Natural convection heat transfer inside vertical cylindrical solar storage tanks

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Abstract

Natural convection inside a stratified solar storage tank significantly contributes to the rate of heat loss from the tank. However, only a limited number of studies in the literature have investigated the means to predict convection heat transfer coefficient inside the tank. This study presents a numerical investigation of natural convection heat transfer inside a vertical cylindrical storage tank. The simulated results were then used as a benchmark to assess the suitability of existing flat plate correlations to predict the rate of heat loss from such tanks. The results showed that using flat plate correlations lead to deviations in rate of heat loss with up to 23% compared to the numerical results. These findings demonstrate that there is still a need to develop a correlation that can accurately predict convective heat transfer coefficient inside cylindrical storage tanks.

Keywords: Internal natural convection; Internal heat generation; Solar storage tanks; Heat loss; Heat transfer coefficient; Flat plate correlations

1. Introduction

One of the main factors that reduces the thermal performance of a solar water heating system is the heat loss from solar water storage tanks. This is particularly the case when the tank is in standby mode at night, due to lower ambient temperatures compared to the day (Yang et al., 2019). Therefore, the ability to predict the rate of heat loss is vital to effectively optimise the performance of such systems.

Static heat loss to the ambient takes place through a series of heat transfer modes, as shown in Figure 1. The first resistance represents the natural convection heat transfer inside the tank due to temperature gradient between the water and the wall. Heat received from the water is then conducted through the tank wall and insulation before it is lost to the ambient. Although the thermal resistance to conduction through the tank wall and insulation can be easily estimated using Fourier's law (Cengel, 2007), the natural convection heat transfer resistance between the water and the wall is difficult to predict since there are only a limited number of studies that exist on understanding its relation to the rate of heat loss.

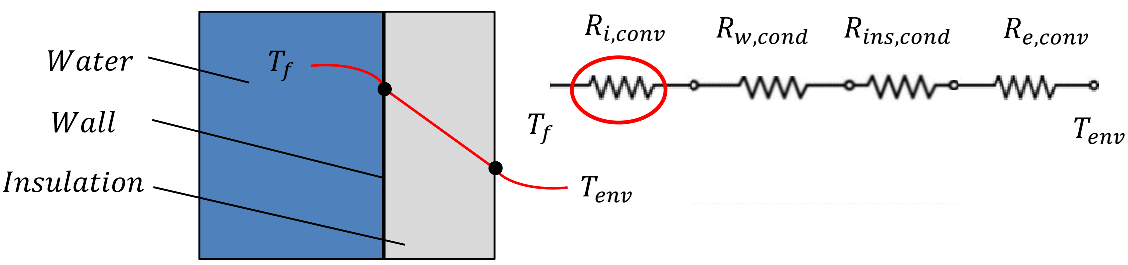

Fig. 1: Thermal resistance network for static heat loss to the ambient

The contribution of natural convection heat transfer inside the tank to the total rate of heat loss varies depending on the temperature profile inside the tank. At the beginning of standby mode, stored water inside a storage tank may be stratified, or 'well-mixed', depending on charging and discharging operations during the day.

Oliveski et al. (2003) conducted a study on transient cooling of well-mixed water in a cylindrical tank. Based on the findings, the authors developed correlations that can be used for estimating convection heat transfer coefficient inside the tank. However, the tank must be initially well-mixed to use the correlation. Furthermore, the correlation also poses limitations in its use since it was tied to a limited range of R-values for both the tank wall and insulation.

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To overcome the limitations from the study of Oliveski et al. (2003), Rodriguez et al. (2009) numerically investigated the cooling of a non-stratified cylindrical storage tank. The results were then presented in the form of a dimensionless correlation to predict convection heat transfer coefficient inside the tank. Although the correlation they developed was valid for a wider range of R-values of tank wall and insulation, it was still tied to the time elapsed from the initial non-stratified state of the tank at the beginning of standby mode.

