Applied Thermal Engineering 29 (2009) 3110–3117

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

A run-around heat exchanger system to improve the energy efficiency of a home appliance using hot water

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ARTICLE INFO

Article history:
Received 2 March 2009
Accepted 2 April 2009
Available online 8 April 2009

Keywords:
Energy efficiency
Heat exchanger
Home appliance
Water heater

ABSTRACT

A significant portion of the energy consumed by many home appliances using hot water is used to heat cold supply water. Such home appliances generally are supplied water at a temperature lower than the ambient temperature, and the supply water is normally heated to its maximum operating temperature, often using natural gas or an electrical heater. In some cases, it is possible to pre-heat the supply water and save energy that would normally be consumed by the natural gas or electrical heater. In order to save the energy consumed by an appliance using water heater, a run-around heat exchanger system is used to transfer heat from the ambient to the water before an electrical heater is energized. A simple model to predict the performance of this system is developed and validated, and the model is used to explore design and operating issues relevant to the run-around heat exchanger system. Despite the additional power consumption by the fan and pump of the run-around heat exchanger system, the experimental data and analysis show that for some systems the overall energy efficiency of the appliance can be improved, saving about 6% of the energy used by the baseline machine.

1. Introduction

The rising demand and cost for electrical energy continues to motivate research directed at improving the energy efficiency of machines that use electrical power. The Association of Home Appliances Manufacturers (USA) [1] reported that a typical home appliance produced in the US in 1990 using electrical energy to heat water was nearly 30% more efficient than such appliances produced in 1972. These efficiency gains were achieved mainly by decreasing the hot water temperature. According to Hojer [2], a contemporary clothes-washing machine using hot water at 60 °C consumes 3.42 MJ (0.95 kWh) per cycle, with 2.74 MJ (0.76 kWh) used to heat the water. As reported by the Swedish Consumer Agency [3], the typical washing machine is used about 200 times per year, and thus the annual electricity demand for the machine becomes 684 MJ, and the electricity for water heating is about 550 MJ, roughly 80% of total electricity consumed by the clothes-washing machine. Thus, the energy used by an electrical heater can be a considerable part of the overall energy consumed in such home appliances. This work investigates the feasibility of one method to save electrical energy in a water heater. In particular, we study the energy savings achieved by using a run-around heat exchanger system. This system was used to pre-heat the supply-water temperature before the water heater was energized.

1.1. Energy saving methods from previous work

Miller et al. [4] reported that the efficiency of an electric-resistance water heater was improved from 82% to 92% through better insulation and the use of heat traps. This efficiency improvement could save about 300 kWh/year in a typical US residence. According to Geller [5], significant improvements in the efficiency of electric water heaters are possible by using heat pumps that consume 50–70% less electricity than simple electric resistance heaters. In terms of economics, it was estimated that with a first cost of $800–$1200 – four times that of a convectional electric water heater – the heat-pump water heater could be expected to have a simple payback period of less than 6 years.

Consumer surveys conducted by the US Department of Energy [6] show that changes in washing habits can result in significant energy savings. For instance, 20% of the average electrical energy use in clothes-washing machines would be saved if the warm rinse option on washing machines in the US were not used. This report also indicated that electricity used by a clothes dryer could be reduced by around 15% with automatic termination controls that sensed dryness and automatically shut off the dryer and improved insulation.

Lewis [7] developed a prototype heat pump clothes dryer (HPCD). Tests of the HPCD showed that electricity consumption could be reduced by 50–60% relative to conventional electric clothes dryers. In addition to the HPCD, Turiel et al. [8] showed that microwave clothes dryers could save 26% of the electricity consumed by standard clothes dryers.

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doi:10.1016/j.applthermaleng.2009.04.005

Nomenclature

A	area (m^2)
c_p	specific heat at constant pressure (kJ/kg K)
C_r	heat rate capacity ratio, $(\dot{m}c_p)_{\min}/(\dot{m}c_p)_{\max}$
D	diameter (m)
E	energy (kJ)
h	heat transfer coefficient (kW/m^2 K)
k	thermal conductivity (kW/m K)
L	length (m)
M	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
N_{TU}	number of transfer unit, $UA/(\dot{m}c_p)_{\min}$
Pr	Prandtl number
$\dot{Q}$	heat transfer rate (kW)
R	thermal resistance (K/kW)
Re	Reynolds number
T	Temperature (°C)
t	time (s)
UA	overall heat conductance (kW/K)
$\dot{V}$	volumetric flow rate (m^3/s)

Subscripts

a	air; air-side
ave	average
c	conduction
f	fin; final
fan	fan
g	gain
i	initial
in	inlet; inner
max	maximum
min	minimum
new	new
o	surface
old	old
out	outlet; outer
pmp	pump
st	stored
sup	supply
sys	system
T	total
tub	tub
tube	tube
w	water; water-side
xchr	heat exchanger

Greek symbols

ε	heat exchanger effectiveness
η	efficiency

Tests conducted by Lebot [9] 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 [10] and Nogard [11] 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 [12] 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 extract heat from a district-heating or central-heating installation. They concluded that heat-fed machines connected to district heating resulted in the highest energy savings compared to washing machines heated directly by electricity.

