Available online at www.sciencedirect.com
SCIENCE DIRECT®

ELSEVIER

International Journal of Heat and Mass Transfer 48 (2005) 4738–4747

International Journal of
HEAT and MASS TRANSFER

www.elsevier.com/locate/ijhmt

Heat transfer enhancement by pin elements

N. Sahiti, F. Durst *, A. Dewan

LSTM-Erlangen, Institute of Fluid Mechanics, Friedrich-Alexander-Universität Erlangen-Nürnberg, Cauerstrasse 4, D-91058 Erlangen, Germany

Received 16 November 2004; received in revised form 25 May 2005
Available online 22 August 2005

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

* Corresponding author. Tel.: +49 9131 852 9501; fax: +49 9131 852 9503.
E-mail address: durst@lstm.uni-erlangen.de (F. Durst).

0017-9310/$ - see front matter © 2005 Elsevier Ltd. All rights reserved.
doi:10.1016/j.ijheatmasstransfer.2005.07.001

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 (air in the present paper) decreases to $(1-\varphi)A_\mathrm{b}$, where A_b denotes the surface area of the bare plate. Hence, to estimate the heat transfer enhancement by augmentation elements, we may write

$$\dot{q}_\mathrm{a} = \dot{q}_\mathrm{fr} + \dot{q}_\mathrm{bp} = -(1-\varphi)k_\mathrm{a}\left(\frac{\partial T}{\partial y}\right)_{\mathrm{a},y=0} - \varphi k_\mathrm{s}\left(\frac{\partial T}{\partial y}\right)_{\mathrm{s},y=0} \tag{2}$$

where $\dot{q}_\mathrm{a}$ is the augmented heat flux (Figs. 1 and 2), $\dot{q}_\mathrm{fr}$ is the heat flux at the interface between solid and fluid (air in the present paper), $\dot{q}_\mathrm{bp}$ is the heat flux through the base area of pin, k_a and k_s is the thermal conductivity of air and solid material, respectively, and $\left(\frac{\partial T}{\partial y}\right)_{\mathrm{a},y=0}$, $\left(\frac{\partial T}{\partial y}\right)_{\mathrm{s},y=0}$ are the temperature gradients at the interface between the free area and air and between the base plate and base area of pin fin, respectively, and φ denotes the ratio of the base pin area and bare plate area (coverage ratio).

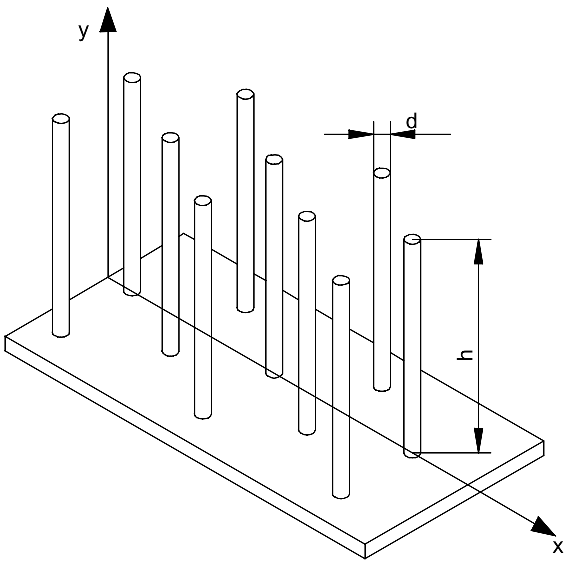

Fig. 1. Heat transfer area covered by pins.

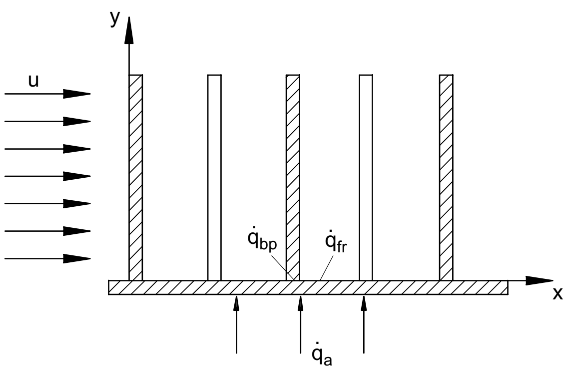

Fig. 2. Heat fluxes from the free and pin base area.

