Energy-efficient *Legionella* control that mimics nature and an open-source computational model to aid system design

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Title
Energy-efficient Legionella control that mimics nature and an open-source computational model to aid system design.

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Highlights
\begin{itemize}
\item A design for efficient Legionella control and general pasteurization is introduced.
\item A freely-available design model developed in Microsoft Excel is described.
\item Model results are compared to experimental measurements.
\item Applications to Legionella control and solar disinfection systems are discussed.
\end{itemize}

Keywords
Thermal Model, Heat Exchanger, Pasteurization, Legionnaire’s Disease, Microsoft Excel

Abstract
Although there is no direct connection, the incidence of Legionnaire’s disease has increased concurrently with increased usage of energy efficient domestic hot water (DHW) systems, which serve as ideal growth environments for Legionella pneumophila, the bacteria responsible for Legionnaire’s disease. The Duck Foot Heat Exchange Model (DFHXM) was developed to aid design of energy efficient thermal pasteurization systems with Legionella control specifically in mind. The model simulates a system design imitating the countercurrent heat exchange in the feet of ducks, an evolutionary adaption reducing environmental heat losses in cold climates. Such systems use a heat exchanger to preheat fluids prior to pasteurization and cool the same
fluid after pasteurization. Thus, the design requires minimal addition of heat to achieve pasteurization temperatures and to cover environmental heat losses. This article describes the underlying principles and use of the freely available Microsoft Excel model, as well as compares results from the DFHXM to measurements of an experimental pilot system. Simulation outputs agreed well with experimental results for transient and steady-state temperatures, the largest discrepancy in steady-state temperatures being 4.6%. Lastly, we discuss the flexibility of the DFHXM to simulate a wide variety of designs with special emphasis on Legionella control and solar-thermal water disinfection.

1. Introduction

Use of energy efficient domestic hot water (DHW) systems has increased in recent years as energy costs and energy efficiency standards have increased. Water heating accounted for 18.7% of household energy usage for houses built in the United States between 1980 and 1989 and reduced to 17.8% for houses built between 2000 and 2009[1]. The United States’ energy efficient water heater market increased from 625,000 ENERGY STAR approved units sold in 2006 to 1 million sold in 2009[2], and the U.S. federal minimum standards for energy factors of hot water storages tanks just increased in 2015. In 2008, the European Union set the goal of reducing the energy requirements of new buildings by 20% by 2020, and the addition of solar DHW systems has been shown as one way to reach this goal[3], and global use of solar hot water systems has been increasing. The annual global volume of solar water collectors increased from 1.8 kW_{th} per 1000 inhabitants in 2000 to 12.0 kW_{th} per 1000 inhabitants in 2012[4], and the annual capacity of solar thermal collectors in the EU28 countries and Switzerland nearly doubled from 2004 to 2013[5].

Cases of Legionnaire’s disease, an acute form of bacterial pneumonia often caused by exposure to DHW systems with Legionella pneumophila infestations, have also been increasing. In the United States, the cases of Legionnaire’s disease per 100,000 persons nearly tripled from 0.39 in 2000 to 1.15 in 2009[6]. During that same time period in Europe, cases of Legionnaire’s disease per 100,000 persons increased from 0.538 in 2000 to 1.10 in 2009[7],[8]. Increases in the occurrence of Legionnaire’s disease cannot be directly connected to trends of energy efficient and renewably powered DHW use. However, Legionella p. grows well in water maintained between
25-45 °C\textsuperscript{[9]} and can survive temperatures of 66°C for several minutes\textsuperscript{[10]}, and these temperatures are typical in energy-efficient and solar DHW systems. As an example, energy efficient DHW systems heated by air-source or ground-source heat pumps typically have operating temperatures of 50-55 °C\textsuperscript{[11]}, which still permits \textit{Legionella} growth. Although water temperatures within solar collectors can reach daily maximums of around 90 °C, storage tanks are typically maintained under 50 °C to minimize standby heat loss\textsuperscript{[12],[13]}.

Many thermal, chemical, and irradiative methods for preventing and treating \textit{Legionella} colonization in hot water distribution systems exist\textsuperscript{[14]}, the most common including hyperchlorination, super-heat-and-flush, and maintain storage tanks above 55°C. All these methods have the downside of high cost; both thermal methods are energy-inefficient while hyperchlorination is potentially hazardous to the health of users\textsuperscript{[15]}.

