optimal solution.

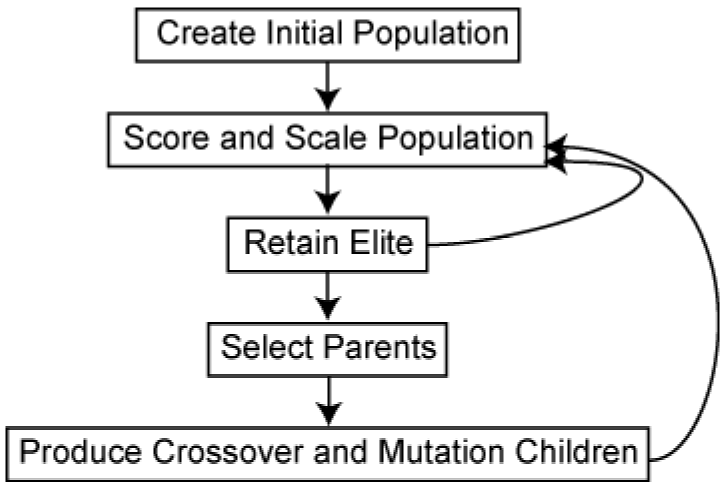


Figure (15): Steps of Genetic Algorithm

- **Economic-environment optimization of the PCM-TES-based cooling system using GA:**

The GA optimization method is utilized to optimize the sizing of the PCM-thermal energy storage cooling system, particularly the storage unit, the tubing system, evaporator, and condenser, by the means of minimizing the total cost. The objective function defined for this problem is as follows:

$$Minimize\ Obj(f) = C_{total}$$

The total cost consists of three terms as following:

C_total = C_investment +C_elect + C_env

1) C_investment which is the summation of the Capital cost of each component in the charging and discharging cycle (except the AHU and the pump in the Charing Charing cycle) C_investment is given as follows:

C_investment=[C_HEX +C_pumping +C_evap + C_cond + C_comp + C_ExpV + C_cooling_tower+C_st]

Note that all cost correlations used are listed in the cost model section elaborated clearly in the MTALB code (check out appendices). 2) C_elect , which is the amount electricity required by the model over a year. The formula of this term is listed in the cost model section. 3) C_env which is the environment penalty due to the carbon dioxide emitted from the cooling system for a year. The correlation of C_elect is also listed in the cost model section.

The mighty of using GA is specifying as many decision variables and constraints as needed. A total of nine decision variables are specified for this problem. Table (3) lists decision variables used along with their bounds and the custom constraint.

Table (3): Setup of the decision variables and customized constraints fed in the GA optimization code

Decision Variable	Lower Bound	Upper Bound	Important Notes and Clarifications
L_ice: Length of the tubes required inside the TES	0.1	6.0960	The length of HTF tube is given at this range because F_L , the tube length cost factor is given for this range of tube length, check out the MATLAB code for more clarification
N : Number of Tubes required inside the TES	10	150	Number of tubes is within a an adequate range to allows more population size for the algorithm to pick from
m_dot_HTF: the mass flow rate of the HTF	0.05	0.2	From literature review, this range is typical for the given cooling load of the system.
T_HTF_in: the temperature of the mass flow rate going into the tank during discharging process	11	13	For the ice is given from 11 to 13 C , and for the hydrated salt used it was rest from 18 to 23. This range is found typical for refrigeration system
T_HTF_out: the temperature of the mass flow rate going into the tank during discharging process	3	5	For the ice is given from 3 to 5 C , and for the hydrated salt used it was rest from 10 to 15. This range is found to be typical for refrigeration system.
D_o: the outer diameter of the HTF tube	6.4mm	101.6mm	This range is standardized commercially. This is given for a custom value within this range. Refer to the custom constraint in this table.
t_tube: the thickness of the HTF tube	9.1mm	16.3 mm	This range is standardized commercially. This is given for a custom value within this range. Refer to the custom constraint in this table.
T_storing: the temperature at which the PCM is stored	-10	T_melting	Where T_melting = 0 C for ice and 5 for the hydrated salt
T_c_b: the evaporative temperature of the evaporator in the charging system	-30	0	This is typical, working saturated temperature for the evaporator in the refrigeration system
Custom Constraints			

<table>
<tr>
<td>D_o and tube_t: are standardized within a given a range and for given valuesof D_o there is a correspondent value of tube_t</td>
<td>
<table>
<tr><th>Tube thickness (mm)</th><th>Outer tube diameters (mm)</th></tr>
<tr><td>0.91</td><td>6.4 9.5 15.9</td></tr>
<tr><td>1.02</td><td>12.7 19.1</td></tr>
<tr><td>1.22</td><td>22.2 25.4 28.6 31.8 34.9 38.1 41.3 44.5</td></tr>
<tr><td>1.63</td><td>50.8 54 66.7 76.2 101.6</td></tr>
</table>
</td>
</tr>
<tr>
<td>(L_ice < height of the tank):

The length of HTF tube inside the TES is strictly smaller than the height of the tank

(V_tube <= (V_tank-V_ice): The volume of the HTF tube must be equal or less the available tank space , The available volume tank is defined s the actual volume of the tank mice the volume accommodated by the pcm</td>
<td>V_tank = V_ice/(1-0.20)% allow an additional 20% for the pipes and clearance, where V_ice is the voulme of PCM required to stoareg the thermal eneregy

height = (V_tank /(pi*0.25))^(1/3) % height of the tank which is equal to its diamter

V_tube = N*pi*0.25*(D_o^2)*L_ice

V_ice_ava= V_tank-V_tube</td>
</tr>
<tr>
<td>(T_c_b < T_storing): The saturated evaporative temperature must be strictly kept under the storing temperature of the PCM .

Also T_storing is strictly smaller than 0 for all pcm</td>
<td>These are recommeded constraints for refrigration system[14]</td>
</tr>
</table>

Results

- **For Ice**

Table (4): GA results for the cooling system when using ice as the PCM for the TES

Execution MATLAB code result	**Optimization terminated: average change in the fitness value less than options.FunctionToleranc eand constraint violation is less than options.ConstraintTolerance.**
Optimal Solutions for the Decision Variables	
Decision Variable	**Result**
L_ice	m
Tube_t	0.00091 m
D_o	0.0064 m
N	10
m_dot_HTF	0.17376 kg/s
T_HTF_in	11.0002 C
T_HTF_out	4.375 C
T_storing	0 C
T_c_b	-29.9 c
The cost objective function Findings	

Minimal Total Cost (C_total)	**214812**
C_investment	124384
C_env	**9.0425e4**
C_elct	**0.0083**
Important Optimal system components' sizing findings	
Number of Plates required for the condenser HEX	**3**
Number of Plates required for the evaporator HEX	**192**

- **For the hydrated salt PCM:**

Table (5): GA results for the cooling system when using hydrated salt (T_m=5) as the PCM for the TES

Execution MATLAB code result	**Optimization terminated: average change in the fitness value less than options.FunctionToleranc eand constraint violation is less than options.ConstraintTolerance**
Optimal Solutions for the Decision Variables	
Decision Variable	**Result**
L_ice	0.1 m
Tube_t	0.00163 m
D_o	0.10106 m
N	20
m_dot_HTF	0.2kg/s

T_HTF_in	20.28 C
T_HTF_out	11.0285 C
T_storing	4.99 C
T_c_b	-29.4 c
The cost objective function Findings	
Minimal Total Cost (C_total)	**155488**
C_investment	59100
C_env	**96385**
C_elct	**0.00886**
Important Optimal system components' sizing findings	
Number of Plates required for the condenser HEX	**3**
Number of Plates required for the evaporator HEX	**49**

- **Results Discussion**

Note the full solutions including the cost of each of the system's component can be found on the appendices. Also, the MATLAB code for GA optimization when using ice is included in the appendices, and the same framework code is used for the hydrated salt with the difference that the initial values are set for this PCM now.

