Simplified model for transient cooling process of a storage tank with an internal gas flue

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Abstract

The transient cooling of a fluid initially at rest, inside a storage tank with an internal gas flue submitted to heat losses to the ambient is studied. In order to identify the relevant non-dimensional groups that define the transient natural convection phenomena that occur, a non-dimensional analysis is carried out. The long-term behavior of the fluid is modeled by formulating a prediction model based on global balances. The global model together with the proposed correlations have been submitted to a validation process by comparing the results with those obtained from experimental set-up and detailed numerical simulations.

1. Introduction

In the last decades, detailed numerical simulations of heat transfer and fluid flow using Computational Fluid Dynamics (CFD) codes have became a powerful tool for studying the complex phenomena taking place in storage tank. Several works describing the unsteady cooling or heating processes of a fluid inside an enclosure can be found in the literature. Among these studies, the work conducted by Lin and Armfield [1]. They studied numerically the cooling process of an initially homogeneous fluid by natural convection in a vertical cylindrical tank. In their work, the top and the bottom walls were maintained adiabatic while the lateral wall was suddenly cooled. In a later paper, the same authors [2] analysed the long-term behavior of cooling a fluid in a cylindrical tank considering two different situations: one with an imposed sidewall temperature and the other with fixed side and bottom wall temperatures while the other walls were maintained adiabatic. Using the numerical results, scaling relations for the mean fluid temperature and Nusselt number have been obtained for each situation. Although their study was in non-dimensional form, because of the imposed boundary conditions, the correlations they found can not be extended to the real case of a cylindrical tank submitted to unsteady natural convection due to heat losses to the environment.

More recently, Rodríguez et al. [3] carried out a scaling analysis and numerical simulation of the transient process of cooling-down of an initially homogeneous fluid in a vertical cylinder submitted to natural convection. The correlations obtained can be extrapolated to other situations as they are expressed in term of non-dimensional parameters governing the phenomena that occur inside the tank.

Almost all the mentioned studies have been carried out in standard cylindrical tanks. However, the results obtained in the aforementioned works can not be extrapolated to the configuration of a storage tank with an internal gas flue. This configuration is not only of fundamental interest to fluid mechanics and heat transfer, but also of importance in many practical applications. In fact, the attempt of the present study is to investigate, by means of numerical CFD simulations, the long-term cooling process of this tank configuration due to unsteady natural convection. A global model has been proposed. To feed this simplified model with the main parameters, some proposed Nusselt correlations have been used [4].

2. Problem definition

A schematic of the cylindrical storage tank under study is shown in Fig. 1. The tank considered, with an aspect ratio (H/D), is made of 3 mm of steel wall and the thermal insulation of polyurethane thermal of 2 cm thickness. In the present analysis, the tank has an external aspect ratio Hc/D, with a gas flue aspect ratio Hc/Dc and with a height ratio Hc/H. The tank is submitted to a cooling process due to temperature gradient between the fluid and the ambient. The tank is initially at uniform temperature of T_0, while the ambient temperature has been remained fixed to T_{env} = 293 K.

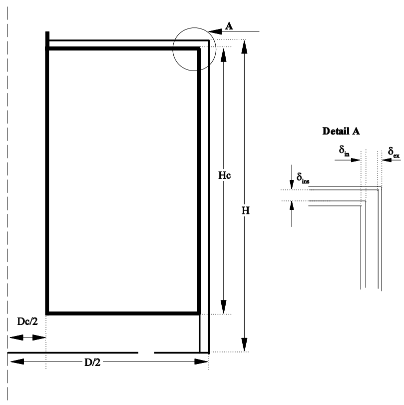


Figure 1. Schematic of the geometry of the storage tank under study.

3. Global model analysis of transient cooling process

To characterise the transient behaviour of a storage tank with an internal gas flue during a long-term cooling process, we have considered one-temperature level for each domain i.e. in the storage tank (T) and in the gas flue (T_F).

3.1. Inside the storage tank

Taking into account the assumptions outlined before, an energy balance of the tank can be formulated as:

$$\rho C_p \frac{\partial \bar{T}}{\partial t}\Omega = -\left(\frac{\bar{U} h_{env} S_{env}}{\bar{U} + h_{env}}\right)(\bar{T} - T_{env}) - \left(\frac{\bar{U}_{WF} h_{WF} S_{WF}}{\bar{U}_{WF} + h_{WF}}\right)(\bar{T} - \bar{T}_F) \quad (1)$$

3.2. Inside the gas flue

The energy balance of the gas flue can be formulated as:

$$\rho C_p \frac{\partial \bar{T}_F}{\partial t}\Omega = -\left(\frac{\bar{U}_F h_F S_F}{\bar{U}_F + h_F}\right)(\bar{T}_F - \bar{T}) + \dot{m} C_p (T_{env} - \bar{T}_F) \qquad (2)$$

From the one-level temperature model represented by Eqns. (1) and (2) it is possible to evaluate the transient evolution of the mean fluids temperatures at the storage tank and in the gas flue, if the values of the transient heat loss coefficients through internal fluid wall and through the internal and external flue walls (h_{env}; h_{WF}; h_F) are known.

