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# Energy Efficiency Improvement in a Water Heater With a Run-Around Heat Exchanger System

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## ABSTRACT

A significant portion of the energy consumed by home appliances that uses hot water is used for heating cold supply water. Such home appliances generally use supply water at a temperature lower than the ambient temperature. Thus, the water is normally heated from below the ambient temperature to its designed maximum temperature by the appliance, often using natural gas or an electrical heater. During the period when the heater is energized, electrical energy is used to increase a water temperature to the designed maximum temperature. It may be possible to save energy by preheating the supply water before this period. In order to save the energy, a run-around heat exchanger system is used to transfer heat from the ambient to the water before the electrical heater is energized. A simple model to predict the dynamic performance of this system is developed and validated using experimental data. Despite the additional power consumption by the fan and pump, the experimental data and analysis show that overall energy efficiency can be improved.

## 1. INTRODUCTION

A typical home appliance produced in the U.S. in the 1990s using electricity for water heating is nearly 30% more efficient than such appliances produced in 1972 (Association of Home Appliances Manufacturers, 1991). These efficiency gains were achieved mainly by reducing hot water use through lower designed maximum temperatures. According to Hojer (2004), a modern washing machine (at 60°C), one of the home appliances using a water heating system, uses 3.42 MJ (0.95 kWh) per cycle, where 2.74 MJ (0.76 kWh) is electricity for water heating. The machine is assumed to be used 200 times/year, as reported by the Swedish Consumer Agency (1996), thus the annual electricity demand for the machine is 684 MJ. Electricity for water heating is about 550 MJ which is 80% of total consumed electricity. Thus, the energy used by an electrical heater can be a considerable part of the overall energy consumed in such home appliances.

### 1.1 Background

Consumer surveys reported by U.S. Department of Energy (1989) indicated that changes in washing habits had resulted in significant energy savings, and that 20% of the average energy use in clothes washing would be saved if the warm rinse option on a clothes washing machine in the U.S. was not used. This work also indicated that clothes dryer electricity could be reduced by around 15% with automatic termination controls that sensed dryness and automatically shut off the dryer and improved insulation.

Lewis (1983) developed a prototype heat pump clothes dryer (HPCD). Test of the HPCD showed electricity savings of 50-60% relative to conventional electric clothes dryers. Furthermore, microwave clothes dryers tested in the U.S. provided 26% electricity savings compared to standard clothes dryers.

Tests conducted by Lebot (1990) showed that horizontal-axis clothes washing machines, which were predominant in Europe, were nearly three times as energy efficient as vertical-axis washing machines of comparable size because they used much less water. Abbate (1988) and Nogard (1989) introduced an innovative horizontal-axis washing machine that eliminated filling the clothes tub with hot or warm water when the heater was on. Instead, hot detergent solution was continuously circulated and sprayed on the spinning clothes. This technique significantly reduced hot water, detergent, and energy use compared to standard European clothes washing machines.

Zegers and Molenbroek (2000) developed heat-fed washing machines. In these washing machines, an extra heat exchanger was placed at the bottom of the machine between the drum and the bottom of the tub to use hot water from a district-heating or central-heating installation as a heating source. They concluded that heat-fed washing machines connected to district heating resulted in the highest energy savings, much more than hot-fill machines heated by electricity, whereas the energy savings of heat-fed washing machines connected to central heating were comparable to hot-fill machines.

Persson (2007) developed a washing machine heated by sources other than electricity, such as a hot-water circulation loop, and its simulation model. The machine used supply water from a cold water pipe that was heated by the hot-water circulation loop via a heat exchanger built into the washing machine (heat-fed machines). This study showed that almost all electricity for heating could be replaced by using a hot-water source at 70°C for the washing machine. It indicated that larger savings occurred with supply temperatures around 55°C. In addition to the hot-water circulation loop, it suggested an alternative way to save electricity, connecting the washing machines to the domestic hot water pipe (hot-water-fed machines). However, the electrical savings with this measurement were much smaller. Persson simulated a hot-water-fed machine and compared the results to simulations of a heat-fed machine. The simulation results showed that 2.2 MJ (0.6 kWh) electricity per cycle was used for the hot-water-fed model and 1.4 MJ (0.4 kWh) electricity per cycle for the heat-fed model, if water at 60°C was supplied.

