# CFD Solution and Experimental Testing of Buoyancy-Driven Convection Caused by Condensers Immersed in a Water Tank of HPWH

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## Abstract

Heat pump water heaters can cut electricity consumption by half, as comparing to the conventional electric resistant water heaters. A conventional heat pump water heater (HPWH) usually has a water circulating pump to sample water temperature every 15 minutes in the tank, and to draw water to a condenser-water heat exchanger outside the tank, if water temperature is below the set point. The pump would be on at least once every 15 minutes, 24 hours a day. The novel design presented in this study was to insert the condenser coil through the opening on the top of the water tank. This design eliminated the need of the water circulating pumps, and may improve the efficiency of the HPWH by eliminating pumping power.

Two types of condenser coil designs were considered; one was a bayonet tube (tube-in-tube) and one was the "U" tube. Previous test data indicated that "U" tube design performed better than the bayonet tube condenser coil, and thus only "U" tube condenser coil was considered in the study.

With straight "U" tubes inserted into the tank, it was found that the convective heat transfer was not strong enough to break water temperature stratification in the tank, which resulted in a temperature differential of 16ºC (30ºF) from top to bottom. However, when the coil was built in an"L" shape, the water stratification disappeared. A computational fluid dynamics code, CFD, was used to study the straight and "L" shaped condenser coils. Results from CFD simulation were compared with the experimental data and found to be close to each other.

### Nomenclature

| | |
|---|---|
| CFD | : Computational fluid dynamics; |
| COP | : Coefficient of performance; |
| G | : Gravity; |
| HPWH | : Heat pump water heater; |
| T | : Temperature; |
| $\Delta T$ | : Temperature difference; |
| t | : Time; |
| P | : Pressure; |
| $\vec{\mathbf{u}}$ | : Velocity vector; |

**Greek symbols**

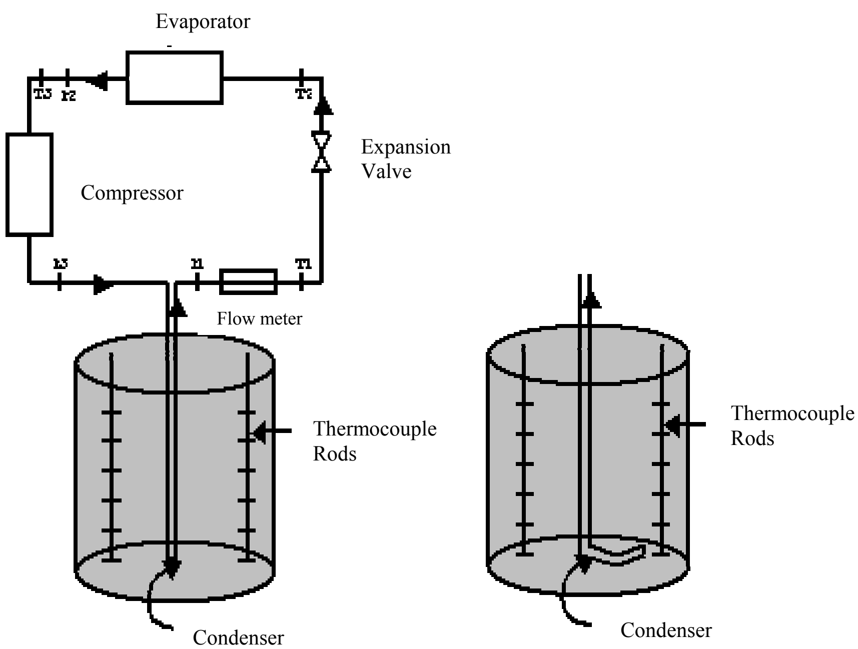
| | |
|---|---|
| $\nu$ | : Kinematic viscosity; |
| $\alpha$ | : Thermal diffusivity; |
| $\rho$ | : Density; |
| $\beta$ | : Thermal expansion coefficient; |

## Introduction

There are two popular types of residential water heaters used in the American market: direct gas fired and electric resistance. Nationwide, sales of gas fired water heaters hold a slight majority with ~54 percent of the market, with the remaining 46 percent being almost entirely electric resistance units. In the year 2000, just over 9.2 million total residential water heaters were sold that year, yet only about 2,000 of which were HPWHs [1,2,3]. Heat pump water heater, however, only finds good market in tropical climate region where mild temperature provides high HPWH efficiency.

