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Design Considerations for Compact Heat Exchangers

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Abstract – *The next generation of nuclear power plants demand highly-effective, high integrity, compact heat exchangers capable of meeting the mechanical challenges posed by the need for improved cycle efficiencies. Heatric offers a range of diffusion bonded compact heat exchangers to meet these challenges, each custom-designed and engineered by Heatric's dedicated Nuclear Engineering Department.*

Heatric currently offers three heat exchanger types: Printed Circuit Heat Exchangers (PCHEs); Formed Plate Heat Exchangers (FPHEs); and Hybrid Heat Exchangers (H^2Xs). The thermal-hydraulic design, selection, and optimisation of these heat exchangers must consider a wide range of factors, including process constraints, cost, size, and mechanical considerations. Each heat exchanger is a bespoke product, designed by iteration and consultation with customers to provide optimal cost and performance (i.e. pressure drop/effectiveness).

This paper discusses the design and surface enhancement considerations that lead to optimal heat exchanger designs. In particular, heat transfer versus pressure drop performance considerations for enhanced surfaces will be discussed, with reference made to other surface types. The versatility of Heatric's manufacturing techniques and their use in reducing heat transfer penalties are also discussed, including mechanical and construction considerations.

The paper closes with an overview of the engineering capabilities and services offered by Heatric's Nuclear Engineering Department.

I. INTRODUCTION

Heatric has been manufacturing compact Printed Circuit Heat Exchangers since 1985, when Heatric was first established in Australia. The term 'compact' is often confused with meaning small; however, individual heat exchangers can be in excess of 8 metres length and 100 tons weight; assemblies can comprise tens of exchangers, so compact heat exchangers can be of appreciable size. 'Compact' more accurately refers to the higher duties that are achieved in smaller sizes than (say) shell and tube heat exchangers. This compactness is achievable through higher surface densities (i.e. heat transfer surface area per unit volume of heat exchanger), and through enhancement of heat transfer coefficients by selection of heat transfer surface geometries.

The compact heat exchangers types that Heatric manufactures are Printed Circuit Heat Exchangers (PCHEs), Formed Plate Heat Exchangers (FPHEs), and Hybrid Heat Exchangers (H^2Xs). These are all formed from alternating layers (typically hot-cold, hot-cold etc., see fig. 1). For PCHEs, layers are etched plates (see fig. 2). For FPHEs, layers consist of fins (see fig. 3) which are bound by side bars and separated by flat parting sheets. H^2Xs are a combination of both, a typical sequence being etched-plate/parting sheet/fins/etched-plate/parting sheet etc. Figures 4, 5, and 6 show sections through a PCHE, an FPHE, and a H^2X respectively.

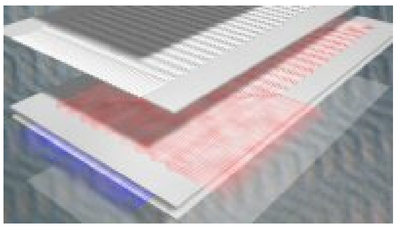

Fig. 1, stacked plates with the hot side shown in red and the cold side shown in blue.

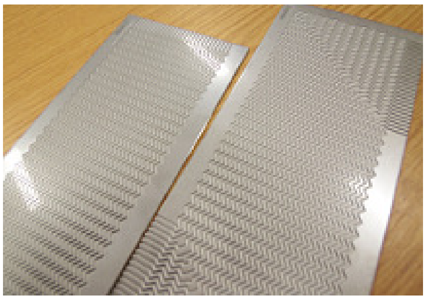

Fig. 2, etched plates (left plate is end-entry, right plate is side-entry).

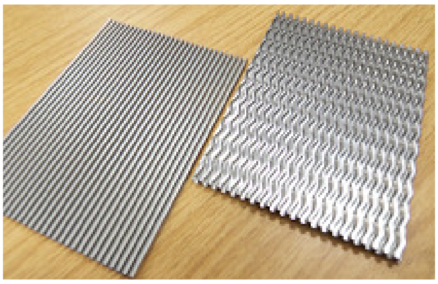

Fig. 3, fins (left side is plain-type, right side is herringbone-type).

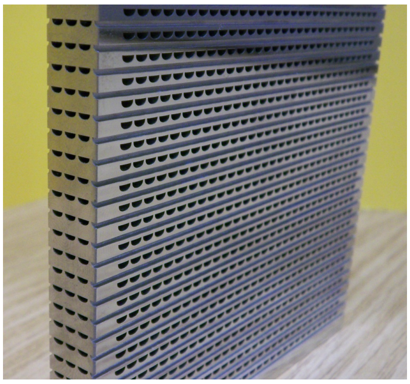

Fig. 4, section through a PCHE (cross-flow).

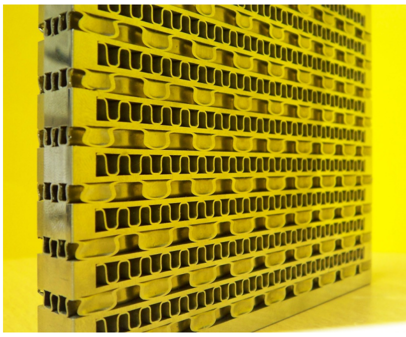

Fig. 5, section through a FPHE (cross-flow).

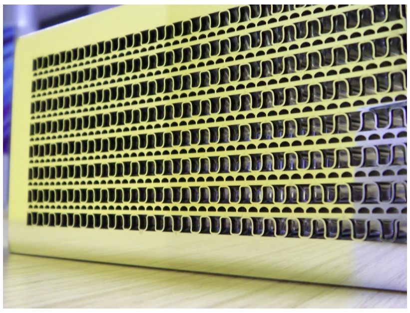

Fig. 6, section through a H^2X (counter-flow).

II. FACTORS INFLUENCING DESIGN

Before starting the design, the requirements of the proposed heat exchanger duty are screened to check for feasibility. This initial screening should review:

1. Design pressure and temperature – are these possible with the materials that are qualified for manufacture?
2. Process fluids – are these compatible with the available materials of construction, and free from excessive blockage-causing dirt/particulates/other fouling mechanisms?
3. Heating/cooling curves – do these satisfy an energy balance for the specified flow-rates and temperatures?
4. Allowable pressure drop – is this so small that required free-flow area will tend to infinity?
5. Required thermal effectiveness – is this so high that required heat transfer area will tend to infinity?

The first two items (particularly item 1) influence material choice and can influence the designer to select one heat exchanger type in preference to another; for example, a specification of a higher design temperature requiring a higher alloy of greater cost may result in selection of a FPHE solution, as FPHE construction tends to require less raw material. Specification of a higher design pressure may result in selection of a PCHE solution, which due to fin-forming limitations can be designed to satisfy a higher design pressure than a FPHE design.