Persson [13] 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 the electricity used for heating could be saved by using a hot water source at 70 °C for the washing machine. In addition to the hot-water circulation loop, Persson 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 system were measured to be 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 was used at 60 °C.

Persson and Ronnelid [14] developed a simulation model of a system using energy from solar collectors, district-heating or a boiler rather than using direct electrical energy. A washing machine equipped as a heat-fed machine system was simulated together with solar heating systems. It was shown that the washing machine integrated with the heat-fed machines and advanced solar "combisystems" gave higher energy savings.

Most prior work on clothes-washing machines has considered significant system changes, including the use of auxiliary heat sources, heat pumps, and integration of multiple appliances. Such approaches will typically lead to substantial increases in the first cost and complexity of the system. Our goal is to examine a much simpler approach of using a run-around heat exchanger system to make use of ambient heat for preheating the supply water. Of course, the energy savings achieved in this simpler system is expected to be smaller than the systems discussed above, but the cost and complexity is also reduced.

2. Experimental apparatus

A washing machine was carefully instrumented and equipped with a run-around heat exchanger system. A schematic diagram of the experimental apparatus is shown in Fig. 1, along with thermocouple and resistance temperature detectors (RTD) locations. In order to measure temperatures, type-T thermocouples (±0.12 °C)[1] and pre-calibrated RTDs (±0.01 °C) 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 in order to measure the amount of water (±10 g) going into the tub. In order to measure the water temperature inside the tub, a thermocouple was placed near the bottom surface of the tub (within a few mm), and the water temperatures at heat exchanger inlet and discharge were measured using thermocouples immersed in the flow. The air temperatures at

[1] Measurement uncertainty with a 95% confidence interval.

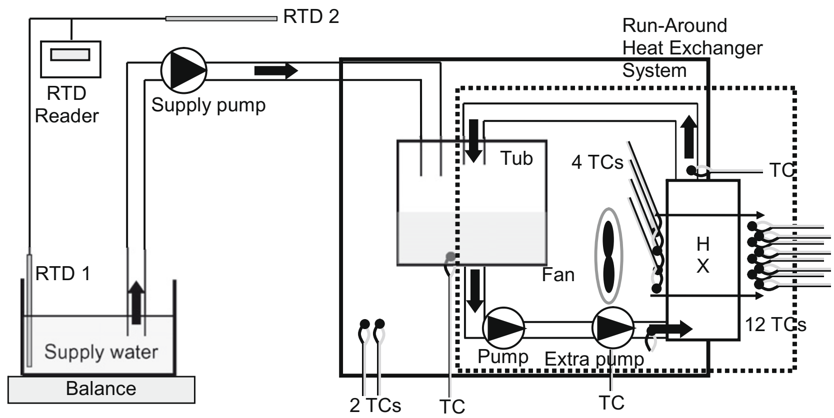

Fig. 1. Schematic diagram of experimental apparatus.

heat exchanger inlet and discharge were measured using four and twelve thermocouples, respectively, on evenly spaced grids. Finally, the air temperature inside the machine was measured using two thermocouples near the water heater inside the machine.

The heat exchanger[2] was placed at one side of the machine, and an air-flow path was accommodated by cutting a hole in the sheet metal forming the side of the machine. Using a simple cardboard shroud, a fan was 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 (see Fig. 1). The water heating system used in this study had 672 s (±1 s) of a "heater-off" period, during which there were 26 s of starting pump-on time plus 17 pump cycles. One pump cycle was composed of 22.5 s of pump-on time and 15.5 s of pump-off time. During the pump-on time, the water in the tub circulated through the pump and, in some cases, the run-around heat exchanger system. All the temperature data were sampled at a rate of 400 samples per second, and 400 samples were averaged so that average temperatures were recorded every second by a data acquisition system.

3. Mathematical model

The model is based on the conservation of mass and energy along with simple quasi-steady heat exchanger models. The simulation uses the ε–N_{TU} method for heat exchanger analysis and a thermal resistance network to calculate the overall heat transfer coefficient, *UA* from established heat transfer correlations. The physical system upon which the simulation is based is shown schematically in Fig. 2. For the tub, there is no internal energy generation, and the tub is assumed to be well insulated. The rates of energy transfer associated with the water flow into and out of the tub are calculated by assuming water is a simple, incompressible substance. The specific heat and mass flow rate of the water are prescribed (taken as known). The thermal capacity of the water in the tub dominates changes in energy storage and an energy balance yields,

$$\dot{m}_{\mathrm{in}} c_{p,\mathrm{w}} T_{\mathrm{w,in}} - \dot{m}_{\mathrm{out}} c_{p,\mathrm{w}} T_{\mathrm{w,out}} = M_{\mathrm{tub}} c_{p,\mathrm{w}} \frac{dT_{\mathrm{tub}}}{dt}, \tag{1}$$

where M_{tub} is the mass of water in the tub. It is further assumed that the flow rates of water in and out of the tub are equal and the water within the tub is thermally well mixed; thus, $\dot{m}_{\mathrm{in}} = \dot{m}_{\mathrm{out}} = \dot{m}_{\mathrm{w}}$ and $T_{\mathrm{tub}} = T_{\mathrm{w,out}}$, and Eq. (1) can be written as

$$\dot{m}_{\mathrm{w}}(T_{\mathrm{w,in}} - T_{\mathrm{tub}}) = M_{\mathrm{tub}} \frac{dT_{\mathrm{tub}}}{dt}. \tag{2}$$

