Effect of Adding Horizontal Rings on the Thermal Behavior of Cylindrical Liquid Enclosure Exposed to High Heat Flux

Dr. Mohammed Hamed Mahmoud
Asst. Professor
Mech. Eng. Dept.
Al- Mustansiriyah University

Eng. Ali Jaber Alkhakani
Mech. Eng. Dept.
Al- Mustansiriyah University
Baghdad - Iraq

Abstract :

Power generation by using concentrated solar thermal energy on liquid enclosures is one of the most promising renewable energy technologies. In this work, a developed liquid enclosure fitted with various number and configurations of horizontal metal rings have been analyzed, fabricated and tested. The influence of adding metal rings arrangement is investigated for its potential to enhance radial heat conduction to the center-line of the enclosure from the side-walls. Experiments were carried out for fluid in both static and dynamic modes of operation inside the enclosure that subjected to high heat flux.

A developed two-dimensional CFD model to predict the transient flow and thermal fields within liquid enclosure subjected to heat flux has been developed and tested. The developed numerical model takes into consideration energy transport between the liquid inside enclosure and the solid material of the enclosure. The numerical simulations have been compared with experimental measurement. The computational code has been found in a good level of agreement with the experimental data except for liquid at the peak part of the enclosure.

The results indicate that adding metal rings produce significant impact on the transient temperature difference inside enclosure during both static and dynamic modes. The six-ring model is found to be more effective for enhancing radial heat transfer than other three models that have been tested. The in-line arrangement is found to provide better thermal effect as compared to the staggered rings.

Two new correlations for natural heat transfer inside liquid enclosures subjected to high heat flux have been formulated (one for no-ring model and the other for six-ring model). The natural Nusselt number is found to be around a constant value for Rayleigh number less than (5×10^8).

The recommended use of metal rings inside liquid enclosures subjected to heat flux, and the predicted Nusselt number correlation, will add to local knowledge a significant mean to gain more heat in large scale concentrated solar power plants.

Keywords : *horizontal rings, liquid enclosure, concentrated solar energy, Nusselt number.*

Introduction:

The heating and cooling processes within liquid cylindrical enclosures often occurs in many engineering applications and industrial processes, such as, industrial food production, domestic solar water heaters, petroleum exploration and metallurgic industry, among others [1] . Another application is related to chilled-water storage unit in air-conditioning systems. This chilled storage is charged at night during hours of maximum coefficient of performance, and discharged at the next day (or in the hours of electricity cutoff) to meet the load demand [2]. Generally, these processes involve a heat transfer of a fluid that has initially a uniform temperature. The design of such systems, demands an understanding of the transient temperature fields that arises in liquid enclosure during charging/discharging and relaxation periods (i.e. in the absence of external flow).

In the recent few years a significant effort has been invested into the determination of the flow profiles and heat flow distribution within the Concentrated Solar Thermal (CST) systems. (CST) is a method where by the solar radiation that falls on a given surface is focused on a smaller surface to increase its intensity. (CST) is a proven renewable energy technology that harnesses solar irradiation in its most primitive form for electricity generation and industrial applications [3]. In many ways, the design and operation of solar thermal power generation systems is more complex and challenging than that of conventional fossil fuel power–plants. Therefore, technologies are required to concentrate, absorb and transfer solar thermal energy to the working fluid [4].

Several works concerning about this field are available in the literature, involving experimental and numerical analysis. Experimental investigations by Shyu et al. [5] showed that degradation of thermocline in storage tank with thicker walls is more pronounced due to larger axial heat conduction in the tank wall. It was concluded that the heat loss to the ambient was the major factor in degradation of the thermal stratification in an un-insulated tank.

Hariharan and Badrinarayana [6] studied the effect of ambient and operating conditions on temperature distribution of hot water storages. They observed that temperature gradient improves with increasing temperature difference and water flow rate.

Zachar et al. [7] studied numerically the impact of a baffle plate facing the inlet jet on temperature fields in vertical tanks. According to their study, the use of large baffle plates allows the preservation of the thermal gradient even for high flow rates.

Altuntop et al. [8] analyzed numerically the effect of using different obstacles on thermal behavior in hot water tanks. The results indicate that the obstacle types having gap in center appear to have better thermal gradient than those having gap near the tank wall. However, the numerical model ignored the effect of tank wall in the calculations. Also, no experimental work has been conducted to validate these numerical results.

Haltiwanger and Davidson [9] investigated experimentally the effect of adding cylindrical baffle in a cylindrical water thermal storage. The results indicate that the baffle increases the storage side convective heat transfer.

Ham et al. [10] concluded that two-dimensional models can take the mixing between layers into account, which includes more factors than that of one-dimensional level. They concluded that further research should be focus on enhancement of temperature gradient between layers.

A comprehensive examination of literature in the field indicates that there are a significant number of studies related to the problem of the thermal water enclosure. However, the majority of these works have ignored the thermal effect of the solid enclosure materials. Also, most of these previous works consider either the enclosure is adiabatic, or the direction of heat flow is out of the enclosure. Also, it seems that most of previous researches have been focused on thermal response for solar domestic hot storage tanks and very little work has been done on thermal gradient evaluations for liquid energy storage packed with solid materials. In fact, just recently Valmili et al. [12] concluded that there are no insufficient experimental data related to heat transfer and energy transport inside liquid enclosures that packed with solid materials.

However, the authors of this paper did not find in the literature any work related to thermal response of liquid enclosure packed with metal material and subjected to heat flux at the same time.

On the other hand, and as far as known, laboratory measurements of natural heat transfer coefficient in conventional liquid enclosures that subjected to heat flux are very limited in open literatures. Actually, it has been noted that there are only a few simple correlations for limited boundary conditions [13-20]. Further, unlike the conventional liquid enclosures, there is a lake for heat transfer correlations that related to liquid enclosures fitted with solid materials and subjected to heat flux at the same time. However, these seem to be proprietary data.

Keeping the above in view, the main objectives of the present research are to:

1- Provide knowledge of the thermal effects of adding several new metal rings inside liquid enclosure that subjected to high heat flux. The rings should be suitable to incorporate into existing liquid enclosures. Also, these metal rings should be in good thermal contact with the inner enclosure wall to increase the enclosure ability to absorb and store more thermal energy from the walls.
2- Develop a two-dimensional numerical model based upon the conservation equations of mass, momentum and energy to represent the transient flow and thermal fields within the liquid enclosure during both static (with no liquid flow) and dynamic (with liquid flow) modes of operation. The developed model should take in consideration the thermal interaction between the liquid and the solid materials of the enclosure.
3- Test and validate the developed numerical model against a directly measured experimental data of a laboratory scale liquid enclosure that is made locally for this purpose. The enclosure should be exposed to heat flux and packed with several metal rings.
4- Formulate suitable correlations for heat transfer coefficient inside liquid enclosure that subjected to high heat flux and fitted with new arrangement of horizontal rings (at the same time).

Numerical Model :

The numerical model proposed in this work consists of cylindrical vertical tank as shown in Figure (1) whose geometrical aspect ratio is (H/2R), and is subjected to heat flux. The governing equations in the enclosure are the mass continuity equation, the momentum equations in the axial and radial directions, the energy equation of the liquid and an additional energy equation for the solid material of the enclosure. Assuming the working fluid is incompressible, Newtonian and two-dimensional, the transient conservation equations can be written as:

$$\frac{\partial\rho}{\partial t}+\frac{\partial}{\partial x}(\rho.u)+\frac{1}{r}\frac{\partial}{\partial r}(\rho.v.r)=0 \qquad \text{...............} \quad (1)$$

$$\frac{\partial\rho}{\partial t}(\rho_L\,.u)+\frac{\partial}{\partial x}(\rho_L\,.u^2)+\frac{1}{r}\frac{\partial}{\partial r}(\rho_L\,.r.u.v)= -\frac{\partial P}{\partial x}+\frac{\partial}{\partial x}\left(\mu\frac{\partial u}{\partial x}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(\mu.r.\frac{\partial u}{\partial r}\right)-\rho_L\,.g \qquad \text{...............} \quad (2)$$

$$\frac{\partial}{\partial t}(\rho_L\,.v)+\frac{\partial}{\partial x}(\rho_L\,.u.v)+\frac{1}{r}\frac{\partial}{\partial r}(\rho_L\,.r.v^2)= -\frac{\partial P}{\partial r}+\frac{\partial}{\partial x}\left(\mu\frac{\partial v}{\partial x}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(\mu.r.\frac{\partial v}{\partial r}\right)-\mu.\frac{v}{r^2} \qquad \text{................} \quad (3)$$

$$\frac{\partial}{\partial t}(\rho_L\,.Cp_L.T)+\frac{\partial}{\partial x}(\rho_L\,.Cp_L.T.u)+\frac{1}{r}\frac{\partial}{\partial r}(\rho_L\,.Cp_L.T.v.r)= \frac{\partial}{\partial x}\left(K_L\,.\frac{\partial T}{\partial x}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(K_L\,.r.\frac{\partial T}{\partial r}\right) \qquad \text{................} \quad (4)$$

$$\frac{\partial}{\partial t}(\rho_S\,.Cp_S.\overline{T})-\frac{\partial}{\partial x}\left(K_S\,.\frac{\partial T}{\partial x}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(K_S\,.r.\frac{\partial T}{\partial r}\right) \qquad \text{................} \quad (5)$$

Equation (5) is an additional energy equation in the present improved numerical model used to include the solid material effect at different enclosure heights.