Many HPWHs adopt pumps to circulate water from the storage tank through the heat pump condenser for water heating. Because of the frequent on/off cycling of the pumps and high temperature operating conditions, the pump reliability may be compromised. It would be costly to replace the water pumps. The design reported in this study eliminates the pump by directly immersing the condenser coil through an opening on the top of the tank without any major change of current tank design. The novel concept could potentially be less expensive than present designs and more efficient because of the elimination of the pump and a heat exchanger.

Preliminary test data showed that a vertical condenser coil inserted into the tank resulted in water temperature stratification within the tank, as high as 20˚C (36˚F) temperature differential from top to bottom, regardless of heat flux [3]. The refrigerant condensing temperature was high because of the inefficient heat transfer between water and refrigerant caused by the temperature stratification, which in turn caused high compressor operating pressure. The temperature stratification also damped the convective water circulating with in the tank [3]. If the temperature stratification could be eliminated, the heat pump could be operated more efficiently with lower discharge pressure. In this study, a hockey stick like "L" shape condenser coil was designed to reduce, or eliminate, the temperature stratification inside the tank. A CFD code was used to simulate the HPWH performance with the new condenser coil design. The CFD calculated data indicated temperature stratification was greatly reduced to only 1˚C (2˚F).




(a) (b)

Figure 1 Schematic of a U-tube condenser immersed in a water tank of HPWH; (a) is a vertical condenser, and (b) is an L-Style condenser [3].

When heat is added to a fluid and the fluid density varies with temperature, a flow can be induced due to the force of gravity acting on the density variations. In the water tank of a HPWH, a buoyancy-driven flow can be induced if an immersed condenser is employed. The buoyancy-driven flow is strongly dependent of the configuration of condensers.

In the present work, CFD simulation and tests were conducted to study the effects of condenser geometry on water temperature stratification and heat pump performance. The schematic of heat pump water heating is shown in Figure 1. In Fig. 1(a), the top part is vapor-compression cycle, including compressor, expansion valve, evaporator, and condenser immersed in a water tank. The geometries of condenser include vertical and L-style condensers, as is also illustrated in Figure 1(a) and 1(b). Both types of condenser coils were tested to confirm the simulated data.

**Physical model and simulation**

Buoyancy-driven flow in enclosed spaces has been studied extensively in the past decades in response to energy-related applications (e.g. nuclear and solar). Ostrach (1982) offered a broad review of the work carried out prior to 1982. Schwab and De Witt (1970) were among the first to study natural convection from a single vertical rod immersed in a concentric cylinder. They predicted the effect of geometrical ratio and Prandtl number on the Nusselt Number. Charmchi and Sparrow (1982, 1983) conducted both experiment and simulation of a small inner cylinder located at various positions within a concentric cylinder. Neumann (1990) studied the steady and oscillatory convection in rigid vertical cylinders heated from below. Later work on natural convection in a vertical cylinder was primarily concerned with the heat transfer coefficient and correction (Lafortune and Meneley, 1990). Up to now, very few cases have been found to investigate thermal stratification reduction and fluid flow field in the water storage of HPWH.

In the current study, a vertical cylindrical enclosure is chosen as a fluid storage tank. Water is the working fluid, which is regarded as an incompressible fluid with constant kinematic viscosity $v$, and thermal diffusivity $\boldsymbol{\alpha}$. Natural-convection flow is modeled with Boussinesq approximation during CFD simulation. In the Boussinesq approximation, variations in fluid density are ignored, except insofar as they give rise to a gravitation force. Therefore, fluid density is treated as a constant value in all solved equations, except for the buoyancy term in the momentum equation:

$$(\rho - \rho_0)g \cong -\rho_0 \beta (T - T_0) g \tag{1}$$

where $\rho_0$ is the constant density of flow, $T_0$ is the operating temperature, and β is the thermal expansion coefficient. The above equation is obtained by using the Boussinesq approximation $\rho = \rho_0 (1 - \beta \Delta T)$ to eliminate $\rho$ from the buoyancy term. This approximation is accurate as long as changes in actual density are small.