One potentially energy and cost efficient method is to thermally disinfect water immediately prior to exiting a hot water outlet. The general system design, shown in Figure 1, mimics countercurrent heat exchange present in the circulatory systems of many birds. Ducks, for example, have a structure of closely connected arteries and veins called \textit{rete} in the legs and feet which act as a biological heat exchanger; warm arterial blood coming from the body donates heat to cold venal blood returning from the feet. This action reduces the energy input necessary to bring blood returning from the feet back to body temperature by a factor as large as 84\textsuperscript{[16]}. Thus, we refer to any design which transfers heat between fluid in a single continuous flow to aid a heating or cooling process a “duck foot” (DF) heat exchange system (see Figure 1). Such systems are ideal for the point-of-use eradication of \textit{Legionella} or other pasteurization applications such as flow through solar water pasteurization.
Figure 1. General design of a “duck-foot” heat exchange system. Fluid entering at temperature $T_1$ changes by $\Delta T$ in the heat exchanger to $T_2$. After passing through some kind of reservoir, fluid re-enters the heat exchanger at a temperature very near $T_2$ (due to heat gains/losses to the environment). Fluid exits the heat exchanger at an output temperature near to the input temperature $T_1$.

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DF heat exchange systems are common design features in dairy pasteurization plants and flow through solar water pasteurization systems\textsuperscript{[17]}. Other applications include processes during which fluids must increase in temperature before cooling. For example, the solar dairy pasteurization system presented by Quijera et al. (2011) and the solar honey pasteurizer presented by Evans and de Schiller (1997) would have improved their efficiencies by including DF heat exchange systems\textsuperscript{[18],[19]}.

Many models exist for the design optimization of plate heat exchangers. McKillop and Dunkley (1960) first presented a thermal model for calculating fluid temperatures within the channels of plate heat exchangers using the Gill-Runge-Kutta method for solving the system of first order ordinary differential equations\textsuperscript{20}. Gut and Pinto (2003) developed an assemblage algorithm for modeling generalized plate heat exchanger configurations which considers the temperature dependence of fluid properties within heat exchange channels and solved the system of first order
ordinary differential equations using the computational software gPROMS\textsuperscript{[21]}. Strelow (2000) presents a non-iterative method for modeling heat exchangers with multiple process streams and unusual geometries\textsuperscript{[22]}. To the authors’ knowledge, however, no models exist specifically describing DF heat exchange systems without manipulation. Additionally, existing plate heat exchanger models require knowledge of solving systems of first order differential equations and often additional, expensive computer software to run; thus, they are inaccessible to many potential users. The aim of this project was to develop a simple, easy-to-use model specific to DF heat exchange systems using widely available computer software. The model is intended to assist designers thinking of implementing DF heat exchange systems for *Legionella* spp. control or for a variety of pasteurization applications including solar water pasteurization.

2. Duck Foot Heat Exchange Model (DFHXM) Description and Use

2.1 Model Description

A numerical Duck Foot Heat Exchange Model (DFHXM) was created in Microsoft Excel to simulate steady-state and transient fluid temperatures within a DF system. The DFHXM assumes a flat plate, single pass heat exchanger with parallel channel flow and uses simple, one-dimensional heat transfer in iterative calculations. It differs from more complicated models of plate heat exchangers by approximating multiple channels flowing in parallel as a single channel. For a single-pass plate heat exchanger with $N$ plates and $N+1$ channels, the DFHXM uses symmetry to approximate the system as two countercurrent fluid channels, each with half the total channel volume, and scales the result by $N$. The idealizations outlined by McKillop and Dunkley (1960) were used while making the model: (1) fluid properties are independent of temperature; thus, the overall heat transfer coefficient is constant within the heat exchanger, (2) no heat is lost by the fluid or heat exchanger material to the external environment, (3) no heat is conducted in the direction of fluid flow, and (4) temperatures and flow rates within control volumes in fluid channels are constant\textsuperscript{[20]}. An additional assumption for DF systems is that mass flow rates are equal for both flow directions within the heat exchanger.
The model breaks the two-channel approximation into \( n \) partitions along the direction of flow to create \( 2n \) fluid control volumes (Figure 2); each fluid control volume represents the sum of channel volumes at a given partition level on each side of the heat exchanger.

![Figure 2](image)

**Figure 2.** Numbering system of partitions and control volumes in the DFHXM spreadsheets. White regions of the heat exchanger refer to fluid control volumes (sums of channels), while grey regions refer to heat exchange plates. Arrows mark the direction of fluid flow.

