1990

The Impact of Compact Heat Exchangers on Refrigeration Technology and CFC Replacement

J. E. Hesselgreaves
National Engineering Laboratory

THE IMPACT OF COMPACT HEAT EXCHANGERS ON REFRIGERATION TECHNOLOGY AND CFC REPLACEMENT

by

J E Hesselgreaves
National Engineering Laboratory, UK

ABSTRACT

Several criteria point to the increased utilisation of compact heat exchangers in refrigeration plant in the near future. Firstly, the requirements to phase out production of existing ozone damaging refrigerants (including HCFC 22) means using less and compact evaporators and condensers allow this through reduced fluid inventory. Secondly, the new replacement refrigerants such as HFC 134a will be up to five times or so more expensive than present ones, so the cost factor in refrigerant charge will make compact exchangers more attractive. A further factor is that of performance. By analogy with other applications of compact heat exchangers, there is considerable potential for reducing temperature differences within size and cost constraints, so that cycle performance can be improved. This could at least partly offset any disadvantages of replacement refrigerants caused by physical property differences.

This paper reviews some recent applications of compact heat exchange technology in refrigeration, including brazed plate and printed circuit (PCHE) types, in terms of their impact on refrigerant inventory and overall size. Factors controlling the heat-transfer characteristics are explored and some aspects of further compact heat exchange development are discussed.

1 INTRODUCTION

The basic background to the CFC and HCFC issue is well known. A reduction in CFC 12 and HCFC 22 usage has already been made possible by the introduction of new heat exchanger types but the phasing out of both refrigerants (HCFC 22 largely for global warming reasons) will need new technology. Other papers in this conference deal with compressor developments, the search for suitable lubricating oils and the differences in heat transfer arising from changing to the alternatives such as HFC 134a. This paper is concerned with the prospects for using less refrigerant for a given thermal duty, that is, for reducing refrigerant charge by increasing compactness.

The direct effect, that of exchanger refrigerant charge, is not the only advantage of compactness. Smaller heat exchangers also give rise to smaller lengths of connecting pipework, thus making a further reduction in fluid inventory. A second advantage, not linked with the CFC issue, is that plant size, weight and thus cost are reduced, which can be a considerable advantage in, for example, transport and offshore applications. It also becomes possible to operate with lower temperature differences between streams, thus improving system economics and reducing primary energy consumption, with its resulting CO_2 effect on the greenhouse problem.

In this paper some basic principles of heat transfer and its enhancement in boiling and condensation are reviewed, in relation to compactness. The main heat exchanger types used in refrigeration are then described and their impact on inventory reduction discussed. Finally, further developments are briefly discussed.

The refrigeration industry lends itself well to the rapid implementation of advanced heat exchange techniques and the introduction of compact heat exchangers, partly because of the advantages of large scale production and partly because the costs to the user of equipment failure are usually small compared with those of the process industries. A further factor is that in order to maintain or improve system efficiency, whether for a refrigeration plant or heat pump system, the temperature differences must be kept low. For evaporator or condenser design, temperature differences (Ts) are often closely specified, and because heat flux- T relationships are often non- linear it is often more pertinent to consider heat fluxes than heat-transfer coefficients in describing performance. With regard to compactness, this is usually characterised by the hydraulic diameter d_h which is defined by

$$d_h = \frac{4\ A_c L}{A_s} = \frac{4 \text{ free volume}}{\text{surface area}}$$

if A_c is the mean flow cross-sectional area along length L.

For a given heat load Q,

$$Q = \dot{q} A_s$$

where $\dot{q}$ is the heat flux.

Thus free volume, V, is given by

$$V = \frac{A_s d_h}{4} = \frac{Q}{\dot{q}}\frac{d_h}{4}.$$

Hence free volume is directly proportional to d_h and inversely proportional to heat flux. For minimum free volume, with its direct effect on refrigerant charge, we seek to maximize heat flux (at a given temperature difference) and to minimise hydraulic diameter. A compact heat exchanger surface is typically defined as one having a surface area to volume ratio of greater than 700 m^2/m^3, which for normal porosities of about 0.8 gives a hydraulic diameter of less than 5 mm.