For static mode (i. e. without liquid flow), the initial condition is null velocities in whole enclosure when the hot water in the upper part and the cold water in the lower part. The boundary conditions are no–slip condition at symmetry line, lateral wall and enclosure top and bottom. The heat gain to the liquid is through lateral walls and enclosure top. The bottom of the enclosure is considered perfectly insulated.

For dynamic mode simulations, the liquid is initially at a prescribed temperature profile with the entry of hot or cold liquid during charging or discharging processes. The boundary conditions are the same for the natural heating simulations, but considering null shear stress at the symmetry line.

The boundary conditions for the momentum and energy equations, at the inlet jet, were (u = u_{in}) ; (v = 0) and (T = Tin). At the outlet jet, boundary conditions were ($\partial u/\partial x$ = 0) ; v = 0 and ($\delta T/\delta x = o$). In simulations, the boundary condition symmetry is applied along the vertical line (r = o), therefore, the calculations domain was just half the physical body.

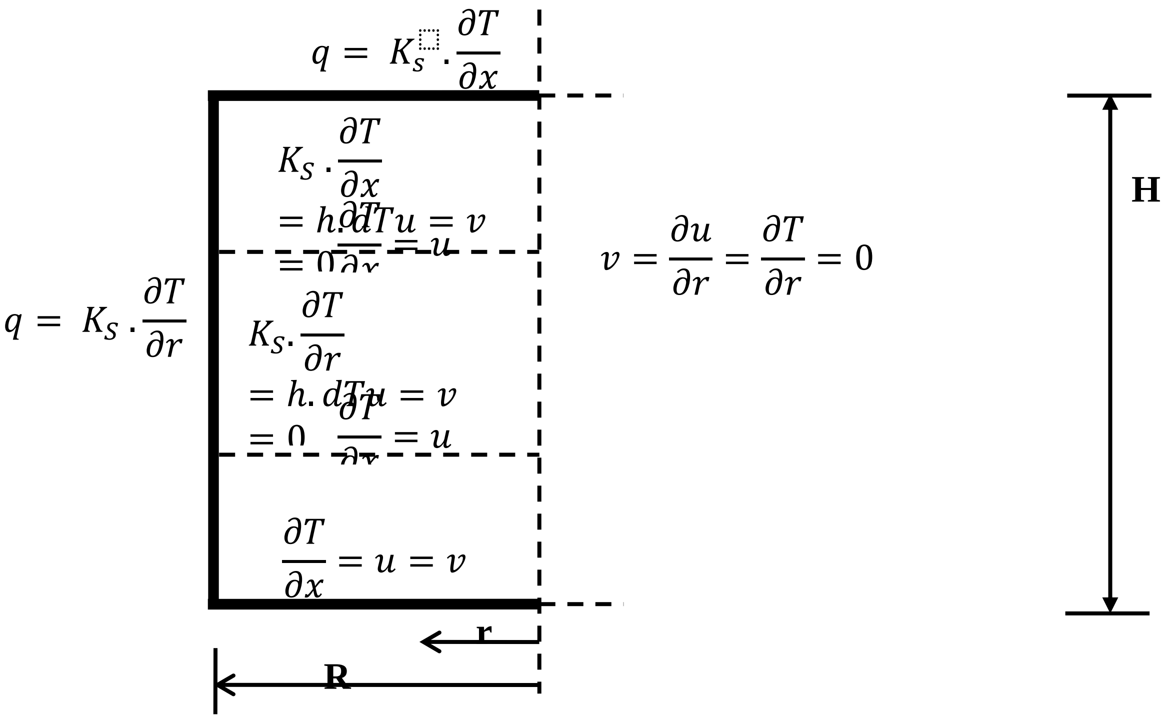


Figure (1): Boundary conditions of the mathematical model.

The above model requires solution of the conservation of the mass momentum and energy equations for the multiple zone enclosure the separated by multi-horizontal metal circular rings. The above transient equations are solved using the finite volume method in cylindrical structured mesh. To solve the algebraic linear equations resulting from the discretization of the governing equations, the Tri-Diagonal Matrix Algorithm (TDMA) was employed.

The solution of the transient problem was carried out in a completely implicit form. For each time step, the iterative process was applied until reaching the convergence criterion. Solutions are assumed to converge when the following convergence criterion is satisfied by every dependent variable at every grid point in the computational domain;

$$\left|\frac{\Phi_{new} - \Phi_{old}}{\Phi_{new}}\right| \le \Phi$$

Where Φ in general could be any dependent variable. In this work, Φ is less than (10^{-3}) for continuity and momentum equations and Φ is less than (10^{-6}) for energy equations. The simulations run with a time step of (1-3) seconds.

In the formulation of the mathematical model, the physical properties of the water (ρ, μ, K) are considered uniform in space and updated in time as a function of water average temperature, ($T_{av.}$) in K.

Density, [kg/m^3] $\qquad \rho = 863 + 1.21 * T_{av.} - 0.00257 * T_{av.}^2$

Dynamic viscosity, [kg/m.s]

$$\mu = 0.0007 * \left(\frac{T_{av.}}{315}\right)^{-5.5}$$

Thermal conductivity, [W/m.K] $\qquad \mathrm{K} = 0.375 + 8.84 \times 10^{-4} * T_{av.}$

Experimental Work:

In this work, thermal behavior of water enclosure subjected to high heat flux is investigated experimentally. The tests were preformed to obtain the temperature profile inside the enclosure during both static (stagnation) and dynamic (charging/discharging) modes of operation. Figure (2) shows the experimented apparatus.

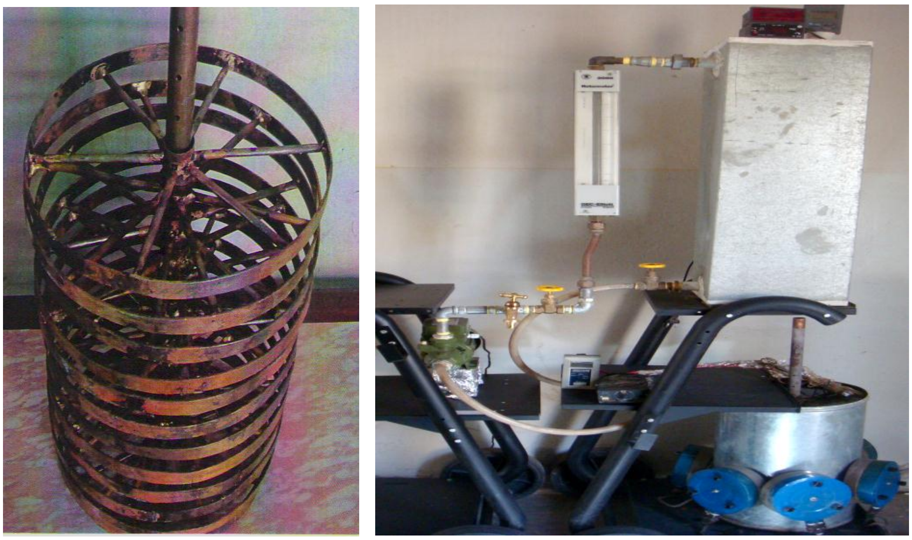
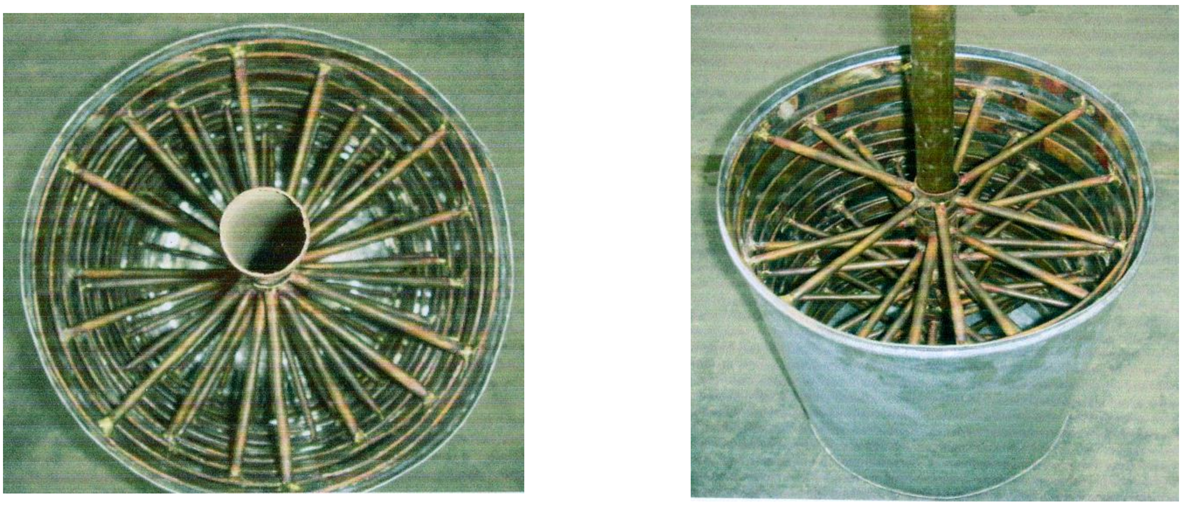


