Original Paper

Measurement of convective heat transfer for various checker systems

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The heat transfer properties of six different checkerworks were studied using a special test equipment. The temperature was deliberately limited to 300 °C in order to measure the local temperature distribution of the air for the first time.

The flow in the regenerator channels shows a considerable non-symmetrical appearance and local strands. Its enthalpy can not be determined by the measurement of only one temperature. The heat transfer depends not only on the velocity of the flow (forced convection) but also on the temperature differences and resulting density differences of the fluid (free convection).

In consequence of the low operating temperature and the necessity to measure the air temperature at many points, sufficiently accurate values for the coefficient of heat transfer could only be achieved for a part of the parameter range of industrial regenerators.

Untersuchung der konvektiven Wärmeübertragung in verschiedenen Kammergitterungen

In einem Versuchsregenerator wurden sechs verschiedene Steinarten auf ihre wärmetechnischen Eigenschaften untersucht. Dabei wurde bewußt nur mit Temperaturen unterhalb von 300 °C gearbeitet, um erstmals die lokale Verteilung der Lufttemperaturen messen zu können.

Die Strömung in den Kanälen ist durch erhebliche Asymmetrien und lokale Strähnen gekennzeichnet. Ihre Enthalpie kann keinesfalls durch Messung einer einzigen Temperatur bestimmt werden. Der Wärmeübergang hängt nicht nur von der Strömungsgeschwindigkeit ab (sogenannte erzwungene Konvektion), sondern auch von Temperaturunterschieden bzw. von daraus resultierenden Dichteunterschieden im Gas selbst (sogenannte freie Konvektion).

Bedingt durch die niedrige Arbeitstemperatur und die erforderliche Berücksichtigung vieler Lufttemperaturen konnten ausreichend zuverlässige Wärmeübergangskoeffizienten nur für einen Teil des Parameterbereichs industrieller Regeneratoren ermittelt werden.

1. Introduction

The industrial glass melting process has a large demand for energy at a relatively high temperature level. For a rational utilization of energy resources regenerators and recuperators are used for heat recovery. Especially the regenerative air preheating has a long tradition in the glass industries, because it was one of the prerequisites for the development of the continuous glass melting furnace in 1867 by Siemens [1]. The principle has not changed very much since that time. The regenerators – always present in pairs – comprise a heat accumulator which stores a part of the thermal energy of the flue for a period of time and gives this energy back to the preheated air afterwards. Refractory bricks with a special arrangement are applied as accumulator. In the past only bricks with a rectangular shape (figures 1a and b) have been in use. Since a couple of years bricks with an intricate shape and thin walls have been preferred (figures 1c to f).

By the use of modern regenerators it is possible to recover 50 to 75 % of the enthalpy comprised in the hot flue gas and take it back to the system [2 to 4]. The exact value depends on the size, on the load and last but not least on the thermal behaviour of the checkerwork, i.e., on the coefficient of heat transfer. A precise knowledge of the rate of heat transfer is necessary for an optimal scaling [5 to 7] and constructing [8 and 9] of regenerators. Therefore, some investigations of this subject shall be presented in this paper.

2. Test scheme

The determination of the coefficient of heat transfer for regenerators is difficult because of the following reasons:

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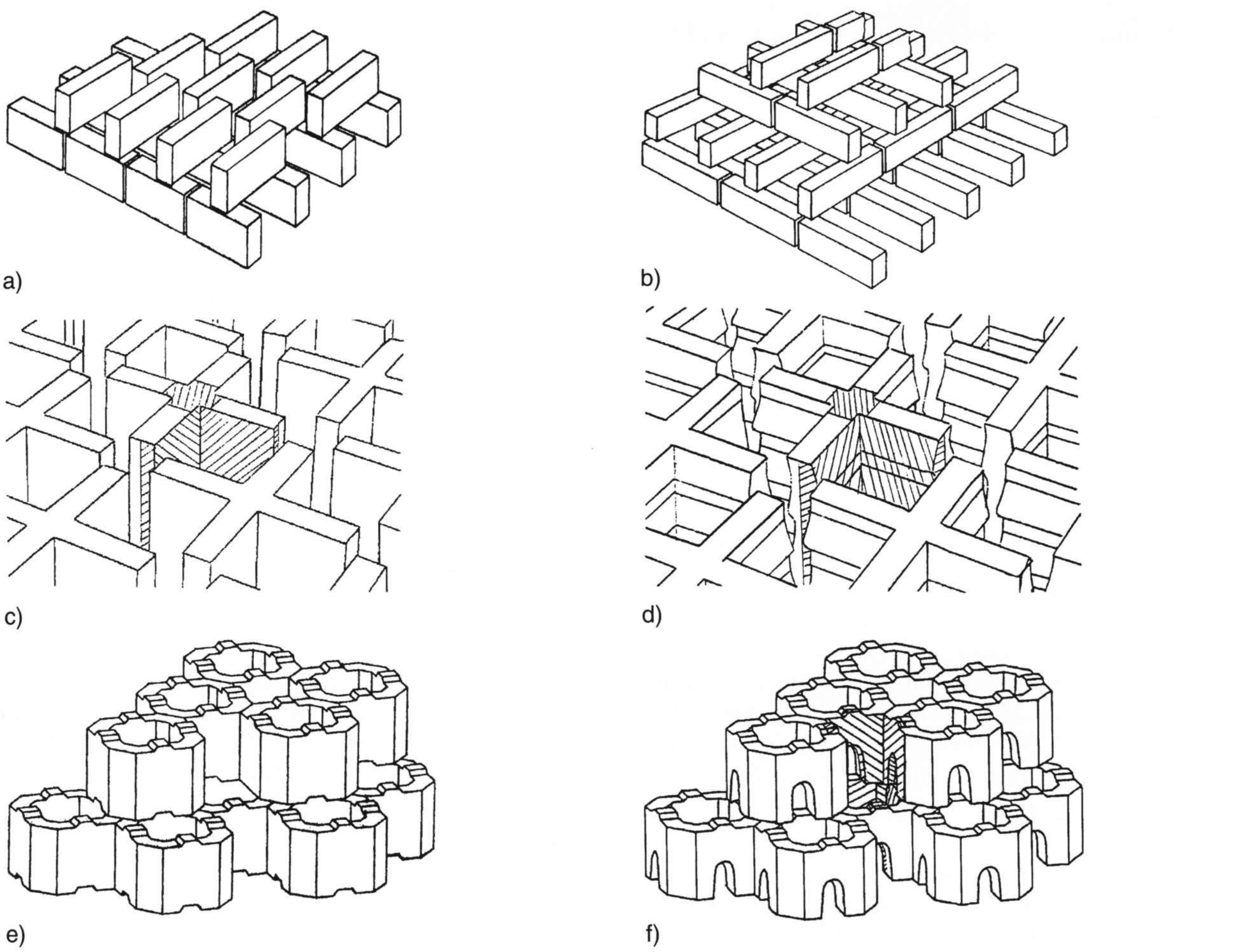

Figures 1a to f. Checkers under investigation, a) diagonally staggered pigeonhole packing; b) straight pigeonhole packing; c) smooth cruciform; d) corrugated cruciform; e) chimney block; f) open chimney block.

a) For investigations on industrial facilities [10] as well as for a realistic simulation in a test equipment [11 and 12] it is necessary to measure temperatures above 1000 °C. Due to the high contribution of the radiation to the whole heat transfer the measurement of these temperatures is quite expensive and not very accurate.
b) Even if one succeeds in measuring the air temperature at one point of the cross-section of a regenerator channel, further problems arise due to the inevitable temperature gradient perpendicular to the main flow. Such a gradient is necessarily present in any regenerator channel as soon as heat is transferred. If the temperature profile is not known exactly, no accurate specification of the heat transfer or of the characteristic temperature difference between brick and air is possible. In the work of other authors [10 to 12] the temperature was measured only at one point of the upper part of the regenerator, and therefore, the significant differences in enthalpy between the near-wall and the main flow simply were neglected.
c) It is well-known that the amount of heat transfer and as a consequence also the temperature in the fluid are affected by the flow. Under the influence of gravity a feedback can arise, which complicates the relations significantly. If a gas has locally different temperatures, these variations correspond with distinct changes in density. Such differences in density between horizontal neighbouring gas volumes lead under the influence of gravity to buoyancy and in consequence to a changed flow and finally to a different heat transfer. One talks of the influence of free convection. The mathematical treatment of the problem is sophisticated. For the measurement this means that for a certain checkerwork not only the mean velocity of the flow has to be varied but also the temperature difference between fluid and wall, as it also influences the flow and in consequence the heat transfer in a non-linear way.

