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Comparative study of the influences of different water tank shapes on thermal energy storage capacity and thermal stratification

Zheng Yang, Haisheng Chen*, Liang Wang, Yong Sheng, Yifei Wang

Institute of Engineering Thermophysics, Chinese Academy of Sciences, Beijing 100190, China

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The influences of different water tank shapes on thermal energy storage capacity and thermal stratification in the static mode of operation is investigated in this study under laminar natural convection. A new experimental apparatus is built, and a numerical model is developed to simulate the flow and heat transfer in the water tank. Computational results agree with the experimental data. Among the 10 different water tank shapes studied, the sphere and barrel water tanks are ideal for thermal energy storage capacity, whereas the cylinder water tank is the least favorable. The thermal energy storage capacity is closely related to the surface area of the water tank. According to the characteristics of the velocity and temperature fields, these shapes can be divided into three categories: shapes with sharp corners, those with hemispheres, and those with horizontal plane surface. Shapes with sharp corners have the highest degree of thermal stratification, whereas the shapes with horizontal plane surface possess the lowest. That of the shapes with hemispheres lies in between these two degrees. The thermal stratification of different shapes is determined by the flow at the bottom of the water tank and the heat transfer from the fluid to the environment.

1. Background

Thermal energy storage in water tanks is important in many engineering fields, such as in the storage systems of supercritical compressed air, solar heating systems, and nuclear reactors. The operation process of the water tank can be divided into the dynamic mode of operation and the static mode of operation. The dynamic mode of operation refers to its operation when thermal energy from the tank is being used, i.e., when the discharging/charging process is taking place. The static mode of operation refers to its thermal behavior when the water in the tank is not being used, i.e., when there is not water flowing in/from the tank [1]. The static mode of operation is also called the cooling process in some literature, because the thermal behavior in the static mode of operation is caused by a natural cooling process (i.e. a passive and not an active one) owing to heat losses to the environment. During the static mode of operation, velocity and temperature boundary layers form along the lateral wall of the water tank as a result of the heat transfer from the fluid in the tank to the environment, thus inducing thermal stratification in the tank. This stratification significantly affects thermal energy storage capacity and even system efficiency [2].

Many researchers are interested in this topic. Rodríguez et al. [3] studied the transient cooling of a fluid that is initially at rest inside a vertical cylinder and is subject to heat loss through the walls. A correlation for the *Nu* number was obtained, and a global thermodynamic model was successfully established. He et al. [4] investigated natural convection heat transfer and flow in a vertical cylinder with two ends at different temperatures. The variation patterns of *Nu* versus *Ra* are consistent with the results obtained by Catton and Edward [5,6], but the application range can be extended to the aspect ratio of 10. Holzbecher and Steiff [7] experimentally and numerically investigated natural convection flow in an internally heated vertical cylinder and noted that laminar and turbulent natural convections differ significantly with the influence of the boundary condition. Papanicolaou and Belessiotis [8] studied the transient state of natural convection in a vertical cylindrical enclosure for water at high *Ra* numbers. Their results showed that the low *Re* $k-\varepsilon$ models often predicted relaminarization but that the high *Re* $k-\varepsilon$ models detected very gradual decaying turbulence.

* Corresponding author.
E-mail address: chen_hs@mail.etp.ac.cn (H. Chen).

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Nomenclature

c_p	specific heat in the fluid region, J/kgK
$c_{p,ins}$	specific heat in the thermal insulation layer, J/kgK
E	internal energy,kJ
E_{env}	internal energy at the environment temperature, kJ
e	internal energy per unit mass, kJ/kg
Ex	internal exergy, kJ
ex	internal exergy per unit mass, kJ/kg
g	acceleration of gravity, m/s^2
Gr	Grashof number, $Gr = g\beta\Delta TL_{ref}^3/\upsilon^2$
h_{ext}	heat transfer coefficient of the external surface, W/mK
k	heat conductivity in the fluid region, W/mK
k_{ins}	heat conductivity in the thermal insulation layer, W/(mK)
L_{ref}	reference length, m
Nu	average Nusselt number,$Nu = hl/k$
p_{env}	environment pressure, MPa
p	fluid pressure, Pa
q_w	heat flux on the fluid wall of the water tank, W/m^{2}K
q_{ins}	heat flux on the internal wall of the thermal insulation layer, W/m^{2}K
Pr	Prandtl number, $Pr=\upsilon/\alpha$
Ra	Rayleigh number, $Ra = g\beta\Delta T_{ref}L_{ref}^3/\upsilon\alpha$
r	radial position, m
T	temperature in the fluid region, K
T_0	fluid initial temperature, K
T_{env}	environment temperature, K
T_∞	main fluid temperature, K
T_w	internal wall temperature, K
T_{ref}	reference temperature,$T_{ref}=(T_\infty+T_w)/2$, K
T_{ins}	temperature in the thermal insulation layer, K
ΔT_{ref}	reference temperature difference,$\Delta T_{ref}=T_\infty-T_w$, K
t	time, s
Δt	time step, s
v_{ref}	reference velocity,$v_{ref}=\alpha/L_{ref}$, m/s
u	axial velocity in the fluid region, m/s
V	volume of the water region, m^3
v	radial velocity in the fluid region, m/s
z	axial position, m

