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Strategy for selection of elements for heat transfer enhancement

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Abstract

The present paper points out that the selection of elements for heat transfer enhancement in heat exchangers requires a methodology to make a direct comparison of the performances of heat exchanger surfaces with different elements. Methods of comparison used in the past are, in many respects, approximate and hence fail to predict accurately the relative performance of conventional heat exchanger surfaces operated with different heat exchanger elements. Owing to the direct use of the Colburn factor for performance assessment, these methods over-predict the relative performance of heat exchangers. In the present paper, a more consistent comparison method is presented and is demonstrated to work by comparison of the performance of an experimentally investigated pin fin heat exchanger with that of a smooth pipe heat exchanger. The method yields results that belong to the volume goodness factors group. It represents a practical approach, as it is applicable to all kinds of heat exchanger surfaces and does not require the conversion of the experimental data in terms of Nusselt number and friction factor for comparison purposes. The present work demonstrates that the suggested method can also be used for performance comparison of existing heat exchanger surfaces with available heat transfer and pressure loss data.

Keywords: Comparison method; Heat exchanger performance; Heat exchanger selection; Heat transfer enhancement

1. Introduction and aim of the work

Continuous efforts to improve the performance of heat exchangers in all fields of applications have resulted in the accumulation of a large amount of data containing the thermal and flow characteristics of many investigated heat exchangers. Among the available data are also those that were obtained for elements of heat transfer enhancements, and engineers usually apply these data for a preliminary selection of elements for heat transfer enhancements. However, the availability of the data for a large variety of elements for different heat exchanger surfaces is of little benefit unless proper methods to compare the final performances of such surfaces are provided. Moreover, during the development of new heat exchangers, one needs to plot the data in an appropriate way in order to assess directly the performance of the proposed heat exchanger compared with an earlier developed one. The development of such comparative methods should result in the selection of a surface or enhancement element which would lead to the most effective heat exchanger within given constraints. The comparison should be as simple as possible but with some confidence that the surface selected by such a comparison will meet the requirements of the heat exchanger under operating conditions.

A common way to present the thermal and fluid dynamic characteristics of heat exchangers is in the form of Nusselt number Nu or Colburn factor j and friction factor f. However, a direct comparison of the dimensionless parameters for different heat exchanger surfaces would not provide the answer as to which surface will perform best under given operating conditions. For example, compact heat exchangers built up with surfaces containing enhancement elements are characterized by higher pressure drop than less compact heat exchangers. Hence, an undesirable consequence of the utilization of elements for heat transfer enhancements is a larger increase in f, often larger in percentage, than the corresponding increase in Nu for a

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Nomenclature

A	heat transfer surface area, m^2
c_p	isobaric specific heat capacity, J/(kg K)
C	heat capacity rate, J/(s K)
d_h	hydraulic diameter, m
e	volume or area reduced power input, W/m^3 or W/m^2
E	power input, W
f	friction factor
h	convective heat transfer coefficient, $W/(m^2 K)$
j	Colburn factor
l	flow length between boundary-layer disturbance, m
L	flow length of heat exchanger, m
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number
NTU	number of heat transfer units
Pr	Prandtl number
$\dot{q}$	volume or area reduced heat transfer rate, W/m^3 or W/m^2
$\dot{Q}$	heat transfer rate, W
Re	Reynolds number
St	Stanton number
T	temperature, K
u_∞	free fluid stream velocity, m/s
U	overall heat transfer coefficient, $W/(m^2 K)$
V	heat exchanger volume, m^3

Greek symbols

β	heat exchanger compactness, m^2/m^3
ε	heat exchanger efficiency
η	fin efficiency
η_t	total extended surface efficiency
μ	dynamic viscosity, Pa s
ν	kinematic viscosity, m^2/s
ρ	density, kg/m^3
σ	ratio of free flow area to the frontal area

Subscripts

b	bare surface area reduced parameter
c	minimum cross-section, cold
in	inflow
h	hydraulic, hot
min	minimum
n	used by Soland et al. [7]
t	total
v	volume reduced parameter

given frontal area and flow rate. Therefore, by comparing the ratio f/Nu of a heat exchanger of interest, one may incorrectly conclude that less compact heat exchangers are more effective heat transfer devices. Obviously, although the dimensionless factors are suitable for scaling purposes among one class of heat exchangers, they do not offer the answer as to which of exchanger surface or enhancement element will meet the performance objectives within the design constraints. Therefore, in a practical application, this form of presentation is not useful, since for such applications one primarily needs to know, among heat exchangers with different heat transfer elements, which one will provide higher heat transfer rates for a given pressure drop or vice versa.