II.A. Thermal specification

Assuming the required thermal effectiveness to be feasible, the effectiveness required will influence the configuration selected by the designer. Thermal effectiveness is given by:

$$\varepsilon = \frac{\left|T_{IN} - T_{OUT}\right|_{LARGER}}{T_{HOT,IN} - T_{COLD,IN}} \qquad (1)$$

Specifying a thermal effectiveness and maintaining a thermal energy balance means that specifying either the hot or cold stream inlet temperature must preset the other three terminal temperatures and thus the heating/cooling curve. The number of heat transfer units (NTU) required can then be found:

$$NTU = \frac{\left|T_{IN} - T_{OUT}\right|_{LARGER}}{(LMTD)_{WEIGHTED}} \qquad (2)$$

Another basic heat transfer relationship is:

$$Q = F_{GEOM.} \cdot UA \cdot (LMTD) \qquad (3)$$

$F_{GEOM.}$ in equation (3) is the geometric correction factor to the log-mean temperature difference (LMTD) due to non-counterflow. Design experience shows that for optimal heat exchanger designs, as $NTU \to \infty$, $F_{GEOM.} \to 1$. For a layer containing more than one cross-flow pass (a 'folded' design), this will lead to an increase in the number of 'folds' required. For a side-entry layer (e.g. the right-hand plate in figure 2), this will place a limit on the relative sizes of the distributing and counter-flow regions of the layer, since the distributing region is in cross-flow. For very high NTU designs, particularly where a low allowable pressure drop is specified, a platelet configuration would be selected (see figs. 7 and 8). Thus, re-specification of thermal-hydraulic duty may lead to a change in the optimal configuration, meaning 'scaling' to estimate heat exchanger volumes/costs should only be used with caution.

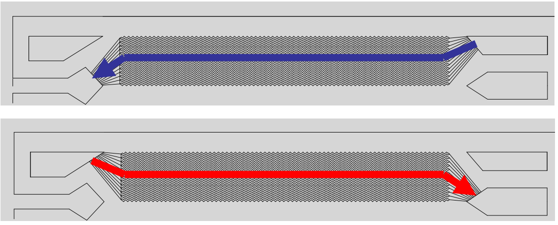

Fig. 7, example platelet configuration.

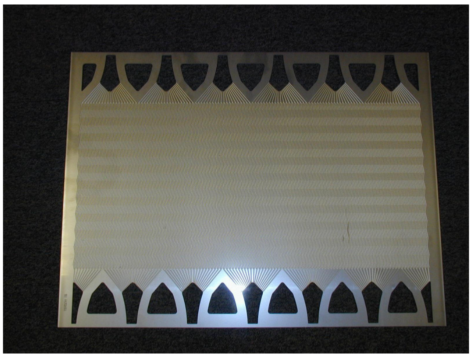

Fig. 8, example full size platelet plate.

II.B. Hydraulic specification

The allowable pressure drop should be specified appropriately. Although lower pressure drops are desirable to reduce operating cost and improve cycle efficiency, an inappropriately small pressure drop can make heat exchanger design very difficult or impossible, capital cost considerations aside. Pressure drop can be calculated by:

$$\Delta P = 4 \cdot f \cdot \left(\frac{l}{d_h}\right) \cdot \left(\frac{\rho u^2}{2}\right) \qquad (4)$$

A low pressure drop will therefore tend to require a short flow length and a low velocity. This will affect the heat transfer film coefficient:

$$h_{film} \propto Nu \propto \mathrm{Re}^a \propto u^a \qquad (5)$$

Referring to equation (3), since $F_{GEOM.}$ is fixed by the configuration and $(LMTD)$ is fixed by the process specification, a reduction is h_{film} (and thus a reduction in U) must be countered by an increase in heat transfer area. Heat transfer area can be increased by increasing flow length, flow width, or the number of layers. Since pressure drop is proportional to length (from equation 4), an increase in length is undesirable. Increasing flow width or the number of layers increases the free-flow area. Since:

$$u \propto \frac{1}{A_{free-flow}} \qquad (6)$$

Increasing the heat transfer area by increasing width or the number of layers ultimately reduces heat transfer coefficient and requires a further increase in heat transfer area. When the allowable pressure drop is too low this becomes a self-defeating iteration. This is compounded further because:

$$f \propto \frac{1}{\mathrm{Re}^b} \tag{7}$$

So reducing velocity also increases friction factor, requiring a further increase in free-flow area. Alternatively, a heat transfer surface with lower friction factor may be selected, but this tends to reduce the heat transfer coefficient and again require an increase in heat transfer area.

III. HEAT TRANSFER SURFACES

Compact heat exchangers are available with a range of surface types, generally intended to enhance surface density and heat transfer coefficients, and which also assist mechanical design (for example, fins form many attachment points between adjacent parting sheets). Some of the available surface types are considered here, these being:

1. Plain fins (formed)
2. Perforated fins (formed)
3. Herringbone & wavy fins (formed)
4. Serrated fins (formed)
5. Plain etched passages
6. Zigzag etched passages

Figs. 9 and 10 illustrate the different surface types. All except for the plain surfaces have features in their geometry to enhance heat transfer film coefficients by thinning/disruption of the hypothetical laminar film. Thermal hydraulic characteristics of fins are influenced by:

1. Fin count/pitch
2. Fin thickness
3. Fin height
4. Perforation fraction (if perforated)
5. Wave angle and wavelength (if herringbone/wavy)
6. Serration length (if serrated)

Thermal hydraulic characteristics of etched passages are influenced by:

1. Etched passage depth
2. Passage line width
3. Zigzag angle

The thermal, hydraulic, and mechanical calculations are inter-related, so all should be considered during optimisation. Designers may approach optimisation in differing ways; however, optimisation time may be reduced by initially screening passage geometries to eliminate configurations that would later fail the mechanical design calculation. Cost estimation should be introduced during thermal/hydraulic design, as most designs are optimised on cost (although in some cases another parameter may be more important for optimisation, for example weight). Surface selection is therefore an important part of design optimisation, as this greatly affects heat transfer and thus heat exchanger size and cost.

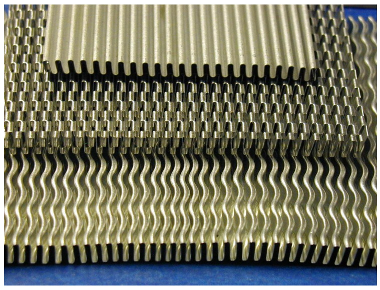

Fig. 9, fins shown top to bottom are plain, serrated, and herringbone (perforated fins not shown).

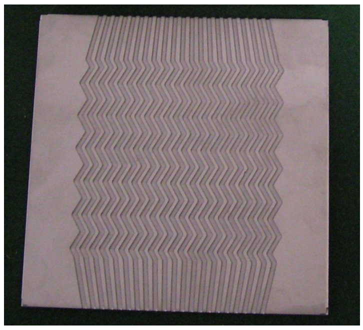

Fig. 10, etched plate showing plain etched passages at entry and exit, zigzag passages in-between.

IV. THERMAL-HYDRAULIC PERFORMANCE

The performance characteristics considered here are the Fanning friction factor, and the Colburn j factor. The Fanning friction factor is defined as:

$$f = \tau \cdot \frac{2}{\rho u^2} \tag{8}$$

i.e. fluid shear stress at the wall divided by velocity pressure. The Colburn j factor is given by:

$$j = \frac{Nu}{\text{Re} \cdot \text{Pr}^{1/3}} \tag{9}$$

From proportionality (7) and equation (9), it can be seen that f and j are both inversely related to Reynolds number. The index b in proportionality (7) is dependent upon regime (i.e. laminar or turbulent flow), therefore so is f; since j is proportional to Nusselt number, whose method of calculation depends upon regime, j is also dependent upon regime. Both are also dependant upon passage geometry.