In order to achieve a high heat transfer rate, one should employ a large number of small elements with a small coverage ratio φ (about 5%), resulting in a substantially increased heat transfer area but without an excessive pressure drop. Therefore Eq. (2) can be rewritten in the following approximate form:

$$\dot{q}_\mathrm{a} \approx -k_\mathrm{a}\left(\frac{\partial T}{\partial y}\right)_{\mathrm{a},y=0} - \varphi k_\mathrm{s}\left(\frac{\partial T}{\partial y}\right)_{\mathrm{s},y=0} \tag{3}$$

Hence, the ratio of total heat flux from a base plate with pins and the bare base plate takes the form

$$\frac{\dot{q}_\mathrm{a}}{\dot{q}_\mathrm{b}} \approx 1 + \varphi\frac{k_\mathrm{s}}{k_\mathrm{a}}\frac{\left(\frac{\partial T}{\partial y}\right)_{\mathrm{s},y=0}}{\left(\frac{\partial T}{\partial y}\right)_{\mathrm{a},y=0}} \tag{4}$$

In order to obtain a possible large value of the ratio, apart from a small coverage ratio one needs to chose a fin which has a large heat transfer surface area and a large heat transfer coefficient. Closely spaced pins with a small diameter and large length to diameter ratio have such features. If heat transfers from pin and flat surface are represented in terms of heat transfer coefficients for the pin and flat surface and respective temperature differences, one can prove that for fluid velocities of the order of 2 m/s and the pin height to pin diameter ratio of the order of 15, the ratio of the two gradients in Eq. (4) becomes approximately 0.1. Further, by assuming the material of the pin to be copper with $k_\mathrm{s} = 380$ W/m K and with thermal conductivity of air $k_\mathrm{a} = 0.026$ W/m K one can show that the following heat transfer ratio results:

$$\frac{\dot{q}_\mathrm{a}}{\dot{q}_\mathrm{b}} \approx 70 \tag{5}$$

This ratio indicates that it is possible, in principle, to increase the heat transfer of a flat surface by a factor of about 70 by placing pin-type elements on to it. Increases of this kind are not achievable by any other means or only by excessive increases in the pressure loss of the fluid flow passing through the heat enhancement elements. The large increase achievable with pin-type heat transfer enhancement elements encouraged the authors to carry out the experimental investigations described below using a counter-flow pipe heat exchanger.

3. Experimental investigations

3.1. Heat exchanger and test rig

The experimental investigations were aimed at demonstrating that heat exchanger surfaces can be built with pin-type enhancement elements that permit the surface heat transfer to be enhanced by a factor of about 70, as estimated above. To carry out the experiments, a counter-flow heat exchanger was chosen, embracing two axis-aligned pipes, the inner one of copper and the outer one of stainless steel. Around the inner pipe a copper wire mesh providing pin-like fins with diameter 0.7 mm and length 28.2 mm was wrapped (Fig. 3). The wires were arranged in a somewhat staggered way and had a mean distance of 3 mm in the stream-wise direction and 6.5 mm in the span-wise direction, measured in a mean annular diameter of 70.8 mm. This double-pipe heat exchanger was mounted in a test rig consisting of an open-end wind tunnel as sketched in Fig. 4. The wind tunnel, set according to DIN norms, was able to provide a flow rate up to 8 m^3/s. The various flow rates of air for the experiments were measured by using a U-tube manometer attached to the wind channel.

To obtain warm fluid for obtaining a certain heat flux, a standard instantaneous water heater was employed. The warm water flow rates were measured with a standard water supply flow meter manufactured by Minol Messtechnik. Temperature changes on the cold fluid side (air side) were recorded by PT100 resistance thermometers manufactured by OMEGA Newport with a perforated coating. On the water side the temperature was measured by the PT100 resistance thermometers but with a normal coating.

To avoid heat losses from the wire mesh into the surroundings and to avoid erroneous readings of temperatures at the inlet and outlet of air due to radiation and convection from the bands of the inner copper pipe, the relevant parts were insulated using a mineral fibre mat with thermal conductivity $k = 0.1$ W/m K (Fig. 5). The sequential readings of four PT100 thermometers used during the experimentation were taken from a reading device with a display and a channel switching unit. In order to obtain high heat transfer rates, the heat exchanger was run in the counter-current mode.