Depending on a specific application, one may identify various performance objectives and constraints that would determine the final heat exchanger configuration. The objectives and constraints for heat exchanger comparisons are the major heat exchanger operating and design variables such as the heat transfer rate, power input, flow rates and heat exchanger volume. Heat exchanger comparison can be performed by selecting one of the operating variables as a performance objective and the rest as constraints, e.g., if the reduction of heat exchanger volume is selected as the objective, the constraint might be the fixed heat transfer rate, fixed power input or both fixed heat transfer rate and fixed power input. Usually, performance objectives in the selection procedure are either the reduction of heat exchanger volume or increase in the overall heat transfer coefficient or reduction of power input for a given heat transfer rate [1]. The last objective could also be reformulated as an increased heat transfer rate for a fixed power input. The importance of reduced heat exchanger volume lies in the reduced material cost, weight and space requirements. The improvement of the overall heat transfer coefficient results in an increased heat transfer rate or in the reduction of heat transfer driving potential (temperature difference). Further, the reduced temperature difference is associated with lower thermodynamic irreversibilities, resulting in lower thermodynamic costs.

Various comparison methods, known as the performance evaluation criteria (PEC) or goodness factor, have been developed in the past while seeking appropriate heat exchanger selection procedures. In order to simplify the analysis, the PEC usually consider only the heat exchanger surface controlling the heat transfer resistance, e.g., air or gas side, and neglect the thermal resistance of the separating walls and fluid flow arrangement. Further, the PEC account only for the core pressure drop, which do not include pressure changes due to the entrance and exit effects and the acceleration effect.

Sahiti et al. [2] have already emphasized the importance of an appropriate comparison method. They presented one of the comparison methods for the performance assessment of different heat exchanger surfaces, without giving much detail of the method. The review of the known PEC, including their advantages, disadvantages and basic relationships, is given in the next section. It has been shown here that most such methods are approximate in nature. Therefore, a new method for the comparison of heat transfer and pressure drop characteristics of heat exchangers is proposed and demonstrated here. Other aspects that might influence the selection of heat exchangers, such as maintenance, reliability, safety and costs were not considered in the present work.

2. Review of comparison methods for heat exchanger selection

Depending on whether the frontal area of the heat exchanger or the heat exchanger volume is the parameter of interest, two major comparison criteria, namely the area goodness factor and volume goodness factor, have been used in the past. The area goodness factor represents basically the direct comparison of the ratio j/f as a function of Re in order to identify which heat exchanger would require the minimal frontal area for a fixed pressure drop. The method does not serve, however, as an effective selection tool in several practical applications where in addition to the pressure drop, the entire heat exchanger volume has to be taken into account.

One of the earliest proposals in the literature, dealing with comparison methods of heat exchangers, was published by Colburn [3] and was adopted by London and Ferguson [4] to plot the heat transfer coefficient h versus flow friction power supply e normalized by the total heat transfer area (wetted area) of the investigated heat exchangers. They plotted the data for the reference gas at 280 °C and proposed a relation needed to predict h and e for fluids at other than reference properties. The method of London and Ferguson [4] belongs to the volume-goodness factor group of heat exchanger performance as it refers to the entire heat exchanger surface within the heat exchanger volume. The advantage of the London and Ferguson [4] method is the direct use of the heat exchanger data, such as Colburn factor $j = StPr^{2/3}$ and Fanning friction factor f to compare heat exchanger performances:

$$h = \frac{\mu c_p}{Pr^{2/3} d_\mathrm{h}} Re\, j \tag{1}$$

$$e = \frac{\mu^3}{2\rho^2 d_\mathrm{h}^3} Re^3 f \tag{2}$$

The Reynolds number is based on the hydraulic diameter and is defined as

$$d_\mathrm{h} = 4\frac{A_\mathrm{c} L}{A_\mathrm{t}} \tag{3}$$

where A_t denotes the total heat transfer area, A_c the minimal cross-sectional area and L the flow length of heat exchanger [5].

It is important to note the indicative character of the expressions for h and e (Eqs. (1) and (2)) regarding the influence of d_h in the performance of heat exchanger surfaces (higher performance with a decrease in the value of the hydraulic diameter). The better performance of a particular heat exchanger based on the London and Ferguson [4] method is characterized by a higher curve position in the plot of h versus e. The method allows only a rough estimation of the relative heat exchanger performance as a large value of h implies a small driving temperature difference and therefore the real advantage of a heat exchanger with larger h will be less than that predicted by the comparison of h and e only. Further, the comparison method leads to the same conclusion regarding the performance of heat exchangers belonging to the same class but with different flow lengths. This is owing to the assumption of a constant h over the heat exchanger flow length. However, in reality the heat exchanger performance is not invariant over the flow length due to the less increase of the heat flow rate compared to the increase of the pressure drop. Therefore, the performance of the heat exchanger decreases with an increase in their flow length.