Figure (2): Experimental setup.

The test section of the present experimental work consists of a vertical cylindrical water enclosure with an immersed, removable, screen rings fitted inside the enclosure. The laboratory scale enclosure is constructed from galvanized steel with an inside diameter of (305) mm and a height of (550) mm. The main purpose of using metal screen rings is to increase the heat transfer from the external enclosure wall to the inside water by conduction. In addition, these screens packing should provide a torturous path for the water flow through the enclosure. The system of removable screen packing includes up to twelve horizontal rings through and around which the storage water should pass. As shown in Figure (3), the screen ring packing is fitted inside the enclosure in two arrangement systems (in–line and staggered).



(a): In-line arrangement (b): Staggered arrangement

Figure (3): Removable in–line and staggered screen ring packing fitted inside the enclosure.

During the experimental measurements, the external wall of the enclosure is heated by electric heating elements to a uniform wall temperature. These heating elements should simulate the solar energy heat flux. The water enclosure is covered by an insulated box, so there is no heat loss from it. Therefore, the heat gain to the enclosure is equal to the power of the electric heating elements. The effect of heat gain on thermal gradient inside enclosure will be investigated experimentally. The temperature range investigated of approximately 28°C to 87°C .

The experiments were divided into two categories:

I) Static experiments (without water flow).

II) Dynamic experiments (with charging /discharging water flow).

In the first experiments the water was heated until a specified temperature was reached, and the temperature was monitored during this process. The temperature measurements were taken with 5-min intervals along three hours. In later experiments, after reaching the specified temperature, hot water was discharged at constant mass flow from the top of the enclosure whereas an identical quantity of cold water was charged through its base. The temperature profiles were also measured along the process. After a predefined volume of cold water was charged and the thermocline formed, the charge/discharge process was interrupted, and the degradation of the thermocline zone was observed by monitoring the temperature profiles.

The filling of the tank produces fluid motion that persists for some time and may affect the results. It has been determined that after about two minutes of waiting time between fillings and test running, the detrimental vertical motion had dissipated. In general, tests were started after five minutes of filling and the decay in the temperature distribution beyond this point is studied. The detailed of the thermocouple distribution inside the water enclosure are shown in Figure (4).

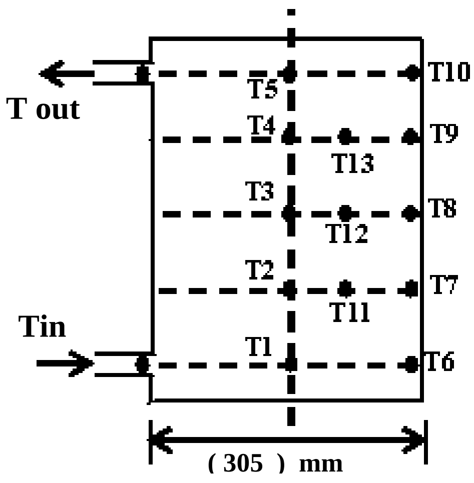


Fig. (4): Locations of the thermocouples inside the water enclosure.

The thermocouples were located at fixed distances of 10 cm intervals along a vertical plane and were attached to a vertical rod, which was inserted into the enclosure at the top. The positions of the thermocouples were chosen based on previous numerical simulation of the temperature profiles. The highest radial temperature gradients occur in the middle part of the enclosure. For this reason, thermocouples were more concentrated in this region, as shown in Figure (4).

Thirteen calibrated thermocouples are distributed at five levels within the enclosure to measure the water temperature. Besides the thirteen thermocouples, twelve others were used; two to measure the water temperature at inlet and outlet of the enclosure, three to measure the water temperature inside the auxiliary water tank, six to measure the external wall temperature and another one for the external environment, where the experiments were performed. The experimental rig has been equipped with an acquisition data system based on a PC to continuously record the temperature values.

Experiments were performed three to four times to check the repeatability and accuracy of the measurements taken. It was found that the temperature data could be repeated to within 4%, indicating a fairly high level of accuracy.

Results and Discussions:

The thermal behavior during static and dynamic operation conditions are important issues and both should be investigated. Static mode represents the most frequent state of the enclosure and dynamic mode has great relations with real time analysis.

The temperature distributions of water in the enclosure that subjected to uniform heat flux are investigated experimentally and theoretically during both static mode of operation (Fig. 5) and dynamic mode of operation (Fig. 6). In general the numerical simulation was able to accurately reproduce the time–dependent heating process in the enclosure. The temperature profiles of the enclosure were examined during 180 min. period with an initial uniform temperature of 31 °C. For both static and dynamic modes of operation, the CFD calculations give an underestimated temperature at the middle of the enclosure height and an overestimated temperature at the top and bottom of the enclosure. However, the numerical and experimental results showed that there is a slight temperature decrease in the middle of the tank height, while a strong thermal stratification exists only at the lower part with a temperature decrease to the bottom of the enclosure.

It can be seen that the numerical model predicts well the transient temperature distribution in the tank during both dynamic and static modes of operation except water at the peak part of the enclosure, where the numerical results were higher than experimental ones.

