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Heat transfer enhancement by pin elements

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Abstract

Heat transfer enhancement is an active and important field of engineering research since increases in the effectiveness of heat exchangers through suitable heat transfer augmentation techniques can result in considerable technical advantages and savings of costs. Considerable enhancements were demonstrated in the present work by using small cylindrical pins on surfaces of heat exchangers. A partly quantitative theoretical treatment of the proposed method is presented. It uses simple relationships for the conductive and convective heat transfer to derive an equation that shows which parameters permit the achievement of heat transfer enhancements. Experiments are reported that demonstrate the effectiveness of the results of the proposed approach. It is shown that the suggested method of heat transfer enhancements is much more effective than existing methods, since it results in an increase in heat transfer area (like fins) and also an increase in the heat transfer coefficient.

Keywords: Heat exchanger; Pin fins; Heat transfer enhancement

1. Introduction and aim of work

Heat exchangers are used in various industrial applications and are devices that are installed to permit the transfer of thermal energy between two (or more) fluids at different temperatures without having direct contact. Parallel-flow, counter-flow and cross-flow heat exchangers are in operation and their overall performance has been treated in many text books, e.g., see [1,2]. The major challenges to the design of a heat exchanger are to make it compact, i.e., to achieve a high heat transfer rate and, at the same time, to allow its operation with a small power loss. These aims of research and development have not changed over the years but, most recently, high energy and material costs have resulted in increased efforts to design and produce more and more efficient heat exchanger equipment. In connection with this, investigations into heat transfer enhancements have attracted new interest including at Institute of Fluid Mechanics, Erlangen. Heat transfer increases have been studied experimentally and theoretically and first results are reported in the present paper.

For heat transfer enhancements in heat exchangers, active and passive methods have been employed. For active methods, some external power is needed to achieve the attempted heat transfer enhancement, usually from a flowing fluid to a heat exchanger wall. If the power is taken from the actual fluid flow, it is possible to drive with this flow instabilities, e.g., see [3], to yield an increased heat transfer coefficient. To increase the heat transfer coefficient, a common passive method is to employ

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Nomenclature

$\dot{q}$	heat transfer per unit area
$\dot{Q}$	heat transfer
$\dot{V}$	volume flow rate
A	surface area
D	diameter of the outer tube
d	diameter of the inner tube
N	number of pin rows
Eu	Euler number
E	specific power input (pumping power per unit bare area)
h	convective heat transfer coefficient
k	thermal conductivity
l	length of tube
Nu	Nusselt number
p	pressure
Pr	Prandtl number
Re	Reynolds number
T	temperature
U	overall heat transfer coefficient
u	mean flow velocity

Greek symbols

ρ	density
φ	coverage ratio
Δ	difference

Subscripts

h	hydraulic diameter
b	bare plate
a	air and augmented heat transfer
s	solid
w	water
fr	free part of the plate (not covered by pins)
bp	base of the pins
i	inner diameter
o	outer diameter
in	inlet
out	outlet

turbulence promoters of different geometry, e.g., see [4,5]. To achieve both an increase in heat transfer coefficient and a moderate increase in heat transfer area, flow inserts such as twisted tapes, wire coils or cross-bar grids are used, e.g., see [4,5]. Recently, the employment of dimples has been suggested, e.g., see [6], to increase the effective heat transfer from surfaces. Extended surfaces characterized with high heat transfer coefficients and a substantial increase in heat transfer area provides appreciable heat transfer enhancement compared with the all above-mentioned passive techniques. Most recently, Dewan et al. [7] presented a review of passive heat transfer augmentation techniques. They mainly discussed the use of twisted tapes and wire coils in laminar and turbulent flow regimes. They also considered other passive techniques, such as ribs, fins and dimples, and showed that the effectiveness of a particular type of technique depends on Reynolds number and other flow properties.

While seeking heat transfer enhancement, apart from the utilization of various surface enhancement elements, efforts have also been made to select an optimal flow arrangement within the heat exchanger in order to obtain the maximum advantage for a given heat exchanger configuration. It may be noted that in a counter-flow arrangement of a heat exchanger, the outlets of fluid streams lie at opposite ends and this enables the outlet temperature of cold fluid to rise above the outlet temperature of the warm fluid. This is not possible in a parallel-flow arrangement, and hence as far as the heat transfer rate is concerned, the counter-flow heat exchangers are superior to the parallel type [8]. Nevertheless, it is not possible to use a counter-flow heat exchanger in all practical situations. Therefore, a common arrangement in practice is a heat exchanger with a cross-flow arrangement and this is characterized by a better temperature distribution compared with the parallel-flow heat exchanger, but the temperature distribution in such a heat exchanger is not as good as in the case of a counter-flow arrangement.

Both effective surface enhancement elements and the optimal flow arrangement were employed during the experimental investigation of the pin fin heat exchanger described in the present work.

2. Estimation of heat transfer enhancements

First for some approximate theoretical considerations of the heat transfer from a surface, the molecularly conducted heat from a plate without any heat transfer enhancement element (bare plate) can be given as

$$\dot{q}_{\mathrm{b}} = -k_{\mathrm{a}} \left(\frac{\partial T}{\partial y} \right)_{\mathrm{a},y=0} \tag{1}$$

where k_{a} is the thermal conductivity of the air, $\left(\frac{\partial T}{\partial y} \right)_{\mathrm{a},y=0}$ is the temperature gradient at the air side of the wall–air interface and $\dot{q}_{\mathrm{b}}$ is the heat transfer rate per unit area of bare plate.

When elements for heat transfer augmentation are placed on the surface to cover an area φA_{b}, the area for the heat transfer from the solid surface to the fluid