Therefore, the continuity of fluid flow then become

$$\nabla \cdot (\rho \vec{\mathbf{u}}) = \rho \nabla \cdot \vec{\mathbf{u}} = \nabla \cdot \vec{\mathbf{u}} = \mathbf{0} \tag{2}$$

The Navier-Stokes equation of fluid flow is described as follows

$$\frac{D\vec{\mathbf{u}}}{Dt} = -\frac{\nabla P}{\rho_0} + v\nabla^2 \vec{\mathbf{u}} - \frac{\Delta \rho g}{\rho_0} \tag{3}$$

After substituting Eq.(1) into the foregoing equation, it becomes

$$\frac{D\vec{\mathbf{u}}}{Dt} = -\frac{\nabla P}{\rho_0} + v\nabla^2 \vec{\mathbf{u}} - g\beta (T - T_0) \tag{4}$$

The addition of an equation for temperature completes the Boussinesq approximation. The energy equation is given by

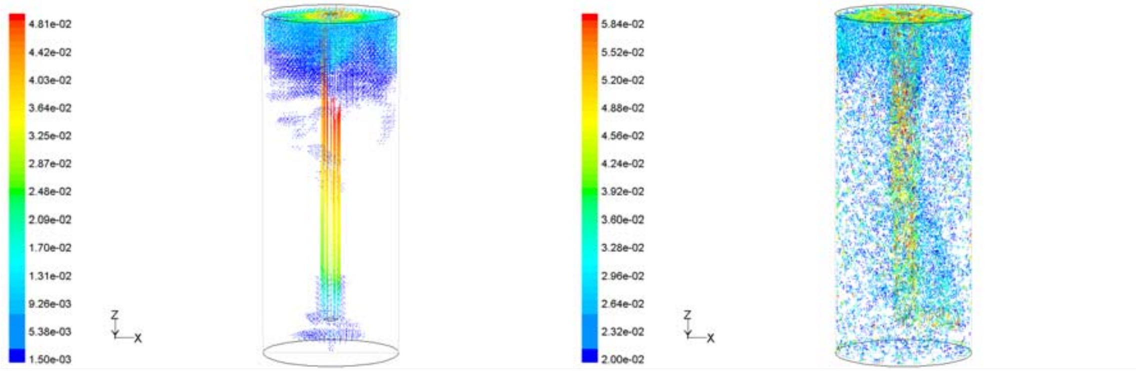
$$\frac{\partial T}{\partial t} + \vec{\mathbf{u}} \cdot \nabla T = \alpha \nabla^2 T$$

At the tank wall, either the no-slip or the shear-free velocity boundary condition is prescribed. The cylindrical sidewall is assumed adiabatic. The boundary condition at condenser wall was based on heat flux obtained experimentally [11]. Tank water and condenser coil temperatures were assumed constant and were measured experimentally.

The buoyancy-driven flow in the present case is considered as a transient process. All simulations are conducted under three-dimension modeling. In the phase of CFD simulation, two geometry types of condensers are considered: vertical and L-style heated rods. To simplify and save computational time, the condenser is assumed to be a single heated rod. The diameter of the rod is 0.03M. The length of rod is 1.14M. Water tank height is 1.27M, and its diameter is 0.5M. The heat capacity is assumed as constant.

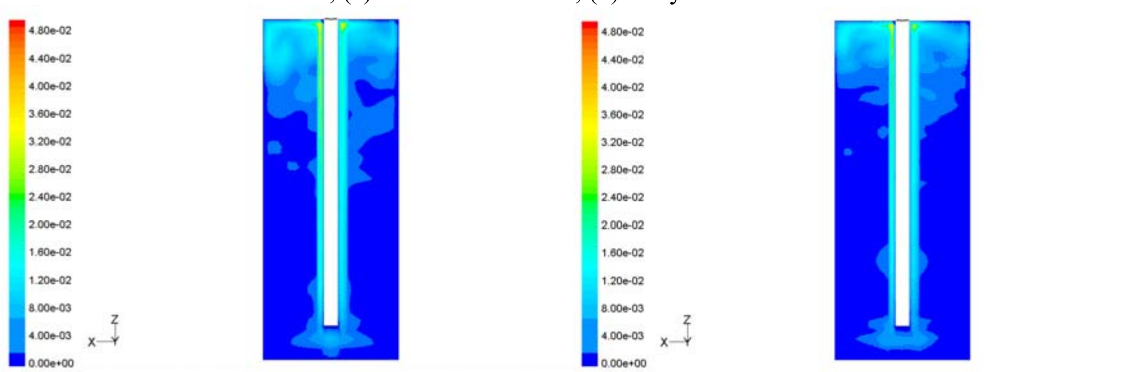
Figure 2 illustrates the velocity vector profile of buoyancy-driven flow caused by a condenser immersed in a water tank after 30 minutes of heating. In figure 2(a), it shows that buoyancy-driven flow is produced and moves up along the vertical heating surface when the vertical condenser is employed. Unlike the situation of a vertical condenser, significant mixed convection through the entire water tank is created when a L-style condenser is immersed in the tank. The difference is totally produced due to mixing caused by the extended non-vertical part of the L-style condenser.