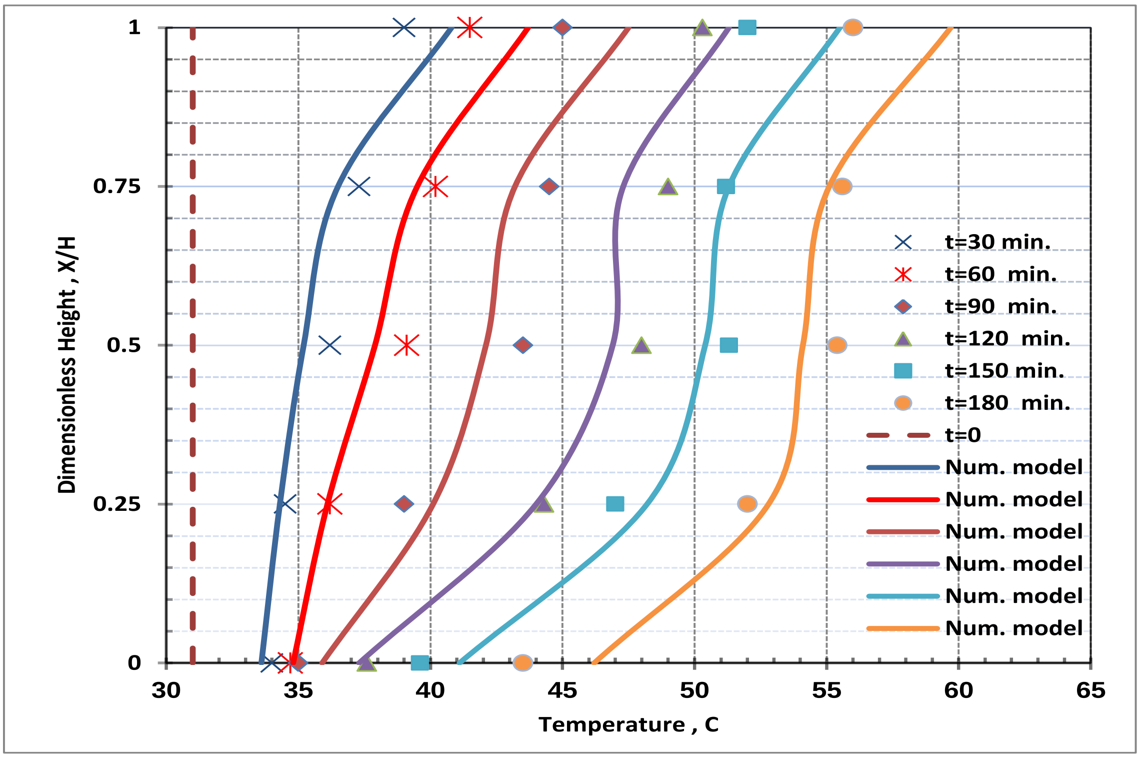


Figure (5) : Comparison between the numerical model temperature profiles and experimental data during static mode of operation.

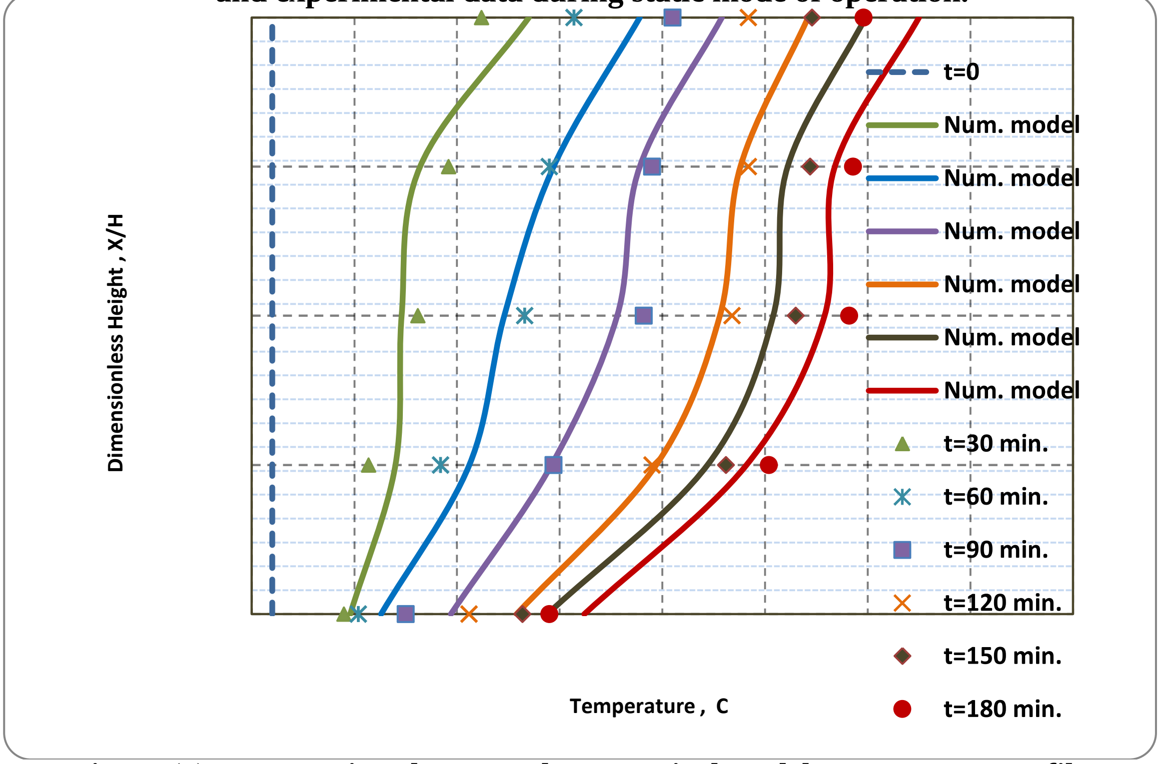


Figure (6) : Comparison between the numerical model temperature profiles and experimental data during dynamic mode of operation.

The influence of ring arrangement is investigated for its potential for enhancing radial conduction heat transfer to the centerline of the enclosure. The impact of adding 3-rings in-line and 3-rings staggered arrangements on axial water temperature profile is shown in Figure (7) during static mode and in Figure (8) during dynamic mode of operation. For the static mode, the thermal response of no rings model is found to have a similar behavior to that for in-line rings and staggered rings but with increase in temperature when adding rings.

However, the enhancement in water temperatures is nearly equal in the enclosure with in-line and staggered arrangements as shown in Figure (7).

For the dynamic mode (Figure 8), the thermal response with rings is clearly different from tank with no rings. However, the in-line arrangement gives higher temperature profile for most of the enclosure height. On the other hand, horizontal rings with staggered configuration add substantial hydraulic resistance to the moving fluid during the dynamic mode. Therefore, it is concluded that using in-line arrangement is better than using staggered rings. During the dynamic mode, an obvious temperature gradient or thermocline has been formed between hot water at the top and cold water in the bottom. It is seems that the thermal stratification within the tank have been achieved because of horizontal rings that increasing the radial heat transfer from the vertical tank wall.

In this work, the water mass flow rate of 1, 2 and 3 liters per minute have been studied. For the studied mass flow rates, the increase of inlet jet velocity do not modifies significantly the thermal response profiles. Therefore, only results with mass flow rate of 2 L/min will be shown up. Also, it may be concluded that in solar energy systems the operational mass flow rate can be increased without damage on stratification profiles. It must be mentioned that the diameter of the inlet and outlet jets have a diameter of ¾ inch, so that, for this mass flow rate, the flow is laminar.

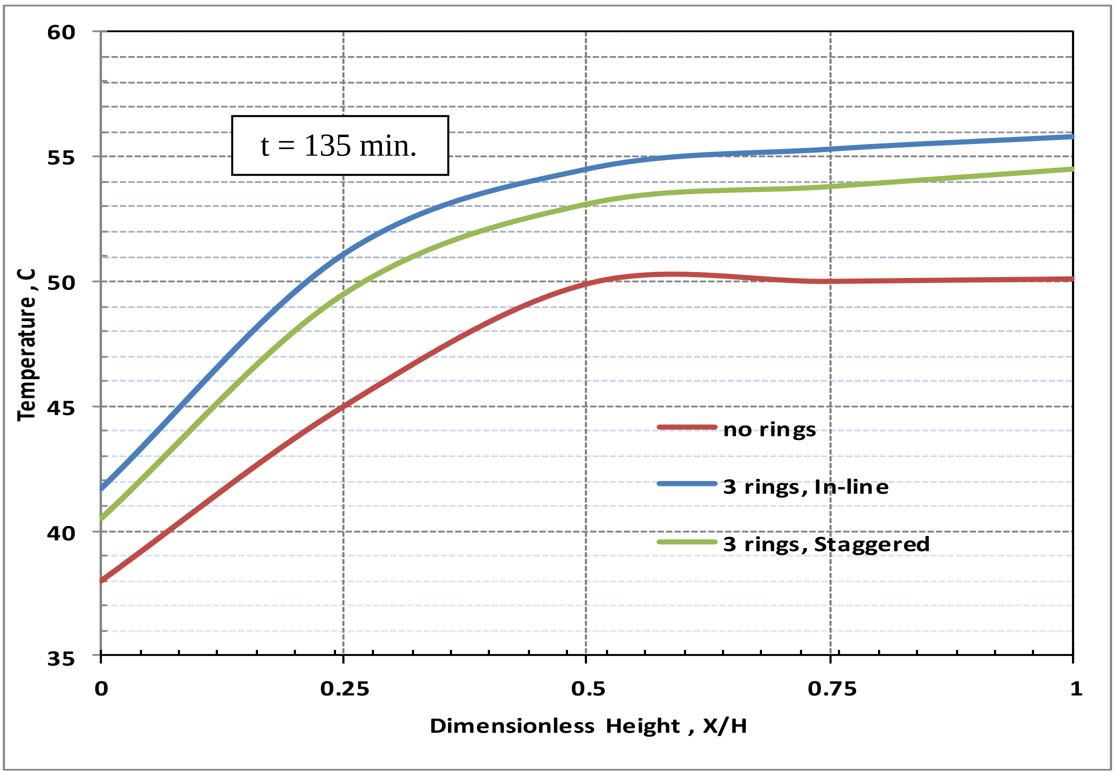


Figure (7) : Effect of metal rings arrangement on the water temperature distribution during static mode of operation.