Greek letters

α	thermal diffusivity in the fluid region, m^2/s
β	thermal expansion coefficient in the fluid region, 1/K
μ	dynamic viscosity in the fluid region, kg/ms
ρ	density in the fluid region, kg/m^3
ρ_{ins}	density in the thermal insulation layer, kg/m^3
τ	non-dimensional time, $\tau = tv_{ref}/L_{ref}$
η_1	thermal energy storage efficiency, %
η_2	thermal exergy storage efficiency, %
υ	kinematic viscosity,m^2/s

Oliveski [9] numerically investigated the transient cooling process in a vertical cylindrical tank to correlate the internal heat transfer coefficient with the main parameters that govern the cooling process. The correlation that combines the aspect ratio and the *Nu*, *Pr*, and *Ra* numbers agreed with the numerical results. Oliveski [10] analyzed the flow and temperature fields inside a storage tank that is subject to natural convection. Two correlations for the *Nu* number were obtained in the numerical results. Fernandez-Seara et al. [1] conducted an experiment to study the static heating and cooling periods of a water storage tank, especially its thermal performance and the degree of thermal stratification.

Previous investigations mostly focused on natural convection and the thermal stratification of square cavities [11–14] and vertical cylinders [3–10]. They ignore other water tank shapes. Thus, the current study investigates and compares the influences of different water tank shapes on thermal energy storage capacity and thermal stratification in the cooling process numerically and experimentally. An experimental apparatus with which to obtain the experimental results is described in detail in Section 2. The mathematical model and the scheme for the numerical solution are established to simulate the cooling process. They are validated by the experimental results presented in Section 3. In Section 4, the thermal energy storage capacity and the thermal stratification of these water tank shapes are compared and discussed. Finally, conclusions are presented in Section 5.

2. Experimental apparatus

This experiment aims to obtain reliable data with which to validate the mathematical model. Fig. 1 shows the structure diagram of the experimental apparatus. The tests were conducted in a vertical cylindrical tank that is 35 cm tall, has an inner radius of 35 cm, and has a wall thickness of 1 mm. The tank is fabricated with stainless steel 304. A thermal insulation layer covers the tank walls, which is 3 cm thick and is composed of glass wool. To avoid crushing by the tank and the water, four small cylinders are placed under the tank at the bottom of the thermal insulation layer. The cylinders are 40 mm in radius and consist of polytetrafluoroethylene. Furthermore, a wooden base supports the experimental apparatus to prevent direct contact with the ground.

The temperature measurement system is composed of temperature points, a data acquisition card, and a computer. The measuring points are generated by a T-type thermocouple with a measuring head of 1 mm. The collected electrical signals are sent to two data acquisition cards (NI 9213, a 16-channel thermocouple input module) and then displayed and recorded by a measurement program based on Labview 11.0. Data are recorded every second. The uncertainty analysis considers the temperature measurement, time effect, and data connection board. It produces a mean uncertainty of ±0.1 K and is evaluated according to the method prescribed by Moffat [15].

Fig. 2 depicts the distribution of the measuring points in the experimental apparatus. To measure the temperature field within the water tank, a bracket is placed inside the tank. This bracket is composed of stainless steel and consists of a backbone that is 350 mm tall and six branches that are 175 mm long. A total of 32 T-type thermocouples are installed on the bracket. The backbone is equipped with seven measuring points for axis temperature distribution, and each radial branch is equipped with four measuring points for radial temperature distribution. Furthermore, six measuring points are installed on the external wall of the tank to measure the external wall temperatures. One measuring point is installed to determine the environmental temperature.

All tests begin with a 353.15 K uniform temperature field in the water tank. They are conducted indoors, and the environment temperature is maintained at 288.15 K–289.15 K. The tests last for 20 h. Four repeated experiments are performed, and one set of ideal data is finally selected as the test result.