Application Of Single Blow Technique For Heat Transfer Measurement In Packed Bed Of Vegetables

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ABSTRACT

The non-invasive measurement of the mean heat transfer coefficient for the packed bed of fruits and vegetables may be thought as still open issue. There is a clear need for the assessment of heat transfer conditions for various types of fruits and vegetables in order to accurately predict the thermal load that is necessary to select refrigeration equipment for cold storage chamber. Additionally, there is significant development in numerical modeling of heat and mass transfer processes in cold storage chambers for fruits and vegetables which requires precise heat transfer description.
The theoretical basis for the indirect measurement approach of heat transfer coefficient for the packed bed of vegetables that is based on single blow technique is presented and discussed in the paper. The approach based on the modified model of Liang and Yang was presented and discussed. The testing stand consisted of a dedicated experimental tunnel along with auxiliary equipment and measurement system are presented. The geometry of the tested matrices were presented. Selected experimental results of heat transfer and pressure drop are presented and discussed for the packed bed of Chinese cabbage. These results were presented as dimensionless relationships.

1. INTRODUCTION

The paper deals with approach of the measurement of heat transfer and flow resistance in packed bed of fruits with Chinese cabbage as an exemplary case. The knowledge on heat transfer conditions during cooling of vegetables and fruits is necessary for the design of the cold storage chamber.

Usually measurement of the flow resistance in packed bed of vegetables or fruits is not very difficult. However, the measurement of the heat transfer coefficient may be thought as a challenge in most cases due to very complicated geometry of the bed elements. This is the reason why it is impossible to apply the simplest direct methods of the measurement of the heat transfer coefficient that are based on the direct measurement of the temperature of the vegetables, gas temperature distribution as well as heat flux density. This means that in this case an indirect method must be applied. The methodology of such measurement is proposed in this paper.

2. METHODOLOGY OF HEAT TRANSFER COEFFICIENT MEASUREMENT

The so-called single blow technique is thought to be an efficient method used to experimentally determine the average heat transfer coefficient α in the packed bed of vegetables. Heat transfer coefficient is based on the actual surface area of the bed elements and takes into account convective heat transfer between gas and vegetable surface. In the discussed method the average heat transfer coefficient α to be found is determined by means of the comparison of the actual temperature profile of gas (that is heated or cooled in the tested packed bed) measured at the outlet of the tested packed bed with the predicted one on the basis of the theoretical model. The agreement between the experimental temperature profiles and theoretical prediction depends on the heat transfer coefficient α that is applied in the theoretical model of heat transfer. Various theoretical approaches may be applied in order to predict temperature profile at the outlet for given inlet conditions and the vegetables geometry. Below short review of theoretical models which may be applied in the single blow method is presented.

The measurement in the single blow method is to record the time variability of gas temperature profile directly at the inlet and outlet of the tested packed bed which is caused by switching the heating section on or off is of key importance for the discussed method. The following conditions need to be met before the measurement:

- gas flow is steady state (constant velocity);
- the temperature in the measurement section is constant and equal in the axial and radial directions.

Temperature profiles should be measured for different velocities, involving the whole range of Reynolds numbers expected in the range of the operating conditions of the packed bed.

