Isı Bilimi ve Tekniği Dergisi, 37, 1, 103-113, 2017
J. of Thermal Science and Technology
©2017 TIBTD Printed in Turkey
ISSN 1300-3615

NUMERICAL INVESTIGATION OF THE EFFECTS OF GEOMETRIC VARIATIONS OF COOLING TUBES FOR ICE FORMATION IN ICE STORAGE TANK

Gökhan GÜNHAN[*] Ahmet FERTELLI and Ertan BUYRUK*****
University of Cumhuriyet, Department of Mechanical Engineering, 58140, Sivas / TURKEY
*gunhangokhan@gmail.com, **fertelli@cumhuriyet.edu.tr, ***buyruk@cumhuriyet.edu.tr

(Geliş Tarihi: 10.07.2015, Kabul Tarihi: 02.11.2016)

Abstract: In the present study, it is aimed to compare the effects of different geometries located in a rectangular ice storage tank on ice formation. For this aim FLUENT package program was used to solve the flow numerically depending on time. Rectangular tank was modeled in two dimensional and calculations were carried out on half of tank due to symmetry. Water temperature was assumed in the tank as 0 °C, 4 °C and 12 °C, cylinder surface temperatures were taken as -10 °C. Ratio of Ai/Ac (Formed ice area/Cross sectional area), temperature distribution, velocity vectors and liquid fraction were computed for different cooler geometries as circular, hexagonal, square, triangle 1 and triangle 2. The highest ice formation rate was seen in circular model, which was followed by hexagonal, triangle 1, square and triangle 2 models. While the highest solidification rate was seen in circular cross-section cylinders, solidification rates in hexagonal cylinders were rather close to the circular cross-section cylinders. However, the difference between square and triangle 2 models increased even more compared to cases when Ti= 0 °C and Ti= 4 °C
Keywords: Cold storage, Phase change, Natural convection, Solidification.

BİR BUZ DEPOLAMA TANKINDA SOĞUTUCU GEOMETRİSİNİN BUZ OLUŞUMUNA ETKİLERİNİN SAYISAL OLARAK İNCELENMESİ

Özet: Bu çalışmada, içerisinde su bulunan dikdörtgen bir soğu depolama tankına yerleştirilen farklı silindir geometrilerinin buz oluşumuna etkilerinin karşılaştırılması amaçlanmıştır Bu amaçla FLUENT paket programı kullanılarak akış alanının zamana bağlı sayısal çözümü yapılmıştır. Dikdörtgen tank iki boyutlu modellenerek analizler akış alanı simetrik olduğundan tankın yarısı için yapılmıştır. Tank içerisindeki su sıcaklığı 0 °C, 4 °C ve 12 °C olarak alınıp, silindir yüzey sıcaklığı -10 °C kabul edilerek, farklı geometrilerdeki silindir modelleri için A_b/A_s oranı (Buz alanı / kesit alanı), sıcaklık dağılımı, hız vektörleri ve sıvı oranları hesaplanmıştır. En yüksek buz oluşum oranları dairesel kesitli silindir modelinde görülmekle beraber, altıgen modeldeki katılaşma oranları da dairesel modele yakındır. Bununla beraber T_{su}=12°C için kare ve üçgen 2 modellerinin katılaşma oranları arasındaki farkın da 0 °C ve 4 °C ile karşılaştırıldığında daha fazla olduğu görülmektedir.
Anahtar Kelimeler: Soğu depolama, Faz değişimi, Doğal konveksiyon, Katılaşma.

Nomenclature

A_c	Cross-sectional area of cylinder, [m^2]
A_i	Solidified areas, [m^2]
c	Specific heat, [J / kg K]
d	Diameter of cylinder, [m]
h	Sensible enthalpy, [J / kg]
k	Thermal conductivity, [W/m K]
K	Permeability, [m^2]
K_o	Empirical constant in Kozeny-Carman equation
L	Latent heat, [J / kg]
P	Pressure, [Pa]
t	Time, [s]
T	Temperature, [K]
u	Velocity component in x direction, [m / s]
V	Velocity vector, [m / s]
x, y	Cartesian coordinates, [m]
β	Liquid volume fraction
μ	Dynamic viscosity, [Ns / m^2]
ρ	Density, [kg / m^3]
υ	Velocity component in y direction, [m / s]

Subscripts

c	Cylinder
f	Fusion state
l	Liquid phase
ref	Reference value
i	Initial
s	Solid phase