This means that prediction of j and f data can be problematic, particularly where surface enhancements (such as deflections and serrations) are used. Manufacturers therefore produce test exchangers containing proposed surfaces of interest, which are then tested thermally and hydraulically to obtain experimental f and j data. These data may then either be tabulated or correlated for use in design.

Sample f and j data have been prepared[1] for various fin types (other geometries such as fin count, height etc. maintained constant) and are presented graphically in appendix A, figs. 1 and 2 respectively. Sample f and j data for various etched geometries of similar aspect ratio are also presented in these figures.

Referring to figs. 1 and 2 in appendix A, it can be seen that the different surface geometries give rise to differing f and j trends with Reynolds number. If follows therefore that the optimal surface type depends upon the Reynolds number of interest. If fins are considered, short serrated fins offer the highest j factors. The f factors are also higher, but since better heat transfer performance tends to result in smaller heat exchangers, the higher f factor may be acceptable if the smaller exchanger geometry lowers the pressure drop.

Referring again to the f curve for a short serrated fin, above a Reynolds number of approximately 3000 the curve begins to plateau. This occurs because the frictional pressure drop through a serrated fin becomes dominated by form drag[2], so f tends to a constant and equation 8 no longer applies. In design terms this means that increasing the fluid velocity lowers the j factor for heat transfer, but not the f factor, so an increasing pressure drop penalty must be paid for diminishing heat transfer benefit.

An important consideration therefore is the ratio of friction factor to j factor (or its inverse, j/f). This ratio can be seen in the following equation for the ratio of heat transfer film coefficient to power loss per heat transfer surface:

$$\frac{h_{film}}{E} = 2 \cdot \left(\frac{j}{f}\right) \cdot \left(\frac{1}{u^2}\right) \cdot \left(\frac{C_P}{\text{Pr}^{2/3}}\right) \tag{10}$$

This is a measure of the effectiveness of a surface at transferring heat for a given pumping power. It therefore follows that surfaces with lower f/j ratios are more effective at transferring heat for a given pumping power.

Fig. 3 in appendix A shows the f/j ratios for the surfaces that have been considered. It can be seen that plain or perforated surfaces have the lowest f/j ratio. This brings into question the advantage of using enhanced surfaces with deflections or serrations, if a higher value of h_{film}/E can be achieved with less-enhanced surfaces. Consider a rearrangement of equation 3 for $F_{Geom} = 1$:

$$\frac{Q}{(LMTD)} = UA \tag{11}$$

The UA criterion is useful in evaluating compact heat exchanger designs, since the required UA is constant for a given thermal duty. For a heat exchanger to perform satisfactorily, the available UA must meet or exceed the required UA. Logically, this may be achieved in two ways:

1. Lower values of mean overall heat transfer coefficient with greater heat transfer areas;
2. Higher values of mean overall heat transfer coefficient with lesser heat transfer areas.

The requirement for a greater heat transfer area under (1) generally requires more raw materials, thus incurring greater cost, than option (2). However, excessively high friction factors that could be incurred by option (2) can lead to increased cost due to a requirement for more layers (to obtain more free-flow area).

The optimum compact heat exchanger is a trade-off between selecting surfaces with higher j factors (to minimise required heat transfer area) and selecting surfaces with lower f/j ratios to achieve better heat transfer for the permitted pressure drop. What is considered 'optimum' is driven by end-user requirements, some examples being minimal cost, weight, installed height, or footprint.

Etched zigzag passages are particularly suitable for this optimisation. The chemical etching process means that a wide range of enhancements can be designed for without retooling. Referring to the j-factor graph, zigzag angles can be selected to include the range of performance offered between plain fins at the lower end and short-serrated at the higher end, with a 'high' zigzag offering higher j factors than the short serrated fin. The f/j ratios of the different zigzag angles are favourable compared to fin surfaces having similar j characteristics. An exception is plain fins versus plain-etched, where the f/j ratio is slightly preferable in the laminar region.

Another feature of heat transfer enhancement is that the f and j curves are 'smoothed' through the transition around the critical Reynolds number. In applications where the outlet temperature of side A is controlled by adjustment of side B flow, and side B flow is near the critical Reynolds number (e.g. in some gas coolers), selection of zigzag angle stabilises performance and thus helps stabilise control.

V. HEAT EXCHANGER CONFIGURATION

As important as surface selection is the selection of heat exchanger configurations. Some examples (with figures in appendix B) are:

1. Counterflow with side-side and end-end flows (app. B fig. 1).
2. Counterflow platelet configuration (app. B fig. 2).
3. Parallel flow (app. B fig. 3).
4. Crossflow (app. B fig. 4).
5. Cross-counterflow (app. B fig. 5).
6. Serpentine counterflow (app. B fig. 6).

The optimal configuration for a given duty is arrived at by iteration; often there are a number of designs that would satisfy the specified requirements, with a smaller number of designs that would be considered optimal. Additionally, it may not be immediately obvious when commencing design that a particular configuration would prove unsuitable, so the designer should use experience and judgement to screen concepts in order to reduce optimisation time. Expanding equation (3) to consider the configurations described above:

$$Q = F_{SC} \cdot F_{GEOM} \cdot UA \cdot (LMTD) \tag{12}$$

F_{SC} is a correction factor for heat 'short-circuiting' across folded passages. Where multiple crossflow or serpentine passes exist, an interpass temperature difference can exist. This temperature difference can result in heat being transferred between passes within a stream's layers, reducing the amount of heat transferred to the other stream. The table below summarises which of these correction factors apply to the configurations given above:

TABLE I

Configurations and correction factors

Configuration	F_{GEOM}	F_{SC}
Counterflow side/end	Yes (part crossflow)	No
Counterflow platelet	No (distributors excluded)	No
Parallel flow	Depends on flow arrangement	Depends on flow arrangement
Crossflow	Yes	No
Cross-counterflow	Yes	Yes
Serpentine counterflow	Yes (but only in small distributor regions)	Yes

Short-circuit can be reduced by reducing the heat transferred per pass, which will tend to increase the number of passes. F_{GEOM} is increased by increasing the number of passes, or by moving to a counterflow design. As discussed in section II.A, a greater value of NTU requires higher values of F_{GEOM}. For example, high effectiveness Helium and sCO2 recuperators tend to have high values for NTU, and so are usually designed in counterflow-platelet type configurations. Gas coolers typically have a lower value for NTU, so they are designed in counterflow side/end configurations or cross-counterflow configurations.

Another possibility with compact heat exchangers is multi-streaming, where three or more streams may be incorporated in a single heat exchanger. Two examples of three stream heat exchanger configurations are shown in appendix B figs. 7 and 8 respectively.