These difficulties have contributed to the fact that the results of other authors [10 to 12] show only a poor coincidence. The present authors therefore tried to manage the problem in the present paper with a new test scheme:

– The maximum temperature of the test equipment was deliberately limited to 300 °C, while in industrial facilities temperatures above 1200 °C are quite common. The investigation of a high-temperature problem at a lower temperature level allowed to use bare thermocouples (Ni–CrNi) instead of suction pyrometers [13]. In consequence, it was possible to provide several measuring points in each cross-section of the checker.

- The periodical operation of a regenerator with a reversal time of regularly 20 min as it is typical for industrial facilities was replaced with an "infinite" reversal time, i.e., an undisturbed cooling of the bricks respectively heating up of the air. By this means a wide range of temperature differences in the fluid could be realized.

If the thermal performance of the checkerwork in the test equipment is sufficiently well-known at low temperatures, the heat transfer and similarity laws allow to calculate the performance of a regenerator with realistic temperatures and under periodical operation [14 to 17].

3. Description of test equipment and measuring system

The essential part of the test equipment (figure 2) is a regenerator with a height of 2 m (in the left part of the figure). This regenerator has a cross-section of $(0.65 \times 0.65)\,m^2$ and provides room for a checkerwork with 3×3 channels. The mean channel is designed for the measurement. The surrounding channels were intended to have a certain symmetry of temperature and flow. This was only achieved partly as the test evaluation has shown.

As shown in table 1 six different checkerworks made up of original bricks have been installed and tested in the test equipment. In order to simulate the charging of the regenerator with flue gas and air, the regenerator is surrounded by a pipe system, fans and a special heating equipment.

Twelve thermocouples have been arranged in each of the two cross-sections in the main channel to measure the temperature distribution in the fluid and in the surrounding bricks. At a certain flow path it is possible to measure the air velocity with a Prandtl tube and the corresponding temperature with a thermocouple in order to calculate the volume and mass flow of the air.

4. Measurement

During the testing period hot air flows from top to bottom through the regenerator (fan 1 in figure 2) in order to raise the temperature of the bricks up to about 250 °C. This phase simulates the waste gas period of an industrial regenerator. Afterwards the bricks are cooled down under the influence of cold air passing the regenerator from bottom to top (fan 2). The temperature of the air rises along its way through the channel. In this period for a range of different air velocities the amount of heat transferred between the two cross-sections containing the thermocouples is determined over a time of 10 min.

The heat flux $\dot{Q}$ is measured as a function of the air mass flow $\dot{m}_L$ and of the mean temperature difference $\Delta\vartheta$ between the surface of the brick and the air. The test equipment is designed to realize small temperature differences as well as large differences. In order to achieve this variety of temperatures, the air is either forced to flow in a circle or is partly or totally taken from the ambient. There is even the opportunity to heat the air at this stage by an electric heating system.

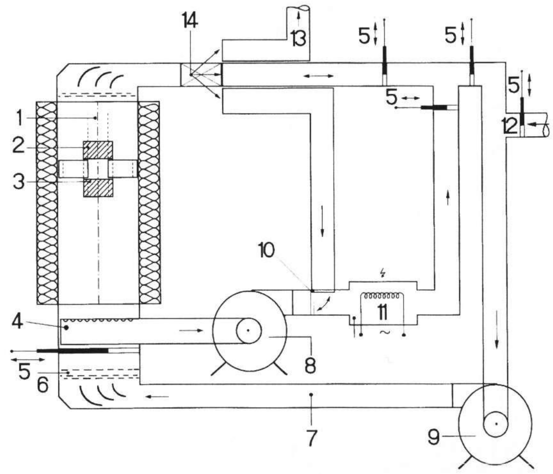

Figure 2. Test equipment. 1: mean channel, 2: upper test brick, 3: lower test brick, 4: hot air suction pad, 5: shut off gate, 6: velocity equalizing section, 7: velocity test section, 8: fan 1, 9: fan 2, 10: feeder gate, 11: electric heater, 12: inlet, 13: exhaust, 14: feeder gate.

