Efficiency of heat transfer in heat exchangers with wire mesh packing

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A R T I C L E   I N F O

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A B S T R A C T

This paper presents the results of studies concerned with heat exchange and pressure drop during the flow of air and water through a channel with and without wire mesh packing. It was stated that using wire mesh packing makes possible increase in heat exchanged flux as well as heat transfer coefficient at simultaneous lowering the temperature of heating surface. A new model for describes heat transfer from channel wall to fluid is proposed. The differences between the fluxes of exchanged heat (in an empty and packed channel) compensates for the higher demand for energy in order to pump the gas through wire mesh packing, in the circumstances of the conducted research the energetic gain resulting from the use of wire mesh packing amounted to 40%.

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1. Introduction

The growing energy consumption of industrial processes imposes the need to search for new engineering and technological solutions aimed at increasing their efficiency. One of such solutions involves the application of porous materials as wire mesh packing in thermal apparatus. The very large specific surface of the porous materials constitutes the development of the surface of heat exchange, which enables the transfer of much higher heat flux (in comparison to apparatus without mesh packing). Porous mesh packing also finds application in mass exchangers. The major role of the wire mesh packing is associated with an increase in the surfaces of contact between the phases. The processes encountered in such apparatus in many cases have exo- or endothermic characteristics. The heat exchange between the walls and fluids is transported with the aid of the porous structure.

One of the most common porous structures used in industry is a mesh packing composed of the several overlying layers of metal wire. The selection of the various mesh size and wire diameter makes it possible to select the packing offering a great diversity in terms of porosity and specific surface. Such packing is characterized by regularity of dimensions and shapes of pores. The opportunity of having mesh packing made with a variety of materials makes it resistant to a variety of unfavorable service conditions, e.g. contact with a high temperature agent or chemically aggressive ones. Wire mesh packing is applied in a variety of processes and devices with flowing media. The examples include high efficiency heat exchangers, heat storage, porous fins, solar-receiving devices, structures in heat pipes, regenerators in Stirling cryocoolers, cooling of electronic equipment, catalytic reactors and others [1–6].

The very large pressure drop of flow through porous materials is associated with the need of installing relatively large power output for the flow of a fluid, which considerably reduces the efficiency of heat exchangers. The results found in some papers, e.g. [7–9] regarding the hydraulic properties of flow and heat exchange encountered during flow through porous packing indicate that the use of an adequate packing may lead to a sufficient increase of heat exchange in order to compensate for the energy losses associated with the higher flow resistance.

The intensity of the heat exchange and pressure losses is functions of both hydrodynamic properties of the flow as well as the internal structure of a porous material. The decisive role is played by such quantities as: velocity and properties of fluids, porosity and permeability of the packing, shapes and sizes of pores/particles in the packing, specific surface of the packing and thermal conductivity of the system fluid–porous material. The effect of the above parameters on thermal and flow phenomena tends to be ambiguous in a lot of cases. As a result, the determination of the optimum course of technological processes is predominantly a complex task.

In the references we have to do with very few papers, whose authors present models describing heat transfer during flow through wire mesh packing. The measures of the intensity of heat exchange include dimensionless criteria, such as Nusselt number or Colburn factor, which enable one to determine heat transfer coefficient from the relations relevant for the classical cases of heat transfer in empty apparatus. The general form of the Nusselt number is presented by Eq. (1)

\[
\text{Nu} = \frac{h_d d_f}{k_f}
\]
Bin-Nun and Manitkos [10] apply the criterion based relation (Eq. (2)) developed by Gedeon according to which Nusselt number is the function of the Peclet number and porosity of wire mesh packing

\[ \text{Nu} = (1 + 0.99\text{Pe}^{0.66})\text{Da}^{0.79}. \]  

(2)