To understand the process of buoyancy-driven flow caused by an immersed condenser in a water tank, a symmetric 2D cut-plane was employed to describe the movement of the buoyancy-driven flow. Figure 3 plots the 2D velocity contour of the buoyancy-driven flow caused by a vertical immersed condenser at 30min and at 60min. It clearly illustrates that a laminar buoyancy-driven flow is formed in the water surrounding the vertical condenser. This water thus rises up because of the force of gravity acting on the density variations. The majority of water in the tank, however, still has not been affected by the convective flow, except at the top and the layer surrounding the surface of a heated condenser. Meanwhile, the motion of the buoyancy-driven flow is gradually reduced with the increase of heating time, as the velocity profile of Fig. 3(b) is less than that of Fig. 3(a). Figure 4 illustrates the transient temperature contour in the water tank. Significant temperature stratification is produced due to the buoyancy-driven flow. When the water surrounding the vertical condenser is heated, this water is forced to rise up due to the buoyancy. After hot water reaches the top of water tank, it exchange heat with cold water below through thermal diffusion only. Thus significant temperature stratification in water tank is formed from top to bottom. From Figures 4(a) and 4(b), it is observed that the temperature stratification in the tank is a strong function of time.



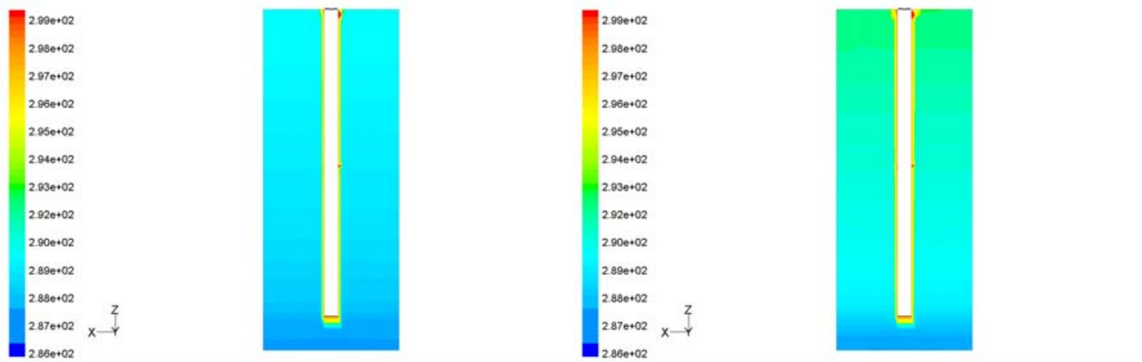

(a) (b)

Figure 2 The velocity (m/s) vector profile of buoyancy-driven flow caused a condenser immersed in a water tank; (a) vertical condenser; (b) L-style condenser.




(a) (b)

Figure 3 The velocity (m/s) contour of buoyancy-driven flow along a symmetric cutplane; (a) at 30min; (b) at 60min; the condenser is vertical.

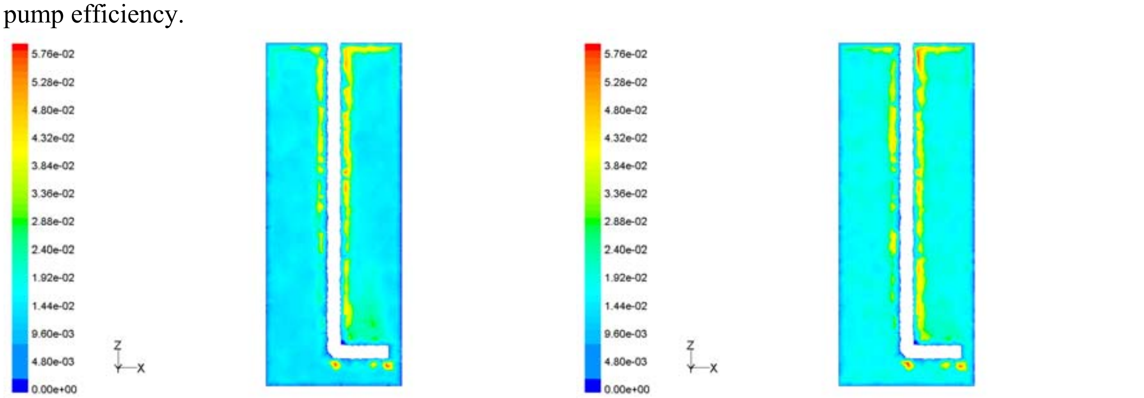



(a) (b)

Figure 4 The temperature (˚K) contour of buoyancy-driven flow along a symmetric cutplane; (a) at 30min; (b) at 60min; the condenser is vertical.

