COMPACT FEEDWATER HEATERS

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Abstract

With few exceptions, heat exchangers found in existing power plants are conventional shell-and-tube heat exchangers. Compact heat exchanger technologies can offer several advantages over conventional designs, including reduced weight, cost and space. When components occupy less space, reactor buildings can be smaller, resulting in further cost savings. This paper summarizes the results of a sizing and costing study for a condensate heater using conventional shell-and-tube, plate-and-shell and minichannel heat exchanger technologies. A parametric analysis was conducted using a range of principal dimensions and Nusselt-type heat transfer correlations to decide the heat transfer length or number of plates or tubes and reasonable constraints were placed on internal flow velocities and pressure drops. Net Present Costs were estimated using the weight of steel required for the design, assuming manufacturing costs scale with weight. It was found that compact technologies can reduce the weight of the required heat exchanger by a factor of 3, reduce the cost by a factor of 2.4, and reduce the space occupied by a factor of 8. With deployment to remote locations and a higher premium placed on floor space, it is postulated that small reactors serve to benefit most from compact heat exchanger technologies.

1. Introduction

Heat exchangers play an integral role in any heat engine. At the most fundamental level, every heat engine must transfer heat to a working fluid, and from there to a heat sink, converting some of this energy to a more useful form. The Rankine cycles employed by most large-scale power reactors often include large steam generators, reheaters, condensers, feedwater heaters, as well as multitudinous heat exchangers forming the cooling water system(s), all the way down to small sample coolers. With few exceptions, designers have chosen to use shell-and-tube technology in these applications. The simplified Brayton cycles of some small reactors use one or two gas-to-gas preheaters and a gas-to-water cooler. Cognizant of the poor thermal conductivity of gases, designers of these reactors have begun to lean toward compact heat exchanger technologies.

Despite the weight, cost and footprint advantages, a lack of familiarity with modern compact heat exchanger technology in the nuclear industry poses risks to its adoption; as there are uncertainties about the mechanisms of failure, means of inspection, costs of maintenance, and suitability as a radiological barrier. These risks may have slowed the introduction of modern compact heat exchanger technology to established large-scale reactor designs, where the benefits are further diluted by economies of scale. The effort needed to gain familiarity and quantify the risks of adoption weighs more favourably for small reactor designs, which serve to benefit most from the technology because the scale of their construction and their deployment to remote locations places a premium on floor space and the cost of freight. Quantifying the potential economic benefit is an important first step.

This paper summarizes the results of a sizing and costing study for a condensate heater using conventional shell-and-tube, plate-and-shell and compact minichannel heat exchanger technologies. A parametric analysis was conducted using a range of principal dimensions and Nusselt-type heat transfer correlations to decide the heat transfer length or number of plates, and reasonable constraints were placed on internal flow velocities and pressure drops. Net Present Costs were evaluated using the weight of steel required, assuming manufacturing costs scale with weight.

1.1 Shell-and-tube Heat Exchangers Explained

Undergraduate courses in heat transfer describe the design and function of shell-and-tube heat exchangers in some detail. Heat is transferred from a higher temperature fluid to a lower temperature fluid with one fluid flowing through tubes contained within a shell and the other flowing over the outside of the tubes within the shell itself. Baffles are often employed on the shell side to increase the number of passes the external fluid takes past the tubes, and header divisions are often used to increase the number of passes on the tube side. Each tube is fixed to the tube sheet, which puts a practicable limit on the number of tubes and the tube spacing.

Figure 1 shows a simplified analysis of a 1000 W heat exchanger that raises the temperature of water from 300 K to 301 K across a constant temperature difference of 1 K. Figure 1(a) shows two clear trends: the mass of tubing required decreases with decreasing tube diameter and decreases with Reynolds number in the turbulent region. Figure 1(b) shows that the penalty for reduced mass is a higher pressure drop. (Note that Figure 1(b) is on its side.) If pressure drop is a limiting design criterion, then large diameter tubes are chosen in conventional shell-and-tube designs. The alternative is a large number of very small and short tubes, which would result in a compact shell-and-tube design.

1.2 Plate Heat Exchangers Explained

Instead of tubes, the stacking of grooved plates forms the basic design of many compact heat exchangers, including the plate-and-shell and printed circuit heat exchangers. By adding plates to the stack, not only does the heat transfer surface area increase, but the flow cross-sectional area increases as well, reducing the Reynolds number. In the printed circuit design, the channels are discreet, etched grooves with no cross-talk between channels (Figure 2(a)). In the plate-and-shell design, the corrugations on neighbouring plates cross each other, and there is significant cross-talk between corrugations (Figure 2(b)). Wavy designs are often incorporated in the plates to enhance boundary layer disturbance, improving local mixing and allowing a smooth transition between laminar and turbulent flow regimes.