A completely different comparison method has been suggested by LaHaye et al. [6]. They used the flow length between the major boundary-layer disturbances to plot the heat exchanger data in a dimensionless form. In this way, they introduced the major factor responsible for the heat transfer increase, namely the ratio obtained by dividing of the flow length between the major boundary-layer disturbance l and hydraulic diameter d_h. Hence, they achieved to plot in one diagram the performance of all possible heat transfer surfaces with a known value of the ratio l/d_h. However, the authors pointed out that the method is valid for the turbulent flow regime where much more orderly behavior of data for j and f could be presented by a constant slope coefficient of the performance line. Further, the method provides only an approximate comparison of heat exchanger surface geometries as it does not account for the fin efficiency, fin thickness, gaps between successive elements in a row, etc. Furthermore, the method of LaHaye et al. [6], like the London and Ferguson [4] method, does not account for the influence of driving temperature difference on the relative performance of a given heat exchanger surface. Hence, both methods should be applicable only if the temperature difference between heat transfer area and the fluid do not change for the surfaces under comparison. However, such conditions usually do not prevail in practical heat exchanger applications.

A more practical method for the comparison of heat transfer surfaces was developed by Soland et al. [7]. They compared the performance for a fixed flow rate and fixed inlet temperatures of the hot and cold fluids. The key of the method lies in conversion of j and f factors of an extended surface to the similar factors based on the bare area of the enhanced surface of the heat exchanger, which is the same as the area of an imagined heat exchanger surface with no fins. Further, Re is derived based on the open area as though the fins were not present, using a definition for hydraulic diameter similar to that given in Eq. (3). In order to avoid confusion, Soland et al. [7] added the subscript n to their parameters. Following the procedure described in detail in their paper, the authors derived j_n, f_n, Re_n and d_n and used these to present the heat exchanger data from Kays and London [8] in the following form:

$$\frac{j_n Re_n}{d_n^2} = \text{function}\left(\frac{f_n Re_n^3}{d_n^4}\right) \tag{4}$$

The variable groups of Eq. (4) were used by the authors as performance variables reduced to the heat exchanger volume V, because such groups are directly proportional to the number of heat transfer units NTU (as a measure of heat transfer ability) and the power input E:

$$\frac{\mathrm{NTU}}{V} \approx \frac{j_n Re_n}{d_n^2} \tag{5}$$

$$\frac{E}{V} \approx \frac{f_n Re_n^3}{d_n^4} \tag{6}$$

Hence, Soland et al. [7] basically compared the number of heat transfer units versus the power input per heat exchanger volume of different plate fin surfaces including surfaces with sand grain roughness. The method of Soland et al. [7] does not require a constant temperature difference between the wall and the fluid and this is a substantial advantage over previously discussed methods. Furthermore, the definitions used by the authors are simple and can be easily derived with the same accuracy as the authors achieved, whereas the definitions used by London and Ferguson [4] cannot be derived exactly without additional information regarding the data reduction procedure, which often is not provided in detail. Further, the method allows the comparison of the surfaces with turbulent promoters and roughness, whereas the other comparison methods normally do not allow such a comparison as they are based on the total heat transfer area (wetted area) and this parameter cannot be predicted accurately for rough surfaces.

In parallel with Soland et al. [7], Shah [9] presented a study of about 30 different heat exchanger comparison strategies. He discussed, compared and assessed methods mainly based on their simplicity, pointing out that no heat transfer surface can be best for all applications. Thus he claims that no fine calculations are needed for the heat exchanger surface comparison. Nevertheless, in the following sections, the authors show that selection procedure of modern heat exchangers cannot rely on approximate performance assessment methods. By employing the total extended surface efficiency η_t, Shah [9] adopted the recommendation of Kays and London [5] to develop a "volume goodness factor" which compares the heat transfer per unit heat exchanger volume and unit temperature difference $\eta_t h\beta$ versus power input per unit heat exchanger volume $e\beta$ at some standard fluid properties. From the viewpoint of compactness, a high plot of $\eta_t h\beta$ versus $e\beta$ will characterize a surface of better performance. In the case with no system or manufacturing restrictions, Shah [9] suggested the usage of London and Ferguson [4] method, whereas for the comparison of heat exchanger surfaces as they are, he suggested the usage of the adopted method of Kays and London [5].

Some papers also discuss the PEC for a particular class of heat exchangers, e.g., Webb [10] reported a comprehensive study of PEC for application to single-phase heat transfer in tube flows. The author provides a detailed procedure based on the main objectives, namely, reduced heat exchanger material, increased heat transfer rate, reduced driving temperature difference and reduced power input to select the optimal surface.

More recently, Cowell [11] presented a general comparison procedure for the heat transfer surfaces used in compact heat exchangers. He developed a detailed method comprising almost all previously given methods and showed that this can be used for a wide range of heat transfer surfaces. However, as the author suggests, the method has only an indicative character, which offers to the user simple procedures for the preliminary selection of a heat transfer surfaces. Similarly to the previously described methods, Cowell [11] also used the Colburn factor j to compare surfaces based on different objectives and restrictions.