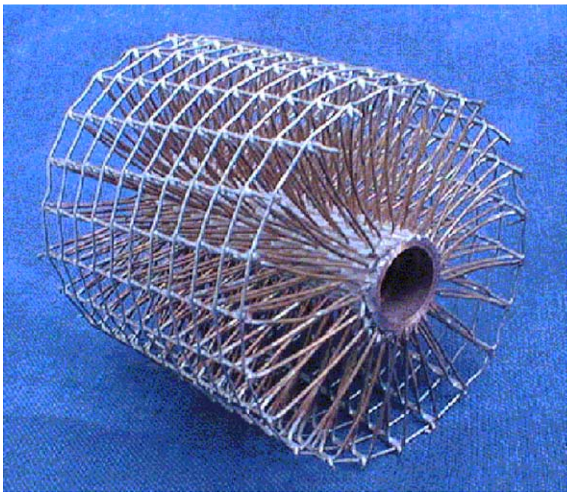

Fig. 3. Core part of the pin fin heat exchanger.

3.2. Experimental procedure and data reduction

A simple way to measure the flow rate of air in the present test facility is by reading the pressure difference at the inlet nozzle and converting it into the mean nozzle inlet velocity. However, this technique could not be applied because of small pressure differences corresponding to the low flow rates considered. Therefore, we used a U-tube manometer connected to the channel wall, which was previously calibrated using a Pitot probe. The velocity profile obtained by the use of Pitot tube was used for the determination of the mean velocity in the tube cross-section and volume flow rate. For each flow rate a reading of the difference in the U-tube manometer was made to obtain a calibration curve.

To cover a wide range of conditions relevant to heat exchangers, the measurements were taken at 15 flow rates of air corresponding to a water flow rate of

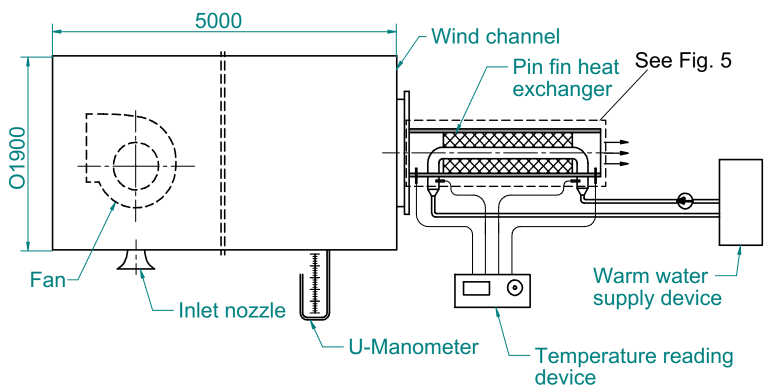

Fig. 4. Schematic of the pin fin heat exchanger measurement setup.

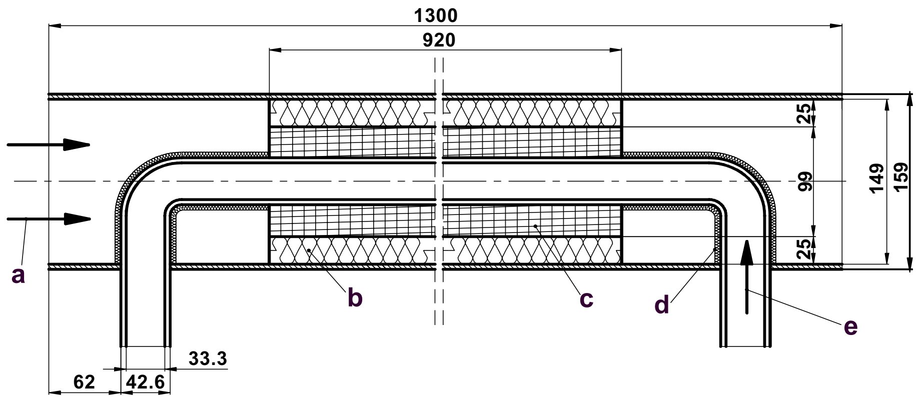

Fig. 5. Geometry of the pin fin heat exchanger (dimensions in mm): (a) cold air; (b) insulation (air-side); (c) wire-like pin fins; (d) insulation (water-side); (e) hot water.

3 l/min. Each experiment was repeated three times to obtain a good representative average value. During the measurements the flow rates of air were chosen in steps of 20 mm of water column of the U-tube. Thus a flow range from 0.016 to 0.062 m^3/s could be obtained. Owing to the heat introduced by the fan, the inlet temperature of the air varied between 19 and 22.5 °C. Higher inlet temperatures were recorded at higher flow rates owing to higher heat losses from the fan motor. At the outlet of heat exchanger, air temperatures between 64 and 51 °C were observed. The inlet water temperature was around 76.7 °C with a variation of less than 0.65%, whereas the outlet water temperature reached values between 69 and 73 °C.

