

COMPACT HEAT EXCHANGERS

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Activities to improve the efficiency of compact heat exchangers concentrated on enhancing the heat transfer by affecting the boundary layer and by enlarging active surfaces. Optical methods like holographic interferometry offer valuable possibilities to assess the effectiveness of measurements for heat transfer augmentation. They are also good touchstones for theoretical models and computer codes. Some recent findings in the development of compact heat exchangers with respect to heat transfer improvement are discussed. The work chosen from the literature is mainly concentrated on plate heat exchangers.

INTRODUCTION

In the literature /1,2/ a heat exchanger is defined as compact if the ratio between its heat transferring surface A and its volume V is larger than $700\text{m}^2/\text{m}^3$. Today compact heat exchangers are available from the market up to a surface-volume ratio of almost $7000\text{m}^2/\text{m}^3$ /1/. With high values of the cross-sections the fluid flow becomes rather small, as easily understandable which has two effects namely a tendency to laminar flow in the heat transferring channels and a high pressure drop.

Laminar or near laminar flow is associated with low heat transfer coefficients and, therefore, the efficiency is improved by enhancing the boundary layer conditions - for example with turbulence promoters - and by additional enlargement of the heat transferring surface for example via fins. Of course, fins can act as turbulence promoters also.

It is not possible to treat the whole subject of compact heat exchangers in a paper of limited space. Therefore, here emphasis is given on methods and their success for improving the heat transport in compact heat exchangers and only a very limited part of the literature published in the last few years can be briefly discussed. There exist excellent survey papers about compact heat exchangers for example by Shah /1/, Shah and Webb /2/, Webb /3/ and Bergles /4,5/.

PLATE HEAT EXCHANGERS

Most of the compact heat exchangers are designed according to the plate type having channels of rectangular cross-section for the fluid flow. In the most simple case the cross-section has the form of a square and for manufacturing the ducts the question rises what is the best compromise for the radius at the corner of the duct with respect to heat transfer behaviour and low production costs. Interferometric studies /6/ confirmed that the thermal boundary layer in narrow corners is thicker than at the plain walls as

the interference fringes, representing isotherms, in Fig. 1 demonstrate. The evaluation of this interferometric studies resulted in large differences of the Nusselt number around the perimeter of the duct, especially at low Reynolds numbers as seen in Fig. 2. In a narrow corner the Nusselt number approaches almost zero with laminar flow. In contrary the corners can act as turbulence promoters with Reynolds numbers above 1800. Now maxima of the Nusselt number can be observed near the corners.

Averaging the Nusselt number versus the circumference of the duct the differences between channels with sharp corners and those with smooth corners are not so high as Fig. 3 conveys. This figure confirms also the good agreement between interferometric measurements and experiments using the classical method of heat balances.

A simple method to improve the cross-transport in a rectangular duct is a periodical deflection of the flow direction, for example by a zigzag arrangement. Asako et. al. /7,8/ performed a theoretical study by applying numerical solutions to the balance equations and predicted the pressure drop and the Nusselt number as a function of the deviation angle and the spacing (distance versus length) of the zigzag elements. Figure 4 demonstrates that a stronger deflection of the flow by the zigzag elements improves the heat transfer and, however, is associated with a large increase in pressure drop. Wider spacing results also in pressure drop increase, probably due to additional turbulence. However the heat transfer is improved only moderately.

Wavy elements for flow deflection let expect a smoother fluid-dynamic behaviour with lower pressure losses, however, the improvement in the heat transfer along the flow path is limited to locally restricted areas as Fahrhanieh and Sunden /9/ proved in a theoretical study and as confirmed in Fig. 5. The peaks in the course of the Nusselt number in Fig. 5 are due to boundary layer separation and vortex-formation, when the flow passes the deflection elements.