INTRODUCTION

Cold thermal energy storage is very important in recent years, due to the increased energy demand in many parts of the world. For example, large office buildings in warmer climates can spend immense amounts of money and energy for only air conditioning. As a result, there can be a great advantage to shift electricity usage from high demand times to lower ones, in order to save money. This is usually done by storing thermal energy from a cooling system, run with electricity at night, so that the cold thermal energy can be extracted during peak cooling periods along the day (Dinçer and Rosen, 2002). Ice storage systems that can be designed two different types as static or dynamic, are one of the most significant storage systems due to offering high storing capacities in small volumes. Numerous experimentally validated mathematical models of phase change material (PCM) in thermal storage units have been developed over the years. Stamatiou and Kawaji (2005) experimentally investigated the convective heat transfer characteristics of ice slurries flowing vertically upward in a rectangular channel. Al-Abidi et. al. (2013) numerically investigates the solidification of a PCM in a triplex tube heat exchanger with and without internal and external fins to enhance heat transfer during the charging and discharging of PCM. The effects of PCM freezing from the inside tube, the outside tube, and both tubes were investigated. Natural convection presence on PCM storage studied experimentally and numerically by Longeon et al. and studies prove that an injection side coupled with free convection heat transfer mechanism influences the evolution of the PCM melting front. To conclude, a top injection for charge and a bottom one for discharge are recommended (Longeon et.al, 2013). Numerical and experimental studies about different containers and PCM options Ismail and Morales (2009) A simplified mathematical representation has been analytically developed using the ε-NTU technique for a cylindrical tank filled with phase PCM, with heat transfer fluid flowing through tubes inside the tank(Tay et.al, 2012) Lipnicki and Weigand (2012) studied experimentally and theoretically the natural convection and solidification of a vertical annular enclosure; the inner cylinder was cooled down below the solidification temperature of the heat transfer fluid (water), whereas the outer was kept at a constant temperature above zero. The thermal resistance of the contact layer between the cooled inner wall and the solidified layer was investigated; they reported that the influence of the contact layer between the frozen layer and the cold surface was of significant importance for the solidification process. The positions of cooling cylinders for single, 2 and 4 tube cases are investigated numerically by Fertelli et. al (2016). The obtained results shows the natural convection effects in the cavity helps to create thicker ice formation on the top half of the tank and increases the water temperature gap from top and bottom halves. Vyshak and Jilani (2007) analyzed the effect of different configurations of latent heat thermal storage systems which have the same volume and surface area of heat transfer numerically. A comparative study of the total melting time of a phase change material packed in three containers of different geometric configurations: rectangular, cylindrical and shell and tube. The results they present showed that cylindrical containers take the least time for equal amounts of energy storage, and this geometric effect is more pronounced with an increase in the mass of the PCM. Guo and Zhang (2008) studied numerically by using the Fluent software the effects of geometry parameters and boundary conditions on the performance of a new type high temperature latent heat thermal energy storage system (HTLHTES). The comparison between the storage with aluminum foils and without foils is based on the same tube diameter and PCM. The study shows that the discharging process is significantly accelerated by adding aluminum foils. Ezan et. al (2011) developed a measurement method to determine the solidification in a thermal energy storage unit. The measurement method is based on observation of electrical conductivity of PCM that changes dramatically in solidification/melting process. Doğan et. al (2012) researched numerically to determine the effects of geometric parameters of annular fin on a horizontal cylinder case and a correlation was obtained for the optimum fin spacing depending on Rayleigh number and fin diameter.

Fertelli et. al. (2009) studied experimental investigation of the charging process in internal melting on thermal energy storage system. The control volume approach used in analysis of external melting on thermal energy storage system provided good results into system dynamics, and could accurately predict the effects of heat transfer fluid on flow rate and inlet temperature on the cool storage characteristics of the tank. Sugawara et.al. (2008) studied effects of super-heating/sub-cooling and the cylinder locations on freezing and melting of water ice around a horizontal cylinder placed in a square cavity.

In the present study, it is aimed to calculate the effect of ice formation and natural convection movements on different cooler geometries placed in a rectangular cavity filled with water. Unseen on the literature, cross section geometries of the tubes which coolant flows are studied in five various shapes as circular, hexagonal, square, triangle 1 and triangle 2 for two different initial water temperatures.

PROBLEM DEFINITION AND MATHEMATICAL MODELS

The main objective of this study is to reveal the influence of different cylinder geometries (circular, hexagonal, square, triangle 1 and triangle 2) and initial water temperatures on the solidification and natural convection in a rectangular cavity. As shown in Figure 1, a two dimensional fixed tank with height 11d and width 4d is considered in this study and diameter of the tube was taken as d (d=0.0254m). These values were chosen because of constituting a similarity to **those** which are in literature. Cylinders were placed at various vertical distances of h=2d from the bottom and the vertical distance between tubes defined as 4d. Initial water temperatures in the cavity (T_i) and cylinder surface temperature (T_c) were assumed as 0 ºC, 4 ºC, 12 ºC and -10 ºC respectively.

Boundary conditions are considered as symmetry at left and right boundaries, except cylinder surfaces and side walls of the cavity is insulated. The following assumptions are made to simulate solidification around cylinders in a fixed space: Flow is two dimensional, laminar and incompressible.

i) Water as a Newtonian fluid for phase changing material;
ii) Unique thermal conductivities and specific heats (k_s, k_l, C_s, C_l) are considered for solid and liquid phases
iii) Tank walls are insulated
iv) At time t=0, water (PCM) is stable (u=v=0 at cylinder surfaces)
v) Effects of viscous dissipation and radiation are ignored.

An enthalpy formulation based fixed grid approach (Shih and Chou, 2005) is employed in this article to study the transport phenomena during solidification. When the temperature is below T_f - ΔT, β is equal of zero and the cell volume is full of solid. The cell volume is all in the liquid region (i.e., β is equal to 1) if T is greater than T_f. While the temperature falls between T_f and T_f - ΔT, β is between 0 and 1, which means that both liquid and solid phases are coexistent in the same region, called the mushy zone (Shih and Chou, 2005).

The governing equations used in the present analysis are the equations proposed by Shih and Chou.

Continuity equation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho V) = 0$$

In developing the momentum equations, Darcy's law and the Kozeny-Carman equation are adopted to model the flow and the permeability within the mushy zone, respectively.

x -momentum equation

$$\frac{\partial}{\partial t}(\rho u) + \nabla \cdot (\rho u V) = -\frac{\partial P}{\partial x} - \frac{\mu}{K} u + \nabla \cdot (\mu \nabla u)$$

The second term on the right hand side of the momentum equations are flow resistances due to the interaction between water and ice in the solid-liquid coexistence control volume which is modeled by using the Darcy Law (Sugawara et.al, 2008). The permeability K is assumed to vary with liquid volume fraction and is required to have the limits of K= 0 - K= ∞ corresponding to pure solid-liquid of β=0- β=1 (Sugawara et.al, 2008).

y-momentum equation

$$\frac{\partial}{\partial t}(\rho \upsilon) + \nabla \cdot (\rho \upsilon V) = -\frac{\partial P}{\partial y} - \frac{\mu}{K} \upsilon + \nabla \cdot (\mu \nabla \upsilon) + (\rho_m - \rho) g$$

Energy equation

$$\frac{\partial}{\partial t}(\rho h) + \nabla \cdot (\rho h V) = \nabla \cdot \left(\frac{k}{c} \nabla h\right) - \frac{\partial}{\partial t}(\rho \beta L) - \nabla \cdot (\rho \beta L V)$$

where the permeability is obtained by Kozeny-Carman equation (Shih and Chou, 2005):

$$K = K_o \left(\frac{\beta^3}{(1-\beta)^2} \right)$$

K_0 (=5.56 x 10^{-11} m^2) is a permeability coefficient which could be adopted from the mushy zone morphology (Sugawara et.al, 2008).