Pressure drop measurements were carried out under isothermal conditions [9] to avoid the fluid property changes due to temperature variations in the heat exchanger. The pressure drop in the heat exchanger corresponding to the flow rates was measured by taking static pressure readings 5*d* in front of the heat exchanger on a differential manometer manufactured by Novodirect. The static pressure on the heat exchanger outlet was considered to be same as the ambient pressure, since the heat exchanger during the pressure measurement procedure was kept at the end of the pipe.

In order to obtain the heat transfer coefficient on the air side, one needs to know the overall heat transfer coefficient, which, considering the heat exchanger as an adiabatic system, can be calculated by the expression

$$U = \frac{\dot{Q}}{A \cdot \Delta T_{\mathrm{lm}}} \tag{6}$$

where A is the heat transfer area. In the present experiments A represents the outside surface area of the inner tube with $d_{\mathrm{o}} = 42.6$ mm. The logarithmic mean temperature difference in the upper equation is calculated by the expression

$$\Delta T_{\mathrm{lm}} = \frac{(T_{\mathrm{w\,out}} - T_{\mathrm{a\,in}}) - (T_{\mathrm{w\,in}} - T_{\mathrm{a\,out}})}{\ln\left(\frac{T_{\mathrm{w\,out}} - T_{\mathrm{a\,in}}}{T_{\mathrm{w\,in}} - T_{\mathrm{a\,out}}}\right)} \tag{7}$$

The heat flux in Eq. (6) was calculated using the enthalpy difference on the water side, because parameters were measured accurately on the water side

$$\dot{Q} = \rho_{\mathrm{w}} c_{p\mathrm{w}} \dot{V}_{\mathrm{w}} (T_{\mathrm{w\,in}} - T_{\mathrm{w\,out}}) \tag{8}$$

where ρ_{w}, c_p and $\dot{V}_{\mathrm{w}}$ are the density, thermal capacity and volume flow rate of the water, respectively.

By knowing the overall heat transfer coefficient, one can derive the heat transfer coefficient on the air side by applying the following expression:

$$h_{\mathrm{a}} = \left(\frac{1}{U} - \frac{d_{\mathrm{o}}}{h_{\mathrm{w}} d_{\mathrm{i}}} - \frac{d_{\mathrm{o}}}{2k_{\mathrm{c}}} \ln \frac{d_{\mathrm{o}}}{d_{\mathrm{i}}}\right)^{-1} \tag{9}$$

where d_{i} and d_{o} denote the inner and outer diameters of the inner heat exchanger tube and h_{w} represents the heat transfer coefficient on the water side.

There are well-established equations derived empirically for the turbulent flow regime or analytically for the laminar flow regime describing the heat transfer on the water side [2,8,10]. However, during the present experiments it was found that the mean Reynolds number $Re \approx 4757$ corresponds to the transition region. Therefore, most expressions reported in the literature would fail to provide physically meaningful values for the heat transfer coefficient in the inner tube of the heat exchanger. An appropriate equation for Nu_{w} on the water side was found to be the following:

$$Nu_{\mathrm{w}} = 0.036 Re^{0.8} Pr^{1/3} (d_{\mathrm{i}}/l)^{0.055} \tag{10}$$

where l represents the length of the inner tube (920 mm in the present case). Eq. (10) proposes a Nusselt number for developing turbulent flows [10] and it also provides accurate results for the transition region of the flow in the inner tube of the present work. Obviously the inlet conditions of the water flow into the heat exchanger section (such as sharp bends) cause an early transition of the laminar flow into turbulent flow.

The heat transfer coefficient on the water side (Eq. (9)) was derived based on the Nusselt number using the following expression:

$$h_w = \frac{k_w Nu_w}{d_i} \tag{11}$$

where k_w denotes the thermal conductivity of water.

The above-described procedure was applied to the derivation of the heat transfer coefficient on the air side by taking into account variation in fluid properties due to the temperature variation. All fluid properties were taken from the VDI Wärmeatlas [11]. The heat transfer results were presented in the dimensionless form by using Nusselt number on the air side calculated as follows:

$$Nu_a = \frac{h_a D_h}{k_a} \tag{12}$$

where $D_h = D_i - d_o$ represents the hydraulic diameter of ring space occupied by pins and D_i denotes the inner diameter of the outer pipe. The same hydraulic diameter was used for the calculation of the Reynolds number on the air side.

