DESIGN AND ANALYSIS OF THERMAL COLD STORAGE SYSTEM USING PHASE CHANGE MATERIAL

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ABSTRACT

To maintain thermal comfort inside the cold storage system by controlling the temperature of secure the product life with using of the phase change materials which is placed inside the storage system and elimination of continuous supplying refrigeration internally transferred to the heat to phase change materials. Heat absorbed by the PCM and it change its phase from solid to liquid. Until freezing temperature attained after that it release the energy to atmosphere and change its phase liquid to solid. As a result the temperature of the cold storage system is maintained in comfort condition. Applied the phase change material thermal cold storage system we should to maintain the constant temperature on the system and also to save the continuous supply of energy sudden change of the closed system for maintaining temperature.

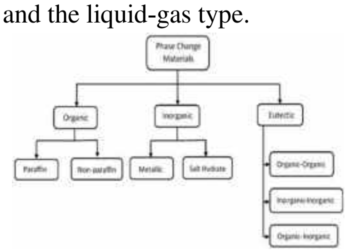
I. INTRODUCTION

The comfortable, modern living conditions are attained by vast energy consumption. The air-conditioning system takes up over half the electricity usage in the building services systems. The traditional air-conditioning system usage is based on the non-renewable energy sources. In contrast, the application of the clean energy can avoid the problem of the environmental pollution and energy consumption. So the application of the phase change material thermal cold energy system has been chosen as a privileged refrigeration system. Because the efficient to usage of the electrical energy from daytime and save the energy for the night, or store the solar energy during summer, and apply for winter, which is not synchronized between the energy generation and consumption. To solve the serious problems, some studies conducted theoretical analysis and experimental studies. The cold storage system is found to store the cold energy using a storage medium and release energy as needed, which is widely used in building air-conditioning system. The cold storage system in the air-conditioning applications is classified by the different storage mediums. In the recent developments, the common methods to achieve a cold storage are water and ice and latent heat storage systems (phase change materials (PCMs)). The latent heat storage uses the latent heat of PCM when the phase changes to energy storage. For a thermal cold storage system, the cold energy produced by based on the advantages of PCMs' energy storage, it is widely used in the different applications in solar energy storage system and building efficiency field. Also, PCM cold storage system helps to reduce the mismatch between the energy supply and demand. At the same time, the system has higher energy. ChuansiGao et.al investigated that the heating effects of three PCM vests (Tmelt=32, 28 and 240C) were tested on a thermal manikin with constant temperature at 300 C in a sub-zero environment (Ta=-40C, Va=0.4 m/s). The results showed that the heating effects lasted about 3-4 hours. The highest heating effects reduced heat lost for 20-30 W/m2 on the torso during the first two hours. **E. Oro, A. De Gracia** investigated TES for cold storage applications using solid–liquid phase change materials has been carried out. The scope of the work was focused on different aspects: phase change materials (PCMs), encapsulation, heat transfer enhancement, and the effect of storage on food quality. Materials used by researchers as potential PCM at low temperatures (less than 200 C) are summarized and some of their thermo physical properties are reported. Over 88 materials that can be used as PCM and about 40 commercially available PCM have been listed. Problems in long term stability of the materials, such as corrosion, phase segregation, stability under extended cycling or sub cooling are discussed. **Justin NingWeiChiu,Viktoria Martin** investigated TES design, erythritol has been characterized rather well as a PCM, while glycerol, erythritol-glycerol blend system and olive oil are novel. However, the thermal cycling using the T-history method, has shown that glycerol as well as the erythritol-glycerol blends up to 30 mol% Er compositions are extremely influenced by glass transition, avoiding phase change even with seeding. The compositions above 60 mol% Er melted indicating decreasing melting points towards the glycerol-rich compositions. However, their freezing accompanied very large super cooling. Due to the absence of phase change in up to 30 mol% Er, deriving the erythritol-glycerol phase diagram was not possible, to find suitable blend compositions. The system in addition underwent thermally activated change. Hence, it appears that the applicability of glycerol or its blends with erythritol as PCMs can be realized only if a fast-crystallization initiation mechanism is found, and thermally activated change is avoided.