And sensible enthalpy-temperature relations (Shih and Chou, 2005)

$$h = \begin{Bmatrix} C_s T & T < T_f \\ C_l T & T \geq T_f \end{Bmatrix}$$

The second and third term on the right hand side of the energy equation is a source term related to phase change. It denotes the generation /absorption due to freezing/melting (Sasaguchi et.al, 1997). In deriving the energy equation, the sensible enthalpy (Shih and Chou, 2005), h, is defined as follows:

$$h = h_{ref} + \int_{T_{ref}}^{T} c dT$$

where h_{ref} is the reference enthalpy, T_{ref} is the reference temperature, and c is the specific heat at constant pressure. Moreover, during the phase-change process, the latent heat can vary between zero (for a solid) and the latent heat of the material, L (for a liquid). A non-Boussinesq approach is applied and to account for the buoyancy-driven flow within the domain, the density-temperature equation derived by Gebhart and Mollendrof is expressed as follows:

$$\rho = \rho_{\mathrm{m}} \left(1 - w \left| T - T_m \right|^q \right)$$

where ρ_m is the maximum density of water (= 999.972 kg/m3), w =9.2793 10-6 °C-q, Tm= 4.0293 °C and q = 1.894816 (Sugawara et.al, 2008).

RESULTS AND DISCUSSION

Finite volume method (FLUENT package program) was used to solve all the natural convection based solidification stages around cylinders. Quadrilateral grid was used for simulation and finer grid distribution was used near cylinder surfaces (Fig. 1). Grid spacing increases away from cylinder surfaces, with total of 20000 to 25000 elements. Three grid sizes at 20208, 39962 and 59750 cells were investigated to validate the independence of grid size from the numerical solution and given in Figure 2. In case of T_i= 4 ºC and T_c= -10 ºC with a circular model (Fig. 1) computed for grid study. When three different grid numbers are compared, it is seen that mesh is independent of grid number between 20000 and 59750.

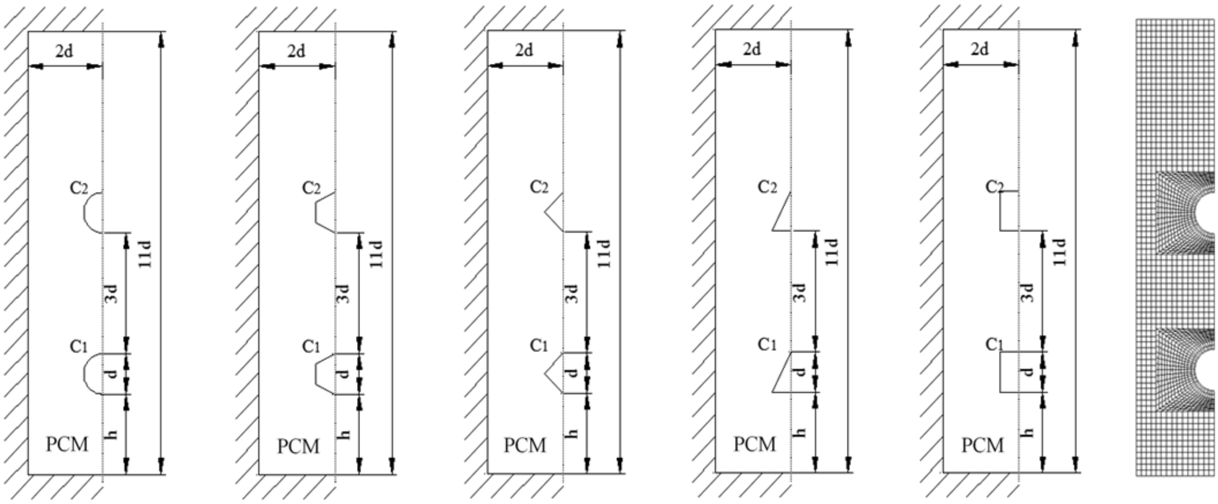

Figure 1. Physical models and grid system

The results of the current study have been compared with the results of a previous experimental study to verify the results obtained with the numerical method used. As a result of time-dependent numerical analyses, the rate of ice area to the cylinder surface area was found and compared with another study as a reference (Sasaguchi et.al, 1997). As it can be seen in Figure 3, when the experimental results are compared with numerical analyses, solidification rates were exactly the same at the beginning of ice formation. In the later stages of solidification, the rates got rather closer and parallel.

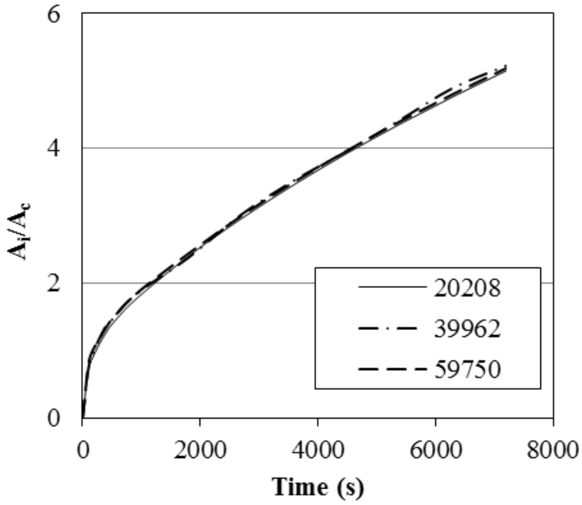

Figure 2. Grid independence results for solidification around a cylinder.