In addition to F_{SC} and F_{GEOM}, there are two other heat transfer penalties that can be important in compact heat exchanger design:

1. Fin efficiency.
2. Longitudinal conduction.[3]

For finned passages, fluid flowing immediately adjacent to the parting sheet only has to overcome the conduction thickness of the primary surface area. Fluid flowing adjacent to the fin leg and remote from the primary area transfers heat to the fin leg, which is secondary area, which by conduction transfers heat to the primary heat transfer area. This means that the secondary area is less effective at transferring heat than the primary; a correction to the secondary area, called fin efficiency, must be evaluated:

$$B = \left(\frac{effective\ \ fin\ height}{true\ \ fin\ height} \right)^2 \tag{13}$$

$$x = b_{fin} \sqrt{\frac{B \cdot h_{film}}{2 \cdot t_{fin} \cdot k_{fin}}} \tag{14}$$

$$\eta_{fin} = \frac{\tanh(x)}{x} \tag{15}$$

(See appendix B, fig. 9, for fin geometry definitions). Fin efficiency worsens with increasing fin height, increasing heat transfer film coefficient, and reducing fin thickness. In materials such as stainless steel where the thermal conductivity is relatively low, this encourages the use of lower fin heights. Fin thickness can also be increased, which will have the benefit of adding mechanical strength, but may also increase blockage fraction and thus pressure drop.

Longitudinal conduction arises when a stream’s temperature span is very much greater than the temperature difference for heat transfer. This establishes a temperature difference in the metal in the direction of flow. A sufficiently high temperature difference will result in heat being transferred along this temperature gradient, reducing the heat transferred between fluids.

Various methods exist for treating longitudinal conduction, including rigorous numerical methods. For brevity, these are not discussed here. However, an estimate of the decrement in thermal effectiveness due to longitudinal conduction can be found from:

$$\frac{\delta\varepsilon}{\varepsilon} \approx \frac{\left(k_{metal} / l \right) \cdot A_{conduction}}{C_{\min}} \tag{16}$$

This can be used as a screening tool during preliminary design to establish if longitudinal conduction will be significant. It is particularly important in designs where longitudinal conduction may be significant to specify an adequate allowable pressure drop; low allowable pressure drops require increased free-flow areas, which tend to increase the metal conduction area, worsening the effects of longitudinal conduction. This can result in a self-defeating iteration of the type discussed in section II.B.

VI. MANUFACTURING/MECHNICAL CONSIDERATIONS

As discussed in section 1, Heatric’s compact heat exchangers are made of stacked layers. In a PCHE, one etched plate can constitute a layer. In an FPHE, a layer consists of fin pads enclosed by side/end bars and enclosed between parting sheets.

PCHE plates are manufactured by producing an etching mask based on technical drawings of the plate design. Solid plates are then photochemically etched to remove material and form flow channels. These plates are then stacked according to the specified stacking sequence; each etched plate constitutes one stacking operation.

For FPHE layers, layers are formed as follows:

1. Fin dies, if not already available, are prepared for the fin geometry required and installed into the fin press.
2. Foil is fed to the fin press, which corrugates the foil.
3. Fin corrugation is initially trimmed off the press into rectangular pads of required length.
4. Fins pads are cut to the required shapes.
5. Fins are cleaned to remove grease/dirt introduced during pressing.
6. Side/end bars are cut to required sizes.
7. Parting sheets are cut to size.
8. Layers are stacked according to stacking drawings, indicating stacking pattern, location of varying fin pads and bar lengths within each layer (see fig. 11 for an example).

Fig. 11, part-stacked FPHE. The exposed layer shows 5 fin pads, 2 side bars, and 2 end bars.

H^2Xs are a hybrid of the PCHE and FPHE. In a hybrid, parting sheets are only required to close layers containing fins and to close the open face of etched plates.

The blocks are then prepared for diffusion bonding. Diffusion bonding is a solid-state joining process; Heatric’s bonding process does not use an interlayer or braze alloy, the bond is formed by using heat and pressure to promote grain growth across the boundary between components. A micrograph of a section through a PCHE core is shown in fig. 12:

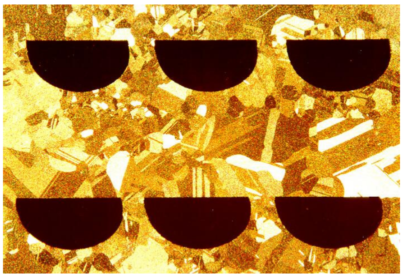

Fig. 12, micrograph of a PCHE section. Note that this is a crossflow PCHE, B side layers are not visible in this section.

The diffusion bonding process used by Heatric results in a return to the parent metal strength properties as listed in ASME. Sections II and III above discuss the need for 'screening' to ensure that bond-qualified materials are available for the requested design pressure and temperature, and to establish limits for feasible surface configurations. This screening exercise must also consider manufacturing capabilities; these are a crucial factor in the optimisation process. Allowable stresses for some of Heatric's bond-qualified materials are shown in fig. 13:

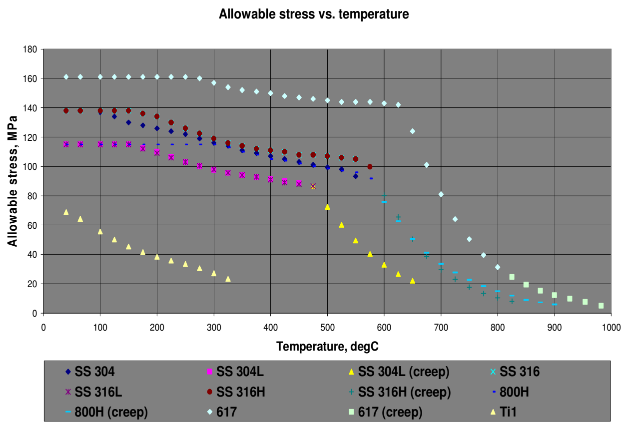

Fig. 13, allowable stress vs. temperature. Creep values are indicative, and based on a 100 000 hour life.

It can be seen from fig. 13 that the fall-off in allowable stress is high at high temperatures. It is therefore important to carefully consider design temperature and pressure requirements; in high-temperature situations, a small change in design temperature or a change to a specified transient condition can alter the material of construction and/or the heat exchanger configuration. Heat exchanger configuration may change due to:

1. Peak stresses (e.g. due to higher thermal stress in steady state or due to transient induced temperature gradients).
2. Cost.

The mechanical design of heat exchanger blocks considers the membrane stress between passages within a layer (i.e. across the ridge in PCHEs and fin legs), the membrane and bending stresses across layers (i.e. across the base of the passage in etched plates, or across parting sheets where used), and membrane plus bending stresses from layers to atmosphere through the sides of the block (i.e. through side margin in PCHEs, or side bars in FPHEs). The width of side bar/margin is also influenced by the header wall thicknesses required, as sufficient land for welding must be provided.

The choice between FPHE and PCHE is not always obvious; there are times when one configuration or the other is more optimal, but there is some design overlap between the technologies, more so with the introduction of the H^2X. Factors influencing choice are:

1. Material cost
2. Fin forming capabilities
3. Scrap factors
4. Labour cost

Fin forming is more limited than chemical etching. There are limitations to fin configurations that can physically be formed, and limits to the amount of load that can be applied by the fin press. This is another cost optimisation; increasing die load can result in a reduction in the size of fin pad that can be pressed. This increases costs, and can be impractical from a manufacturing standpoint. Chemical etching is less limited, since all that is required is a different etching mask.

Chemical etching favours more complicated configurations (e.g. serpentine, complex distributors) than forming. Formed surfaces can be used, but there can be a significant increase in labour costs as there are more components to stack.

Forming can be favourable in terms of materials costs; foil is pressed into shape, with scrap only being introduced by fin press start-up/shut-down and by cut-off. Chemical etching involves the removal of material from a solid plate, which is lost.