Boiling heat-transfer data are most often presented in the form of heat flux versus temperature difference (or wall superheat). Tube surface treatments for promoting nucleate boiling on tubes (Fig. 1a) are now well established in refrigeration plant, Hitachi's Thermoexcel-E, Wieland's GEWA-T and Linde's High Flux surface being among the best known (Bergles 1988). At a given superheat the heat flux is much higher (by up to an order of magnitude) for enhanced surfaces on single tubes with pool boiling but the cumulative advantage in bundles is much less pronounced. This is thought to be (Bergles 1988) because, with enhanced bundles, the boiling remains nucleate throughout the bundle with little convective augmentation, whereas with plain bundles there is considerable two-phase convective augmentation in the upper rows. The overall advantage in heat flux for a given superheat may thus be limited to a factor of 2-3, with its corresponding effect on inventory.

Compact exchangers of the brazed plate (Fig. 1b), plate fin (Fig. 1c) and printed circuit type (Fig. 1d), described in the following sections, display rather different behaviour to the shell-and-tube types. There is consistent evidence (Robertson 1980, Bergles 1988) that nucleate boiling is suppressed in the narrow passages of these types and that convective two-phase heat transfer dominates. A high convective component is a natural consequence of the much higher mass fluxes characteristic of these types. A contributory factor to a lower nucleate boiling component is that wall superheat is often reduced. It has yet to be established, however, whether any advantage would accrue from superimposing, for example, a high flux coating on a compact surface. At present, overall heat fluxes of compact heat exchangers are of the same order as those of enhanced flooded bundles.

Developments in condensing heat transfer have followed similar trends to those in evaporation, with the requirement now for condensate removal instead of nucleation promotion. Numerous studies (Marto (1988), Honda and Nozu (1990), Wang et al (1990)) have been made of condensate retention between fins of conventional shell-and-tube condensers. Various devices to assist condensate drainage have been tested (Tanasawa (1990)). Other developments consist of special tubes such as the Thermoexcel-C, which incorporates both fins and drainage channels.

3 HEAT EXCHANGER TYPES

3.1 Shell-and-tube Evaporators and Condensers

At present shell-and-tube types are the work horses of the commercial and industrial refrigeration industries. The use of augmented tubes, as described above, is now commonplace. It is normally recommended to use structured or sintered nucleate boiling surfaces on the tubes in the bottom few rows of a flooded evaporator to generate two-phase flow and to use low-fin tubes for convective augmentation in the remaining rows. The nucleate boiling surfaces also serve to overcome the increased saturation temperature (up to 0.5°C) caused by the head of liquid, which is greater in the bottom rows. The evaporator then performs rather like a fluidised bed, with the liquid phase being fluidised by the upward flowing vapour. It is this that marks the essential difference between a flooded evaporator and the kettle reboiler type used for process evaporation, in which the recirculating flow is exterior to the bundle and the two-phase flow through the bundle is entirely upward.

Direct expansion evaporators, in which the refrigerant evaporates inside the tubes, are also exploiting enhanced surfaces, chief among them being the various forms of microfin. Enhancement factors of about 2 are typical for this type.

Condensers typically employ surfaces such as low fins on the shell side or microfins on the tube side, which enable reductions of size of up to about 40 per cent. Condensation pressure drop is generally small.

The hydraulic diameter of typical tubular refrigeration bundles is about 20 mm, so that the enhanced surfaces of all types have limited value in reducing inventory. Clearly the hydraulic diameter could be reduced by using smaller tubes but at the expense of higher fabrication costs. It thus appears likely that tubular types will be progressively phased out as design and operating experience grows with the more compact types described below.

3.2 Brazed and Welded Plate Heat Exchangers

This type (Syed (1990)) has achieved a remarkable impact on refrigeration technology in the last few years, being used for both evaporation and condensation duties, particularly in the process industries. This market growth appears to have occurred independently of the CFC issue and to have been due to price advantage rather than compactness per se. A development of the gasketted plate heat exchanger, the brazed version (Figs 1b and 2) consists of the usual formed plates vacuum brazed at the edges and internal contact points to form a sealed block without gaskets. Hydraulic diameters are typically about 5 mm, with the result that both weight and inventory are 10-30 per cent of shell-and-tube values. Heat-transfer coefficients are also reported to be higher than shell-and-tube values (Syed (1990)). In boiling applications, the multiple parallel channels give rise to maldistribution which has proved difficult to eradicate, extra resistance upstream of each channel being necessary.

The welded plate version (Fig. 3) is used for larger duties (typically above 250 kW). It consists of pairs of plates welded on the reverse side of the gasket grooves to form the refrigerant channels and each pair is gasketted together to form the process channels. This version is thus suitable for potentially fouling process fluids. Both types are used for condensing and for flooded or forced convection boiling.