To examine the effect of different cylinder geometries (circular, hexagonal, square, triangle 1 and triangle 2)in rectangular cavity on transient natural convection in water, analysis were carried out for different initial water temperatures (T_i= 0 ºC, 4 ºC, 12 ºC).

The results of temperature contours, liquid fractions and velocity vectors are observed for all the models at the 900 s, 1800 s, 3600 s and 7200 s seconds and variation of solidified area ratio (A_i/A_c) which is defined as the ratio of the solidified area to the total cross sectional area of the cylinder obtained by the time.

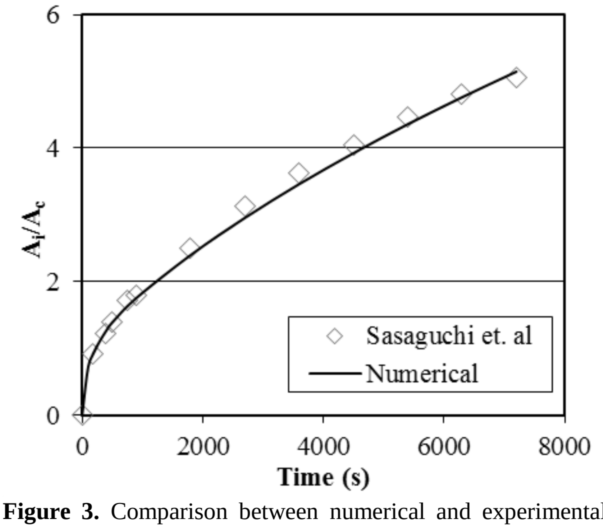

Figure 3. Comparison between numerical and experimental data for solidification

Figure 4 shows the timewise variation of the isotherms, velocity vectors, and liquid fractions for circular and hexagonal coolers and initial water temperature of 4 °C. The water in the tank started to get cooler by the end of t= 900 s, water at the bottom of the tank preserved initial 4 °C temperature. As water had the maximum density value at this temperature, the water cooled on the cylinder surface started to flow upwards as its density decreased.

The temperature of the water around cylinder C_1 decreases and gets even colder as it moves around the upper cylinder. Meanwhile, the water contacted with the ice layer around the upper cylinder and made swirling movements under the cylinder. At the same time, the water getting cooler around cylinder C_2 moves towards the upper part of the tank and cools the upper part of the tank faster. While water temperature in the area between the cylinders was at about 3 °C, the water flowing towards tank walls was colder. Besides, it was seen that thermal stratification started in this region. At t= 3600 s and t= 7200 s, the water started to let up and the temperature of the water decreased with the conduction at the bottom of the tank and with the completion of thermal stratification. When liquid rates and velocity vectors were examined it was seen that while natural convection movements played an active role until t= 1800 s, the convection movements around