A simple way to promote turbulence and to separate the thermal boundary layer from the wall would be a grooving of the channel wall. Interferometric studies /10,11/ of the boundary layer conditions in such a grooved channel confirmed the separation of the boundary layer at the beginning of the groove and its new formation at the edge of the next plateau. The local Nusselt number experiences large and sharp changes at the border between groove and plateau as can be seen from Fig. 6. At the plateau, however, the thermal boundary layer is building up quickly again and, therefore, the Nusselt number deteriorates rapidly. The influence of the grooves decreases along the flow path of the fluid due to turbulence effects. Because of the strong reduction of the heat transfer in the grooves the average Nusselt number in such a design is not so much better than that of a flat, plane channel having the same length. There is, of course, potential for improvement by optimizing the length of the grooves and that of the plateaus.

Another possibility to augment the heat transfer in a rectangular channel with longitudinal flow is to insert cylindrical segments and fix them on the walls in a staggered or in-line arrangement as shown in Fig. 7. These cylindrical segments promote deflections and velocity changes in the flow and can act as turbulence promoters if the Reynolds numbers are above 1500. These configurations were studied by interferometric methods /12/ also. Taking into account the heat conductivity in the material of the cylindrical segments one can argue that this arrangement is a mixture of plate heat exchanger and tube heat exchanger.

For an in-line arrangement of the cylindrical segments the course of the local Nusselt number at and behind the first, third, and fifth element along the flow path is given in Fig. 8 with $Re = 1500$ of the fluid. By comparing the Nusselt number behind the first 3 segments with that behind the fifth segment one can draw the conclusion that the flow is

starting to become turbulent due to the upstream perturbation effect which is confirmed¹ by looking at the interference fringes of the interferometric studies. For Reynolds numbers above 1000 this turbulence promoting cylindrical segments enhance the heat transfer at the wall to a large extent compared to that of a flat smooth duct as Fig. 9 demonstrates. The augmentation is not much depending on the arrangement of the segments - staggered, non-staggered or inclined (diagonal to the main flow direction) - but the non-staggered configuration gives the best results.

Similar conditions as with inclined cylindrical segments at the walls experiences the flow by passing through a duct which is formed by wavy-sinusoidal sheets of metal (Fig. 10). Gaiser and Kottke /13,14/ studied the heat transfer and the pressure drop in such configurations as a function of the amplitude of the waves. They used an experimental technique based on the analogy between heat and mass transfer. The surface of the wavy sheets was paved with a coloured component, and the chemical reaction of the gas carrying a reactive component produced a change in the colour of the surface which was used as a measure of the mass transfer in the duct. They found a strong influence of the wavelength and the steepness of the waves with respect to the flow direction. The best heat transfer was achieved with wavy configurations perpendicular to the flow direction. For $Re=2000$ this fact is demonstrated in Fig. 11. Orthogonal waves, however, create a much larger pressure drop than inclined ones as the course of the pressure drop coefficient versus the inclination angle presented in the right diagram of Fig. 11 demonstrates.

Finally one can think about the effect of perforations in wavy or meander-like sheets a design, which was used to study regenerative heat exchanger - periodic heating and cooling of storage material - by Fuji et al. /16/. The different arrangement of perforating holes in the meandering metallic sheets as studied by Fuji et al. is shown in Fig. 12. The perforation can result in a sucking off of the boundary layer as well-known from aviation studies. Perforated elements, however, can induce acoustic vibrations and by this may produce noise. Fuji et al. /16/ found only slight improvement of heat transfer with increasing porosity of the sheets, but the contraction and the enlargement of the flow area by the meandering duct affected the heat transfer and the pressure drop strongly as shown in Fig. 13. The Fanning friction factor is more influenced by the meandering duct shape than the Nusselt number.

HEAT TRANSFER AUGMENTATION BY FINS

There exist many proposals for fin-design in the literature. Figure 14 gives a few examples of different forms and arrangements. Fins are used, as mentioned before, not only for enlarging the heat transferring area, but also for promoting turbulence in the flow and by this enhancing the heat transfer coefficient. The principal geometrical forms are quite similar to that discussed before as plane, wavy or perforated flow ducts. New forms are strip offset fins, louvered fins and pin fins. Kays and London /16/, Maltson et al. /17/, Joshi and Webb /18/, Sunden and Svantesson /19/ and Sparrow et al. /20/ present newer results of heat transfer studies with fins of different form and arrangement.