Because heat capacity of water in the tub, $M_{\mathrm{tub}} c_{p,\mathrm{w}}$ is much greater than total heat capacity of the heat exchanger composed of fins, tubes, and water in tubes, the energy storage change in the tub is dominant. Thus, the conditions of the heat exchanger are taken as quasi-steady, energy storage changes are neglected, and a thermal resistance network can be used for the heat exchanger. In the ε–N_{TU} method, the rate of heat transfer in the heat exchanger can be related to the effectiveness using the following equation:

$$\dot{Q} = \varepsilon \dot{Q}_{\mathrm{max}}, \tag{3}$$

where ε is heat exchanger effectiveness, which for the case under study is related to the heat exchanger N_{TU} by [15]

$$\varepsilon = 1 - \exp\left(\frac{N_{TU}^{0.22}}{C_r}\{1 - \exp(-C_r N_{TU}^{0.78})\}\right), \tag{4}$$

where $C_r = (\dot{m}c_p)_{\mathrm{min}}/(\dot{m}c_p)_{\mathrm{max}}$. Assuming the pipe between the tub and the heat exchanger to be well insulated, the maximum heat transfer rate in the heat exchanger can be written as

$$\dot{Q}_{\mathrm{max}} = \min\left\{\dot{m}_{\mathrm{a}} c_{p,\mathrm{a}}(T_{\mathrm{a,in}} - T_{\mathrm{tub}}), \dot{m}_{\mathrm{w}} c_{p,\mathrm{w}}(T_{\mathrm{a,in}} - T_{\mathrm{tub}})\right\}. \tag{5}$$

The number of transfer units, N_{TU}, for the heat exchanger is defined

$$N_{TU} = UA/(\dot{m}c_p)_{\mathrm{min}}, \tag{6}$$

where the overall heat conductance, *UA*, is calculated using the thermal resistance network and appropriate heat transfer correlations for the heat exchangers.

$$UA = (R_{\mathrm{a}} + R_{\mathrm{c}} + R_{\mathrm{w}})^{-1} \tag{7}$$

$$R_{\mathrm{a}} = \frac{1}{\eta_{\mathrm{o}} h_{\mathrm{a}} A_{\mathrm{T,a}}} \tag{8}$$

$$R_{\mathrm{c}} = \frac{\ln(D_{\mathrm{out}}/D_{\mathrm{in}})}{2\pi k_{\mathrm{tube}} L_{\mathrm{tube}} N_{\mathrm{tube}}} \tag{9}$$

$$R_{\mathrm{w}} = \frac{1}{h_{\mathrm{w}} A_{\mathrm{T,tube}}}, \tag{10}$$

where R_{a}, R_{c}, and R_{w} are the thermal resistance of air-side convection, the conduction through the tube, and the water-side convection, respectively. The surface efficiency, η_{o}, in Eq. (8) is related to the fin surface area (A_{f}), total surface area ($A_{\mathrm{T,a}}$), and fin efficiency (η_{f}) according to

$$\eta_{\mathrm{o}} = 1 - \frac{A_{\mathrm{f}}}{A_{\mathrm{T,a}}}(1 - \eta_{\mathrm{f}}). \tag{11}$$

[2] Geometric data for the heat exchangers are given in the Appendix A.

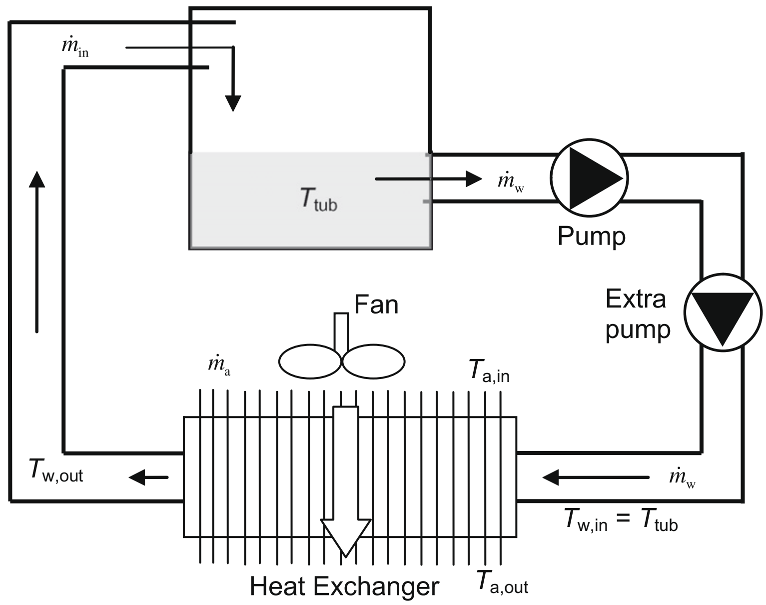

Fig. 2. System schematic of the simulation model.