Arena Simone investigated two different approaches were analyzed to perform the behaviour of PCM during phase transition. The first approach considers heat transfer only by conduction in both solid and liquid PCM phases. The second approach considers heat transfer by conduction and natural convection in liquid PCM domain. To model the phase transition, the apparent heat capacity formulation was used in both approaches. In this method, the latent heat was implemented by increasing the heat capacity of the material, while the phase change process was modelled by assuming that the transition takes place in a narrow range of temperature rather than at a fixed temperature. The two approaches were separately evaluated for two different cases and then were validated by comparing the numerical results with experimental data. For each model temperature behavior, phase change evolution and melting fraction were evaluated and analyzed for both charge and discharge processes.

II. PHASE CHANGE MATERIALS

Thermal energy storage systems based on phase change materials are considered to be an efficient alternative to sensible thermal storage systems. Furthermore, these systems have high energy density compared to sensible heat storage systems. As said before PCM include the solid-solid, the solid-liquid, the solid-gas and the liquid-gas type.



Classification of PCM

III. PROBLEM INVESTIGATED

The ventilation aspects in a cross ventilation room are investigated. Fig. 1 shows the computational domains (two-dimensional – 2D). A 10 x 20 x 30 ft room of height H= 10 ft and width L=10 ft is considered for investigation. The room has an opening at height (h1) for inlet and another opening at height (h2) for outlet from the floor and is located on the vertical (building) walls. As it is cross-sided naturally ventilated room, the air flow is through the inlet opening on the left side, passes through the room and flows out through the right opening using buoyancy effect. For this room, the opening shape is varied and the analysis is done. Also, analysis is made to compare the air change rate obtained from the analytical methods. The problem is investigated numerically using heat transfer concepts and analyzed by CFD softwares.

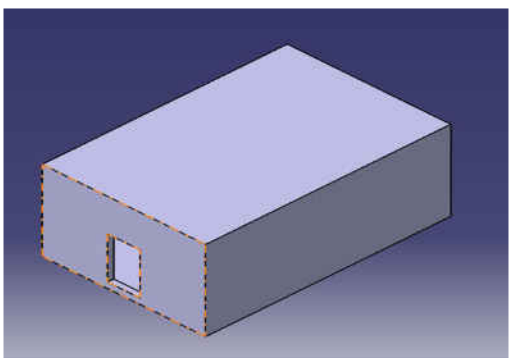


Fig. 1. Computational domains

Design package: CATIA V5

IV.THEORETICAL CALCULATIONS

Dimensions of the storage system

Length = 20 ft

Breath = 30 ft

Height = 10ft

A. Analytical methods

<u>LOAD CALCULATIONS</u>

We are taking a 10 x 10 x 10 ft room as a sample room and the results are investigated.

The area of the ventilation opening used here is 2 x 2 ft opening.

A building or room gains heat from many sources. Inside occupants, computers, copiers, machinery, and lighting all produce heat. Warm air from outside enters through open doors and windows, or as 'leakage' though the structure. However the biggest source of heat is solar radiation from the sun, beating down on the roof and walls, and pouring through the windows, heating internal surfaces. So the major heating loads for a room are the envelope load, people load and fresh air load.

<u>Envelope load:</u>

We know that, heat transfer $Q = UA\Delta T$

The value of overall heat transfer coefficient U can be obtained from ASHRAE 62.1 standards [1].

Considering the outdoor temperature of the room to be 30℃ and the desired indoor temperature to be 25℃, the change in temperature appears to be 5℃.

Total envelope load, Q = (Q1 + Q2 +Q3 + Q4 + Q5 + Q6) / 6

Where Q1 = Heat generated on wall with inlet

Q2 = Heat generated on wall with outlet

Q3 = Heat generated on ceiling

Q4 = Heat generated on the floor

Q5 = Heat generated on other wall

Q6 = Heat generated on other wall

Total envelope load, Q = 1.146 kW

<u>People load:</u>

The internal cooling load due to occupants consists of both sensible and latent heat components. The rate at which the sensible and latent heat transfer take place depends mainly on the population and activity level of the occupants. Hence a portion of the heat transferred by the occupants in the form of radiation.