Heat is transferred between adjacent fluid control volumes according to

\[
Q = U A_p (T_h - T_c) \Delta t,
\]

where \((T_h - T_c)\) is the temperature difference between adjacent fluid control volumes which determines the direction of heat flow, \( U \) is the overall heat transfer coefficient of the heat exchanger, and \( A_p \) is the area of heat exchange plate between adjacent fluid control volumes.

The time step \( \Delta t \) is dependent on mass flow rate \( \dot{m} \), the volume \( V_{cv} \) of control volumes, and the fluid density (assumed constant) \( \rho \) as

\[
\Delta t = \frac{\rho V_{cv}}{\dot{m}}.
\]

The time step can be adjusted for a given fluid by changing the mass flow rate or the number of partitions \( n \), which is related to control volume size by

\[
V_{cv} = \frac{w H N (L_s - 1)}{2n} \text{ (see Table 1).}
\]

Beginning each time step temperature changes within adjacent control volumes are calculated by accounting for heat transfer between fluids as

\[
T_{int} = T_{old} + \frac{Q}{\rho V_{cv}},
\]

where \( T_{old} \) represents the fluid temperature at the beginning of the time step, and \( T_{int} \) is an intermediate fluid temperature within each control volume. The sign of the equation depends on relative temperatures of the adjacent control volumes – the control volume with lower starting temperature gains energy, while the control volume with higher starting temperature loses energy. Next, heat transfer occurs between control volumes and heat exchange plates within the partition. Because conductive energy transfer occurs quickly within the metal relative to the length of time steps, the model assumes (1) the metal to have a linear temperature gradient.
and (2) an average temperature equal to the average temperature of the surrounding control volumes at the end of the time step. The model checks for a temperature difference between the metal plate at the beginning of the time step and the average of intermediate control volume temperatures. If the metal temperature \( T_m \) is lower than the average intermediate temperatures of the control volumes, metal temperature increases by \( \Delta T_m \) while the control volume with higher temperature decreases by \( \Delta T_f \) according to the system of equations:

\[
T_m + \Delta T_m = \frac{1}{2} (T_{c,int} + T_{w,int} - \Delta T_f)
\]

\[
|c_m \Delta T_m| = |c_f \Delta T_f|
\]

where \( T_{c,int} \) and \( T_{w,int} \) are the intermediate temperatures in adjacent control volumes, and \( c_m \) and \( c_f \) are the heat capacities of the metal and of fluid within a given control volume, respectively. If intermediate control volume temperatures are equal and the metal plate temperature is lower, the control volumes lose energy equally to the metal, both decreasing in temperature by \( \Delta T_f \) according to the equations:

\[
T_m + \Delta T_m = \frac{1}{2} (T_{c,int} + T_{w,int} - 2\Delta T_f)
\]

\[
|c_m \Delta T_m| = 2|c_f \Delta T_f|
\]

If the average of intermediate fluid temperature is lower than the metal temperature, the metal loses energy equally between the surrounding control volumes according to the equations:

\[
T_m + \Delta T_m = \frac{1}{2} (T_{c,int} + T_{w,int} + 2\Delta T_f)
\]

\[
|c_m \Delta T_m| = 2|c_f \Delta T_f|
\]

Depending on the scenario of energy transfer between control volumes and metal, the control volumes and metal may increase in temperature, decrease, or remain unchanged as,

\[
T_{c,new} = T_{c,int} \pm \Delta T_f
\]

\[
T_{h,new} = T_{h,int} \pm \Delta T_f
\]

\[
T_{m,new} = T_m \pm \Delta T_m
\]

where the subscript \( new \) signifies the control volume and metal temperatures at the end of each time step. For the \( i \)th control volume, the fluid temperature concluding a given time step is assigned to the \((i+1)\)th control volume for the following time step, while metal temperatures remain within a given partition stay in place for the next time step. The DFHXM simulates a heating reservoir placed between the heater inlet and heater outlet of a DF system. Fluid is assumed to have uniform
temperature within the heater and enters and exits at the same mass flow rate \( \dot{m} \). Thermal mixing occurs in the heating reservoir during each time step as water enters from the heater inlet; heat energy determined by the heater power \( P \) is transferred to the volume as \( Q = P \Delta t \), causing a temperature change of \( T_{Res,new} = T_{Res,old} + \frac{P \Delta t}{\rho c V_{Res}} \). The model simulates a hysteresis of the heating reservoir such that the heater turns off when temperatures are higher than a set maximum and does not reengage until temperatures fall below the hysteresis range set by the user.