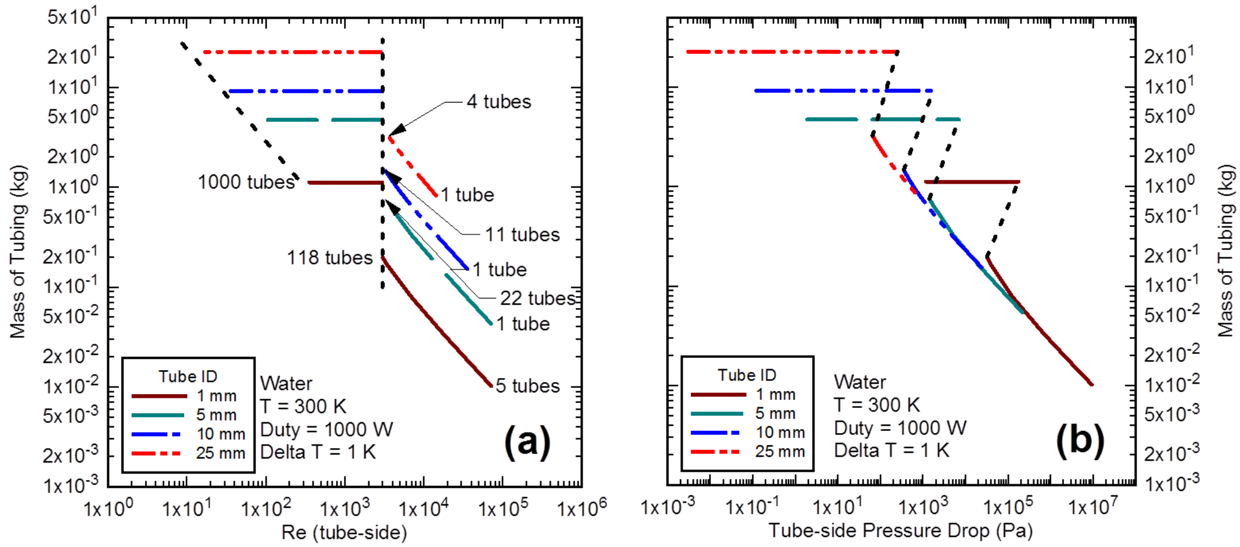

Figure 1 (a) Calculated mass of tubing and (b) tube-side pressure drop for an example 1000 W heat exchanger.

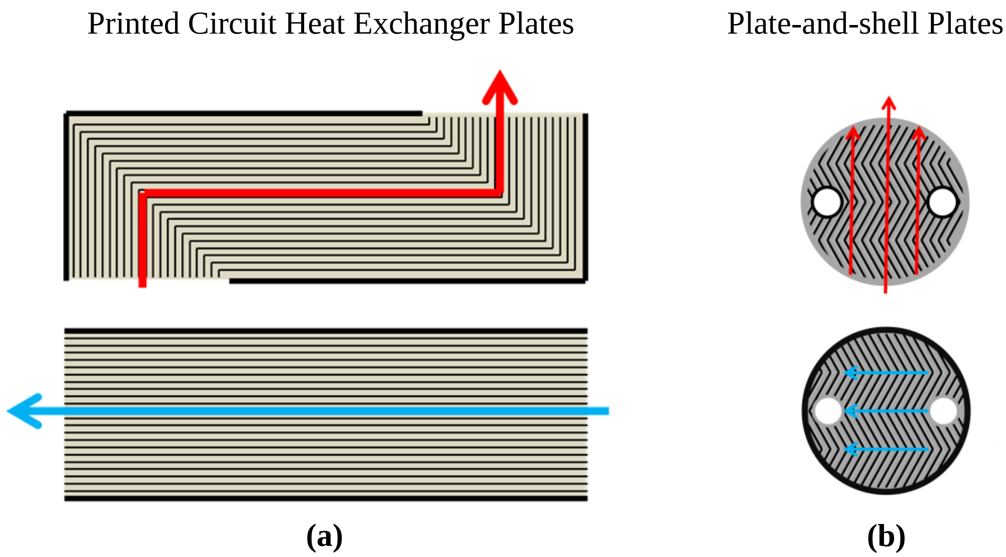

Figure 2 (a) Examples of printed circuit heat exchanger plates (counter-cross-flow), and (b) examples of plate-and-shell plates (cross-flow). In printed circuit heat exchangers, each plate has a flat face and a grooved face, and adjacent plates are different. In Plate-and-shell heat exchangers, each plate is the same and the seals allow flow distribution from either the shell (top) or the ports (bottom).