Persson and Ronnelid (2007) developed a simulation model using heating energy from solar collectors, district heating or a boiler rather than using electric energy. A washing machine equipped with heat-fed machine system was simulated together with solar heating systems. It showed that the washing machine integrated with the heat-fed machines and advanced solar “combisystems” gave higher energy savings.

## 1.2 Objectives

It is obvious that there has been some research on reducing the electrical energy consumption of home appliances. However, most of prior work has focused on the supply water temperature. Energy conservation can be achieved by increasing the water temperature during operation of the machine. For that reason, research is necessary to study methods to increase energy efficiency during the operation time. The objectives of this study were as follows; 1) Quantify the energy savings possible with a run-around heat exchanger system used before the electrical heater is energized. 2) Increase the water temperature in the tub through heat transfer from the environment using fans, a heat exchanger, and a recirculation pump integrated into the water heating system. 3) Use two different types of heat exchangers to explore the impact of heat exchanger efficiency on energy efficiency. 4) Develop a simulation model to predict the system performance and estimate the performance of a near-optimum design.

## 2. EXPERIMENTAL DESCRIPTIONS

The machine having a water heating system was carefully instrumented and equipped with a run-around heat exchanger system. A schematic of experimental apparatus is shown in Figure 1. In order to measure temperatures ( $\pm 0.12^\circ\text{C}$ ), type-T thermocouples and precalibrated RTDs (Resistance Temperature Detectors) were used. The temperatures of the supply water were measured by placing an RTD in a supply reservoir. The reservoir was located on the balance to measure the amount of water ( $\pm 10\text{g}$ ) going into the tub. The heat exchanger was placed at one side of the machine by cutting the side of the machine, and fans were connected to the heat exchanger. A supply pump was used to connect the reservoir and the input pipe of the machine and to make the supply water enter the tub, and an extra pump was used optionally. The water heating system used in this study had 672 seconds ( $\pm 1\text{s}$ ) of the heater-off period which is 17 pump cycles. One cycle of this period was composed of 22.5 seconds of pump on time and 15.5 seconds of pump off time. During the pump on time, the water in the tub circulated through the pump and the run-around heat exchanger system.

## 3. MODELING APPROACH

The model is based on conservation of mass and energy along with simple quasi-steady heat exchanger models. The simulation uses the  $\varepsilon - NTU$  method for heat exchanger analysis and a thermal resistance network to calculate the overall heat transfer coefficient. The physical system upon which the simulation is based is shown schematically in Figure 2. An energy balance applied to the tub of the machine is given by

$$\dot{Q}_{\text{in}} - \dot{Q}_{\text{out}} + \dot{Q}_{\text{gen}} = \dot{Q}_{\text{st}} \quad (1)$$