Avoiding the issue of time-elapsed, Lin and Akins (1986) studied pseudo steady state natural convection inside a vertical cylinder by dynamically adjusting the wall temperature to maintain a constant temperature difference between the wall and the fluid following an instantaneous change of temperature of the cylinder wall. This boundary condition resembles the heating of a stratified storage tank They proposed a Nusselt number correlation, as a function of Rayleigh number, for different aspect ratios. However, the relationship is only valid for Rayleigh numbers (based on volumetric heat generation) of less than 10^7, thus, limiting its applicability to solar domestic storage cylinders, where Rayleigh numbers are often three orders of magnitude higher than 10^7.

Thus, there is a lack of generalized relationship that can predict convective heat transfer coefficient inside initially stratified tanks, based on the temperature difference between the stored water and the wall, independent of cooling time and R-values of tank insulation. Therefore, it is interesting to see if a combination of the well-established vertical flat plate correlation of Churchill and Chu (1975), as well as the upward facing horizontal plate and downward facing horizontal plate mentioned in Çengel (2007), would be able to predict the natural convection heat transfer coefficient inside cylindrical tanks for a given temperature difference between the wall and the water.

As such, this work aims to investigate the suitability of using existing flat plate correlations to estimate the natural convective heat transfer coefficient inside vertical cylindrical storage tanks, having different volumes and aspect ratios that fall within the range of solar domestic hot water tanks.

2. Methodology

To obtain steady state numerical results that can be compared with flat plate correlations, a computational fluid dynamics (CFD) approach was used. In doing this, a three-dimensional cylindrical tank with internal uniform heat generation was modelled using the CFD code, ANSYS Fluent 19.2. Two tanks, having aspect ratios between 1 and 3, and volumes of 169 L and 269 L, were considered since these fall within the typical range of solar domestic hot water cylinders. The rate of heat generation was adjusted to maintain an average water temperature of 60°C, while the insulated end walls and isothermal sidewall were held at a temperature (T_w) of 59°C, as shown in Figure 2.

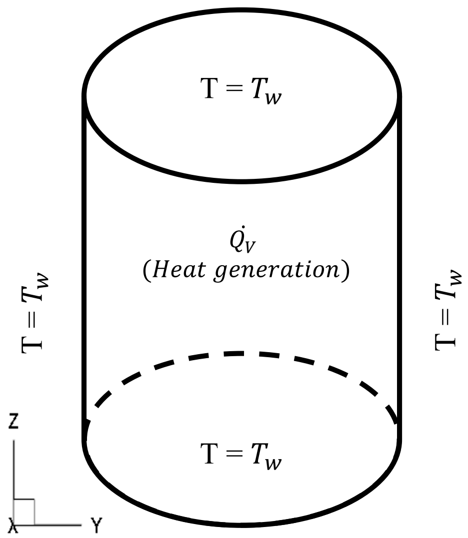


Fig. 2: Schematic of the computational boundary conditions

In analysing the natural convection, the Rayleigh number (based on volumetric heat generation) can be defined as shown in equation (1).

$$Ra_Q = \frac{g\beta\rho c_p \dot{Q}_v H^5}{\upsilon k^2} \quad \text{(eq. 1)}$$

Where g is the acceleration due to gravity (m/s^2), β is the volume expansion coefficient (1/°C), ρ is the density of fluid (kg/m^3), c_p is the specific heat capacity of fluid (J/kgK), Q_v is the volumetric heat generation rate (W/m^3), H is the height of the tank (m), υ is the kinematic viscosity (m^2/s), k is the thermal conductivity of fluid (W/mK).

On the other hand, Rayleigh number (based on temperature difference) is defined by equation (2).

$$Ra_T = \frac{g\beta(T_f - T_w)H^3}{\upsilon\alpha} \quad \text{(eq. 2)}$$

Given the cases analysed, the Rayleigh number (based on the temperature difference) lies in the range $10^{10} < Ra_T < 10^{11}$. Referring to Oliveski et al. (2003) and Rodriguez et al. (2009), the convective heat transfer coefficient on the sidewall for such cases is in the order of $10^2\ W/m^2K$. Taking this into consideration, the rate of volumetric heat generation required to keep the average temperature of water at 60°C are also in the order of $10^3\ W/m^3$. Thus, the expected Rayleigh number (based on volumetric heat generation) will be in the order of $10^{13} < Ra_Q < 10^{14}$. Kulacki and Richards (1985) reported that the laminar regime can be obtained for $Ra_Q < 10^7$ in their numerical study on cooling of rectangular fluid layers with internal heat generation and insulated bottom wall. Solutions for cylindrical hot water tanks may also apply to rectangular hot water tanks provided that the condition in equation (3) is met.

$$D_c \geq \frac{35H_c}{(\frac{Ra_T}{Pr})^{\frac{1}{4}}} \quad \text{(eq. 3)}$$

Where D_c is the diameter of the cylinder (m), H_c is the height of the cylinder (m), Pr is the Prandtl number of the fluid.