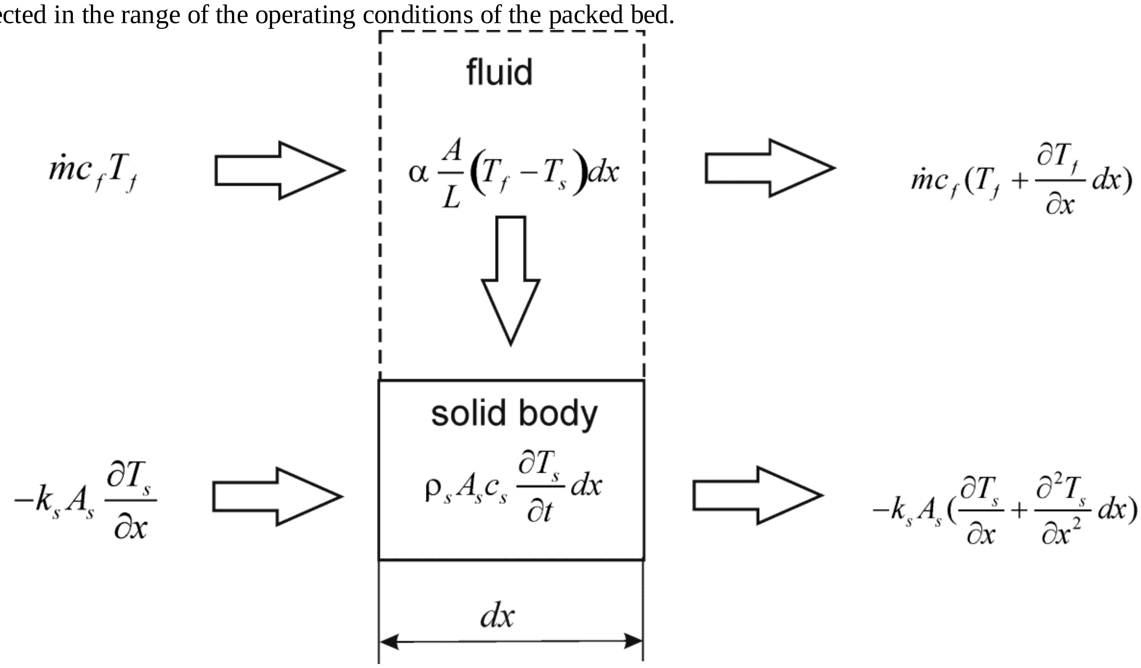

Figure 1. Energy balance for fluid (*f*) and solid body (*s*) for heater exchanger element

Temperature profiles recorded during the single blow technique measurements are used when solving equations modelling heat exchange in the tested packed bed. The measured temperature leap or profile at the inlet to the test section is the boundary condition for the model equations. The measured temperature profile at the outlet of the test section should be predicted on the basis of the model. However, the theoretical and the measured outlet temperature profiles may be compared in several ways that may be thought as equivalent.

The most basic one-dimensional model of transient heat transfer between a porous body and a fluid flowing through it is the model proposed by Anzelius (1926). This model became the basis for many further modifications and is still being developed by removing the numerous simplifications assumed by the author in order to obtain analytical solution of the model equations. Anzelius (1926) model is based on the energy balance for a solid body element (porous or perforated packing of the exchanger) in the situation presented in Fig. 1.

The following assumptions were made:

- the physical properties of fluid are independent of temperature;
- gas flow is steady state ($\dot{m}$ = const);
- the solid has a homogeneous structure;
- the thermal conductivity perpendicular to the flow direction is infinite (both for the fluid and for the solid body);
- the thermal conductivity of the fluid in the flow direction equals to zero;
- the external casing of the packed bed is adiabatic;
- initially the temperature in the exchanger is uniform ($T_s = T_f = T_i$);
- initially the fluid temperature leaps from T_i to T_{f1} and then it remains steady.

Balance equations of the element of fluid and solid body in the non-dimensional form are as follows:

$$\frac{\partial T_f}{\partial z} = T_s - T_f \,, \tag{1}$$

$$\frac{\partial T_s}{\partial \tau} = \lambda NTU \frac{\partial^2 T_s}{\partial z^2} + \left(T_f - T_s\right), \tag{2}$$

where non-dimensional time is defined as follows:

$$\tau = t \frac{\alpha A}{m_s c_s} \,, \tag{3}$$

and non-dimensional coordinate is defined as:

$$z = NTU \frac{x}{L} \,. \tag{4}$$

After the transformation of the above equations, two new coefficients appear in the energy balance for the solid body, namely the number of heat transfer units:

$$NTU = \frac{\alpha A}{\dot{m} c_f} \,, \tag{5}$$

and a parameter connected with solid body thermal conductivity k_s in the direction θ is parallel to the flow direction:

$$\lambda = \frac{k_s A_s}{\dot{m} c_f L} \,. \tag{6}$$

If the conductivity can be ignored, the above equations have analytical solutions developed by Schumann (1929) in the form:

$$\theta_s = 1 - e^{-(z+\tau)} \sum_{n=0}^{\infty} z^n \frac{d^n}{d\left(z\tau\right)^n} \left(J_0\left(2i\sqrt{z\tau}\right)\right), \tag{7}$$