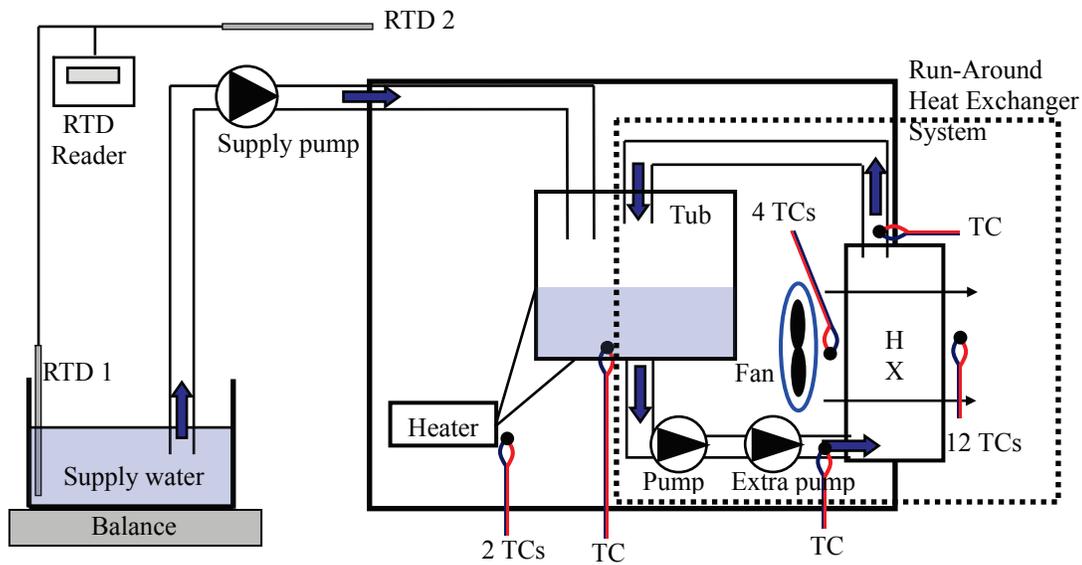


Figure 1: Schematic of experimental apparatus

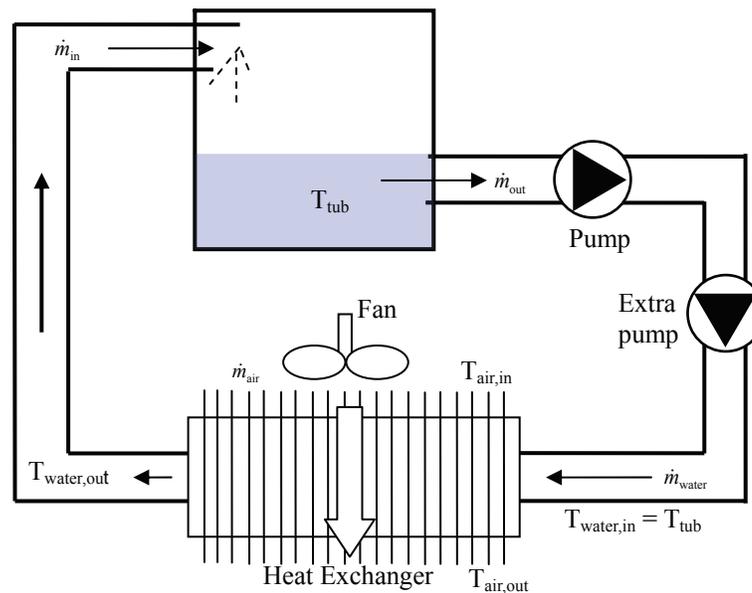


Figure 2: System schematic of the simulation model

For this system, the energy generation term is zero. The inlet and outlet energy flows are calculated by assuming that water is incompressible and formulating the enthalpy fluxes. The specific heat and mass flow rate of water are prescribed (taken as known). Since an important parameter is the temperature of the water in the tub, the energy balance takes changes in stored energy into account. The energy balance equation becomes,

$$\dot{m}_{in} c_{p,water} T_{water,in} - \dot{m}_{out} c_{p,water} T_{water,out} = M_{tub} c_{p,water} \frac{dT_{tub}}{dt} \quad (2)$$

where  $M_{\text{tub}}$  is the mass of water in the tub. It is assumed that the flow rate of water is constant ( $\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$ ), and  $T_{\text{tub}} = T_{\text{water,out}}$  because water in the tub is well mixed during heater-off period.

The conditions of the heat exchanger are taken as quasi-steady and energy storage changes are neglected. Thus, a thermal resistance network can be used in the steady state for the heat exchanger. In the  $\varepsilon - NTU$  method, the actual rate of heat transfer in the heat exchanger can be related to the effectiveness using the following equation:

$$\dot{Q} = \varepsilon \dot{Q}_{\text{max}} \quad (3)$$

where  $\dot{Q}_{\text{max}}$  is the maximum possible heat transfer rate of the heat exchanger. Because flow rates of air and water can vary by using different fans and the extra pump, the maximum heat transfer rate in the heat exchanger can be given by Equation (4). The number of heat transfer units ( $NTU$ ) is defined as Equation (5).

$$\dot{Q}_{\text{max}} = \min\{\dot{m}_{\text{air}} c_{p,\text{air}} (T_{\text{air,in}} - T_{\text{water,in}}), \dot{m}_{\text{water}} c_{p,\text{water}} (T_{\text{air,in}} - T_{\text{water,in}})\} \quad (4)$$

$$NTU = UA / C_{\text{min}} \quad (5)$$

where  $C$  is a heat capacity rate ( $\dot{m}c_p$ ) and the  $C_{\text{min}}$  can be the heat capacity rate of air or water. The overall heat transfer coefficient,  $UA$ , is calculated using the thermal resistance network and appropriate heat transfer correlations for the heat exchangers. Two different types of heat exchangers were used in the experiments, one was a plain-fin-round-tube heat exchanger, and the other was a slit-fin-round-tube heat exchanger. An appropriate  $\varepsilon - NTU$  relation was obtained, depending on the heat exchanger geometry from Incropera *et al.* (2006).