The hydraulic diameters encountered in Eq. (1) has been defined on the basis of the diameter of mesh wire and porosity of the packing. The Peclet number (product of Reynolds and Prandtl numbers) is the quantity specific for the description of heat transfer in laminar flow. Such characteristics of flow are specific for porous packing due to the characteristics of flow, such as pore size and small velocity of the flow. Özdemir and Özgüc [11] propose that additionally the effect of Darcy number. Darcy number, which is based on the permeability of porous material makes it possible to account for heat transfer in the inherent structure of the packing. Such a necessity was noted by several authors, including [12–14]. The criterion based equation developed by Özdemir and Özgüc for the description of the Nusselt number is presented by Eq. (3).

\[ \text{Nu} = 0.2211\left(\text{PeV}\text{Da}^{0.5992}\right). \]  

(3)

Since this relation has been designed for the description of flow through mesh packing situated in a flat orifice, the Nusselt number was defined on the basis of the height of the orifice. In contrast, in paper [15] the proposed model of heat exchange accounts only for the hydrodynamic criterion of the flow (Re number) and porosity of the wire mesh packing (Eq. (4))

\[ J = \left(\frac{h_f}{k_f}\right)^{1/2} = 0.2\text{Re}^{-n}/(1 - \varepsilon), \]  

(4)

where

\[ n = 0.37 + 0.0136\varepsilon/(1 - \varepsilon). \]  

(5)

Pressure drop of the flow through wire mesh packing may be described, just as in the case of other porous materials by Darcy’s (Eq. (6)) or Forchheimer (Eq. (7)).

\[ \frac{\Delta p}{\Delta L} = \eta_f K^{-1} w_f. \]  

(6)

\[ \frac{\Delta p}{\Delta L} = \eta_f K^{-1} w + \rho_f Cw^2. \]  

(7)

The Pressure drop of the flow through porous packing increases proportionally to the velocity of the fluid (in accordance with Darcy’s law) within the range of small flow velocities. Following an increase of velocity, we have to do with the effect of inertial forces and pressure losses are described by a quadratic equation. The references do not contain appropriate guidelines to describe the boundary values of Darcy and Forchheimer flow for various types of porous packing. Permeability and inertia coefficient for the particular porous packing is determined on the basis of the measurement of the pressure drop of flow. There are also analytical methods applicable for the determination of the quantities based on porosity, diameters of pores and other parameters which characterize the structure of the packing [16,17].

It is quite popular to describe flow through wire mesh packing with the aid of the classical Darcy–Weisbach equation described by Eq. (8). The effect of the packing on the value of pressure drop is expressed as a result of adequate definition of the friction factor.

\[ \frac{\Delta p}{\Delta L} = \frac{f}{2} \frac{w^2}{\text{Da}} \rho. \]  

(8)

Such a course of action is followed by authors in papers [1,15,18].

The presented models of heat transfer and hydrodynamics of the flow through wire mesh packing have been developed on the basis of criteria of probability. The comparison between them is, however, rendered difficult due to the various methods used for defining the basic parameters which characterize the flow. The particular differences are discernible in the description of hydraulic diameter. Individual authors assume this value to take the form of either diameter of a channel, diameters of pores, particles in the packing or various combinations involving diameter with porosity to mention a few. For this reason the methods applied for the determination of heat transfer coefficient, friction factor or
permeability of the packing do not enable one to determine these quantities in a uniform way under various flow conditions. The supplement to the few available empirical papers in this regard is in the form of numerical modeling of thermal and flow phenomena in wire mesh packing. The subject matter of numerical simulations include such elementary values as distribution of pressure, velocity, temperature or the local values of heat transfer coefficient. The practical application of the findings of such studies is largely limited despite the quickly growing computing power of the current machines.

2. Scope of experiment

The author undertook original experimental research aimed at the determination of the heat flux exchanged between the heated wall of a channel with wire mesh packing and the fluid passing through it. For the purposes of comparison similar studies were conducted in an empty channel.

The studies, which applied water and air as the working media, were conducted using an installation presented in Fig. 1. The basic component of the installation was an electrically heated channel made of a steel pipe 21.7 mm in internal diameter and the total length of 2.5 m. The heat releasing section of the channel of length 1 m was preceded by a section to stabilize flow. Following the heat releasing section there was a half-meter section to reduce the nozzle effect. The heating of the channel was undertaken using a 2880 W electrical heating cord winding around the wall of the pipe. A transformer was fitted in order to adjust the voltage which energized the heater and hence it power output.

