Desktop Learning Module Heat Exchanger Performance

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Abstract

Researchers at Washington State University have developed miniaturized hands-on learning stations or Desktop Learning Modules (DLM) to help demonstrate most basic fluid and heat transfer concepts in the classroom. Low-cost, 1 ft^3 modules have been developed with interchangeable cartridges for dye injection into a flow stream; flow measurement with venturi, orifice and pitot tube meters; shell-and-tube, extended area and double pipe heat exchange; and packed bed and fluidized bed performance. The DLMs are effective learning tools, but are they useful in collecting laboratory data?

An experimental study was performed to determine the duty of the DLM shell-and-tube heat exchanger, and then to compare the results to theoretical predictions. Although a few minor modifications of the apparatus were necessary in order to obtain accurate data, experimental heat transfer rates on the tube side (539-831 W) were within 15-20% of theoretical predictions. Similarly, experimental heat transfer rates on the shell side (681-1,068 W) were within 1-11% of theoretical.

Introduction

One of the main objectives of engineering education is to effectively transfer subject information to the engineering students. A number of methods have been developed for enhancing student learning including multimedia developments,[1,2] active, problem-based learning,[3] collaborative learning,[4,5] and participation in cooperative education.[6] Several papers have specifically addressed methods for improving or supplementing the teaching of heat transfer including the use of spreadsheets to solve two-dimensional heat transfer problems,[7] the use of a transport approach in teaching turbulent thermal convection,[8] the use of computers to evaluate view factors in thermal radiation,[9] implementation of a computational method for teaching free convection,[10] and the use of an integrated experimental/analytical/numerical approach that brings the excitement of discovery to the classroom.[11] Supplemental heat transfer experiments for use in the laboratory or classroom have also been presented, including rather novel experiments such as the drying of a towel[12] and the cooking of French fry-shaped potatoes.[13] Suggestions for the integration of heat transfer material into the laboratory and classroom have been described by Penney and Clausen,[14-20] who presented a number of simple hands on heat transfer experiments that can be constructed from materials present in most engineering departments. This cross-course integration of course material has been shown to be a very effective learning tool that causes students to think beyond the content of each individual course.[21]

Golter *et al.*[22] recently described the development and use of Desktop Learning Modules (DLMs) in the teaching of fluids and heat transfer, which may be used as classroom demonstration units or as modular hands-on learning tools. The DLMs are small (most fit into a 1 ft^3 space), portable and relatively inexpensive. DLMs have been developed with interchangeable cartridges for dye injection into a flowing stream; flow measurement with venturi, orifice and pitot tube meters; heat exchange with shell and tube, extended area or double pipe heat exchangers; and pressure drop measurement through packed or fluidized beds. A photograph of the DLM containing a shell and tube heat exchanger module is shown in Figure 1, and a close-up of the shell and tube heat exchanger module is shown in Figure 2.

