Theoretical and Experimental Analysis of Dynamic Heat Exchanger: Retrofit Configuration

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Abstract

This paper theoretically and experimentally describes a dynamic plate heat exchanger configuration that decreases or even eliminates heat exchanger losses in performance and efficiency associated with transient flow rates of hot and cold streams passing through its interior or other sources of imbalance. Heat exchanger constraints are some of the most restrictive transient response constraints in a process and thereby inhibit the process's agility and responsiveness. These constraints include temperature changes, expansion, or thermal stresses in the heat exchangers or neighboring process equipment. Despite any changes in inlet conditions, the proposed configuration is capable of leveling the varying parameters such that the exit temperatures remain fixed. Theoretical and experimental results show that the proposed configuration can respond to changes in process flow rates with a near-zero time constant.

Keywords: plate heat exchanger, transient flow, efficiency, response time, mathematical modeling, orthogonal collocation

1. Introduction

Plate heat exchangers (PHE) are becoming increasingly applicable because of their efficient and compact design. They serve as key components for various processes in industry, including the food processing, pharmaceutical, dairy, chemical and petrochemical industries. Aminian et al. [1] reported on a PHE that weighed 95% less than a comparable shell-and-tube exchanger and still provided 300-400 square feet of surface for heat exchange per cubic foot of volume. Many processes involve scheduled changes in load or other process conditions but experience unscheduled ones as well. The heat exchangers of a process are commonly the most sensitive components to such changes.

Heat exchangers react in a non-linear way with changes in inlet properties, making them difficult to control effectively. Therefore, sophisticated models are used to prominently predict the steady-state and transient performance of different heat exchanger designs [2, 3]. Current commercial software can design heat exchangers and accurately evaluate their performance. However, these models do not commonly provide optimization solutions [4, 5]. Heat exchanger design typically involves specifying the inlet temperatures and flow rates as well as the target hot and cold fluid outlet temperatures. The figures of merit include efficiency, heat exchanger size, and flow resistance. The optimization of plate heat exchangers involves manipulating a large number of design variables, some of which are inherently discrete, causing the task to be non-trivial [6-8].

Thermodynamic irreversibility in a heat exchanger comes from the transfer of heat across a finite temperature difference between the hot and cold streams. Based on this concept, the optimization of a heat exchanger is equivalent to minimizing the amount of useful power that is lost [9-12]. Mishra et al. [13] used a genetic-algorithm-based optimization technique on a cross-flow plate and fin heat exchanger to minimize the entropy generation while maintaining a constant heat duty. Cheng [14] introduced the concept of entropy resistance which uses entropy generation analysis as an alternative method for heat exchanger analysis. Fakheri [15] used the second law of thermodynamics to calculate the thermal

1 efficiency of a heat exchanger as the ratio of the actual heat transfer rate to the optimum heat transfer
2 rate. Fakheri [16] also extended this concept to determine the efficiency of heat exchanger networks
3 excluding the need for charts, or complicated performance statements.

4 There are several studies addressing the issue of liquid temperature control in heat exchangers. Luyben
5 [17] studied the issues of heat exchanger bypass control. Almutairi and Zribi [18] explored the
6 applications of three different terminal sliding mode (TSM) controllers to a plate heat exchanger.
7 Gonzalez et al. [19] presented a model predictive control (MPC) for a network of heat exchangers.
8 Bonivento et al. [20] compared the performance of a standard PID controller with a generalized predictive
9 control. Ramirez et al. [21] incorporated the min-max model predictive control (MMMPC) for an
10 industrial heat exchanger. Additional applications of MPC to heat exchangers can be found elsewhere
11 [22-26].

12 Artificial intelligent control schemes such as artificial neural network (ANN) and fuzzy logic systems
13 have been extensively applied for the thermal control of heat exchanger structures. Jian et al. [27]
14 presented an adaptive fuzzy sliding mode controller for the temperature control of heat exchangers.
15 Skrjanc and Matco [28] proposed a new predictive control structure based on a nonlinear fuzzy model.
16 Ramirez et al. [29] explored the application of neural network technique for the min-max predictive
17 control of a heat exchanger. Further studies in the area of heat exchanger control based on neural network
18 method appear elsewhere in the literature [30-33].

19 Changes in the opposite fluid flow rate or temperature control the fluid temperatures in all the above-
20 mentioned investigations. Flow rate changes may cause the flow regime to shift from laminar to turbulent,
21 significantly decreasing the heat transfer coefficient. To address this issue, a minimum bound should be
22 imposed on the input flowrate, and the controller should be retuned each time the flowrate changes, which
23 is a challenging task. Although manipulating the opposite fluid inlet temperature guarantees a turbulent
24 hydrodynamic regime inside the exchanger, it suffers from a time delay [34, 35].

1 The device described in this document, however, exhibits a heat exchanger technology that significantly
2 reduces or eliminates compromises in heat exchanger outlet temperatures and performance associated
3 with transient process perturbations without changing the thermal power at the inlet of the opposite
4 stream. Furthermore, the presented technology responds to changes in process transients with a very
5 short-lived transients in the exiting flows, which minimizes thermal stress in the heat exchanger during
6 process changes.

7 The majority of design in engineering focuses on the performance of steady-state processes. In this work,
8 steady-state analysis is expanded upon to include detailed transient behaviors. The objective of the work
9 is to aid the development of agile energy-storage systems and load-following processes, which are both
10 transient in nature. Transient analyses regularly identify heat exchangers as the unit operations that are
11 most vulnerable and sensitive to fluctuations in process conditions. During transient moments, heat
12 exchanger performance often undermines process safety, product quality, environmental threat, and
13 equipment reliability. Reliability threats stem from both materials constraints inside the heat exchanger
14 and from downstream ramifications of streams that fail to meet their design conditions.

15 The energy storage and process load following that inspire this work are only two of several applications
16 that profit from transient analyses. Transient analyses point out process susceptibilities to load changes,
17 upsets, control complexity, and broad responses prompted by local changes in densely integrated systems.
18 The solutions proposed and exhibited in this research greatly decrease and even eliminate these process
19 susceptibilities, at least with regards to heat exchangers. Specifically, these procedures simplify controls,
20 conserve process set points during transients, confine process upsets, and decrease process variable
21 cycling. This work's motivation is to describe a dynamic heat exchanger that will eliminate the exchanger
22 efficiency and performance losses that originate from the transient flow rates of entering streams. The
23 discussion here presumes a heat exchanger that is initially balanced, that is, initially has approximately
24 equal values of $\dot{m}c_p$ in each direction. The methods described here can also prevent further efficiency
25 losses in an imbalanced heat exchanger if it experiences a flow perturbation that further imbalances it.

However, flow perturbations that tend to balance it will improve its efficiency and generally would not benefit from this technology.