The pressure drop Δp, which in addition to friction and form drag pressure losses includes the entrance and exit losses, was expressed as a function of the flow rate in the dimensionless form by using the Euler number

$$Eu = \frac{2\Delta p}{\rho_a u_a^2 N} \tag{13}$$

where ρ_a represents the mean air density, u_a the mean air velocity and N the number of pin rows in the streamwise direction.

For comparison purposes, heat transfer and pressure drop estimations were also made for a smooth double-pipe heat exchanger, in the counter-flow arrangement, with the same geometric characteristics as those given in Fig. 5. The Nusselt number for the convective heat transfer in the air flowing through the smooth annular space was derived based on the following equation (VDI Wärmeatlas [11]):

$$Nu = 0.86\left(\frac{d_o}{D_i}\right)^{-0.16} \frac{(\xi/8)RePr}{1 + 12.7\sqrt{\xi/8}(Pr^{2/3} - 1)} \times \left[1 + \left(\frac{D_h}{l}\right)^{2/3}\right] \tag{14}$$

where $\xi = (1.8 \log Re - 1.5)^{-2}$ and D_h = hydraulic diameter of the annular space.

For the determination of the pressure drop in the smooth double-pipe heat exchanger, the following equation was used:

Table 1
Uncertainty of the individual sources

Variable	Symbol	Uncertainty (%)
Inner diameter of outside tube	D_i	0.1
Outer diameter of inside tube	d_o	0.2
Inner diameter of inside tube	d_i	0.3
Difference pressure drop (Digital manometer)	Δp	(1.6–4.6)
Difference pressure drop (U-tube manometer)	Δp	(3.2–8.9)
Water flow rate	V_w	2.0
Inlet air temperature	$t_{a\,in}$	0.9
Outlet air temperature	$t_{a\,out}$	0.4
Inlet water temperature	$t_{w\,in}$	0.3
Outlet water temperature	$t_{w\,out}$	0.3

$$\Delta p = \rho \frac{l}{D_h} \frac{fu^2}{2} \tag{15}$$

where the friction factor f is defined based on the Haaland equation [12]

$$f = \left[-1.8 \log\left(\frac{k}{3.7D_h}\right)^{1.11} + \frac{6.9}{Re}\right]^{-2} \tag{16}$$

where k is the pipe wall roughness (=0.0015 mm).

3.3. Uncertainty analysis

The uncertainty analysis was performed by using the method of Kline and McClintock [13]. By this method the uncertainty of a variable R which is a function of independent variables $x_1, x_2, \ldots, x_n$, can be estimated by taking root-sum-square of the contributions of individual variables. The individual uncertainties of different variables measured in the present work are provided in Table 1. The effects of individual variables resulted in the following uncertainties for the dimensionless variables measured in the present work: $Re = (3.2–10)\%$, $Nu = (16–21.1)\%$, and $Eu = (6.6–20.5)\%$.

4. Discussion of results

One objective of the present experiments was to demonstrate that an enhancement of the heat transfer rate by an increase in fluid flow velocity (and thus resulting in flow turbulence) is not an effective method since the heat flux varies with the velocity approximately as $\dot{Q} \approx u^{0.5}$ whereas the pressure drop varies as $\Delta p \approx u^2$. Thus due to an increase in velocity and hence Reynolds number, the pressure losses would rise faster than the rise of the heat flux. In order to bring out the relatively weak influence of the fluid velocity on the heat transfer rate, Nu for the smooth double pipe with the same dimensions as that of the present heat exchanger is also

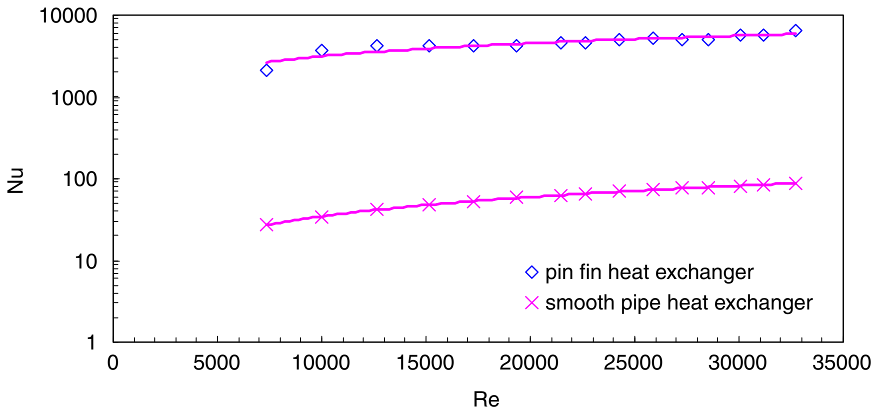

Fig. 6. *Nu* as a function of *Re* for pin fin heat exchanger and a smooth pipe heat exchanger.

presented in Fig. 6. The results clearly show the advantage of using pin fins to increase *Nu*. It is important to note that increased flow velocities result in 2–3 times higher *Nu*, whereas, by employing the pins it is possible to obtain 65–105 times higher values of *Nu* compared with those for the smooth pipe heat exchanger.