Table (4) and Table (5) highlight the important results and findings from the optimization code implemented in MATLAB for both ice and the hydrated salt PCM. For both PCMs, the GA optimization was executed successfully and converged to a solution. The message shown in the MTLAB output widow indicates that the average change in the fitness value is less “options.FunctionTolerance” which means that algorithm stopped because the average change in the objective function over all generations fell underneath a specific threshold. On other words, the GA optimization process has achieved a point I which generating more iterations does not result in minimizing the objective function any further. The constraint violation was found to be less than constraint tolerance which implies that the solution obtained by the GA is satisfied the constraint defined to an acceptable scale. Speaking of the optimization result using ice as PCM, it was, the total cost was calculated to be 214812 $, and the objective function is dominated by the capital cost of the components, specifically the evaporator capital cost. The length, outer diameter, tube thickness and the number of tubes for the HTF inside tank were optimized based on the cost and were found to be close to the bounds defined for them. Using the hydrated salt for TES in this cooling system resulted in less total cost which is 155488. The execution is initially associated with reduction in the evaporator cost because of having a smaller number of the plates. The estimated total cost of the of the cooling system that using 5 C melting point PCM is 32 % than o that ice one. Moreover, the penalty due to the carbon dioxide emissions is somehow significant in the objective function. The latter was found larger in the hydrated salt PCM due to the higher required compressing power in the charging cycle. Furthermore, the optimal storing point was found to be equal to the melting point of the PCM. C_elc value is extremely small in both cases. The correlation used might have underestimated the electricity consumption cost.

- **Statement of Error and Accuracy**

Since this project is tended to establish a preliminary framework for PCM-TES cooling system. Many assumptions and shortcuts have been taken which would have reduced the accuracy of the GA findings. For example, the GA optimization would have been more accurate if the cost module is evaluated over a given cooling profile of an actual office building during the entire day and for entire year. The same thing is true for the weather condition. Also, the cost correlations used might have underestimated or overestimated the actual cost, such as the electric cost. Therefore, more time needs be invested in validating the framework established through intensified literature review of existing papers or asking experts in the field. Moreover, the model established did not consider the different control strategies for the cooling system. Also, while the GA converges to an optimal solution, there is no guarantee that a global optimal solution exists unless sensitivity analysis is conducted. The latter is not done for this project for the lacking the technical knowledge to do so. Finally, one of the major things that has lacked the accuracy of accuracy of the sizing of the component is ignoring the thermodynamics inefficiencies.

- **Conclusion**

The main objective of this paper is to establish a preliminary framework to optimize the sizing an office building's cooling system integrated with latent thermal energy storage. A comprehensive literature review was conducted to explore the state of art the latest active and passive cooling technologies, types of thermal energy storage, and classifications and applications of phase change materials. The literature concluded that passive cooling technics are more suitable for dry and got climates. Also, latent thermal energy storage is much more efficient than the sensible ones because it requires less storage capacity to store the same amount of thermal energy, However, there are many challenges associated with using PCM for latent thermal energy storage, such as finding the proper PCM that has a melting point suits the targeted application. Speaking of achieving the project's goal, Generic Algorithm optimization techniques was used to size the cooling system based on minimizing the overall cost. The objective function includes three cost terms, which are Capital cost, environmental cost, and the electric cost. The algorithm was assessed for two PCMs which are ice and the inorganic hydrated salt with a melting point of 5 C. It was found that using a the 5 Celsius melting point PCM for the cooling system specified for this application results in having 32 % less total cost that the ice-based cooling system. While this project presents a framework for optimizing based on cost, many future refinements and improvements need to be considered, including adding a thermodynamic model to the objective function, using accurate cost correlations, and running sensitivity analysis for the GA code.

References:

[1]:https://www.sciencedirect.com/science/article/pii/S235248472101427X#:~:text=HVAC%20accounts%20for%2038%25%20of,to%2012%25%20of%20final%20energy

[2]: https://www.nrel.gov/news/features/2023/nrel-researchers-reveal-how-buildings-across-the-united-states-do-and-could-use-energy.html

[3]: https://www.eia.gov/tools/faqs/faq.php?id=1174&t=1

[4]: https://www.researchgate.net/profile/R-Boukhanouf/publication/281437940_Space_Cooling_in_Buildings_in_Hot_and_Humid_Climates_-_a_Review_of_the_Effect_of_Humidity_on_the_Applicability_of_Existing_Cooling_Techniques/links/56430f7508ae9f9c13e01874/Space-Cooling-in-Buildings-in-Hot-and-Humid-Climates-a-Review-of-the-Effect-of-Humidity-on-the-Applicability-of-Existing-Cooling-Techniques.pdf

[5]:https://www.researchgate.net/publication/271881424_Passive_ground_cooling_system_for_low_energy_buildings_in_Malaysia_hot_and_humid_climates

[6]:https://www.researchgate.net/publication/272179312_SEASONAL_THERMAL_ENERGY_STORAGE_CONCEPTS

[7]: Thermal Energy Storage: Systems and Applications - Ibrahim Dinçer, Marc A. Rosen - Google Books

[8]: PCM thermal storage in buildings: A state of art - ScienceDirect

[9]: Shell-and-Tube Latent Heat Thermal Energy Storage Design Methodology with Material Selection, Storage Performance Evaluation, and Cost Minimization (ntu.edu.sg)

[10]: Phase change materials integrated in building walls_ A state of the art review (sciencedirectassets.com)

[11]: Phase change materials (PCM) for cooling applications in buildings: A review - ScienceDirect

[12]: https://www.sciencedirect.com/science/article/pii/S1364032106001651

[13]: Bejan, Adrian. *Advanced engineering thermodynamics*. John Wiley & Sons, 2016.

[14]: https://www.sciencedirect.com/science/article/pii/S0378778813000133

[15]: http://cms.ashrae.biz/weatherdata/STATIONS/722040_p.pdf

[16]: Performance assessment of some ice TES systems (sciencedirectassets.com)

[17]: Phase change materials for thermal energy storage | Climate Technology Centre & Network | Tue, 11/08/2016 (ctc-n.org)

[18]: Designing a PCM storage system using the effectiveness-number of transfer units method in low energy cooling of buildings - ScienceDirect

[19]: B3-115A 60 Plate Heat Exchanger with M8 Mounting Studs [HX115060] | DudaDiesel Biodiesel Supplies

[20]: Design of plate heat exchangers for use in medium temperature organic Rankine cycles | Heat and Mass Transfer (springer.com)

[21]: **https://towardsdatascience.com/introduction-to-optimization-with-genetic-algorithm-2f5001d9964b**