1.1. Background

Figure 1 shows a common, countercurrent, two-stream heat exchanger that could be of any design, i.e. brazed-plate, plate-and-frame, or shell-and-tube. Hot and cold streams enter the exchanger on the left and right sides, respectively, and exchange heat such that the temperature of the hot stream decreases while that of the cold stream increases. The outlet temperature of the cold stream often exceeds that of the hot stream, leading to possible confusion about which stream is the hot or cold stream. In this discussion, hot and cold classifications accompany the inlet temperatures, as is most common in practice and as shown in the figure. In this discussion, the outlet temperature of the cold stream is commonly higher than that of the hot stream.

Many industries use multi-stream heat exchangers in which more than two streams enter and exit at different positions along the heat exchanger. Industries that prioritize heat integration commonly use multi-stream heat exchangers, e.g., the cryogenic processing industry [36-39]. The analyses demonstrated here also apply to multi-stream systems assuming the hot and cold streams are regarded as the mass-flow-weighted average properties of all hot and cold streams.

1.2. Heat Exchanger Efficiency

The technical literature does not define a heat exchanger efficiency. This document suggests one such useful definition. Insulated heat exchangers conserve enthalpy, so a first-law definition is problematic. That is, the enthalpy flowing out of the system equals that flowing in. Heat exchangers do not involve shaft work, so a second-law definition (work over heat) does not naturally come to mind. Nevertheless, this document proposes a second-law definition for heat exchanger efficiency that is both useful and simple.

Conceptually, this efficiency describes the consequence of heat exchange on the flow streams' (a) ability to do work (availability or exergy) or, equivalently, (b) entropy. Quantitatively, the heat exchanger efficiency is one minus the difference in exergy between the streams exiting and entering a heat exchanger normalized by the largest achievable difference in such exergy. The largest achievable difference with respect to heat exchange occurs if all the streams come to the same temperature. These analyses and this definition assume no heat transfer between the exchanger and its surroundings. Therefore, the difference in exergy is proportional to a difference in entropy, with the ambient temperature as a proportionality constant. This expression depends on conditions at thermal equilibrium, that is, at the temperature the streams would become if the inlet streams were to equilibrate at a single temperature

$$\eta = 1 - \frac{\sum_{outlet} m_i \hat{b}_i - \sum_{inlet} m_i \hat{b}_i}{\sum_{eq} m_i \hat{b}_i - \sum_{inlet} m_i \hat{b}_i} = 1 - \frac{\Delta_{obs} m_i \hat{s}_i}{\Delta_{eq} m_i \hat{s}_i} = \frac{\sum_{eq} m_i \hat{s}_i - \sum_{outlet} m_i \hat{s}_i}{\sum_{eq} m_i \hat{s}_i - \sum_{inlet} m_i \hat{s}_i} \quad (1)$$

where $\hat{b}_i = \hat{h}_i - T_0 \hat{s}_i$ represents the specific exergy of stream i and $\Delta_{eq} m_i \hat{s}_i$ is the difference between the entropy in all the streams at equilibrium temperature and entropy of the streams at the inlet temperature. The temperature T_0 represents the temperature of the ambient environment and, since it drops out of the equation, is irrelevant to the exchanger efficiency. This ambient temperature should not affect a well-posed efficiency definition for reasons discussed below. The middle form of the equation results from the sum of the inlet enthalpy flows equaling the sum of the outlet enthalpy flows.

This efficiency becomes unity if the total exergy of the outlet streams equals that of the inlet streams, which can only occur in the limit of the hot and cold streams transferring heat with no temperature difference at any given point along the exchanger (ideal heat exchanger). The efficiency is zero if all outlet streams approach the same equilibrium temperature, which represents the poorest possible heat exchanger performance from an efficiency or entropy standpoint. An efficiency of 50% indicates that heat exchanger generates half of much entropy it would generate if all streams come to the same temperature.

1 This concept of heat exchanger efficiency is different from effectiveness, availability and exergy [7, 9,
2 40-48] in some crucial ways. Effectiveness is the ability to reach a given temperature in a stream(s)
3 quickly. For instance, quenching hot steel in water or oil is highly effective but highly inefficient heat
4 transfer because the heat transfer rate is rapid (effective) but much of the heat exchanges across a large
5 temperature difference (inefficient).

6 An alternative definition of efficiency is the ratio of the exergy of the stream before and after heat
7 exchange. This, however, would lead to confusing and unfortunate results in that it would depend on the
8 heat rejection or ambient temperature, which requires a situational definition of a material that is not
9 involved in the heat exchanger operation. A heat exchanger efficiency should only depend on temperature
10 differences associated with its operation, not on an ambient temperature. For example, if two heat
11 exchangers operate with identical temperature differences between the inlet and outlet streams and
12 between the hot and cold streams, they should have identical efficiencies. However, if one of them
13 operates near the local heat rejection temperature and the other operates at much higher temperatures, the
14 definition using the ratio of exergies would result in the high-temperature system efficiency greatly
15 exceeding the low-temperature efficiency.

16 When a heat exchanger is ideally balanced, the entering flows undergo equal but opposite temperature
17 changes. Additionally, if the specific heat capacities are independent of temperature, the product of mass
18 flow rate and specific heat, $\dot{m}\hat{c}_p$, must be the same between streams, which is an achievable goal. Heat
19 exchanger efficiency also increases as the instantaneous temperature difference between the streams is
20 reduced. A smaller stream temperature difference generally requires design changes such as increasing
21 the heat transfer area, obtaining a more uniform flow distribution, or decreasing the thickness or
22 temperature drop in plate materials. These design choices can only approach a zero temperature
23 difference, and therefore 100% efficiency, asymptotically, though some industrial designs achieve
24 differences of less than 1 K.

1.3. Balanced heat exchanger

Figure 2 illustrates counter-current flow arrangement of two fluids past a solid plate, with heat transferring from fluid 1 to fluid 2. The unsteady-state energy balance is applied to describe the temperature profile of stream 1, as follows,

$$\rho_1 c_{p,1} \left(\frac{\partial T_1(x, t)}{\partial t} + v_{x,1} \frac{\partial T_1(x, t)}{\partial x} \right) = -k \frac{\partial^2 T_1(x, t)}{\partial x^2} + q_1 \quad (2)$$

where a positive value of q_1 denotes heat increase, and the fluid is assumed well-mixed, so the effect of heat transfer is represented by a volumetric heat source term q_1 with typical units of J/cum.s that relates the heat flux, q_1'' , (J/sqm.s), as,

$$q_1 = \frac{A_1}{V_1} q_1'' = \frac{q_1''}{L_1} \quad (3)$$

where L_1 is $\frac{1}{2}$ the distance to an adiabatic channel wall. If we assume that the operation is steady-state and the conduction term is included in the heat transfer coefficient term,

$$\rho_1 c_{p,1} v_{x,1} L_1 \frac{\partial T_1(x)}{\partial x} = q_1 L_1 = q_1'' \quad (4)$$

where $\dot{m} = \rho L_1 v_{x,1} \Delta z$. Similarly for the stream 2,

$$\rho_2 c_{p,2} v_{x,2} L_2 \frac{\partial T_2(x)}{\partial x} = q_2 L_2 = q_2'' \quad (5)$$