However, *Nu* is not the only parameter to assess the performance of a heat exchanger. Rather in the design procedure particular care should be given to the pressure losses as these are directly proportional to the operating costs. It may happen that owing to high pressure losses the expenditure on mechanical pumping power is as much as the enhancement in heat transfer rate of a heat exchanger. Pressure drop in general in all heat exchangers is approximately proportional to ρ^{-2} for laminar or turbulent flow conditions [14]. Therefore, the pressure drop in a heat exchanger with one or both working fluids as a gas is usually critical and thus the pressure drop on the air side was only measured in the present experiments. It was ensured that during the pressure drop measurements no heat transfer took place, in order to prevent the effect of changing fluid properties. For the flow over tube banks or over pin fins, it is convenient to give the pressure drop results in term of pressure loss coefficient or Euler number for tube row, since for such flows the pressure drop varies linearly with number of pin rows crossed by the fluid.

The non-dimensional form of the data presented in Fig. 7 enables one to obtain the pressure drop for geometrically similar heat exchangers but with different lengths and hence different numbers of pin rows. We emphasise again that the plotted *Eu* includes the pressure drop introduced due to the pins (the major part of the pressure drop) in addition to the pressure drop associated with the entrance and exit effects in the heat exchanger.

The dimensionless form of the presentation of heat transfer results is suitable for scaling purposes, e.g., if one needs to apply results from a small test heat exchanger to a real heat exchanger with larger dimensions but within the requirements of similarity analysis. However, in a practical application, this form of presentation is not useful, since for such applications one primarily needs to know heat transfer rates for a given pressure drop or vice versa. Moreover, it is often necessary to choose the most effective fins for the heat transfer

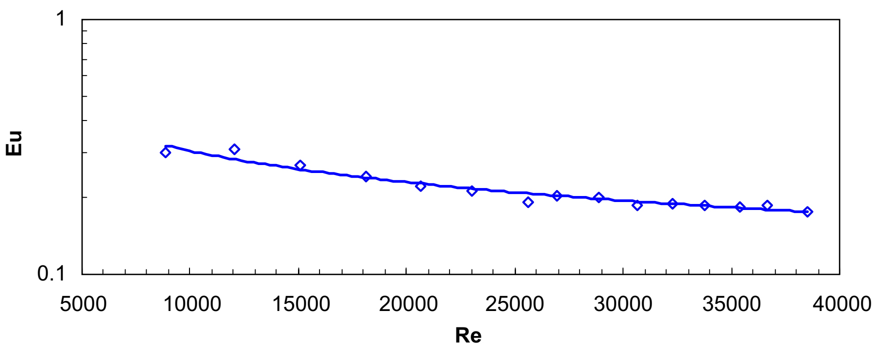

Fig. 7. *Eu* and *Re* correlation for the investigated heat exchanger.

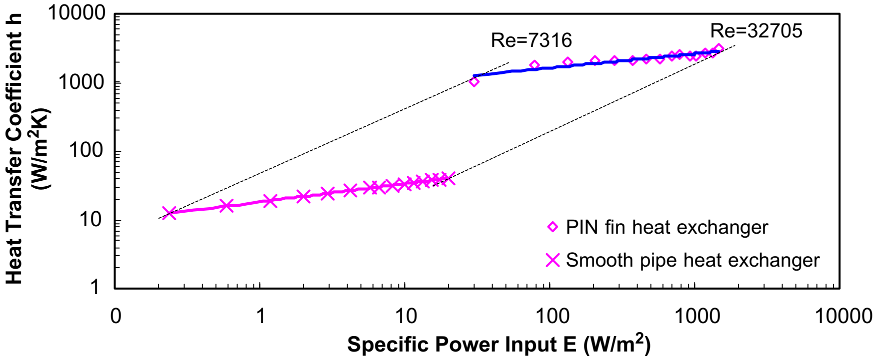

Fig. 8. Heat transfer coefficient vs. power input of the investigated heat exchanger.

enhancement taking into account both thermal and pressure drop characteristics. The way of presenting a huge amount of experimental data from heat exchanger tests is very important and this will be discussed in more detail in a forthcoming publication. Nevertheless, heat transfer and pressure data for both the present pin fin and smooth pipe heat exchangers are presented in Fig. 8 in the form of the heat transfer coefficient (heat transfer per unit bare area and unit temperature difference) as a function of the specific power input (pumping power per unit bare area). We believe that this form of presentation of data provides a very simple and effective way to select the best performing heat transfer surface.