Two different air-side designs were considered in the study. One design was a plain-fin-and-round-tube design, for which η_f was determined following Schmidt [16]. The other design was a slit-fin-and-round-tube design, for which h_a was determined using the correlations due to Wang and co-workers [17,18], and η_f was found using the well-known sector method. When the water-side flow was laminar, h_w was found using the correlation presented by Kays [19], and when the water flow was turbulent the correlation due to Gnielinski [20] was used. Thus, for prescribed water and air-flow rates, the air-side and water-side convection thermal resistances were known.

Energy balances on the air and water streams yield the following two equations:

$$\dot{Q} = \dot{m}_a c_{p,a}(T_{a,in} - T_{a,out}) \tag{12}$$

and

$$\dot{Q} = \dot{m}_w c_{p,w}(T_{w,in} - T_{tub}), \tag{13}$$

where the return pipe from the heat exchanger to the tub is also assumed to be well insulated.

The ordinary differential equation given in Eq. (2) is provided an initial condition on T_{tub}, and then subject to the algebraic equations given in Eqs. (3)–(13), with thermophysical properties and air and water-flow rates taken as known, the system of equations is marched forward in time using a Runge–Kutta integration and Newton–Raphson iteration available in a commercial software package (Engineering Equation Solver).

4. Experimental and simulation results

The wash cycle used by the machine in this study has a designed maximum water temperature of 95 °C. The water temperature in the tub for the 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. 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. Therefore, the most important parameter to measure or predict in this study is the temperature increase of water in the tub during water heater-off period.

4.1. Baseline experiment

In order to investigate the effect of component performance, such as that of the fan, pump, and heat exchanger on the system performance, experiments with different component arrangements were conducted at different supply-water temperatures and with different average air temperatures inside the machine. For the baseline experiments, no fan, pump, or heat exchanger were used. The results of baseline experiments are given in Fig. 3. In the figure, ΔT_g is the temperature increase from the initial tub-water temperature to the final water temperature during heater-off period, and ΔT_i is the difference between the supply-water temperature and the air temperature inside the machine. As one might expect, the trends reflect a linear

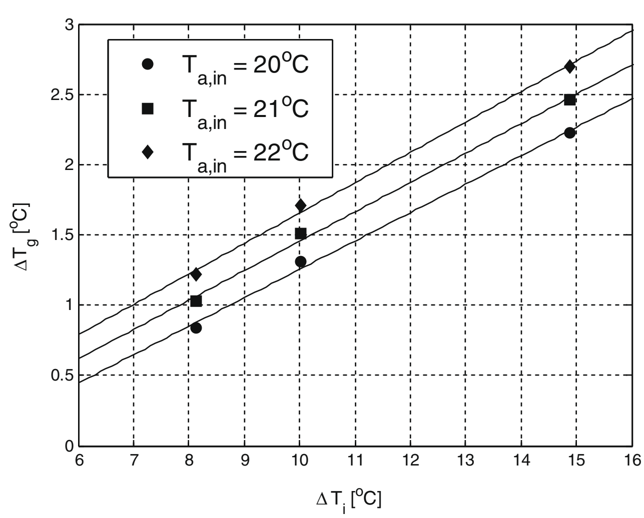

Fig. 3. Baseline results (lines shown for readability).

increase in ΔT_g as ΔT_i increases and a decrease in ΔT_g as ΔT_i decreases. These findings are consistent with a lumped-capacitance behavior over a fixed time interval, with a constant thermal resistance and capacitance. The results show that, contrary to the modeling assumption, heat transfer does occur to the water in the tub, even when the run-around heat exchanger system is not used. Thus, the tub is not perfectly insulated. These results will provide a baseline that allows for a meaningful calculation of the overall energy efficiency achieved with the prototype run-around heat exchanger system.

4.2. Prototype experiment

First of all, 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 $5.6(10^{-2})\,m^3/s$ (with a 15-cm blade fan) and $17(10^{-2})\,m^3/s$ (with a 23-cm blade fan). Fig. 4 shows the temperature increase of water in the tub with different air-flow rates. The data show that increasing the air-flow rate from $5.6(10^{-2})$ to $17(10^{-2})\,m^3/s$ (almost a three times increase) produces a relatively small change in tub-water temperature. There are several reasons for this small effect: since the mass flow rate of water is small, the heat transfer coefficient of the water-side is relatively small due to laminar flow. Furthermore, although the dominant resistance is typically on the air-side in air-to-liquid heat exchangers, the dominant resistance in this prototype occurs on the water-side for these conditions.

