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ASSESSMENT OF SINGLE PHASE CONVECTION HEAT TRANSFER ENHANCEMENT

Naser Sahiti
University of Prishtina
Faculty of Mechanical Engineering
Kodra e diellit p.n., 10000 Prishtina
Kosovo

Florent Bunjaku
University of Prishtina
Faculty of Education
Rr. Agim Ramadani p.n., 10000
Prishtina, Kosovo

Drenusha Krasniqi
University of Prishtina
Faculty of Mechanical Engineering
Kodra e diellit p.n., 10000 Prishtina, Kosovo

ABSTRACT

Improvements in the efficiency of heat exchangers respectively increase of exchanged heat transfer rate, can lead to substantial cost, space and materials savings. The current paper describes the single phase heat transfer enhancement by simultaneous increase of heat transfer area and heat transfer coefficient. Further, the paper provides an analysis of a simple heat transfer assessment method, demonstrated on pin fins as elements for heat transfer enhancement but which can be applied also to other fin forms. The applicability of the analytical assessment method is checked by experimental investigations of a double-pipe pin fin heat exchanger. The order of the magnitude of heat transfer enhancement obtained experimentally was similar to that obtained analytically. The heat transfer and pressure drop results for the pin fin heat exchanger were compared with the results for a smooth-pipe heat exchanger. It was found that by a direct comparison of Nu and f-factor no conclusion regarding the relative performances could be made. Hence, in the current work a more practical comparison method, consisting in comparison of heat transfer rate per heat exchanger volume versus pumping power per heat exchanger volume is provided.

Keywords: heat transfer enhancement, pin fins, performance comparison

1. INTRODUCTION

The major heat transfer enhancement techniques that have found widely spread commercial application are passive techniques characterized with various types of heat transfer enhancement elements. The most effective heat transfer enhancement can be achieved by using fins as elements for heat transfer surface area extension. Fins are effective in heat transfer enhancement only if they exceed the boundary layer thickness, resulting in a major part of the heat transfer area being exposed to a free fluid stream. The heat transfer coefficient on the extended surface may be lower or higher than that on an unfinned surface, e.g. the plain fins increase the heat transfer surface area but may result in a slight decrease in heat transfer coefficient, whereas interrupted fins (strip, louvered, etc.) provide both an increased surface area and increased heat transfer coefficient.

2. ASSESSMENT OF POTENTIAL FOR HEAT TRANSFER ENHANCEMENT

It has been pointed out that the extension of heat exchanger surfaces by utilization of fins is a widely used method to enhance heat transfer passively. Different fin geometries have been used in practical applications such as plain, strip, louvered and pin fins. It would be helpful to know previously what order of magnitude of heat transfer enhancement can be expected with such fins. The analysis presented here will show that such a prediction is possible by utilizing the basic law of conductivity of heat transfer for the bare surface area and for the surface where the fins are attached to the bare surface area. The analysis is based on the pinned fin surface but it can be applied also to any other fin geometry.

The molecularly conducted heat from a plate without any heat transfer enhancement element (bare plate) can be given as

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

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

When elements for heat transfer enhancement are placed on the heat transfer surface to cover an area φA_b, the area for the heat transfer from the solid surface to the fluid (air in the present work) decreases to $(1-\varphi)A_b$, where A_b denotes the surface area of the bare plate. Hence, to estimate the heat transfer enhancement, we may write

$$\dot{q}_e = \dot{q}_{up} + \dot{q}_{bp} = -(1-\varphi)k_a \left(\frac{\partial T}{\partial y} \right)_{a,y=0} - \varphi k_s \left(\frac{\partial T}{\partial y} \right)_{s,y=0} \tag{2}$$

where $\dot{q}_e$ is the enhanced heat flux, $\dot{q}_{up}$ heat flux through the unpinned portion of the base plate (free portion of the base plate), $\dot{q}_{bp}$ heat flux through the base surface area of pin, k_a, k_s thermal conductivities of air and solid material, respectively, $\left(\frac{\partial T}{\partial y} \right)_{a,y=0}$, $\left(\frac{\partial T}{\partial y} \right)_{s,y=0}$ temperature gradients at the interface between the free surface and air and at the base of the pin fin, respectively, and φ denotes the ratio of the base pin surface area and bare plate surface area (coverage ratio).

