Investigation of flow and heat transfer in corrugated-undulated plate heat exchangers

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Abstract  An experimental and numerical investigation of heat transfer and fluid flow was conducted for corrugated-undulated plate heat exchanger configurations under transitional and weakly turbulent conditions. For a given geometry of the corrugated plates the geometrical characteristics of the undulated plates, the angle formed by the latter with the main flow direction, and the Reynolds number were made to vary. Distributions of the local heat transfer coefficient were obtained by using liquid-crystal thermography, and surface-averaged values were computed; friction coefficients were measured by wall pressure tappings. Overall heat transfer and pressure drop correlations were derived.

Three-dimensional numerical simulations were conducted by a finite-volume method using a low-Reynolds number $k-e$ model under the assumption of fully developed flow. Computed flow fields provided otherwise inaccessible information on the flow patterns and the mechanisms of heat transfer enhancement.

List of symbols

- $A_c, A_u$: cross-section of corrugations/undulations normal to furrows, $m^2$
- $D_{eq}$: hydraulic diameter, $4S/V$, m
- $f$: equivalent friction coefficient, see Eqn. (6)
- $h$: convective heat transfer coefficient, $q/(T_w - T_i)$, W m$^{-2}$K$^{-1}$
- $H$: external height of corrugation or undulation, m
- $H_i$: internal height of corrugation or undulation, m

Greek symbols

- $\theta$: angle between plate and main flow direction, rad
- $\lambda, \lambda_p$: thermal conductivity of fluid and of plates, W m$^{-1}$K$^{-1}$
- $\mu$: fluid viscosity, N s m$^{-2}$
- $\rho$: fluid density, kg m$^{-3}$
- $\sigma$: uncertainty
- $\psi$: inclination of furrow sides in straight-circular approximation, rad

Subscripts

- b: thermo-static bath
- c: corrugated plate
- f: fluid
- t: turbulent
- u: undulated plate
- w: wall

1 The role of experiments and CFD in the design of compact heat exchangers

Plate heat exchangers, though less common than the more traditional shell-and-tube exchangers, are widely used in the chemical, process and food industry [1]. Closely related to surface-type plate exchangers are also the regenerative air preheaters used in fossil-fuelled power stations to recover the heat of the fuel gases by preheating the combustion air [2].

Recently, plate heat exchangers have been the subject of increasing research due to the attractive possibility of improving their performances, thus reducing volume and cost, by a relatively simple rational re-design of the basic heat transfer elements [3].
For any given overall flow rate, it is desirable to attain the highest possible heat transfer coefficients (thus allowing the exchanger to be more compact, light and cheap) and the lowest possible frictional pressure losses (thus reducing the necessary pumping power). However, since more twisted configurations usually increase both heat transfer and friction, a compromise must be sought by weighting the two performance parameters in a way which depends, in turn, upon the specific application. The optimization problem in rotary air heaters was discussed, for example, in [4].

Other considerations – related to the manufacturing, operation, and maintenance of the exchanger – may also be important. For example, in rotary air preheaters the geometry of the flow passages must be such that the ducts do not get obstructed by fouling too readily, and that they can be easily cleared from fouling and deposits when these occur by using air or steam jets (“sootblowing”). Aspects like leakage between the hot and cold streams, corrosion, and structural stiffness must also be taken into account.

Thus, the efficient design of compact exchangers requires a detailed knowledge of heat transfer and friction coefficients as functions of the geometry of the flow passages and of the flow rate. Moreover, it would be desirable to understand the mechanisms by which changes in the geometry or in the flow rate affect the exchanger’s performance; this, in turn, requires a knowledge of the local distribution of the heat transfer coefficient over the active walls, as well as of the whole flow field (including mean and fluctuating components, i.e. turbulence quantities).

Experimental techniques such as liquid crystal thermography, combined with digital image processing, do provide the complete distribution of the heat transfer coefficient [5]. The investigation of the flow field is much more difficult, due to the complex geometry of the flow passages [6]. Not surprisingly, heat transfer results are much more common than flow measurements in the literature.

Computational Fluid Dynamics can be of great help under such circumstances. Its main advantage lies in its unique capability to give a holistic picture of flow field, temperature field and local heat transfer rates, and in the flexibility by which changes in geometry, Reynolds number or boundary conditions can be described and their effects can be studied.

However, laboratory experiments are still necessary in order to validate and calibrate models and numerical methods, and a complete picture of flow and heat transfer in the exchanger’s passages can only be given by parallel experimental and computational studies.

2 Geometries considered in this study

2.1 The corrugated-undulated configuration

The configurations considered in the present study can be reconducted to the basic corrugated-undulated geometry shown in detail in Fig. 1. Plates bearing shallow furrows (undulated plates, subscript “u”) alternate with plates bearing deeper furrows (corrugated plates, subscript “c”), thus delimiting open flow passages which intersect one another at an angle.

The upper and the lower plate of a generic couple share the same basic geometry, which is characterized by the following parameters:

- pitch \( P \);
- thickness \( s \) of the plate;
- external height \( H \) or, equivalently, internal height \( H_i = H - s \);
- angle \( \theta \) between the main flow direction and the axes of the furrows.

In industrial applications, corrugations are usually aligned with the main flow direction (\( \theta_c = 0 \)), while undulations form an angle \( \theta_u \) varying from a few degrees to 90°. A “see-through” geometry is therefore obtained.

If the two plates of each couple are identical and form equal but opposite angles with the main flow direction (\( \theta_u = \theta_c \)), the so called crossed-corrugated configuration is obtained. Therefore this latter, which has been the subject of previous papers by the present authors [6–8],

![Fig. 1a-c. Geometry of corrugated-undulated (CU) heat transfer elements. a End view of element pack; b cross section and planform of undulated plate; c cross section and planform of corrugated plate](image-url)