It was found that all the cases used in this study satisfy the criteria given in equation (3) and thus, turbulence modelling is needed since all the cases are expected to have $Ra_Q > 10^7$. For the treatment of turbulence in volumetrically heated enclosures, Dinh and Nourgaliev (1997) showed that reasonable results consistent with experimental data can be obtained using low-Reynolds (Re) number $k - \varepsilon$ turbulent models. Having said that, the low-Re turbulence model of Lam Bremhorst was used in this study.

A grid convergence index (GCI) with a safety factor of 3 was applied to perform a grid independence analysis, as reported by Roache (1998). The resulting GCIs between the coarse mesh (15 mm) and the fine mesh (7.5 mm), calculated based on the average temperature of the tank, showed a maximum error of 0.01%, indicating that a mesh with 15 mm cell size was adequate.

Given the temperature differences, the Boussinesq density model was used, this considers the density of water as a linear function of temperature only during the computation of body force in the momentum equation. The maximum error in density resulting from the Boussinesq approximation was found to be less than 0.002% in comparison to the non-Boussinesq data when using the thermal expansion coefficient of water evaluated at a reference temperature of 59.5°C.

To validate the computational methodology of modelling a cylindrical tank with volumetric heat generation, a three-dimensional rectangular tank model was modelled and the resulting Nusselt number on each wall was compared with the experimental data of rectangular layer of fluid subjected to isothermal cold walls, as reported by Steinberner and Reineke (1978). Six validation points were chosen from the study of Steinberner and Reineke (1978) and the CFD tank models with different volumes with the same aspect ratio of 1 were developed to match the Rayleigh numbers of chosen validation points, as indicated in Table 1.

Table 1. Chosen validation points along with dimensions of CFD models considered

H (m)	AR	Volume (L)	$\dot{Q}_V (W/m^3)$	Ra_Q
0.5	1	125	2000	6.28×10^{12}
0.52	1	140	2000	7.64×10^{12}
0.55	1	157	2000	1.01×10^{13}
0.6	1	216	2000	1.56×10^{13}
0.63	1	250	2000	1.99×10^{13}
0.8	1	512	2000	6.58×10^{13}

From figure 3, it can be seen that the simulated CFD results agree reasonably well with the published data of Steinberner and Reineke (1978) for each tank wall, except the bottom wall which indicated differences of up to 55%. However, it is important to note that rate of heat loss through the bottom wall only contributes less than 6% of the total heat loss and thus, it is not likely to cause significant errors when predicting the rate of heat loss from the tank.