In order to mitigate the water-side convection resistance, the mass flow rate of water was increased by using an extra pump. At sufficiently high mass flow rates of water, the flow in the tube becomes turbulent causing greater heat transfer rate than that of laminar flow. This condition prevailed at a water-flow rate of $8.5(10^{-2})$ kg/s. The temperature increase of water in the tub with and without the extra pump is shown in Fig. 5, where the dramatic impact of this design parameter is evident. The tub-water temperature increase was essentially doubled when the water mass flow rate was increased from $1(10^{-2})$ kg/s to $8.5(10^{-2})$ kg/s at an air-flow rate of $17(10^{-2})\,m^3/s$.

Because the dominant thermal resistance occurred on the air-side at higher water-flow rates, an enhanced fin design was used to further enhance the system performance. Instead of using the plain-fin heat exchanger, a slit-fin geometry was adopted (see Appendix A). Although the water-side geometry was unchanged, the slit-fin design was expected to reduce the air-side thermal resistance by about 41%. In Fig. 6 the temperature increase of water in the tub for plain-fin and slit-fin heat exchangers is compared at the largest air-flow and water-flow rates. As the slit-fin enhanced the air-side performance, ΔT_g increased.

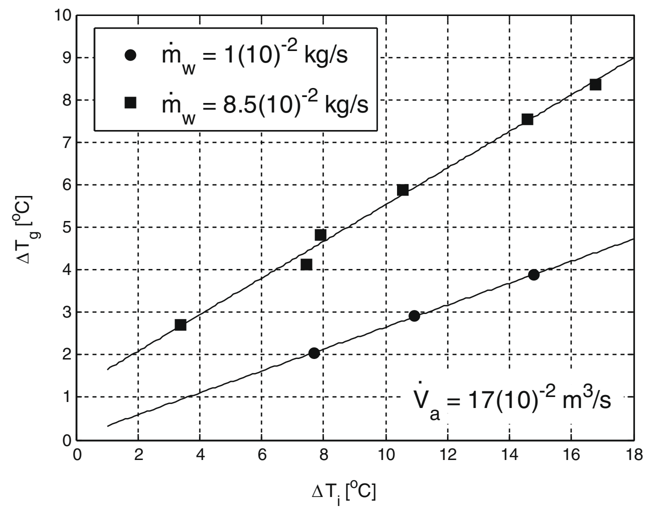

Fig. 5. Effect of water-flow rate (lines for readability).

4.3. Simulation results

Figs. 7 and 8 show that a simulation model tends to underestimate the temperature increase compared to experimental data; however, trends are predicted very well. Differences in ΔT_g between experimental data and simulation predictions are largely because in the baseline, as seen in Fig. 1, there is a temperature increase in the tub due to a heat leak. In order to estimate the water temperature increase in the tub more exactly, these effects must be considered in the simulation model.

By using the simulation model, the effects of air and water-flow rates on system performance were investigated. Fig. 9 shows the effect of the air-flow rate. Because the flow in the tube is turbulent and the dominant thermal resistance occurs on the air-side in the case of $\dot{m}_w = 8.5(10)^{-2}$ kg/s (use of the extra pump), ΔT_i increases much more than that of the case of $\dot{m}_w = 1(10)^{-2})$ kg/s. In Fig. 10,

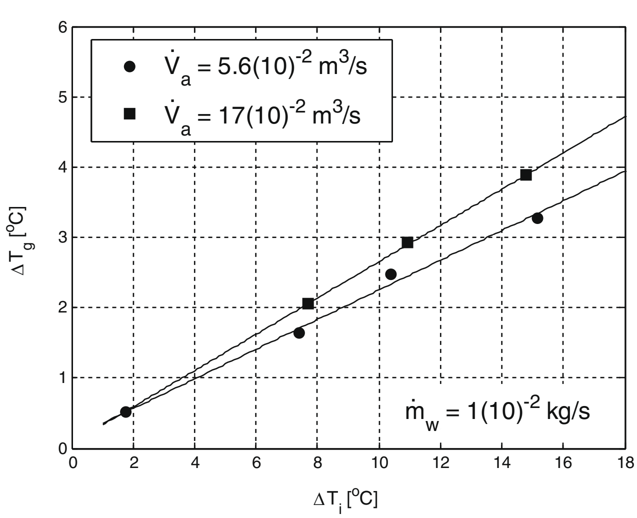

Fig. 4. Effect of air-flow rate (lines for readability).

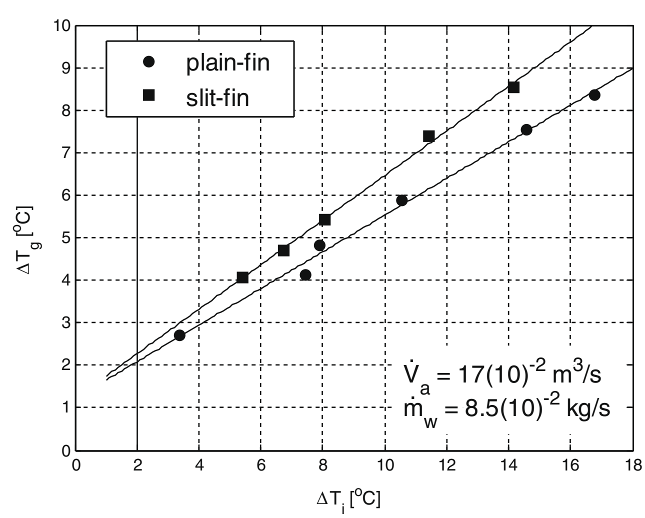

Fig. 6. Effect of heat exchanger (lines for readability).