When higher design temperatures are specified, more costly alloys will be required. If an FPHE design is feasible for the specified design pressure, this may be a more cost-effective solution, as the raw material cost may dominate over the labour cost. If the design pressure is increased, a

PCHE solution may be more cost effective due to lower labour costs, particularly where a more 'conventional' alloy (e.g. 300 series stainless steel) can be used. This is an important consideration when considering steady-state operation versus transient operation. A steady-state estimate of design temperature may select for a lower cost alloy, but a peak transient temperature may select for a higher alloy. This affects mechanical design and also may affect the choice of heat exchanger type, both of which consequently affect thermal-hydraulic design optimisation.

VI.A. Mechanical selection pitfalls

As design temperature and pressure increase, the thickness of the fin leg (FPHE) or ridge (PCHE) increase. This reduces the number of passages per unit width of a layer. The temptation is to increase the passage size to increase free-flow area. This however tends to increase the ridge/fin leg thickness required, and so the iteration becomes self-defeating. A more optimal solution is to reduce passage size – this has the following effects:

1. Ridge/fin thickness can be reduced.
2. Passage base/parting sheet thickness can be reduced.
3. Free-flow area per unit face area is increased.
4. Heat transfer surface density is increased.

Another potential pitfall is related to exchanger assemblies. Where the heat exchanger size required exceeds manufacturing capabilities, heat exchanger blocks may be welded in parallel to achieve the desired size. However, at higher design temperatures, and particularly where high temperature gradients exist, thermal stresses may dictate that block-to-block welds are not possible. Blocks must therefore be manifolded together into a piped assembly; this incurs a pressure drop penalty that should not be overlooked.

VI.B. Mechanical selection tweaks

In addition to the other considerations discussed, some other optimisation options are:

1. Use of 'ported' designs.
2. Use of HIP/cast components for headers/nozzles (extra cost, but overcomes forming difficulties).
3. Differential pressure design (i.e. gauge design pressure referenced to a higher than atmospheric pressure).
4. Use of sentinel layers (to provide extra protection against cross-contamination).
5. Use of PCHEs to provide discrete flow channels (to minimise inventory loss should a leak occur).

A 'ported' design is one where headers are integral to the heat exchanger. An example is the platelet arrangement shown in fig. 7; the active passages are etched part-way into the plate, but the headers are etched right through the plate.

HIP/cast headers can be required if header diameter is larger and so a thicker header walls are required (larger headers may be required to reduce header pressure drop, so that more pressure drop may be used in the heat exchanger block). Pressure drop distribution is another degree of freedom in thermal-hydraulic/mechanical optimisation.

Differential pressure design is useful where: the compact heat exchangers are installed in a containment vessel; one of the sides (e.g. A side) discharges to the vessel atmosphere; the fluid discharging to the vessel atmosphere is always present when the other side (e.g. B side) is present in the compact heat exchanger. If the A side is the vessel atmosphere, then the design pressure of the A side of the compact heat exchanger is the allowable pressure drop, and the new design pressure of the B side is the original B side design pressure minus the A side operating pressure. This is generally advantageous to the thermal-hydraulic design of the compact heat exchangers (less metal is required for mechanical integrity, reducing metal wall resistance and enhancing compactness); mechanical design is also simplified, as the compact heat exchangers may then be designed to ASME VIII and the containment vessel to ASME III.

Sentinel layers provide an extra layer of protection where cross-contamination is a concern. For example, consider a CO_2/sodium IHX:

1. High thermal effectiveness calls for a counterflow platelet-style design.
2. Leakage of CO_2 into sodium is a concern.
3. Pure counterflow leaves side-header positions free for use.

Heatric's manufacturing process and post- bond leak testing minimise the risk of cross-leakage. However, extra security is available by design where required. PCHE options may be selected for this kind of duty, but H^2X designs are also available where blockage-causing impurities are anticipated in the sodium side (the sodium side can be selected to have a formed surface of larger hydraulic mean diameter).

The CO_2 side would usually be selected to be of PCHE-style. PCHE channels are discrete, and with potentially hundreds of lines per plate and hundreds (or even thousands) of plates, the magnitude of leakage from a small number of leaking passages is small.

Where concern still exists, sentinel passages can be included. The counterflow arrangement in App. B fig. 8 could have a stacking pattern of AB AB AB etc.; if sentinel layers were required, then a C layer of side-entry type (see fig. 2) could be introduced. If the A side is CO_2 and the B side sodium, the stacking pattern could then run ACBC ACBC ACBC etc. In the event of an A to C cross-leak failure, the C side would pressurise and therefore be detectable before a C to B cross-leak failure occurred.

VI.C. on-going programmes

The majority of Heatric's designs consider 300 series stainless steels, although other materials are fully qualified for manufacture (including those mentioned previously). 300 series stainless steels are more cost-favourable than higher alloys, so part of Heatric's materials development program develops their usage, in parallel with qualification of higher alloys (for example Inconel 617).

As standard, 316/316L stainless steel is used. This is a material that qualifies for 316 strength with 316L carbon chemistry. 316H tends not to be used, for supply chain reasons and weld sensitisation concerns.

Fig. 13 shows ASME VIII allowable stresses for 316 stainless steel. Although the allowable stresses are listed up to a maximum temperature of 550^{o}C, the use of 316 is permitted up to 600^{o}C. Allowable stresses are not quoted by ASME VIII for the range 550 – 600^{o}C, but an in-house programme is in progress at Heatric for design in this temperature range.

VII. HEATRIC'S NUCLEAR ENGINEERING DEPARTMENT

Heatric has an engineering department dedicated to Nuclear Enquiries. This multi-skilled department includes people from a range or disciplines, experience, and nationalities. The department consists of:

1. Nuclear Engineering Manager (background - Materials Science).
2. Assistant Project Manager (background - Mechanical Engineering).
3. Senior Engineer (background - Mechanical Engineering).
4. Mechanical Engineer (background - Mechanical Engineering).
5. Design Engineer (background - Mechanical Engineering).
6. Design Engineer (background - Chemical Engineering).
7. Design Engineer (background - Mechanical Engineering).
8. Senior Development Engineer (background - Materials Science).
9. Trials Technician (background - Manufacturing).

Heatric's Nuclear department can support programmes from concept stage where steady-state design is required, through to undertaking engineering studies at the detail engineering stage. The department has the expertise to work in the following fields:

1. Thermal design.
2. Hydraulic design
3. Mechanical design.
4. Materials selection.
5. Manufacturing/product development.

Thermal-hydraulic design is initially carried-out in steady state using proprietary methods, but can be expanded to evaluate transient response as transient conditions are identified. Calculations can be expanded to include the assembly in which the heat exchangers are installed; these calculations can be further supported/confirmed by CFD analysis.

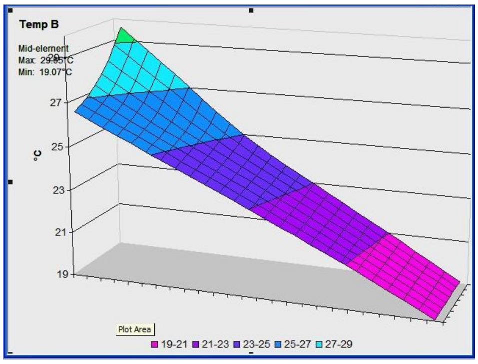

Fig. 14, example steady-state map of fluid bulk temperature.