[22]: **https://www.mathworks.com/help/gads/what-is-the-genetic-algorithm.html**

Appendix (A): EES code


```
// Primary Set-up & Givens:
// Cooling Load :
Q_c_dot = 2e3  [W]
// Operating hours:
time_dc = 11 [h]
// Cooling load for 11 hours, working hours from 7 AM to 6 PM, assuming the system is idle in the rest of the day
Q_c = Q_c_dot *time_dc
// the efficinecy of the storage is typically= 0.9
eta_st = 0.9
// the required storage capacity :
Q_st = Q_c / eta_st
// Ambient and operating inputs:
// For AHU ( Air Handling unit ):
// Comfort Temp
T_2 = 23 [C]
// The max  ambient dry bulb Temp in the hottest day in summer for Melbourne, Fl:
T_1  = 32.7 [C]
// the mass flow rate of the AHU:
T_avg_AHU =( T_2+T_1)/ 2
// Capacity of air at T__avg_AHU , specific enthalpy h_g_1 of water vapor at T_1 , h_g_2 at T_2, omega absolute humidity at
T_1 and T_2

cp_air_AHU = 1.005e3 [J/kg-C]
P_amb = 100e3 [Pa]

relative_humidity_1 = 0.7   // ambient relative humidity ( measured by weather station)
relative_humidity_2=0.55   //  needed relative hunidity inside the building

P_sat_1 =   4.951e3[Pa]       //P_sat(Water,T=T_1) water vapor saturation pressure at ambient dry bulb
P_sat_2 =   2.811e3[Pa]       // P_sat(Water, T=T_2)  water vapor saturation pressure at comfort T

omega_1 = (0.066 *P_sat_1 *  relative_humidity_1)/ ((P_amb)-  (P_sat_1 *  relative_humidity_1)       )
omega_2 = (0.066 *P_sat_2*  relative_humidity_2)/ ((P_amb)-  (P_sat_2*  relative_humidity_2)      )

h_g1=  2560e3  [J/kg]          //enthalpy(Water,T=T_1, x=1)
h_g2=  2543e3 [J/kg]           // Enthalpy (Water, T= T_2 , x=1)

delta_h_AHU = cp_air_AHU * ( T_1- T_2) + (   ( h_g1*omega_1)-  (h_g2*omega_2)   )
 //m_dot_air =(Q_c_dot )/ ( delta_h_AHU) // where delta_h_AHU = h_1-h_2
// so the mass flow rate Required for AHU is 0.15 kg/s

// the Discharing Cycle in which water with 20 % Glycol is used as heat transfer fluid (HTF) for the heat storage tank
// for refrigeration systems, Typical Temp for  T_HTF_out which the temp of the HTF discharged from PCM-TES into AHU is (
3 to 5 C)
// for refrigeration systems, Typical Temp for  T_HTF_in which the temp of the HTF going into PCM-TES from AHU is (11 to
13 C)
//  We  fix T_HTF_out and T_HTF_in  at the average of their typical ranges, ususally those values are optimized for annual
operation cycle
// The melting temp of the choosen PCM must be greater than the freezing point of the HTF( GLYC)
T_fp_HTF=     -0.07306[C]          // freezingpt(GLYC,C=0.2)  the heat transfer flluid  20 % Glycol
cp_HTF = 4191 [J/kg*C]               //cp(GLYC,T=T_HTF_avg,C=0.2)
T_HTF_out =  (3+5)/2
T_HTF_in  = (11+13)/2
T_HTF_avg = ( T_HTF_out+ T_HTF_in )/2
k_HTF=0.5759 [W/m-C]
Pr_HTF =        10.17//prandtl(GLYC,T=T_HTF_avg,C=0.2)
mu_HTF=0.001398[Pa-s]
rho_HTF =    1000        //Density(GLYC,T=T_HTF_avg,C=0.2)
// First: Ice
T_melting_ice = 0 [C]  // to be adjusted for another pcm
h_l_ice = 336e3 [J/kg]// to be adjusted for another pcm
rho_ice = 917 [kg/m^3]// to be adjusted for another pcm
k_ice = 2.22 [W/m-C]// to be adjusted for another pcm
```

```
//m_dot_HTF = 0.1 [kg/s] // the mass flow rate of heat taransfer fluid will varied for optimal studies
 Q_st_J = 87984e3 [J]   // Q_st * 3600
V_ice = Q_st_J /(rho_ice *h_l_ice )  //
A_st = 6* pi# *(V_ice /(2*pi# )  )^(2/3)  // It should be mentioned that the shape of ice storage tank wasassumed to be cylindrical (with diameter equal to the height) inorder to minimize the heat leakage rate
R_insulated = 1980e-3 [m^2*C/W]
t_dc = 11 [h]
t_ch= 13[h]
t_dc = Q_leakage_dc *R_insulated/ (( T_1- T_HTF_out)*A_st)
t_ch = Q_leakage_ch  *R_insulated/ (( T_1- T_melting_ice)*A_st)

//Design of heat the Heat Exchanger useg effectivness_NTU method -Analisys-Based Design
//epsilon = (T_HTF_in - T_HTF_out)/(T_HTF_in-T_melting_ice)
//epsilon = 1- exp(-NTU_ice)
 //epsilon = 1-exp(-0.0199*(A_ice/m_dot_HTF))

N = 150// 20, 50, 150
D_o =  6.4e-3    [m]      // Typical range , from 6.40 mm to 101.6 mm
tube_t =      9.2e-4     [m]//thickness , from 0.91 mm to 1.63 mm
D_i = D_o-tube_t
p_pitch=  max( (1.25e-3)*D_o ,(D_o + 6.4e-3[m]) ) //ch, ( aka r_max ), assume square pitch arrangement
k_wall =  398 [W/m-C]          // copper
 L_ice_i =0.1  // I'm using the iterative method, initiating a value of L_ice and then resolve everthing with the correct value

R_t_ice =  R_HTF + R_wall+ R_ice
R_HTF = 1/ ( D_i *pi#*h_f_ice*L_ice_i )
R_wall = ln(D_o/D_i)/ (2*pi#*k_wall*L_ice_i)
 R_ice = (    ln(  ( pc_f*(  (p_pitch*0.5)^(2)  -   (D_o*0.5)^(2)  ) +  (D_o*0.5 )^(2)     )^(0.5)              /     (D_o*0.5)              )       )
          /(  2*pi#*k_ice*L_ice_i     )          // pc_f is the The phase change fraction  for a tube rounded by a cylindrical volume of PCM
 pc_f =   0.90

A_cross_tube = 0.25*pi#*(D_i^2)*N
Re_f = m_dot_HTF*D_i/ (A_cross_tube *mu_HTF)
h_f_ice  = Nusselt_f_ice *k_HTF / D_i
Nusselt_f_ice = 3.66 + (0.0668 * (D_i / L_ice_i) * Pr_HTF * Re_f) / (1 + 0.04 * ((D_i / L_ice_i) * Re_f * Pr_HTF)^(2/3))
U= 1/( R_t_ice*D_o*pi#*L_ice_i)



Delta_T_1 = T_HTF_in - T_melting_ice ;
Delta_T_2 = T_HTF_out - T_melting_ice;
LMTD = (Delta_T_1- Delta_T_2)/ ln(Delta_T_1/Delta_T_2 );
Q_st_dot = Q_st/ time_dc
Q_st_dot =  U* D_o*pi#*L_ice_new *N*LMTD
A_ice_i= N*pi#*D_o *L_ice_new



// using L_ice_new

R_t_ice_new=  R_HTF_new + R_wall_new+ R_ice_new
R_HTF_new = 1/ ( D_i *pi#*h_f_ice*L_ice_new  )
R_wall_new = ln(D_o/D_i)/ (2*pi#*k_wall*L_ice_new )
 R_ice_new = (    ln(  ( pc_f*(  (p_pitch*0.5)^(2)  -   (D_o*0.5)^(2)  ) +  (D_o*0.5 )^(2)     )^(0.5)              /     (D_o*0.5)              )
    )             /(  2*pi#*k_ice*L_ice_new     )          // pc_f is the The phase change fraction  for a tube rounded by a cylindrical volume of PCM



h_f_ice_new  = Nusselt_f_ice_new *k_HTF / D_i
Nusselt_f_ice_new= 3.66 + (0.0668 * (D_i / L_ice_new ) * Pr_HTF * Re_f) / (1 + 0.04 * ((D_i /L_ice_new ) * Re_f * Pr_HTF)^(2/3))
U_new= 1/( R_t_ice_new*D_o*pi#*L_ice_new)
```

```
Q_st_dot =  U_new* D_o*pi#*L_ice *N*LMTD
A_ice= N*pi#*D_o *L_ice


L_plus_ice = L_ice/ (D_i*Re_f)
f_avg_ice = (4/Re_f)* (   (3.44/sqrt(L_plus_ice))      +  ( (  (1.25/(4*L_plus_ice))+(64/4)- (3.44/sqrt(L_plus_ice) )   )/( 1+(
0.00021/L_plus_ice^2) )      )      )


//f_fd =  1/ (0.79*ln(Re_f)-1.64)^2
//f= f_fd*(1+(D_i/L_ice)^0.7)
P_drop =  f_avg_ice*(L_ice/D_i)*(m_dot_HTF^(2))/(2*rho_HTF*A_cross_tube^(2))
C_B =16648.3 *A_ice^(0.6123)


// second : pcm_1 ; Potassium Flouride


T_melting_pcm_1 = 5 [C]  // to be adjusted for another pcm
h_l_pcm_1 = 231e3 [J/kg]// to be adjusted for another pcm
rho_pcm_1 = 1445[kg/m^3]// to be adjusted for another pcm
k_pcm_1 = 0.584[W/m-C]// to be adjusted for another pcm


V_pcm_1 = Q_st_J /(rho_pcm_1 *h_l_pcm_1 )  //
A_st_pcm_1 = 6* pi# *(V_pcm_1 /(2*pi# )  )^(2/3)  // It should be mentioned that the shape of ice storage tank wasassumed to
be cylindrical (with diameter equal to the height) inorder to minimize the heat leakage rate


t_dc = Q_leakage_dc_pcm_1 *R_insulated/ (( T_1- T_HTF_out_pcm_1)*A_st_pcm_1)
t_ch = Q_leakage_ch_pcm_1  *R_insulated/ (( T_1- T_melting_pcm_1)*A_st_pcm_1)


 L_pcm_1_i =0.1  // I'm using the iterative method, initiating a value of L_ice and then resolve everthing with the correct value

R_t_pcm_1 =  R_HTF_pcm_1 + R_wall_pcm_1+ R_pcm_1
R_HTF_pcm_1 = 1/ ( D_i *pi#*h_f_pcm_1*L_pcm_1_i )
R_wall_pcm_1 = ln(D_o/D_i)/ (2*pi#*k_wall*L_pcm_1_i)
 R_pcm_1 = (    ln(  ( pc_f*(  (p_pitch*0.5)^(2)  -   (D_o*0.5)^(2)  ) +  (D_o*0.5 )^(2)     )^(0.5)              /     (D_o*0.5)            )
  )            /(  2*pi#*k_pcm_1*L_pcm_1_i     )          // pc_f is the The phase change fraction  for a tube rounded by a
cylindrical volume of PCM


h_f_pcm_1  = Nusselt_f_pcm_1 *k_HTF / D_i
Nusselt_f_pcm_1 = 3.66 + (0.0668 * (D_i / L_pcm_1_i) * Pr_HTF * Re_f) / (1 + 0.04 * ((D_i / L_pcm_1_i) * Re_f * Pr_HTF)^(2/
3))
U_pcm_1= 1/( R_t_pcm_1*D_o*pi#*L_pcm_1_i)

 T_HTF_in_pcm_1= (18+23)/2
T_HTF_out_pcm_1 = (10+15)/2

Delta_T_1_pcm_1 = (T_HTF_in_pcm_1 -T_melting_pcm_1) ;
Delta_T_2_pcm_1 = (T_HTF_out_pcm_1- T_melting_pcm_1);
LMTD_pcm_1 = (Delta_T_1_pcm_1- Delta_T_2_pcm_1)/ ln(Delta_T_1_pcm_1/Delta_T_2_pcm_1 );

Q_st_dot =  U_pcm_1* D_o*pi#*L_pcm_1_new *N*LMTD_pcm_1
A_pcm_1_i= N*pi#*D_o *L_pcm_1_new
```



```
// using L_pcm_1_new

R_t_pcm_1_new=  R_HTF_pcm_1_new + R_wall_pcm_1_new+ R_pcm_1_new
R_HTF_pcm_1_new = 1/ ( D_i *pi#*h_f_pcm_1*L_pcm_1_new  )
R_wall_pcm_1_new = ln(D_o/D_i)/ (2*pi#*k_wall*L_pcm_1_new )
 R_pcm_1_new = (    ln(  ( pc_f*(  (p_pitch*0.5)^(2)  -   (D_o*0.5)^(2)  ) +  (D_o*0.5 )^(2)     )^(0.5)              /     (D_o*0.5)
  )      )              /(  2*pi#*k_pcm_1*L_pcm_1_new     )          // pc_f is the The phase change fraction  for a tube rounded by a
cylindrical volume of PCM


h_f_i_pcm_1_new  = Nusselt_f_pcm_1_new *k_HTF / D_i
Nusselt_f_pcm_1_new= 3.66 + (0.0668 * (D_i / L_pcm_1_new ) * Pr_HTF * Re_f) / (1 + 0.04 * ((D_i /L_pcm_1_new ) * Re_f *
Pr_HTF)^(2/3))
U_pcm_1_new= 1/( R_t_ice_new*D_o*pi#*L_ice_new)



Q_st_dot =  U_pcm_1_new* D_o*pi#*L_pcm_1 *N*LMTD_pcm_1
A_pcm_1= N*pi#*D_o *L_pcm_1

L_plus_pcm_1 = L_pcm_1/ (D_i*Re_f)
 f_avg_pcm_1 = (4/Re_f)* (   (3.44/sqrt(L_plus_pcm_1))      +  ( (  (1.25/(4*L_plus_pcm_1))+(64/4)- (3.44/sqrt(L_plus_pcm_1)
)   )/( 1+(0.00021/L_plus_pcm_1^2) )      )      )

//f_fd_pcm_1 =  1/ (0.79*ln(Re_f)-1.64)^2
//f_pcm_1= f_fd_pcm_1*(1+(D_i/L_pcm_1)^0.7)
P_drop_pcm_1 =  f_avg_pcm_1*(L_pcm_1/D_i)*(m_dot_HTF^(2))/(2*rho_HTF*A_cross_tube^(2))
C_B_pcm_1 =16648.3 *A_pcm_1^(0.6123)
```

