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Investigating The Thermodynamic Characteristics of Energy Saving Refrigerating System with Cold Accumulator Under Tropical Conditions

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ABSTRACT

The increasing demand for efficient refrigeration in tropical regions highlights the importance of integrating thermal energy storage systems into conventional refrigeration units. This study presents a simulation-based investigation of a refrigeration system equipped with a cold accumulator, designed to enhance performance under tropical climatic conditions using R134a refrigerant. The cold accumulator serves as a thermal buffer, enabling load shifting, peak demand reduction, and improved energy efficiency. Using a thermodynamic simulation model, the system's cooling capacity, coefficient of performance (COP), and energy consumption were analyzed under variable tropical ambient temperatures ranging from 25 °C to 45 °C. The results demonstrate that the integration of the cold accumulator stabilizes evaporator temperature fluctuations, reduces compressor cycling frequency, and enhances COP by up to 8% compared to conventional systems without storage. Furthermore, the system exhibits potential for peak load management, allowing for partial charging during off-peak hours and discharging during peak demand.

Keywords: *Cold accumulator, thermal buffer, coefficient of performance, cooling capacity.*

I. INTRODUCTION

Poorly designed, improper usage, poor servicing or maintenance methods, badly or poorly recycling methods of refrigerant fluids with high global warming potential (GWP) leakage from abandoned refrigeration equipment at the end of their lifespan is regarded as part of "direct" contributions to greenhouse emission (Ikem, 2017). These emissions can be reduced by more efficient energy conversion, use of natural refrigerants with excellent GWP based on the general refrigerant scale, design of refrigeration machines for maximum efficiency by reducing the absorbed power of the refrigeration system (Ikem, A. Ikem, *et al.,* 2016). In addition to these, it is known that the overall energy consumption of refrigeration, air-conditioning, or heat pump system during its service life is a considerable cost factor and frequently is a multiple of the initial investment (Akintunde, 2008; Ikem *et al.,* 2017). International concern over relatively high (GWP) of some frequently used refrigerants has resulted in some European countries removing them from the cooling systems and sort for alternative refrigerant as replacement in domestic chillers. For this reason, they are canvasing for the production and use of these refrigerants to be terminated in the near future (Wongwises S, Chimres N, (2005), Bolaji BO, (2008)). Generally, refrigeration systems are designed for fixed capacity to achieve better cooling effect based on the maximum demand at the highest ambient temperature. The consequence is that the refrigeration system which delivers high cooling capacity, is selected to overcome the worst condition, and needs to be cycled on/off when normal conditions occur. This is best achieved when the system incorporates cold accumulator.

The vertical-tube evaporator and heat pipes have their own peculiarities in the course of heat and mass transfer processes, the boiling processes in the vertical-tube evaporator of heat pipes and the accumulation of cold during freezing on its surface can be modeled. (Hasan S., Ali S., 2011). One of the most frequently used refrigerants (R134a) in the refrigeration machines was used in this model to carry out the calculations using the vertical-tube evaporator in the accumulation of cold on the heat exchange surface while boiling the refrigerants within the tube (Jaroschek *et al.,* 2017). It is therefore necessary to model the accumulation process that can give optimum results. A heat exchanger tube shown in figure 1, was used in the modeling process.

II. MATERIALS AND METHODS

A. Model Equation

The mathematical description of the processes of the accumulator operation from heat pipes in cold accumulation is necessary. The schematic diagram of the heat exchange tube for the mathematical experiment is shown in Figure 1.