2. Design Conditions and Design Criteria

A condensate heater example was chosen, just upstream of a deaerator, with one extraction steam inlet and no cascading drains to simplify the analysis. Table 1 gives the design condition details, with the total duty divided into conventional zones of desuperheater, condenser and drains cooler. Figure 3 provides a T-q diagram of these conditions.

Table 1
Design Conditions of Condensate Heater

Hot Fluid Mass Flow Rate (kg/s)				57.4				
Section	**Description**	**$T_{hot\ in}$ (ºC)**	**$P_{hot\ in}$ (MPa)**	**$H_{hot\ in}$ (kJ/kg)**	**$T_{hot\ out}$ (ºC)**	**$P_{hot\ out}$ (MPa)**	**$H_{hot\ out}$ (kJ/kg)**	**Duty (MW)**
1	Desuperheater	179.4	0.2402	2826.8	125.1	0.233	2603.8	12.8
2	Condenser	125.1	0.233	2603.8	125.1	0.233	634.9	113.0
3	Drains Cooler	125.1	0.233	634.9	92.3	0.233	386.8	14.24
Cold Fluid Mass Flow Rate (kg/s)				927.6				
Section	**Description**	**$T_{cold\ in}$ (ºC)**	**$P_{cold\ in}$ (MPa)**	**$H_{cold\ in}$ (kJ/kg)**	**$T_{cold\ out}$ (ºC)**	**$P_{cold\ out}$ (MPa)**	**$H_{cold\ out}$ (kJ/kg)**	**Duty (MW)**
1	Desuperheater	119.8	0.550	503.2	123.0	0.550	517.0	12.8
2	Condenser	91.0	0.550	381.3	119.8	0.550	503.2	113.0
3	Drains Cooler	87.3	0.550	366.0	91.0	0.550	381.3	14.24

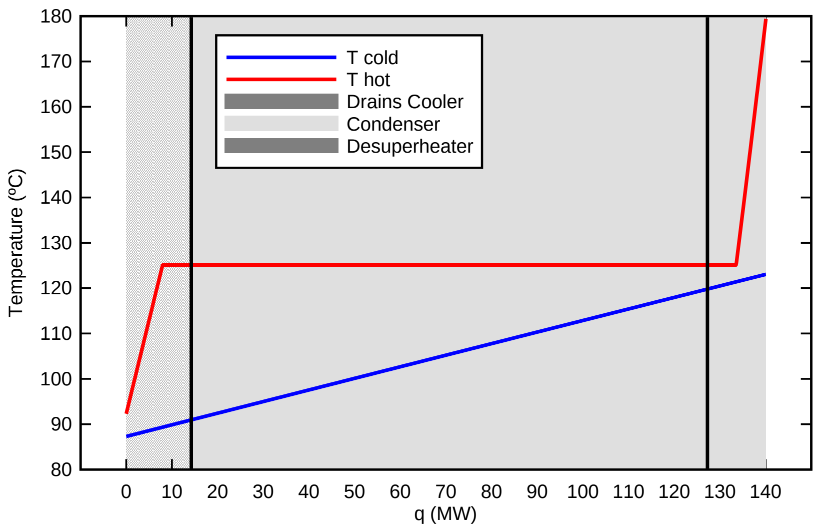

Figure 3 T-q diagram of condensate heater used in the analysis.

One design criterion used was to limit the cold fluid pressure drop to 10% of the operating pressure, ΔP = 55 kPa. Although somewhat arbitrary, the pressure drop affects the pressure of the deaerator, and to allow a much higher pressure drop could significantly alter the steam flow and heat balance in the event of a transient change in flow rate, leading to instabilities.

Another design criterion used was to limit the design velocities to V_g = 30 m/s for steam flow and V_l = 2.5 m/s for condensate flow. These were simply rough numbers based on rules of thumb [1] used to limit noise generation and flow erosion. No attempt was made to characterise the vibration or accelerated corrosion that might result from higher velocities.