4. Mathematical formulation

The domain involves the two fluids (water inside the store and air inside the flue), an internal steel wall, and a polyurethane insulating material covered by an uniform thin layer of steel at external walls. The fluid flow and heat transfer phenomena for the problem under study is governed by the Navier-Stokes and the energy conservation equations. In the model, laminar flow and constant physical properties except density variations in buoyancy terms of the momentum equations (Boussinesq approximation) have been assumed.

4.1. Boundary conditions

Considering the symmetry of the problem defined, the computational domain has been assumed to be axisymmetric. No-slipping conditions have been used as boundary conditions for the momentum equations at all solid walls, while at the axis, symmetry boundary condition has been imposed. In the energy equation, every external tank wall (side and top) has been submitted to external convection actions.

4.2. Numerical approach

The numerical solution has been obtained over the whole domain by discretising the governing equations by means of finite volume techniques as described by Patankar [5]. Fully implicit first order temporal differentiation using cylindrical grids on a staggered arrangement has been implemented. Diffusive terms have been evaluated using second order central differences scheme, while convective terms have been approximated by means of high order SMART scheme using a deferred correction approach [6]. The pressure-velocity coupling has been solved by using a SIMPLE-C algorithm [7]. The resulting equation for pressure has been solved using the direct Band LU solver [8] while the algebraic system of linear equations for the velocities and temperature have been calculated using TDMA line by line solver [5].

5. Results

5.1. The heat transfer coefficient

According to a non-dimensional analysis done, the heat transfer coefficients at the different walls can be expressed in terms of the non-dimensional groups identified [4]. The numerical results have been adjusted using the Levenberg-Marquardt algorithm [9]. This method provides the solution to the non-linear function, by minimising the sum of squares of the deviations.

- The external heat transfer coefficient for the storage tank (ambient side)

$$h_{env} = 4.18t^{-0.188}Ra^{0.076}\left({}^{H_C}/_{D}\right)^{0.378}\left({}^{H_C}/_{D_C}\right)^{-0.221}\left({}^{H_C}/_{H}\right)^{0.614}\hat{U}^{0.374}\frac{k\alpha^{-0.188}}{H_C^{0.624}} \quad (3)$$

- The internal heat transfer coefficient for the storage tank (chimney side)

$$h_{WF} = 0.96t^{-0.204}Ra^{0.064}\left({}^{H_C}/_{D}\right)^{0.333}\left({}^{H_C}/_{D_C}\right)^{0.371}\left({}^{H_C}/_{H}\right)^{-1.184}\hat{U}^{-0.324}\frac{k\alpha^{-0.204}}{H_C^{0.592}} \quad (4)$$

- The heat transfer coefficient for the gas flue

$$h_F = 3.66t^{0.066}\left({}^{H_C}/_{D}\right)^{-0.281}\left({}^{H_C}/_{D_C}\right)^{0.898}\left({}^{H_C}/_{H}\right)^{-3.96}\hat{U}_F^{-0.028}\frac{k\alpha^{0.066}}{H_C^{1.11}} \quad (5)$$

5.2. The transient mean fluids temperatures

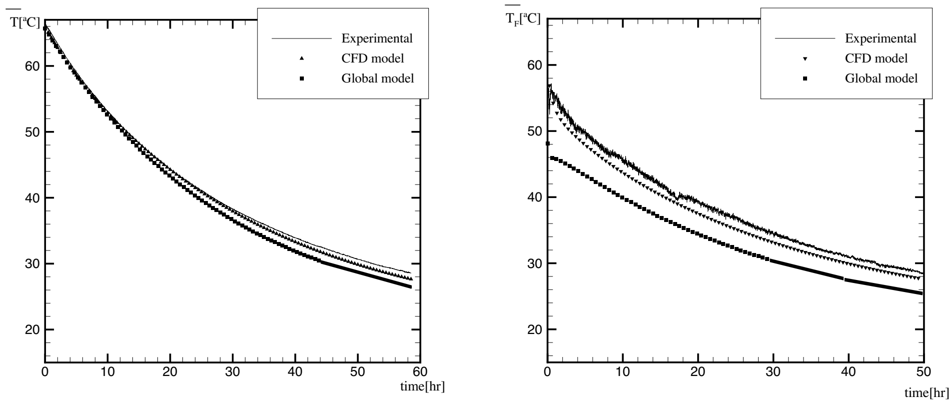
Substituting the heat transfer coefficients by their expressions (Eqns. 3, 4 and 5) in the one-level temperature balances (Eqns. 1 and 2), the transient mean fluid temperature in the water and in the gas flue domains can be defined as function of time.