As all thermo-physical parameters are known, the equations are then integrated in time using a Runge-Kutta method to obtain predictions of the tub temperature in time,  $T_{\text{tub}}(t)$ .

## 4. RESULTS AND DISCUSSION

The cycle used in this study has a designed maximum temperature of 95°C. The water temperature in the tub for baseline experiments was measured using different supply-water temperatures. The temperature with the prototype system was measured using two different fans and an extra pump at different supply-water temperatures. The primary parameter for this study is the temperature increase of water in the tub during heater-off period. Because raising the tub-water temperature before the heater is energized will reduce the length of time it is on, raising the tub water temperature directly affects energy usage during the heater-on period.

### 4.1 Baseline Experiment Results

In order to investigate the effect of parameters such as a fan and a pump on the system, experiments without the fan and the pump were conducted at different supply-water temperatures and different average air temperatures inside the machine. Figure 3 shows the results of baseline experiments. The vertical axis,  $\Delta T$ , is the temperature increase from initial temperature to final temperature of water in the tub during heater-off period. As one would expect, the trends reflect a linear decrease in  $\Delta T$  as the supply-water temperature increases, and  $\Delta T$  increases as the average air temperature inside the machine increases. The baseline experiment results will be used to calculate the energy efficiency of the prototype system.

### 4.2 Effect of the Flow Rate of Air (without an extra pump, $\dot{m}_{\text{water}} = 0.01$ kg/s)

Prototype experiments with different flow rates of air were performed using a plain-fin heat exchanger. Two fans were used to produce air-flow rates of 0.056 m<sup>3</sup>/s (six-inch blade fan) and 0.17 m<sup>3</sup>/s (nine-inch blade fan). Figure 4 shows the temperature increase of water in the tub with different flow rates of air. Increasing the air-flow rate from 0.056 m<sup>3</sup>/s to 0.17 m<sup>3</sup>/s (almost 3 times) produces the maximum water temperature increase of 25%. There are several reasons for this small effect: Since the flow rate of water is small,  $C_{\text{min}}$  occurs at the water-side and the heat transfer coefficient of water-side is relatively small due to laminar flow. Furthermore, although the dominant resistance is typically on air-side in air-cooled heat exchangers, the dominant resistance in this prototype occurs on the water-side.

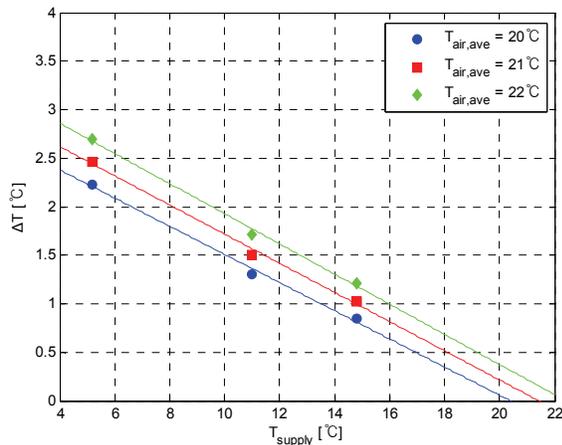


Figure 3: Baseline results (lines shown for readability)

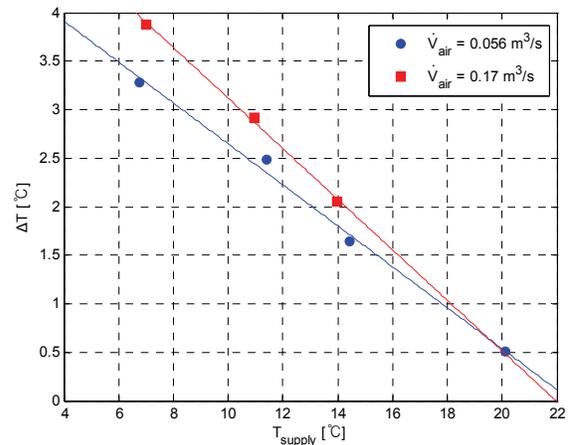


Figure 4: Effect of air-flow rate (lines for readability)

#### 4.3 Effect of the Flow Rate of Water (with $\dot{V}_{air} = 0.17 \text{ m}^3/\text{s}$ )

Because of the water-side resistance discussed in section 4.2, the flow rate of water was increased to improve performance. At sufficiently high flow rates, the flow in the tube becomes turbulent causing greater heat transfer rate than that of laminar flow, and  $C_{min}$  changes to the air-side. These conditions prevailed at a water-flow rate of 0.085 kg/s. The temperature increase of water in the tub with and without the extra pump is shown in Figure 5. A maximum increase of 135% in  $\Delta T$  was achieved by increasing the flow rate of water from 0.01 kg/s to 0.085 kg/s.