The last design criterion was to ensure the designs meet ASME Boiler and Pressure Vessel Code Section III (Rules for Construction of Nuclear Facility Components) [2]. A minimum shell thickness of 6.35 mm and a minimum tube thickness of 0.71 mm were assumed and were thicker than required by code. A minimum plate thickness, t_p, of 0.5 mm was assumed with plate-and-shell plate thickness evaluated by Eqn (1), based on Roark's equation for a parallelogram plate with two free edges and two simply supported edges [3]. Printed circuit plate thickness, t_f, was evaluated by Eqn (2), as recommended by Hesselgreaves [4] (where λ is the corrugation wavelength, d_{mc} is the minichannel diameter, P is the pressure and S is the maximum allowable stress for 316L stainless steel under the design conditions). The minimum plate thickness was thicker than required by Eqn (2).

$$t_P = \lambda \cdot \sqrt{3.05 \frac{P}{S}} \tag{1}$$

$$t_f = \frac{P \cdot d_{mc}}{S - 0.6P} \tag{2}$$

3. Calculation Methods

3.1 Software and Formulations

The calculations were performed in Mathcad Prime 2.0 and used the IAPWS Industrial Formulation 1997 for the Thermodynamic Properties of Water and Steam and related formulations for viscosity and thermal conductivity [5-7].

3.2 Solution Approach to the Heat Transfer Problem

The Log Mean Temperature Difference Method was used to calculate required heat transfer area in the desuperheater and drains cooler sections of the heat exchanger. The condenser section of the heat exchanger was divided into 10 stages (with each stage assumed to be at the same temperature, pressure and flow), and convection coefficients were evaluated based on average cold fluid temperature and average hot fluid quality. Surface temperatures were found by applying a golden-ratio iterative search algorithm, rather than a Mathcad solve block. The surface temperatures were used to calculate the required heat transfer length.

3.3 Convection Coefficients

Cold fluid convection coefficients, h_t, were calculated using the Gnielinski correlation [8] for tube flow and printed circuit minichannel flow, Eqn (3), where k is the fluid thermal conductivity, D_t is the tube diameter, t_t is the tube thickness, $\mathrm{Nu}_{laminar}$ is 3.66 for a circular tube and 2.96 for a semicircular channel [4] (since the heat transfer problem is neither constant temperature nor constant heat flux, the constant temperature estimation for Nusselt number was used), f_F is the Fanning friction factor, Re_D is the Reynolds number, and Pr is the Prandtl number. The Martin correlation [9] was used for convective heat transfer coefficient in the plate-and-shell design, Eqn (4), where D_h is the hydraulic diameter, f_M is a unique friction factor described by Martin for plate-and-frame type heat exchangers, β is the chevron angle (30º was used), μ is the bulk viscosity, and μ_s is the viscosity at the surface temperature.

$$h_t = \begin{cases} \dfrac{k}{D_t - 2t_t}(3.66) & \text{Re} \leq 3000 \\ \dfrac{k}{D_t - 2t_t}\left(\dfrac{\left(\dfrac{f_F}{2}\right)(\text{Re}_D - 1000)\text{Pr}}{1 + 12.7\left(\dfrac{f_F}{2}\right)^{0.5}\left(\text{Pr}^{\frac{2}{3}} - 1\right)}\right) & \text{Re} > 3000 \end{cases} \tag{3}$$

$$h_{Martin} = \frac{k}{D_h} \cdot 0.205 \cdot \text{Pr}^{\frac{1}{3}} \cdot \left(f_M \cdot \text{Re}_D^2 \cdot \sin(2\beta) \right)^{0.374} \left(\frac{\mu}{\mu_s} \right)^{\frac{1}{6}} \tag{4}$$

Hot fluid convection coefficients, h_S, in the single-phase sections of the desuperheater and drains cooler were calculated using a correlation recommended by Lee [10] for shell-side flow, Eqn (5), where D_e is the equivalent diameter. For plate-and-shell, the Martin correlation was again used, Eqn (4). For printed circuit, the Gnielinski correlation was again used, Eqn (3).

$$h_S = \frac{k}{D_e} \cdot 0.36 \cdot \text{Re}_D^{0.55}\, \text{Pr}^{\frac{1}{3}} \cdot \left(\frac{\mu}{\mu_s} \right)^{0.14} \tag{5}$$

Hot fluid convection coefficients in the two-phase condenser section, h_{cond}, were calculated using the laminar film condensation correlation of Dhir and Lienhard [11] for shell-side flow, Eqn (6), where g is the acceleration due to gravity, ρ is the density, h'_{lg} is a corrected heat of vaporization, N_{col} is the number of columns of tubes, T is the temperature, subscripts *sat* and *s* refer to saturation and surface conditions, and subscripts l and g refer to liquid and vapour states. Due to the relative complexity of calculating condensation of water in plate-and-shell or printed circuit heat exchangers, a shear factor multiplier, χ_{shear}, was applied to the single-phase correlations, following the approach described by Taylor [12], Eqn (7), where x_g is the vapour quality.