5. Evaluation

The heat flux $\dot{Q}$ at the measuring point can be determined from the differences in enthalpy for the air between the two cross-sections. $\dot{Q}$ was calculated from the mass flow $\dot{m}_L$, the mean air temperatures ϑ_1 and ϑ_2 in each cross-section, and the specific heat capacity for air c_{pL} using the following formula

$$\dot{Q} \equiv \dot{m}_L\, c_{pL}\, (\vartheta_2 - \vartheta_1)\,. \tag{1}$$

To calculate the coefficient of heat transfer α, it is necessary to divide the value for the heat flux $\dot{Q}$ by the area A taking part in the heat exchange and by the mean temperature difference $\Delta\vartheta$ between surface and fluid

$$\alpha = \dot{Q}/(A\,\Delta\vartheta)\,. \tag{2}$$

In this paper the whole surface has been used to calculate the heating surface area A, i.e., even the parts in the lee are included equally in spite of their limited contribution to the heat exchange. The heating surface area was constant for the testing of each checkerwork as well as the cross-section area, while the fluid velocity u and the temperature difference brick/air $\Delta\vartheta$ were varied widely.

The evaluation at first gives values for the coefficient of heat transfer α as a function of the mean temperature difference $\Delta\vartheta$ and of the mass flow of the air $\dot{m}_L$, respectively of the mean flow velocity u. In the interest of a universal use of the results a non-dimensional presentation can be given. A non-dimensional equivalent of the coefficient of heat transfer α is given by the Nusselt number Nu, which is defined as

Table 1. Characteristics of the checkerwork

characteristics	chimney block		pigeonhole packing		cruciform	
	open	closed	diagonally staggered	straight	smooth	corrugated
height of brick in mm	150	150	124	150	420	420
thickness of brick in mm	40	40	64	64	30	30
length of brick in mm	–	–	250	300	130	130
passage width in mm	140	140	140	140	140	140
Reynolds factor D_H in mm	143	143	149	166	121	121
heating surface in area in m^2	0.1739	0.1642	0.1584	0.1370	0.2484	0.2659[1])
cross-section of channel in m^2	0.0191	0.0192	0.0271	0.0286	0.0195	0.0195

[1]) The heating surface area of the corrugated cruciform represents the corrugated area and not as also possible the projected surface, which would have the same value as the smooth cruciform.

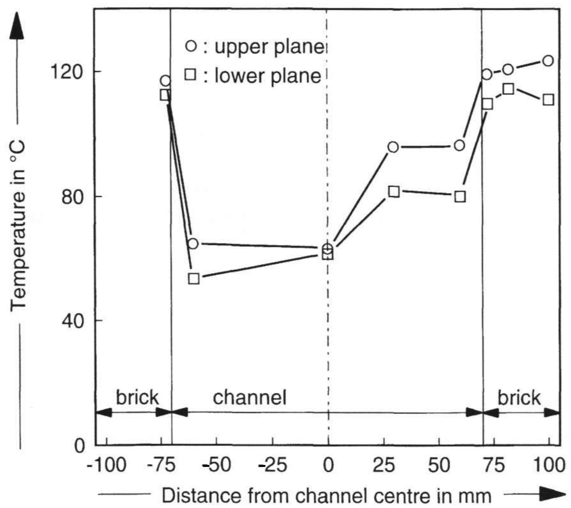

Figure 3. Temperatures in the regenerator channel.

$$Nu = \alpha D_H/\lambda \,. \tag{3}$$

In this equation λ means the thermal conductivity of air and D_H the Reynolds factor, which is a commonly used parameter for describing the flow in channels. The Reynolds factor is defined as the quotient of the fourfold of the channel cross-section F and the circumference in contact with the fluid U, i.e.,

$$D_H = 4F/U \,. \tag{4}$$

For the parameters u and $\Delta\vartheta$ a non-dimensional equivalent can also be given, namely in the form of the Reynolds number Re and the Grashof number Gr. The Reynolds number – as a non-dimensional flow velocity – describes the influence of the forced convection, and is defined by

$$Re = u\,D_H/\nu \,. \tag{5}$$

The Grashof number represents the influence of the free convection and is defined by

$$Gr = g\beta\Delta\vartheta D_H^3/\nu^2 \,. \tag{6}$$

In this formula g represents the acceleration due to gravity, ν the viscosity and β the coefficient of thermal expansion of air at a mean temperature.

6. Results

Figure 3 gives the temperatures in a vertical cross-section through the regenerator channel for an example. The two graphs show for the upper and lower cross-section the development of the temperature for the air and for the surrounding bricks. The test presented was performed with a diagonally staggered pigeonhole packing. The air entered the channel with ambient temperature and had in the channel a velocity of 0.19 m/s (normal state) corresponding to a Reynolds number of 1700.