### 2.2 Using the Duck Foot Heat Exchange Model

The DFHXM can be run in versions of Microsoft Excel 2007 and newer and is available without cost\(^{[23]}\). Users must input the parameters listed in
Table 1 to match their desired design. Heat exchanger manufacturers readily provide design parameters for plate heat exchangers made for specific temperature programs and flow rates. These typically include the heat exchange area, the overall heat transfer coefficient (OHTC), plate dimensions, plate materials, and the number of plates. The OHTC is specific to flow rate and plate configuration, but users can adjust heat load by changing the number of plates, thereby adjusting heat exchange area within the model while leaving the OHTC and flow rate constant. The model uses “effective” plate width and height because it assumes flat rectangular plates; however, the plate regions in contact with the fluid are most often partially rectangular with irregular geometry near the corner holes of each plate. Plates are also commonly corrugated to enhance heat transfer. Effective plate height $H$ can be determined from effective plate width $w$, which is generally easier to obtain from schematic drawings, and area of individual plates defined as total heat exchange area $A$ divided by the number of plates $N$. Manufacturers typically report the thickness of the plate pack plus end covers in the form $a + L_s N$, where $a$ is the thickness of the end covers and $L_s$ is the thickness of the plate stack per plate. Some manufacturers do not report plate thickness $l$ in schematic drawings; in such cases users must directly contact the manufacturer. As an example, SWEP International reported their heat exchangers have plates between 0.25-0.45 mm thick$^{[24]}$. Increasing the number of partitions $n$ improves model performance by decreasing the size of control volumes and the length of time steps, but calculation time also increases.
Table 1. User input parameters of the DFHXM.

*Parameter not necessary for every mode of the model.

| Parameter Name                              | Symbol | Unit               |
|---------------------------------------------|--------|--------------------|
| **Heat Exchanger Parameters**              |        |                    |
| Overall Heat Transfer Coefficient           | $U$    | W °C$^{-1}$ m$^2$  |
| Heat Exchange Area                          | $A$    | m$^2$              |
| Effective Plate Width                       | $w$    | m                  |
| Effective Plate Height                      | $H$    | m                  |
| Plate Stack Thickness per Plate             | $L_s$  | m                  |
| Plate Thickness                             | $l$    | m                  |
| Number of Plates                            | $N$    |                    |
| Specific Heat Capacity of Plate Material    | $c_m$  | J °C$^{-1}$ kg$^{-1}$ |
| Density of Plate Material                   | $\rho_m$ | kg m$^{-3}$     |
| **Fluid Parameters**                        |        |                    |
| Density of Fluid                            | $\rho$ | kg m$^{-3}$        |
| Specific Heat Capacity of Fluid             | $c$    | J °C$^{-1}$ kg$^{-1}$ |
| Fluid Flow Rate                             | $\dot{m}$ | kg s$^{-1}$     |
| Temperature of Inlet Fluid                  | $T_i$  | °C                 |
| Ambient Temperature                         | $T_a$  | °C                 |
| Constant Heater Outlet                      | $T_h$  | °C                 |
| **Heating Reservoir Parameters**            |        |                    |
| Power of Heating Reservoir $^*$             | $P$    | W                  |
| Volume of Heating Reservoir $^*$            | $V_{Res}$ | m$^3$         |
| Initial Temperature of Heating Reservoir $^*$| $T_{Res,Ini}$ | °C       |
| Maximum Temperature of Heating Reservoir $^*$| $T_{Res,Max}$ | °C       |
| Hysteresis of Heating Reservoir $^*$        | $T_{Hys}$ | °C       |
| Initial Heater Power $^*$                   | $P_{Ini}$ | W              |
| Ramp Rate of Heater Power $^*$              | $P_{Ramp}$ | W s$^{-1}$     |
| **Model Parameters**                        |        |                    |
| Number of Partitions                        | $n$    |                    |
| Time Length of Simulation                   | $t_{Run}$ | s              |

Four functional spreadsheets comprise the DFHXM – one for inputting parameters and three for recording the calculated results of fluid temperatures within the heat exchanger, heat exchange plate temperatures, and fluid temperature within the heating reservoir through time. Control volumes and heat exchange plate partitions are numbered to correspond with regions in the heat
exchanger as shown in Figure 2. Control volumes 1 through \( n \) refer to the inlet side of the heat exchanger while \( n+1 \) through \( 2n \) refer to the outlet side. Temperatures at the inlet and heater inlet are represented by control volumes 1 and \( n \) respectively. Similarly, heater outlet and outlet temperatures are represented by \( n+1 \) and \( 2n \) respectively. The \( n \) partitions of heat exchange plates are represented by numbers identical to adjacent inlet side control volumes as shown in Figure 2. Depending on the desired results, users may choose from four simulation modes of the DFHXM summarized in Table 2.