#### 4.4 Effect of Heat Exchanger Type

Because the dominant resistance changed to the air-side by using the extra pump and the plain-fin heat exchanger, an enhanced fin design could be used to effectively improve the air-side heat transfer performance. In this study, the slit-fin and round-tube heat exchanger was investigated. The surface efficiency was obtained using the sector method, and the heat transfer coefficient was calculated by the correlations provided by Wang *et al.* (2001). In Figure 6, the temperature increase of water in the tub for plain-fin and slit-fin heat exchangers is compared when the extra pump and the nine-inch blade fan were used. The slit-fin enhanced the air-side performance and  $\Delta T$  increased.

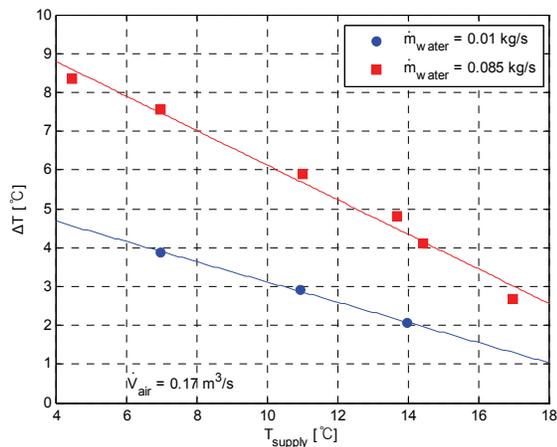


Figure 5: Effect of water-flow rate (lines for readability)

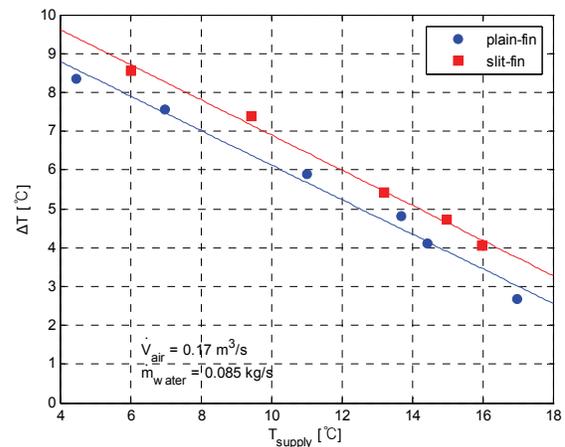


Figure 6: Effect of heat exchanger (lines for readability)

#### 4.5 Comparison Between Experimental and Simulation Model Results

Figure 7 and 8 show that simulation tends to underpredict the temperature increase compared to experimental data; however, trends are predicted very well. Differences in  $\Delta T$  between experimental and simulation results are largely

because in the baseline, as seen in Figure 1, there is a temperature increase in the tub due to heat gains from the inside wall of the tub which has higher temperature than the supply water and from the heat transfer between the tub and air inside the machine. In order to estimate the water temperature increase in the tub more exactly, these effects must be considered in the simulation model.

#### 4.6 Predictions from Simulation Model

The effect of the air-flow rate is shown in Figure 9. Because the flow in the tube is turbulent and the fluid of  $C_{min}$  is the air for  $\dot{V}_{air} < 0.3$  m<sup>3</sup>/s in the case of  $\dot{m}_{water} = 0.085$  kg/s,  $\Delta T$  increases much more than that of the case in which  $C_{min}$  is water-side (beyond 0.3 m<sup>3</sup>/s). The effect of the water-flow rate is shown in Figure 10. As the flow rate of water increases, the  $\Delta T$  becomes constant.