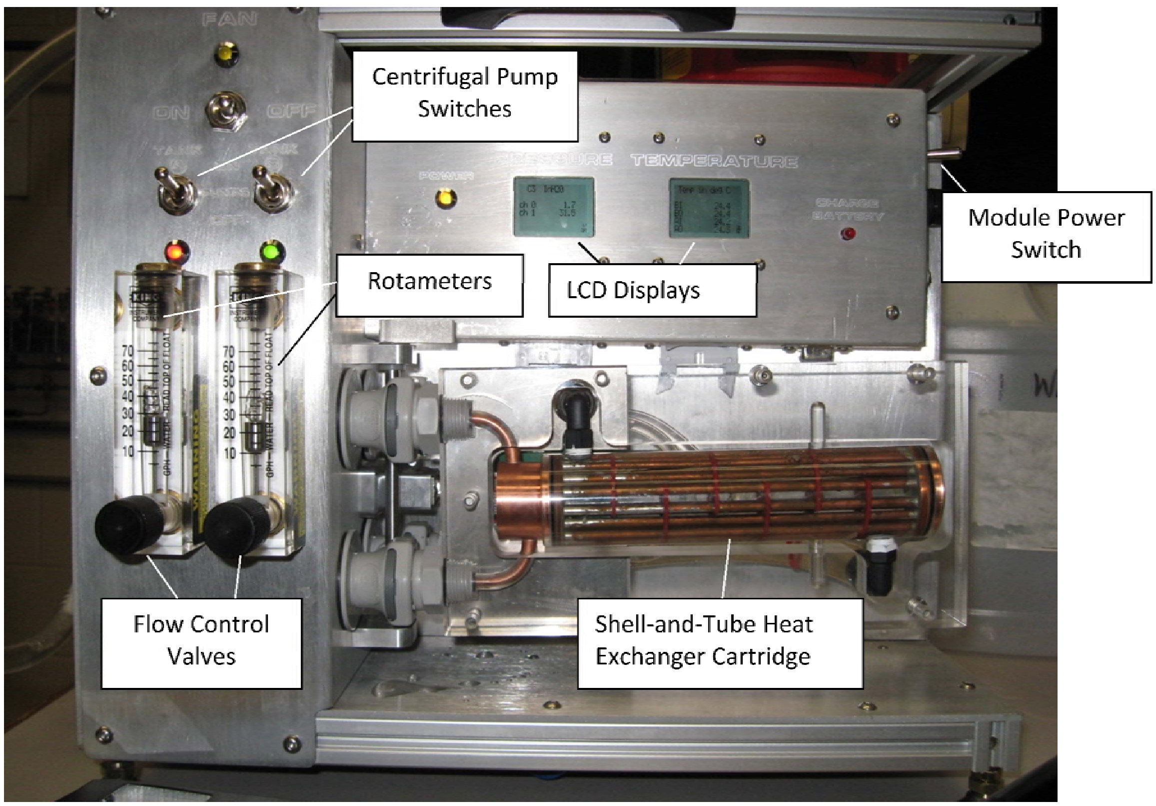

Figure 1. Photograph of DLM with Shell and Tube Heat Exchanger

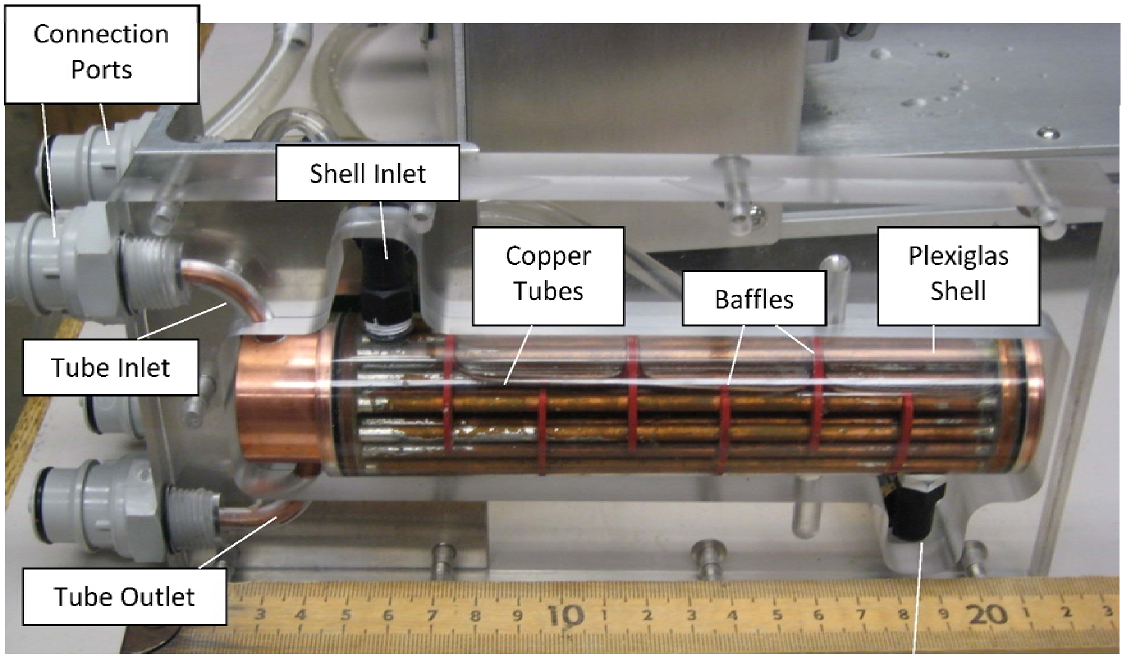

Figure 2. Close-up of Shell and Tube Heat Exchanger Module

DLMs are well accepted by students and have been shown to enhance student learning. In utilizing DLMs in teaching the characteristics of open-channel flow, flow control and measurement in a Civil Engineering Water Resources class, one section of students was given hands-on active learning (the test group) and the other section (as a control) received traditional lecture.[23] Concept inventory performance for the test group improved 52.1% over pre-test results. In addition, a flashlight survey showed that 29.5% of the students in the test group were very satisfied and 65.9% were satisfied with the hands-on active method when compared with traditional lecture. In another flashlight study, 98% of the surveyed students had the opinion that hands-on group learning with DLMs helped them remember important facts, 93% said they had a more thorough understanding of ideas and concepts, and 93% said they were better able to visualize ideas.[24]