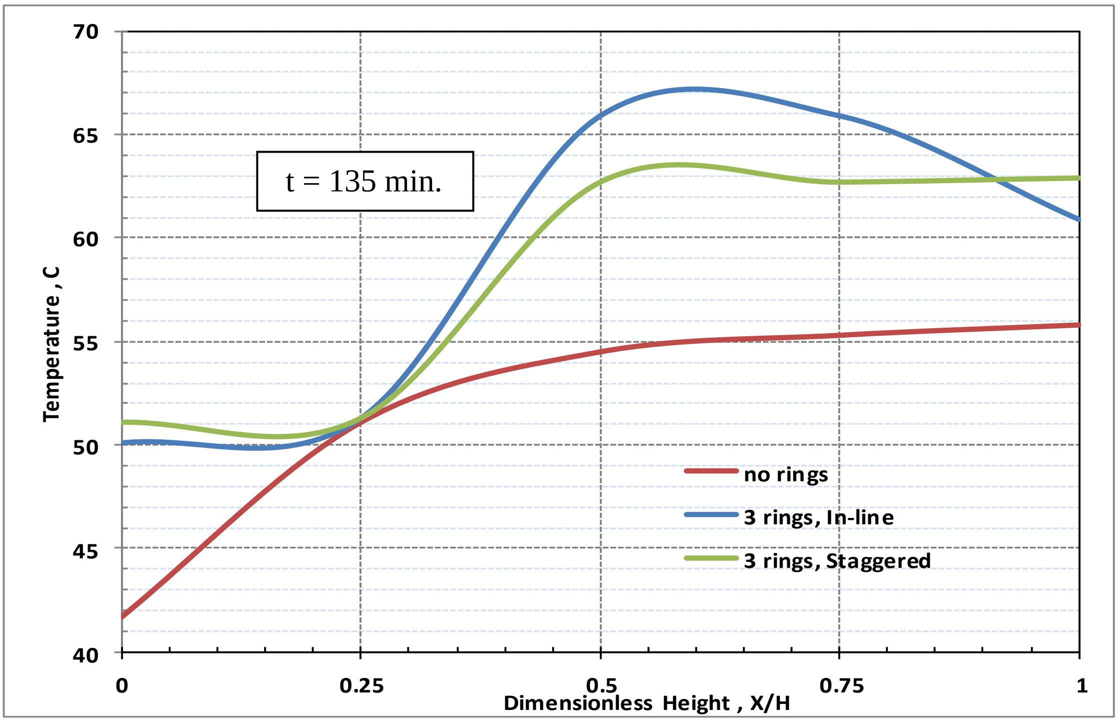


Figure (8) : Effect of metal rings arrangement on the water temperature distribution during dynamic mode of operation.

Figure (9) and (10) show the effect of adding metal rings on the transient temperature distribution at center-line and near wall during static mode (Fig. 9) and during dynamic mode (Fig. 10). In these figures data with and without the rings are compared. It is clear from these figures that water enclosure with no rings behaves differently from that contains metal rings, that is the rings have a significant impact on the center line temperature distribution. The horizontal rings are produce irregular, turbulent flows that additionally enhance heat transfer, and therefore creates rise in the water temperature of the enclosure after about 5 min.

An indication of the complexity of the temperature profiles during the static mode can be seen in Figure (9). It is clear from this figure that using (12) rings is useful only after about 45 min.. However, using 3 and 6-rings give a similar response in enhancing the heat transfer but the 6 rings model produce high fluctuation in the temperature profile. With the progress of time, it's noticed that the 3-rings model always enhance the heat transfer as compared to tank without rings.

During dynamic mode, as can be seen in Figure (10), the general trends are the same but the differences between the predictions increase with the increase in time. The use of (3) and (6) metal rings led to a greater degree of enhancing conduction heat transfer. Using (6) rings inside the enclosure produce high fluctuation in the temperature profile. This secondary effect may be due to a local instability in the flow near wall (such as a transition to turbulence) which causes the hot, near wall, fluid to be injected into the center line at intermediate levels along the enclosure height.

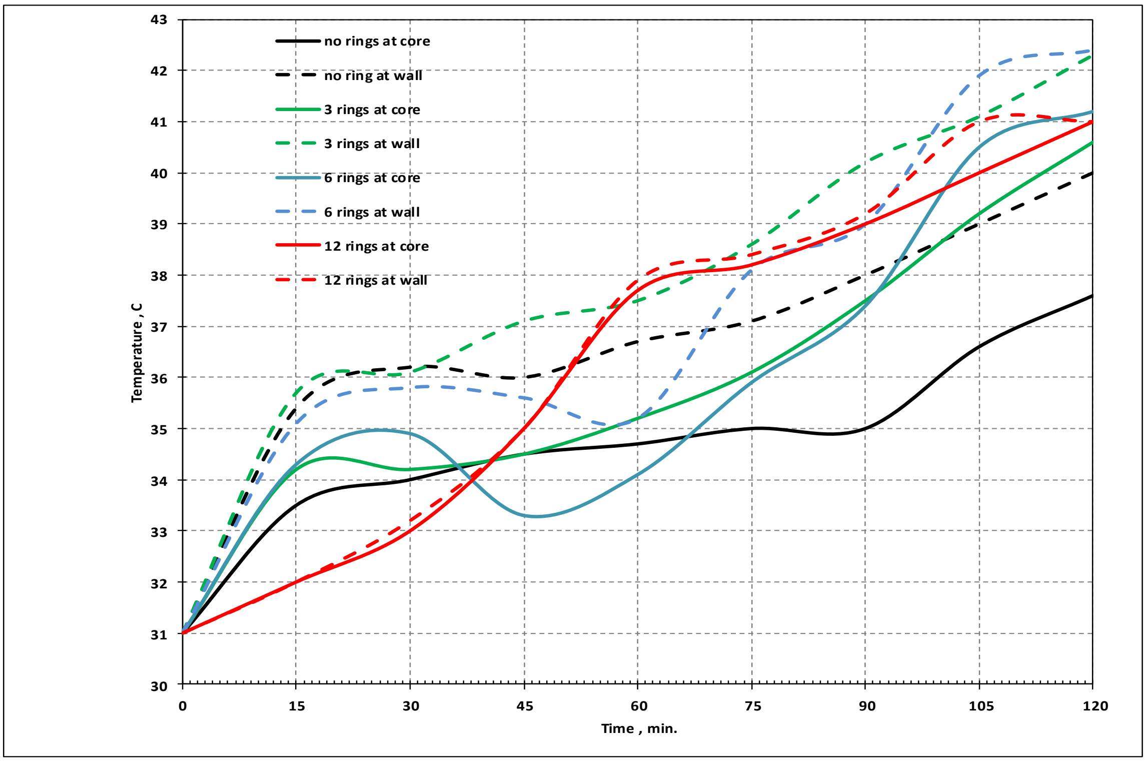


Figure (9) : Effect of adding horizontal rings on water transient temperature at center-line and near-wall during static mode of operation.