$$\theta_f = 1 - e^{-(z+\tau)} \sum_{n=1}^{\infty} z^n \frac{d^n}{d\left(z\tau\right)^n} \left(J_0\left(2i\sqrt{z\tau}\right)\right), \tag{8}$$

where θ is non-dimensional temperature defined as follows:

$$\theta = \frac{T - T_i}{T_{f1} - T_i} \,. \tag{9}$$

The distributions of temperatures obtained from the above Schumann analytical solution can be used to determine the heat transfer coefficient α if the experiment well agrees with the boundary and initial conditions assumed in the model. Then, comparing the measured temperature distribution $T_f(t)$ at the packed bed (packed bed) outlet with the theoretical temperature distribution the spot $z = NTU$, the best fit will be achieved if the value α specific for the tested exchanger is used (Furnas, 1932).

Instead of comparing the temperature profiles, their derivatives can be used, and in particular their maximum values. Locke (1950) differentiated the solution for the outlet temperature $T_{f2} = T_f(\tau, z{=}NTU)$ at constant NTU and obtained the relationship for the slope S of the curve describing evolution of temperature T_{f2} in dimensionless coordinates. This makes it possible to determine the derivative from the experimental profile of outlet temperature $T_{f2}(t)$ and the experimental value S_{max}. For that value it is possible to obtain the corresponding NTU value, and then find the needed value of coefficient α.

Solving the model equations taking into consideration heat conductivity along the solid body is only possible with numerical methods. The necessary calculations were done e.g. by Howard (1964), and his findings in the form of tables and graphs of function $NTU = f(S_{max})$ for selected values λ within the range of [0.005 …10, ∞] are also included in the paper presented by Pucci et al. (1967). The effect of the parameter λ on the determination of NTU is significant, especially when $NTU > 10$. Due to the considerable slope of the curve $NTU = f(S_{max})$ this approach of

determination of α with the use of maximum slope S_{max} involves considerable error for certain combinations of parameters S_{max} and λ.
The accuracy of determining of heat transfer coefficient using the single blow method can be improved by extending the Anzelius model, providing that:

- the leap of the inlet temperature of the fluid occurs not immediately;
- heat conductivity in the solid body occurs not only in the axial direction but also in the radial direction;
- the external wall of the packed bed (packed bed) is not adiabatic; consequently there is a radial temperature gradient in the packed bed;
- pressure drop of the fluid flowing through the packed bed (packed bed) causes change of its temperature due to Joule-Thomson effect;
- the distribution of fluid velocity in the axial direction is heterogeneous due to disturbances of gas flow occurring at the elements of the packed bed (packed bed).

The modification of the initial conditions of Schumann solution proposed by Liang and Yang (1975) minimizes the problems connected with experimental realisation of the non-immediate temperature leap at the tested exchanger inlet. Liang and Yang model was further extended by Cai *et al*. (1984) by taking into consideration also the conductivity of the solid body in the axial direction. The solution of this model could be obtained by numerical procedure, however. In this case the temperature profile at the exchanger inlet can be any function of time. Chen and Chang (1996) added to the model an equation describing the heat transfer between fluid and the casing. They also took into account the Joule-Thomson effect (Chen and Chang, 1997), and finally also radial conductivity (Chang *et al*., 1999). Luo *et al*., 2001, highlighted the impact of disturbances of fluid flow caused by the elements of exchanger packing (so-called axial dispersion). Taking into consideration that the usually the measurements cover large number of test runs then the most useful will be the analytical solution of the temperature profile at the outlet of the tested packed bed, however.

One of the sources of inaccuracy of the heat transfer coefficient α determined by means of the single blow method is the difficulty of the realisation of the immediate temperature leap at the exchanger inlet. Due to the thermal capacity of the heater any temperature changes always occur at a certain time span. Liang and Yang (1975) proposed the dimensionless inlet temperature profiles obtained in the discussed test method after switching the heater on or off as the exponential function:

$$\theta_f(\tau,0) = 1 - e^{-\tau/\tau^*}, \tag{10}$$

where τ^* is an experimentally determined dimensionless constant. The above temperature profile as the boundary condition for modified Anzelius model equations (Liang and Yang, 1975). In the equation for fluid energy balance its thermal capacity was taken into consideration so the equation for gas energy balance was obtained as follows:

$$\frac{\partial \theta_f}{\partial \tau} + b_1 \frac{\partial \theta_f}{\partial z} = b_2\left(\theta_s - \theta_f\right), \tag{11}$$

and in the solid body the thermal conductivity was ignored so that Eq. (2) was used. In the dimensionless form this equation can be presented as follows:

$$\frac{\partial \theta_s}{\partial \tau} = \theta_f - \theta_s . \tag{12}$$

The coefficients b_1 and b_2 in Eq. (11) are constant and defined as follows:

$$b_1 = b_2 \frac{v_f}{A_{f,\min}}, \qquad b_2 = \frac{m_s c_s}{m_f c_f}, \tag{13}$$

where v_f is the volume of fluid inside the packed bed per length in the flow direction, and $A_{f,\min}$ is the smallest cross-section surface area of the packed bed available for the flowing fluid. It must be remembered that temperature T_{f1}, necessary to calculate θ should be the value settled at the inlet after a sufficiently long time after switching the heater on or off. Equations (11) and (12) with boundary condition Eq. (10) and the following initial conditions:

$$\theta_s(0, z) = \theta_f(0, z) = 0, \tag{14}$$

were solved (Liang and Yang, 1975) with the use of the Laplace transform which gave a set of equations:

$$p\bar{\theta}_f + b_1 \frac{d\bar{\theta}_f}{dz} = b_2\left(\bar{\theta}_s - \bar{\theta}_f\right), \tag{15}$$

$$p\bar{\theta}_s + = \bar{\theta}_f - \bar{\theta}_s, \tag{16}$$

$$\bar{\theta}_f\left(p,0\right) = \frac{1}{\tau^* p\left(p + \frac{1}{\tau^*}\right)}, \tag{17}$$

where $\bar{\theta}$ denotes the transform, and p is its argument. The solution to the set of Eqs. (15)–(17) in relation to $\bar{\theta}_f$ is as follows (Liang and Yang, 1975):

$$\bar{\theta}_f = \frac{\exp\left(-\frac{p}{b_1}\left(1 + \frac{b_2}{1+p}\right)z\right)}{\tau^* p\left(p + \frac{1}{\tau^*}\right)}. \tag{18}$$

Finding the inverse Laplace transform for Eq. (18) at $z = NTU$, the following relationships describing the profile of dimensionless temperature at the packed bed outlet are obtained:
a) if $\tau < t^*$ (i.e.: $t < L/w_f$):

$$\theta_f(\tau, NTU) = 0 \tag{19}$$

b) when $\tau \geq t^*$ (i.e.: $t \geq L/w_f$):

$$\theta_f\left(\tau, NTU\right) = \frac{1}{\tau^*}\int_{t^*}^{\tau} e^{-(\tau-\eta)/\tau^* - b_2 t^*}\left\{e^{-(\eta - t^*)} I_0\left(2\sqrt{b_2 t^*\left(\eta - t^*\right)}\right) + \int_0^{\eta - t^*} e^{-\xi} I_0\left(2\sqrt{b_2 t^* \xi}\right) d\xi\right\} d\eta. \tag{20}$$

In the above relationships the symbol t^* denotes dimensionless time defined as follows:

$$t^* = \frac{NTU}{b_1}, \tag{21}$$

and w_f is the mean velocity of fluid in the packed bed.

Provided the relatively high numerical cost connected with the application of more complex mathematical models of heat transfer in regenerators, the authors decided that the use the above modified Liang and Yang model (Liang and Yang, 1975). The key element in this model is the consideration of the exponential character of the inlet temperature profile which allows to avoid a considerable error resulting from the impossibility to execute an immediate leap of temperature at the inlet of the tested packed bed. It is recognized that the accuracy of this model should be satisfactory for indirect measurement of the heat transfer coefficient.