Although DLMs are excellent demonstration and modular learning tools, the quality of the data from the modules has not yet been demonstrated. The purpose of this paper is to determine the duty of the DLM shell-and-tube heat exchanger, and then to compare the results to theoretical predictions. Specifically, experiments were performed to:

- Measure the inlet and outlet shell-side and tube-side fluid temperatures at ten shell-side to tube-side flow rate ratios, and then calculate the duty of the heat exchanger based on these measurements;
- Calculate the duty of the heat exchanger using the ideal tube bank approximation; and,

- Compare the duties from these two methods to evaluate the performance of the heat exchanger.

Equipment and Procedures

A schematic of the DLM shell and tube heat exchanger is shown in Figure 3. The test cartridge contained two tube passes and one shell pass. The 3.175 mm OD copper tubes were housed in a TEMA NEN shell geometry inside a 32 cm ID x 14.3 cm long Plexiglas® shell. The lateral tube pitch was 4.72 mm and the vertical tube pitch was 4.26 mm. The shell contained 6 baffles, with a baffle spacing of 2 cm and a baffle cut of 25%. In addition to the heat exchanger, the DLM module also contained:

- thermocouples, located at the inlet and outlet shell-side and tube-side streams, to measure inlet and outlet water temperatures;
- pressure transducers, with pressure taps also located at the inlet and outlet shell-side and tube-side streams, to measure the pressure drop across the tube- and shell-sides;
- two Plexiglas® tanks, to serve as reservoirs for the shell-side and tube-side fluids;
- a tank equilibration valve, connecting the tanks, which allows equilibration of the water temperature;
- two centrifugal pumps;
- flow-control valves, to control the shell-side and tube-side flow rates;
- two rotameters;
- a recycle line, to recirculate tube-side fluid; and
- a control and display panel, to operate the system and display pressure drop and temperature.

Additional equipment, not connected to the module, included a Dickson TC200 thermocouple probe and temperature indicator and a Fisher Automerse immersion heater. These additional items were required to obtain accurate temperature readings from the DLM.