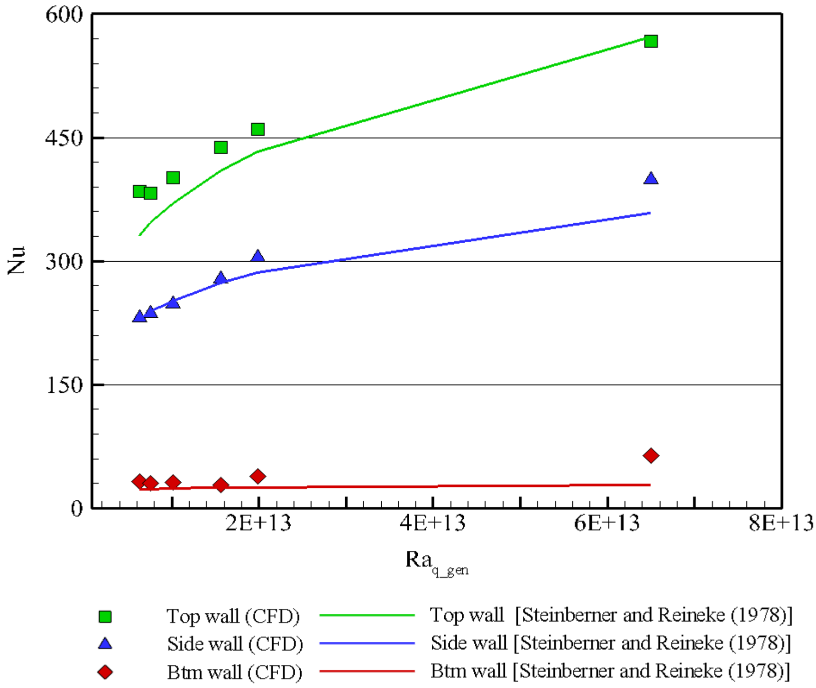


Fig. 3: Validation of CFD method with published Nusselt numbers on top, side and bottom walls from Steinberner and Reineke [6]

3. Results and Discussion

In considering the results in Figure 4 (a), it was apparent that increasing the aspect ratio of the tank led to a decrease in convective heat transfer coefficient on the side wall. This is associated with the degree of thermal stratification in the tank, which becomes more pronounced with increasing aspect ratio. The degree to which thermal stratification changes with aspect ratio can be represented by the maximum temperature difference of water inside the tank, as shown in Figure 5. It appears that this may suppress the buoyancy driven boundary layer flow, which leads to a lower convection heat transfer on the side wall. This relationship between aspect ratio and heat transfer agrees with the findings of Kulacki and Richards (1985) and Holzbecher and Steiff (1995) who investigated a similar case, but for laminar flows.

Unlike the sidewall, numerical simulations showed that the convective heat transfer coefficient on the top wall increased with increasing aspect ratio, as shown in Figure 4 (b). As the aspect ratio increases, water that is cooled by the top wall interacts with the boundary layer on the sidewall. This results in stronger convective cells close to the top wall, which increase the natural convection heat transfer near the top wall. Conversely, on the bottom wall, no notable variations in convective heat transfer coefficient with aspect ratio was observed, as can be seen from Figure 4 (c). This is owing to weak natural convection on the bottom wall due to the small temperature difference between water in that region and the tank wall.