$$h_{cond} = 0.729 \cdot \left(\frac{g \cdot \rho_l (\rho_l - \rho_g) k_l^3 h'_{\text{lg}}}{N_{col} \cdot \mu_l \cdot (T_{sat} - T_s) D_t} \right)^{0.25} \tag{6}$$

$$\chi_{shear} = \left(1 + x_g \cdot \left(\frac{\rho_l}{\rho_g} - 1 \right) \right)^{0.5} \tag{7}$$

3.4 Cold Fluid Pressure Drop

Cold fluid pressure drops were calculated for each heat exchanger section using the calculated heat transfer length. The Martin friction factor was used for the plate-and-frame design. For the shell-and-tube and printed circuit heat exchangers, the Fanning friction factor was calculated using the appropriate laminar relations, Eqn (8) top, or the Petukhov correlation [13], Eqn (8) bottom.

$$f_F = \begin{cases} \dfrac{16}{\mathrm{Re}_D}(circle) \text{ or } \dfrac{15.78}{\mathrm{Re}_D}(semicircle) & \mathrm{Re}_D \le 2300 \\ (1.58 \cdot \ln(\mathrm{Re}_D) - 3.28)^{-2} & \mathrm{Re}_D > 2300 \end{cases} \tag{8}$$

3.5 Operating Costs

The present value of the operating costs were estimated from the heat exchanger pressure drop and the power consumed by a pump operating at 85% efficiency over a 60-year life. The off-peak energy cost of 7.2¢/kWh from 2014 March was used.

3.6 Capital Costs

The capital costs were estimated from the weight of the heat exchanger, assuming stainless steel construction. The model described by Turton et al. [14] for the bare module cost of a u-tube shell-and-tube heat exchanger was used as the normative model for costing. This model was converted from cost in terms of surface area to cost in terms of weight, and indexed to inflation. Implicit in the use of this model is the assumption that the cost of manufacturing scales with weight regardless of the heat exchanger technology.

3.7 List of Assumptions and Simplifications

This parametric study made many assumptions in the details of the design of the three heat exchanger types that may not apply to a more detailed and sophisticated design. Some caveats:

- The maximum flow velocity criteria are assumed to be adequate and limiting for the designs. A practical designer might alter the flow direction or permit higher velocities based on experience or detailed calculations.
- Straight flow paths have been assumed for the printed circuit design. The use of wavy flow patterns could further improve performance and reduce cost.
- Specific correlations for shell-and-tube and plate-and-shell heat exchangers are expected to be reasonably accurate (±20%).
- Adapted correlations were used for the condenser region of a plate-and-shell and for the entire printed circuit heat exchanger. While these are thought to be adequate, their accuracy has not been assessed.
- The use of only 10 stages in the condenser regions is a simplification that will lead to inaccuracies in the calculated results. Hesselgreaves [4] recommends 100 stages for more detailed design.

4. Results and Discussion

4.1 Shell-and-tube

Shell diameters (ID) of 1 m to 3 m were evaluated containing tubes with diameters (OD) ranging from 3.175 mm (1/8") to 25.4 mm (1"). Several combinations had pressure drops exceeding the design limit of 55 kPa. In these cases, the flow was divided among several identical heat exchangers to remain within the desired pressure drop. At the design limit of 55 kPa, the present value of the operating costs was only $400,000, which had little bearing on the net present value, which was $5,000,000 at a minimum. The calculated heat transfer length (multiplied by the number of heat exchangers) is shown in Figure 4, along with the net present value. The analysis reveals that for the scenario proposed, the lowest cost units contain a large number of small diameter tubes (Table 2, left); that is, they are compact. The results were filtered to those with less than 20 000 tubes (which is still a large number of tubes) to obtain designs that might be considered representative of conventional shell-and-tube designs (Table 2, right).