**Table 2.** Simulation modes of the DFHXM. All modes use a constant temperature for fluid entering the inlet of the heat exchanger.

| Simulation modes                      | Explanation                                                                                                                                 |
|---------------------------------------|---------------------------------------------------------------------------------------------------------------------------------------------|
| Constant Heater Outlet Temperature    | Fluid passes the heater outlet at a constant temperature \( T_h \), which is a necessary input parameter. All parameters relating to the heating reservoir may be omitted. |
| Heating Reservoir without Thermostat  | Fluid passes the heater outlet from a heating reservoir with volume \( V_{res} \), constant power \( P \), and initial temperature \( T_{res,ini} \). All other heating reservoir parameters and \( T_h \) may be omitted. |
| Heating Reservoir with Thermostat     | Identical to Heating Reservoir without Thermostat with the addition of a thermostat maintaining reservoir temperature between \( T_{res,max} \) and \( T_{res,max} - T_{hys} \) after initial warm-up or cool-down of the heating reservoir. The user may omit \( P_{ini} \), \( P_{ramp} \), and \( T_h \). |
| Heating Reservoir Ramped Power        | Identical to Heating Reservoir without Thermostat, but the heater begins with an initial power \( P_{ini} \) and ramps linearly at a rate \( P_{ramp} \) to a final power \( P \). No maximum temperature is set for the reservoir, and the user may omit \( T_{res,max} \), \( T_{hys} \), and \( T_h \). |

The **Constant Heater Outlet Temperature** mode allows users to investigate the effects of heat exchange area and heat load on the transient and steady-state characteristics of a DF system by simulating a “limitless” heating reservoir. The **Heating Reservoir without Thermostat** mode allows users to investigate the effects of heating reservoir volume, power, and temperature on the transient and steady-state characteristics of a DF system, as well as differences in energy usage.
No upper limit is placed on reservoir temperature to ensure that equilibrium temperatures reflect the heat transfer capacity of the heat exchanger. The *Heating Reservoir with Thermostat* mode is the most realistic simulation of DF heat exchange system and includes a reservoir maximum temperature and hysteresis. The final mode, *Heating Reservoir Ramped Power*, was created to mimic DF systems with non-constant power sources such as wood or solar heating.
3. Simulation Results and Discussion

3.1 Model Validation

An experimental DF heat exchange system was built by Infjärdens Värme AB in Piteå, Sweden, in 2009 and is shown in Figure 3. Measurements were conducted at the Luleå University of Technology (LTU) as part of a Senior Design Project in Renewable Energy in 2013[25]. The heating reservoir was a type VB OEM 3000 manufactured by Värmebaronen in Kristianstad, Sweden, had a volume of 2.8 L and was rated to deliver 3 kW power. The tank was insulated in an 18 mm type K-060 cylindrical jacket made by the same manufacturer. A SWEP International heat exchanger with heat exchange area 0.64 m² was used, but exact plate dimensions were unknown. Effective plate width and height were inferred from photographs to be 0.2 m and 0.4 m respectively, though accuracy with these parameters is not necessary for proper model functioning. Heat exchange plates were assumed to be stainless steel and have thickness of 0.3 mm. Piping between the heat exchanger and reservoir was 22 mm diameter copper with a total length of 2 m. Armaflex insulation was used around the heat exchanger (20.5 mm) and the copper pipes (12 mm). Four type-K thermocouples monitored temperatures to the nearest 1 °C at inlets and outlets to the heat exchanger as shown in Figure 3. A tap water faucet in the laboratory was connected to the inlet of the experimental system. Flow was adjusted to an approximately constant rate by adjusting the tap and was measured to the nearest 0.1 kg min⁻¹ with a balance and stopwatch at the beginning, middle, and end of each experimental trial.
Figure 3. Experimental DF heat exchange system tested at LTU in 2013. Labels mark the locations of thermocouple temperature measurements and corresponding locations in the DFHXMS. The dashed line and arrows mark the locations and direction of water flow.