TABLE I
VALUES OF REFRIGERANT BOILING POINTS FOR DIFFERENT

Regime	**Boiling Point t_0 (°C)**
1st	-5
2nd	-10
3rd	-15

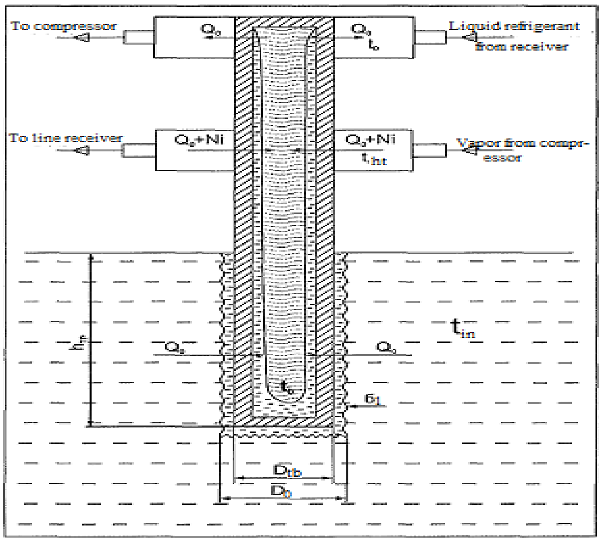

FIGURE 1: DESIGN DIAGRAM OF THE HEAT EXCHANGE TUBE

This is to know how much ice is possible to freeze on the heat exchange surface. For this purpose, a calculation is made for the following initial data:

Determining the quantity of ice that freeze on the heat exchange surface during the operation of the chiller is the key concern here. Initial conditions are set to achieve this aim. During the ice storage process, the following initial input data for the different regimes of the simulation are taken into account:

– freezing point:$t_{fr} = 0$ °C;
– melting point: $t_{def} = 0$ °C;
– refrigerant boiling point:

TABLE II.

ICE TEMPERATURE AT DIFFERENT REGIMES

Regime	**Boiling Point t_0 (°C$t_l = (t_0 + 3)$°C)**
1st	-2
2nd	-7
3rd	-12

Thickness of thawing layer: $\delta_{th} = 0.5 \text{ x } 10^{-3} m = 0.5mm.$

Refrigerating agent: - Freon 134a.

Regime	Melting temperature: t_{ht}, (°C)
1st	25
2nd	35
3rd	45

B Heat of Freezing Ice

Given that the ice formed is collected into the tank:

The amount of heat q_{fr}, required for the formation of 1 kg of ice at the initial temperature t_{in}, is determined by the formula:

$$q_{fr} = c_w.(t_{in} - t_{fr}) + r_w + c_l.(t_{fr} - t_l) \qquad (1)$$

Where,

c_w - heat capacity of water, kJ/kgK;

c_l - specific heat of ice, kJ/kgK;

r_w- heat of ice formation, kJ/kgK;

t_{in}-initial temperature of ice formation,°C; $c_w = 4197$ kJ/kgK; $c_l = 2120$ kJ/kgK;

$r_w = 334000$ kJ/kg; $t_{in} = 5$°C.

These compositions are used to determine the freezing and thawing time of ice.

C Freezing Perion of Ice

$$\tau_{fr} = \frac{q_{fr} \cdot \rho_l}{t_{fr} + |t_0|} \cdot \left\{ \frac{\delta_l^2}{2.\lambda_l} + \frac{\delta_{th}}{\alpha_a} \right\} \qquad (2)$$

where, ρ_l ice density, kg/m^3;

λ_l - thermal conductivity of ice, W/mK;

λ_w - thermal conductivity of water, W/mK;

δ_{th} - wall ice thickness, m;

α_a - heat transfer coefficient of the refrigerant to the wall of the accumulator, W/m^{2}K;

$\rho_l = 917$ kg/m^3; $\lambda_l = 2.22$ W/mK; $\lambda_w = 0.560$ W/mK; $\delta_{th} = 0.003$ m;

$\alpha_a = 380$ W/m^{2}K − R134a;

D Time for Partial Defrosting of Ice

The essence of ice accumulation is for storage. The resulting ice is stored in a tank. Heat of defrost, q_{ht} is determined by the formula:

$$q_{ht} = c_l.(t_{ht} - t_l) + r_w \qquad (3)$$

E Ice Defrosting Time

$$\tau_{def} = \frac{q_{def} \cdot \rho_l}{t_{def} + |t_{ht}|} \cdot \left(\frac{\delta_{def}^2}{2.\lambda_w} + \frac{\delta_{th}}{\alpha_a} \right) \qquad (4)$$

E Mass of Ice Accumulated

Many factors affect the performance of a cold accumulator. Consider each factor individually. The mass m_{cyc} of ice kg, frozen on the heat exchange surface during a work cycle, is determined by the formula:

$$m_{cyc} = №_{tub}.\left[\pi.\left(\frac{D_0^2 - D_{tub}^2}{4}\right).h_{tub} + \frac{\pi.D_0^2}{4}\right].\rho_l \qquad (5)$$

where $№_{tub}$ number of accumulator tubes;

D_0 is outer diameter of the frozen ice layer, m; D_{tub} - tube diameter, m;

h_{tub} is tube height in solution, m.