Joshi and Webb /18/ studied strip offset fins of a design shown in Fig. 15a and developed correlations for predicting the Fanning friction and Colburn factors for such arrangements. They defined a critical Reynolds number

$$Re^* = 257.2 \left(\frac{l}{s} \right)^{1.23} \left(\frac{t}{l} \right)^{0.58} \left(\frac{t}{d_{hyd}} + 1.328 \left(\frac{l}{d_{hyd}} \right) Re^{-0.5} \right)$$

at which the flow in the wake of the fins becomes turbulent and consequently classify

their correlations in two regions, namely a laminar ($Re < Re^*$)

$$f_l = 8.12 Re^{-0.74} \left(\frac{l}{d_{hyd}} \right)^{-0.41} \left(\frac{s}{h} \right)^{-0.02}$$

$$j_l = 0.53 Re^{-0.50} \left(\frac{l}{d_{hyd}} \right)^{-0.15} \left(\frac{s}{h} \right)^{-0.14}$$

and a turbulent ($Re > Re^* + 1000$)

$$f_t = 1.12 Re^{-0.36} \left(\frac{l}{d_{hyd}} \right)^{-0.65} \left(\frac{t}{d_{hyd}} \right)^{0.17}$$

$$j_t = 0.21 Re^{-0.40} \left(\frac{l}{d_{hyd}} \right)^{-0.24} \left(\frac{t}{d_{hyd}} \right)^{0.02}$$

one.

Dubrovski and Vasiliev /21/ studied a little different arrangement of offset fins compared to that of Joshi and Webb, which is shown in Fig. 15b. They also developed correlations for the Nusselt number and the Fanning friction factor and distinguished two regions which are subdivided by limiting Reynolds numbers

$$Re_{lim, Nu} = 3960(\delta/d)^{0.25}(l/d)^{0.42}$$

$$Re_{lim, f} = 448(\delta/d)^{-0.053}(l/d)^{0.09}$$

The heat transfer correlations not only take into account Reynolds number, but also describe the influence of the spacing of the fins and the equivalent diameter.

$Re < Re_{lim, Nu}$:

$$Nu = 0.000437(\delta/d)^{-2.6}(l/d)^{-0.15} Re^{2.2(\delta/d)^{0.56}(l/d)^{-0.63}}$$

$Re > Re_{lim, Nu}$:

$$Nu = 0.00723(\delta/d)^{-1.6}(l/d)^{-0.9} Re^{1.2(\delta/d)^{0.34}(l/d)^{0.15}}$$

The pressure drop is also given as a function of the geometrical conditions in the fin duct.

$Re < Re_{lim, f}$:

$$f = 1.05(\delta/d)^{-1.06}(l/d)^{-0.217} Re^{-0.277(\delta/d)^{-0.288}(l/d)^{0.064}}$$

$Re > Re_{lim, f}$:

$$f = 0.131(\delta/d)^{-0.44}(l/d)^{-0.234} Re^{-0.0042(\delta/d)^{-1.76}(l/d)^{0.39}}$$

Pin fins are mainly used for cooling electronic equipment. Incropera et al. /22/ studied the heat transfer augmentation with such arrangements. They used two different materials

for the fins, one with high thermal conductivity (copper) and another with low thermal conductivity (Lexan) to be able to distinguish between heat transfer augmentation by surface enlargement and by turbulence promotion. Examples of their findings on the heat transfer in the different rows of the cylindrical fins are shown in Fig. 16. Elements of such fin arrangements should not consist of too many pin rows as this figure conveys.

TUBE AND SHELL DESIGN

To reduce the volume of a tube and shell heat exchanger the tubes can be equipped with fins of different shape as well-known. The literature about heat transfer at fin tubes is so numerous that no attempt can be made here to do justice to this large field of research by describing only a little part of it. Therefore, the discussion will be restricted to some information from the literature on elliptical tubes only.