Water at a constant flow rate and temperature was run through the system prior to taking measurements, and temperatures were recorded each minute after power was supplied to the heating reservoir. Steady-state temperatures for each flow rate are listed in Table 3; however, it should be noted that flow rates varied up to 6% from reported averages during experimental trials. The DFHXMS heater power was set to 2700 W to account for an estimated 10% energy loss in the system, which is reasonable considering calculated power delivery from the experimental heater averaged 8.7% lower than the rated level of 3000 W. Remaining thermal losses were assumed to occur through pipes and connections. The overall heat transfer
coefficients used for simulations were calculated from experimental steady-state temperatures for each flow rate; parameters for simulation runs are also listed in Table 3. The number of partitions \( n \) was adjusted for different simulations to ensure nearly equal time steps of 0.11 s.

Table 3. Experimental steady-state temperatures and DFHXM simulation parameters for the four investigated flow rates. Measurement locations of \( T_1, T_2, T_3, \) and \( T_4 \) are shown in Figure 3.

| Flow Rates | 3.4 | 3.8 | 5.0 | 7.5 | kg min\(^{-1}\) |
|------------|-----|-----|-----|-----|----------------|
| **Experimental Steady State Temperatures** | | | | | |
| \( T_1 \)  | 24  | 41  | 39  | 38  | °C            |
| \( T_2 \)  | 56  | 71  | 66  | 53  | °C            |
| \( T_3 \)  | 68  | 81  | 73  | 58  | °C            |
| \( T_4 \)  | 35  | 49  | 47  | 43  | °C            |
| **DFHXM Simulation Parameters** | | | | | |
| \( U \)     | 1039| 1415| 1910| 2498| W °C\(^{-1}\) m\(^2\) |
| \( T_i \)   | 24  | 41  | 39  | 39  | °C            |
| \( T_a \)   | 23  | 40  | 39  | 39  | °C            |
| \( T_{Res,Ini} \) | 23  | 40  | 39  | 39  | °C            |
| \( n \)     | 100 | 90  | 68  | 44  | -             |
| \( \dot{m} \) | 0.0567| 0.0633| 0.0833| 0.128| kg s\(^{-1}\) |
| **Unchanging Simulation Parameters** | | | | | |
| **Heat Exchanger Parameters** | | | | | |
| \( A \)     | 0.64 m\(^2\) | | | | |
| \( w \)     | 0.2 m    | | | | |
| \( H \)     | 0.4 m    | | | | |
| \( L_s \)   | 0.00229 m| | | | |
| \( l \)     | 0.0003 m | | | | |
| \( N \)     | 8        | | | | |
| \( c_m \)   | 500 J °C\(^{-1}\) kg\(^{-1}\) | | | | |
| \( \rho_m \) | 8000 kg m\(^{-3}\) | | | | |
| **Fluid Parameters** | | | | | |
| \( \rho \)   | 988 kg m\(^{-3}\) | | | | |
| \( c \)     | 4178 J °C\(^{-1}\) kg\(^{-1}\) | | | | |
| **Heating Reservoir Parameters** | | | | | |
| \( V_{Res} \) | 0.0028 m\(^3\) | | | | |
| \( P \)     | 2700 W    | | | | |

The **Heating Reservoir with Thermostat** mode of the DFHXM was run for each experimental flow rate; although, steady-state temperatures in the reservoir were always below thermostat temperature, and the simulated thermostat never engaged. The DFHXM simulations agreed well with experimental results as shown in Figure 4 for both transient and steady-state temperatures.
Excluding the results for inlet temperature $T_1$ which was an input parameter ($T_i$) for the DFHXM, the most inaccurate simulation was of the 3.8 kg min$^{-1}$ trial. The average deviance from experimental steady-state temperatures for $T_2$, $T_3$, and $T_4$ at this flow rate was 4.6%, while the average deviance for simulations of the other flow rates ranged from 0.5 to 1.0%. Both the model and experiment show how increasing flow rate with a fixed heater power and heat exchange capacity decreases both temperature spread and the time necessary to reach equilibrium. The time for temperatures to reach 95% of steady-state in simulations reduced from 10.6 min for 3.4 kg min$^{-1}$ to 5.0 min for 7.5 kg min$^{-1}$.