Appendix (B) : MATLAB code

Script(1):

```
function c_total = calculate_cost(x, filename)

    opts = detectImportOptions(filename, 'VariableNamingRule', 'preserve');
    dataTable = readtable(filename, opts);
    % descion Variables : input vector
    L_ice = x(1);
    N = x(2);
    m_dot_HTF = x(3); % Mass flow rate of HTF
    T_HTF_in = x(4); % Inlet temperature of HTF
    T_HTF_out = x(5); % Outlet temperature of HTF
    D_o = x(6); % Outer diameter of the tube
    tube_t = x(7); % Tube thickness
  T_storing = x(8); % Eighth decision variable
    T_c_b = x(9);     % Ninth decision variable

% The discharging cycle:
% The following may be used to design AHS in future
P_sat_1 = 4.951e3; % [Pa] this is the staurated pressure at T_1
P_sat_2 = 2.811e3; % [Pa] this is the saturated pressure at T_2
relative_humidity_1 = 0.7;
relative_humidity_2 = 0.55;
P_amb = 100e3; % Pa
T_1 = 32.7; % [Celsius]  % this is the max ambient temp in the hottest day in Melbourne
Fl
T_2 = 23; % Celsius
cp_air_AHU = 1005; % J/kg*C
h_g1 = 2560e3; % J/kg
h_g2 = 2543e3; % J/kg
% the cooling load of the office building in whuch the integrated pcm-TES
% is used for
Q_c_dot = 2e3; % Watts %the peak cooling load required,  in futuure I can design my
system over the daily profile of the cooling load
% Operating hours:
time_dc = 11  %[h] Cooling load for 11 hours, working hours from 7 AM to 6 PM, assuming
the system is idle in the rest of the day
Q_c = Q_c_dot *time_dc  % taragted cooling capacity in kWh
eta_st = 0.9 % the storage efficiency
% the required storage capacity :
Q_st = Q_c / eta_st     %in kWh
Q_st_dot = Q_st/time_dc  % in kW
% Calculations for AHS that may be used for future calculations
omega_1 = (0.066 * P_sat_1 * relative_humidity_1) / (P_amb - (P_sat_1 *
relative_humidity_1));
omega_2 = (0.066 * P_sat_2 * relative_humidity_2) / (P_amb - (P_sat_2 *
relative_humidity_2));
delta_h_AHU = cp_air_AHU * (T_1 - T_2) + ((h_g1 * omega_1) - (h_g2 * omega_2));
m_dot_air = Q_c_dot / delta_h_AHU; % this is the required mass flow rate of the air
inside teh AHU

% Discharging Cycle- Heat transfer fluid used to facilate the heat
% transfer between the pcm inside the tank and the discharing process is
```

```
% mixture of glycol-water mixture with 20 % concentration SSpecification of
% specifications regarding the HTF

T_fp_HTF = -0.07306; % Celsius
cp_HTF = 4191; % J/kg*C
T_HTF_out = (3 + 5) / 2;  % [C]this is initiated value and it will be optimzed later
T_HTF_in = (11 + 13) / 2;  %[C] this is initiated value and it will be optimized later
T_HTF_avg = (T_HTF_out + T_HTF_in) / 2; % [C]assume constant fluid proprties at this Temp↙
value regardless of teh adjusmenrts that the optz will make
k_HTF = 0.5759; % W/m-C
Pr_HTF = 10.17;
mu_HTF = 0.001398; % Pa-s
rho_HTF = 1000 %kg/m^3

% PCM  Properties(ice)
T_melting_ice = 0; % Celsius
h_l_ice = 336e3; % J/kg    % latent heat
rho_ice = 917; % kg/m^3
k_ice = 2.22; % W/m-C
m_dot_HTF = 0.1; % [kg/s ] , this is the initiated value of the mass flow rate of HTF, it↙
will be optimzed
Q_st_J = 87984e3; % J      % stoarge capacity required in J
V_ice = Q_st_J / (rho_ice * h_l_ice);
A_st = 6 * pi * (V_ice / (2 * pi) )^(2/3);  % teh shape of the storage is cylnder with↙
height equals to teh Diamter for better heat transfer raesons as concluded from the↙
litrature review
R_insulated = 1980e-3; % m^2*C/W  ., this typical value for thermal insulation of the↙
thermal  stoarge tank
t_dc = 11; % [h] discharing time
t_ch = 13; % [h] charging time

Q_leakage_dc=  t_dc * ((T_1 - T_HTF_out) * A_st)/R_insulated ; % estimated leakage of the↙
tank dueing discharing process
Q_leakage_ch=t_ch *((T_1 - T_melting_ice) * A_st)/ R_insulated;% estimated leakage of the↙
tank dueing charing process

% the pcm thermal stoarge can be treated as a heat exchanger and design
% for its tubing system length
% using the LMTD method to design for the length of the tubes carrying HTF
% inside the tank,

D_o = 6.4e-3; %[ m ] , outer tube diamter, initiated value,     it's going to be optimzed
tube_t = 9.2e-4; % [m] , thickness of the tube , initiated value, it's going to be↙
optimized
D_i = D_o - tube_t;   % [m] inner tube diamter
p_pitch = max((1.25e-3) * D_o, (D_o + 6.4e-3)); % m this repersnts r_max , which is tehg↙
maximum distance taht teh PCM will soldifies
k_wall = 15 ; % [W/m-C] stainless steel is choosen for the material of tube, it's not↙
typiacl but it's the only materail that I founf cost correlatuoions for it
L_ice = 3   % this an initiated value of tube length carrying the HTF required inside the↙
pcm-stoarge tank , will be optimized based on cost and Pressure drop
```

```
N=60        % number of tube requid, will be optimzzed based on cost and pressure drop

% Calculate resistances for PCM-HEX
A_cross_tube = 0.25 * pi * (D_i^2) * N;
Re_f = m_dot_HTF * D_i / (A_cross_tube * mu_HTF); % where Re_f is teh Rynold of HTF ,
depnds on mass flow rate
L_plus = L_ice/ (D_i*Re_f);
f_fd = 1/( (0.72*log(Re_f))-1.64           )^2;
if Re_f <= 2300
    Nusselt_f_ice = 3.66 + (0.0668 * (D_i / L_ice) * Pr_HTF * Re_f) / (1 + 0.04 * ((D_i /
L_ice) * Re_f * Pr_HTF)^(2/3));
else
    Nusselt_f_ice = 0.023 * Re_f^0.8 * Pr_HTF^0.4;
end
if Re_f <= 2300
f_avg = (4/Re_f)* (    (3.44/sqrt(L_plus))       +  ( (  (1.25/(4*L_plus))+(64/4)- (3.44
/sqrt(L_plus) )    )/( 1+(0.00021/L_plus^2) )       )       );
else
   f_avg= f_fd*(1+(D_i/L_ice)^0.7);
end
 h_f_ice  = Nusselt_f_ice *k_HTF / D_i ; % this is the heat trasnfer coeff in the HTF
side

% Thermal resistances:

R_HTF = 1 / (D_i * pi * h_f_ice * L_ice);% [C/W] thermal resistance due to convection of
HTF inside the tubes
R_wall = log(D_o / D_i) / (2 * pi * k_wall * L_ice);  % [C/W] conductive thermal
resistance
pc_f_values = 0.005:0.01:0.995; % Range of pc_f values, which is typically an alternative
for time that takes pcm to completely soldifes and reach r_max, this will alter the value
of thermal resistance of teh PCM