It is seen that the temperatures in the upper cross-section are generally higher than the corresponding ones in the lower plane, that means an increase of enthalpy of the air in the direction of the flow. This corresponds with the temperature difference between brick and air. The increase in temperature for the air is in the centre of the channel less marked compared with the regions in the neighbourhood of the bricks. In some tests even a decrease of temperature could be observed for the measuring points at the channel axis. This points to a low convective cross exchange between the centre and the near-wall layers.

The temperature development of the bricks could only be measured at one side because the thermocouples were defective at the other side. The development of the temperatures of the bricks shows the expected gradient in the direction of the channel, corresponding with the heating-up of the air. The non-symmetrical appearance of the air temperatures relating to the channel axis is quite conspicuous. Even if further measuring points from planes behind or in front of the cross-section mentioned are introduced, none of the checkerworks shows a smooth graph. Also the symmetry with respect to the channel axis is quite poor. As a reason for the irregular appearance of the temperature profile, a non-symmetrical flow and local strands can be assumed as the thermal boundary conditions are almost symmetrical. The irregular appearance of the flow is surprising because the air is equalized by the means of sieves and baffle plates and the measuring path is located after a running-in path of about $10 \times D_H$. Unfortunately, it was not possible to measure the flow velocity in the channel, therefore, the explanation must remain uncertain.

As a consequence of the present measurements, the advice can be given to further investigation to provide even much more test points for the temperature and the corresponding velocity in order to determine and summarize the local enthalpy flow.

Figure 4 presents a summary of the results of all the testing. It shows the heat transfer presented by the Nusselt number *Nu* as a function of the Reynolds number *Re* for the six checkerworks under investigation. The Grashof number *Gr* was varied too as a second parameter.

The six different checkerworks show distinct differences in their thermal behaviour. The classic checkerwork types as the diagonally staggered or the straight pigeonhole packing show a higher Nusselt number than the modern bricks, which also show differences between each other. The good thermal behaviour of the rectangular bricks has to do with the forced or at least promoted flow exchange between the different channels. This special feature results in an improved heat transfer but also in an improved mass transfer, i.e., these bricks are more likely to be blocked and finally destroyed by deposits than the modern checkerworks. In the past as the lifetime of a glass melting tank was about two years, this was not as critical as nowadays where running times of over ten years are aimed at. The flow exchange between the channels complicates also the accurate measuring of the heat transfer for these types of checkerwork.

With regard to the differences between the various checkerworks one has also to take into account that for a calculation of the energy transported per square metre of regenerator volume not only the Nusselt number is important but also the specific surface, the wall thickness and the channel width.

For each checkerwork a distinct improvement of the heat transfer can be seen corresponding with an increase of the Grashof number, while the increase related to the Reynolds number is quite small. The Reynolds number as well as the Grashof number change considerably over the size of an industrial regenerator. Therefore, both parameters should be considered. For that purpose suitable data and basics for calculation have to be obtained first. The objective of this work is to contribute to this.

7. Numerical analysis and comparison with data from literature

If the results of this paper shall be the basis for numerical simulations then the development of a formula is necessary in addition to the presentation in a chart. A formula enables – in contrast to a diagram – to calculate the Nusselt number for any values of the independent variables *Re* and *Gr*. In addition, the comparison with data from literature becomes easier.

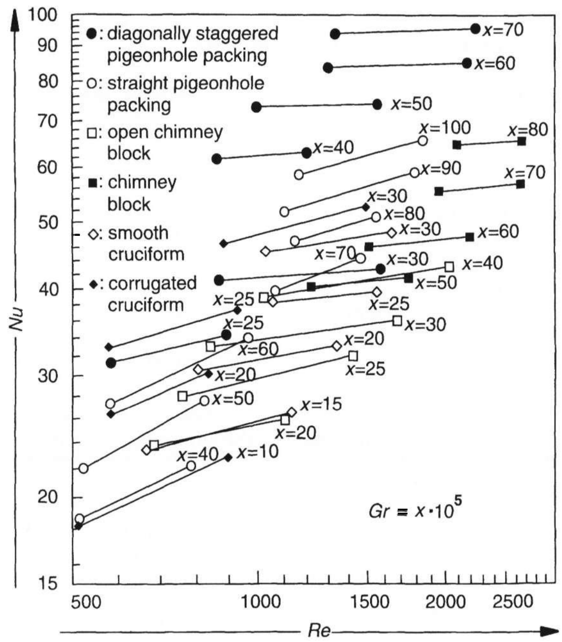

Figure 4. Nusselt number *Nu* as a function of the Reynolds number *Re* with the Grashof number *Gr* as a further parameter.