The following options are considered acceptable for the accumulator in the Federal Republic of Nigeria:

$№_{tub} = 16$ pieces, $D_{tub} = 0.025$ m; $h_{tub} = 0.6$ m; $D_0 = D_{tub} + 2\delta_{th}$, m.

F Number of Ice Freezing and Defrost Cycles During Storage

$$№_{cyc} = \frac{\tau_{acc}}{(\tau_{fr} + \tau_{def})} \qquad (6)$$

where τ_{acc} is accumulation time (at night), Adopted τ_{acc} in Nigeria = 9 x 3600 = 32400 s.

The maximum mass of ice frozen during this period is given as

$$m_{ax} = \mathrm{m_{cyc}} . \text{№}_{\mathrm{cyc}} \quad (7)$$

G Determination of Accumulator Discharge Time

A mass of ice accumulated during the night, and then spent on production needs reduces the condensation temperature of the refrigeration system.

Heat entering the discharge accumulator in accordance with Figure 1 is:

$$Q = Q_0 + Q_k = Q_0 + (Q_0 + N_i) = 2Q_0 + N_i \quad (8)$$

where Q is the capacity of the refrigeration unit, kW;

N_i is indicated power of the compressor, kW.

Accumulator discharge time:

$$\tau_{dis} = \frac{2.Q_0 + N_i}{m_{fr}.r_l} \quad (9)$$

where r_l is the latent heat of melting ice.

III. RESULTS AND DISCUSSIONS

The results of the accumulator simulation for temperature conditions $t_0, t_l \; and \; t_{ht}$ for the freezing time and defrosting time using Engineering Equation Solver (EES) software (version 2) for the different regimes are shown in figures (2 - 5) and the results for the performance of the accumulator using the same temperature conditions for the number of freezing and defrosting cycles and the maximum amount of ice frozen on the heat exchange surface are shown in figures (6 – 11).

1. The defrosting time (τ_{def}, s) for the refrigerant R134a was determined and approximated to the equation $\tau_{def} = 2596.9t_{ht}^{-1}$ in the boiling temperature range of $t_0[(-5) - (-15)]$ and in the heating temperature range t_{ht} $[(+25) - (+45)]$
2. An increase in the thickness of ice during accumulator charging period from 0.008 to 0.01 m leads to an increase in charging and discharging time and a decrease in the mass of ice frozen during this time.
3. The maximum performance of the accumulator during accumulation is obtained by freezing ice of 7-9 mm thick.
4. The performance of the ice accumulator, its operating mode and thermal load are the basis for calculating the geometric parameters of the cold accumulator.
5. Computer simulation and calculation during operation of a cold accumulator took into account the heat transfer coefficient α of the refrigerant, the heat of formation q_o, the freezing time τ_{fr} and defrosting time τ_{def} of ice, the number of cycles №_{cyc}, the periodic mass m_{cyc} of ice accumulated during a full cycle, the maximum mass m_{ax} of the accumulated ice during the whole process etc.
6. Any change in pressure will affect the compression ratio of the machine. The value of the compression ratio affects the overall performance of any chiller.