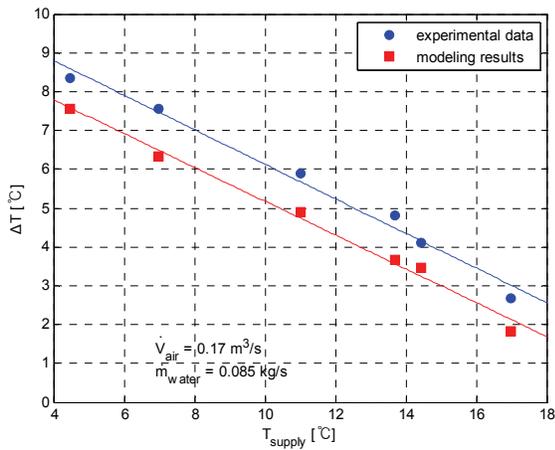


Figure 7: Comparison of the simulation to experiments (nine-inch blade fan and plain-fin heat exchanger with the extra pump—lines shown for readability)

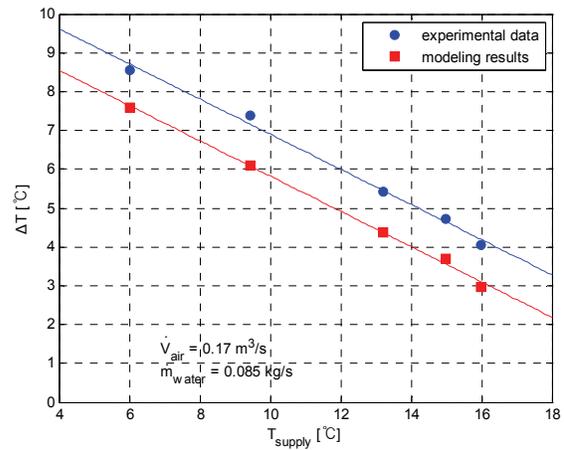


Figure 8: Comparison of the simulation to experiments data (nine-inch blade fan and slit-fin heat exchanger with the extra pump—lines shown for readability)

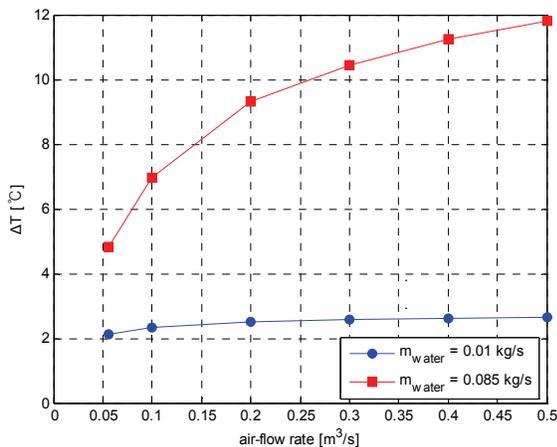


Figure 9: Simulated results with various air flow rates (curves shown for readability)

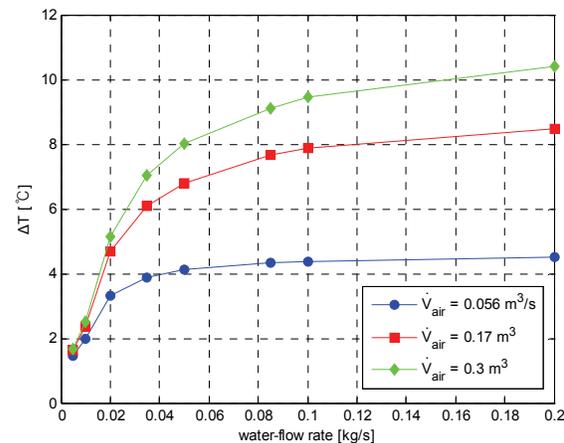


Figure 10: Simulated results with various water flow rates (curves shown for readability)

## 5. PERFORMANCE COMPARISONS

In order to estimate the optimum design in terms of the energy efficiency, the following parameter is considered:

$$\Delta E_{\text{saved}} = \frac{E_{\text{old}} - E_{\text{new}}}{E_{\text{old}}} = \frac{-(E_{\text{fan}} + E_{\text{pump}})}{M_{\text{tub}} c_p (T_f - T_{i,\text{old}})} + \frac{T_{i,\text{new}} - T_{i,\text{old}}}{T_f - T_{i,\text{old}}} \quad (6)$$

where

$$E_{\text{old}} = M_{\text{tub}} c_p (T_f - T_{i,\text{old}}) \quad (7)$$

$$E_{\text{new}} = M_{\text{tub}} c_p (T_f - T_{i,\text{new}}) + E_{\text{fan}} + E_{\text{pump}} \quad (8)$$