These types of exchanger are inherently lower in flow area and greater in flow length than shell-and-tube types, so that two-phase shear forces and hence pressure drops are higher. The increased pressure drop does not in general affect the saturation pressure and temperature enough, however (Syed (1990)), to offset the performance advantages of lower temperature differences, which can be as low as 1.5°C.

3.3 Air-conditioning Evaporators and Condensers

Air to refrigerant heat exchangers at present use circular tubes for the refrigerant flow, flat tubes being avoided historically because of pressure containment requirements. In-tube enhancements, whilst enabling the reduction of the total length of tube, make only a relatively modest impression on inventory. In the author's opinion any substantial improvement here will need a move away from circular tubes towards flat, galleried or other internally supported tubes, the supports doubling as extended surfaces. Provided that judicious circuiting allows the avoidance of excessive refrigerant-side pressure drop, this should yield both inventory saving and material saving, through the adoption of highly interrupted automotive radiator type air-side surfaces.

3.4 Plate-fin Heat Exchangers

Brazed aluminium heat exchangers (Figs 1C and 4) are now almost exclusively used for gas separation (cryogenic) duties and are finding increasing applications in the process field (Taylor (1987)). They are highly compact, having hydraulic diameters of 1.5 mm or less, can contain high pressures (up to 90 bar) and can support very low temperature differences between streams. Like the brazed plate type, they are subject to maldistribution and instabilities of operation and great care has to be taken in distributor design for two-phase flow. They are not used at present for general refrigeration duties but are being actively considered for advanced heat pump systems (Shohtani (1990)) and the type offers potential advantages in inventory reduction over the brazed plate type. In addition, because a highly extended surface is being used, it makes economical use of material and total weight is lower than other types.

The pressure drop of plate-fin heat exchanger surfaces in two-phase flow will be, like the brazed-plate type, higher than shell-and-tube types. The PFHE has, however, more flexibility in shape than the brazed-plate type, so that by adjustment of flow area and flow length a good design could be achieved for any specified maximum pressure drop. Refrigerant flow distribution could be a problem for extreme cases (high flow area, low flow length).

3.5 Printed Circuit Heat Exchangers

This new type of exchanger has been developed by Heatric Pty Ltd over the last seven years (Johnston (1985), Johnston (1986), Reay (1990)). Flat plates are photochemically etched with heat-transfer passages and then diffusion bonded together to form a solid block. The exchanger is illustrated in Fig. 5 and a schematic cross-section is shown in Fig. 1d. It is clear that the heat-transfer surface is effectively all primary. Entry and exit ports may be formed within the block, or alternatively headers may be welded onto the edges as with plate-fin exchangers.

This type is the most compact of those presently available, with hydraulic diameters of 0.5 mm to 1 mm, supporting power densities up to 5 MW/m^3. With such small fluid passages, the type is particularly suited to non-fouling duties such as refrigerant evaporation and condensation and there are considerable prospects for advanced absorption cycle applications, in which high performance of the heat exchangers is at a premium. Substantial reductions in both fluid inventory and overall plant size are possible as illustrated in Fig. 6. The potential problems of flow instability are relatively easily dealt with by etching appropriate (equal pressure drop) distributor passages upstream of the heat-transfer section. This distributor pressure drop can be shared with that of the expansion device in DX applications. The core pressure drop can be relatively easily controlled by the fact that this type has similar flexibility in shape to the plate-fin type above. In most cases the advantage gained in temperature difference through the better heat transfer will outweigh the small loss in saturation temperature caused by pressure drop. At present this type is more expensive than conventional types for normal duties but becomes very economical when exotic materials are called for in corrosive applications. The inherently small size makes aggressive chemical cleaning viable in fouling situations.

4 FURTHER DEVELOPMENTS

The immediate future is likely to see, as already mentioned, a rapid progress towards saturation of the mainstream refrigeration market with the brazed or welded-plate type. This process will be hastened by the replacement of CFC refrigerants with HFCs such as 134a. Looking further ahead and particularly for more demanding applications, the more compact plate-fin type is likely to make an increasing market impact. This type has a minimum hydraulic diameter of about 1 mm, so that further reductions giving lower inventories may only come through new surface forms related to PCHEs.