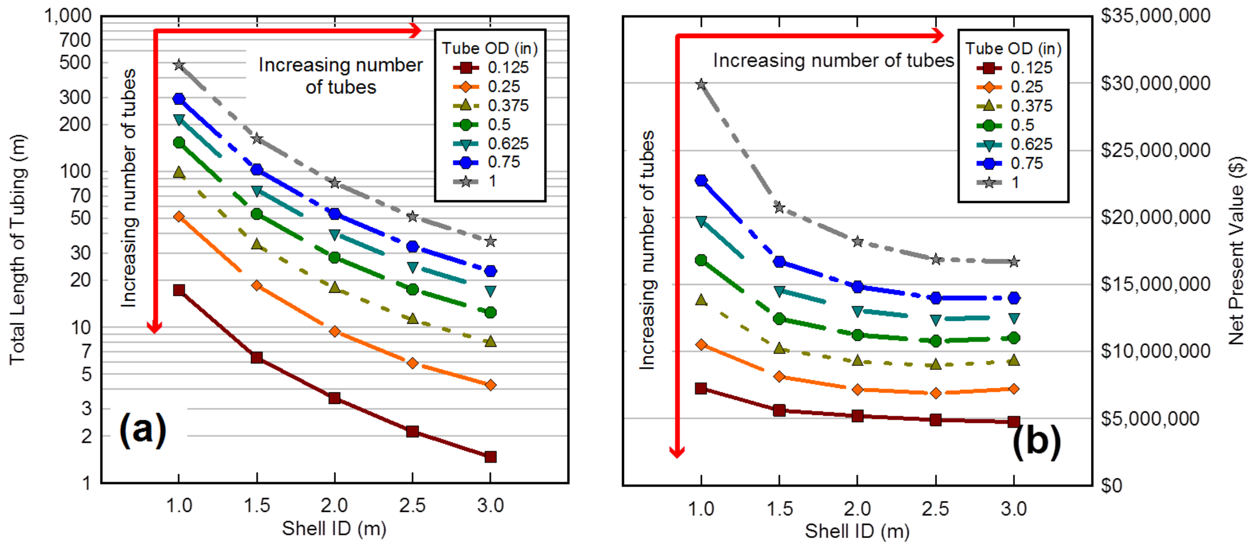

Figure 4 (a) Total heat transfer length, and (b) Net Present Value vs Shell ID for Shell-and-Tube heat exchangers.

Table 2
Summary of Shell-and-Tube Heat Exchanger Results

Option	1 Compact	2 Compact	3 Conventional	4 Conventional
Number of Heat Exchangers	2	3	2	3
Shell ID (m)	3.0	2.5	1.5	2.0
Tubing Length per HX (m)	0.73	0.71	8.87	11.19
Number of Tubes per HX	323 873	224 912	15 994	8 996
Tube OD (in.)	0.125	0.125	0.375	0.375
Weight per HX (kg) Total Weight (kg)	12 100 24 200	8 200 24 600	24 400 48 800	17 600 52 800
Total Cost	$4 750 000	$4 910 000	$9 240 000	$10 170 000
Single HX Floor Space (m^2) Total Floor Space (m^2)	2.2 4.4	1.8 5.3	17.7 35.5	16.8 50.4

4.2 Plate-and-shell

Shell diameters (ID) of 1 m to 3 m were evaluated containing plates with corrugation amplitudes between 1 and 3 mm. The port diameter to plate diameter ratio was limited to 1/8. Plate diameters were 0.94x the shell diameter as a result. The number of identical heat exchangers required to produce reasonable flow velocities in the shell and ports was quite high; flow was divided among thirty-six 1 m (ID) heat exchangers, sixteen 1.5 m (ID), nine 2 m (ID), six 2.5 m (ID) or four 3.0 m (ID). Although the number of heat exchangers may seem high, a bank of a large number of small heat exchangers may provide a means of segregating flow in the event of a failure and add to the safety rating of the system.

For some combinations, the number of plates required to meet the pressure drop requirement exceeded the number of plates required to meet the heat transfer area requirement. These were also the most cost-effective designs. The calculated number of plates (multiplied by the number of heat exchangers) is shown in Figure 5, along with the net present value. The analysis reveals a somewhat parabolic relationship between shell diameter and cost. It is worth noting that the largest shell diameter has the smallest range of costs among the evaluated corrugation amplitudes. Three options have very nearly the same cost, as shown in Table 3. Although these designs cost around 14% more than their shell-and-tube rivals, they could be arranged vertically to occupy less floor space.