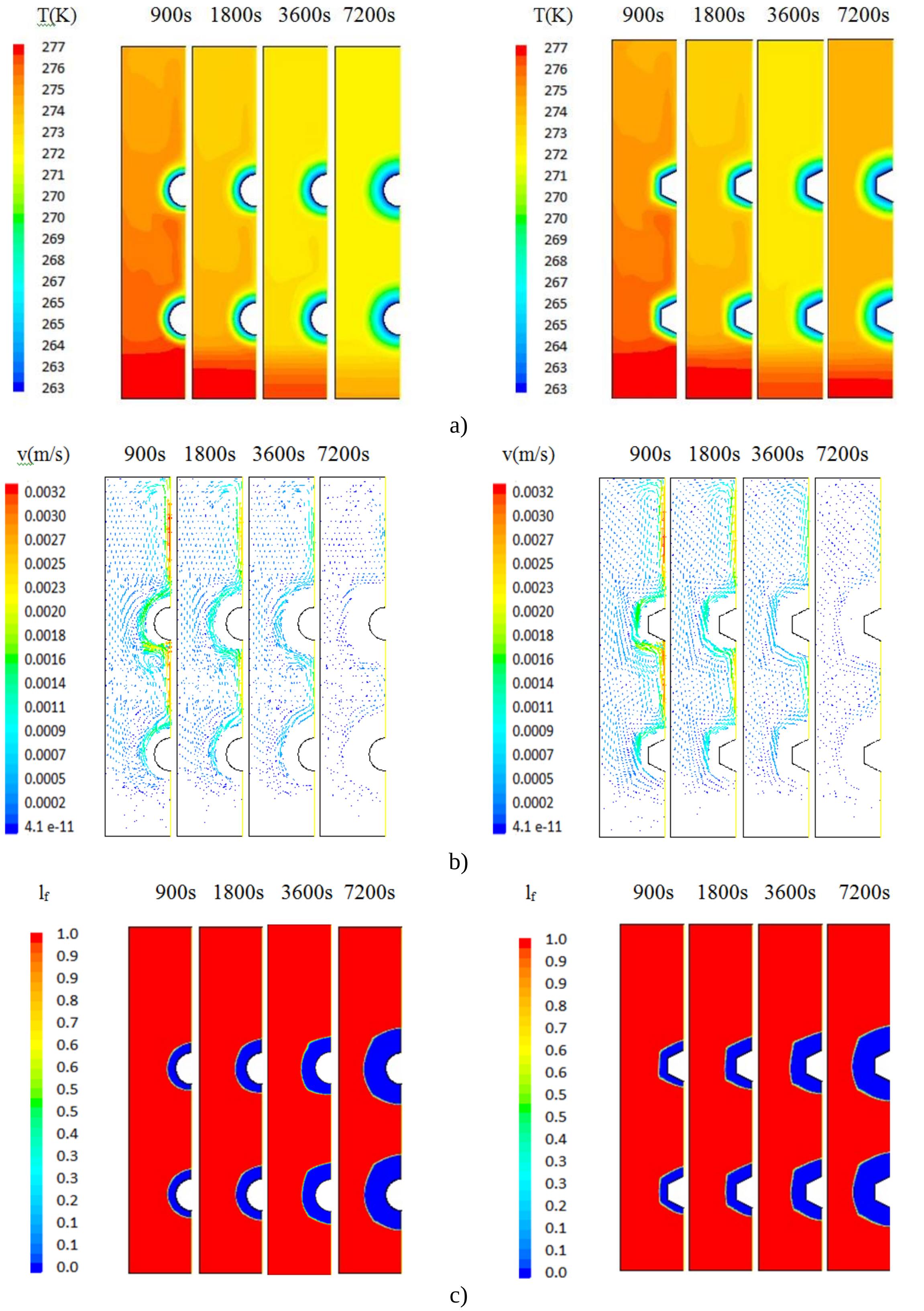

Figure 4. Variation of isotherms, velocity vectors and liquid fractions for circular and hexagonal geometries (T_i = 4 °C) a) isotherms b) velocity vectors c) liquid fractions.

the cylinders and throughout the tank started to weaken later and gradually terminated. t= 900 s onwards, ice layers started to form around the cylinders and the ice formations did not contact each other as the distance between the cylinders was larger. While ice formations around the cylinders had a circular shape, ice formations in hexagonal model started by following the cooler geometry as in one cylinder. At the end of t= 7200 s ice formations evolved to have a circular form.

Isotherms, velocity vectors and liquid fractions at T_i= 4°C for triangle, square and triangle 2 geometries are shown in Figure 5. At t= 900 s the water in the tank started to become cooler in each of the three models and an upward flow movement was observed between the cylinders as in circular and hexagonal models. The water contacting the upper cylinder goes towards the inner areas of the tank by swirling counterclockwise. As it can be understood from the liquid rates, a significant difference in ice formation is seen as a result of this swirling movement particularly in the square model. At the end of t= 1800 s, asymmetric ice formation at the bottom part of cylinder C_2 in square model was seen. This can attributed to the fact that the water cooling down around cylinder C_1 cannot move upwards and cools down to solidification temperature, and instantly starts freezing. While the fastest velocity from cylinders towards the upper part of the tank reached at t= 900, flow movements ended at t= 3600 s. In triangle 1 model, ice formation, which followed cooler profile at the beginning, followed a more shapely form as the water in the tank remained stable and almost all the water in the tank cooled down to phase change temperature at the end of t= 7200 s. In triangle 2 model, ice layer did not form circular shape as in other models but an ice mass shining upward was formed.

Figure 6 shows the timewise variation of the isotherms, velocity vectors, and liquid fractions for circular and hexagonal coolers and initial water temperature of 12 °C. Large temperature gradients were observed and more effective natural convection movements occurred. The water around both cylinders moves towards the lower part of the tank because of the temperature difference between water and surface temperature. The water whose temperature decreased moved down the lower parts of the tanks because of density resulting from the temperature difference between warm water around it and cold cylinder surface.

Meanwhile, the water accumulated in the lower parts of the tank at 4 °C continued to cool down and made a counter flow movement following the cylinder surface and started an upward flow movement when its temperature decreased below 4 °C. On the one hand, cold water reached to solidification temperature and started to form ice layer around the cylinder; on the other hand, after the cold water leaving cylinder surface reached to the tank walls, it got closer to the upper cylinder by making clockwise swirl. In the lower parts of cylinder C_2, ice formation was larger because the temperature difference between cooler cylinder surface and water was smaller compared to upper areas.

At the end of t= 1800 s, upward flow movements were seen around both cylinders and the water cooling down around the upper cylinder as well reached to the walls of the tank and started swirling clockwise. As it can be seen from the general temperature distribution of the tank, while the water at the bottom of the tank is 4 °C, in the upper parts of the tank the water was at 12 °C just as at the beginning. The temperature of the water in the area close to the cylinders decreased down to about 1 °C.

At the end of t= 3600 s, natural convection movements increased even more and in the area between the two cylinders the velocity vectors reached to their highest values. In the circular cross section, the flow moving upwards following the geometry of the cooler went towards the walls of the tank rather than the upper parts of the tank because of hexagonal cross section geometry. Meanwhile, water at 1 °C to 2 °C around the upper cooler cylinder moved towards the upper part of the tank because of the density created under the influence of temperature difference between it and the warmer water around it.

At the end of t= 7200 s, all of the water in the tank cooled down to solidification temperature and natural convection movements finished. As it can be understood from liquid rates, ice formations were more in the lower parts of the cylinders since cold water initially accumulated in the lower part later on asymmetric ice formation was seen in the upper cylinder because of water movements between the cylinders. At the end of 7200 s, more ice formation was seen because solidification shapes followed cylinder geometry and water temperatures in the parts of the cylinders facing each other were closer to phase change temperature.