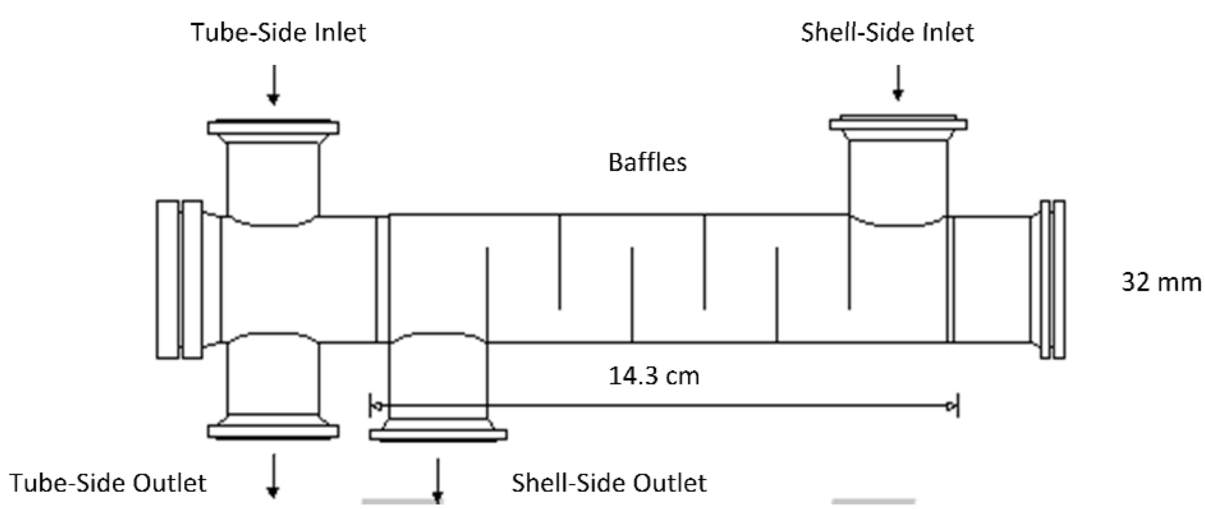

Figure 3. Schematic of DLM Shell and Tube Heat Exchanger

Figure 4 shows a schematic of the DLM with the heat exchanger cartridge. To operate the system, the Plexiglas® tanks were filled with tap water. The water for the tube-side was heated to 50°C using the immersion heater, and water in for the shell-side was cooled to just above 25°C through the addition of a small amount of ice. The system was operated by first holding the shell-side flow rate at an intermediate value (15 gph) and adjusting the tube-side flow rate (10-22.5 gph), and then holding the tube-side flow rate at an intermediate value (15 gph) and adjusting the shell-side flow rate (10-22.5 gph). When steady state was reached at each setting (in just a few minutes because of the small size of the apparatus), the inlet and outlet temperatures on the shell-side and tube-side were recorded.

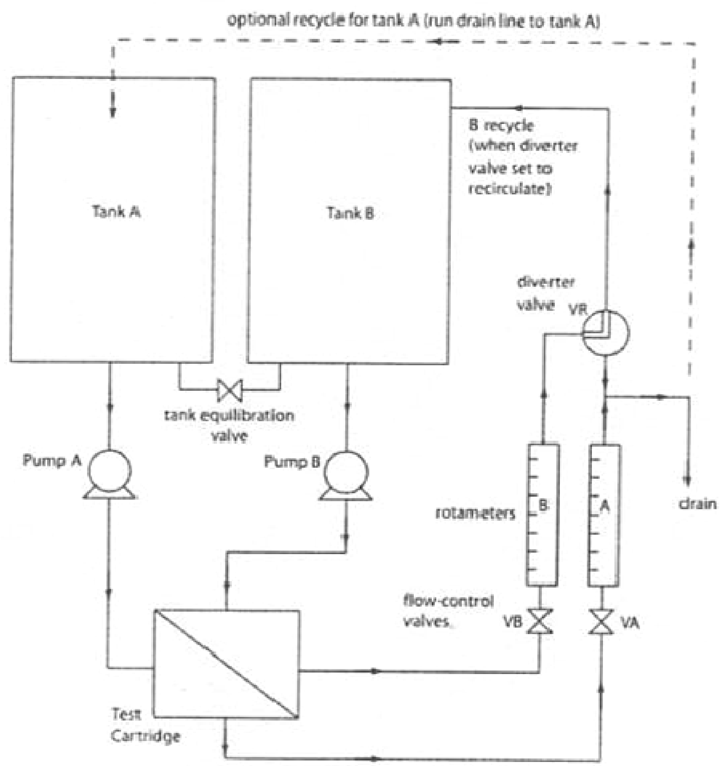

Figure 4. Schematic of DLM with Heat Exchanger Cartridge

Results and Discussion

Table 1 shows the experimental data from the DLM system. Energy balance closure, calculated as $\dot{m}C_p\Delta T$ for both the tube- and shell-sides, ranged from 73-91%.

Table 1. Inlet and Outlet Water Temperatures at Varying Shell- and Tube-Side Flow Rates

	Shell Side			Tube Side		
Run	Flow Rate (gph)	Inlet Temperature (°C)	Outlet Temperature (°C)	Flow Rate (gph)	Inlet Temperature (°C)	Outlet Temperature (°C)
1	15	16.8	28.1	10	53.7	41.3
2	15	15.4	29.4	15	54.2	43.3
3	15	16.8	33.0	20	54.0	44.8
4	15	17.8	31.7	17.5	52.2	43.0

5	15	18.1	34.3	22.5	52.9	44.4
6	10	16.5	32.0	15	51.9	42.5
7	15	16.3	28.7	15	51.9	41.3
8	17.5	16.4	27.1	15	51.3	40.4
9	20	17.7	27.1	15	51.6	40.5
10	22.5	18.6	27.5	15	51.3	40.4

Morphing the Cylindrical Tube Bundle to an Ideal Square Tube Bank

To calculate the duty of the heat exchanger, the cylindrical tube bundle was morphed to an ideal square in-line tube bank. The inside cross-sectional area of the cylindrical exchanger shell was first set equal to the cross-sectional area of the ideal tube bank, as noted in Equations (1) and (2).