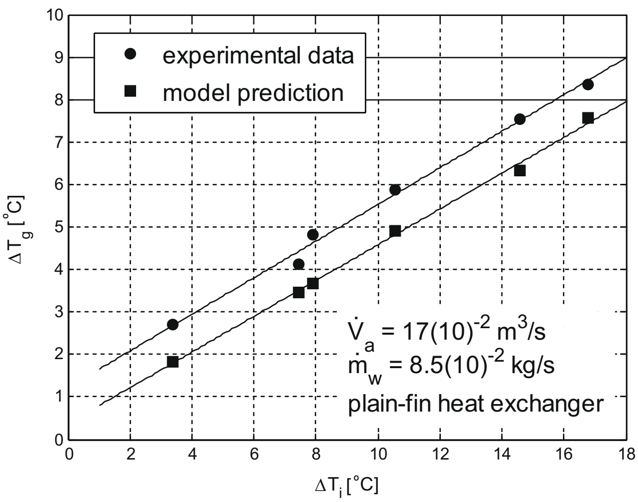

Fig. 7. Comparison of the simulation to experiments (plain-fin: lines shown for readability).

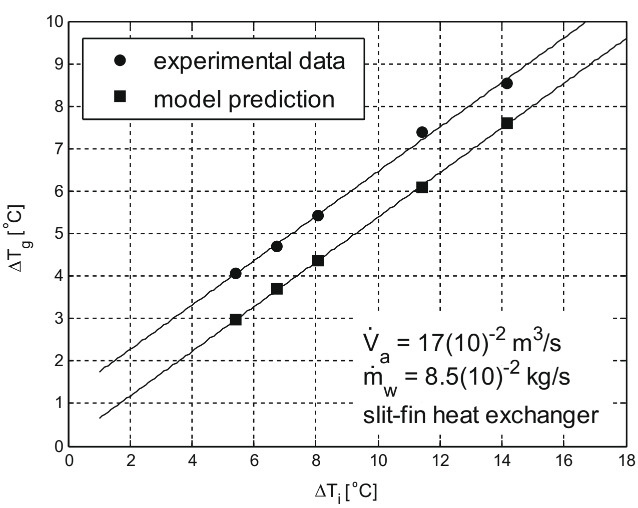

Fig. 8. Comparison of the simulation to experiments (slit-fin: lines shown for readability).

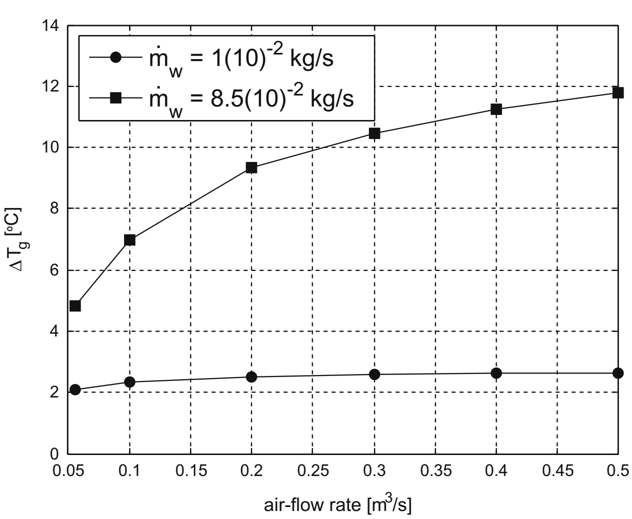

Fig. 9. Simulated results with various air-flow rates (curves shown for readability).

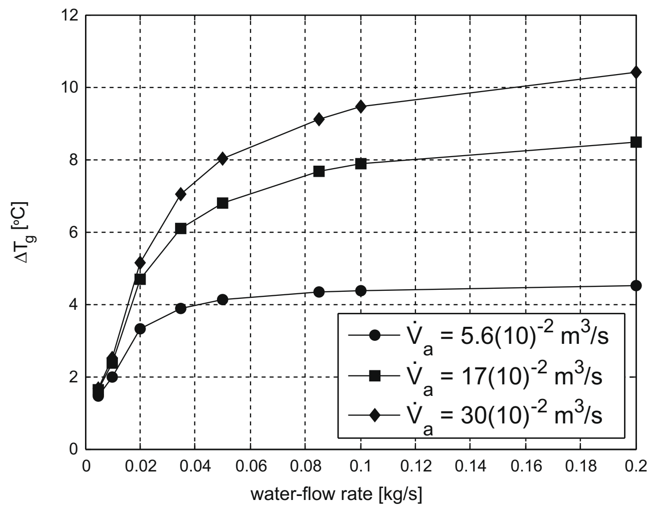

Fig. 10. Simulated results with various water-flow rates (curves shown for readability).

it can be seen that ΔT_g becomes constant as the mass flow rate of water increases.