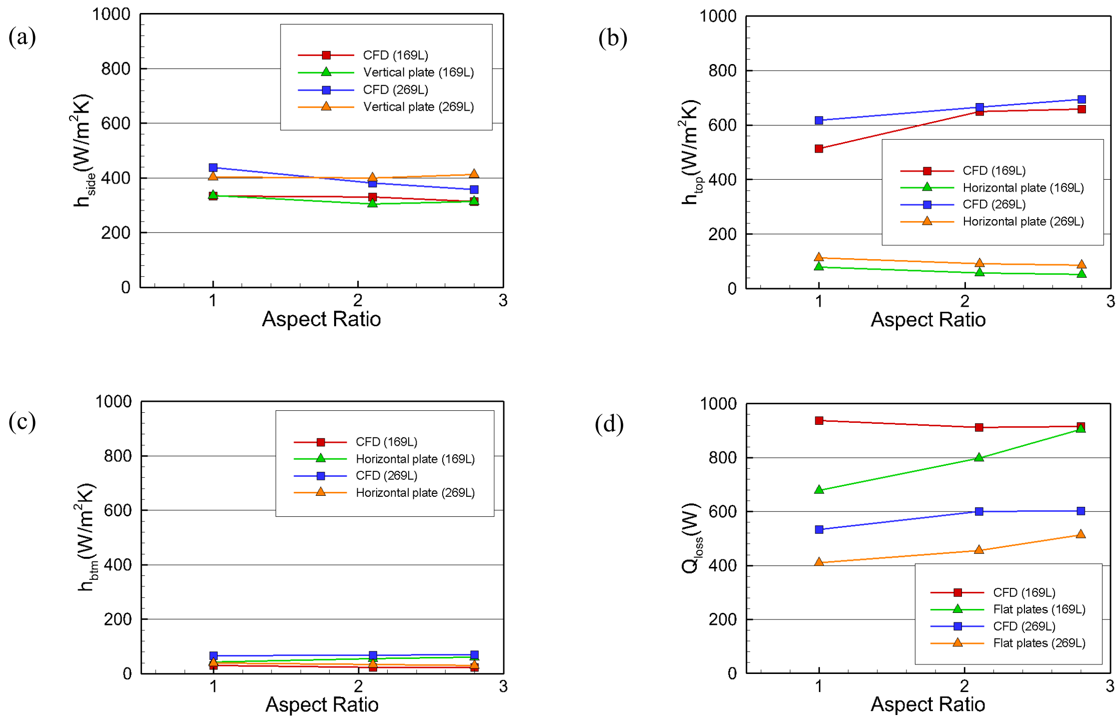


Fig. 4: Convection heat transfer coefficients on the (a) side, (b) top and (c) bottom tank walls, and (d) total rate of heat loss from storage tank models

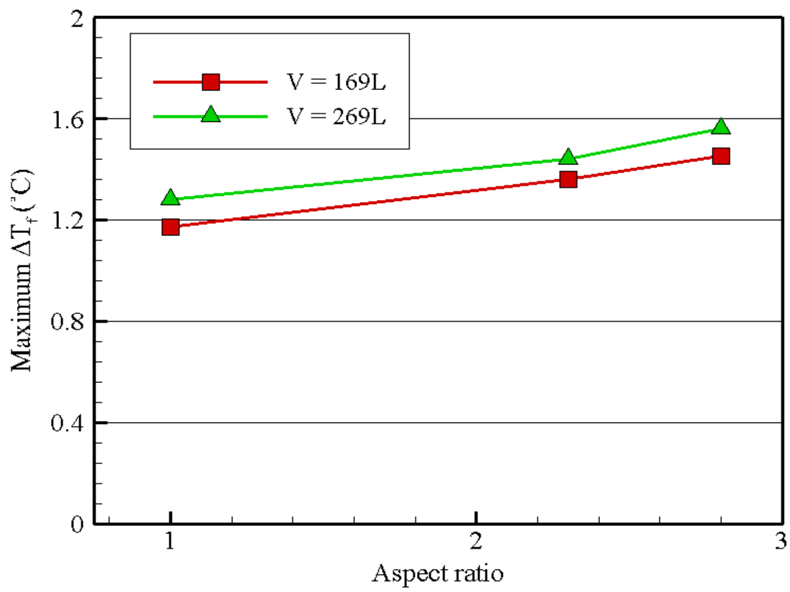


Fig. 5: Maximum temperature difference of water inside the tank reflecting the degree of thermal stratification for tank models with different aspect ratios

From observation of the differences between the numerical results and those predicted from flat plate correlations in Figure 4 (a), it is worth noting that estimations of convective heat transfer coefficients from a vertical plate is the closest for the tank with an aspect ratio of 2.3 while noticeable deviations were observed for aspect ratios of 1 and 3 for tank models with higher volume of 269 L. High convection heat transfer in low aspect ratio tanks can be attributed to mixing inside the tank generated by the sinking motion of hot water upon losing heat to the top cold wall, as illustrated in Figure 8 and 9. This phenomenon tends to be confined to the top region of the tank as the degree of thermal stratification increases with increasing aspect ratio, resulting in a low convective heat transfer coefficient. Though this effect is less pronounced in the tank model with a lower volume, of 169 L (see Figure 6 and 7).

Interestingly, the predicted heat transfer coefficients on the top wall, using flat plate correlations, are significantly lower than the numerical results, as shown in Figure 4 (b). This is associated with the formation of convective cells near the top section of the tank (see Figure 6-9) since the top wall is bounded by side walls to form an enclosure-like configuration, unlike the case of an isolated horizontal flat plate. Given the weak natural convection on the bottom wall, very little discrepancies between simulated and predicted results was observed, as can be seen from Figure 4 (c).