Research work is in progress at NEL on a new form of highly compact surface (Patent applied for). Illustrated schematically in cross-section in Fig. 1e it consists of a stack of contiguous thin perforated sheets, or fins, sandwiched between stream separation plates. The perforations, which can be of any shape, are out of phase in adjacent fins so that the fluid flows in and out of the planes of the fins on its passage through the surface. Two or more fin sheets can be used, depending on the proportion of extended surface required. The flow is highly three-dimensional, resulting in a high heat transfer at low pressure drop. Single-phase experimental results for a thick-finned version, expressed in terms of Colburn j-factor and Fanning friction factor f, versus Reynolds number, are shown in Fig. 7 (j and f are defined by: $j = hPr^{2/3}/GC_p$, where h = heat-transfer coefficient, Pr = Prandtl number, G = mass velocity and C_p = specific heat; and $f = 2\rho\Delta PA_c/G^2A_s$, where ρ = density and ΔP = pressure drop). This form of surface, bridging the gap between PFHE and PCHE types whilst allowing very small hydraulic diameters, is suitable for single-phase (either liquid or gas) and two-phase applications.

The new compact forms of heat exchanger, in addition to being increasingly used in evaporators and condensers, are also of relevance for intercoolers (or economisers), solution heat exchangers and the other exchangers used in absorption cycle equipment, for which performance demands are high.

5 CONCLUSIONS

Current market trends, together with developments both of existing compact heat exchanger types and of new types, are expected to play a strong role in the reduction of refrigerant usage in the foreseeable future.

ACKNOWLEDGEMENTS

This paper is presented by permission of the Director, National Engineering Laboratory, East Kilbride, Glasgow and is Crown copyright.

REFERENCES

BERGLES, A. E. (1988). Some perspectives on enhanced heat transfer second generation heat transfer technology. Trans ASME, 110, 1082.

HONDA, H. and NOZU, S. (1990). Effects of bundle depth and working fluid on the optimised fin geometry of a horizontal low-finned condenser bundle. Second International Symposium on Condensers and Condensation, HTFS, University of Bath, UK.

JOHNSTON, A. et al. (1985). Printed circuit heat exchangers for energy conservation. End of Grant Report No 519, National Energy Research, Development and Demonstration Program, Australia.

JOHNSTON, A. et al. (1986). Miniaturised heat exchangers for chemical processing. The Chemical Engineer, Dec.

MARTO, P. J. (1988). An evaluation of film condensation on horizontal integral-fin tubes. J. Heat Transf., 110, 1287.

REAY, D. A. (1990). Novel techniques in heat exchanger design including printed circuit units. Symposium on Advanced Heat Exchanger Design, Inst. Chem. Engr., Leeds, UK.

ROBERTSON, J. M. (1980). Review of boiling, condensing and other aspects of two-phase flow in plate-fin heat exchangers. In Compact Heat Exchangers - History, Technological Advancement and Mechanical Design Problems. ASME HTD, Vol. 10.

SHOHTANI, H. (1990). The experimental research on condensation of the plate-fin heat exchangers, using non-azeotropic refrigerant mixtures. Second International Symposium on Condensers and Condensation, HTFS, University of Bath, UK.

SYED, A. (1990). The use of plate heat exchangers as evaporators and condensers in process refrigeration. Symposium on Advanced Heat Exchanger Design, Inst. Chem. Engr., Leeds, UK.

TANASAWA, I. (1990). Condensation heat transfer - Japanese research in 1980s. Second International Symposium on Condensers and Condensation, HTFS, University of Bath, UK.

TAYLOR, M. A. (1987). Plate-fin heat exchangers, guide to their specification and use. 1st Edn., HTFS.

WANG, S. P., HILIKATA, K. and DENG, S. J. (1990). Experimental study on condensation heat-transfer enhancement by various kinds of integral finned tubes. Second International Symposium on Condensers and Condensation, HTFS, University of Bath, UK.