5. Performance comparison and discussion

In order to quantify the energy savings provided by the run-around heat exchanger system, the following parameter is considered:

$$\eta_{sys} = \frac{E_{old} - E_{new}}{E_{old}} = \frac{-(E_{fan} + E_{pmp})}{M_{tub} c_{p,w}(T_f - T_{i,old})} + \frac{T_{i,new} - T_{i,old}}{T_f - T_{i,old}}, \tag{14}$$

where

$$E_{old} = M_{tub} c_{p,w}(T_f - T_{i,old}), \tag{15}$$

$$E_{new} = M_{tub} c_{p,w}(T_f - T_{i,new}) + E_{fan} + E_{pmp}. \tag{16}$$

In Eq. (14), T_f is the designated maximum water temperature of the cycle (95 °C), and $T_{i,new}$ is the new initial temperature of heater-on period. In this analysis, $T_{i,old}$ can be calculated using the baseline experiment results (Fig. 3). In this way, the energy savings that is calculated accounts for the real heat leak in the tub. Based on previous results, four systems are compared:

(a) Plain-fin heat exchanger with $\dot{V}_a = 17(10)^{-2}$ m^3/s and $\dot{m}_w = 1(10)^{-2}$ kg/s.
(b) Plain-fin heat exchanger with $\dot{V}_a = 17(10)^{-2}$ m^3/s and $\dot{m}_w = 8.5(10)^{-2}$ kg/s.
(c) Slit-fin heat exchanger with $\dot{V}_a = 17(10)^{-2}$ m^3/s and $\dot{m}_w = 1(10)^{-2}$ kg/s.
(d) Slit-fin heat exchanger with $\dot{V}_a = 17(10)^{-2}$ m^3/s and $\dot{m}_w = 8.5(10)^{-2}$ kg/s.

To compare system performance, ΔT_i was set near 15 °C for all of these experiments. The results are presented in Table 1. For comparison, a "$\eta_{sys,max}$" is calculated by assuming the water in the tub is raised to the average air temperature inside the machine. Energy consumptions of the fan and the added pump were 34 kJ and 49 kJ, respectively. It turns out that system (d) is the best design for this study. Although the energy efficiency, η_{sys}, is 5.8% of the total baseline usage in system (d), it has 46% potential compared to the maximum case (12.8%). A propagation of error analysis shows the uncertainty in η_{sys} to be 0.19%.

Table 1
Comparison of system performance in four systems.

System	ΔT_i (°C)	$T_{i,old}$ (°C)	$T_{i,new}$ (°C)	$T_{i,max}$ (°C)	η_{sys} (%)	$\eta_{sys,max}$ (%)
a	13.2	9.3	10.9	20.2	1.35 ± 0.20	12.21
b	16.8	7.0	12.8	21.2	5.39 ± 0.18	14.93
c	15.4	7.8	9.5	20.8	1.45 ± 0.18	14.34
d	14.6	8.4	14.6	20.6	5.85 ± 0.19	12.85

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, a significant savings might be possible by circulating water through the run-around loop for the entire heater-off time. From the simulation model, such operation under the system (d) scenario would result in $T_{i,new}$ = 16.3 °C (as compared to 14.6 °C in the current system (d)).

6. 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 and to explore design changes. In this study, the slit-fin and round-tube heat exchanger with $\dot{V}_a = 17(10)^{-2}$ m^3/s and $\dot{m}_w = 8.5(10)^{-2}$ kg/s has the best energy efficiency of 5.8%. For achieving more energy efficiency, not only are using a single, more powerful pump and elimination of the pump off time recommended, but also a more powerful and efficient fan should be considered. 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).

Appendix A. Heat exchanger specifications

The heat exchanger used in this study had plate fins and round tubes. In order to obtain specific parameters of heat exchanger, Eqs. (A.1)–(A.5) were used. The plain-fin and round-tube heat exchanger specifications are noted in Figs. A.1 and A.2, and Table A.1. The specifications for slit-fin heat exchanger are summarized in Table A.2, and specific geometry of slit-fin is shown in Fig. A.3.

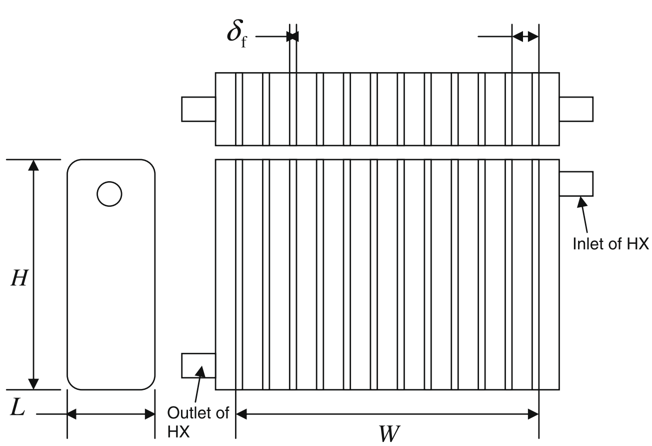

Fig. A.1. Geometry of plain-fin and round-tube heat exchanger.