In general, there are many types of formula suitable for the empirical description of the heat transfer between a solid and a fluid. Formulas which consist of a product of powers of the independent non-dimensional factors are largely preferred [18]. For some elementary problems of this type this approach can be derived theoretically. For the laminar boundary layer at a slab [19]

$$Nu = c\,Re^{0.5} \tag{7}$$

or for the isotherm perpendicular wall [18]

$$Nu = c\,Gr^{0.25}. \tag{8}$$

Regarding the problem with two independent factors *Re* and *Gr* [18] something like

$$Nu = a Re^{m}\,Gr^{n} \tag{9}$$

will be obtained using the same pattern. Jeschar [12] has introduced this approach in 1961 for the non-dimensional description of the heat transfer of regenerators. He developed a new interpretation of some investigations of Kistner [20] regarding the pigeonhole packing. Kistner himself presented his results for the coefficient of heat transfer as a function of the standardized flow velocity and the Reynolds factor as follows:

$$\alpha = 8.7 \cdot \frac{u^{0.5}}{D_H^{0.333}} \tag{10}$$

with α in $W/(m^2 \cdot K)$, u in m/s, and D_H in m.

This formula and the corresponding measurements show no dependence on the temperature. Because such a dependence can be assumed from general considerations,

Table 2. Formulas for the Nusselt number Nu as a function of the Reynolds number Re and the Grashof number Gr for different checkerworks

type of checkerwork	formula
diagonally staggered pigeonhole packing	$Nu = 1.398 \cdot 10^{-2} \cdot (Re^{0.3} + 1.248 \cdot 10^{-3} \cdot Gr)^{0.9759}$
straight pigeonhole packing	$Nu = 1.715 \cdot 10^{-1} \cdot (Re^{0.3} + 9.44 \cdot 10^{-7} \cdot Gr)^{2.0256}$
smooth cruciform	$Nu = 5.174 \cdot 10^{-1} \cdot (Re^{0.3} + 5.602 \cdot 10^{-6} \cdot Gr)^{1.392}$
corrugated cruciform	$Nu = 2.727 \cdot 10^{-1} \cdot (Re^{0.3} + 1.916 \cdot 10^{-6} \cdot Gr)^{1.959}$
open chimney block	$Nu = 1.379 \cdot (Re^{0.3} + 2.856 \cdot 10^{-6} \cdot Gr)^{1.1264}$
chimney block	$Nu = 2.962 \cdot 10^{-1} \cdot (Re^{0.3} + 7.03 \cdot 10^{-6} \cdot Gr)^{1.2848}$

Jeschar has splitted the formula of Kistner in an exponent term containing the temperature-dependent Reynolds number and a second exponent term containing the Grashof number. The second term has the function to compensate the temperature dependence of the first term for the measured values of Kistner. The result was a formula of the type:

$$Nu = 0.2 \cdot \left(\frac{l}{D_H}\right)^{-0.166} \cdot Re^{0.5} \cdot Gr^{0.1} \cdot Pr^{0.333} . \qquad (11)$$

For the present problem a formula being composed of a product of exponent terms has a flaw in some respects. The formula is not capable to give correct results for the three important borderline cases $Re = 0$ and/or $Gr = 0$ (pure free or pure forced convection or pure conduction of heat). The numeric result always gives $Nu = 0$. It is well-known from physics that any convection – forced or free – improves the heat transfer, which is always present as soon as a temperature difference exists. There is even a certain amount of heat transfer without any motion ($Gr = 0$, $Re = 0$) which would be equivalent to a Nusselt number of about 5.

This aspect is quite important for the regenerators of the glass industries since the heat transfer is characterized by a relatively large variety of the independent variables. At the entrance of the regenerator the cold air has a high density, a low velocity and the temperature difference between brick and air is usually large. As the air temperature rises on the way though the regenerator, the density decreases, while the velocity speeds up and the temperature difference diminishes. As a result quite different conditions for the heat transfer and also different variables over the length of the regenerator will be found. In order to cope with this special feature a modified exponent formula has been developed, which is able to cover the borderline cases $Re = 0$ or $Gr = 0$,

$$Nu = a(Re^{b} + c \cdot Gr)^{d} . \qquad (12)$$

This approach has an additional factor to adapt the formula to the values. By calculating the minimum of the residues for the values of figure 4 the coefficients and exponents of the formula were determined. The results are shown in table 2.