3. TEST APPARATUS

The experimental part of the single blow method is carried out in a wind tunnel. Temperature and pressure measurement points are placed at the inlet and outlet of the tested packed bed. Static pressure difference is measured in order to determine the frictional flow resistance. In addition, the velocity of gas flow through the packed bed should be also measured. It may be suggested that the temperature of gas flowing through the packed bed should be raised above the ambient temperature by approximately 10–20 K. An electrical heating coil made of resistance wires may be applied for this purpose. Due to the required velocity homogeneity, the heating coil should disturb the flow as little as possible. The structure of the coil also affects the gas temperature profile obtained after switching it on or off which is important from the point of view of the theoretical model applied to draw up the results of the measurements. The simplest models offer the analytical solutions for which an immediate temperature leap at the measurement section inlet is assumed. Some of the theoretical models assume the inlet temperature profile that is described by an exponential function achieved in the experiment. In more complicated numerical models there is no limitation concerning the profile of inlet temperature so the restrictions concerning the heating section (mainly its length in the direction of gas flow) are less important. Due to the homogeneity of velocity profiles required in the case of the theoretical models, it is recommended to place a flow straightener (in the form of a honeycomb packed bed) before the test section. In order to eliminate the influence of disturbances generated at the tunnel outlet geometry, a similar flow straightener should be also placed at the outlet of the test section. Reduction of the channel hydraulic diameter between the tunnel inlet and the test section is required in order to reduce turbulence level in the flow.

The test tunnel consists of three sections as it is show in Fig. 2. Air flow rate in the tunnel was controlled by means of change of the centrifugal exhaust fan rotational speed. The electric heater capacity was also controlled. The test stand consists of three parts:

- the inlet section with an electrical heater;
- the measurement section with the tested packed bed of vegetables;
- an exhaust fan with a diffuser.

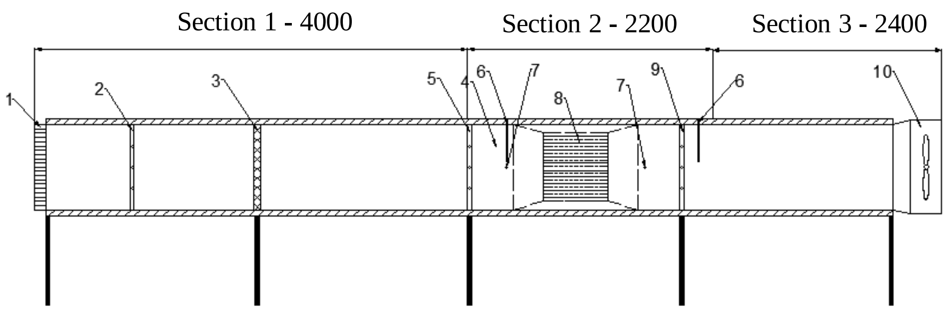

Figure 2. Schematic diagram of the test tunnel:
1 – flow rectifier; 2 – air flow rate and temperature sensors; 3 - electric air heater; 4 – confusor/diffuser; 5 – thermocouples net at the tested bed inlet; 6 – moisture sensors; 7 – pressure gauges; 8 – tested packed bed of vegetables; 9 – thermocouples net at the tested bed outlet; 10 –exhaust fan

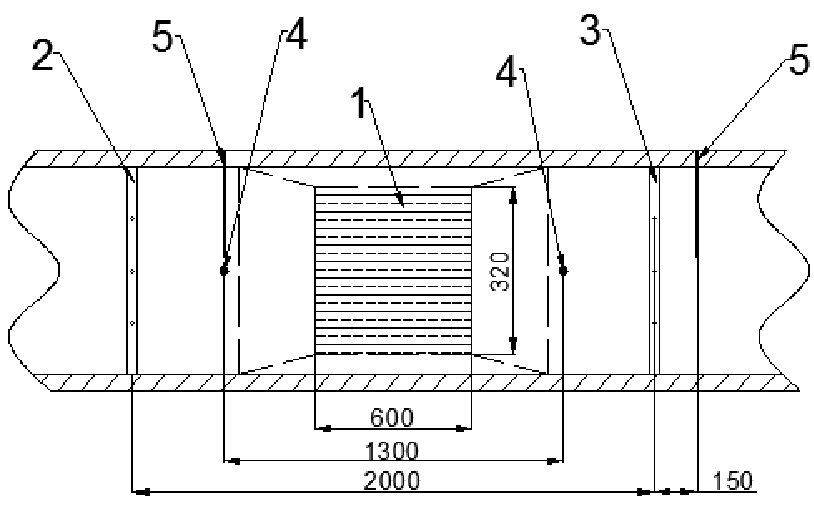

Figure 3. Schematic of the test section: 1- tested packed bed; 2- thermocouples net at the bed inlet; 3 - thermocouples net at the bed outlet; 4- pressure gauges; 5- air moisture sensors