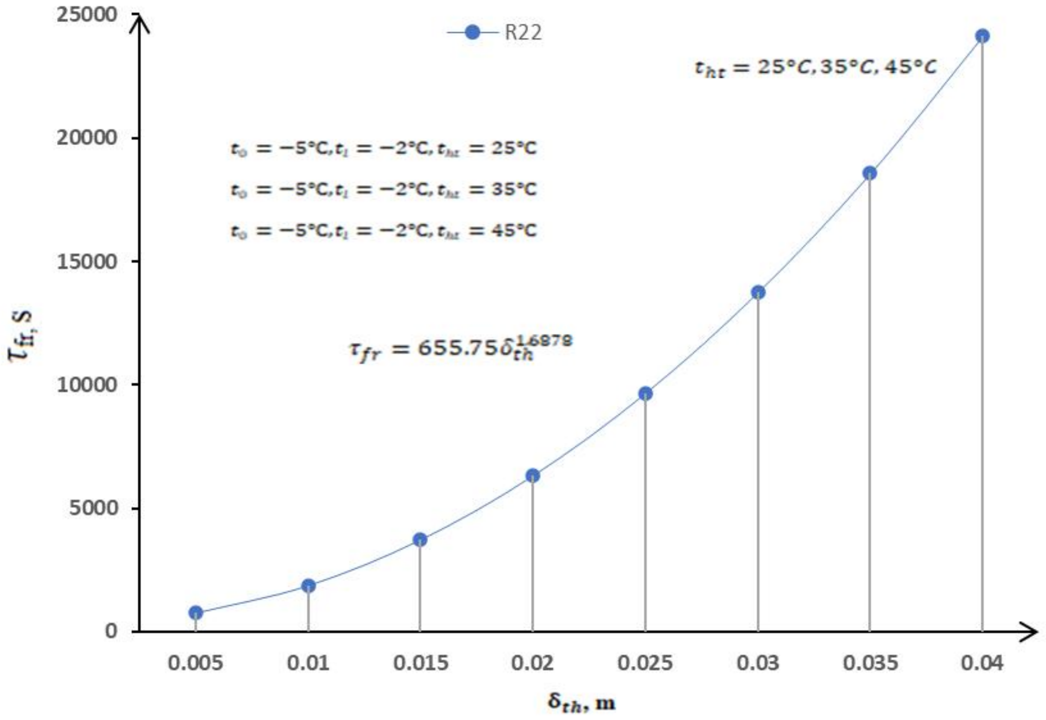

FIGURE 2. VARIATION OF ICE FREEZING TIME WITH THICKNESS AT $t_0 = -5°C$.