$$A_{shell} = \frac{\pi D_{shell}^2}{4} \qquad (1)$$

$$A_{shell} = A_{square} \qquad (2)$$

The side length of the square tube bank was calculated from the bank area, as is shown in Equation (3).

$$W^2 = A_{square} \qquad (3)$$

The ideal tube bank has the same number of tubes in each row and column. Therefore, the number of tubes in a row of the ideal tube bank was found by taking the square root of the total number of tubes in the exchanger, as noted in Equation (4).

$$N_{tr} = \sqrt{N_t} \qquad (4)$$

The center-to-center tube spacing in the ideal tube bank (transverse and longitudinal pitch) was found by dividing the side length of the bank by the number of tubes per vertical row, as is noted in Equation (5).

$$S_t = \frac{W}{N_{tr}} \qquad (5)$$

The gap distance between the tubes in the ideal bank was then calculated using Equation (6).

$$G = S_t - D_t \qquad (6)$$

There are an equal number of gaps and tubes in the bank, with a half gap between the outermost tube of each row and the theoretical square shell. The minimum shell-side flow area occurs when the fluid enters the bank and flows between the tubes. This area is equal to the product of the gap distance and the distance between the baffles, or

$$A_g = N_{tr} G B \qquad (7)$$

The maximum fluid velocity over the tube bank is equal to the volumetric flow rate on the shell-side divided by the minimum shell-side flow area as is shown in Equation (8).

$$V_{max} = \frac{\dot{v}_o}{A_g} \qquad (8)$$

Calculation of Heat Transfer Coefficients

Correlations from Çengel[25] were used to determine the shell-side and tube-side heat transfer coefficients, and the overall heat transfer coefficient of the exchanger was then calculated. The Nusselt number for cross-flow over a tube bank is known to be related to the Reynolds number of the flow by the correlation shown in Equation (9):

$$Nu_{o,16} = 0.52Re_o^{0.5}Pr_o^{0.36}(\frac{Pr_o}{Pr_{o,s}})^{0.25} \quad (9)$$

Equation (9) is valid for Reynolds number between 100 and 1000. For flow over a tube bank, the Reynolds number is given as

$$Re_o = \frac{\rho_o V_{max} D_t}{\mu_o} \quad (10)$$

Equation (9) is applicable only for tube banks in which the number of tubes per vertical row is greater than 16, and this was not the case in this experiment. Therefore, a correction factor, shown in Equation (11), was included to determine the correct Nusselt number.

$$Nu_o = F_{bank} Nu_{o,16} \quad (11)$$

The shell-side heat transfer coefficient was calculated from the definition of the Nusselt number, noted in Equation (12).

$$Nu_o = \frac{h_o D_t}{k_o} \quad (12)$$

A similar correlation was used to find the Nusselt number on the tube side. For turbulent flow in tubes, the Nusselt number is related to the Reynolds number by the Dittus-Boelter equation:

$$Nu_i = 0.023Re_i^{0.8}Pr_i^{0.3} \quad (13)$$

The exponent on the Prandtl number was 0.3 for this experiment because the tube-side fluid was cooled. The Reynolds number for flow in a circular tube is given by Equation (14).

$$Re_i = \frac{\rho_i V_i D_t}{\mu_i} \quad (14)$$

The tube-side fluid velocity is dependent on the volumetric flow rate on the tube side and the cross-sectional tube flow area. The total cross-sectional flow area in the tubes is given by Equation (15).

$$A_{c,i} = \frac{\pi D_t^2}{4}(\frac{N_t}{N_p}) \quad (15)$$

The tube-side fluid velocity was calculated by dividing the tube-side volumetric flow rate by the flow area, as is shown in Equation (16).

$$V_i = \frac{\dot{v}_i}{A_{c,i}} \quad (16)$$

The tube-side heat transfer coefficient was then found by applying the definition of the Nusselt number,

$$Nu_i = \frac{h_i D_t}{k_i} \quad (17)$$

Because the tube thickness is small and the thermal conductivity of the tube material (copper) is high, the thermal resistance of the wall can be neglected. Thus, the inner and outer heat transfer areas can be assumed to be equal. With these assumptions, the overall heat transfer coefficient was calculated by Equation (18).