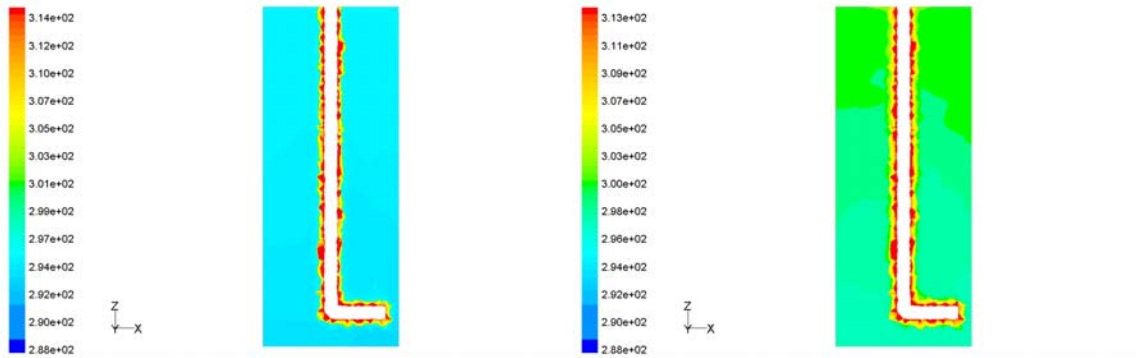
Strong convective heat transfer is produced in a water tank if a L-style condenser is immersed. Fig. 5 illustrates the 2D velocity vector profile when an L-style condenser is immersed in a water tank. It should be noted that a laminar buoyancy-driven flow surround the surface of a heated condenser is replaced by a strong convective fluid flow across the entire tank. The result is a well mixed fluid with close to uniform water temperature distribution, except for the region near the surrounding area of the condenser coil. The reason for such phenomenon is mainly due to the non-vertical portion of the L-style condenser. The buoyancy-driven flow caused by the extension part of a L-style condenser leads to the circulating flow in the water tank. As a result, mixed convective zone for the entire tank is formed. The temperature profile in the case of using L-style condenser is also different since an evenly distributed temperature exists in the tank. Compared to Fig. 4, a nearly uniform temperature profile is formed, as shown in Fig. 6, although a very tiny temperature differential still exists between the top and the bottom of the water. During the test, the water tank was instrumented with twin parallel columns of thermocouples at a radius of three inches and six inches respectively. Each column had seven thermocouples spaced at four-inch intervals, spanning the height of water to within one inch of the top and bottom. Testing data indicate that the temperatures at the same water levels are close to each other, as shown in both Figures 4 and 6. Figure 7 shows the temperature validation between testing data and predicted result. The experimental and simulated data matched well, which confirms that calculated data by a CFD code was accurate. Fig. 7(a) shows that the temperature difference in a water tank could be up to 15$^{o}$C after 5 hours heating. The temperature remains nearly uniform at any time in the case of the L-style condenser.

As mentioned earlier, the temperature stratification in the water tank, in general, gives rise to less than optimal performance of a given design of heat pump. If the tank is evenly mixed, the total energy in the tank does not change, but the hottest temperature is lower, and the heat pump thus runs at a more efficient state. Therefore, L-style condenser is helpful to eliminate the penalty imposed by the temperature stratification and improves the heat pump efficiency.