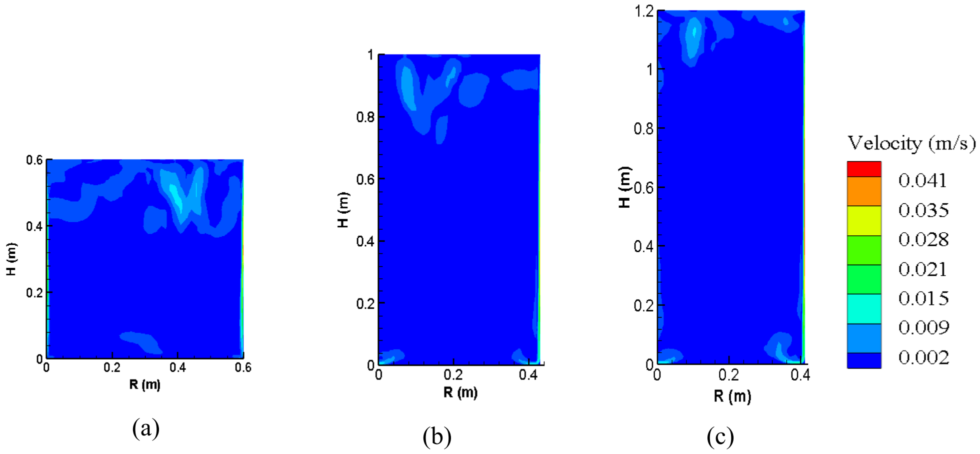


Fig. 6: Velocity contours inside 169L tank models (a) AR = 1, (b) AR = 2.3 and (c) AR = 2.8

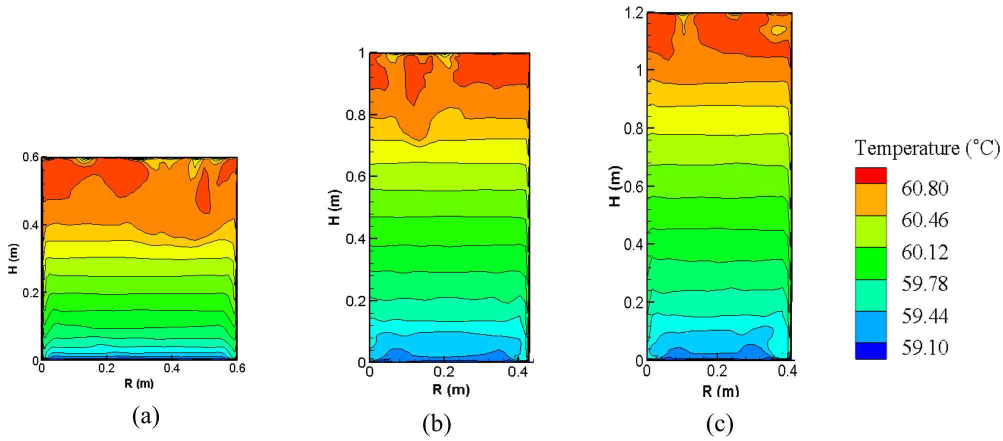


Fig. 7: Temperature contours inside 169L tank models (a) AR = 1, (b) AR = 2.3 and (c) AR = 2.8