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} \quad (18)$$

Determination of Heat Exchanger Duty and Fluid Outlet Temperatures

The theoretical heat transfer rate in the heat exchanger was calculated by three methods. The calculated heat transfer rates were then set equal to conserve energy, resulting in a solvable system of simultaneous equations. The heat transfer rate across the tube surface between two fluids in a multipass or cross-flow heat exchanger is given by Equation (19).

$$Q_s = F_{corr} UA\Delta T_{lm} \quad (19)$$

The tube surface heat transfer area in Equation (19) was found from the equation

$$A = \pi D_t L N_t \tag{20}$$

The log mean temperature difference is the appropriate average temperature difference for the analysis of heat exchangers,[25] and is calculated by Equation (21).

$$\Delta T_{lm} = \frac{\Delta T_{max} - \Delta T_{min}}{\ln\left(\frac{\Delta T_{max}}{\Delta T_{min}}\right)} \tag{21}$$

The maximum temperature difference in the heat exchanger occurs at the shell inlet, and was calculated by the equation

$$\Delta T_{max} = T_{h,in} - T_{c,out} \tag{22}$$

The minimum temperature difference in the heat exchanger occurs at the shell outlet, and was calculated by the equation

$$\Delta T_{min} = T_{h,out} - T_{c,in} \tag{23}$$

The correction factor in Equation (19) is necessary for multipass exchangers and was determined from Figure 11-18(a) in the textbook by Çengel[25]. The correction factor is a function of the parameters P and R, shown in Equations (24) and (25).

$$P = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}} \tag{24}$$

$$R = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}} \tag{25}$$

The heat transfer rates to the tube- and shell-side fluids as they flowed through the reactor were also determined. The mass flow rate of the tube-side fluid was found using the continuity equation

$$\dot{m}_t = \rho_i V_i A_{ci} \tag{26}$$

The heat transfer rate to the tube-side fluid is given by Equation (27) as

$$Q_{f,i} = \dot{m}_t Cp_i (T_{h,in} - T_{h,out}) \tag{27}$$

Similarly, the mass flow rate of the shell side fluid, found from the continuity equation, is

$$\dot{m}_s = \rho_o V_{max} A_g \tag{28}$$

The heat transfer rate to the shell-side fluid is given by Equation (29) as

$$Q_{f,o} = \dot{m}_s Cp_o (T_{c,out} - T_{c,in}) \tag{29}$$

By the first law of thermodynamics, energy is conserved. Therefore, the heat transfer rates in the heat exchanger must all be equal, or

$$Q_s = Q_{f,i} = Q_{f,o} \tag{30}$$

This condition was imposed, and hot and cold fluid outlet temperatures were manipulated until the heat transfer rate calculated by each method was equal.

Evaluation of Heat Exchanger Performance

To evaluate heat exchanger performance, the conformity of the predicted theoretical heat transfer rate to the heat transfer rates calculated from experimental data was evaluated. The outlet tube- and shell-side fluid temperatures were recorded during the experiment. The experimental heat transfer rate to the shell-side fluid, using the mass flow rate as calculated in Equation (28), is given by Equation (31).

$$Q_{exp,o} = \dot{m}_s Cp_o (T_{c,out\_exp} - T_{c,out}) \tag{31}$$

Similarly, the experimental heat transfer rate to the tube-side fluid is given by Equation (32).

$$Q_{exp,i} = \dot{m}_t Cp_i (T_{h,in} - T_{h,out\_exp}) \tag{32}$$

The ratios of the experimental heat transfer rates to the theoretical heat transfer rate, found using Equation (19) were computed using the equations

$$ratio_i = \frac{Q_{exp,i}}{Q_s} \quad (33)$$
$$ratio_o = \frac{Q_{exp,o}}{Q_s} \quad (34)$$

The heat transfer rate in the exchanger was also calculated using the HTRI Xchanger Suite software package in the Rating Case Mode. The geometry of the shell-and-tube heat exchanger in the HTRI software package was set to mimic the geometry of the experimental heat exchanger. An NEN shell geometry with 3.2 cm shell inner diameter was chosen, with the tube pitch set at 4.72 mm and a tube layout angle of 30°. The tubes were selected to be thin-walled (0.3 mm wall thickness) copper. The baffle spacing was set at 20 mm, with a baffle cut of 25%. Water properties were calculated by the program on a component-by-component basis using the built-in property database. Mass flow rates and water inlet and outlet temperatures were entered, and heat exchanger duty was calculated. When the program was run with the tube length set at 14.3 cm, an error message indicated that the tubes were too short. The tube length had to be extended to 16 cm to obtain a solution, and the simulations were performed with this tube length. Therefore, the heat transfer rate for the exchanger calculated by HTRI was based on a slightly inflated heat transfer area. The ratio of the heat transfer rate calculated by HTRI to the theoretical heat transfer rate was found using Equation (35).