The packing was made of wire mesh with mesh size of 1.3 mm and diameter of the wire of 0.25 mm. The overlaying mesh formed a multi-layer porous packing with the porosity of 76.4%. Due to difficulties in obtaining uniform porous packing over the entire cross section of the channel, the space near the axis of the pipe was closed with non-porous mass over the entire length of the channel. As a result, ring section of the channel followed (Fig. 2).

The conducted research involves the measurement of fluid properties before and after the heating section. In these two spots the channel was empty, which enabled the partial balancing of the fluid mass. Due to the profile of the temperature in the cross section of the channel the temperature measurement was performed at the distance of 2 mm from the wall of the channel and in its axis (Fig. 2). In addition, the temperature of the wall was registered directly at the internal surface of the channel at the distance of 70 mm from the beginning and end of the heated section.

Simultaneous with temperature measurement the pressure drop values were recording along the entire length of the heating section and values of pressure in the channel. Collected data made it possible to determine the propriety of air present in the channel.

Fig. 1. Diagram of experimental installation.

Fig. 2. Distribution of wire mesh packing and thermocouple in the channel. (a) Cross-section. (b) Longitudinal section.
Pressure drop was measured as the difference between the pressures at the start and end of the channel’s heating section. The measurements of pressure involved electronic pressure gauges (differential and absolute).

The installation was supplied with air directly from the installation of compressed air. Water was fed into the system using an impeller pump. The flow of the medium was controlled using constricting valves and measured using electronic flowmeters. For the air the procedure involved measurements with a mass flowmeter due to huge pressure differences. Pressures sensors and flowmeters selected had inaccuracy levels below 1.5%. Temperature was measured using Ni–Cr–Ni thermocouple with precision range of 0.2 K. Prior to measurements a calibration of the gauges was undertaken.

The output signals from measurement devices were registered by a computer acquisition system. After the conditions of the flow stabilized the recording of data from experiment was taken at 1 Hz for 3 min. Table 1 contains the summary of the conditions at which the research was conducted.

The results of measurements presented in this paper were gained during studies conducted at a constant thermal output of the heating element $q_a = 12,245$ W/m$^2$ for heating air and $q_w = 32,654$ W/m$^2$ for heating water. The difference of the values of heat flux density and heat flux density absorbed by the fluid results from dissipation of heat into the environment from the heating element.

### 3. Discussion of results

#### 3.1. Heat exchange

An increase of the temperature of the fluid ($\Delta t_f$) resulting from its passing through the heated channel forms direct parameter of the effectiveness of the heating system. The results of the conducted measurements indicate that the use of wire mesh packing give a different result for gas than for fluid. For the case of air the increase in the temperature of gas was greater for the packed channel in comparison to the empty one. The situation is reverse for the case of water. The increment of temperature was defined as the difference between the temperature before and after the heated section of the channel (Eq. (9)), while the temperature of the fluid was assumed to be the mean of temperatures measured near the channel’s wall ($t_f^1$) and the one in the axis of the channel ($t_f^2$)

$$\Delta t_f = t_{f2} - t_{f1}, \quad (9)$$

where $f = a, w$;

$$t_f = \frac{t_f^1 + t_f^2}{2}. \quad (10)$$

The presence of the wire mesh packing does not, however, affect the nature of the variations of the increase of fluid’s temperature. Fig. 3 presents the characteristic curve of temperature increase of air following a growing flux of the gas. In both channels this value initially increases and gains a maximum, upon which starts to decrease following a further increase of flux. The initial, rising tendency of $\Delta t_a$ results from an increase in the turbulence of the flow following an increase of flux and hence the velocity of the air.

The increase in the temperature of the water deteriorates over the entire range of the variation of the flux, as illustrated in Fig. 4. As a result of the high thermal capacity of the liquid the increase of the heat transfer coefficient (at increasing velocity of fluid) does not compensate for the rising demand for the heat needed for heating water.