sum_R_ice = 0; % initiated
% Loop through pc_f values
for pc_f = pc_f_values

R_ice = (log((pc_f * ((p_pitch * 0.5)^(2) - (D_o * 0.5)^(2)) + (D_o * 0.5)^(2))^0.5 /
(D_o * 0.5))) / (2 * pi * k_ice * L_ice);
sum_R_ice = R_ice - sum_R_ice
end
R_t_ice =  R_HTF + R_wall+ sum_R_ice; % total thermal resistance
Delta_T_1 = T_HTF_in - T_melting_ice ;
Delta_T_2 = T_HTF_out - T_melting_ice;
LMTD = (Delta_T_1- Delta_T_2)/ log(Delta_T_1/Delta_T_2 );
U = 1 / (R_t_ice * D_o * pi * L_ice);
Q_st_dot = (U*pi*D_o* L_ice *LMTD *N);
A_ice = N *pi*D_o *L_ice ;
L_ice =A_ice/N *pi*D_o   % required length of HTF tubes
%% the following tank diminsioning is to set up a latter constraint:
V_tank = V_ice/(1-0.20)% allow an additional 20% for the pipes and clearance
height = ( V_tank /( pi*0.25) )^(1/3) % height of the tank which is equal to its diamter
```

```
V_tube = N*pi*0.25*(D_o^2)*L_ice
V_ice_ava= V_tank-V_tube

%cost calculations of the tubing system inside the tank:
F_m =  2.70 + (10.7639*A_ice)^(0.07) % material cost factor, for stainless steal
F_p = F_m % low gauge pressure , pressure gage pressure factor


%FL is a cost factor depends on  tube length, gioven for differnt tube
%lengths
tubeLengths = [0.1;0.4;1.6;2.4384; 3.6576; 4.8768; 6.0960];
FLValues = [1.6;1.45;1.3;1.25; 1.12; 1.05; 1.00];

% New tube lengths for interpolation
newTubeLengths = [L_ice];

% Initialize array to store interpolated FL values
interpolatedFL = zeros(size(newTubeLengths));

%  interpolation:
for i = 1:length(newTubeLengths)
    % Finding the two closest data points for interpolation:
    [~, idx] = min(abs(tubeLengths - newTubeLengths(i)));

    % Linear interpolation formula % I got this code structure online% I
    % checked its ability for interpolation ina seperate script.
    interpolatedFL(i) = FLValues(idx) + ...
                        (newTubeLengths(i) - tubeLengths(idx)) / ...
                        (tubeLengths(idx + 1) - tubeLengths(idx)) * ...
                        (FLValues(idx + 1) - FLValues(idx));
end
% now calcualting the pressure drop for pumping cost:
P_drop = f_avg*(L_ice/D_i)*(m_dot_HTF^(2))/(2*rho_HTF*A_cross_tube^(2));
V_rate = m_dot_HTF/rho_HTF; % this is teh volumtric rate
eta_pump = 0.8 % efficiancy of the pump, ignore other etas for pump
W_pumping_inlet  = P_drop*V_rate/eta_pump
C_pumping = 705.48* (W_pumping_inlet^(0.71))* (1+ (0.2/(1-eta_pump)));%pumping cost;   %↙
storage unit cost
C_st = 8.67 *10^(2.911*exp(0.1416*log(V_ice))) % cost of the storage tank unit

%Design of the components of the charing cycle:
%stadrized given dimensions of the plate HEX:
W_e = 0.253 ; %[m] width of the plate
L_p = 0.456 ;%[m]
t_p = 0.002 ;%[m] // Plate thickness
beta_chovron = 0.785 ; % // Chevron angle
b = 0.002     ;   %[m]
pco = 0.004   ;   %[m] // corrugation pitch
X_x= (b*pi)/ pco ; % // Wavenumber
phi = (1/6) * ( 1+ (1+(X_x^2) )^(0.5)+ 4*(1+( (X_x^2)/2) )^(0.5) );  %// Arae Increase↙
Factor
```

```
D_h = (2*b)/ phi ; %[m]
d_eq = 2*b  ;     %// eqivalint diameter
k_wall = 15;% [W/m-K]  %//conductivity of stainless steel % note that I forgot that I
used copper in the EES code for teh paramteric studies




% Given Temps
T_c_b = -20 ;    % this will be optimized  T_c_b is the saturated evaporator temp and it
varies from -30 to T_storing_Temp
T_wb= 26.6          % this is the wet bulp temp opteined from the design cheet of A/C
system in Melboure Fl
T_c_in_cond = 35 %The temp of the cooling water coming from the cooling tower to the
condenser
T_c_out_cond = 40.56   % the temp the cooling water coming from out teh condenser
T_coooling_tower_df =T_c_out_cond-T_c_in_cond % a temp differnce of 5.56 is typical for
cooling tower
T_h_cond = T_c_out_cond+5; % The saturated temp of condenser,T_h_cond must be kept above
T_wetbulp and making sure does not exceed 60 C
Temp1= T_h_cond;   % Temp1, calling the proprties of R134a at Temp_1, this value is fixed
and not going to be optz for now
Temp2 = T_c_b;      % Temp2, calling the proprties of R134 at Temp_2,  this valu will be
varied, optimized

% proprties of R-134a at saturated Temp for evaporator % I added an excel sheet to work
director, and did thedata extracting function and introplation in other scripts

propertiesAtTemp2 = getPropertiesForTemp(Temp2 , dataTable);
cp_h_evp = propertiesAtTemp2.cpLiquid;
k_h_evp = propertiesAtTemp2.kLiquid;
pr_h_evp = propertiesAtTemp2.PrLiquid;
mu_h_evp = propertiesAtTemp2.muLiquid;
rho_h_evp_L = propertiesAtTemp2.DensityLiquid;
rho_h_evp_V = propertiesAtTemp2.DensityVapor;
h_f_evp = propertiesAtTemp2.hf;
h_g_evp = propertiesAtTemp2.hg;
P_evap = propertiesAtTemp2.Pressure

% proprties of R-134a at saturated Temp for condenser
propertiesAtTemp1 = getPropertiesForTemp(Temp1 , dataTable);
cp_h_c = propertiesAtTemp1.cpLiquid;
k_h_c = propertiesAtTemp1.kLiquid;
pr_h_c = propertiesAtTemp1.PrLiquid;
mu_h_c = propertiesAtTemp1.muLiquid;
rho_h_c_L = propertiesAtTemp1.DensityLiquid;
rho_h_c_V = propertiesAtTemp1.DensityVapor;
h_f_c = propertiesAtTemp1.hf;
h_g_c= propertiesAtTemp1.hg;
```

```
P_cond = propertiesAtTemp1.Pressure

% storing the proprties in the following variables for evaporator:
cp_c_b =cp_h_evp ;
k_c_b = k_h_evp ;
pr_c_b = pr_h_evp;
mu_c_b = mu_h_evp;
rho_c_b_L =   rho_h_evp_L ;      %density(R-134a,P=P_c_b, x=0)
rho_c_b_V = rho_h_evp_V  ;       %density(R-134a,P=P_c_b, x=1)
h_in_evap = h_f_evp      ;
h__out_evap = h_g_evp;

% Design the evaporator:

Q_evap_dot = Q_st_dot / t_ch ; %W
m_dot_c_b = Q_evap_dot  /(h__out_evap - h_in_evap )   ; % [kg/s], the mass flow rate in
teh charing cycle
G_c_b= m_dot_c_b/(b*W_e) ;
x_c_b =0.75      ;  %vapor quality
G_c_b_eq = G_c_b*( (1-x_c_b)+( x_c_b *(rho_c_b_L/rho_c_b_V)^(0.5) ) );
Re_c_b_eq = (G_c_b_eq*D_h)/(mu_c_b);

%Evaporating Heat Transfer Coeff:
h_c_b = 5.323*(k_c_b/D_h)*Re_c_b_eq^(0.42)*pr_c_b^(1/3) ;
%heat tranfer area of (Evaporator)
h_h_b = sum_R_ice/ (2*pi*L_ice) %haet transfer coefficient due to soldification of the