![Figure 4](image)

**Figure 4.** Comparison between heat exchanger inlet and outlet temperatures versus time for experimental and simulation results. The *Heating Reservoir with Thermostat* mode of the DFHXM was used for every simulation, and parameters are listed in Table 3. Experimental steady-state temperatures and DFHXM simulation parameters for the four investigated flow rates. Measurement locations of $T_1$, $T_2$, $T_3$, and $T_4$ are shown in Figure 3. Temperature numbers ($T_x$) correspond to labels in Figure 3. The investigated flow rates were (a) 3.4 kg min$^{-1}$; (b) 3.8 kg min$^{-1}$; (c) 5.0 kg min$^{-1}$; and (d) 7.5 kg min$^{-1}$.

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A drawback to using the DFHXM is the necessity to estimate heat losses to the external environment. Had the heating reservoir’s rated power of 3000 W been used, the model would
have overestimated steady-state temperatures by 9.3 % at 3.8 kg min\(^{-1}\) and as much as 2.1 % with the other flow rates. However, accounting for thermal losses by decreasing the simulated heater power, as was done in this simulation, gives users the freedom of investigating different levels of system efficiency.

3.2 Example Simulations
The DFHXM calculates the temperature profile along the heat exchanger and within the reservoir volume through time. Sample results from a simulation run of the *Heating Reservoir without Thermostat* mode with the input parameters in Table 4 are shown graphically in Figure 5 as transient temperature profiles in the heat exchanger (Figure 5a) and heating reservoir (Figure 5b) as the system approaches steady-state. The chosen parameters are reasonable for a DF system installed for *Legionella* control with a single shower. After 10 seconds, water in the middle regions of the heat exchanger remained near the starting ambient temperature of 20 °C, while the reservoir was still near its starting temperature of 75 °C, being not much yet diluted by influx of cooler water. Water entering the heater inlet had already risen to nearly 50 °C due to heat received from water exiting the heating reservoir. After 30 seconds, ambient temperature water had been fully flushed from the system, but areas near the inlet and outlet still required more energy input to rise above the temperature of incoming water at 40 °C. The reservoir temperature further reduced due to thermal mixing of entering water. After 60 seconds, the system was nearly at equilibrium. The temperature profile along the heat exchanger was nearly linear, and temperature within the reservoir approached its horizontal asymptote.

Table 4. Input parameters for example DFHXM simulation.
Fluid Parameters
\( \rho \) 988 kg m\(^{-3}\)
\( c \) 4178 J °C\(^{-1}\) kg\(^{-1}\)
\( \dot{m} \) 0.3 kg s\(^{-1}\)
\( T_i \) 40 °C
\( T_a \) 20 °C

Heating Reservoir Parameters
\( P \) 5000 W
\( V_{Res} \) 0.025 m\(^3\)
\( T_{Res,Ini} \) 75 °C

Model Parameters
\( n \) 50

Figure 5. (a) Temperature profile within fluid channels of the heat exchanger at 10, 30, and 60 s in the DFHXM Heating Reservoir without Thermostat simulation. The far left represents the inlet, and the far right represents the outlet. The temperature spike in the center results from the energy added in the heating reservoir; the peak indicates temperature of fluid re-entering the heat exchanger from the reservoir (\( T_2 \)). (b) Heating reservoir temperature through time for the same simulation.

Because the intended purpose of DF heat exchange systems is for disinfection by pasteurization, the ability to monitor the temperature within the reservoir tank for different system arrangements is important for design. For example, imagine a system with the parameters listed in Table 4 equipped with a 10 kW heating reservoir and a thermostat set to a maximum temperature of 78 °C with a hysteresis of 6 °C. If the desired function is *Legionella* spp. eradication, the temperature of the reservoir should never drop below 70 °C to ensure no infected water exits the tap to create dangerous aerosols. Heating reservoir volume is an important design consideration to system function and energy consumption; with too small a tank the water temperature will sink...
below 70 °C due to thermal mixing of cooler water entering the tank, while standby energy loss increases with tank volume. The DFHXM was used to find an optimal tank volume to ensure safe function and minimize energy losses. Figure 6 shows the reservoir temperature versus time for different reservoir volumes.

Figure 6. Reservoir temperature versus time for 10, 20, and 30 L heating reservoirs in the DFHXM Heating Reservoir with Thermostat simulation. Input parameters were identical to those in Table 4 except $P$ was 10,000 W, maximum reservoir temperature was 78 °C, and hysteresis was 6 °C.