The heat absorbed by the fluids during the passage along the heated section of the channel is expressed in terms of the heat flux density. This quantity was determined on the basis of fluid flux and the temperature of the fluid at the inlet and outlet from the heating section (Eq. (11))

$$q_f = \frac{m_f(c_f t_{f1} - c_f t_{f1})}{A}. \quad (11)$$

In case of both channels, the heat transfer area $A$ is considered as internal area of heated section of the channel.

For the case of the air, the flux of heat is transferred during the flow through a packed channel. Heat flux density increase monotonically following the increase in the gas mass flux (Fig. 5). In the course of the variation of $q_a$ the maximum of increment of air temperature is not reflected. The density of heat flux is more affected by the mass flux increase of the heated liquid than the increase of the temperature.

During the heating of water the density of heat flux accommodated by the liquid $q_w$ is smaller for the case of the packed channel, as illustrated in Fig. 6. However, just as in the case of heating air, the value of $q_w$ increases along with the rise of water flux, despite

### Table 1

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Mass flux</th>
<th>Velocity</th>
<th>Reynolds number</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air (a)</td>
<td>0.12E–04 to 9.80E–04</td>
<td>0.05–3.91</td>
<td>3–250</td>
</tr>
<tr>
<td>Water (w)</td>
<td>0.78E–02 to 7.55E–02</td>
<td>0.04–0.36</td>
<td>51–409</td>
</tr>
</tbody>
</table>

![Fig. 3. Increase of air temperature in the heated section of the measurement channel.](image)

![Fig. 4. Increase of water temperature in the heated section of the measurement channel.](image)
the reduction in the increase of the temperature of the liquid following an increase in its flux. The situation was reverse during the observation of heating water in an empty channel. For this case the increase in water flux is followed by an initial increase in the flux of exchanged heat followed by a decrease.

In all cases the nature of the variability of heat flux density is reflected in the value of the heat transfer coefficient between the wall and the liquid. This coefficient was determined on the basis of heat flux density and mean temperatures of the walls ($t_b$) and the fluid ($t_f$) from Eq. (12), and the resulting values are presented in Figs. 7 and 11.

$$\alpha_{a,\text{exp}} = \frac{q_f}{t_b - t_f}.$$  \hspace{1cm} (12)

The mean temperatures of the fluid (Eq. (13)) and the wall (Eq. (14)) were calculated on the basis of the measured temperatures at the beginning (subscripts 1) and at the end (subscripts 2) of the heated section

$$t_f = \frac{t_{f1} + t_{f2} + t_{f1} + t_{f2}}{4},$$  \hspace{1cm} (13)

$$t_b = \frac{t_{b1} + t_{b2}}{2}. \hspace{1cm} (14)$$

For the two media the value of heat transfer coefficient is higher for the case of the passage through a packed channel in comparison to an empty one. This tendency is likely to monotonically follow an increase in the mass flux of the medium (Fig. 7). This is particularly discernible for the case of heating air. For relatively small fluxes of this medium the value of heat transfer coefficient tends to be similar in both channels. Starting from the value of gas mass flux of $1.7 \times 10^{-4}$ kg/s we can note a considerable increase of heat transfer coefficient in the packed channel, whereas its value in the empty channel $\alpha_{a,\text{exp}}$ goes up insignificantly. Under the greatest values of air flux the coefficient of heat transfer in an empty channel is seven times higher than in an empty channel. The decisive role is played by the temperatures of the wall and the passing fluid.

As we can see in Fig. 8, the difference between the mean temperatures of the wall and the air is comparable in the two channels within the range of small gas fluxes. The recorded differences are revealed for higher air temperatures (similar as in the case of different $\alpha_a$ values). In a packed channel the exchange of heat occurs under considerably smaller differences of temperature between the fluid and the heating surface. The temperature of heating surface is as lower than in channel without packing.