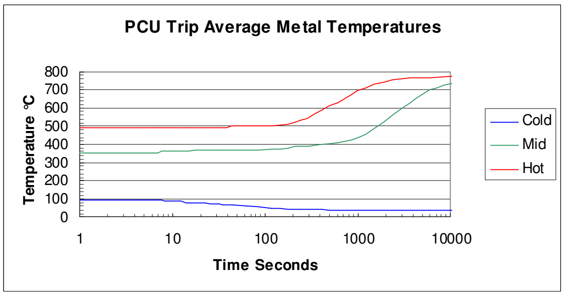

Fig. 15, example average metal temperature response in transient.

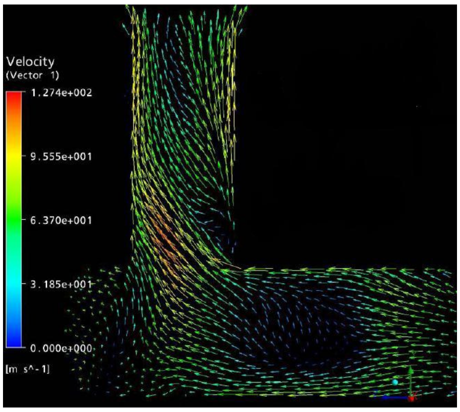

Fig. 16, example CFD velocity analysis.

Mechanical design in the first instance uses 'hand-calculation' methods per the applicable code, but can be further supported by FEA calculations, for example where high temperature gradients are induced by transients.

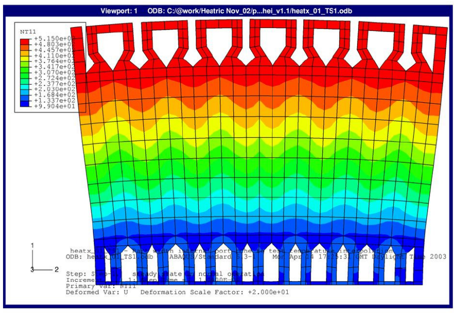

Fig. 17, example FEA output.

Manufacturing and product development are aimed and anticipating and responding to emerging product needs. Counterflow platelet-style configurations have been developed to meet requirements for high thermal NTU designs with high thermal stresses, and H^2X designs are being developed to meet the needs of (for example) liquid metal programmes.

Product trials can be also be carried out, for example to support calculations relating to cyclic duties. An in-depth program to analyse transients/upsets could include thermal-hydraulic analysis to predict heat exchanger response/cycles, mechanical analysis to estimate the number of cycles to failure, plus cyclic testing of representative samples to validate the results of calculations.

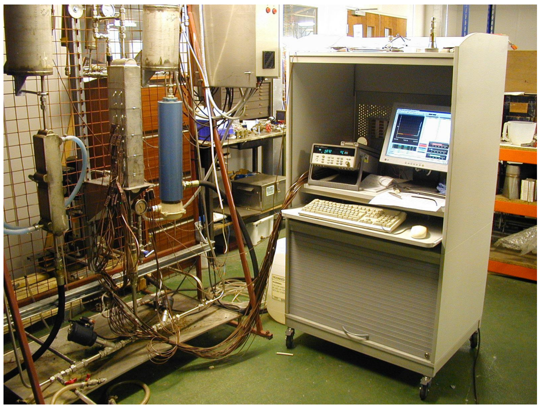

Fig. 18, example transient testing rig.

These capabilities mean that Heatric are well-placed to support the engineering phase of nuclear programmes.

VIII. CONCLUSIONS

Design of compact heat exchangers is not a straight-forward process. For a given thermal-hydraulic duty, a number of solutions may exist which appear to meet the specified requirements. However, the number of optimal solutions is much smaller. Design is further complicated by mechanical design requirements, and the effects of transience.

Initial thermal-hydraulic and mechanical screening is crucial to reducing the time spent evaluating infeasible options, and to approach optimisation in a more streamlined and efficient manner. A wide range of configurations and surface types are available to meet these challenges. Heatric offer a range of products and configurations, whose advantages are exploited to provide optimal bespoke-designed heat exchangers.

Heatric are well-placed to support the nuclear market, both through this range of custom-designed products, and through its engineering and testing support capabilities.

NOMENCLATURE

A	-	heat transfer area, m^2
$A_{conduction}$	-	metal cross-sectional area for longitudinal conduction, m^2
$A_{free-flow}$	-	fluid free-flow area, m^2
B	-	fin banking factor
b_{fin}	-	subchannel fin height, m
$C_{\min}$	-	minimum of hot and cold stream heat capacity rates, W/^{o}C
C_P	-	isobaric specific heat capacity, J/kg.^{o}C
d_h	-	hydraulic mean diameter, m
E	-	power loss per heat transfer surface, W/m^2
f	-	Fanning friction factor
F_{GEOM}	-	geometric correction factor to log-mean temperature difference for non-counterflow
F_{SC}	-	short-circuit correction factor
h_{film}	-	heat transfer film coefficient, W/m^2.^{o}C
j	-	Colburn factor for heat transfer
k_{fin}	-	thermal conductivity of fin, W/m.^{o}C
k_{metal}	-	metal thermal conductivity, W/m.^{o}C
l	-	length, m
$LMTD$	-	log-mean temperature difference, ^{o}C
NTU	-	number of transfer units for heat Transfer
$1/n$	-	fin pitch, m
Nu	-	Nusselt number
$\Pr$	-	Prandtl number
Re	-	Reynolds number
Q	-	heat transferred, W
t_{fin}	-	fin thickness, m
T	-	temperature, ^{o}C
u	-	velocity, m/s
U	-	overall heat transfer coefficient, W/m^2.^{o}C
x	-	parameter in eqns. (14) & (15)
ΔP	-	pressure drop, N/m^2
ε	-	thermal effectiveness
η_{fin}	-	fin efficiency
ρ	-	fluid density, kg/m^3
τ	-	fluid shear stress at wall, N/m^2

APPENDIX A.