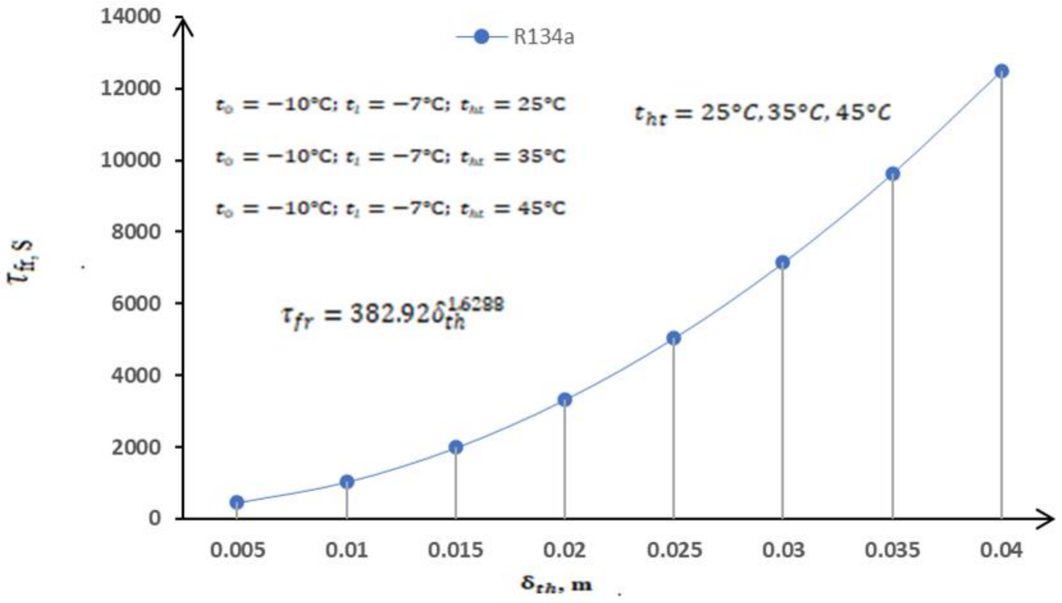

FIGURE 3. VARIATION OF ICE FREEZING TIME WITH THICKNESS

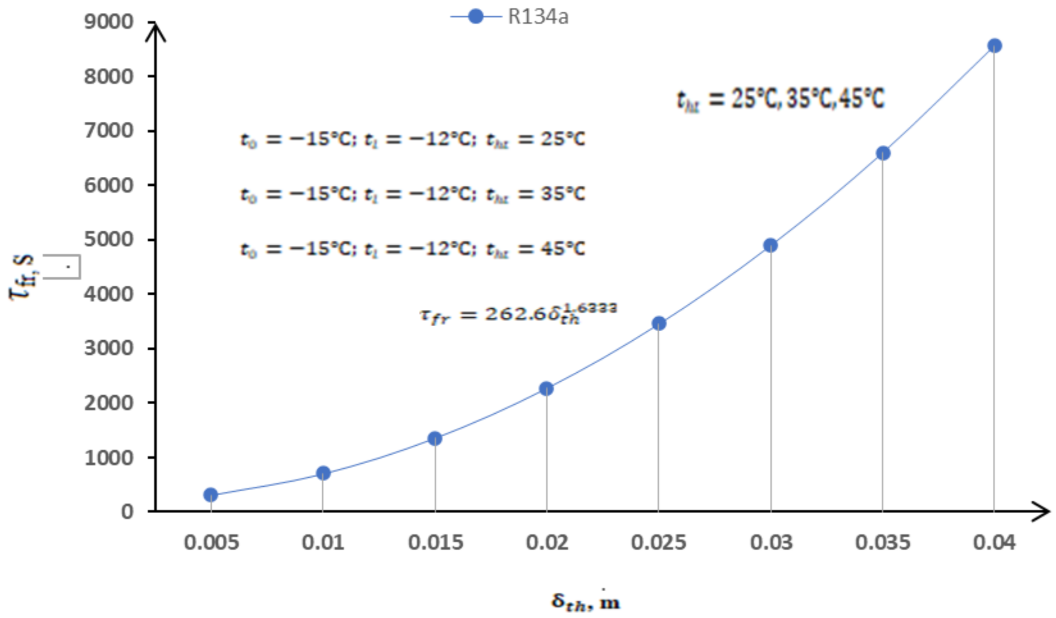

FIGURE 4. VARIATION OF ICE FREEZING TIME WITH THICKNESS AT $t_0 = -15°C$

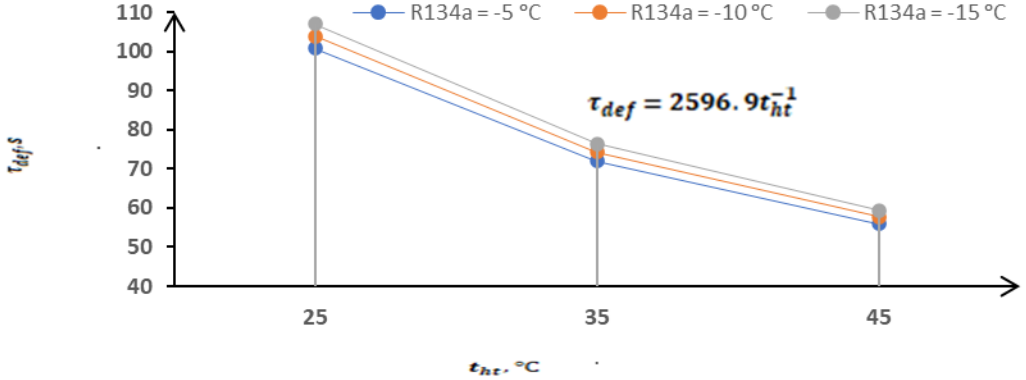

FIGURE 5. VARIATION OF ICE DEFROSTING TIME WITH THE HEATING TEMPERATURE AT $t_0 = (-5, -10, -15)°C$

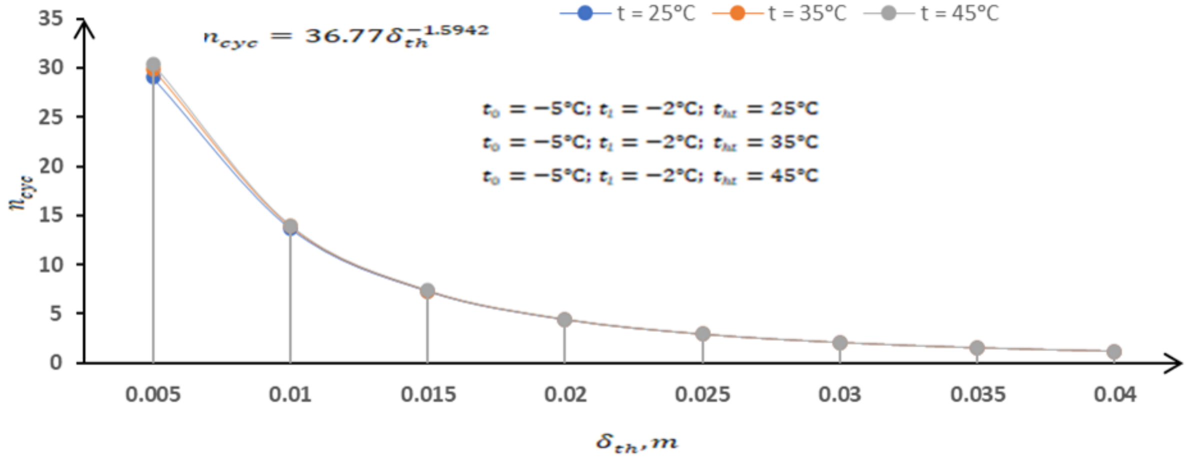