For a comparison of the results of the present work with data from literature the measurements of Kistner [20] respectively Jeschar [12] for the straight pigeonhole packing, the measurements of Gramatte [11] for the straight pigeonhole packing, the smooth cruciform and the open chimney block, and the measurements of Barklage-Hilgefort [21] for the smooth cruciform have been selected. These works have been chosen because they offer the possibility to calculate the heat transfer not only as a function of the Reynolds number but also as one of the Grashof number according to the intention of the present paper. For the evaluation the factors presented by the authors in their works were not used but the complete evaluation was again done in a uniform way. The Reynolds factor D_H was used as the characteristic diameter and the mean of the air and brick surface temperature for calculating the air properties. The calculation of the heat exchange surfaces area was based as far as possible on the definitions of the present paper. The differences between these new values and the data given in the original papers have been quite small.

In the data from literature for the Grashof number mainly values below 10^6 are found as they are typically for the regenerators of the glass industries, while in this paper most values are above 10^6. For the few points where the Grashof numbers overlap, the Reynolds numbers in the literature are above 2000, while the Reynolds numbers in this paper are between 500 and 1500. It was not possible to find any data from literature fitting the authors' values exactly.

In order to carry out the comparison in spite of those difficulties a weighed Reynolds number Re_g was defined as follows $Re_g = (Re^{b} + c \cdot Gr)^{1/b}$. If $Gr = 0$, the weighed Reynolds number has the same value as the normal Reynolds number. With this definition the formulas from table 2 can be transformed as seen in table 3.

Figure 5 shows the data from literature (closed symbols) and the authors' values (open symbols) as a function of the Reynolds number Re. The data from literature form a band with a not negligible deviation. A slight increase of the Nusselt number corresponding with the Reynolds number can be seen. The values of the present paper are generally higher although they have lower Reynolds numbers. Those differences could

Table 3. Formulas for the Nusselt number Nu as a function of the weighed Reynolds number Re_g for different checkerworks

type of checkerwork	formula	weighed Reynolds number
diagonally staggered pigeonhole packing	$Nu = 1.398 \cdot 10^{-2} \cdot Re_g^{0.2928}$	$Re_g = (Re^{0.3} + 1.248 \cdot 10^{-3} \cdot Gr)^{1/0.3}$
straight pigeonhole packing	$Nu = 1.715 \cdot 10^{-1} \cdot Re_g^{0.6077}$	$Re_g = (Re^{0.3} + 9.44 \cdot 10^{-7} \cdot Gr)^{1/0.3}$
smooth cruciform	$Nu = 5.174 \cdot 10^{-1} \cdot Re_g^{0.4176}$	$Re_g = (Re^{0.3} + 5.602 \cdot 10^{-6} \cdot Gr)^{1/0.3}$
corrugated cruciform	$Nu = 2.727 \cdot 10^{-1} \cdot Re_g^{0.5877}$	$Re_g = (Re^{0.3} + 1.916 \cdot 10^{-6} \cdot Gr)^{1/0.3}$
open chimney block	$Nu = 1.379 \cdot Re_g^{0.3379}$	$Re_g = (Re^{0.3} + 2.856 \cdot 10^{-6} \cdot Gr)^{1/0.3}$
chimney block	$Nu = 2.962 \cdot 10^{-1} \cdot Re_g^{0.3854}$	$Re_g = (Re^{0.3} + 7.03 \cdot 10^{-6} \cdot Gr)^{1/0.3}$

be explained if different Grashof numbers have a substantial influence on the heat transfer as asserted in this paper.

Figure 6 shows the data from literature (closed symbols) and the authors' values (open symbols) for the Nusselt numbers as a function of the weighed Reynolds number Re_g. First the result of the adaptation is seen, i.e., the good fitting of the values of the authors with the graphs representing the functions of table 3. An improved coincidence of the values can be stated especially for the straight pigeonhole packing. The values from literature are slightly higher than the authors' ones but they show a similar course. The relative differences increase for small values of the weighed Reynolds number.

The coincidence is quite excellent for the open chimney block, however, the overlapping for the weighed Reynolds number is quite small for the data from literature and values of the authors. For the smooth cruciform the coincidence is relatively poor. In this case the values from literature scatter independently of the abscissa chosen.