Compact and efficient heat transfer surfaces are usually associated with manufacturing and design difficulties. Hence, the potential user usually needs to know as accurately as possible the relative performance of the proposed heat transfer surfaces in order to assess the final benefit. Owing to their indicative and general character, the comparison methods described in this section cannot satisfy these requirements. Hence, one of the objectives of the present work was the development and application of heat transfer surface comparison methods which are simple, more accurate and suitable for the selection of heat exchangers.

3. Approximate comparison of pin fin heat exchanger versus smooth pipe heat exchanger

The first step in selecting a heat exchanger is to define its characteristic shape and the applied working fluids. Usually, both of these selection criteria are dictated by the specific application, e.g., the gas–liquid plate fin heat exchanger and tube fin heat exchangers have been established in the automobile industry and in air-conditioning units, whereas the shell and tube heat exchangers have found wide applications in power plants and the chemical industry. There is no benefit from a complex comparison method which allows the comparison of heat exchangers for all possible applications but which does not offer the required accuracy for the relative performance of a heat exchanger for a particular application. The accuracy of some of PEC presented in Section 2 is given below based on the performance comparison of the tested pin fin heat exchanger with the geometrically similar smooth pipe heat exchanger. The most widely used method for comparison purposes was found to be the method of London and Ferguson [4] and therefore, this was selected in the present work to demonstrate the limited possibilities of approximate methods for the selection of heat exchangers. Additionally, the performance comparison was performed also based on the method of Soland et al. [7], as this has some similarities to the method proposed in the present work.

3.1. Heat transfer coefficient as performance variable

In Section 2 it was shown that the comparison of the heat exchanger performance based on the method of London and Ferguson [4] reduces to the comparison of the heat transfer coefficient versus the power input to drive the heat exchanger with both parameters reduced to the entire heat transfer area. However, in the present work the demonstration of such method (Fig. 1) was performed with the heat transfer coefficient and power input reduced to the bare area of the surface covered with pins h_b and e_b as in the authors' opinion it has some advantages [2,12]. The heat transfer coefficient of the pin fin heat exchanger was derived experimentally whereas that of the smooth pipe was obtained analytically. The evaluation of the power input was also done in the same way, whereas in general Eq. (2) should be used if the friction factor f is available.

Fig. 1 clearly shows the major advantage of pin fins as far as the heat flux rates for the same temperature difference and same power input are concerned. It was found that the ratio of heat transfer for the pin fin heat exchanger ($h_{b,pin}$) to that for the smooth pipe heat exchanger ($h_{b,smooth}$) for the same power input is in the range $h_{b,pin}/h_{b,smooth} = 23$–$30$. Higher values of the ratio are obtained for lower values of e_b.

3.2. Number of heat transfer units as performance variable

Despite the similarity of the performance comparison method of Soland et al. [7] with the current method, it will be shown that such method enables also only an approximate performance prediction of heat exchanger surfaces.

The key parameter in the Soland et al. [7] method is the NTU factor defined as

$$\mathrm{NTU} = \frac{UA}{C_{\min}} \tag{7}$$

where U is the overall heat transfer coefficient, A the heat transfer area upon which U is based and $C_{\min} = (\dot{m}c_p)_{\min}$ the minimal heat capacity rate [13].

It was already emphasized in Section 1 that the PEC criteria takes into account only one side of the heat exchanger surface, assuming the heat transfer resistance of the other side to be negligible. The error in the performance assessment introduced by such an assumption has been found to be within 5%. Furthermore, the thermal resistance of the wall separating fluids in the heat exchanger is small owing to the use of thin and high thermal conductivity wall materials (usually aluminium or copper). A further simplification was the assumption of the perfectly conducting fins ($\eta = 1$). Hence, Soland et al. [7] used an approximate form of NTU:

$$\mathrm{NTU} = \frac{hA}{\dot{m}c_p} \tag{8}$$

where h denotes the heat transfer coefficient and $\dot{m}c_p$ the heat capacity rate of the fluid on the side under consideration. They derived the NTU from Eq. (8) by expressing h in term of the Colburn factor j. Eq. (8) would be an exact expression of NTU for a heat exchanger with a uniform wall temperature, e.g., single-phase flow in the side under consideration and two-phase flow in the opposite side (condensation or evaporation). The use of an approximate form of NTU for comparison of heat exchanger surfaces for which j and f factors are available is a reasonable choice, as the evaluation of exact NTU requires the arbitrary assumption of fluid type, fluid flow rate, fluid inlet temperature and channel flow geometry of the side not considered and the gain in accuracy (~5%) would not be worth the effort required. However, the performance comparison of the present heat exchanger was carried out based on the exact NTU values, as all required data were available (Fig. 2).