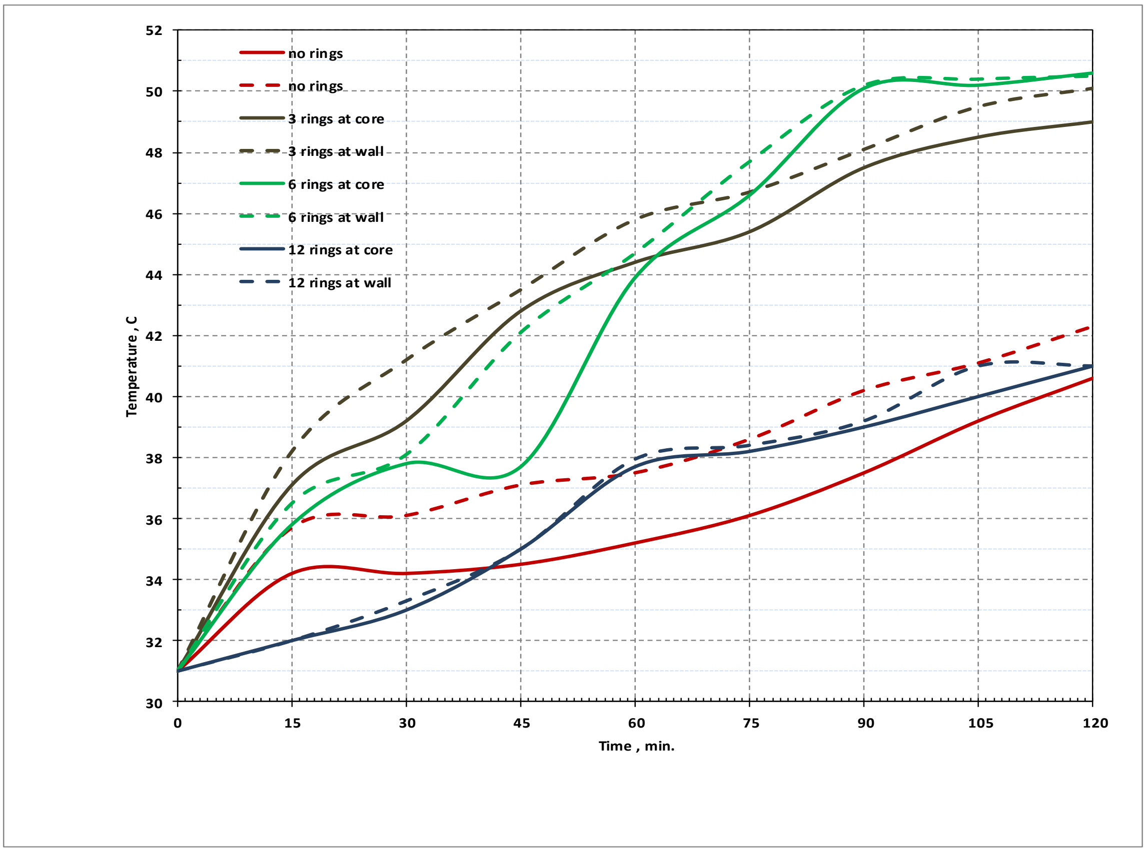


Figure (10) : Effect of adding horizontal rings on water transient temperature at center-line and near-wall during dynamic mode of operation.

It is clear that there is thermal boundary layer develops initially along the heated walls. Due to this heat gain to the conducting side-wall, the fluid close to the enclosure wall has a higher temperature than the fluid at the center of the enclosure. The relative hotter fluid flows up along the tank wall while the fluid with lower temperature flows downwards. It is noticed that at the beginning of heating the side wall water temperature begins to rise immediately, but the center–line temperature does not begin to change until warm water sinks to its level .

In general, it is found that using horizontal metal rings led to less difference between near–wall and center-line temperature due to increasing radial heat conduction from side-wall to the core water. It is clear that using 12 rings make this difference very low during both dynamic and static modes. However, when using (3) or (6) rings this difference became higher but with an increase in the main water temperature, especially during dynamic mode. From all above, it is concluded that using 6-rings model is more useful during both static and dynamic modes for enhancing radial heat conduction from side-wall to water.

Figure (11) and (12) show the temperature distribution along the enclosure radius during static and dynamic modes of operation. As expected, the water close to the conductive metallic wall has a higher temperature than the fluid at the center of the enclosure due to heat gain at wall. The relative hotter fluid close to the wall rises up along the wall, while in the center of the tank, the water with lower temperature flows down to a lower level. During the static mode (Fig.11), it is observed that the center-line temperature is much lower than the near-wall temperature for the case of no metal rings. In the other side, it is clear that there was a very little radial temperature variation during the dynamic mode for the case of no rings. It seems that the side-wall fluid is mixed toward enclosure core due to dynamic motion of water. In general, for both modes of operation, the presence of positive horizontal metal rings increases the radial heat conduction from side-wall to center-line. It is observed that using 3 and 6 rings provide higher radial conduction during static mode. It is found that using 6 rings during dynamic mode provide the higher temperature rise for main enclosure core among the three considered models.

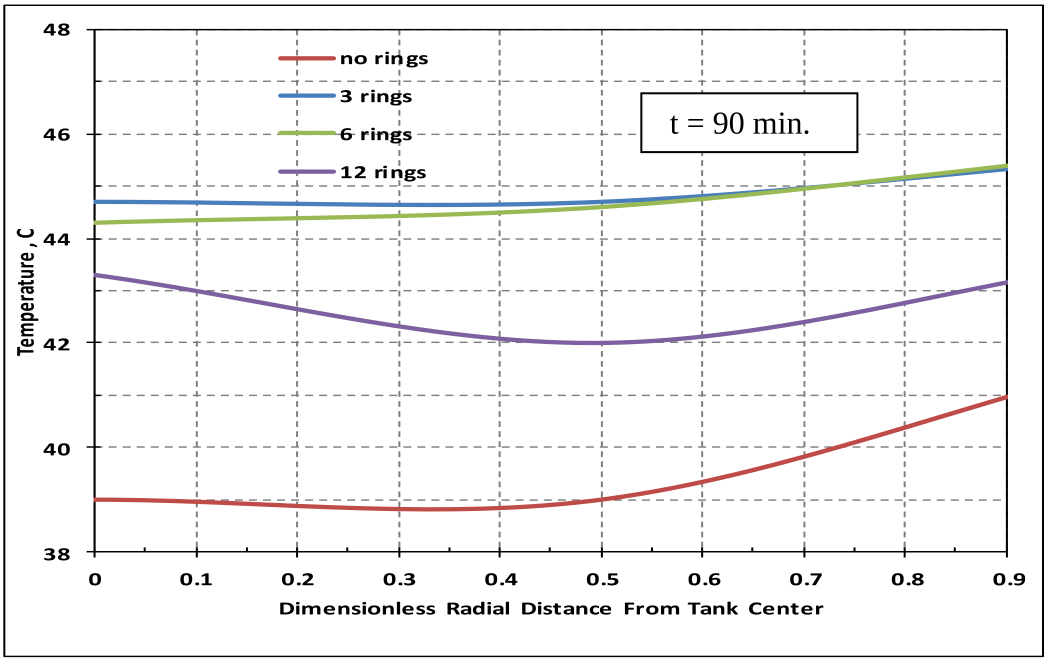


Figure (11) : Radial temperature distribution during dynamic mode .

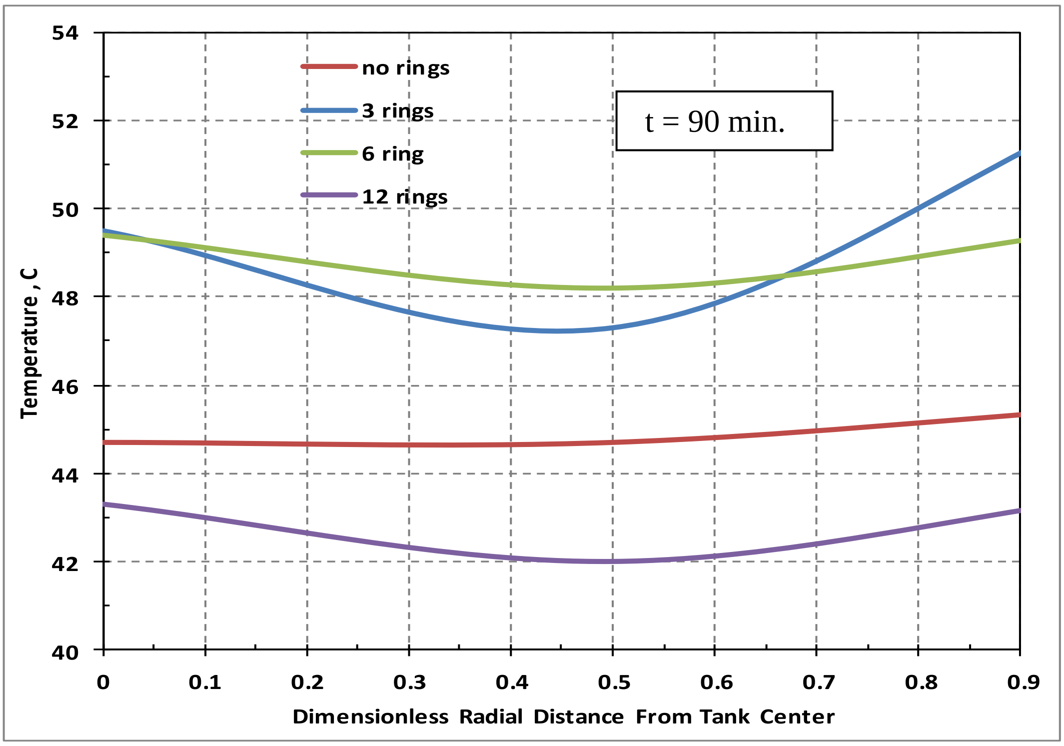


Figure (12) : Radial temperature distribution during dynamic mode .