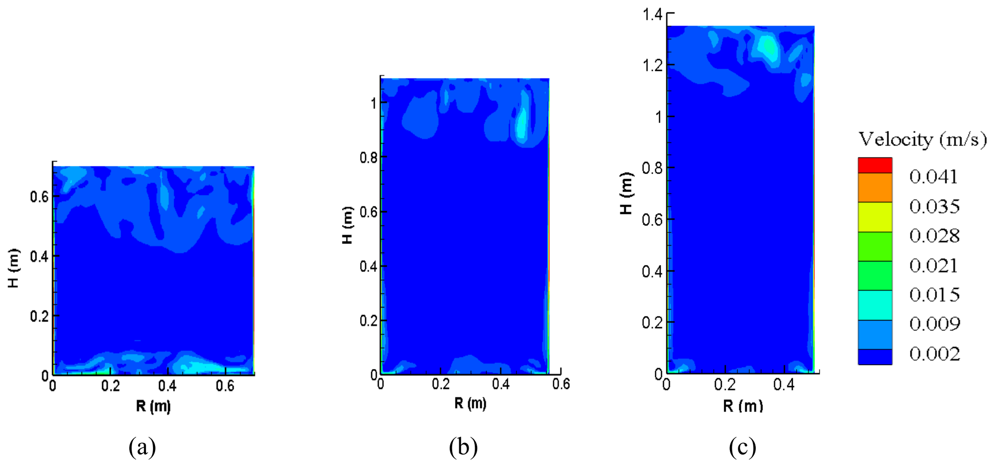
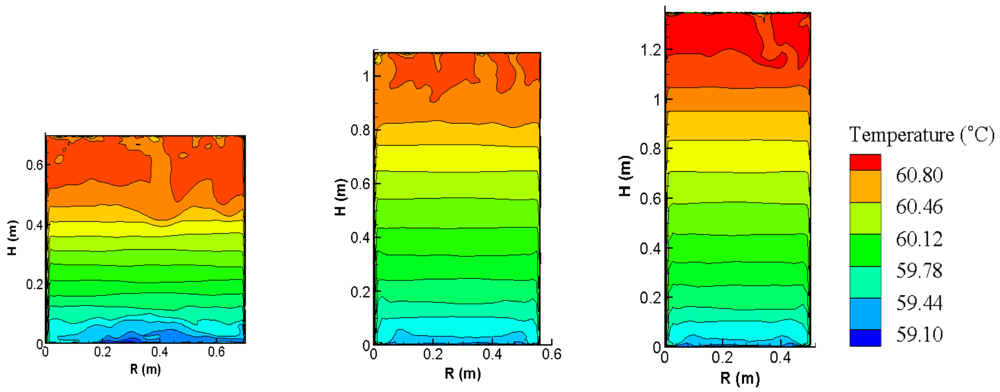


Fig. 8: Velocity contours inside 269L tank models (a) AR = 1, (b) AR = 2.3 and (c) AR = 2.8



(a) (b) (c)

Fig. 9: Velocity contours inside 269L tank models (a) AR = 1, (b) AR = 2.3 and (c) AR = 2.8

From the observation of the difference in overall rate of heat loss between numerical results and predictions from flat plate correlations in Figure 4 (d), it is apparent that the existing flat plate correlations cannot accurately estimate of rate of heat loss with the maximum deviation of 23% was observed for the case of 169L tank with aspect ratio of 1, owing to the discrepancies in convective heat transfer coefficients on the side and top walls.

4. Conclusion

Natural convection heat transfer inside cylindrical enclosures, which is applicable to stratified solar water storage tanks, has received little attention and because of this, there is a lack of relationships that can be used to determine the rate of heat loss from such systems. To address this, this study examined the suitability of using a combination of existing flat plate correlations to predict convective heat transfer coefficient inside stratified vertical cylindrical tanks with different volumes and aspect ratios that fall in the range of solar domestic hot water tanks. The results showed that there are discrepancies in convective heat transfer coefficients on the top and side walls between flat plate correlations and those of simulated data. This highlights the fact that there is still a need to develop a correlation that can predict convective heat transfer coefficient inside these stratified tanks independent of time elapsed from the initial non-stratified state of the tank and R-values of tank insulation.

5. Symbols and Abbreviations

Symbols

AR	Aspect ratio
c_p	Specific heat capacity (J/kgK)
D	Diameter (m)
g	Gravitational acceleration (m/s^2)
h	Local convection heat transfer coefficient (W/m^2K)
H	Height (m)
k	Thermal conductivity (W/mK)
Nu	Local Nusselt number
Pr	Prandtl number
$\dot{Q}_V$	Rate of volumetric heat generation (W/m^3)
Q	Rate of heat loss (W)
R	Thermal resistance (K/W)
Ra_Q	Rayleigh number (volumetric heat generation)
Ra_T	Rayleigh number (temperature difference)
T	Temperature (°C)
ΔT_f	Temperature difference of fluid within the tank (°C)
β	Volume expansion coefficient ($1/°C$)
υ	Kinematic viscosity (m/s^2)
ρ	Density (kg/m^3)

Subscripts

i, conv	Internal natural convection
w,cond	Conduction (wall)
ins, cond	Conduction (insulation)
e, conv	External natural convection
f	Fluid (water)
env	Ambient
w	Wall
c	Cylinder
side	Side wall
top	Top wall
btm	Bottom wall
loss	Heat loss

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