Note that the exact NTU/V factor decreases with increase in power input, whereas by determination of NTU/V from Eq. (5) the opposite behavior would result. This is because Soland et al. [7] cancelled the mass flow rate $\dot{m}$ (the same for both heat exchangers) and this results in a change in the behavior of the number of heat transfer units with power input. Regarding the relative merit of the heat

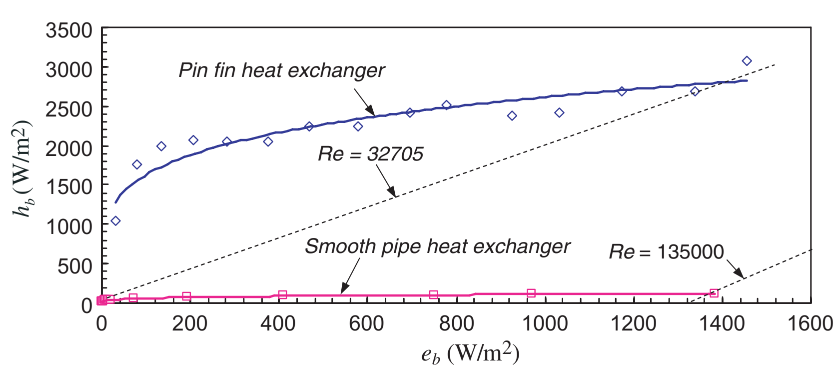

Fig. 1. Performance comparison of heat exchangers based on the London and Ferguson [4] method.

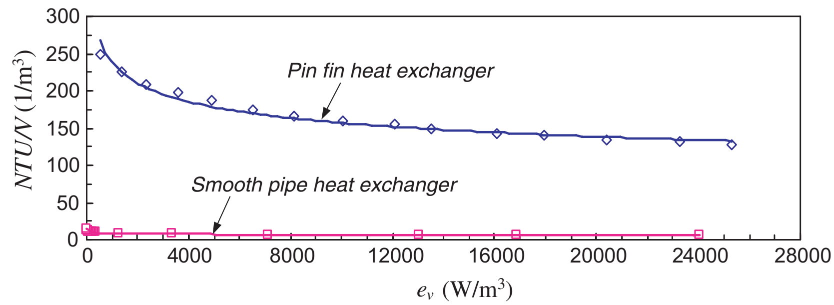

Fig. 2. Heat exchanger performance graph after the Soland et al. [7] method.

exchanger surfaces, it was found that $(\mathrm{NTU}/V)_{\mathrm{pin}}/(\mathrm{NTU}/V)_{\mathrm{smooth}} = 3$–$9$ over the same range of e_v, where higher values were obtained for lower values of e_v.

4. Consistent comparison of pin fin heat exchanger versus smooth pipe heat exchanger

The basis of the method developed by Soland et al. [7] for the performance evaluation criteria by employing NTU was the exponential relationship of NTU with heat exchanger efficiency ε, which for the simplified case of fluid flow through heat exchanger channels with uniform temperature follows the relationship

$$\varepsilon = 1 - e^{-\mathrm{NTU}} \tag{9}$$

Otherwise, the efficiency usually follows a direct relationship to the heat transfer rate:

$$\dot{Q} = \varepsilon(\dot{m}c_p)_{\min}(T_{\mathrm{h,in}} - T_{\mathrm{c,in}}) \tag{10}$$

Soland et al. [7] concluded that higher NTU means higher ε and therefore higher heat transfer rates (Eq. 10) and one can therefore use the number of heat transfer units to characterize the heat exchanger performance. However, as NTU and ε are not related to each other through a proportional relationship, the performance assessment by Soland et al. [7] method would result in an approximate comparison of the heat exchanger configuration. Further, similar to the London and Ferguson [4] method, the Soland et al. [7] method cannot consider the performance decrease with the heat exchanger flow length due to the assumption of a constant h in the relationship used to derive NTU (Eq. (8)). It should also be noted that the simple relationship for the efficiency given by Eq. (9) applies only in special cases of heat exchangers, whereas for common heat exchangers the form of Eq. (9) is much more complicated.

The behaviour of the present heat exchanger efficiency versus the number of heat transfer units is plotted in Fig. 3, which indicates that for NTU > 1 quite a small increase in ε can be obtained and therefore in these regions the Soland et al. [7] method might fail to predict heat exchanger performance accurately. The method might particularly be critical for modern heat exchanger where the performance improvement possibilities are limited. Hence, the selection of heat exchangers based on approximate methods will result in heat exchangers that might not meet the required performance objectives under the operating conditions. From the viewpoint of accuracy, a direct plot of ε against the power input would result in the accurate assessment of heat exchanger performance. However, the present authors suggest a physically more meaningful direct plot of the heat transfer rate reduced to the heat exchanger volume $\dot{q}_v$ versus the required power input also reduced to the heat exchanger volume e_v (Fig. 4).