pcm
T_storing  = 0; %[C]  % initiated ,, will be optimzed , perfectly ranges from melting
point of the pcm  to -10 C

T_h_in_b = T_HTF_in ; % assume that the chiller operates when ice temp is equal to
T_HTF_in( when the ice temp equals to the temp of the glycol going into storage in teh
discharing cycle
T_h_out_b  = T_storing ;  % the goal is get te tank into stoarge tempeature which
typicall vary from -10 to T_melting, this will be optimized
delta1_b = T_h_in_b- T_c_b;
delta2_b = T_h_out_b - T_c_b;
LMTD_b = (delta1_b-delta2_b)/log(delta1_b/delta2_b);
U_b = 1/ ((1/h_h_b)+(1/h_c_b)+ (t_p/k_wall) );
A_b = Q_evap_dot/(LMTD_b*U_b);
%Number of palte requied
 N_p_b = ( A_b/(L_p*W_e) +2 ) ;

%Design of condenser,
cp_h_cond = cp_h_c    %cp(R134a,T=T_h_cond,P=P_h_cond)
k_h_cond =  k_h_c     %conductivity(R134a,T=T_h_cond,P=P_h_cond)
pr_h_cond =  pr_h_c    % prandtl(R134a,T=T_h_cond,P=P_h_cond)
mu_h_cond =  mu_h_c    %viscosity(R134a,T=T_h_cond,P=P_h_cond)
rho_h_cond_L = rho_h_c_L   %density(R134a,P=P_h_cond, x=0)
rho_h_cond_V = rho_h_c_V   %density(R134a,P=P_h_cond, x=1)
m_dot_h_cond = m_dot_c_b   %
```

```
Q_cond_dot = m_dot_h_cond *(h_g_c-h_f_c);
G_h_cond= m_dot_h_cond/(b*W_e);
x_h_cond =0.75 %vapor quality
G_h_cond_eq = G_h_cond*( (1-x_h_cond)+( x_h_cond *(rho_h_cond_L/rho_h_cond_V)^(0.5) ) );
Re_h_cond_eq = (G_c_b_eq*D_h)/(mu_h_cond);
% Condensing Heat Transfer Coeff:
h_h_cond = 4.118*(k_h_cond/D_h)*Re_h_cond_eq^(0.4)*pr_h_cond^(1/3);

% proprties of cooling water at 48.28 C (average Tof water entering and
% leaving the condenser) % becuase I didn't take HVAC , I just knew as I'm conducting my↙
literature review it is
% not necessary to use cooling tower, it becomes necessary for larger scale
% HVAC systems
cp_c_cond = 4180
k_c_cond = 0.63
pr_c_cond = 4.3
mu_c_cond = 0.55e-3
rho_c_cond = 988
m_dot_cooling_tower = 43.2e-3*Q_cond_dot
G_c_cond = m_dot_cooling_tower/(b*W_e)
Re_c_cond = (G_c_cond *d_eq)/ ( mu_c_cond)
f_c_cond= ( (1.82*log(Re_c_cond) ) - 1.64)^(-2)
Nusselt_c_cond = ( (f_c_cond/ 8)* (Re_c_cond-1) *pr_c_cond ) / ( ( 12.7*(f_c_cond/8)^↙
(0.5) *(pr_c_cond^(2/3) -1) ) +1.07 )
h_c_cond= (Nusselt_c_cond*k_c_cond)/(d_eq)

% heat tranfer area of (Condenser)

delta1_cond = T_h_cond - T_c_out_cond ;
delta2_cond = T_h_cond- T_c_in_cond;
LMTD_cond = (delta1_cond-delta2_cond)/log(delta1_cond/delta2_cond);
U_cond = 1 / ((1/h_h_cond)+(1/h_c_cond)+ (t_p/k_wall) );
A_cond = Q_cond_dot/(LMTD_cond*U_cond);
%  Number of palte required for zone a
N_p_cond = ( A_cond/(L_p*W_e) +2 )

% requied compressor power
W_dot_compressor = m_dot_h_cond *(h_g_c-h_g_evp)
COP= Q_evap_dot/W_dot_compressor

%cost calculations for the charing cycle(Capital cost)
% Evaporator:
C_evap = 16648.3 *A_b^(0.6123);
%Condenser:
 C_cond = (516.621*A_cond)+268.45;
 %Compressor :
 C_comp =( (39*m_dot_h_cond)/(0.9-0.6))*( P_cond/P_evap)*log(P_cond/P_evap);
% Expansion valve:
C_ExpV = 114.5*m_dot_h_cond;
%cooling twoerL:
C_cooling_tower = 746.749 *(m_dot_cooling_tower^0.79)*(T_coooling_tower_df^0.57)*↙
```

```
((T_c_out_cond- T_wb)^(-0.9924))*( (0.022*T_wb)  +0.39  )^(2.447);
% electricity cost, note that, for now I ignored the electric consumption
% cost of the pump in the charing cycle, fans in the AHS and Cooling Tower.
c_elc_off_peak=  0.09 ;  % [$/kWh] off-peak rating for Melbourne FL
c_elc_on_peak = 0.14 ;% [$/kWh] on-peak rating for Melbourne Fl
C_elect = (W_dot_compressor*(c_elc_off_peak/3600)) +(   W_pumping_inlet  *
(c_elc_on_peak/3600)    )

N_on= 11*5*4*12; % number of operating(discharing) cycles (hours) for a 5 days a weak in
12 months
N_off = 13*5*4*12; % number of charing cycles (hours) for a 5 days a weak in 12 months

% The penalty cost of CO2 emission (CO2 )
E_cons_A = (W_dot_compressor*N_off)+ (  W_pumping_inlet*N_on)
emm_factor_co2 = 0.968 % [kg/kWh]this is  CO2 emission factor
mass_co2= emm_factor_co2*( E_cons_A)   % [kg] this the annual mass of co2 produced by the
system
c_co2 = 90 % The penalty cost of CO2 emission (cCO2 ) was considered as 90 USdollars per
ton of carbon dioxide emissions
 C_env =( (mass_co2/1000 )* c_co2 ) % the cost of co2 emission , this is a rough estimate
as I ignored many components on the system

 C_HEX = (516.621*A_ice)+268.45*F_p* F_m  * interpolatedFL(i) %cost of tank HEX

C_investment= C_HEX +C_pumping +C_evap + C_cond + C_comp + C_ExpV + C_cooling_tower+C_st
c_total = C_investment +C_elect + C_env % the  main objective function to be minimized,
includes enviromental aspects
% main results for discharing cycle:

disp(['Pumping Cost: ', num2str(C_pumping)]);
disp(['Storage Unit Cost: ', num2str(C_st)]);
disp(['Tank Heat Exchanger Cost: ', num2str(C_HEX)]);
disp(['mass flow rate of HTF: ', num2str(m_dot_HTF)]);
disp(['Volume of ice : ', num2str(V_ice)]);
disp(['Heat Transfer Area : ', num2str(A_ice)]);
disp(['Length required for tank tubes  based on cost: ', num2str(L_ice)]);
disp(['Number of tube in Tnak driven by cost criteria: ', num2str(N)]);
disp(['Outer Diamter: ', num2str(D_o)]);
disp(['HTF Temp into the Tank: ', num2str(T_HTF_in)]);
disp(['HTF Temp out from the Tank: ', num2str(T_HTF_out)]);
% main results for charing cycle:
disp(['Capital cost of condenser: ', num2str(C_cond)]);
disp(['Capital cost of Evaporator : ', num2str(C_evap)]);
disp(['Capital cost of compressor: ', num2str(C_comp)]);
disp(['Capital cooling tower: ', num2str(C_cooling_tower)]);
disp(['Capital cost of Expnasion Valve : ', num2str(C_ExpV)]);
disp(['Heat transfer area of the condenser : ', num2str(A_cond)]);
disp(['Heat transfer area of the evaporator : ', num2str(A_b)]);
disp(['Number of plates for the condenser : ', num2str(N_p_cond)]);
disp(['Number of plates for the Evaporator  : ', num2str(N_p_b)]);
disp(['cost of electricity: ', num2str(C_elect)])
```



```
disp(['cost of investment: ', num2str(C_investment)]);
disp(['cost of co2 emission: ', num2str(c_co2)]);
disp(['Total Cost: ', num2str(c_total)]);


end
```