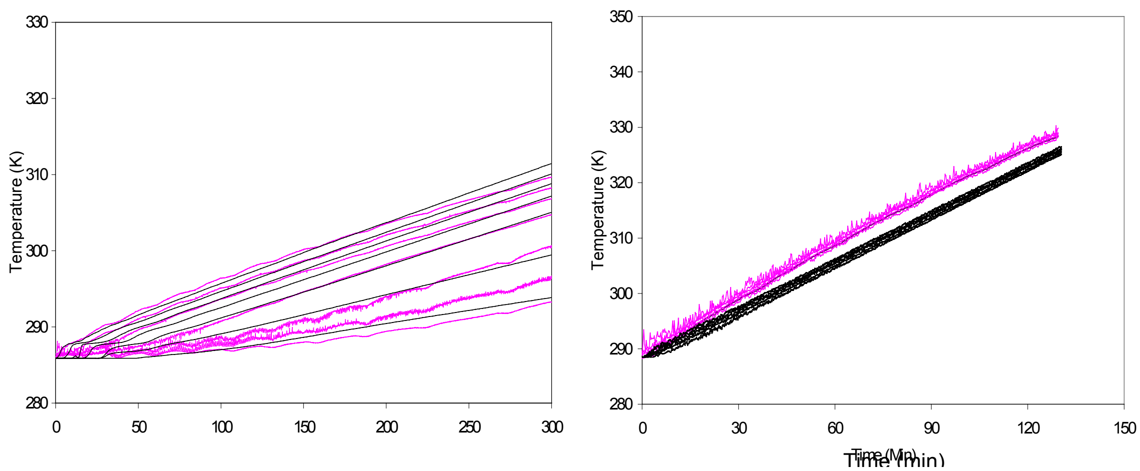

(a) (b)

Figure 5 The velocity contour of buoyancy-driven flow along a symmetric cut-plane; (a) at 30min; (b) at 60min; the condenser is L-style.




(a) (b)

Figure 6 Temperature (˚K) contour of buoyancy-driven flow along a symmetric cut-plane; (a) at 30min; (b) at 60min; the condenser is L-style.




(a) (b)

Figure 7 Tank temperature (˚K) comparisons of testing data and predicted result; Red line is testing data; Black line is simulated data; (a) vertical case; (b) L-style case.

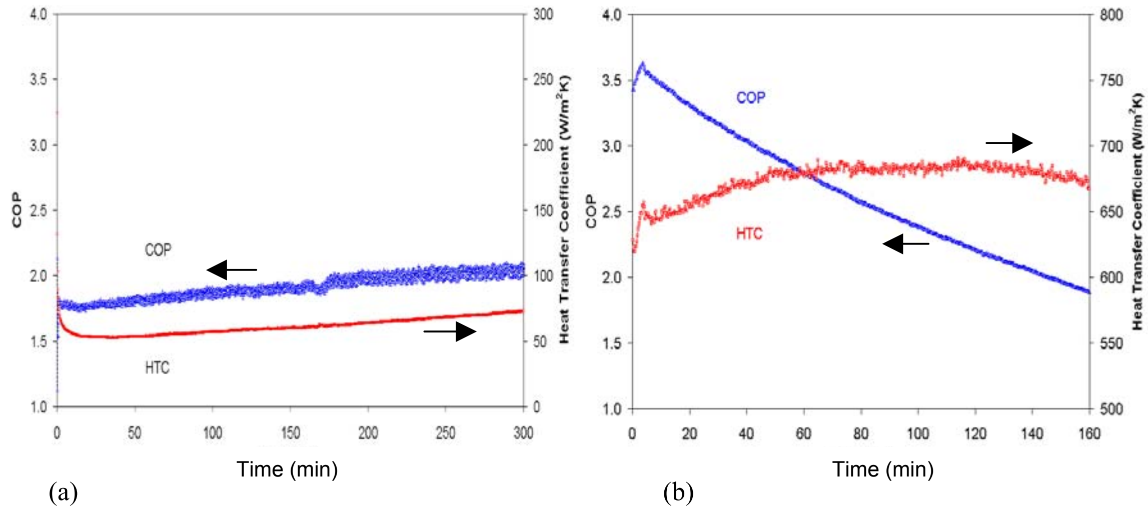



Figure 8. Comparisons of system COP and condenser heat transfer coefficient, (a) vertical condenser; (b) L-style condenser

Figure 8 shows the comparison of system COPs and heat transfer coefficients with vertical and L-shape condensers. Clearly, the L-shape has much higher heat transfer coefficients, which resulted in higher COP than that of the vertical condenser.

## Conclusions

A CFD code was used to study water temperature stratification inside a water tank that was heated by a heat pump. The temperature stratification was caused by the shape of the condenser coil that was inserted into the tank from an opening on the top of the tank.

Two different condenser coils, one vertical and an "L" shape, were analyzed and tested. The CFD simulated data indicated that the vertical coil would result in water temperature stratification. But, the L shaped coil showed the breaking up of the stratification and resulted in a very uniform water temperature inside the tank. Test data confirmed the simulated results. This study will be useful for future designs of heat pump water heaters that employ internal condensers.

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