Due to the coupling between the fluids temperatures in the storage tank and the gas flue, analytical solutions are not available. Thus, the only way to solve these equations is by using an iterative algorithm. In order to obtain the transient evolution of the mean fluids temperatures, the solving process performed has been done by means of the Runge Kutta 4th order algorithm [10].

5.2.1. Validation of the global model

Aiming the validation of the scaling relations obtained in the relevant parameters range studied, the same case which has been tested experimentally and numerically has been simulated using the proposed global model.

For detailed numerical simulation, the considered case account for a tank volume of 0.144m^3 with an external aspect ratio of Hc/D = 2.04 (Hc=0.932m, D=0.455m). Composite tank walls are made of 3 mm steel, 2 cm of polyurethane thermal insulation and 1mm of steel at the external part. The chimney has a diameter (Dc) about 0.1m. The overall heat transfer coefficient used has been set to h_{ext} = 10W/m^{2}K and the ambient temperature has been fixed at the average value of experimental records i.e. at 19 ºC [11].



(a) (b)

Fig. 2: Mean transient temperature of the fluid inside the tank and inside the gas flue obtained with correlation. Comparison with numerical results and experimental results: (a) Inside the storage tank; (b) Inside the gas flue.

In Fig. 2 (a), the results predicted by the numerical code and the simplified model are compared with experimental data. It shown that, even after 60 hours of the cooling process, the simplified model is able to reproduce the experimental data with a good accuracy.

Regarding the air in the chimney, the comparison between the numerical and experimental results is given in Fig. 2 (b). By analysing the figure, it can be shown that, during the 60 hours of the cooling process, a good agreement between the numerical and the experimental results have been obtained. In comparison with global model results, the same tendency as the experimental data has been obtained with some discrepancies.

In general, the temperature profiles are comparable but minor discrepancies between experiments and calculations have been observed especially in the gas flue. Those differences can be due to different factors such as: i) the inconsistency between the experimental rate of air entering the chimney and that imposed in the calculations; ii) another source of discrepancies is the approximations used in the global code: the same heat transfer coefficients have been used along the gas flue and the storage tank; iii) the global model is based on an iterative algorithm.

5.2.2 Verification of the global model

For further verification of the proposed global model together with the correlations for the Nusselt numbers within the range of the relevant parameters studied, the transient evolution of the temperatures inside the tank and inside the gas flue for a new set of cases have been carried out using the simplified model and the detailed numerical simulations. This set of cases account for tank volumes of: 0.15 and 0.1m^3, with an external aspect ratios of: Hc/D = 2.0 and 2.4 and with a gas flue aspect ratios of: Hc/Dc = 9.32 and 15.53. For these cases, initial temperatures have been of 40, 50 and 60 ºC, while the ambient temperature has been fixed to 20ºC. The external heat transfer coefficients of

h_{ext} = 2 and 10W/m^{2}K have been used. All the selected cases have been within the studied range of the relevant parameters which define the problem.

The verification has consisted in comparing the simplified data versus the numerical results. The agreement between detailed calculations and global model results has been quantified by using the mean relative and maximum errors for the mean fluid and gas flue temperatures.

Table 1: Mean fluid temperature and mean gas flue temperature maximum and average relative errors between numerical and global model results.

						Maximum relative errors		**Mean relative errors**	
Case	**Ra**	**Ǔ**	**Hc/D**	**Hc/Dc**	**Hc/H**	**T [%]**	T_F **[%]**	**T [%]**	T_F **[%]**
Case 1	$7.5 \cdot 10^{11}$	5.84	2.00	9.32	0.823	3.81	11.4	2.89	4.94
Case 2	$1.2 \cdot 10^{12}$	5.83	2.42	9.32	0.823	4.09	20.0	3.17	7.08
Case 3	$4.0 \cdot 10^{11}$	2.46	2.42	9.32	0.823	1.68	14.35	1.17	5.88
Case 4	$1.2 \cdot 10^{12}$	2.34	2.00	15.53	0.823	0.77	14.76	0.39	3.71

6. Conclusion

The global models together with the proposed correlations have been submitted to a validation process for a specific case of a storage tank. In fact, the global model has been verified at the first stage by comparing the results with those obtained from experimental set-up and from detailed numerical simulations. Satifactory results have been obtained especially for the fluid temperature.

In a second stage, a new set of cases have been tested using detailed numerical simulations and compared to the global model. Considering all the results, it can be concluded that the proposed methodology together with the correlations lead to satisfactory results within the range of the relevant parameters studied.

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