In order to achieve a high heat transfer rate, one should employ a large number of small elements with a small coverage ratio φ (5-10%) resulting in a substantially increased heat transfer surface area but without an excessive pressure drop. In that that case $\varphi << 1$, and therefore the ratio of the total heat flux from a base plate with pins and the bare base plate takes the form

$$\frac{\dot{q}_e}{\dot{q}_b} \approx 1 + \varphi \frac{k_s}{k_a} \frac{\left(\frac{\partial T}{\partial y} \right)_{s,y=0}}{\left(\frac{\partial T}{\partial y} \right)_{a,y=0}} \tag{3}$$

For pins as elements to enhance the heat transfer, Eq. (3) takes the form

$$\frac{\dot{q}_e}{\dot{q}_b} \approx 1 + 4\varphi\eta \frac{h_p}{h_b} \frac{l}{d} \tag{4}$$

where, h_p, h_p, η, l, and d refer to the heat transfer coefficient of pins, heat transfer coefficient of bare plate, the pin efficiency, the pin length and pin diameter respectively

For a flow velocity of 2 m/s, by selecting the pin length to diameter ratio to be $l/d \approx 15$ and taking the plate length to be ~120 mm (e.g. for a heat sink), one obtains $\dot{q}_e / \dot{q}_b \approx 70$.

3. HEAT EXCHANGER TEST RIG AND RESULTS

To carry out the experiments, a counter flow heat exchanger was chosen, involving two axis-aligned pipes, the inner one consisting 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. 1).

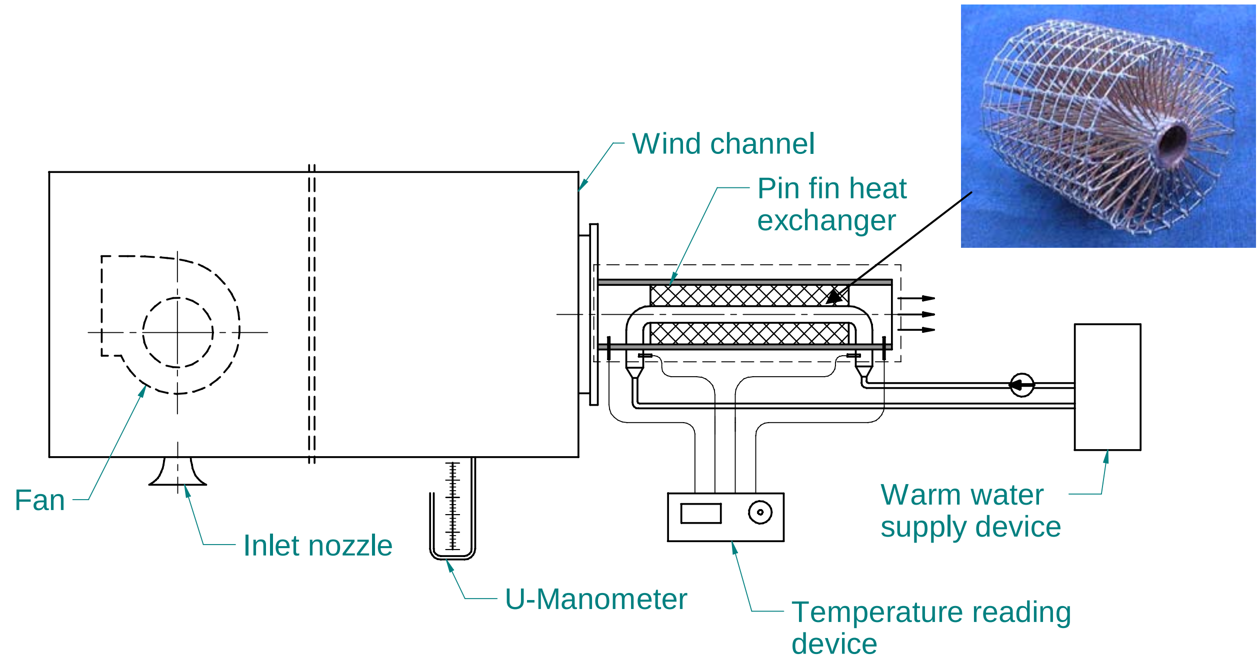

Figure 1. Heat exchanger test rig

The experimental results (Fig. 2) 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. In order to have a complete picture regarding the performance of pin fin heat exchanger compared with a smooth one, the pressure drop characteristics for pin fin and smooth pipe heat exchangers should be compared. Hence similarly to the comparison of heat transfer characteristics in terms of *Nu*, the friction factors *f* of the two considered heat exchangers were compared (Fig. 2). As one can see from Fig. 2, the ratio of the friction factors of the pin fin heat exchanger and the smooth pipe heat exchanger was even larger than the corresponding ratio of *Nu*, e.g. for smaller Re the ratio of friction factors was found to be 180 whereas for higher *R*e this ratio took the value 140.