Isotherms, velocity vectors and liquid fractions at T_i= 12°C for triangle, square and triangle 2 geometries are shown in Figure 7. While the warm water in top half of the tank preserved its initial temperature at t= 900 s, water temperature decreased in the lower parts and it cooled down to 4 °C - 5 °C at the bottom of the tank. No flow movement was seen around cylinder C_2 in three models. While the cold water around the lower cylinder created a downward flow movement at the beginning, it made counter flow movement and flowed toward the upper cooler and tanks' walls after the water temperature decreased down to 4 °C. Cold water which faces warmer water made a swirling movement by hitting the walls of the tank. While these movements were more severe in square model, they were weaker in triangle 1 and triangle 2 models. In triangle 1 and triangle 2 models, the swirl movement in the area between the cylinders weakened at t= 1800 s and turned into an upward flow movement. In square model, water temperature decreased and the same movement continued. At that time, the water whose temperature decreased around the upper cylinder started an upward movement with the influence of density. Cold water left the surface of the cooler cylinder and moved towards the tank's walls. As there was still warm water in the upper part of the tank swirling movement went on till t= 3600 s. The top half of the tank cooled down to 1 °C in square and triangle 2 models and natural convection movements resulting from temperature difference weakened.

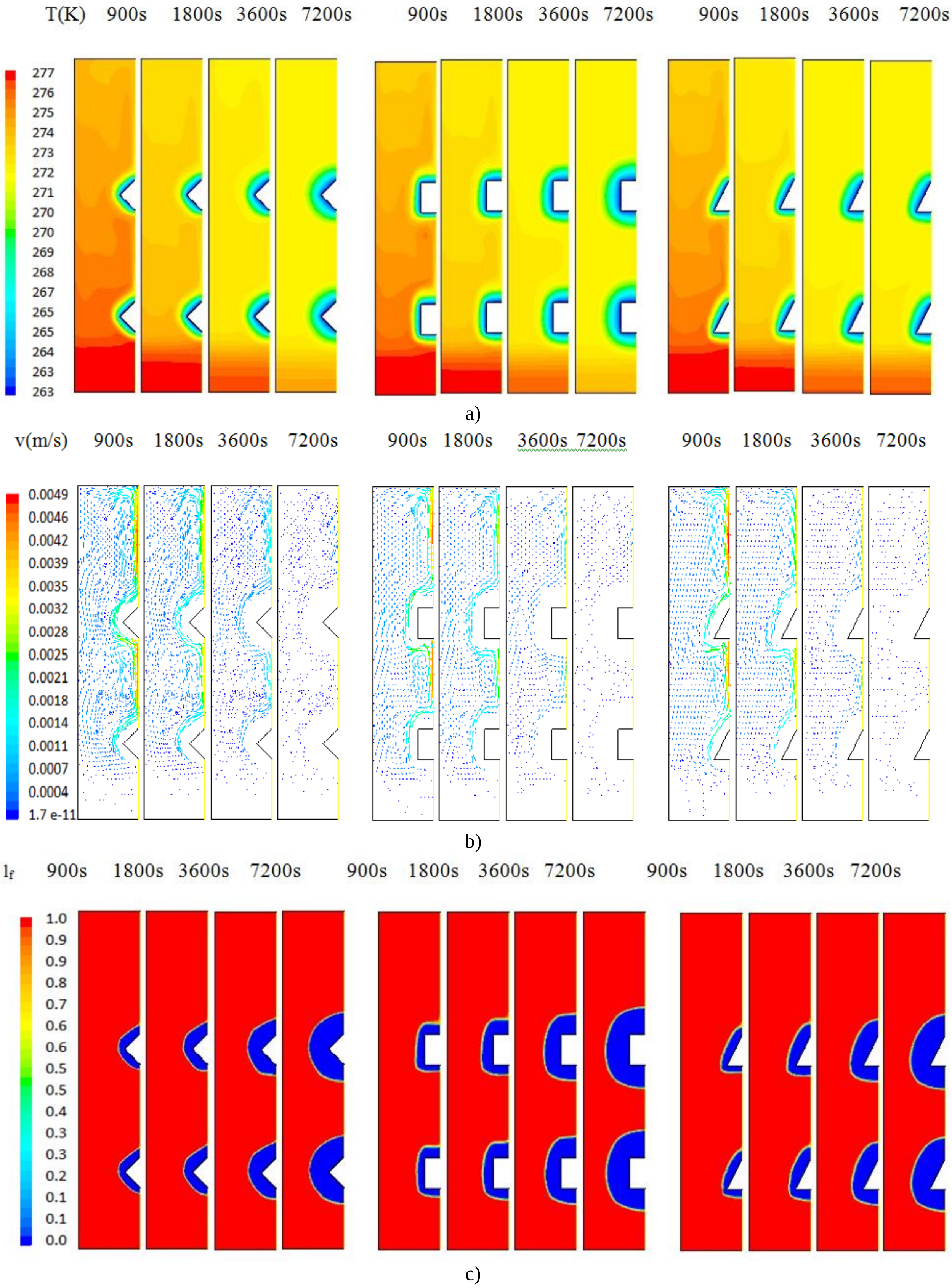

Figure 5. Variation of isotherms, velocity vectors and liquid fractions for triangle 1, square and triangle 2 geometries (T_i=4 °C) a) isotherms b) velocity vectors c) liquid fractions.