Assuming h_i as the local heat transfer coefficient,

$$q_1'' = -h_1 (T_1(x) - T_p(x)) \quad (6)$$

$$q_2'' = -h_2 (T_2(x) - T_p(x)) \quad (7)$$

Writing the heat balance for the separation solid leads to,

$$q_1'' + q_2'' = -\Delta y \rho_p c_{p,p} \frac{dT_p(x)}{dt} = 0 \quad (8)$$

1 Consequently, the energy transport equations are converted to a system of ordinary differential equations
2 (ODEs), as follows,

$$(h_1 + h_2) \rho_1 l_1 C_{p,1} v_{x,1} \frac{dT_1(x)}{dx} + h_1 h_2 (T_1(x) - T_2(x)) = 0 \quad (9)$$

$$(h_1 + h_2) \rho_2 l_2 C_{p,2} v_{x,2} \frac{dT_2(x)}{dx} - h_1 h_2 (T_2(x) - T_1(x)) = 0 \quad (10)$$

$$x = 0 : T_1(x) = T_1^0 \quad (11)$$

$$x = L : T_2(x) = T_2^0 \quad (12)$$

3 where x represents the distance through heat exchanger.

4 The highest efficiencies in heat exchangers are achieved when the flows are balanced. A balanced heat
5 exchanger will transfer heat across streams with the same temperature difference at all locations along the
6 heat exchanger. This is usually identical to when both hot and cold streams have the same product of
7 $\dot{m}c_p$. For multi-channels, the sum of the $\dot{m}c_p$ products is used to determine if the system is balanced. The
8 temperature profiles for a simple, balanced heat exchanger are straight and parallel (Figure 3). In more
9 complex exchangers, the temperature profiles may not necessarily be straight. However, they remain
10 equidistant throughout and in a sense are still parallel. In either case, the exchanger is balanced as long as
11 the temperature difference between hot and cold streams is constant. That is,

$$T_h(x) - T_c(x) \equiv \Delta T(x) = a \quad (13)$$

12 where $T_h(x)$ and $T_c(x)$ are the temperatures of the hot and cold streams and functions of position inside
13 the heat exchanger, respectively, and a is a constant. In this definition, the interior temperature profiles
14 are generally unknown. A more reasonable denotation depends only on the usually known inlet and outlet
15 temperatures, namely,

$$T_{h,inlet} - T_{c,outlet} = T_{h,outlet} - T_{c,inlet} \quad (14)$$

This second denotation thoroughly describes the balanced condition for simple heat exchangers with straight temperature profiles and remains a practical definition for balanced conditions when the temperature profiles are not straight.

1.4. Heat Transfer Rate ratio

Several convenient dimensionless numbers were derived from theoretical analyses of heat exchanger performance. One of these numbers is the ratio of mass flow rates multiplied by the heat capacities in each channel. Identified as ζ in this paper, this parameter can be represented by the equation below for systems with unit width in both streams,

$$\zeta(x) = \frac{\dot{m}_1 c_{p,1}}{\dot{m}_2 c_{p,2}} = \frac{L_1 v_1 \rho_1 c_{p,1}}{L_2 v_2 \rho_2 c_{p,2}} \quad (15)$$

where $\dot{m}_i$, $c_{p,i}$, L_i , v_i , and ρ_i represent the mass flow rate, specific heat capacity, $\frac{1}{2}$ the channel thickness, velocity, and density of stream i . The index i is 1 for the hot stream and 2 for the cold stream. The parameter $\zeta(x)$ is positive for co-current and negative for countercurrent heat exchangers because of the directions and signs on the velocities. Thus, to be consistent, the mass flow rates in Eq. (15) are signed values, with flows in opposite directions having opposite signs. For heat exchangers that involve more than one hot or cold stream, ζ uses the flow-weighted sum of the flows, turning Eq. (15) into,

$$\zeta(x) = \frac{\sum_{hot} A_i v_i \rho_i c_{p,i}}{\sum_{cold} A_i v_i \rho_i c_{p,i}} \quad (16)$$

where A_i is the cross sectional heat transfer area occupied by stream i .

In a simple, balanced, counter-current heat exchanger, the value of ζ is generally at or nearly -1 and is practically independent of x . ζ is a function of position only when the physical fluid properties vary with temperature and thus position. When this is the case, an average ζ -value close to -1 can be assumed to be

1 a balanced condition provided that no complexities, such as phase changes, occur in the heat exchanger.

2 Inside the heat exchanger, heat transfer coefficients may also depend on position.

3 Should the hot stream flow rate increase by 40%, the value of ζ would decrease from -1 to -1.4 and the
4 ensuing heat exchanger temperature profiles would look approximately like those shown Figure 4. The
5 same change in ζ would be observed after increasing the hot-side heat capacity, density, or channel
6 thickness. Equivalently, after decreasing any of those same properties of the cold stream by 40% relative
7 to the values that produced the results in Figure 3 effects the same change in ζ . As Figures 3 and 4
8 portray, the practical result is that the difference between the hot inlet and cold outlet temperatures
9 decreases by about 10 °C, from about 16 °C to about 6 °C. The difference between the cold inlet and hot
10 outlet temperatures increases by about 11 °C, from about 16 °C to about 27 °C. The temperature increase
11 on the cold inlet end is slightly larger than the decrease on the hot inlet end.

12 A conceptually similar but quantitatively opposite result would occur if the cold stream flow rate were to
13 increase by 40% from its balanced value (Figure 5). That is, the hot inlet temperature difference increases
14 from 16 °C to about 27 °C while the cold inlet temperature difference decreases from about 16 °C to
15 about 6 °C.

16 Changes in heat exchanger outlet temperatures such as those described above occur in transients affiliated
17 with load changes, process upsets, and similar real-world adaptations. The real-world behaviors commonly
18 persist for a finite time as the process returns to normal operation. It is also common that flows in both
19 streams change simultaneously. For example, if the flow rates of both hot and cold streams increase by
20 40%, the heat exchanger would go back to a balanced condition, but with moderately higher temperature
21 differences between the streams (about 18 °C, or 2 °C higher) compared with the conditions in Figure 3,
22 as shown in Figure 6. Many times changes or upsets cause the heat exchanger performance to deviate
23 from starting conditions like Figure 3 making them shift in time through conditions similar to Figure 4
24 and/or Figure 5, ending back at the conditions in Figure 3 or Figure 6. When the conditions in Figure 4

and Figure 5 prevail, the product or process will often not meet specifications or fail to be produced at all. This results in lost production, increased waste, and commonly increases the safety or environmental hazards. The dynamic heat exchangers proposed in this paper minimize the effects of transients and apply to many types of heat exchangers and fluids.

2. Experimental System

2.1. Device Description

The dynamic heat exchanger system introduced here can retrofit an existing, classical heat exchanger and applies most conveniently to a system that involves relatively high volumetric heat capacities, such as two liquid streams.