As per the standards, 20-50 ft / person floor area is required for comfortable occupation and hence the no. of occupants for a 10 x 10 ft room is to be 5.

The sensible heat and latent heat of the occupants can be obtained from ASHRAE 62.1 standards [1].

People load = Total sensible load + Total latent load

= 0.55 kW

<u>Fresh air rate:</u>

As per ASHRAE standard 62.1:6.2.2.1 [2], the design outdoor airflow (or) ventilation rate required in the occupiable space is given by

Ventilation rate, V = Rp.Pz + Ra.Az

V = 31 cfm

<u>Fresh air load:</u>

The fresh outdoor air enters the room through the openings. A load is added by the incoming fresh air which can be calculated using the sensible and latent heat of the air.

Fresh air load = 0.18 kW

Total load in the room = Envelope load + People load + Fresh air load

= 1.146 + 0.55 + 0.18

= 1.876 kW

<u>Amount of heat required to raise the temperature of the room:</u>

We know that, heat transfer can also be expressed as

Q = Cp.m.dT

For a room of volume 1000 ft3,

dT = 40.6 K

<u>Required air flow rate:</u>

Inorder to find the air change per hour it is essential to find the fresh air rate necessary for comfortable occupancy of the occupants.

So, the required air flow necessary for the desired ventilation level can be found using the equation,

Required air flow rate, $L = \frac{Q}{Cp*\rho*(th - tr)}$

Fresh air rate, L = 79.45 ft3/min

<u>Ventilation Rate (m^3/s):</u>

As per CIBSE Guide B2, The rate at which air is exchanged is an important property for the purpose of ventilation design and heat loss calculations. This property is expressed in ventilation rate (m^3/s).

Ventilation rate (m^3/h) = Air change rate (/h) x Room volume (m^3)

$$Ach = \frac{\text{fresh air rate}*100}{\text{volume of room}}$$

$$= \frac{79.45*60}{1000}$$

$$Ach = 4.76\ h^{-1}$$

Ventilation rate (m^3/s) = 4.76 x 1000/3600 =1.32

Therefore the air change rate for the required room has been calculated using the standards for ventilation and heat transfer concepts.

B. Numerical methods

The given analysis involves the following assumptions:

a) The analysis is in a two-dimensional Cartesian coordinate system and steady state condition.
b) The fluid properties are constant and flow is isothermal.
c) The flow is turbulent.
d) Flow is incompressible and Newtonian.

Under these assumptions, the governing equations to be solved are as follows:

a) Continuity equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 1$$

b) Momentum equation in x and y direction

$$u\,\frac{\partial v}{\partial x} + v\,\frac{\partial u}{\partial y} = \frac{1}{\rho}\,\frac{\partial p}{\partial x} + \gamma\,(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 u}{\partial y^2})$$

$$u\,\frac{\partial v}{\partial x} + v\,\frac{\partial u}{\partial y} = \frac{1}{\rho}\,\frac{\partial p}{\partial y} + \gamma\,(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 v}{\partial y^2})$$

where *ρ is the density, u the velocity in x-direction, v the velocity in y-direction, and p is the pressure.* For modeling the turbulent quantity, standard two-equation k-ε model is used. The above equations are solved numerically using the FLUENT 6.0 CFD software.

The boundary conditions to solve the problem are:

a) The air flow is given in the inlet,
b) No slip boundary condition is applied on the floor
c) The building surface and the outlet has zero pressure boundary condition (p = 0).

The model is created using the SOLIDWORKS software. Boundary layer type mesh is applied on the floor and building surfaces. The uniform or expanding grid spacing is applied over the remains regions. The entire region is treated as fluid (air) continuum. The basic conservation equations are solved numerically using the FLUENT 6.0 CFD software. Steady state solver is activated.