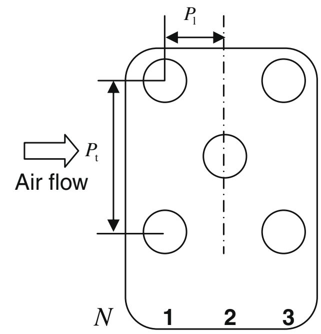

Fig. A.2. Geometric parameters related to tubes.

Table A.1
Plain-fin heat exchanger specifications.

A_c	$3.313(10^{-2})$	{m^2, minimum flow area}
A_f	2.864	{m^2, total fin area}
$A_o = A_{tube} + A_f$	3.062	{m^2, total surface area}
A_{tube}	$1.982(10^{-1})$	{m^2, total tube area}
D_c	$7.620(10^{-3})$	{m, fin collar outside diameter}
L	$3.815(10^{-2})$	{m, flow depth of the heat exchange}
D_{in}	$6.604(10^{-3})$	{m, tube inner diameter = 0.26 inches}
D_{out}	$7.366(10^{-3})$	{m, tube outer diameter = 0.29 inches}
F_P	$1.632(10^{-3})$	{m, fin pitch}
H	$1.988(10^{-1})$	{m, height of heat exchanger}
N	3	{number of tube rows}
N_f	190	{number of fins}
N_{tube}	29	{number of tubes}
P_t	$2.078(10^{-2})$	{m, transverse tube pitch}
P_l	$1.244(10^{-2})$	{m, longitudinal tube pitch}
W	$3.096(10^{-1})$	{m, width of heat exchanger}
δ_f	$1.270(10^{-4})$	{m, fin thickness}

$$A_{fr} = WH, \tag{A.1}$$

$$A_f = (HL - N_{tube}\pi D_c^2/4)N_f, \tag{A.2}$$

$$A_{tube} = (\pi D_c W - N_f \pi D_c \delta_f)N_{tube}, \tag{A.3}$$

$$\sigma = \frac{(P_t - D_c)(F_p - \delta_f)}{P_t F_p}, \tag{A.4}$$

$$A_c = A_{fr}\sigma. \tag{A.5}$$

Table A.2
Slit-fin heat exchanger specifications.

A_c	$3.288(10^{-2})$	{m^2, minimum flow area}
A_f	3.120	{m^2, total fin area}
$A_o = A_{tube}+A_f$	3.317	{m^2, total surface area}
A_{tube}	$1.967(10^{-1})$	{m^2, total tube area}
D_c	$7.620(10^{-3})$	{m, fin collar outside diameter}
L	$3.815(10^{-2})$	{m, flow depth of the heat exchange}
D_{in}	$6.604(10^{-3})$	{m, tube inner diameter = 0.26 inches}
D_{out}	$7.366(10^{-3})$	{m, tube outer diameter = 0.29 inches}
F_P	$1.632(10^{-3})$	{m, fin pitch}
F_s	$1.375(10^{-3})$	{m, fin spacing}
H	$1.988(10^{-1})$	{m, height of heat exchanger}
h_s	$7.600(10^{-4})$	{m, height of slit}
N	3	{number of tube rows}
N_f	207	{number of fins}
N_{tube}	29	{number of tubes}
P_t	$2.078(10^{-2})$	{m, transverse tube pitch}
P_l	$1.244(10^{-2})$	{m, longitudinal tube pitch}
W	$3.096(10^{-1})$	{m, width of heat exchanger}
δ_f	$1.270(10^{-4})$	{m, fin thickness}

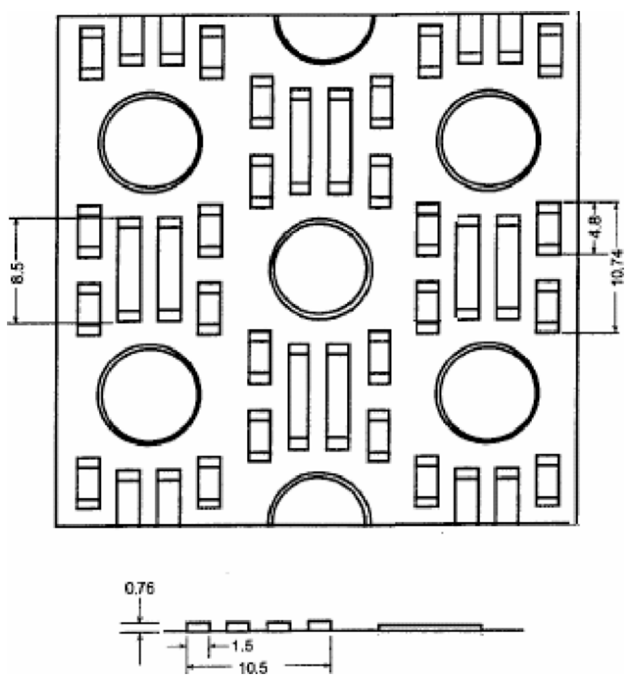

Fig. A.3. Geometry of slit-fin and round-tube heat exchanger.

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