The basic advantages of the use of the present heat exchanger performance diagram (Fig. 4) are (1) accuracy in the performance prediction, (2) no volume or surface geometry constraints, (3) no need to convert the data into j (or h) and f factors as far as the performance of heat exchanger is concerned and (4) same units for the performance parameters according to their similar physical basis.

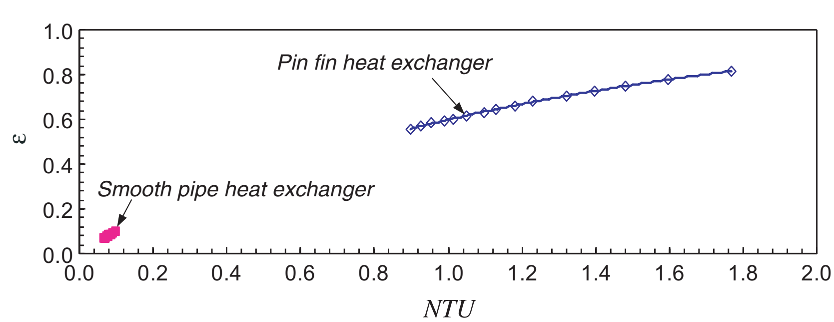

Fig. 3. Efficiency versus number of heat transfer units for the present heat exchanger.

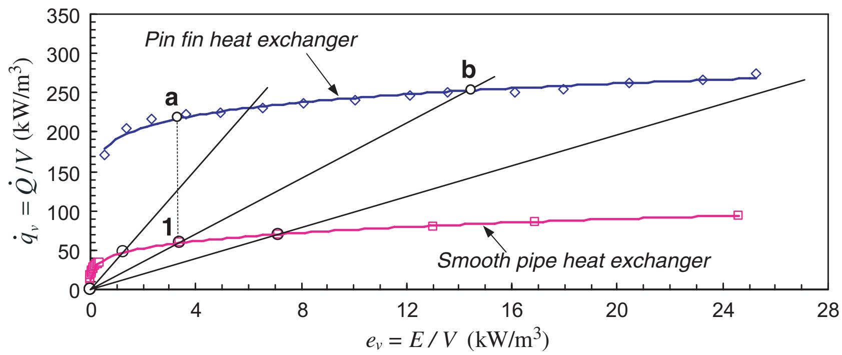

Fig. 4. Heat exchanger performance diagram based on heat transfer rate versus power input per unit heat exchanger volume.

Similar to the Soland et al. [7] method, the diagram allows performance comparisons for

(a) same heat exchanger volume and power input,
(b) same power input and heat transfer rate.

Note that instead of heat transfer rate in (b), Soland et al. [7] used the number of heat transfer units.

The performance comparison of the newly developed heat transfer surfaces with any of the existing surfaces should be performed by plotting of $\dot{q}_v$ versus e_v for the following operating and design constraints:

- same mass flow rate,
- same inlet temperature of the hot fluid stream,
- same inlet temperature of the cold fluid stream,
- same heat exchanger flow length.

The performance comparison by a plot of $\dot{q}_v$ versus e_v for the specified constraints would allow the selection of the heat transfer surface with the highest performance, because in such case the performance depends only on the geometric characteristics of the surface. Namely the heat transfer rate calculated from Eq. (10), provided that specified constraints are satisfied, depends only on the value of the heat exchanger efficiency ε, which on the other hand for constant flow rates and constant flow lengths depends only on the geometric characteristics of the heat transfer surface. Since the operating point of the higher performance surface (point b, Fig. 4) is usually obtained for lower velocities compared to those of the operating point of a lower performance surface (point 1, Fig. 4), the operating constraint about the same mass flow rate can be provided by increasing the cross-sectional area of the higher performance surface. Thereby, the fluid flow velocity corresponding to point b and hence also the corresponding heat transfer coefficient h are kept constant. The predicted smaller volume of the heat exchanger containing the better performing heat transfer surface can be obtained after the comparison is finished by reducing the heat exchanger flow length.

A similar diagram can be obtained by a plot of $\dot{q}_v$ and e_v without any constraints regarding the mass flow rate, inlet temperatures of the fluid streams and the heat exchanger volume. In such a case, one compares the performance of the entire heat exchanger in the actual state and not only of their heat transfer surfaces. For the development of new heat transfer surfaces, this kind of comparison is not suitable as it does not offer an answer to the question of whether the eventual improvement in the performance results from the newly developed surface or from different inlet fluid stream temperatures or from different flow rates used to drive the heat exchangers under comparison.