Script (2):


```
close all
clear all
clc
filename = 'R134a_properties.xlsx';
fitnessFunction = @(x) calculate_cost(x, filename);


nvars = 9; % Number of variables

% The bounds for the decsion variables
LB =[0.1, 10, 0.05, 11, 3, 6.4e-3, 0.91e-3,-10,-30]; % Lower bounds
UB = [6.0960, 150, 0.2, 13, 5, 101.6e-3, 1.63e-3,0,0]; % Upper bounds

% Now Setting up the  options for the genetic algorithm
options = optimoptions('ga', 'Display', 'iter', 'UseParallel', true);



% Runing the genetic algorithm with custom constraints
[x, fval] = ga(fitnessFunction, nvars, [], [], [], [], LB, UB, @customConstraints,↙
options);

% Display the results
disp('Optimal Solution:');
disp(['L_ice: ', num2str(x(1)), ' N: ', num2str(x(2)), ' m_dot_HTF: ', num2str(x(3)), '↙
T_HTF_in: ', num2str(x(4)), ' T_HTF_out: ', num2str(x(5)), ' D_o: ', num2str(x(6)), '↙
tube_t: ', num2str(x(7)), ' T_storing: ', num2str(x(8)), ' T_c_b: ', num2str(x(9))]);
disp(['Minimum Cost: ', num2str(fval)]);

% Custom constraints function that accepts T_fp_HTF as a parameter
function [c, ceq] = customConstraints(x)
    % Extracting  the decision variables that are relevant for the constraints
    D_o = x(6);
    tube_t = x(7);
    T_storing = x(8);
    T_c_b = x(9);
    N= x(2);
     L_ice = x(1);
  h_l_ice =336e3 ;
 Q_st_J = 87984e3; % J, storage capacity required in J
    rho_ice = 917; % kg/m^3
    V_ice = Q_st_J / (rho_ice * h_l_ice); % m^3, actual ice volume
    V_tank = V_ice/(1-0.20); % m^3, total tank volume (includes 20% extra for pipes and↙
clearance)
    V_tube = N*pi*0.25*(D_o^2)*L_ice; % m^3, volume of tubes
    height = (V_tank/(pi*0.25))^(1/3); % m, height of the tank



    % Initialize the constraints array
```

```
    c = zeros(3, 1); % Now we have 5 constraints

    % constaraint (1):  D_o and its correspinng  tube_t

    if any(D_o == [6.4e-3, 9.5e-3, 15.9e-3])
        valid_tube_t = 0.91e-3;
    elseif any(D_o == [12.7e-3, 19.1e-3])
        valid_tube_t = 1.02e-3;
    elseif any(D_o == [22.2e-3, 25.4e-3, 28.6e-3, 31.8e-3, 34.9e-3, 38.1e-3, 41.3e-3,
44.5e-3])
        valid_tube_t = 1.22e-3;
    elseif any(D_o == [50.8e-3, 54e-3, 66.7e-3, 76.2e-3, 101.6e-3])
        valid_tube_t = 1.63e-3;
    else
        valid_tube_t = NaN; % Invalid combination
    end
    % Constraint is satisfied if tube_t matches the valid value
    c(1) = (tube_t ~= valid_tube_t);

   % constarint (2), the evaporator presuure anx T_storing :
    % c(2) will be < 0 if T_c_b < T_storing
    c(2) = T_c_b - T_storing;
    % c(3) will be < 0 if T_storing > 0
    c(3) = -T_storing ;  % where 1e-6 is the smallest value you consider safely above
zero

    % Geometric constraints for the tank , 4 and 5 % it's ignored for this
    % project , assume the dimentionsns of the tank is unknown
c(4) = V_ice - V_tank + V_tube; % V_tube <= (V_tank - V_ice)
   c(5) = L_ice - height; % L_ice < height

    ceq = [];

end
```