The data in Fig. 8 allow the comparison of the performance of different heat transfer surfaces assuming the same fin height and same fin material. As the heat transfer and power input are based on the bare tube area, no estimation of convective heat transfer area and hydraulic diameter of the flow passage is required in order to compare the surfaces. This is of practical interest as the hydraulic diameter of the flow passage is not uniquely defined and the procedures for the estimation of heat transfer area, such as in a compact heat exchanger may be cumbersome. It may be noted that the influence of both of these parameters as well as other factors, such as fin efficiency and fin thickness, is already lumped into h and E. As the values of the parameters for the heat exchangers under comparison differ considerably, the only way to present them in the same figure is to use a logarithmic scale. Fig. 8 allows a comparison of heat transfer rates for the same bare area and temperature difference vs. power input for both heat exchangers. It may be noted that it is not practical from Fig. 8 to compare the heat transfer coefficients of the two heat exchangers for the same value of Re. However, to avoid the confusion, we have presented the lines of the lowest and the highest Re, based on which one can also compare the heat transfer coefficients of both heat exchangers. Such a comparison shows that the heat transfer coefficients behave similarly to the previously compared Nu (Fig. 6) with a small difference related to the changes in the air properties for the smooth and pin fin heat exchangers.

5. Conclusions, final remarks and outlook

Heat exchangers are usually characterized according to their method of operation, i.e., parallel-flow, counter-flow or cross-flow arrangement. Further considerations include the elements that are used to increase the heat transfer rate in a particular heat exchanger. Several heat transfer enhancement elements, such as ribs, twisted tapes, wire coils, cross-bar grids and dimples, have been suggested in the literature. However, such kinds of elements are usually not very efficient at heat transfer enhancement since they usually increase the heat transfer coefficient but the heat transfer area basically remains constant. Much higher values of the heat transfer rates can be obtained by applying fins of different geometry, such as strip, corrugated or louver fins, since by these elements one can obtain a larger heat transfer coefficient and larger heat transfer area compared with bare plates and plates with enhancement elements mentioned above. High performance of the wavy, strip or louvered fins has been investigated and optimized for a long time, resulting in very compact heat exchangers, particularly in thermal power plants, air conditioning units and the automobile industry. However, we consider that these enhancement elements have reached their limit and no further improvement in heat exchanger performance seems possible. Hence enhancement of heat transfer continues to be a challenging problem in different industrial fields. Therefore, an attempt to improve the performance of heat exchangers by using pin fins was undertaken in the present work. We consider that properly distributed pins with an optimal height to diameter ratio will further substantially improve heat exchanger performance as pin fins are

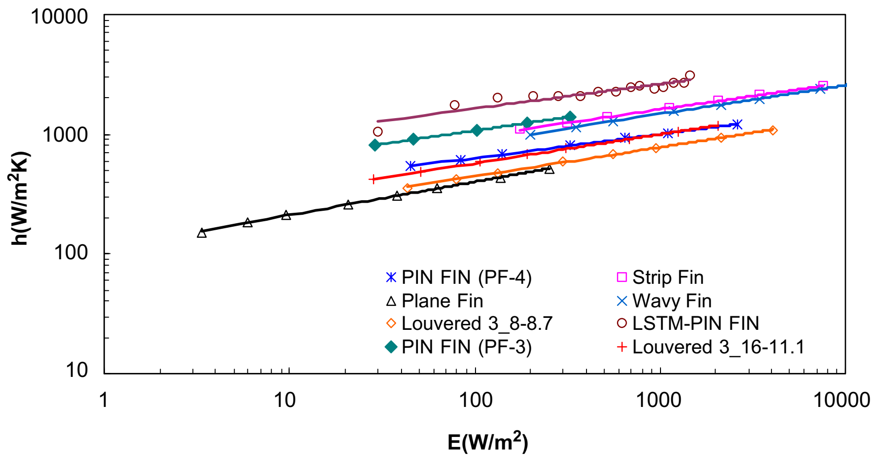

Fig. 9. Comparison of heat transfer coefficient of the investigated pin heat exchanger with some other compact heat exchanger surfaces found in the literature.

characterized with higher heat transfer than strip or louvered fins. In spite of generally higher pressure drop of the flow through the pins compared with fins, the form of presentation of heat exchanger data applied within the present work enables one to prove that for the same pumping power properly arranged pin fins are able to transfer higher heat rates than all other common fin geometries.