Air as fluid is defined and the Boussinesq's model is selected. Gravity is activated. In boundary condition, value for the inlet-velocity magnitude has given. Iterated still the solution is calculated. The grid display, contour of properties, velocity vector and stream function are viewed from the 'Display' option.

IV. RESULTS AND DISCUSSION

Initial and boundary condition,

Initial system temperature = 380C

Velocity of air =10 m/s

Outside system temperature = 300C

A.Adiabatic boundary condition.

We assumed that the environmental conditions were remained unchanged during the numerical simulation. The wall of the comportment was considered as a adiabatic while radiation heat transfer was considered through the wind shield and other glasses. The ambient temperature was considered as 380C and isothermal. Fluent software continuum, energy and transport equations numerically with a natural convection effects. The result shows that thermal accumulation inside the storage system when it maximum temperature distribution.

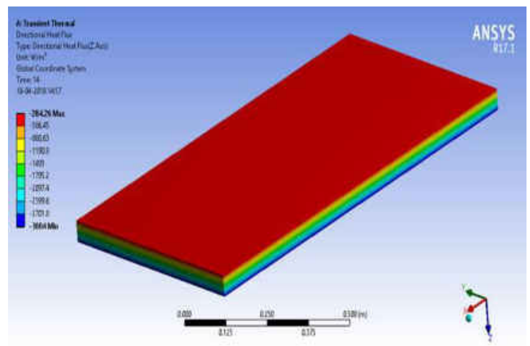


Fig. 2. CFD Results of Heat flux

First, the room is set with a rectangle shaped opening for both the inlet and outlet passages. The opening area is assumed to have an area of 6x 4 ft. The openings are kept at heights h1 = 6 ft. From the floor and the results are taken

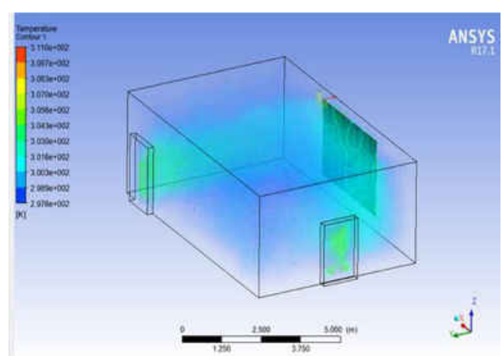


Fig. 3. CFD Results of temperature contour profile

Secondly, the inlet and the outlet has a as these structures could increase the velocity of the air passing through the room at10 m/s. Here in the inlet, the opening has an area of 6 x 4ft and 38℃. The outlet has the exact opposite dimensions on its both sides. The apertures are kept at heights h1 =6ft. from the floor and the results are analysed.

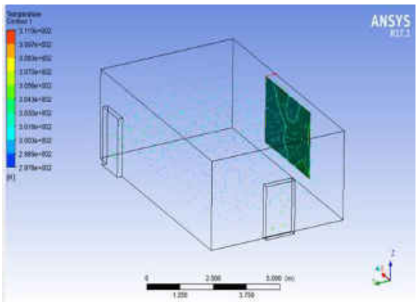


Fig. 4. PCM contour results of temperature

V. CONCLUSION

Study on the “thermal cold storage” had been carried out analytically. The analytical result has been obtained by the temperature measurement conducted on cold storage plant. Three dimensional ANSYS simulation using ANSYS R17.1 code was carried out under “direction of heat flux”. Simulation of the temperature distribution inside the cold storage plant has been done with analytically. The both ANSYS and simulation results indicate that the usage of thermal cold storage system reduce the temperature about 320C to 30.40C in per square meter area of the PCM material. And also result reveals that potential reduction in AC cooling capacity and electrical consumption. Accordingly, its use is highly recommended.

Based on the electrical consumption analysis of a cold storage plant with an air conditioning system shows a better result.

The former condition analysis of an aluminium plate with sodium sulfate gives a 32^0C to 30.4^0C reduction in the efficiency. In future event consideration can be done using the materials which are thermally les conductive like polymer and composites with the sodium sulfate these will show even better than the correct result analysis. A modification in the cold storage plant can even show good results.

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