Merker et al. /23,24/ studied with an experimental method, using the analogy between heat and mass transfer, the transfer conditions in bundles of elliptical tubes of various shape and different arrangement. For gas as coolant with a Prandtl and Schmidt number approximately 1, the measured Sherwood numbers are equal to the Nusselt numbers which would define the heat transfer. Figure 17 presents the measured data averaged over the circumference of the elliptical tubes and over all rows in the bundle as a function of spacing in longitudinal and orthogonal direction and of the Reynolds number, which is referred to the equivalent diameter of the tubes and the velocity of the gas upstream of the bundle. A wider spacing in longitudinal distance and a compact arrangement in orthogonal direction gave the best results.

The local heat transfer coefficient around the surface of an elliptical cylinder behaves different from that around a circular tube, not only with forced, but also with natural convection. Huang and Mayinger /25/ found with holographic studies a peak in the local heat transfer coefficient approximately 90 degrees upstream of the stagnation point with natural convection around elliptical cylinders as Fig. 18 demonstrates, whereas the heat transfer coefficient around a cylindrical tube is continuously falling under the condition of natural convection. Elliptical cylinders provide a much higher heat transfer surface in a given volume than cylindrical ones and their heat transfer coefficients are not much worse. They have the benefit of lower flow resistance.

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FIGURES

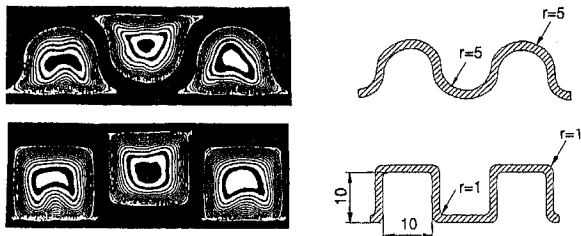


Figure 1 Interferograms of square ducts with radii at the corners

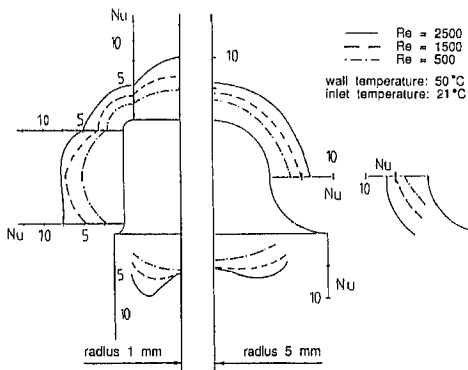


Figure 2 Distribution of local Nusselt number in square ducts with corner radii of 1 and 5 mm

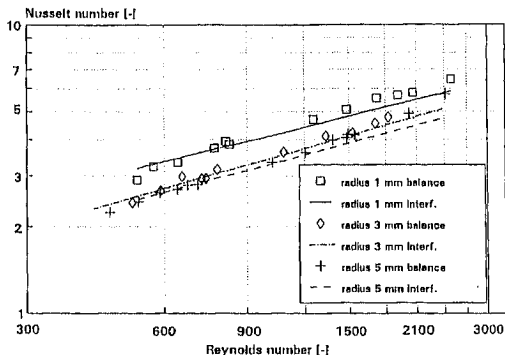


Figure 3 Average Nusselt number as a function of Reynolds number for square ducts with corner radii of 1, 3 and 5 mm

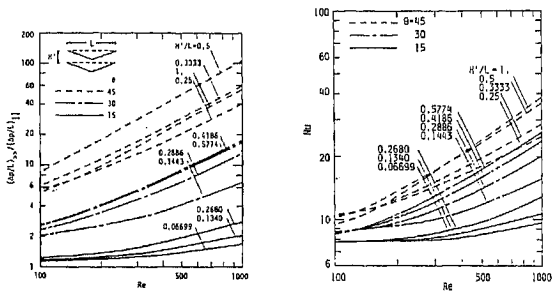


Figure 4 Pressure drop increase and Nusselt number enhancement in zigzag elements /7,8/