An attempt to show the trend of the relative performance of the pin fin heat exchanger surface and other surface geometries was made by using the data for some compact heat exchanger surfaces given in Kays and London [15]. Although the compared heat exchangers are of different geometries, the curves exemplify the advantages of such a kind of presentation of heat exchanger data, e.g., provided there is similar heat exchanger geometry, it allows a direct comparison of heat fluxes per unit bare area and unit temperature difference of all heat exchanger surfaces, taking into account the power input for the same unit bare area. Furthermore, Fig. 9 shows that not all modes of pin fin arrangement perform better than other heat exchanger surfaces. A careful selection of the pin fin arrangement and the ratio of pin height to pin diameter is required to obtain a high-performance pin fin heat exchanger surface. A pin arrangement is considered optimal if the associated pressure drop does not exceed the benefits of enhanced heat transfer rate resulting from high population of the base plate with pins. Otherwise the pin to height diameter ratio influences the pin efficiency and the heat transfer rate of pins. A detailed evaluation of these two parameters is quite complicated owing to their dependence on several other factors and their opposite behavior. Such evaluation is beyond the scope of the present work. However, it is known that for pin fins a good compromise between the pin efficiency and pin heat transfer rate is achieved if the ratio of pin height to pin diameter is of order of 15.

The authors believe that the present work will serve as the basis of further work concerning the optimization of pin fin geometry for application in air heat exchangers and it will encourage the development of simple and cheap procedures to build pin fin heat exchangers for industrial applications.

References

[1] R.L. Webb, Principles of Enhanced Heat Transfer, John Wiley & Sons, New York, 1994.

[2] F. Incropera, D. DeWitt, Introduction to Heat Transfer, fourth ed., John Wiley & Sons, New York, 2002.

[3] M. Greiner, P.F. Fischer, H.M. Tufo, Two-dimensional simulations of enhanced heat transfer in an intermittently grooved channel, Trans. ASME J. Heat Transfer 124 (2002) 538–545.

[4] A.E. Bergles, Techniques to augment heat transfer, in: W.M. Rosenhow, J.P. Hartnett, E.N. Ganic (Eds.), Handbook of Heat Transfer Applications, McGraw-Hill, New York, 1985 (Chapter 3).

[5] A.E. Bergles, The imperative to enhance heat transfer, in: S. Kakac, A.E. Bergles, F. Mayinger, Yüncü (Eds.), Heat Transfer Enhancement of Heat Exchangers, Kluwer Academic Publishers, Dordrecht, The Netherlands, 1998.

[6] L. Zhihua, J.H. Davidson, S.C. Mantell, Heat transfer enhancement using shaped polymer tubes: fin analysis, Trans. ASME J. Heat Transfer 126 (2004) 211–218.

[7] A. Dewan, P. Mahanta, K.S. Raju, P. Suresh Kumar, A review of passive heat transfer augmentation techniques, Proc. Inst. Mech. Eng.: Part A J. Power Energy 218 (2004) 509–527.

[8] H. Baehr, K. Stephan, Wärme- und Stoffübertragung-fourth, fourth ed., Springer-Verlag, Berlin, 2004.

[9] W.M. Kays, A.L. London, Heat-transfer and flow-friction characteristics of some compact heat-exchanger surfaces—Part I, Trans. ASME 72 (1950) 1075–1085.
[10] J.P. Holman, Heat Transfer, eighth ed., McGraw-Hill, New York, 1999.
[11] V.D.I. Wärmeatlas, Springer-Verlag, Berlin, 2002.
[12] F.M. White, Fluid Mechanics, fourth ed., McGraw-Hill, New York, 1999.
[13] S.J. Kline, F.A. McClintock, Describing uncertainties in single-sample experiments, Mech. Eng. 75 (1953) 3–8.
[14] R. Shah, D. Sekulic, Fundamentals of Heat Exchanger Design, Wiley & Sons, New York, 2003.
[15] W.M. Kays, A.L. London, Compact Heat Exchangers, third Reprinted ed., Krieger Publishing Company, FL, USA, 1998.