FIGURE 6. VARIATION OF NUMBER OF CYCLES WITH THE HEATING TEMPERATURE AT $t_0 = -5°C$

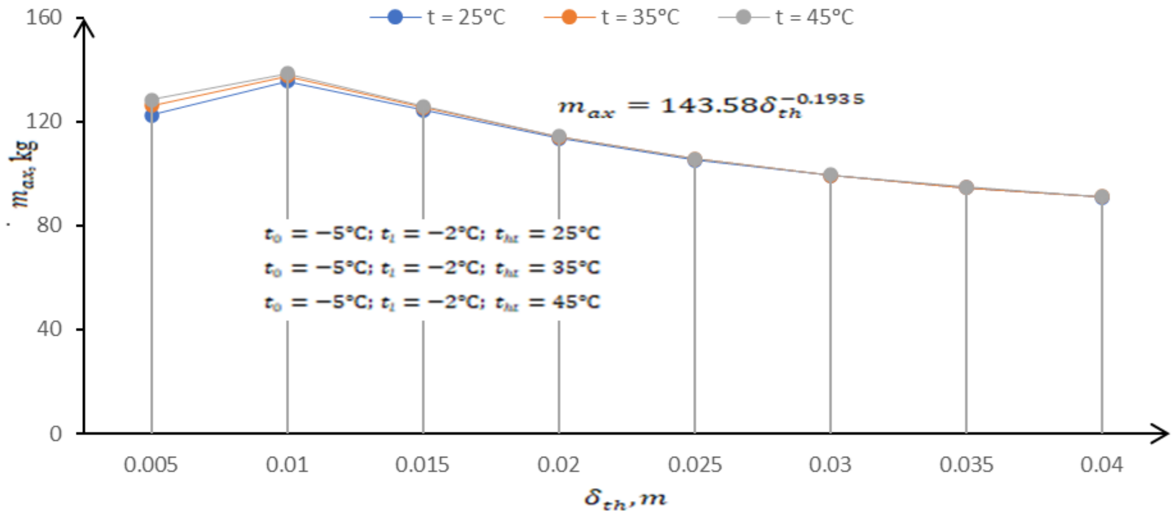

FIGURE 7. VARIATION OF ICE FREEZING MASS WITH THICKNESS AT $t_0 = -5°\mathrm{C}$

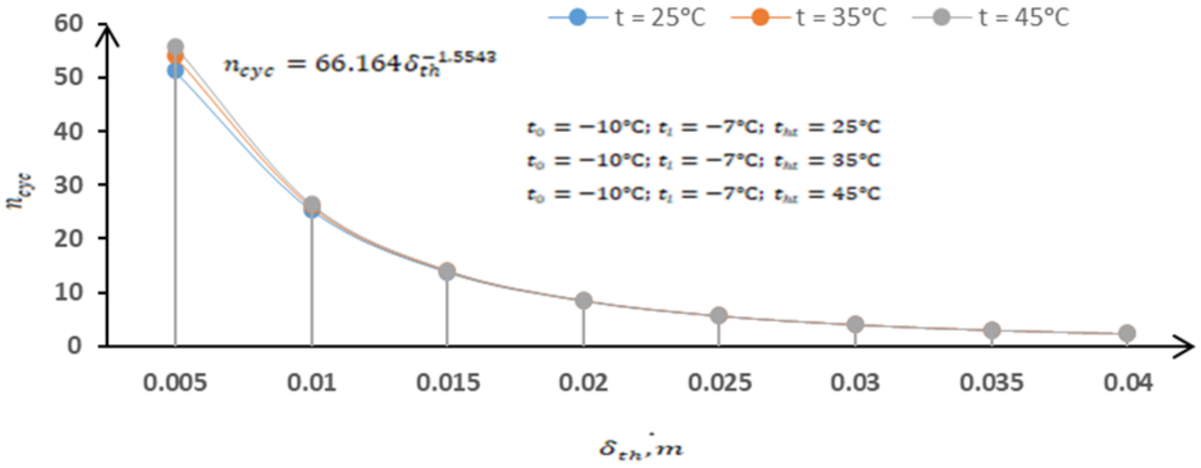

FIGURE 8. VARIATION OF NUMBER OF CYCLES WITH THE HEATING TEMPERATURE AT $t_0 = -10°\mathrm{C}$

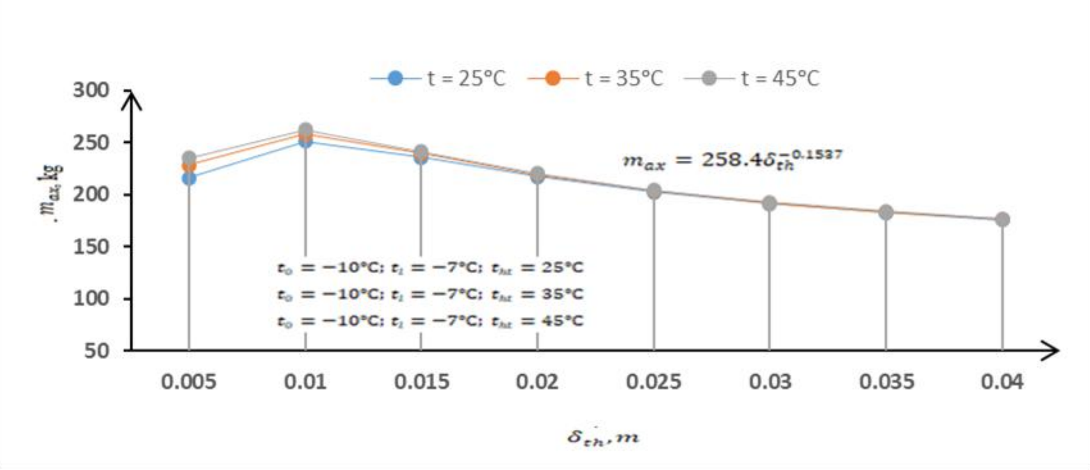