At the end of t= 7200 s, thermal stratifications was completed in square, triangle 1 and triangle 2 models and water stagnantly continued solidification at phase change temperature. When liquid rates were examined, it was seen that while ice layer formed around the cylinders as early as the beginning, asymmetrical

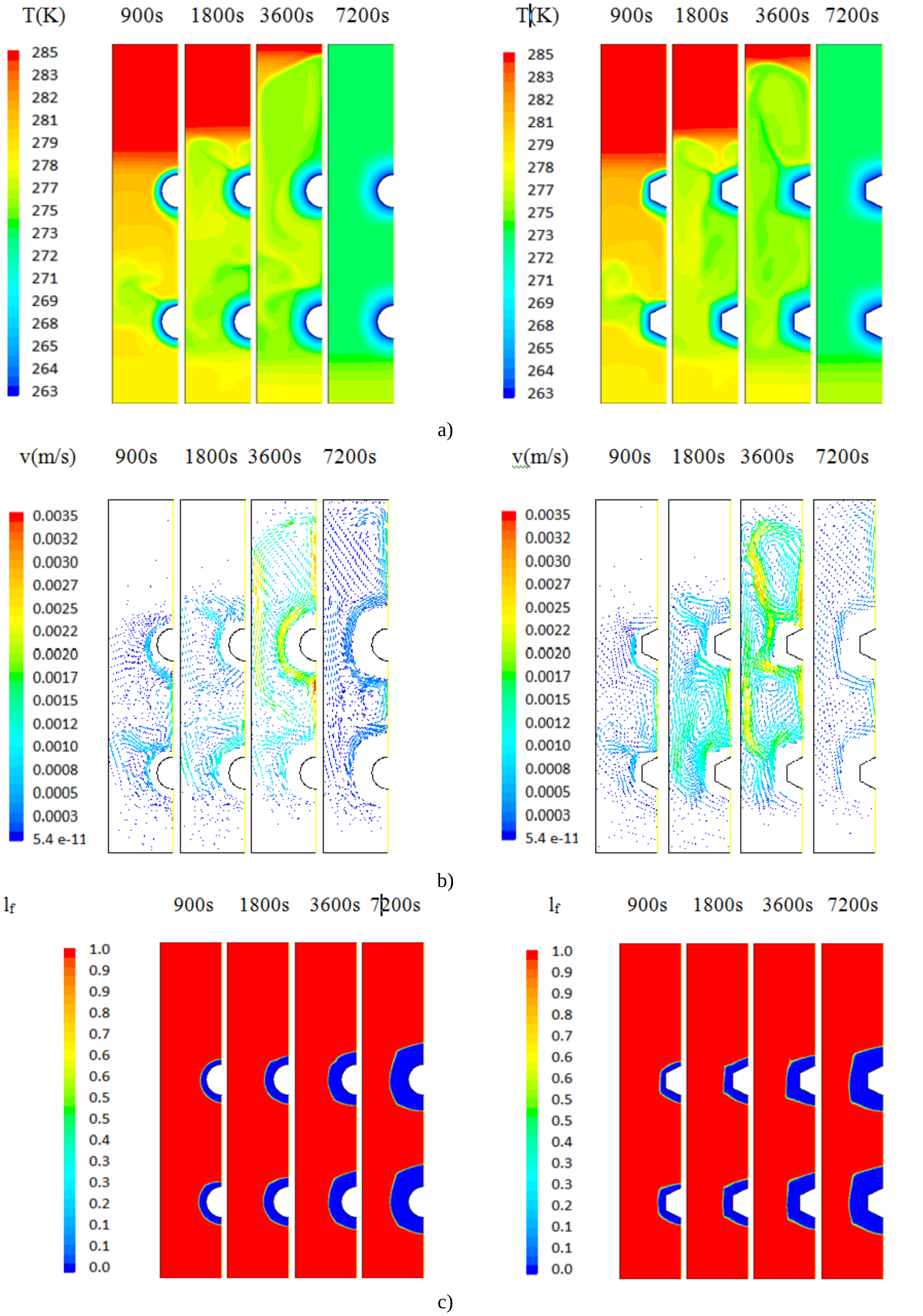

Figure 6. Variation of isotherms, velocity vectors and liquid fractions for circular and hexagonal geometries (T_i = 12°C) a) isotherms b) velocity vectors c) liquid fractions

solidification was seen in the area between the cylinders as a result of flow movements. The thickness of ice following the cooler geometry increased as time passed, while in triangle 1 model it took a circular shape at the end of t= 7200 s, it took ellipse shape in triangle 2 cross section.

The effect of cylinder geometries on the growth rates of the ice layer is shown in Figure 8 for T_i = 0 °C, 4 °C, 12 °C. As is seen from figure, It was observed that when T_i= 0 °C, flow movements were seen and ice formation started in the first seconds and solidification rates increased in time. When water temperature was at 0 °C and 4 °C, the highest solidification rate was seen in circular cross-section model.