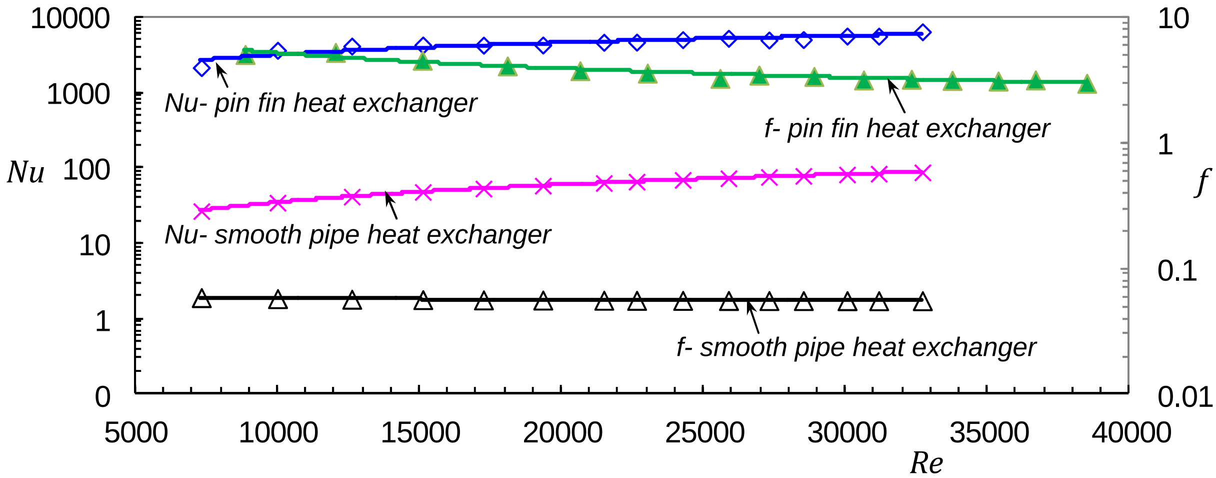

Figure 2. Nu and f as function of Re for pin fin heat exchanger and smooth pipe heat exchanger

Such performance comparison method is confusing, since one might come to the conclusion that utilization of fins is not recommended for heat exchanger performance improvement. The reason lies in the fact that the dimensionless form of 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 enhancement taking into account both thermal and pressure drop characteristics. Therefore, for heat exchanger performance comparison, a more meaningful direct plot of the heat transfer rate per unit volume $\dot{q}_v$ versus the required pumping power per unit heat exchanger volume e_v is recommended.

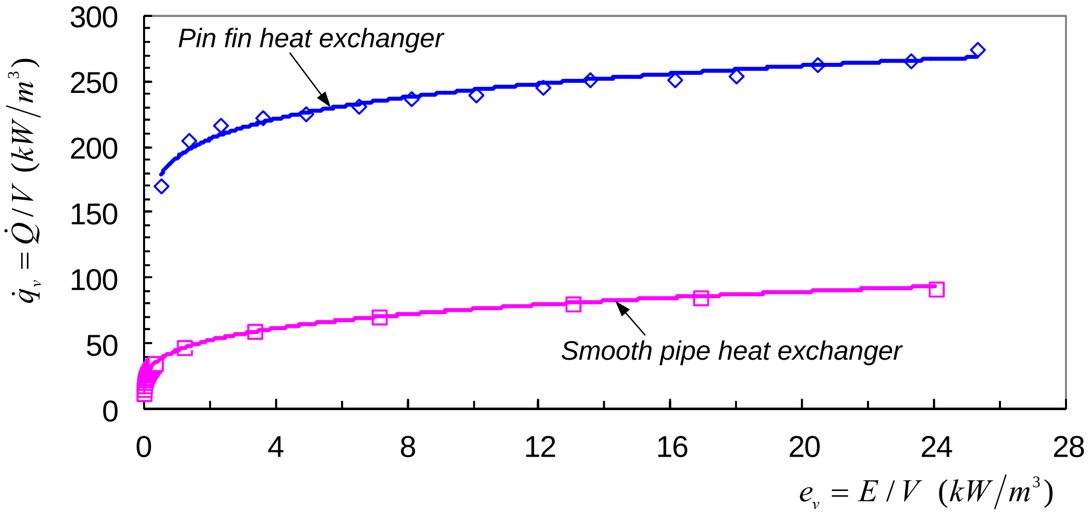

Figure 3. Heat exchanger performance diagram based on heat transfer rate versus pumping power input per unit heat exchanger volume.

4. CONLUSIONS

Current paper has shown that simple analytical calculation might provide insight on potential of particular fin forms for heat transfer enhancement. It is demonstrated that under certain conditions, a heat transfer enhancement factor up to 70 might be achieved with pin fins compared to heat exchanger with no fins. Similar ratios are obtained by experimental investigation of a pin fin heat exchanger. It is also demonstrated that a direct comparison of heat transfer factors such as *Nu*-numbers and friction factors *f* for heat exchangers of different types might lead to confusion regarding the advantages of fins for heat transfer enhancement. Therefore in current paper an alternative performance comparison method which provides the comparison of heat transfer rate per unit heat exchanger volume against pumping power per heat exchanger volume is recommended and practically demonstrated.

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