During the heating of water the difference between the temperatures of the wall and liquid is lower for the case of a packed channel for the entire range of flow variations (Fig. 9). One can note that the density of the flux absorbed by water and the measured increment in temperature during the passage through a packed channel is considerably smaller than in an empty channel (Figs. 4 and 6). However, heating of water occurs at considerable lower temperatures of channel wall. The difference of the temperature between the heating area and the liquid is about two times lower than in channel without packing. For this purpose, the coefficient of heat transfer presented in Fig. 10 gains higher values in a packed channel.

The lower value of the temperature of the empty channel's wall results from heat transfer from the wall into the packing as well as more intense acquisition of heat by the fluid, which passes through a packed channel with a higher velocity in comparison to an empty one (at the same mass flux).
3.2. Pressure drop

The results of measurements indicate that the applied wire mesh packing displays flow properties specific for porous materials. The characteristics of the variation of pressure losses (Fig. 11) conforms with Forchheimer law. Despite the fact that the research was conducted at relatively high velocities of phases, the deviations from Darcy model of flow are small.

On the basis of the measured pressure drop of flow the values of permeability and inertial coefficient were determined for the applied packing. Wire mesh packing applied display different properties for the case of gas and liquid flow. The value of permeability and inertial coefficient for water and air are equal to $K_w = 10.63 \times 10^{-9} \text{ m}^2$, $C_w = 559 \text{ m/s}$, $K_a = 9.43 \times 10^{-9} \text{ m}^2$, $C_a = 710 \text{ m/s}$, respectively. Differences within this range are also note for other types of porous packing.

4. Effectiveness of heat exchange

Both the increase in the temperature of the fluid and flux heat are absolute values and do not reflect the mechanism of heat transfer. The more reliable parameter of the effectiveness of heat exchange is the Colburn factor ($J$). This value was calculated on the basis of the experimentally determined coefficient of heat transfer (Eq. (15)).

$$J_{f, exp} = \frac{\gamma_f \cdot \exp}{g_f c_f}.$$  \hfill (15)

The data presented in Fig. 12 indicate that for the case of both fluids the Colburn factor gains the maximum values during flow through a packed channel. This means that in this channel it is pos-
sible to transfer greater amount of heat from and to a unit of mass flux density of a fluid under a given gradient of temperatures between the wall and the fluid.

For the case of water the difference between the empty and packed channel is small (Fig. 12b). The application of wire mesh packing offers considerable benefits during heating of water. For the higher fluxes of air the relation between Colburn factor between the packed and empty channels reaches 4.7 times.

The effectiveness of heat transfer is largely relative to the thermal contact between the individual particles and the contact in the system between the packing and the wall of the apparatus. Jiang et al. [8] indicate that the value of the thermal conductivity coefficient may be 3–4 times higher for the case of heat transfer through a channel filled with sintered bronze balls in comparison to non-sintered ones.

Probably also in case of meshed packing the reduction of the thermal resistance, e.g. gained as a result of welding wire in the mesh and soldering of mesh onto the heating surface would very likely result in gaining much greater values of flux heat exchange.

The effectiveness of the operation of a heat exchanger is not only relative to the intensity of heat exchange. The efficiency of flow apparatus is reduced by energetic effort associated with pumping of fluids. From the point of view of energetic effectiveness the use of packing in heat exchangers is justified only for the case when the flux of the heat increases to a higher degree in comparison to the power output needed for overcoming the additional hydraulic resistance of the porous material. The resulting energetic gain ($\Delta E$) can be expressed in terms of the differences of heating power ($Q$) less the power of the pumping ($N_f$) of two hypothetical heat exchangers, one of which is an apparatus packed with porous material (wire mesh packing) (Eq. (16)) and the second (with the same size and geometry) is empty without packing.

$$
\Delta E_f = (Q_{f,p} - N_f) - (Q_{f,o} - N_{f,o}).
$$

Subscripts ($p$) denotes exchanger with packing, ($o$) denotes exchanger without packing.

In the conducted analysis the role of heat exchangers is performed by heating sections of measurement channels. For the case of electrical heating of channel flow exist only in the internal space, formed by heating sections of measurement channels. For the case of heat exchange in a packed and empty channel (in comparison to an empty one) as well as high pressure drop of liquid passage through wire mesh packing result in the fact that the use of wire mesh packing is not justified. Over the entire range of variation of mass flux of water the energetic gain is negative (losses), as illustrated in Fig. 13b.