In figure 6 it can be seen that the parameter range of the authors' measurements and that found in typical industrial regenerators presented by the data from literature only slightly overlap. The authors' values cover the parameter range of relatively high weighed Reynolds numbers, and can only provide an extrapolation for the sector of small and medium weighed Reynolds numbers. Therefore, no guarantee can be given for this parameter range which is also important for industrial regenerators. In order to get accurately measured values for all working conditions characteristic to industrial regenerators it would be necessary to adjust and measure temperature differences between wall and air of less than 0.2 K [22]. This is a consequence of the scaling of the present model and could not be achieved with the test equipment used in this work.

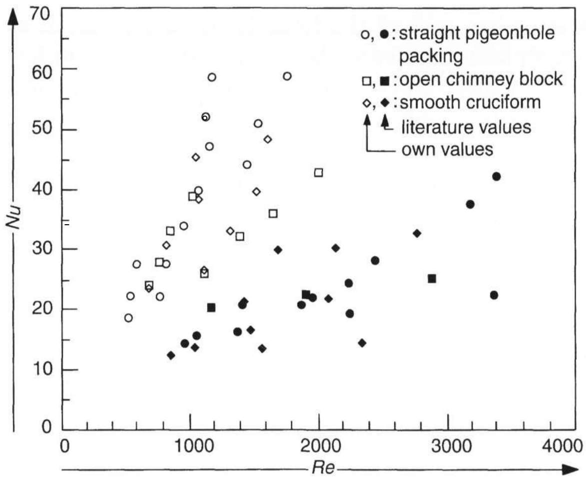

Figure 5. Nusselt number Nu as a function of the Reynolds number Re.

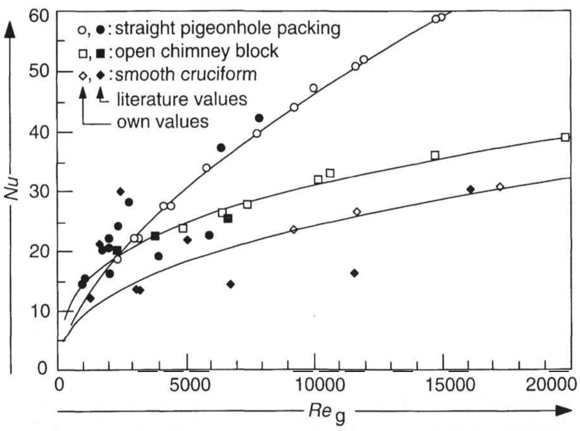

Figure 6. Nusselt number Nu as a function of the weighed Reynolds number Re_g.

8. Summary

The thermal behaviour of six different checkerworks typically used in the regenerators of the glass industries was tested. By means of measuring the local temperature distribution in the channels it could be shown that the flow has a considerable non-symmetrical appearance and local strands. A new numerical approach allows the calculation of the heat transfer dependent on the forced and free convection for a certain range of parameters.

The measurements at the test equipment and the calculation of the coefficient of heat transfer respectively the non-dimensional factors were performed by the Hochschule für Architektur und Bauwesen (HAB), Weimar (Germany). The Hüttentechnische Vereinigung der

Deutschen Glasindustrie (HVG), Frankfurt/M. (Germany), did the numerical analysis and the comparison with the data from literature.

9. Nomenclature

9.1 Symbols

A	heating surface area in m^2
c_{pL}	specific heat capacity for air in J/(kg·K)
D_H	Reynolds factor in m
F	cross-section area of the channel in m^2
g	acceleration due to gravity in m/s^2
Gr	Grashof number
l	thickness of one layer of bricks
$\dot{m}$	mass flow of air in kg/s
Nu	Nusselt number
$\dot{Q}$	heat flux in W
Re	Reynolds number
u	flow velocity of the air in m/s
U	circumference of the channel in m
α	coefficient of heat exchange in $W/(m^2 \cdot K)$
β	coefficient of thermal expansion in 1/K
λ	thermal conductivity of air in W/(m·K)
ϑ_1, ϑ_2	air temperature in °C
$\Delta\vartheta$	temperature difference between surface and fluid in K
ν	viscosity of air in m^2/s

9.2 Indices

1	cross-section 1 (bottom)
2	cross-section 2 (top)
L	air
p	isobar

*

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