$$ratio_{HTRI} = \frac{Q_{HTRI}}{Q_s} \quad (35)$$

Reduced Results

Calculated heat transfer rates (the theoretical heat transfer across the tube surface using the ideal bank morphing method, Q_s; the heat transfer based on the experimental tube-side fluid temperature change, Q_{expi}; the heat transfer based on the experimental shell-side fluid temperature change, $Q_{exp,o}$; and the calculated heat transfer in the exchanger using HTRI, Q_{HTRI}) and the ratios of the heat transfer rates to the theoretical heat transfer rates ($r_i = Q_{exp,i}/Q_s$, $r_o = Q_{exp,o}/Q_s$, and $r_{HTRI} = Q_{HTRI}/Q_s$) are shown for each run in Table 2. The theoretical heat transfer rate was within 20% of the experimental heat transfer rates for each experimental run, indicating that the morphing method was valid for the analysis of the exchanger. The theoretical analysis overpredicted the heat transfer based on the tube-side fluid analysis, but underpredicted the heat transfer based on the shell-side analysis. Ideally, these heat transfer rates should be equal because the heat transferred to the cold fluid must equal the heat transferred from the hot fluid to conserve energy. The heat transfer rates calculated using HTRI compared well with the theoretical and experimental heat transfer predictions. The ideal tube bank morphing method slightly overpredicted the heat transfer, giving heat transfer ratios ranging from 0.92 to 0.98. Overdesign was expected because the theoretical treatment does not account for bypass around the tube bank or parallel flow in some sections of the tube bank. Bypass likely occurred because sealing strips were not used in the exchanger assembly.

Table 2. Comparison of Calculated Heat Transfer Rates

Run	Heat Transfer Rates, (W)				Ratio of Heat Transfer Rates		
	Q_s	$Q_{exp,i}$	$Q_{exp,o}$	Q_{HTRI}	$Q_{exp,i}/Q_s$	$Q_{exp,o}/Q_s$	Q_{HTRI}/Q_s
1	674	539	745	642	0.799	1.11	0.95
2	882	710	923	816	0.805	1.05	0.93

3	979	799	1068	933	0.817	1.09	0.95
4	856	699	917	808	0.817	1.07	0.94
5	964	831	1068	949	0.862	1.11	0.98
6	701	613	681	647	0.874	0.97	0.92
7	809	691	818	754	0.853	1.01	0.93
8	828	710	823	766	0.856	1.00	0.93
9	837	723	827	775	0.864	0.99	0.93
10	830	710	880	795	0.856	1.06	0.96

Required DLM Modifications/Further Recommendations

Although the DLM with shell and tube heat exchanger module worked well and gave heat transfer rates which were comparable with theoretical predictions, a few modifications in the unit are required to yield satisfactory data:

- The disagreement between tube-side and shell-side duties suggests that there were either errors in the flow rate or temperature measurements, or there were heat losses between the exchanger and thermocouples. The test cartridge was not insulated, introducing the possibility of heat losses. It is recommended that a longer equilibration time be allowed once a flow rate is changed so that the heat exchanger can reach steady-state and water outlet temperatures can be measured accurately.
- The cold water temperature in the shell-side reservoir tank was measured with an electronic thermometer that was not an original part of the DLM. It was noted that the temperature reading from the thermometer was approximately 10°C greater than the displayed inlet cold fluid temperature. Additionally, it was observed that the thermocouple did not function properly if the nominal shell-side fluid temperature was less than 15°C. Therefore, it is possible that this thermocouple was not properly calibrated and introduced errors in the temperature measurement, and it is recommended that the thermocouple be replaced.
- Recycle of the heated shell-side stream back to the reservoir quickly increased the cold water temperature, and additional ice had to be added to maintain the temperature at about 25°C. The temperature went through periodical cooling and heating stages, and a consistent temperature was really never achieved. This, along with inadequate mixing in the tank, made it difficult for the exchanger to reach steady state on each run, and may be an additional cause of the discrepancy between the duty calculated on the basis of the shell-side and tube-side fluids. It is recommended that the recycle stream to the shell-side reservoir tank be eliminated. Instead, the shell-side drain hose should be run to a drain. The tube-side recycle stream need not be changed because the thermostat on the electric heater ensures that the hot water is kept at a nearly constant temperature.