This device description suggests using t-joints and three-way valves in various locations. Any functional equivalent to these systems is suitable. In the case of t-joints, t-joints with valves or three-way valves would be suitable alternatives. The three-way valves require a system that can stop flow in all three streams or allow flow between two of the three streams. Such valves are common, but a larger number of even more common two-way valves can provide the same control.

The configuration requires one or more additional valves, pumps, and tanks but does not require replacing or moving an existing heat exchanger. Its simplest form (Figure 7) modifies an existing heat exchanger (E-102), hot and cold inlet flows (P-649/637 and P-639/636, respectively), outlet flows (P-643 and P-651), and flow controllers. The modifications include a valve, bi-directional pump, and tank with optional heating or cooling or insulation for each of the hot and cold inlet lines (V-209, E-107, E-105 and E-106, respectively for the hot flow and V-210, E-108, E-104, and E-103, respectively, for the cold flow) connected with bi-directional pipes (P-654 and P-644 for the hot flow and P-653 and P-642 for the cold flow). The pumps would normally have associated controllers (not illustrated). In its absolute simplest

1 form, the system needs only the equipment connected to the hot or the cold flow, not both, though both
2 appear in this illustration.

3 Under normal conditions, there is no flow to or from the tanks and the heat exchanger operates as usual. If
4 an upset or transient changes one or both of the flows, the dynamic system creates a compensating flow
5 using the ancillary systems such that the flow through the heat exchanger remains balanced, the level of
6 material in the tanks changes, and the heat exchanger operates at its highest efficiency. When the upset
7 passes or the transient subsides, flows in the ancillary systems return to zero. A future transient or upset in
8 the opposite direction allows the system to return to its original condition, including the original levels of
9 stored material in the tanks. Thus, the system levels oscillating changes in demand, random transients,
10 and other dynamic perturbations around the long-term average conditions of the system.

11 In practice, bidirectional pumps are rare and both inefficient and expensive compared to traditional, uni-
12 directional counterparts. Bidirectional controllers are even more rare and expensive. Furthermore, pumps
13 and controllers are expensive compared to valves. The remainder of this discussion presumes a
14 configuration that allows a single, uni-directional pump to accomplish the same thing, removing the
15 complexity of two pumps and controllers but adding more valves. The plumbing complicates the flow
16 diagram, but the operating principle is the same as indicated in the less complex system illustrated in
17 Figure 8. The following discussion describes the hot inlet and associated equipment first, followed by the
18 cold inlet and associated equipment.

19 The incoming hot stream (P-533) connects to a t-junction, one line from which (P-590) connects to the
20 heat exchanger (E-76) and represents the heat exchanger inlet hot flow. The stream connected to the heat
21 exchanger (P-590) may have different properties, commonly a different flow rate, than the incoming hot
22 stream (P-533), and this difference is germane to this invention. The second line (P-542) from the t-joint
23 in the incoming hot stream connects to a three-way valve (V-177). One line (P-605) from this valve (V-
24 177) connects to the cold side of this process through a three-way valve (V-196), and is discussed later in

1 conjunction with the remaining cold-side components. The other line (P-602) from this valve (V-177)
2 connects to a t-joint (joint connecting P-602, P-606, and P-568). One line (P-568) from this t-joint leads
3 to the pump (E-79). The second line (P-606) from this t-joint connects with a three-way valve (V-189).
4 One line (P-603) from this valve connects to a different three-way valve (V-197) which lies on the cold-
5 side of the system and is discussed later. The second line (P-583) connects to a hot storage tank (E-86).

6 The inlet cold stream (P-545) connects to a t-junction, one line from which (P-551) connects to the heat
7 exchanger (E-76) and represents the heat exchanger inlet cold flow. This flow may differ, specifically in
8 flow rate, from the incoming cold stream (P-545), and this difference is germane to this invention. The
9 other line (P-570) from the t-junction connects to a three-way valve (V-196). One line (P-605) from this
10 three-way valve (V-196) has already been discussed above. The second line (P-607) from the three-way
11 valve (V-196) connects to another t-junction. One line (P-550) from this t-joint is the pump (E-79) outlet.
12 The other line (P-604) from this t-junction connects to a three-way valve (V-197). One line from this
13 three-way valve (V-197) connects to the cold storage (E-81). The other line (P-603) from this three-way
14 valve (V-197) was previously discussed.

15 A flow controller/meter for the pump controls the outlet stream (P-550) but could also be on the inlet
16 stream (P-568). Temperature indicators (TI-1, TI-2, TI-3, and TI-4) for the heat exchanger incoming and
17 outgoing hot and cold streams (P-590, P-551, P-594, and P-585, respectively) could similarly be located
18 in a variety of locations, though locating close to the heat exchanger is preferred. Flows illustrated with
19 no directional arrows could be in either direction and flow in different directions, depending on
20 operational modes.

21 As illustrated in this diagram, a small portion of the hot and cold streams can mix during dynamic heat
22 exchanger operation. Specifically, the amount of material in the pump inlet and outlet, P-568 and P-550,
23 respectively, and in the pump E-79 and some of the surrounding lines could mix when operation changes
24 modes. Different embodiments are discussed later that prevent such mixing by either using a single

stream to do the balancing or using separate pumps and dedicated lines. Also, if the pump (E-79) is bidirectional, meaning it can pump in either direction, the plumbing could be simplified. All of these alternative plumbing schemes are obvious to one skilled in art of process design. Also, the three-way valves illustrated here could be replaced with more common in-line valves located in pipes P-602, P-603, P-604, and P-605 and accomplish the same control. However, this would slightly increase the potential mixing of materials from the cold and hot streams when operating modes change.

A heater (E-86) and a chiller (E-81) for the hot and cold storage tanks (E-80 and E-77, respectively) optionally provide means to overcome heat losses and slight temperature differences if needed. The tanks also would optionally be insulated.

2.2. Operation

The dynamic heat exchanger returns the heat exchanger closer to its balanced operation by filling or emptying the storage tanks during an operational transient, using the flow rate difference to change the relative flow rates in the heat exchanger. There is a preference for accomplishing this by filling a tank, but the choice of filling or emptying the tank will usually be dictated by current level of fluids in the tanks. It is probable that the system will manage some transients by both filling one tank and emptying the other. The specific operations in a few specific scenarios appears below.

The first case considers a transient in which the hot incoming stream (P-533) transports more heat or enthalpy than the cold incoming stream (P-545) can balance in the heat exchanger ($|\zeta| > 1, \zeta < -1$). In this case, a portion of the hot stream flows to the hot storage tank (E-80), or a portion of the stored cold material in the cold storage tank (E-77) flows to the cold stream, or both, until the transient subsides. These scenarios appear below (Figure 9), with bolded lines representing lines with flow, arrows representing the flow direction, and dashed lines representing lines with no flow.

The converse case involves the hot incoming stream (P-545) transporting less heat or enthalpy than the cold incoming stream (P-533) can balance in the heat exchanger ($|\zeta| < 1, -1 < \zeta < 0$). In this case,

either a portion of the cold stream flows to the cold storage tank (E-77) or a portion of the stored hot material in the hot storage tank (E-80) flows into the hot stream, or some combination, until the transient subsides. These scenarios appear below (Figure 10), with bolded lines representing lines in which material is flowing, arrows representing the flow direction, and dashed lines representing lines with no flow.