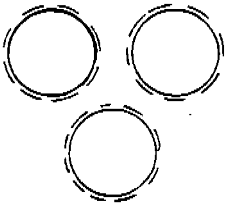

FIG 1a SHELL AND TUBE BUNDLE (ENHANCED)
$d_h = 20$ mm

FIG 1b BRAZED PLATE HEAT EXCHANGER
$d_h = 5\text{–}8$ mm

FIG 1d PRINTED CIRCUIT HEAT EXCHANGER (PCHE)
$d_h = 0.5\text{–}1.0$ mm

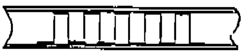

FIG 1c PLATE-FIN HEAT EXCHANGER (PFHE)
$d_h = 1.5\text{–}3.0$ mm

FIG 1e PERFORATED MATRIX HEAT EXCHANGER
$d_h = 0.3\text{–}1.0$ mm

FIG 1 HEAT TRANSFER SURFACE TYPES, ILLUSTRATING PROGRESSION OF COMPACTNESS

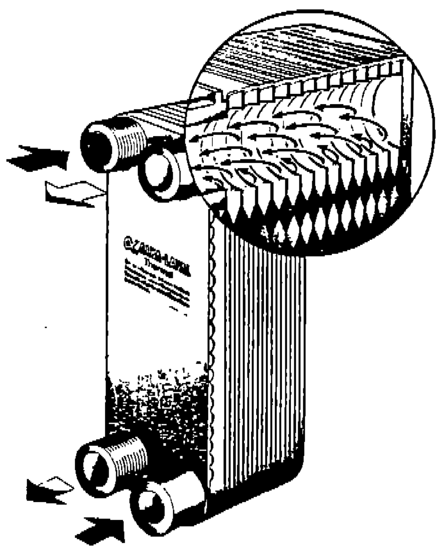

FIG 2 BRAZED PLATE HEAT EXCHANGER

(from Syed (1990))

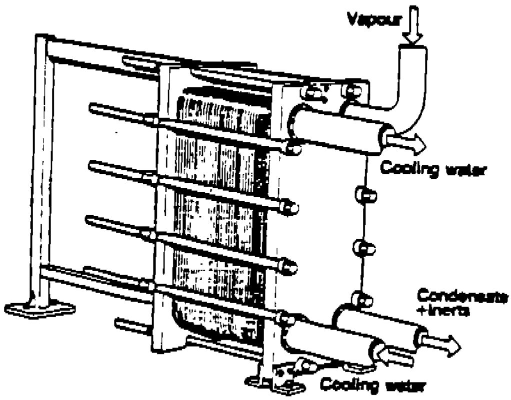

FIG 3 WELDED PLATE HEAT EXCHANGER

(from Syed (1990))

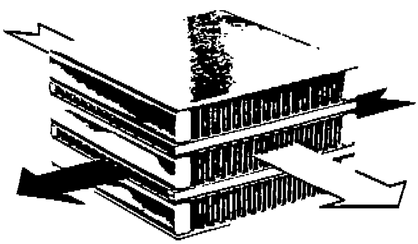

FIG 4 PLATE-FIN HEAT EXCHANGER (CROSSFLOW TYPE)

(courtesy Marston Palmer Ltd)

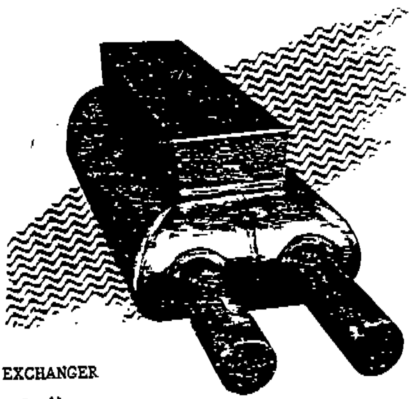

FIG 5 PRINTED CIRCUIT HEAT EXCHANGER
(courtesy Heatric Pty Ltd)

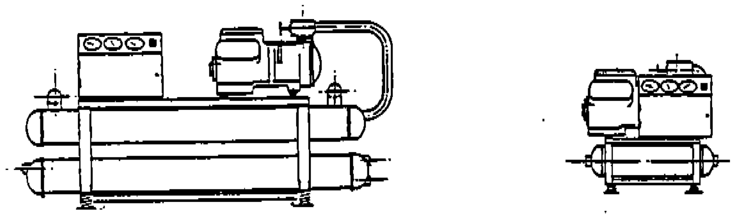

FIG 6 EFFECT OF PCHE ON PLANT SIZE (from Johnston (1985))

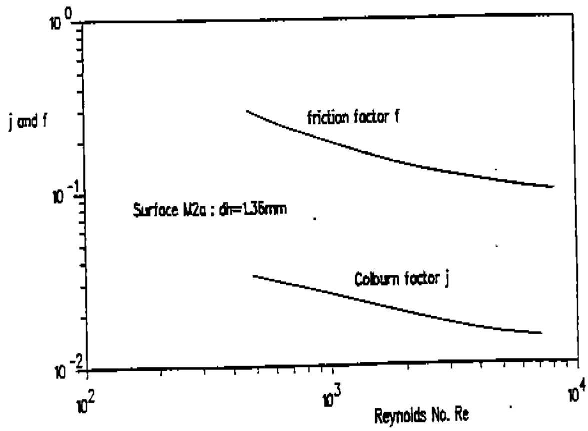

FIG 7 PERFORMANCE OF NEW COMPACT SURFACE