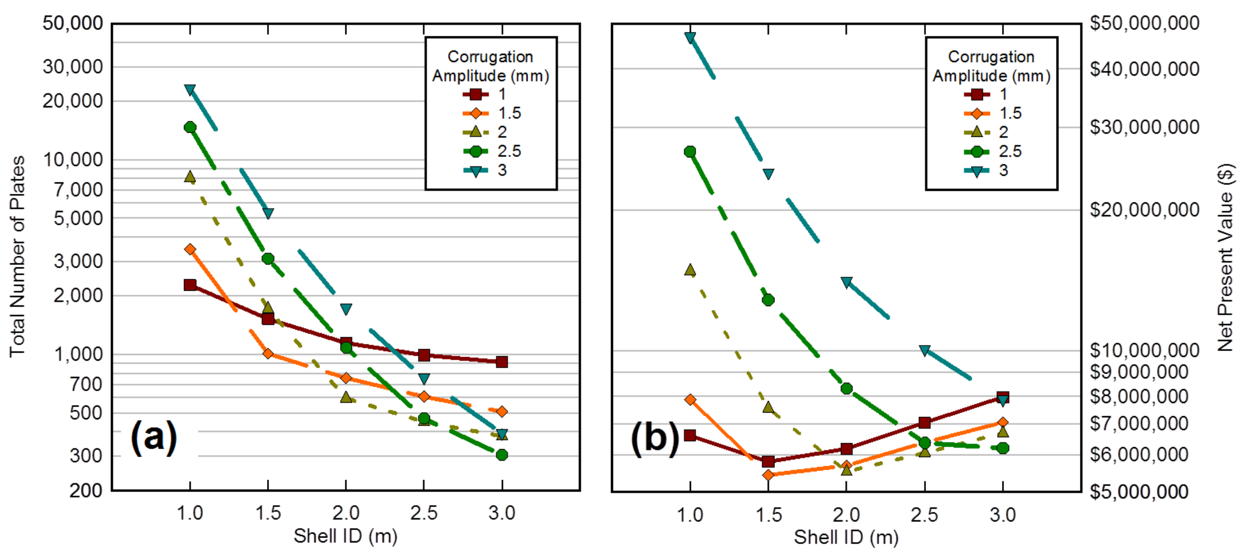

Figure 5 (a) Total number of plates and (b) Net Present Value vs Shell ID for Plate-and-Shell heat exchangers.

Table 3
Summary of Plate-and-Shell Heat Exchanger Results

Option	1	2	3
Number of Heat Exchangers	16	9	9
Shell ID (m)	1.5	2.0	2.0
Length of Plate Stack (m)	0.19	0.27	0.26
Number of Plates per HX	63	66	84
Corrugation Amplitude (mm)	1.5	2.0	1.5
Weight (kg) Total Weight (kg)	1 100 17 500	2 400 21 900	2 400 21 200
Total Cost	$5 430 000	$5 530 000	$5 690 000
Single HX Floor Space (m^2) Total Floor Space (m^2)	0.29 4.6	0.54 4.8	0.51 4.6

4.3 Printed Circuit Heat Exchangers

Plate dimensions of 0.4 to 1.6 m (H) x 0.6 m (W) x 0.5 mm (T) were evaluated with minichannel diameters ranging from 0.5 to 3 mm. The flow was assumed to be distributed among two identical heat exchangers. A staggering number of plates were needed to limit the flow velocities to reasonable values; between 2712 and 27816, depending on the hot minichannel diameter (more plates were needed at smaller diameters). Except at short plate lengths and large hot minichannel diameter, the number of plates did not change.

The net present value of printed circuit heat exchangers is shown in Figure 6 as a function of minichannel diameter and length of plate. A general trend observed is that the required number of plates and the ultimate cost of a unit shows greater variation for a plate length of 0.4 m than for a plate length of 0.8 m. In fact, the number of plates needed for the 0.4 m (L) plates were determined mostly by the heat transfer area needs, whereas the number of plates needed for the 0.8 m (L) plates were determined mostly by the flow velocity criteria. This suggests that the longer plates may actually be more cost effective if the flow velocities in the distribution headers could increase without detriment to safety or component life. Thus, while the most cost-effective design was found for the shortest plate evaluated, Table 4, this might be improved for a longer plate under different design criteria.