Nomenclature

A	Tube surface heat transfer area, m^2
A_{ci}	Cross-sectional area for tube flow, m^2
A_g	Ideal tube bank gap area (minimum flow area), m^2
A_{shell}	Cross-sectional area of shell, m^2
A_{square}	Cross-sectional area of equivalent square ideal tube bank, m^2

B	Baffle spacing, m
Cp_i	Specific heat of tube-side fluid, J/kg·K
Cp_o	Specific heat of shell-side fluid, J/kg·K
D_{shell}	Shell bundle diameter, m
D_t	Tube diameter, m
F_{bank}	Correction factor for flow over tube bank with N_{tr}<16
F_{corr}	Multipass heat exchanger correction factor
G	Gap distance in ideal tube bank, m
h_i	Tube-side heat transfer coefficient, W/m^2·K
h_o	Shell-side heat transfer coefficient, W/m^2·K
k_i	Tube-side fluid thermal conductivity, W/m·K
k_o	Shell-side fluid thermal conductivity, W/m·K
L	Tube length, m
$\dot{m}_s$	Mass flow rate of fluid through shell, kg/s
$\dot{m}_t$	Mass flow rate of fluid through tubes, kg/s
N_p	Number of tube passes
N_t	Number of tubes in exchanger
N_{tr}	Number of tubes in a vertical row of the ideal tube bank
Nu_i	Tube-side Nusselt number
Nu_o	Shell-side Nusselt number
$Nu_{o,16}$	Shell-side Nusselt number for N_{tr}>16 (before correction)
P	Temperature ratio; $P = (T_{c,out} - T_{c,in})/(T_{h,in} - T_{c,in})$
Pr_i	Prandtl number of tube-side fluid
Pr_o	Prandtl number of shell-side fluid
$Pr_{o,s}$	Prandtl number of shell-side fluid at average tube-side fluid temperature
$Q_{exp,i}$	Heat transfer on basis of experimental tube-side fluid temperature change, W
$Q_{exp,o}$	Heat transfer on basis of experimental shell-side fluid temperature change, W
$Q_{f,i}$	Flow heat transfer to tube-side fluid, W
$Q_{f,o}$	Flow heat transfer to shell-side fluid, W
Q_{HTRI}	Heat transfer in exchanger calculated using HTRI program, W
Q_s	Heat transfer across tube surface, W
$ratio$	(shell-side volumetric flowrate)/(tube-side volumetric flowrate)
$ratio_i$	Ratio of experimental to theoretical heat transfer, tube-side basis; $ratio_i = Q_{exp,i}/Q_s$
$ratio_o$	Ratio of experimental to theoretical heat transfer, shell-side basis; $ratio_o=Q_{exp,o}/Q_s$
$ratio_{HTRI}$	Ratio of heat transfer by HTRI to theoretical heat transfer; $ratio_{HTRI} = Q_{HTRI}/Q_s$
R	Temperature ratio; $R = (T_{h,in} - T_{h,out})/(T_{c,out} - T_{c,in})$
Re_i	Reynolds number for flow in tubes
Re_o	Reynolds number for flow in shell
S_t	Tube-to-tube spacing in ideal tube bank, m
$T_{c,in}$	Cold fluid inlet temperature, °C
$T_{c,out}$	Theoretical cold fluid outlet temperature, °C
T_{c,out_exp}	Experimental cold fluid outlet temperature, °C
$T_{h,in}$	Hot fluid inlet temperature, °C
$T_{h,out}$	Theoretical hot fluid outlet temperature, °C
T_{h,out_exp}	Experimental hot fluid outlet temperature, °C
U	Overall heat transfer coefficient, W/m^2·K

V_i	Water velocity through tubes, m/s
V_{max}	Maximum velocity of water through tube bank, m/s
W	Side length of equivalent ideal square tube bank, m
ΔT_{lm}	Log mean temperature difference, °C
ΔT_{max}	Temperature difference at shell-side inlet, °C; $\Delta T_{max} = T_{h,in} - T_{c,out}$
ΔT_{min}	Temperature difference at shell-side outlet, °C; $\Delta T_{min} = T_{h,out} - T_{c,in}$
μ_i	Dynamic viscosity of tube-side fluid, kg/m·s
μ_o	Dynamic viscosity of shell-side fluid, kg/m·s
ρ_i	Density of tube-side fluid, kg/m^3
ρ_o	Density of shell-side fluid, kg/m^3

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