In Equation (6),  $T_f$  is the designated maximum temperature of the cycle (95°C), and  $T_{i,\text{new}}$  is the new initial temperature of heater-on period. Additionally,  $T_{i,\text{old}}$  can be calculated using the baseline experiment results (Figure 3). Based on previous results, four cases are compared:

- 1) Plain-fin heat exchanger with the nine-inch blade fan and no extra pump
- 2) Plain-fin heat exchanger with the extra pump and the nine-inch blade fan
- 3) Slit-fin heat exchanger with the nine-inch blade fan and no extra pump
- 4) Slit-fin heat exchanger with the extra pump and the nine-inch blade fan

The supply-water temperature for all of these experiments was set near 5°C. The results are given in Table 1. For comparison, a “ $\Delta E_{\text{saved,max}}$ ” is calculated by assuming the water in the tub is raised to the average air temperature inside the machine. Energy consumptions of the nine-inch blade fan and the extra pump were 34 kJ and 49 kJ, respectively. It turns out that case 4 is the better design for this study. Although the energy saved,  $\Delta E_{\text{saved}}$ , is 5.85% of the total baseline usage in case 4, it is 46% of the maximum case (12.85%).

If the existing pump has sufficient power to increase the flow rate of water to 0.085 kg/s or above,  $E_{\text{pump}}$  in Equation (6) can be zero. In this case, the energy saved becomes 6.6% which is 51% of the maximum possible energy savings. Such an approach also avoids the cost for the extra pump.

During the pump-off time, water in the tub is still and does not circulate through the heat exchanger. This time consists of approximately 40% of the total heater-off time. As a result, it is recommended to reduce the pump-off time, or keep turning the pump on to circulate water in the tub continuously. From the simulation model in the same situation as case 4, the temperature increase would result in  $T_{i,\text{new}}=16.28^\circ\text{C}$  (as compared to 14.56°C in the current case 4) if pump is operated continuously during heater-off period (672 seconds). Moreover, the actual temperature increase could be higher, because of baseline effects discussed in section 4.5.

Table 1: Comparison of  $\Delta E_{\text{saved}}$  in four cases

Case	$T_{\text{supply}} [^\circ\text{C}]$	$T_{i,\text{old}} [^\circ\text{C}]$	$T_{i,\text{new}} [^\circ\text{C}]$	$T_{i,\text{max}} [^\circ\text{C}]$	$\Delta E_{\text{saved}} [\%]$	$\Delta E_{\text{saved,max}} [\%]$
1	6.99	9.26	10.87	20.18	1.35	12.21
2	4.46	6.97	12.82	21.22	5.39	14.93
3	5.41	7.82	9.53	20.77	1.45	14.34
4	6.02	8.39	14.56	20.62	5.85	12.85

## 5. CONCLUSIONS

Motivated by the importance of saving energy in home appliances, this study of energy performance with a run-around heat exchanger was undertaken. The work reported in this study has made progress in saving energy by increasing the water temperature in the tub of a hot-water-using home appliance during the heater-off period. In addition, it provides a useful simulation model to predict the water temperature increase in the tub under different circumstances. In this study, the slit-fin and round-tube heat exchanger with the nine-inch blade fan and the extra pump has the best energy savings of 5.68%. For achieving more energy efficiency, using a single, more powerful pump and elimination of the pump off time are recommended. The run-around heat exchanger system investigated in this study could be applied to clothes washing machines, dishwashers, and even some clothes dryers (steam-type dryers).

## NOMENCLATURE

$c_p$	specific heat at constant pressure (kJ/kg-K)	<b>Subscripts</b>			
$E$	Energy (kJ)	air	air	old	old
$M$	mass (kg)	ave	average	pump	pump
$\dot{m}$	mass flow rate (kg/s)	f	final	saved	saved
$NTU$	number of transfer unit, $UA/C_{\min}$	fan	fan	st	stored
$\dot{Q}$	heat transfer rate (kW)	gen	generation	supply	supply
$UA$	overall heat transfer coefficient (kW/K)	i	initial	tub	tub
$t$	time (s)	in	inlet	water	water
$\dot{V}$	volumetric flow rate (m <sup>3</sup> /s)	max	maximum		
		min	minimum		
		new	new		
		out	outlet		
<b>Greek symbol</b>					
$\varepsilon$	heat exchanger effectiveness				

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