FIGURE 9. VARIATION OF ICE FREEZING MASS WITH THICKNESS AT $t_0 = -10°C$

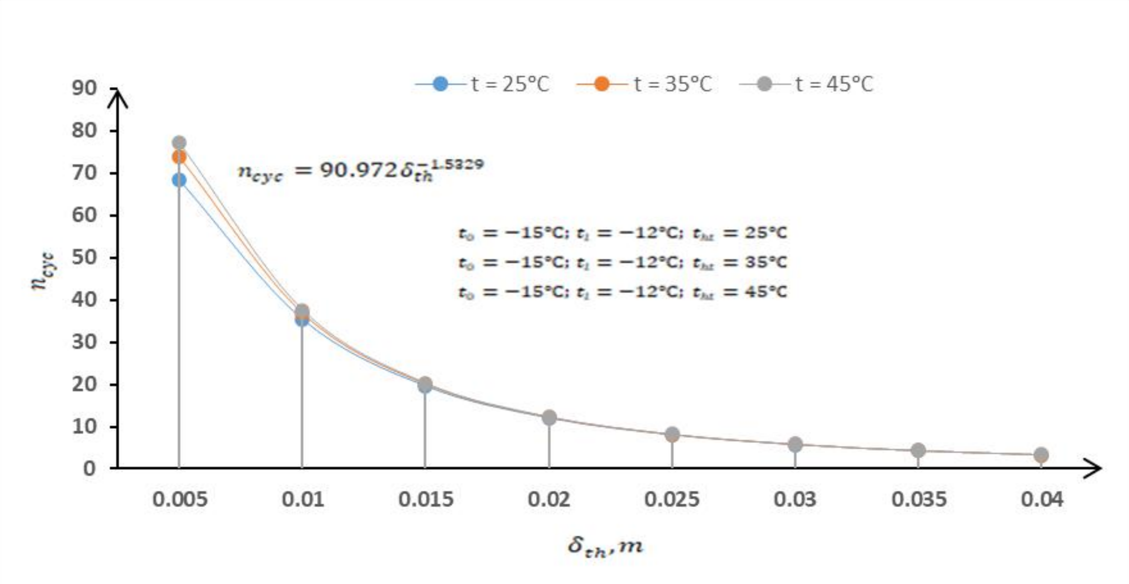

FIGURE 10. VARIATION OF NUMBER OF CYCLES WITH THE HEATING TEMPERATURE AT $t_0 = -15°C$

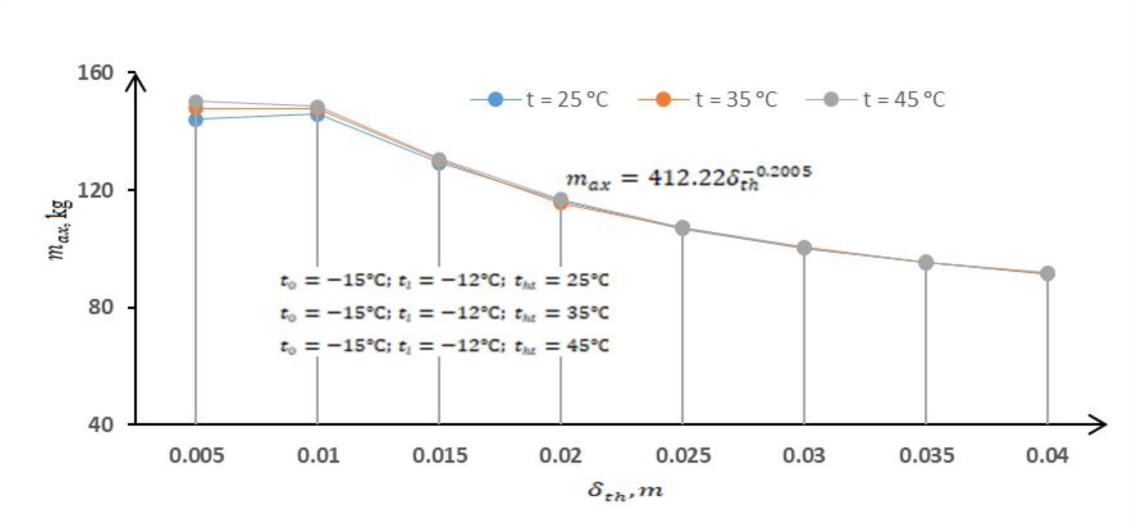

FIGURE 11. VARIATION OF ICE FREEZING MASS WITH THICKNESS AT $t_0 = -15°C$

IV. CONCLUSION

This study investigated the performance of cold accumulator using heat tube for improvement of refrigeration. Refrigerant R134a was used in the simulation by varying the freezing, melting and boiling points of the refrigerant. The results show that optimal performance of the accumulator occurs at ice thickness of 8 mm before melting can begin. The geometric parameters of the cold accumulator can be determined from the parametric equations.

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