Temperatures in larger reservoir volumes were less affected by thermal mixing of entering water than smaller volumes. Larger reservoirs also had longer cycling times of the heater, but this did not affect the energy usage – the heater was powered 77% of the time regardless of reservoir volume. Larger reservoir volumes, however, use more energy due to increased surface area and standby heat losses. In this scenario, a 26 L heating reservoir was ideal, minimizing standby heat losses while always maintaining the reservoir temperature above 70 °C.

Another potential application for the DFHXM is solar pasteurization system design. Imagine that a 25 L reservoir with a flow rate of 1.0 L min$^{-1}$ receives linearly increasing power from 0 to 750 W per m$^2$ solar collector area over a span of 6 hours. Such a system could be used to disinfect drinking water, and designs could consist of solar collectors focusing energy upon the 25 L flow-
through reservoir. The *Heating Reservoir with Ramped Power* mode of the DFHXM was used to simulate changing solar power and monitor temperatures in the heating reservoir through time. The heat exchanger and fluid parameters were identical to those in Table 4 except that fluid flow rate $\dot{m}$ was 0.0167 kg s$^{-1}$ and both initial reservoir temperature $T_{\text{Res,Ini}}$ and inlet fluid temperature $T_i$ were 20 °C. Figure 7 shows the temperature of water exiting the reservoir over 6 hours for three different solar collector areas. The initial power $P_{\text{int}}$ supplied by the three solar heat collectors is 0 W to simulate night. Solar power is assumed to increase linearly over a 6 h period from 0 to 750 W m$^{-2}$, and the ramped powers $P_{\text{ramp}}$ depend upon solar collector areas. The respective $P_{\text{ramp}}$ for the 1.0 m$^2$, 2.5 m$^2$, and 5.0 m$^2$ simulated collectors are 0.0347, 0.0868, 0.174 W s$^{-1}$.

**Figure 7.** Water temperature exiting a flow-through solar pasteurization tank simulated by the *Heating Reservoir with Ramped Power* mode of the DFHXM. Solar collectors of 1.0, 2.5, and 5.0 m$^2$ directed heat energy from the sun, which changed linearly from 0 to 750 W m$^{-2}$ over a 6 h period, to a reservoir with volume 25 L and flow rate 1 L min$^{-1}$. The inlet fluid and reservoir start temperatures matched the ambient temperature of 20 °C. Unmentioned parameters were identical to those in Table 4.

With 1.0 m$^2$ solar collector area, water exiting the reservoir reaches 85.5 °C after 6 hours. Much is gained by increasing solar collector area to 2.5 m$^2$, as water in the reservoir reaches the boiling point after 3.7 h, while system with a 5 m$^2$ collector reaches the boiling point after 2.9 h. Thus, the DFHXM could help in choosing a solar collector to best balance the benefits of larger collectors with the drawbacks of increased cost and size. Although the DFHXM makes the simplifying assumptions of uniform thermal mixing within the heating reservoir, it is nevertheless useful as a rough indicator of the performance of this type of solar thermal pasteurizer. Users can
tailor their simulation parameters and utilize the DFHXM answer their particular design questions.

4. Conclusions
The DFHXM produces accurate models of a variety of DF heat exchange systems, but the user should note that it functions best in situations with small reservoirs and high flow rates because temperature in the heating reservoir is approximated as uniform throughout. The user should also note that both temperature and residence time affect the elimination of *Legionella*; special care should be taken to ensure that fluid reaches a killing temperature for a specified time in the DF system. Although the DFHXM makes many simplifying assumptions, it nevertheless accurately models the transient and steady-state behavior of DF heat exchange systems. It has potential to be a valuable but inexpensive resource to anyone with access to Microsoft Excel. Legionnaires’ disease will likely be an increasing problem as more and more energy efficient hot water systems are installed, and a DF heat exchange system may be an energy and cost effective treatment alternative to other control methods. Additionally, use of solar water pasteurization will likely increase as global populations rise and the need for safe water becomes larger. The DFHXM has the potential to assist many users with a wide variety of design problems. Current and future work includes examining the energy-usage and costs of DF heat exchange systems for pasteurization in *Legionella* spp. infected hot water systems, and initial results show that in certain design scenarios DF heat exchange systems are more energy and cost effective than current disinfection methods.

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