However, the performance of the present pin fin heat exchanger versus the smooth pipe heat exchanger was measured without any constraint regarding the mass flow rate (Fig. 4). This is related to the large difference in the performance of the heat transfer surfaces of such heat exchangers, which results in performance curves that are shifted significantly from each other. Such a shift prevents the performance comparisons on the heat exchanger performance plot for the same mass flow rate.

The performance comparison of heat exchangers for the same volume and same power input can be easily performed by evaluating the ratio $\dot{q}_{v,pin}/\dot{q}_{v,smooth}$ for the curve points connected with the line of the same power input. This case is illustrated in Fig. 4 by the line connecting points 1 and a, which may lie everywhere else in the curve. Since the pressure drop of the pin fin heat exchanger was much higher than that of the smooth pipe one, the same amount of the power input for the pin fin heat exchanger compared with that of the smooth pipe heat exchanger was obtained for smaller Re and hence also for smaller flow rates. Similar behaviour can be expected also for other high-performance surfaces which usually are characterized with higher pressure drop compared with lower performance surfaces. This might be a disadvantage of high-performance surfaces in some applications, e.g., in air-conditioning systems, where the thermal comfort requires a certain amount of air with fixed or variable temperature and relative humidity parameters. For such applications, in order to obtain the required flow rate without an increase in the pressure drop and without changing the heat exchanger volume, as already explained, one changes the shape of the high-performance heat exchanger by increasing the frontal cross-section area and reducing the heat exchanger length. In this way, one obtains the shapes characteristic for high-performance heat exchangers such as in automobiles, units of air conditioning system and elsewhere where liquid–gas heat exchangers are utilized. Such shapes are characterized with their shorter flow length and larger frontal cross-section area.

The performance evaluation of heat exchangers with the same volume and same shape (but different flow rates) resulted in $\dot{q}_{v,pin}/\dot{q}_{v,smooth} = 3$–$4.7$. This means that for the same power input in the region of lower *Re* (NTU > 1) the pin fin heat exchanger would be able to transfer up to 4.7 times more heat, whereas based on the Soland et al. [7] method one might conclude that the pin fin heat exchanger would provide up to nine times higher heat transfer rates. Note that the London and Ferguson [4] method resulted in up to 30 times higher heat transfer rate per unit temperature difference for the pin fin heat exchanger.

The comparison of heat exchanger performances for the same power input and same heat transfer rate represents basically the comparison of the heat exchanger volume for the same duty. In such case the performance comparison results in the comparison of heat exchanger volume for the operating points (e.g., the points 1–b) lying in the straight lines with the slope $\dot{q}_v/e_v$ and which passes through the origin of the plot. This is because in such a case the comparison constraints are constant $\dot{q}_v$ and e_v and this mean that the values of both axes in Fig. 4 are inversely proportional to the heat exchanger volume. The relative size of the heat exchanger volume can be obtained by a comparison of either the ordinates or the abscissas of the operating points and it will result in a smaller volume for a surface with a higher lying curve in the performance diagram, e.g., the performance comparison of the present heat exchanger for curve points 1 and b resulted in the volume of the pin heat exchanger being 0.23 times of the volume occupied by the smooth heat exchanger.

5. Consistent comparison of heat exchangers surfaces with known characteristics

The heat exchanger performance plot (Fig. 4) was obtained by directly plotting of experimentally and analytically estimated heat transfer rates and pressure drop (power input). Similar diagrams can be plotted also for heat exchanger surfaces for which data in the form of *j* (or *h*) and *f* are available. The most comprehensive study of various heat exchanger surfaces was carried out by Kays and London [8]. The comparison of a single air channel comprising some of their surfaces presented in a reprinted edition of their book [14] was performed by assuming the condensing steam channels on the other side. The width of the air channels was assumed to be 200 mm. The performance curves were derived for

- the same inlet air temperature (=20 °C),
- the same fluid inlet temperature on the other side (condensing steam = 100 °C),
- the same flow length (=30 mm).

If one wants to compare the heat transfer surfaces for the same power input and same heat exchanger volume, then the constraints of the same mass flow rate have to be fulfilled. The same is true also in the case when one wants to compare the volume occupied by heat transfer surfaces for the same heat transfer rate and same power input. As already mentioned in the previous section, the fulfilment of the constraints for the same mass flow rate requires a decrease of the heat exchanger flow length and increase of their cross-section area.

In order to plot the performance characteristics of surfaces with the available data (Fig. 5), one has firstly to present the performance parameters ($\dot{q}_v$ and e_v) as function of operating variables. By following the procedure similar to that described by Soland et al. [7], the following forms of relationship for the heat transfer rate ($\dot{q}_v$) and power input (e_v) were obtained [12]:

$$\dot{q}_v = \frac{\varepsilon \rho v \beta \, Re c_p (T_w - T_{a,in})}{4l} \tag{11}$$

$$e_v = \beta \frac{Re^3 \mu^3}{2\rho^2 d_h^3} f \tag{12}$$

where l denotes the air channel length (=30 mm) and T_w the wall temperature of the air channel which was considered to be the same as the temperature of the condensing steam (=100 °C).