<u>Predication of laminar natural Nusselt number correlations :</u>

A laminar natural convection inside vertical cylinder was mainly investigated by numerical methods of solving the Navier-Stokes and energy equations [13]. Different heat transfer open literatures and textbooks present empirical correlations for natural convection heat transfer that related to limited boundary conditions [13-20]. These correlations usually account for steady-state processes and comprise cases of either constant surface temperature or constant heat flux. However, in practice, none of these theoretical cases normally occurs. Seara et al.[21] stated that data and correlations to calculate transient natural convection coefficients obtainable in textbooks or in more specific open literature are quite different from the experimentally data. Therefore, they conclude that more research should be carried out in this area.

In this work, it is evident from results that the predication produce by the present numerical model are reasonable in general, however, it is obvious that there are some discrepancy between the experimental data and the computational results for the static mode of operation (Fig. 5). This discrepancy is believed to be due to the effect of natural Nusselt number correlation used in the numerical model. Therefore, the experimental results will be used to formulate new suitable correlations for natural heat transfer inside liquid enclosures subjected to high heat flux.

The local heat transfer coefficient of water inside the enclosure (h_x) is defined as;

$$h_x = K / \Delta T \ . \ \partial T / \partial r \,|_r$$

The reference temperature difference (ΔT) is the difference between the average temperature of the cross-section of the enclosure (Tav_x) and the side-wall temperature ($T_{x,R}$) at the same height ;

$$\Delta T = T_{x,R} - Tav_x$$

Where, $Tav_x = 1/\pi \, . \, R^2 \cdot {}_o^R\!\int T\,|_x .2\,\pi\, r\, .dr$

So, the Nusselt number is calculated by;
$Nu_x = h_x \, . \, x \; / \; K$

Qureshi and Gebhart [22] have proposed a range of ($Ra = 1.2 \times 10^{13} - 4 \times 10^{13}$) for the start of transition from laminar to turbulent flow for water tank with uniform heat flux. Holzbecher and Steiff [17] observed that the transition to turbulent regime is at ($10^{13} < Ra < 10^{14}$). So in this work, the laminar Rayleigh number is taken below 10^{13} .

The mean Nusselt number as a function of enclosure height is shown in Figure (13) at $Ra = 5.1 \times 10^9$. Two models are considered, water enclosure with no-rings and enclosure fitted with 6-rings inside it. The effect of metal rings is clear for enhancing the free convection inside water tanks subjected to high heat flux. However, the lower heat transfer coefficient may be expected near the enclosure bottom for both cases due to lower temperatures.

The results of the calculations are presented in Figure (14) for both no-ring model and 6-ring model. The results of the predicted Nusselt correlations are of a power fit to the data, and may be expressed for no-ring model as ;

$$\mathbf{Nu = 29.98 \times Ra^{0.0975} \qquad at \quad 8 \times 10^7 \leq Ra \leq 5 \times 10^{10}}$$

and for six-ring model as ;

$$\mathbf{Nu = 125.5 \times Ra^{0.0633} \qquad at \quad 8 \times 10^7 \leq Ra \leq 5 \times 10^{10}}$$

As shown in figure (14), for ($Ra \leq 5 \times 10^8$) the dependence of $Nu = f(Ra)$ is found to tend to the value of (Nu = constant) corresponding to thermal conductivity of motionless liquid. The Nusselt number is found to fluctuate about some average value of 200 for no-ring model and 400 for 6-ring model. The validity of the predicted correlations determined in this work is shown in Figure (14) by comparing them with heat transfer correlation available in open literatures (dashed lines). To the best of authors' knowledge, no previous work has determined natural Nusselt number correlation for liquid enclosure fitted with solid material and subjected to heat flux at the same time. So, no comparison with other results can be done.

The improvement in natural Nusselt number of liquid enclosure subjected to heat flux and fitted with 6-rings arrangement is found to be about two times the value for conventional liquid enclosure with no-rings.

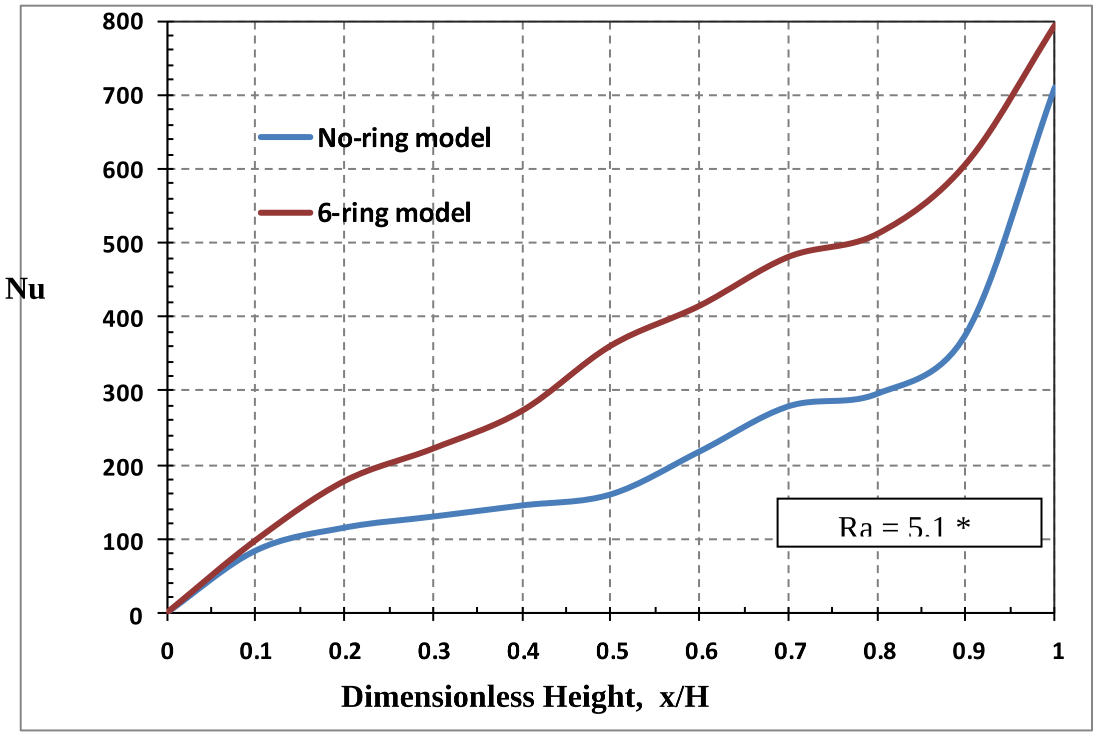


Figure (13) : Relation between vertical height with predicted Nusselt number inside water enclosure subjected to high heat flux.