The heating power is described by Eq. (17)

$$
Q_f = \frac{m_f(C_1 - C_2)}{C_0}.
$$

The pumping power $N_f$ was determined under the assumption of the efficiency of the pumping device exceeding $\eta = 0.7$.

$$
N_f = \frac{\Delta P V_f}{\eta}.
$$

Relation (19) describes relative energetic gain ($\delta E$) expressed in percent

$$
\delta E_f = \frac{\Delta E_f}{Q_{f,o} - N_{f,o}} \times 100\%.
$$

The obtained results are presented in Fig. 13. The use of wire mesh packing for channel in which air was heated has enabled the air mass flux of $(4.5–5.0) \times 10^{-4}$ kg/s. The optimum conditions of flow are encountered under a wide range of change in air flux. The relative energetic gain exceeds 30% for $m_w$ from $3.0 \times 10^{-4}$ kg/s to $7.5 \times 10^{-4}$ kg/s, which in packed channel corresponds to the range of air velocity $w = 1.0–3.2$ m/s.

The smaller heat flux exchanged during heating water in a packed channel (in comparison to an empty one) as well as high pressure drop of liquid passage through wire mesh packing result in the fact that the use of wire mesh packing is not justified. Over the entire range of variation of mass flux of water the energetic gain is negative (losses), as illustrated in Fig. 13b.

5. Model of heat exchange

The references contain little information regarding heat exchange during flow through wire mesh packing. Some of the papers in this regard include [10,11,15].

Fig. 14 presents the comparison between the calculated heat transfer coefficients with the values gained on the basis of author’s experimental research (Eq. (12)). The values of $\delta E_{cal}$ were determined with the aid of the methods for the calculation of this quantity described by Refs. [10,11,15]. The large differences between the calculated and measured results could be attributed to the characteristics of the type of packing used in this research author and by other researchers. In addition, the conditions in which testing was conducted were different. The methods in the referred papers were conducted on gas flow [10,15] or liquid [11]. In contrast, this author undertook concurrent research of both phases.

For water the highest conformity was obtained for the results calculated using Özdemir and Özgüc’s method [11] (Fig. 14b). The researchers conducted studies in the conditions most similar to the ones presented in this paper. Heat transport occurred between the fluid and channel wall with overlaying wire mesh.
The models of heat exchange presented by Bin-Nun and Manitakos [10] and Barron Randall [15] concern recuperators in which heat exchange occurs directly between fluid and the entire area of a wire mesh packing. In research by the current author wire mesh packing was only an element acting as an agent in heat transfer between the wall and the fluid. It is also relevant to note the direction of the flow. In recuperators the layers of the wire mesh are situated perpendicular to the direction of the fluid passage. In the current research the fluid passes along the layers of the mesh forming the packing in the channel, which limits the surface of the contact between fluid and the heated wall. For these purposes the values of the heat transfer coefficient calculated according to guidelines in [10,15] tend to exceed several times the measured values, in particular for water (Fig. 14b). The method put forward by Barron Randall [15] enables one to offer a precise mapping of an actual heat transfer coefficient during passage of large air fluxes (Fig. 14a).

The unsatisfactory results of the application of reference methods for calculation of the coefficient of heat transfer during passage through the examined packing and the special characteristics of this packing indicate a necessity of development of a new model of heat exchange. A number of models [15,19–21] used for heat transport account the effect of the parameters of porous packing as well as hydrodynamic flow conditions (described by only Reynolds number). Hence, in accordance with proposition put forward by other authors [10,15] it was assumed that the hydraulic diameter \( d_h \) of the channel with wire mesh packing is the function of the diameter of the wire \( d \) and porosity \( e \) of the packing (Eq. (22)).

\[
d_h = \frac{e d}{1 - e}.
\]  

(22)

Reynolds number is defined by Eq. (23) on the basis of hydraulic diameter and mean velocity of fluid \( w_f \) in the pores

\[
Re_f = \frac{w_f d_h \rho_f}{\mu_f},
\]

(23)

where

\[
w_f = \frac{m_f}{F_{ch} \rho_f}.
\]

(24)

Reynolds number determined in such a way achieves reaches higher values than those used frequently in relation to flow through granular porous media i.e. the Re defined based on permeability \( K - Re_{K_f} = \frac{w_f K_f \rho_f}{\mu_f} \) or particle diameter \( d - Re_{d_f} = \frac{w_f d_f \rho_f}{\mu_f} \).