Appendix(C): Results for ICE

Pumping Cost: 0.55802 **Storage Unit Cost: 2374.9465** **Tank Heat Exchanger Cost: 8926.692** **mass flow rate of HTF: 0.1** **Volume of ice : 0.28556** **Heat Transfer Area : 3.6191** **Length required for tank tubes based on cost: 0.0012128** **Number of tube in Tnak driven by cost criteria: 60** **Outer Diamter: 0.0064** **HTF Temp into the Tank: 12**

HTF Temp out from the Tank: 4
Capital cost of condenser: 316.0568
Capital cost of Evaporator : 110265.569
Capital cost of compressor: 2.2333
Capital cooling tower: 2501.7359
Capital cost of Expnasion Valve : 0.10157
Heat transfer area of the condenser : 0.09215
Heat transfer area of the evaporator : 21.926
Number of plates for the condenser : 2.7988
Number of plates for the Evaporator : 192.0531
cost of electricity: 0.0083168
cost of investment: 124387.8931
cost of co2 emission: 9.0425e+04
Total Cost: 214812.641
3 29900 214813 0 2
Optimization terminated: average change in the fitness value less than options.FunctionTolerance
and constraint violation is less than options.ConstraintTolerance.
Optimal Solution:
L_ice: 0.10008 N: 10.2539 m_dot_HTF: 0.17376 T_HTF_in: 11.0002 T_HTF_out: 4.375
D_o: 0.0064 tube_t: 0.00091 T_storing: 0 T_c_b: -29.9954
Minimum Cost: 214812.641
IdleTimeout has been reached.

Appendix (D): results for Hydrated Salt

Pumping Cost: 0.55802
Storage Unit Cost: 2229.3302
Tank Heat Exchanger Cost: 7106.649
mass flow rate of HTF: 0.1
Volume of ice : 0.26359
Heat Transfer Area : 3.6191
Length required for tank tubes based on cost: 0.0012128
Number of tube in Tnak driven by cost criteria: 60
Outer Diamter: 0.0064
HTF Temp into the Tank: 20.5
HTF Temp out from the Tank: 12.5
Capital cost of condenser: 318.1176
Capital cost of Evaporator : 46811.6665
Capital cost of compressor: 2.3806
Capital cooling tower: 2631.2056
Capital cost of Expnasion Valve : 0.10826
Heat transfer area of the condenser : 0.096139

Heat transfer area of the evaporator : 5.411
Number of plates for the condenser : 2.8333
Number of plates for the Evaporator : 48.9018
cost of electricity: 0.0088653
cost of investment: 59100.0158
cost of co2 emission: 90
Total Cost: 155488.62
3 29900 155489 0 2
Optimization terminated: average change in the fitness value less than options.FunctionTolerance
and constraint violation is less than options.ConstraintTolerance.
Optimal Solution:
L_ice: 0.1 N: 20 m_dot_HTF: 0.2 T_HTF_in: 20.2812 T_HTF_out: 11.0285 D_o: 0.1016 tube_t: 0.00163 T_storing: 4.9946 T_c_b: -29.4578
Minimum Cost: 155488.62