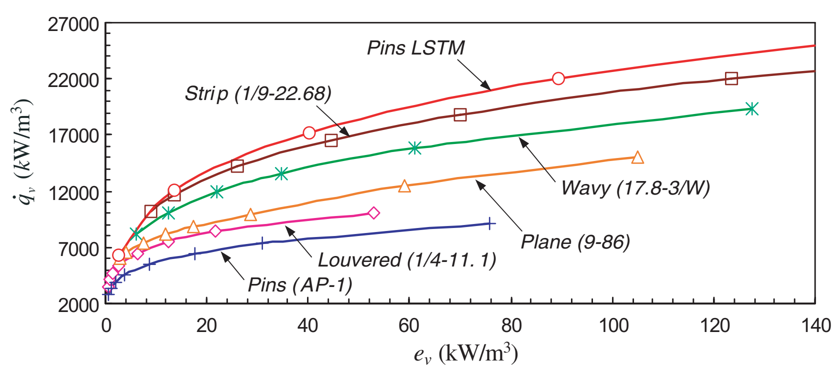

Fig. 5. Performance plot of some of high performance surfaces with known characteristics.

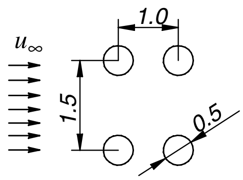

Fig. 6. Pin fin arrangement used for comparison with the Kays and London [14] surfaces.

In addition to the Kays and London [14] data, the characteristics of an in-line pin fin arrangement according to Fig. 6 were plotted. The pin length (=7 mm) was chosen to be of the order of the length of fins for surfaces from Kays and London [14].

An extensive search of the literature revealed that there is no empirical correlation which can be used for the evaluation of heat transfer and pressure drop from the current arrangement of the pin fins for laminar flow conditions. Hence, the heat transfer and pressure drop characteristics of a hypothetical heat transfer surface with pin fins having an arrangement like that in Fig. 6 were derived from equations for *Nu* and *Eu* obtained numerically [12].

The performance plot (Fig. 5) shows that the pin fin heat transfer surface investigated by Kays and London [14] performs worse than all other kinds of fins. This is because that pin fin arrangement was characterized with large streamwise and transverse pin spacing (=3.125 mm) and because the pin diameter (~1 mm) was much larger than the thickness of other fins (~0.1 mm). Similar reasons led to a performance of the chosen louvered fins (1/4–11.1) which is below the performance of other fins. Otherwise, the current pin arrangement (Fig. 6) with dimensions and compactness close to those of other fins presented in Fig. 5 performs better than all other fin geometries.

6. Conclusions, final remarks and outlook

In the present work, a literature survey of methods used for the comparison of heat exchanger performances was performed and it was shown that these methods basically plot the heat exchanger data in terms of Colburn factor j or heat transfer coefficient h versus power input. In general, all listed methods were found to be limited as they assume a constant driving temperature potential for heat transfer from surfaces with different heat transfer enhancement elements. An exception is the Soland et al. [7] method, as instead of heat transfer coefficient as heat exchanger performance variable it uses the number of heat transfer units. However, the Soland et al. [7] method also offers only an approximate comparison, as it does not account for the non-proportional relationship between the heat transfer rate and number of heat transfer units. Further, all performance comparison methods used in the past do not consider the heat exchanger performance decrease with the flow length.

In the present work the accuracy of the London and Ferguson [4] method and the Soland et al. [7] method was tested by the comparison of the performance of an experimentally tested pin fin heat exchanger with that of a smooth pipe heat exchanger evaluated analytically. It was found that both methods over-predict the performance of pin fin heat exchanger compared with the performance of the smooth pipe heat exchanger. Therefore, a direct comparison of heat transfer rate and power input per unit heat exchanger volume was proposed in the current work as a more accurate method for the selection of modern heat exchangers. The proposed method allows performance comparisons of new tested heat exchangers without the need for data conversion into h and f. The present work demonstrated that the method can be also successfully applied for the selection of heat exchanger surfaces based on the data available in the literature. The method is not limited to a particular extended surface geometry or heat exchanger volume provided that the inlet fluid temperatures, mass flow rates and heat exchanger flow length are kept constant. All constraints of the proposed method are parameters which ensure a fair heat exchanger comparison. It is therefore believed that the method offers more confidence for the accurate selection of modern heat exchanger surfaces and configurations and thus allows a design engineer to predict accurately the costs and benefits of the heat exchanger design to be used in a particular application.

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