The results of the flow through the filling of the structure which is close to wire mesh packing [eg. [22–24]] show that the limit of laminar flow is for \( Re_{d_f} = 120–150 \). The own investigations for air phase were performed for \( Re_{s,a} = 6–250 \) which correspond to the range \( Re_{d,a} = 1–45 \), as well as for water \( Re_{s,w} = 51–409 \) and \( Re_{d,w} = 9–74 \) respectively. These are therefore the values within the scope of laminar flow.

Prandtl number, which characterizes the properties of the fluid is defined by Eq. (25). The considerable difference between the Pr value for liquid and gas enables one to state a generalization of heat exchange for the two fluids.

\[
Pr_f = \frac{C_f \mu_f}{\kappa_f}.
\]

(25)

The considerable difference between the Pr value for liquid and gas enables one to state a generalization of heat exchange for the two fluids. For the same purpose, the characteristic for laminar flow module \( (\mu_l/\mu) \) was introduced. Relatively small pores size of

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**Fig. 14.** Comparison between measured and calculated [10,11,15] values of coefficient of heat transfer. (a) For heating air. (b) For heating water.

**Fig. 15.** Nusselt number for water and air in the function of the Reynolds number.
the packing in the channel contributes to laminarization of fluid flow. The relation between the viscosity ($\mu_b$) of the fluid in the temperature of the wall to the viscosity ($\mu$) in the temperature of the fluid is higher for liquids than for gases.

The exponential character of the increase of Nusselt number noted in Fig. 15 was obtained as a result of the introduction of Froude number $Fr$ (Eq. (26)) into the model. Concurrently, due to the considerable reduction of velocity of the liquid this quantity affects the modeling of the conditions of transfer of water heat to a small degree. As a result, the increase of Nu number of both fluids is ensured in accordance with experimental data

$$Fr = \frac{w_f^2}{gd_b}. \quad (26)$$

In analyses case, use of the Froude number in reference to heat transfer it appears full reasonable. Many authors of research in this area, taking into account considerable tortuosity of pores, accent the significance of inertial force in flows through porous packings. The most often, it assumes that the inertial force is essential for $Re > 1$ according to Mischevic et al. [17] inertial effects appear already when fluid velocity exceed 0.01 m/s.

After a detailed analysis of the experimental data as well as thermal and hydraulic phenomena the final form of the relation for the description of the Nusselt number of this type of flow was established in Eq. (27)

$$Nu_f = 0.28Re^{0.25} \frac{1.84Pr^{1.11}}{\mu} + 0.22Fr^{0.85}. \quad (27)$$

The values of the constant and exponents in Eq. (27) come as a result of their optimization in terms of the highest convergence between the calculated $Nu_{cal}$ and measured $Nu_{exp}$ results. The relative error in the calculation of Nusselt number is defined by Eq. (28)

$$\delta Nu_f = \frac{|Nu_{exp} - Nu_{cal}|}{Nu_{exp}} \times 100\%.$$  

The mean relative error amounts to 9.26%, while the standard deviation 14.48%. The least precise data was collected during the flow of small water fluxes.

Eq. (27) along with the complementary relations (21)–(26) can be used to predict the intensity of heat transfer during the heating fluid (from the wall of the channel) in laminar flow through the horizontal section of the annular metal wire mesh packing while hydrodynamic conditions of flow are described by Reynolds number (Eq. (22)) below 400. Taking into account the fluids used in research made into the possible uses of wire mesh packing may promote the increase of the intensity of heat exchange. It was concluded that the greater advantages of using this type of packing are gained for the case of rates of gas flow. In the channel with wire mesh packing the air was heated more and the flux of the exchanges heat was greater in comparison to an empty channel. In addition, the higher value of heat transfer coefficient takes the possibility to transport of definite heat flux at a smaller temperature difference between heating surface and fluid.