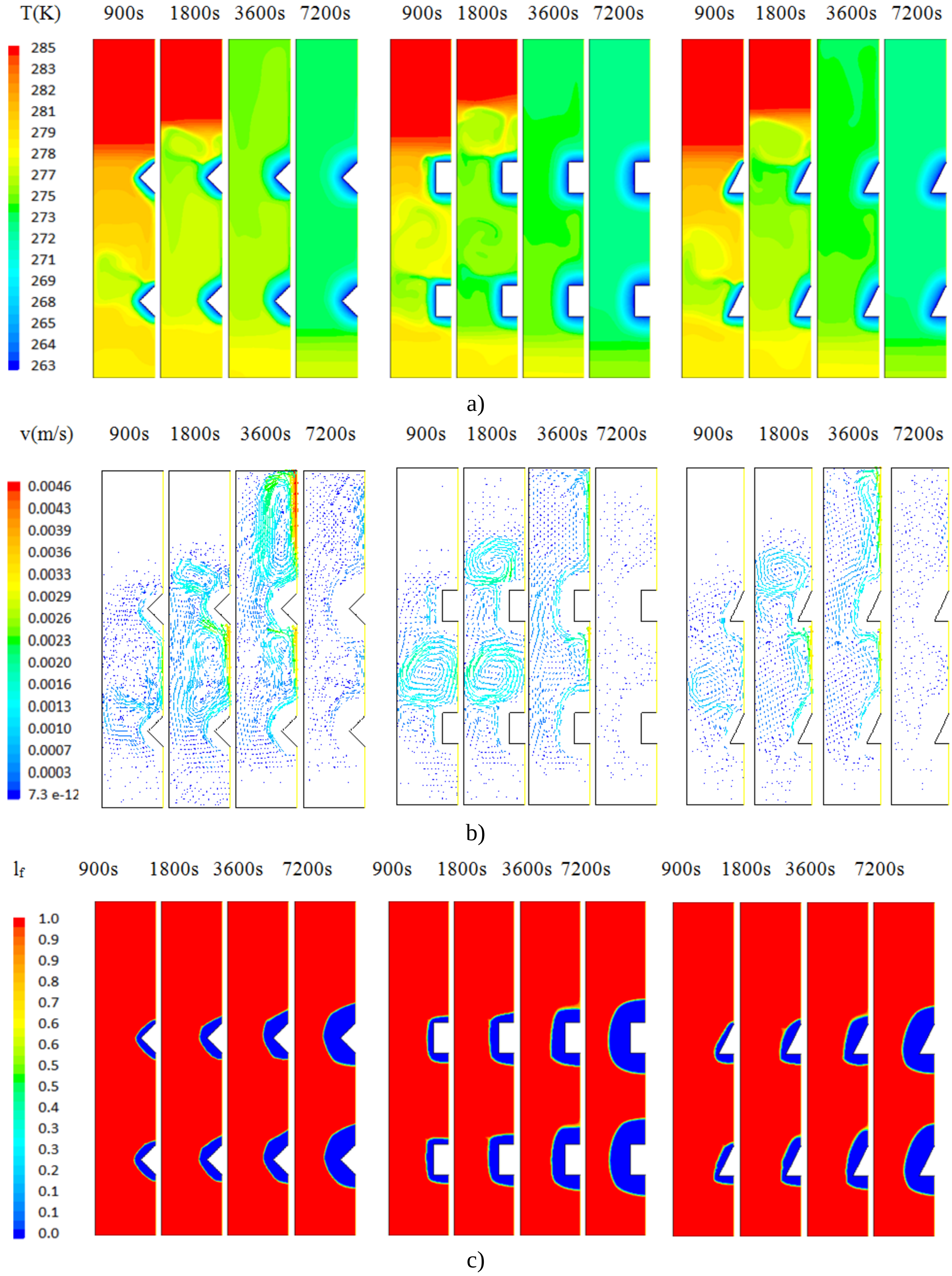

Figure 7. Variation of isotherms, velocity vectors and liquid fractions for triangle 1, square and triangle 2 geometries (T_i=12 °C) a) isotherms b) velocity vectors c) liquid fractions.

When the water temperature at the beginning was T_i= 12 °C and when the time passed for the water in the tanks to cool down to solidification temperature was considered, lower solidification rate was seen compared to those of other initial temperatures. While the highest solidification rate was seen in circular cross-section

cylinders, solidification rates in hexagonal cylinders were rather close to the circular cross-section cylinders. However, the difference between square and triangle 2 models increased even more compared to cases when T_i= 0 °C and T_i= 4 °C.

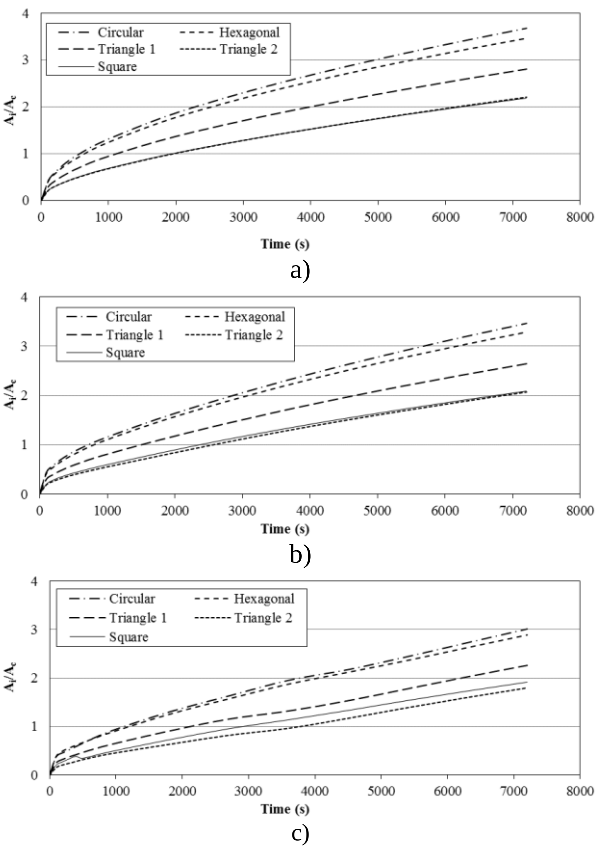

Figure 8. Effects of cylinder geometries on growth of the solidified area for two cylinder a) T_i = 0 °C b) T_i = 4 °C c) T_i = 12 °C.

CONCLUSIONS

In this study, phase change and natural convection movements in a cold storage tank used in ice storage method, which is one of the cryogenic storage methods, were examined. Ice layers formed in time around cooler cylinders of different cross section geometry were examined, temperature distributions, liquid rates and velocity vectors throughout the tank were monitored at 900 s, 1800 s, 3600 s and 7200 s. When solidification rates at all temperatures were examined, it was seen that the highest ice formation rate was seen in circular model, which was followed by hexagonal, triangle 1, square and triangle 2 models. It was seen that initial water temperatures were important factors in ice formation; no flow movement occurred in the tank for 0 °C water temperature and stratification started very early. In other water temperatures, stratification rates decreased by 6% and 18% compared to water at phase change temperature, respectively.

ACKNOWLEDGMENTS

The authors acknowledge the CUBAP (The Scientific Research Projects Council of Cumhuriyet University) for the financial support of the project M-491

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