The schematic of the test section is show in Fig. 3. The temperature distribution at the inlet and outlet to the test section were measured by means of the thermocouples nets. Based on these measurements the average temperature of air at the inlet and outlet were obtained under transient operating conditions.
The following basic parameters are measured on the test stand:

- air temperature at the tunnel inlet;

- air temperature at the tested bed inlet ;
- air temperature at the tested bed outlet;
- static pressure drop at the tested bed;
- dynamic pressure at the tunnel inlet;
- air humidity at the channel inlet and outlet;
- electric heater on/off signal.

Thermocouples of the type of TP201 J of the diameter 0.5 mm were applied. The thermocouples of the open junction type were applied which enabled low thermal inertia. The thermocouples were calibrated for three temperature levels with use of the calibration Pt100 sensor. The maximum difference between calibration sensor readings and thermocouples readings were not exceed ±0.15 K.

The reported measurements were carried out for the case of Chinese cabbage. The prepared packed bed of Chinese cabbage before insertion in the test tunnel were show in Fig. 4. The tested cabbage was inserted in the gauze container.

Figure 4. The packed Bed of Chinese cabbage prepared for insertion to the test channel

The dimensions of the prepared packed bed were as follows: width 0.400 m; height 0.320 m; length 0.600 m; number of cabbages 12. The detailed parameters of the tested bed were presented in Table 1.

Table 1.Parameters of the tested bed of Chinese cabbage

packed bed	mass [kg]	total volume [dm^3]	volume occupied by vegetables [dm^3]	volume of free space [dm^3]	porosity	outer surface area of vegetables [m^2]	density [kg/m^3]
	m_m	V	V_m	V_p	ε	A_p	ρ
I	1.209	2.88	1.12	1.76	0.61	0.145	1007.5
II	1.470	3.32	1.51	1.81	0.55	0.142	973.5
III	1.304	3.01	1.41	1.6	0.53	0.127	924.8
average	1.328	3.7	1.35	1.72	0.56	0.138	968.6

The average mass of the tested bed was 16.15 kg; the average mass of the single cabbage was 1.35 kg; average heat transfer surface area of the single cabbage 1.656 m^2. The average hydraulic diameter of the packed bed d_h may be assessed on the basis of porosity:

$$d_h = \frac{4A_p}{U_p} = 4\frac{V_p}{A_p} = 4\frac{\varepsilon}{a}, \tag{22}$$

where: A_p – surface area of free space; V_p – volume of free space; U_p – perimeter of free space; V - total volume of free space; ε - porosity (V_p/V); a – specific surface area (A_p/V).

4. TEST RESULTS

The obtained results based on the application of the described single blow technique for the packed bed of Chinese cabbage were presented in Table 2. It should be noted that Reynolds number *Re* was based on flow parameters in the equivalent channel inside the packed bed and *w* is velocity at the tested bed inlet. On the basis of the obtained experimental results the following correlation describing heat transfer in packed bed of Chinese cabbage may be proposed:

$$Nu = 0.374\,\mathrm{Re}^{0.934}\,\mathrm{Pr}^{0.33}. \tag{23}$$

Table 2. Results of heat transfer measurements

w [m/s]	*Δp* [Pa]	*Re*	*α* [W/m²K]	*Nu*	*Pr*	*j*	f_D
0.058	6.68	591	22	78.92	0.815	0.143	145.1
0.313	40.35	3176	62	200.96	0.736	0.070	30.27
0.507	83.24	5148	95	318.50	0.761	0.068	23.76
0.164	6.09	1677	30	111.69	0.821	0.071	17.97
0.307	23.66	3134	49	171.41	0.771	0.060	19.99
0.498	55.50	5063	80	272.78	0.751	0.059	17.94
0.699	120.0	7109	117	391.73	0.737	0.061	19.69