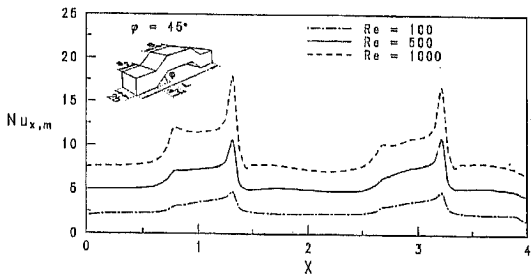


Figure 5 Circumferential average local Nusselt number /9/

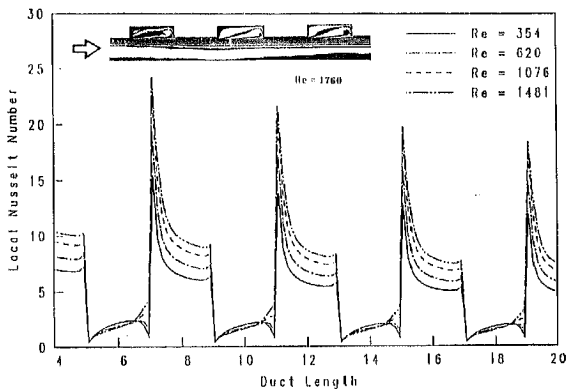


Figure 6 Local Nusselt number in a grooved channel /10,12/

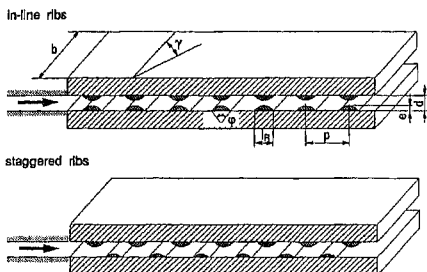


Figure 7 Staggered and in-line arrangement of cylindrical segments

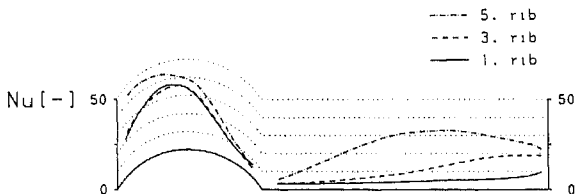


Figure 8 Local Nusselt number distribution at and behind the 1., 3., and 5. rib for in-line arrangement

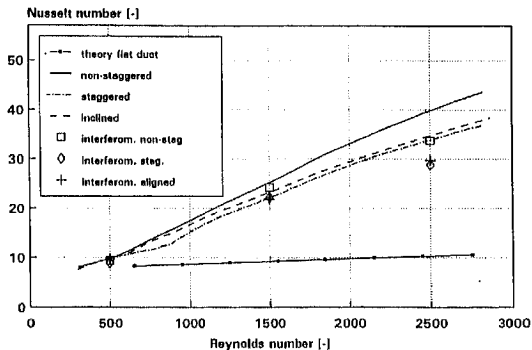


Figure 9 Average Nusselt number as a function of Reynolds number for a rectangular duct with cylindrical segments

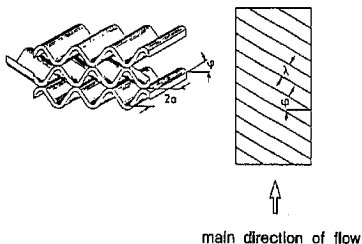


Figure 10 Duct formed by wavy-sinusoidal walls /13,14/

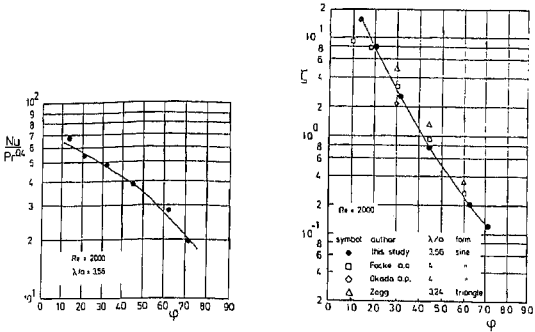


Figure 11 Influence of inclination angle on heat transfer and pressure losses /13,14/

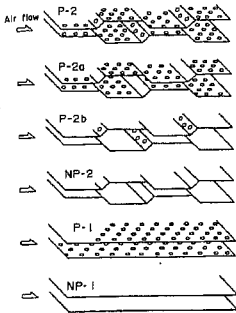


Figure 12 Arrangements of perforating holes in meandering ducts /15/