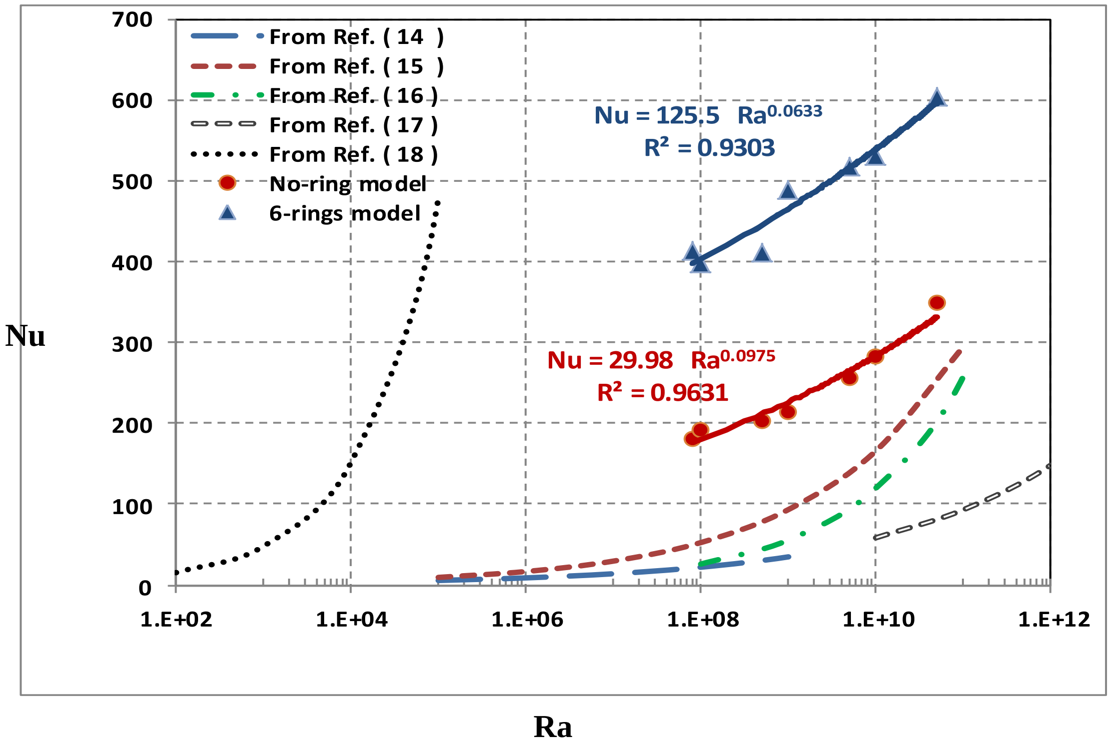


Figure (14) : Comparison between predicted Nusselt number correlations with available open literature correlations.

Conclusions :

In this work, various metal rings were added inside vertical cylindrical enclosure that subjected to high heat flux. The primary objective of the rings is to gain more heat transfer to the liquid inside enclosure. The transient temperature distributions of water were investigated experimentally and theoretically during both static and dynamic modes of operations. The two-dimensional computational code was based upon the conservation equations of mass, momentum and energy, and developed by adding fourth equation that take in consideration energy transport between liquid and the solid material of the enclosure.

The predicted profiles have been found close to those obtained experimentally, except for water at peak part of the enclosure, where the numerical results were higher than experimental ones. The developed numerical simulation has been provided a better understanding and planning of the experimental tests.

Adding metal rings has been found to produce a significant enhancement in radial heat transfer from side-wall to the main liquid inside enclosure. The results indicated that using in-line arrangement is better for enhancing heat transfer than using staggered rings. It has been found that the optimum number of rings is six. This six-ring model is found to be more effective for enhancing heat transfer as compared to the other cases under investigation during both static and dynamic modes of operation.

In order to provide the practical engineers with more reliable Nusselt number correlations, two new correlations for natural heat transfer inside liquid enclosures subjected to high heat flux have been formulated as ;

$$Nu = 29.98 \times Ra^{0.0975} \quad , \text{(for no-ring model)}$$

and, $$Nu = 125.5 \times Ra^{0.0633} \quad , \text{(for six-ring model).}$$

The new correlations are applicable for local Rayleigh number within the rang of (8×10^{7} to 5×10^{10}). The validity of the predicted correlations have been compared to correlations available is open literatures. The natural Nusselt number is found to be around a constant value for Rayleigh number below (5×10^{8}). The improvement in natural Nusselt number of liquid enclosure with six-ring model is found to be about two times the value for conventional liquid enclosure with no-rings.

The recommended use of metal rings inside liquid enclosures subjected to heat flux, and the predicted Nusselt number correlation related to it, will add to local knowledge a significant mean to gain more heat in large scale concentrated solar power plants.

References:

1- Dehghan, A. and A. Barzegar, "Thermal performance behavior of a domestic hot water solar storage tank during consumption operation", Energy Conversion and Management , 52, 2011.

2- Abdulsada, Ghanim. K and Mohammed H. Mahmoud, "Enhancing Thermal Stratification in Liquid Storage Tanks During Relaxation Periods", Engineering and Development Journal, Vol. 12, No.1, 2008.

3- Akinjiola, O. P. and U. B. Balachandran, "Concentrated Solar Thermal (CST) system for fuelwood replacement and household water sanitation in developing countries", Journal of Sustainable Development; Vol. 5, No. 6, 2012.

4- Mc Govern R. and W. Smith, "Optimal concentration and temperatures of solar thermal power plants", Energy Conversion and Management, 60, 2012.

5- Shyu RJ. Lin JY. And Fang LJ., "Thermal analysis of stratified storage tanks", ASME J. Solar Energy Eng., 111, 1989.

6- Hariharan K., and K. Badrinarayana, "Temperature stratification in hot water storage tanks", Energy, 16 (7), 1991.

7- Zachar A. Farkas I. and Szlivka F., "Numerical analyses of the impact of plates for thermal stratification inside a storage tank with upper and lower inlet flows", Solar Energy, 74, 2003.

8- Altuntop, N., M. Arslan, V. Ozceyhan and M. Kanoglu, "Effect of obstacles on thermal stratification in hot water storage tanks", Applied Thermal Eng., Vol. 25, 2005.

9- Haltiwanger, J. and J. Davidson, "Discharge of a thermal storage tank using an immersed heat exchanger with an annular baffle", Solar Energy, Vol. 83, pp 193-201, 2009.

10- Han, Y., R. Wang and Y. Dai, "Thermal stratification within the water tank", Renewable and Sustainable Energy Reviews, Vol. 13, pp 1014-1026, 2009.

11- Mawire, A. and S. H. Taole. "A comparison of experimental thermal stratification parameters for an oil/pebble-bad thermal energy storage (TES) system during charging", Applied Energy, 88, 2011.

12- Valmiki, M., W. Karaki, P.Li, J. Lew, C. chan and J. Stephens , "Experimental investigation of thermal strong processes in a thermocline tank", Journal of Solar Energy Engineering, Vol. 134, 4 , July 2012.

13- Martynenko, O. and P. Khramtsov, Free Convection Heat Transfer, Springer Berlin Heidelberg, 2005.

14- Polezhaev, V.I., " Non-stationary Laminar Heat Convection in a Closed Domain for a Given Heat Flux", Nlekh. Zhidkosti Gaza, No.4, 109, 1970.

15- Hiddink, J, J. Schenk and S. Bruin, "Natural convection heating of liquids in closed container", Appl. Sci. Res., 32 , 217 , Aug. 1976.

16- Goldstkin, R. and S. Tokuda, "Heat transfer by thermal convection at high Rayleigh numbers", Int. J. Heat Mass Transfer, 23, 5, 1980.

17- Holzbecher, M. and A. Steiff, "Laminar and turbulent free convection in vertical cylinders with internal heat generation", Int. J. Heat Mass Transfer, 38, 15, 1995.

18- Lemembre, A. and J.P. Petit, "Laminar natural convection in a laterally heated and upper cooled vertical cylindrical enclosure", Int. J. Heat Mass Transfer, 41, 16, 1998.

19- Bejan, A. and A.D. Kraus, Heat Transfer Handbook, John Wiley and Some Inc., USA, 2003.

20- Incropera, F.P., D.P.De Witt, T. Bergman and A. Lavine, Fundamentals of Heat and Mass Transfer , 7th edition , John Wiley and Sons Inc., USA , 2011.

21- Seara, J., F. Uhia and A. Dopazo, "Experimental transient natural convection heat transfer from a vertical cylindrical tank", Applied Thermal Eng., Vol.31, Aug. 2011.

22- Qureshi, Z.H. and B. Gebhart, "Transition and transport in buoyancy driven flow in water adjacent to a vertical uniform flux surface", Int. J. Heat Mass Transfer, 21, 1978.

Nomenclatures:

Cp : specific heat at constant pressure
D :diameter of the enclosure
G : gravitational acceleration
Gr : Grashof number
H : convection heat transfer coefficient
H : height of the enclosure
H/D: aspect ratio
K :thermal conductivity
Nu :Nusselt number
P :pressure
Pr :Prandtl number
q : heat flux
r : radial coordinate
R : radius of the enclosure
Ra : Rayleigh number
t : time
T : Temperature
u : velocity in x-direction
v : velocity in r–direction
x : Axial direction

Greek Letters:

α : thermal diffusivity
β : thermal expansion coefficient
μ : dynamic Viscosity
ρ : density
ν : kinematic viscosity

Subscripts:

L : Liquid
S : Solid