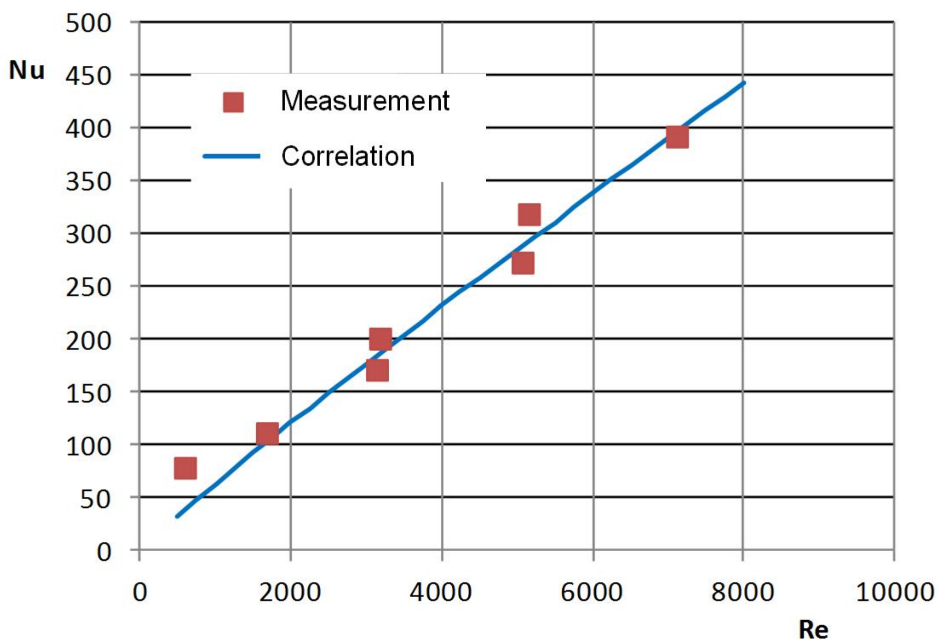

Figure 5. Comparison of measured Nusselt number *Nu* with proposed correlation eq. (23)

The comparison of the measurement results with proposed dimensionless relationship eq. (23) is presented in Fig. 5. It is worth to note that relatively high heat transfer coefficients were achieved in the reported tests. This may be attributed to the high heat transfer surface area enhancement of the tested cabbage since the average outer surface area was taken into account in the measurement procedure.

5. SUMMARY

This paper presents the methodology of the measurement of heat transfer coefficients for packed bed composed of the vegetables. The reported measurements were carried out for the case of Chinese cabbage. It is necessary to emphasize the necessity to develop an indirect method of measurement of the heat transfer coefficient by means of single blow technique. The current results involve the modification of this method proposed by Liang and Yang (Liang and Yang, 1975). The authors believe that the proposed methodology proved to be useful and produced reliable values of coefficients given by relationships eq. (23) for the tested packed bed of Chinese cabbage. Further development of the proposed methodology may be proposed for the case of the other vegetables.

NOMENCLATURE

A	– heat transfer area, m^2
A_s	– cross section, m^2
b_1,b_2	– coefficients (constants)
c	– specific heat at constant pressure, J/(kg·K)
d_h	– average hydraulic diameter of the packed bed, m
f	– Fanning flow resistance factor
f_D	– Darcy flow resistance factor
$\dot{G}$	– mass flux density, kg/(m^2·s)
I_0	– modified Bessel function of the first kind of zero order
j	– Colburn coefficient
J_0, J_1	– Bessel function of the first kind of zero and first order, respectively,
k	– heat conduction, W/(m·K)
L	– length of the packed bed, m
m	– mass, kg
$\dot{m}$	– mass flow rate, kg/s
Nu	– Nusselt number
NTU	– number of thermal units
p	– argument
Pr	– Prandtl number
Re	– Reynolds number
S_{max}	– maximum slope
t	– time, s
t^*	– dimensionless time
T	– temperature, °C, K
w	– mean velocity of fluid, m/s
x	– distance between inlet and outlet of the packed bed, m
z, y	– dimensionless distance to the packed bed inlet
α	– heat transfer coefficient, W/(m^2·K)
η	– variable integration
θ	– dimensionless temperature
λ	– dimensionless axial conduction parameter
ξ	– variable integration
ρ	– density, kg/m^3
τ	– dimensionless time
v	– volume of fluid inside the packed bed per length, m^3/m

subscripts:
f – fluid
i – initial value
s – solid (packed bed)
w – wall of exchanger
1, 2 – packed bed inlet and outlet, respectively

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