Reference to the parameter ζ in the above figure captions indicates a summation in the numerator of all high-temperature streams, and in the denominator of all low-temperature streams:

$$\zeta = \frac{\sum_{hot} L_i v_i \rho_i c_{p,i}}{\sum_{cold} L_i v_i \rho_i c_{p,i}} = \frac{\sum_{hot} \dot{m}_i c_{p,i}}{\sum_{cold} \dot{m}_i c_{p,i}} \quad (17)$$

The main objectives of this heat exchanger is to minimize the entropy production and the Gibbs energy loss in the heat exchanger and to allow process flows to change quickly without compromising the exit flow temperature targets. In simple heat exchangers, this occurs as ζ approaches -1. In complex systems involving large variations in heat capacity, condensation, vaporization, reactions, streams entering one side of the heat exchanger at significantly different temperatures, or other complicating factors, entropy and Gibbs energy loss might be minimized when ζ differs somewhat from -1.

In normal heat exchanger operation, the temperature at which fluid is stored in the tank is somewhat closer to the average temperature (cooler in the hot tank and warmer in the cold tank) than the incoming streams. If the heat exchanger is balanced by withdrawing flows, this difference will be smaller than if it is balanced by adding flow. The heating/cooling elements and well-insulated tanks can eliminate this difference.

2.3. Apparatus

The dynamic heat exchanger uses a small, commercial two-stream brazed-plate heat exchanger (Swep B10THx8), with traditional incoming and outgoing hot and cold streams configured essentially identical to the system shown in Figure 8. It functions at reasonably good efficiency when balanced, similar to

most installed heat exchangers in industrial practice. Table 1 summarizes geometrical features of this plate heat exchanger. Water is the working fluid in this experiment. A bi-directional pump, with appropriate valves and plumbing, moves fluid in the appropriate direction. Flow meters (Great Plains Industry TM050-N ½" NPT) record the two incoming stream flowrates. Digital control valves (Vallworx 5610A-DPS) coupled with the meters through the control system provide flow control. Type T thermocouples (Omega TMTSS-062U-6) record temperatures of the streams flowing into and out of the heat exchanger as well as the tank fluid temperatures. A national Instruments board (NI cDAQ-9188XT) connected to a personal computer provides data acquisition and control.

2.4. Experimental Procedure

The general experimental approach demonstrated here involves establishing a steady, balanced flow through the exchanger, and then changing one stream flow so the exchanger is in an off-balance condition and then returning it back to the balanced condition while monitoring the exchanger performance. The flow perturbations are very nearly step functions, followed by a second step function back to the original flow condition, which provides an accurate indication of the exchanger dynamic response. Experiments reported here include heat exchanger performance with the dynamic system idle and active.

Flow conditions during active system operation depend on whether they reference the process streams or the heat exchanger streams. Process streams refer to calculations based on flow rates of streams approaching the heat exchanger prior to the location of their interface with the dynamic heat exchange system. Heat exchanger streams refer to calculations based on the flows in the heat exchanger itself. When the dynamic system is idle, these are identical. When it is active, they differ significantly. All discussions here refer to the system stream flow rates, not the flow rates in the exchanger.

2.5. Rapid System Response

This system, in addition to maintaining optimal efficiency and meeting set points during transients, also allows the system to move through these transients much more rapidly than is common for traditional

1 heat exchangers. Since the exchangers discussed here maintain essentially the same temperature profile
2 through the transient, the heat exchanger experiences essentially no thermal stress and presents little if
3 any barrier to the speed of the transient. Unlike traditional heat exchangers, which experience thermal
4 stresses and other temperature-profile-related performance issues during transients, these heat exchangers
5 have no such variations. This allows the process to move through transients, change loads, and follow set
6 points at much more rapid speeds than is otherwise the case. In the experimental verifications that follow,
7 step changes in stream properties, which are the fastest possible changes in conditions, illustrate this
8 improved response time. Step changes are the fastest possible changes in conditions and illustrate the
9 effectiveness of this process. Many, probably most, actual flow changes occur more slowly and probably
10 over smaller ranges. The success with step changes in flows illustrated here indicate that any slower or
11 smaller flow perturbations should be at least as effectively controlled.

12 3. Mathematical Modeling

13 Mathematical modeling and optimization of plate heat exchangers are widely discussed in the literature
14 [49]. The pioneering works in the field are those of Watson et al. [50] and Jackson and Troupe [51]. Wolf
15 [52], Bounopane et al. [53] and Zaleski [54] investigate the analytical solution of plate heat exchanger
16 using the eigenvalues. Further analytical studies in the field are done by Marano and Jechura [55]. Settari
17 and Venart [56] applied the polynomial function, and Zaleski and Klepacka [57] applied linear
18 exponential form to predict the thermal efficiency and steady-state temperature profile of plate heat
19 exchangers. Kandlikar and Shah [58] summarized all these steady-state analyses.

20 The literature dealing with the transient analysis of plate heat exchangers is widely available. McKnight
21 and Worley [59] pioneered the unsteady-state analysis of plate heat exchangers by applying the feedback
22 control of high-velocity flow. Khan et al. [60] explored the experimental and theoretical dynamic analysis
23 of a counter-current plate heat exchanger under flow transients. Sharifi et al. [61] and Lakshmanan and
24 Potter [62] presented a cinematic model to estimate the unsteady-state behavior of single-pass plate heat

exchanger. Das and his co-workers [63] explored the dynamic of plate heat exchangers incorporating several phenomena in their models such as dispersion in both fluids [64], axial thermal dispersion in fluid [65, 66], and port to channel maldistribution [67-72]. None of these studies address the issue of the reduced losses in the efficiency associated with transient process perturbations.

Transient energy balances model the heat exchanger performance. The following assumptions lead to a tractable mathematical model of a plate heat exchanger:

1. Counter-current flow;
2. No heat-in-leakage;
3. Negligible heat conduction along the plate;
4. No heat transfer through sealing edges and ports;
5. Fluids thermophysical and transport properties are independent of temperature;
6. No phase change;
7. Heat transfer coefficients that vary with Reynolds number (velocity) but not explicitly with time;
8. A plug-flow velocity profile for the fluids.