During the flow of water through the channel it was noted that the use of wire mesh packing does not bring satisfactory results in terms of an increase of the heat flux absorbed by through the liquid, however like for air, use packing reduce temperature of heating surface.

The reason for the lower intensity of heat exchange during the flow of water in the channel is associated with the additional thermal resistance encountered by water passing the layers of wire which are untied and not attached the channel's walls. The resistance of heat transfer in the packing plays a much less important role for the case of air passage. Under the conditions of considerably less intensive transfer of heat into gas than into liquid, the more important function is performed by the surface of heat exchange.

The high pressure drop of flow does decide on the considerably greater energy use in order to pump the fluids through wire mesh packing. Higher energy consumption (than in the empty channel) required for water pumping through the wire mesh packing is not compensated by an increase in the heat exchange of that packed channel. Therefore, from an energy point of view the use of wire mesh packing for heating liquids is unreasonable.

However, for the case of the air the high intensity of heat transfer in a packed channel leads to a 40% energetic gain (Eq. (19)) in comparison to an empty channel. The energetic gain exceeds 30% for the flow of air in the range of $Re_a = 74–190$.

The undertaken analysis of heat transfer conditions indicates that heat exchangers with wire mesh packing could find application, e.g. in the processes of heat accumulation from low temperature energy resources. In particular, they could be applied in the cases when heat exchangers are limited in terms of overall dimensions for numerous purposes.

Pressure losses during fluid flow through wire mesh packing is described by the Forchheimer equation. Concurrently, it noted that the value of permeability and inertial coefficient are different for water and air and amount to $K_w = 10.63 \times 10^{-9} \text{ m}^2$, $C_w = 559 \text{ m}^{-1}$, $K_a = 9.43 \times 10^{-9} \text{ m}^2$, $C_a = 710 \text{ m}^{-1}$, respectively.

The developed model of heat exchange (Eqs. (21)–(27)) is a development of the model of heat exchange during the passage of a fluid through a channel with wire mesh packing. This model enables one to forecast the value of heat transfer of gases and liquids with the precision range of around 10%.

6. Conclusions

Despite the potentially wide applicability in industrial apparatus the research made into the possible uses of wire mesh packing is scarce. The conducted experimental studies indicate the statement that use of wire mesh in the internal surface of exchangers may promote the increase of the intensity of heat exchange. It was concluded that the greater advantages of using this type of packing are gained for the case of rates of gas flow. In the channel with wire mesh packing the air was heated more and the flux of the exchanges heat was greater in comparison to an empty channel. In addition, the higher value of heat transfer coefficient takes the possibility to transport of definite heat flux at a smaller temperature difference between heating surface and fluid.

During the flow of water through the channel it was noted that the use of wire mesh packing does not bring satisfactory results in terms of an increase of the heat flux absorbed by through the liquid, however like for air, use packing reduce temperature of heating surface.

The reason for the lower intensity of heat exchange during the flow of water in the channel is associated with the additional thermal resistance encountered by water passing the layers of wire which are untied and not attached the channel's walls. The resistance of heat transfer in the packing plays a much less important role for the case of air passage. Under the conditions of considerably less intensive transfer of heat into gas than into liquid, the more important function is performed by the surface of heat exchange.

The high pressure drop of flow does decide on the considerably greater energy use in order to pump the fluids through wire mesh packing. Higher energy consumption (than in the empty channel) required for water pumping through the wire mesh packing is not compensated by an increase in the heat exchange of that packed channel. Therefore, from an energy point of view the use of wire mesh packing for heating liquids is unreasonable.

However, for the case of the air the high intensity of heat transfer in a packed channel leads to a 40% energetic gain (Eq. (19)) in comparison to an empty channel. The energetic gain exceeds 30% for the flow of air in the range of $Re_a = 74–190$.

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References