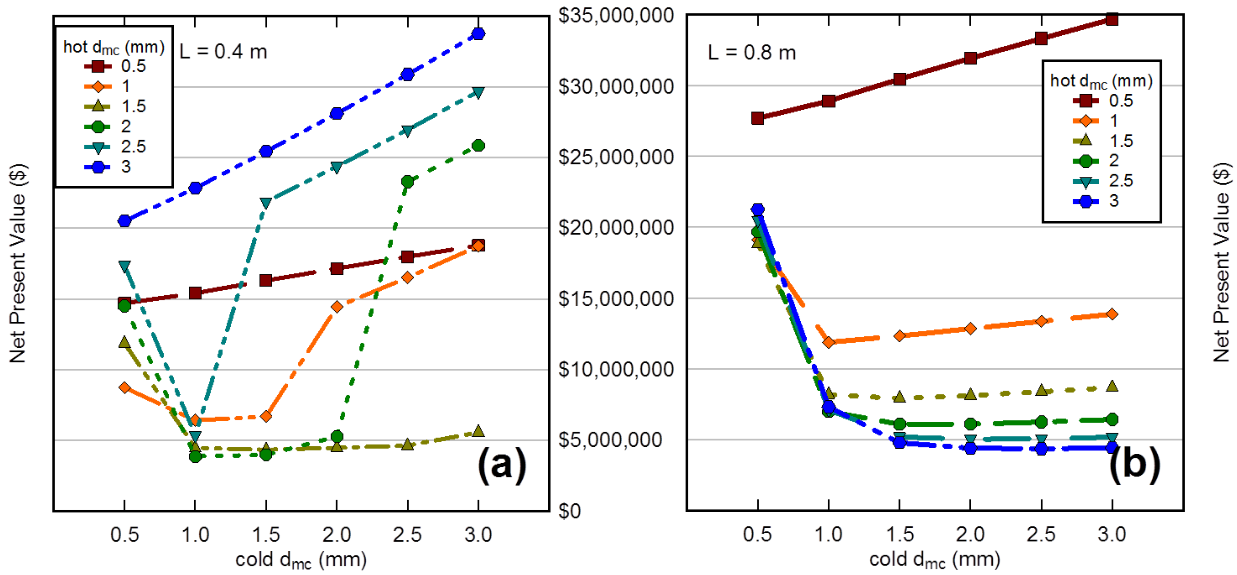

Figure 6 Net Present Value vs minichannel diameter (cold fluid) for Printed Circuit Heat Exchangers. (a) Plate length of 0.4 m. (b) Plate length of 0.8 m.

Table 4
Summary of Printed Circuit Heat Exchanger Results

Option	1	2	3
Number of Heat Exchangers	2	2	2
Plate Width (m)	0.6	0.6	0.6
Plate Height (m)	0.4	0.4	0.8
Length of Plate Stack (m)	5.5	7.3	4.7
Number of Plates per HX	4424	5314	2712
Hot/Cold Minichannel Diameter (mm)	2.0 / 1.0	2.0 / 1.5	3.0 / 2.0
Weight (kg) Total Weight (kg)	8 000 16 000	10 100 20 200	12 700 25 400
Total Cost	$3 880 000	$3 990 000	$4 410 000
Single HX Floor Space (m^2) Total Floor Space (m^2)	2.2 4.4	2.9 5.8	2.8 5.6

5. Summary and Conclusions

A parametric study was conducted on the size and cost of a condensate heater using three heat exchanger technologies: shell-and-tube, plate-and-shell and printed circuit heat exchangers. The most cost-effective designs are summarized below:

- Conventional shell-and-tube: two 1.5 m (ID) x 8.87 m (L) shells with 15 994 x 3/8" tubes each at $9.24M, weighing 48 800 kg total, occupying 35.5 m^2 of floor space.
- Compact shell-and-tube: two 3 m (ID) x 0.73 m (L) shells with 323 873 x 1/8" tubes each at $4.75M, weighing 24 200 kg total, occupying 4.4 m^2 of floor space.

- Plate-and-shell: sixteen 1.5 m (ID) x 0.19 m (L) x 1.5 mm (amp) at $5.43M, weighing 17 500 kg total, occupying 4.6 m^2 of floor space.
- Printed circuit heat exchanger: two 0.4 m (W) x 0.6 m (H) x 5.5 m (L) using 1 mm cold channels and 2 mm hot channels at $3.88M weighing 16 000 kg, occupying 4.4 m^2 floor space on its side or 0.5 m^2 stood vertical.

The results highlight the cost savings possible with compact technologies (up to 2.4x), as well as the reduction in mass (up to 3x) and occupied floor space (a factor 8 or greater). For the situation analyzed in this study, compact heat exchanger technologies present an opportunity to reduce the footprint and overall building size for a small modular reactor, as well as the cost of freight. It is suggested from the similarity of the results among compact designs, however, that all design options be considered through a more detailed analysis at the outset of a new project as different design constraints could result in a different heat exchanger style being chosen.

The ultimate selection of a technology type would need to factor in the risks of adoption as well as the benefits. Quantifying the risks requires a certain familiarity with the technology, a knowledge of the mechanisms of failure and their rates, the ability to predict failure through inspection, and the associated costs of maintenance, inspection, repair and replacement. There may also be nuclear safety risks and benefits to consider, certainly where a heat exchanger is used as a radiological barrier or redundancy of multiple exchangers can be a mitigating safety factor, but also indirectly through effects on process dynamics caused by reduced residence times.

6. References

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