While this study addresses steady-heat transfer descriptions, it can be extended to problems with stronger unsteadiness [73-79]. Figure 11 presents the diagram of a single-pass, counter-current plate heat exchanger using the abovementioned assumptions. The plates range from number 1 to M , and the channels from 1 to N . Each plate exchanges heat with fluids passing on both of its sides, except the first and the last plates, which are in contact with one fluid. Considering small control volumes for the internal fluids and the plates, the transport equations are,

For fluid 1,

$$\rho_1 C_{p1} \left(\frac{\partial T_{1,j}}{\partial \tau} - v_1 \frac{\partial T_{1,j}}{\partial x} \right) + \frac{h_1}{L_1} (2T_{1,j} - T_{w,i-1} - T_{w,i}) = 0, \quad j = 2, 4, 6, \dots, N-1; i = 1, 2, \dots, M \quad (18)$$

For fluid 2,

$$\rho_2 C_{p_2} \left(\frac{\partial T_{2,j+1}}{\partial \tau} + v_2 \frac{\partial T_{2,j+1}}{\partial x} \right) - \frac{h_2}{L_2} (T_{w,i} + T_{w,i+1} - 2T_{2,j+1}) = 0, \quad j = 1, 3, 5, \dots, N; i = 1, 2, \dots, M \quad (19)$$

1 For the plates,

$$\rho_w C_{p_w} \delta \frac{\partial T_{w,i}}{\partial t} - h_h (T_{h,j} - T_{w,i}) + h_2 (T_{w,i} - T_{c,j+1}) = 0, \quad j = 1, 2, \dots, N; i = 2, \dots, M - 1 \quad (20)$$

2 For the 1st plate,

$$\rho_w C_{p_w} \delta \frac{\partial T_{w,i}}{\partial t} - h_1 (T_{1,j} - T_{w,i}) = 0, \quad j = 1, 2, \dots, N; i = 1 \quad (21)$$

3 For the Mth plate,

$$\rho_w C_{p_w} \delta \frac{\partial T_{w,i}}{\partial t} + h_2 (T_{w,i} - T_{2,j+1}) = 0, \quad j = 1, 2, \dots, N; i = M \quad (22)$$

4 where δ is the plate thickness, L_i is $\frac{1}{2}$ the channel thickness, and h_i is the heat transfer coefficient that
 5 needs to be determined by experiment. The general correlation of plate heat exchangers describes the
 6 dependence of the heat transfer coefficient on flow rate,

$$Nu = C Re^n Pr^m \quad (23)$$

7 A least-squares analysis determines the constants and exponents for the heat transfer coefficient
 8 correlation resulting in parameters of $C = 0.312$, $n = 0.397$, and $m = \frac{1}{3}$.

3.1. Discretization of model: orthogonal collocation approach

In this work, the method of lines method provides the numerical solution of the model. Discretized spatial derivatives, $\frac{\partial}{\partial x}$, require implicit, iterative solutions but the time-dependent derivatives are explicit. This converts the system of partial differential equations (PDEs) to a set of time-dependent ordinary differential equations (ODEs). The orthogonal collocation approach provides the spatial discretization. This is an efficient and accurate method for the approximation of PDEs by their equivalent system of ODEs [80-86].

The orthogonal collocation method normalizes a finite domain (a, b) to the domain $(0,1)$ through the following transformation,

$$z = \frac{x - a}{b - a} \quad (24)$$

where z is the new independent variable in the $(0,1)$ range. Then, N interior collocation points in the $(0,1)$ domain lie between these limits while $z_1 = 0$ and $z_{N+2} = 1$ correspond to the boundary points. The choice of the collocation points affects solution accuracy. The roots of the Jacobi orthogonal polynomial provide the optimum positions of the interior collocation points. The Rodrigues formula provides the Jacobi polynomial of N^{th} degree as follows,

$$J_N^{(\alpha, \beta)}(z)(z^\beta(1-z)^\alpha) = \frac{(-1)^N \Gamma(\beta+1)}{\Gamma(N+\beta+1)} \frac{d^N}{dz^N} (z^{N+\beta}(1-z)^{N+\alpha}) \quad (25)$$

where α and β characterize the polynomial, and Γ is the Gamma function. The interpolation polynomial passing through the $N+2$ points provides the set of collocation points $(x_1, y_1), (x_2, y_2), \dots, (x_N, y_N), (x_{N+1}, y_{N+1}), (x_{N+2}, y_{N+2})$,

$$y_{N+1}(z) = \sum_{i=1}^{N+2} y_i l_i(z) \quad (26)$$

- 1 where y_{N+1} is the $N + 1^{\text{th}}$ degree polynomial, y_i is the state variable that corresponds to z_i , and $l_i(x)$ is
 2 the Lagrange interpolation polynomial, which is expressed by the following equations,

$$l_i(x_j) = \begin{cases} 0 & i \neq j \\ 1 & i = j \end{cases} \quad (27a)$$

$$l_i(z_j) = \prod_{\substack{j=1 \\ j \neq i}}^{N+2} \frac{(z - z_j)}{(z_i - z_j)} = \frac{p_{N+2}(z)}{(z - z_i) \left(\frac{dp_{N+2}(z_i)}{dz} \right)} \quad (27b)$$

- 3 where p_{N+2} is the node polynomial and is expressed as,

$$p_{N+2}(x) = (x - x_1)(x - x_2) \dots (x - x_{N+1})(x - x_{N+2}) \quad (28)$$

- 4 Knowing the Lagrange interpolation polynomial, the state variables at any point in the domain becomes,

$$y_{N+1}(z^*) = \sum_{i=1}^{N+2} y_i l_i(z^*) \quad (29)$$

- 5 As the interpolation polynomial described in Eq. (29) is a continuous function, it has a defined derivate,
 6 that is,

$$\frac{dy_{N+1}(z_i)}{dz} = \sum_{j=1}^{N+2} \frac{dl_j(z_i)}{dz} y_j, \quad i = 1, 2, \dots, N + 2 \quad (30)$$

- 1 Therefore, the vector of the state variables Y , and the vector of the first spatial derivatives Y'_{N+1} can be
 2 simplified as,

$$Y = (y_1, y_2, \dots, y_{N+1}, y_{N+2})^T \quad (31)$$

$$Y'_{N+1} = \left(\frac{\partial y_{N+1}}{\partial z} \Big|_{z=z_0}, \frac{\partial y_{N+1}}{\partial z} \Big|_{z=z_1}, \dots, \frac{\partial y_{N+1}}{\partial z} \Big|_{z=z_{N+1}}, \frac{\partial y_{N+1}}{\partial z} \Big|_{z=z_{N+2}} \right)^T \quad (32)$$

- 3 Applying Eqs. (30) and (32), the derivative equation can be rewritten as,

$$Y' = A_z \cdot Y \quad (33)$$

- 4 where A_z is a $(N + 2, N + 2)$ square matrix and is stated as,

$$A_z = \left\{ a_{ij} = \frac{dl_j(z_i)}{dz}; \quad i, j = 1, 2, \dots, N + 2 \right\} \quad (34)$$

- 5 Accordingly, the spatial derivatives of the state variables can be rearranged as,

$$\frac{\partial y}{\partial z} \Big|_{z=z_i} = \sum_{k=1}^{N+2} A_z(i, k) y(k) \quad i = 1, 2, \dots, N + 1 \quad (35)$$

- 6 After substitution of Eq. (35) into Eqs. (18) and (19), the system of PDEs are converted to the following
 7 set of ODEs:

$$\begin{aligned} \rho_1 C_{p_1} \left(\frac{\partial T_{1,j}}{\partial \tau} - v_1 \left(\sum_{k=2}^{N+2} A_z(i, k) T_1(k) + A(i, 1) T_1^0 \right) \right) + \frac{h_1}{L_1} (2T_{1,j} - T_{w,i-1} - T_{w,i}) = 0, \quad j \\ = 2, 4, 6, \dots, N - 1; i = 1, 2, \dots, M \end{aligned} \quad (36)$$

$$\rho_2 C_{p_2} \left(\frac{\partial T_{2,j+1}}{\partial \tau} + v_2 \left(\sum_{k=1}^{N+1} A_z(i, k) T_2(k) + A_z(i, N+2) T_2^0 \right) \right) - \frac{h_2}{L_2} (T_{w,i} + T_{w,i+1} - 2T_{2,j+1}) \quad (37)$$

$$= 0, \quad j = 1, 3, 5, \dots, N; i = 1, 2, \dots, M$$

where T_i^0 denotes the inlet flow temperatures. The Runge-Kutta-Fehlberg method solves the resulting system of ODEs simultaneously.