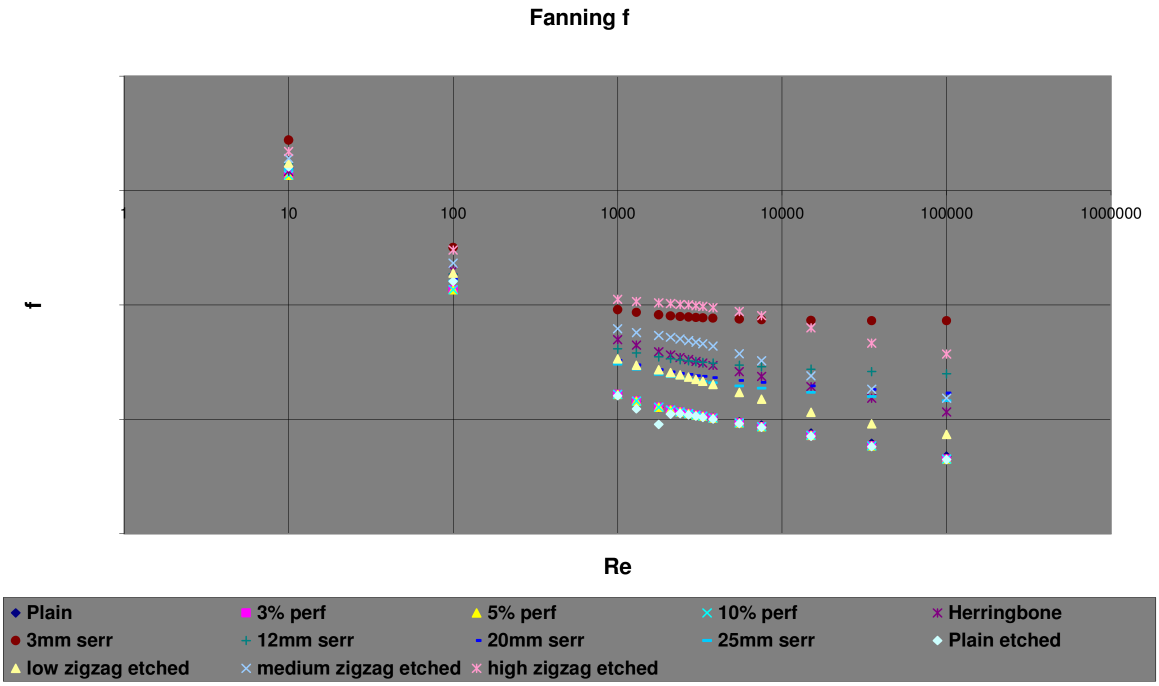

Fig. 1, Fanning friction factor vs. Reynolds number for some compact heat transfer surfaces.

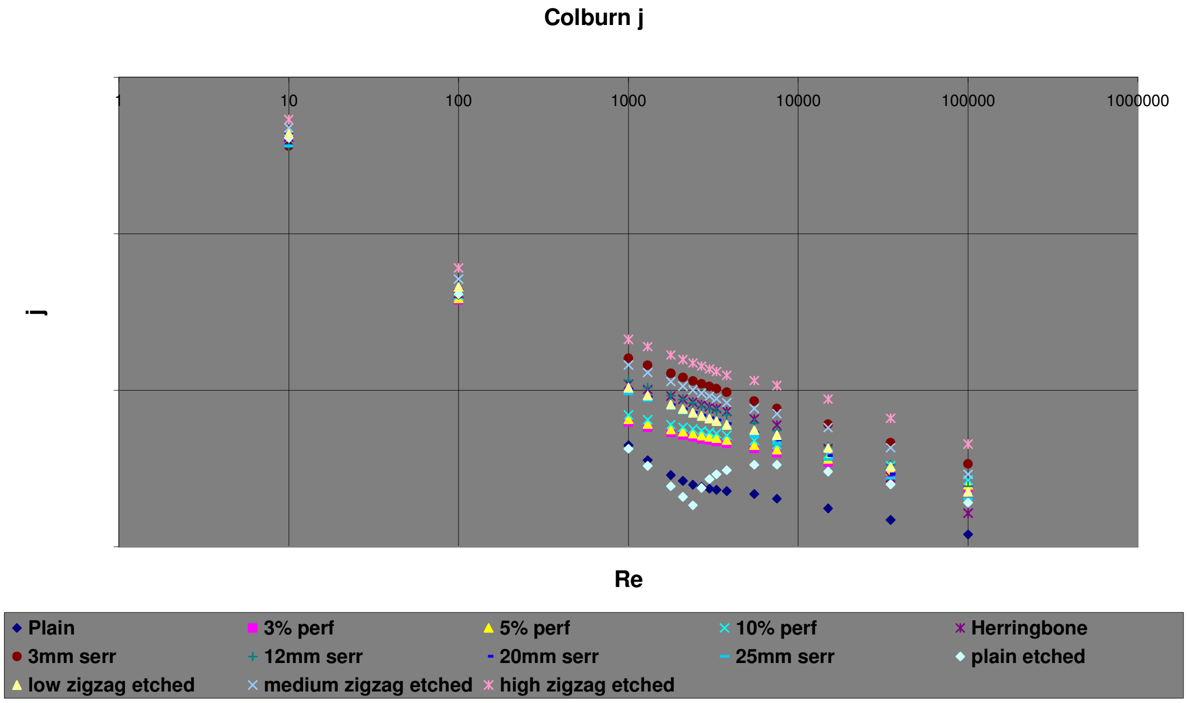

Fig. 2, Colburn j factor vs. Reynolds number for some compact heat transfer surfaces.

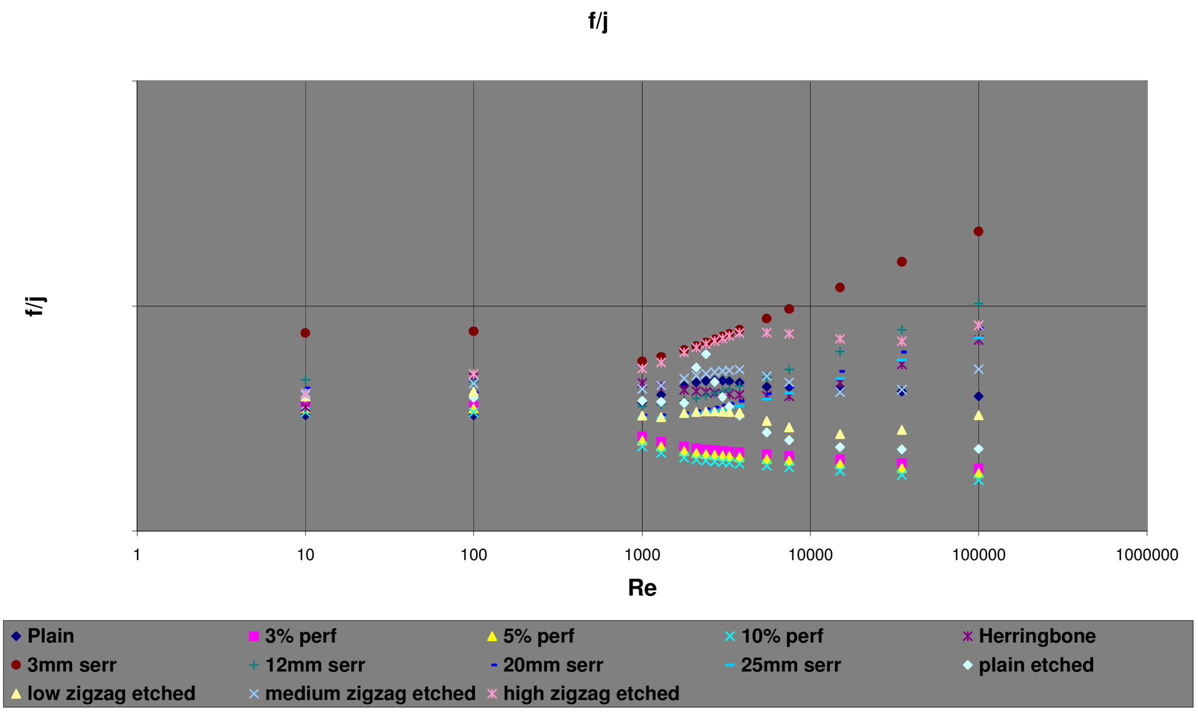

Fig. 3, f/j ratio vs. Renolds number for some compact heat transfer surfaces.