4. Results and Discussion

All of the results included here illustrate the system response to a step change in ζ . The change is accomplished by changing one or both of the fluid velocities such that ζ changes by about a factor of 2. Many practical systems change velocities more slowly and over a more limited range than this. A large step change presents the most difficult perturbation for this system to manage. All other changes will be slower and over a more limited range in most practical systems. Such practical systems should lend themselves to easier management with the techniques discussed here.

The changes modeled and measured here include changes in ζ , that is, changes in flow rate (velocity) or heat capacity (composition). Presumably step changes in heat exchanger design could also be included, though it is difficult to imagine a transient situation where this would be relevant. However, ζ does not capture changes in inlet temperatures. Such temperature changes shift the optimal efficiency of a heat exchanger higher or lower as the difference in the inlet temperatures decreases or increases, respectively. However, a balanced and optimized heat exchanger will still be at its optimal operating point if ζ remains constant, even if the inlet temperatures change. The only way to improve its efficiency is to change the heat exchanger design. If both inlet temperature and ζ change, these techniques can maintain optimal conditions by managing ζ but the temperature change will still potentially affect the heat exchanger efficiency.

4.1. $|\zeta| > 1$

The experiments involve a step change in the conditions of a steady-state, nominally balanced ($|\zeta| \approx 1$) heat exchanger, causing $|\zeta|$ to increase by a factor of nearly 2. This increase is accomplished by changing the flowrate of one or both streams, however in principle, shifts in temperature or composition of either stream would accomplish the same thing. Figure 12 shows typical data from one of the experiments in which the stream flow rates vary abruptly and then return to their baseline values. The flow rate changes occur mainly by almost halving the cold stream flow rate and secondarily by a slight increase in the hot flow rate during the idle case. The results of the active and idle conditions come from two separate experiments and show negligible differences in timing. However, the cold and hot flow rate differences prior to, during, and after the perturbation correspond closely to each other in the idle case and the active case, providing an appropriate baseline for comparison in the efficiencies.

Figure 13 displays the outlet temperatures (left ordinate) and $|\zeta|$ (right ordinate) for the idle case. The outlet temperatures and $|\zeta|$ do not change noticeably in the active case. That is, the outlet temperatures change rapidly, but more slowly than $|\zeta|$ and the flow rates change. The slow response is in part because the heat exchanger's mass and heat capacity prevent temperature from changing instantaneously and in part because it takes some time for the flow to transit the heat exchanger. The value of $|\zeta|$ changes from close to unity to slightly below 2 and then returns to unity. Figure 14 illustrates the outgoing temperature profiles when the dynamic technology is active. As mentioned, the stream temperatures do not change noticeably.

Figure 15 illustrates the heat exchanger efficiency variation in time during operations with the dynamic heat exchanger technology active and idle for a two-stream, counter-current heat exchanger operating with perturbations in $|\zeta|$ ranging from 1 to 1.8, similar to the graphs shown in the previous section. There are large transient perturbations in the computed efficiencies associated with the rapid changes in $|\zeta|$ at the beginning and end of the perturbation. The most important and obvious trend in the figure is that the

dynamic heat exchanger technology essentially removes the drop in efficiency associated with the transient.

4.2. $|\zeta| < 1$

In the converse case, the cold flow increases abruptly by about 50% from its original value, followed by a similar decrease in the cold flow in the idle case (Figure 16). Changes in outlet temperatures (left ordinate) and $|\zeta|$ (right ordinate) appear in Figure 17 for the idle case. Figure 18 and Figure 19 illustrate the outlet temperature profiles and the heat exchanger efficiency alteration through the process, respectively, when the dynamic technology is active and idle.

Similar features of the spikes in the efficiency and experiments having somewhat different starting and stopping times for the $|\zeta|$ perturbation apply to Figure 19 as applied to Figure 15. The fact that the efficiency spikes occur in the opposite directions here as in the previous scenario results from the $|\zeta|$ perturbation being in the opposite direction, less than one, here as in the previous case, where it was greater than one.

These two scenarios ($|\zeta| > 1$ and $|\zeta| < 1$) clearly demonstrate that the dynamic heat exchanger technology maintains the heat exchanger efficiency at its balanced value through the perturbation, and this is germane to this work. The internal temperature profiles of the exchanger are basically constant during the transient, so the process can move through the transient at arbitrarily high rates. The only concern is that the tanks must be capable of holding enough fluid to last through the transient.

In these transients, the efficiency may exceed unity because of the heat exchanger internal temperature changing the flow temperatures and the finite transit time of the flow through the heat exchanger. The effect of the internal temperature profile is only significant when the dynamic heat exchanger technology is idle. The finite transit time has significant effect. The flow controller sensing thresholds and response time also lead to some efficiency spikes. The two results shown here come from separate experiments and

the beginning and ending of the $|\zeta|$ perturbation correspond quite closely but not precisely in the two cases.

5. Conclusions

This investigation includes the theoretical and experimental analysis of a heat exchanger technology that (1) significantly reduces or eliminates compromises in heat exchanger outlet temperatures so they do not propagate through the system, and (2) responds to changes in process transients with a near-zero time constant, which minimizes thermal stress in the heat exchanger during process changes. The technology can retrofit an existing heat exchanger by connecting two holding tanks to the incoming streams to maintain nearly balanced conditions through the process transient. This dynamic system maintains outlet temperatures very near their design points and nearly eliminates compromises in performance and efficiency. This allows the process to maintain product specifications, so far as heat transfer is concerned, as it moves through transient conditions. This heat exchanger reduces the process perturbation effective response time to near zero with potentially large contributions to process flexibility and load change ability. Additionally, the comparison between the experimental and simulated results suggests that the presented theoretical model can provide a good estimate of the dynamic heat exchanger behavior.

Acknowledgment

Sustainable Energy Solutions (SES) is greatly acknowledged for the material, financial support and technical cooperation to the authors.

Nomenclature

A	Heat transfer area [m^2]
a	Model parameter, dimensionless
b	Model parameter, dimensionless
n	Model parameter, dimensionless

$\dot{m}$	Mass flow rate [kg/s]
$\hat{g}$	Specific Gibbs energy [J/kg]
T	Temperature [K]
$\hat{s}$	Specific entropy [J/kg.K]
v_x	Fluid velocity [m/s]
x	Distance through the heat exchanger[m]
t	Time [s]
q	Volumetric heat source [J/m ³ .s]
q''	Heat flux [J/m ² .s]
C_p	Specific heat capacity [J/kg.K]
D_{eq}	Equivalent diameter [m]
L	Half of channel thickness [m]
k	Thermal conductivity [W/m.K]
h	Convective heat transfer coefficient [W/m ² .K]
Re	Reynolds number, dimensionless
Pr	Prandtl number, dimensionless
$LMTD$	Logarithmic mean temperature difference [K]
Nu	Nusselt number, dimensionless
U	Overall heat transfer coefficient [W/m ² .K]
y	State variable
l	Lagrange polynomial
N	Channels number
m	Number of experimental run
Q	Heat load [W]

Greek letters

δ	Plate thickness [m]
φ	Quadratic error function
ζ	Heat transfer rate ratio
ρ	Density [kg/m ³]

Subscript

in	Fluid inlet
out	Fluid outlet
cold	Low-temperature fluid
hot	High-temperature fluid
pred	Predicted
exp	Experimental

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7

Figure 1 Schematic diagram of a typical two-stream, counter-current heat exchanger

Figure 2 Flow arrangement for a counter-current plate heat exchanger

Figure 3 Temperature profiles for the hot and cold streams (top and bottom, respectively) and for the plate separating them (middle line) in a counter-current, steady-state, balanced heat exchanger with $\zeta = -1$ and temperature boundary conditions/set points set at the hot ($x = 0$) end for the hot stream and the cold ($x = 30$) end for the cold stream.

Figure 4 Temperature profiles for the hot and cold streams (top and bottom, respectively) and for the plate separating them (middle line) in a counter-current, steady-state heat exchanger with $\zeta = -1.4$ and temperature boundary conditions/set points set at the hot ($x = 0$) end for the hot stream and the cold ($x = 30$) end for the cold stream.

Figure 5 Temperature profiles for the hot and cold streams (top and bottom, respectively) and for the plate separating them (middle line) in a counter-current, steady-state heat exchanger with $\zeta = -1/1.4 = -0.71$ and temperature boundary conditions/set points set at the hot ($x = 0$) end for the hot stream and the cold ($x = 30$) end for the cold stream.

Figure 6 Temperature profiles for the hot and cold streams (top and bottom, respectively) and for the plate separating them (middle line) in a counter-current, steady-state, balanced heat exchanger with $\zeta = -1$ but with both hot and cold flow rates 40% higher than the conditions in Figure 3 and temperature boundary conditions/set points set at the hot ($x = 0$) end for the hot stream and the cold ($x = 30$) end for the cold stream.

Figure 7 Simplest form of the dynamic heat exchanger

Figure 8 Schematic drawing of a dynamic, two-stream heat exchanger

Figure 9(b) Dynamic heat exchanger operation when the hot flow stream carries more heat or enthalpy than the cold stream can balance ($\zeta < -1$) and the fluid in the cold storage tank supplements the cold flow to help balance the heat exchanger. Bold, solid lines contain flow moving in the direction indicated by the arrows and dashed lines have no flow.

Figure 10(a) Dynamic heat exchanger operation when the cold flow stream carries more heat or enthalpy than the hot stream can balance ($-1 < \zeta < 0$) and a portion of the cold flow is stored in the cold storage. Solid lines contain flow moving in the direction indicated by the arrows and dashed lines have no flow.

(b)

Figure 10(b) Dynamic heat exchanger operation when the cold flow stream carries more heat or enthalpy than the hot stream can balance ($-1 < \zeta < 0$) and the stored hot fluid is used to supplement the hot flow rate. Solid lines contain flow moving in the direction indicated by the arrows and dashed lines have no flow.

Figure 11

Control volume of the fluid inside the channel and the solid control volume in the plate

Figure 12 Typical temporal variation in the flowrates of a two-stream, counter-current heat exchanger involving a perturbation from $|\zeta| \approx 1$ to $|\zeta| \approx 1.8$ (more hot flow than cold flow). The active and idle conditions represent results from two separate experiments and show minor differences in conditions. However, the hot and cold flowrate differences, both during the perturbation and before and after it, are nearly identical, providing a good basis for comparison in the efficiencies.

Figure 13 Typical temporal variation in the outlet temperatures (left ordinate) and $|\zeta|$ (right ordinate) in a two-stream, counter-current heat exchanger with an idle dynamic heat exchanger system involving a perturbation from $|\zeta| \approx 1$ to $|\zeta| \approx 1.8$ (more hot flow than cold flow).

Figure 14 Outlet temperature profiles in a two-stream, counter-current heat exchanger with an active dynamic heat exchanger system involving a perturbation from $|\zeta| \approx 1$ to $|\zeta| \approx 1.8$ (more hot flow than cold flow).

Figure 15 Efficiency variation in time during operations with the dynamic heat exchanger technology active and idle for a two-stream, counter-current heat exchanger operating with perturbations in $|\zeta|$ ranging from one to 1.8. These are separate experiments and the beginning and ending of the time perturbation correspond quite closely but not exactly in the two cases.

Figure 16 Typical temporal variation in the flowrates of a two-stream, counter-current heat exchanger involving a perturbation from $|\zeta| \approx 1$ to $|\zeta| \approx 0.6$ (more cold flow than hot flow). The active and idle conditions represent results from two separate experiments and show minor differences in conditions. However, the hot and cold flowrate differences, both during the perturbation and before and after it, are nearly identical, providing a good basis for comparison in the efficiencies.

Figure 17 Typical temporal variation in the outlet temperatures (left ordinate) and $|\zeta|$ (right ordinate) in a two-stream, counter-current heat exchanger with an idle dynamic heat exchanger system involving a perturbation from $|\zeta| \approx 1$ to $|\zeta| \approx 0.6$ (more cold flow than hot flow).

Figure 18 Outlet temperature profiles in a two-stream, counter-current heat exchanger with an active dynamic heat exchanger system involving a perturbation from $|\zeta| \approx 1$ to $|\zeta| \approx 0.6$ (more cold flow than hot flow).

Figure 19 Efficiency variation in time during operations with the dynamic heat exchanger technology active and idle for a two-stream, counter-current heat exchanger operating with perturbations in $|\zeta|$ ranging from 1 to 0.6. These are separate experiments and the beginning and ending of the time perturbation correspond quite closely but not exactly in the two cases.

Table 1 Main dimension of Sweb B10THx8 plate heat exchanger

Dimension	Value
Plate length (in/mm)	11.45/291
Plate width (in/mm)	4.72/120
Plate thickness (in/mm)	0.0118/0.3
Heat transfer area (ft ² /m ²)	2.04/0.19
Height (in/mm)	1.16/29.4