APPENDIX B.

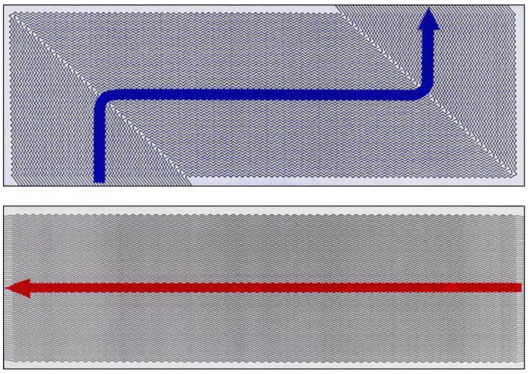

Fig. 1, counterflow example with side-side and end-end flows.

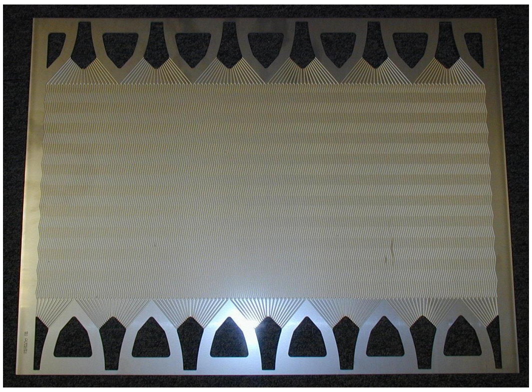

Fig. 2, counterflow example in a platelet configuration.

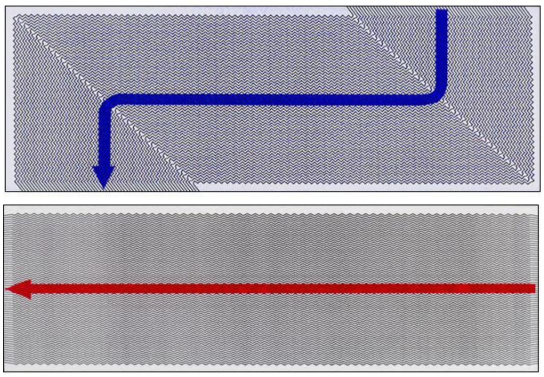

Fig. 3, parallel flow example with side-side and end-end flows.

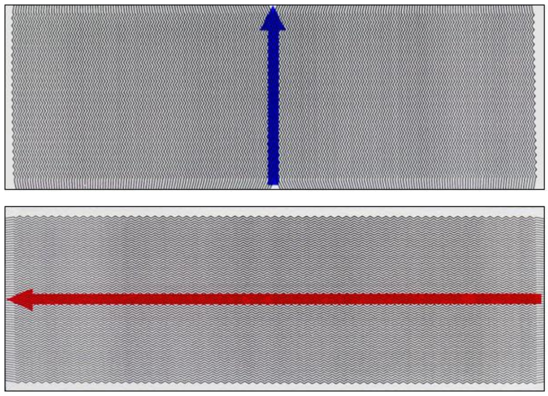

Fig. 4, crossflow example.

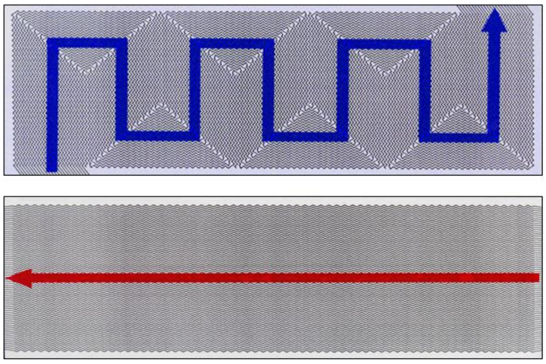

Fig. 5, cross-counterflow example.

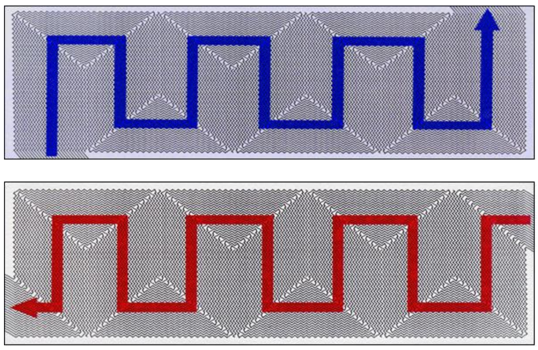

Fig. 6, serpentine counterflow example.

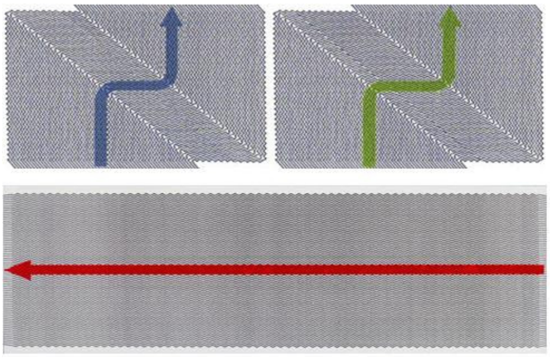

Fig. 7, three stream example, one hot stream with two cold streams in series.

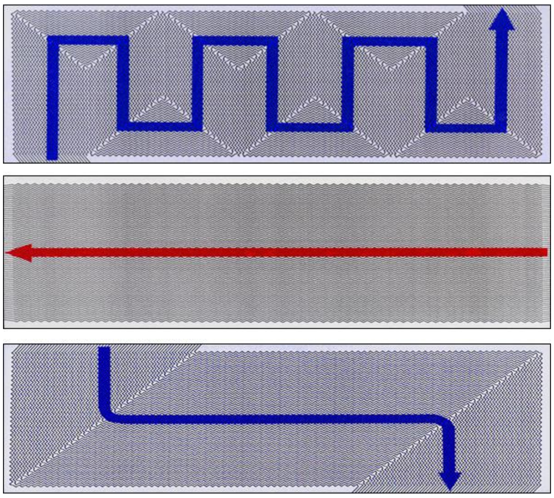

Fig. 8, three stream example, one hot stream with two cold streams in parallel.

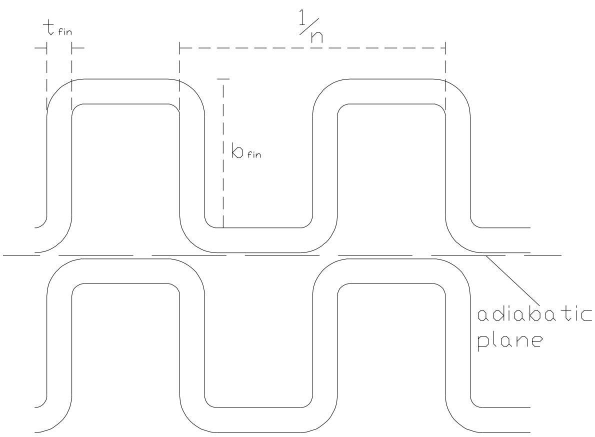

Fig. 9, sketch showing fin geometries. If the adiabatic plane is midway between the two fin layers (as shown), the ratio of "effective fin height" to "true fin height" = 2.

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