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On the prediction of single-phase forced convection heat transfer in narrow rectangular channels

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ABSTRACT

In this paper, selected heat transfer correlations for single-phase forced convection are assessed for the case of narrow rectangular channels. The work is of interest in the thermal-hydraulic analysis of the Jules Horowitz Reactor (JHR), which is a research reactor under construction at CEA-Cadarache (France).

In order to evaluate the validity of the correlations, about 300 tests from the SULTAN-JHR database were used. The SULTAN-JHR program was carried out at CEA-Grenoble and it includes different kinds of tests for two different vertical rectangular channels with height of 600 mm and gap of 1.51 and 2.16 mm. The experimental conditions range between 2 - 9 bar for the pressure; 0.5 - 18 m/s for the coolant velocity and 0.5 - 7.5 MW/m² for the heat flux (whose axial distribution is uniform). Forty-two thermocouples and eight pressure taps were placed at several axial locations, measuring wall temperature and pressure respectively.

The analysis focused on turbulent flow with Reynolds numbers between 5.5×10^3 - 2.4×10^5 and Prandtl numbers between 1.5 - 6. It was shown that standard correlations as the Dittus-Boelter and Seider-Tate significantly under-estimate the heat transfer coefficient, especially at high Reynolds number.

Other correlations specifically designed for narrow rectangular channels were also taken into account and compared. The correlation of Popov-Petukhov in the form suggested by Siman-Tov still under-estimates the heat transfer coefficient, even if slight improvements could be seen. A better agreement for the tests with gap equal to 2.16 mm could be found with the correlation of Ma and the one of Liang. However the heat transfer coefficient when the gap is equal to 1.51 mm could not be predicted accurately. Furthermore these correlations were based on data at low Reynolds numbers (up to 13000) and low heat flux, so the use of them for SULTAN-JHR may be questionable.

According to the authors' knowledge, existing models of heat transfer coefficient in narrow channels have not been developed for high Reynolds number and high heat fluxes. Therefore, a new modified version of the Dittus-Boelter correlation was derived from a best-fitting of the SULTAN-JHR data with a multiple linear regression approach.

The current study highlights that the channel geometry can impact the heat transfer. In particular, a reduction in gap size leads to an enhancement in the heat transfer coefficient.

KEYWORDS

Narrow rectangular channel, Heat transfer correlation, Single-phase forced convection, Nuclear research reactor, SULTAN-JHR

1. INTRODUCTION

The study of the thermal-hydraulics of narrow channels is of interest in several research nuclear reactors, e.g. the Jules Horowitz Reactor (JHR). The JHR [1] is a material testing reactor under construction in France, at CEA-Cadarache. The nuclear fuel assemblies consist of cylindrical concentric fuel plates arranged in such a manner that the coolant flows through narrow channels. In normal operations, these fuel assemblies are designed to experience large heat fluxes (up to 5.5 MW/m^2), thus high core coolant velocities (up to 15 m/s) are also needed.

For the safety analysis of the reactor, the thermal-hydraulic system code CATHARE [2] is employed. This code is based on a transient 2-fluid 6-equation model, complemented with proper correlations for single-phase and two-phase flows. The CATHARE code has been extensively validated for the simulation of conventional Light Water Reactors (LWRs) [3]. However, further work is required to assess the modeling capabilities of CATHARE in respect of the JHR, whose core design and operational conditions differ from the case of commercial reactors. For this purpose, the SULTAN-JHR experimental database [4] can be used. These experiments were carried out at CEA-Grenoble in order to investigate the thermal-hydraulics of narrow rectangular channels under conditions that are typical of the JHR and to provide data for code validation. The rectangular geometry was chosen to avoid the technical difficulties in building a test section with curved plates (such as in JHR). The curvature of the plates in JHR is however relatively small, so that it is believed to influence only marginally the flow and the heat transfer. The geometry of the test sections was designed to have gap sizes and hydraulic diameters representatives of the JHR channel geometry.

A preliminary analysis of the SULTAN-JHR database was already performed [4]. This study evaluated friction correlations for pressure drop and the correlation of Colburn for single-phase forced convection heat transfer, which was found to significantly under-estimate the heat transfer coefficient.

The objective of the current work was to test the correlation of Dittus-Boelter [5] that is implemented in CATHARE for the prediction of the heat transfer in single-phase forced convection, against the SULTAN-JHR experimental data. In view of possible improvements of CATHARE, additional correlations were considered, namely the correlation of Sieder-Tate [6], Gnielinski [7], Popov-Petukhov [8], Liang et al. [9] and Ma et al. [10]. The analysis showed that these correlations provide unsatisfactory results, especially for flows with high Reynolds numbers and high heat fluxes. Therefore ad-hoc correlations were developed from a best-fitting of the SULTAN-JHR data.

The paper is organized as follows: in the next section a description of the SULTAN-JHR facility and of the experiments is given; in section 3 the methodology applied is explained; in section 4 the results obtained for the various correlations are discussed; in section 5 conclusions are drawn.

2. THE SULTAN-JHR EXPERIMENTAL CAMPAIGN

The SULTAN-JHR experimental campaign was performed at CEA Grenoble (France) during the years 2001-2008. One of the objectives was to provide a reliable set of data for the validation of the thermal-hydraulic codes when applied to the modeling of the JHR.

The test section consisted of a narrow vertical rectangular channel that is uniformly heated and where demineralized and degassed water flows upward.

About 300 steady-state tests were carried out. The experimental conditions, which are summarized in Table 1, were selected according to those expected in the JHR core.

Table 1. Selected range of parameters in the SULTAN-JHR experiments.

Outlet pressure [bar]	2 - 9
Inlet water temperature [°C]	25 - 160
Mass flow rate [kg/s]	0.05 - 2.0
Flow velocity [m/s]	0.5 - 18
Uniform heat flux [MW/m²]	0.5 - 7.5

2.1. Test Section Geometry

Two different test sections were used: section 3 (SE3) and section 4 (SE4) with channel gap equal to 1.509 mm and 2.161 mm respectively. As shown in Fig. 1, the channel is delimited by two Inconel-600 plates that are approximately 1 mm thick. The power was then supplied via direct electrical heating of the plates. The extremities of the walls are thinner in order to avoid heat concentration effects that may cause early boiling and potential thermal crisis at the corners.

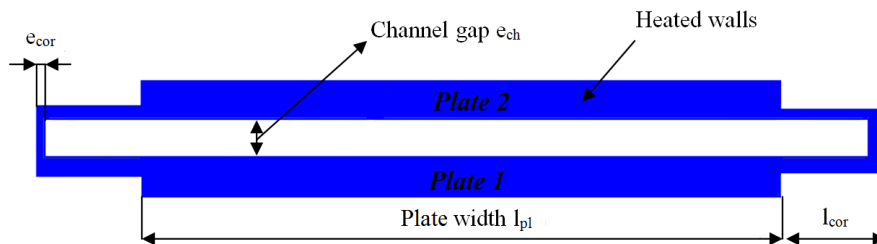


Fig. 1 Cross-section of the SULTAN-JHR test section.

The test section is encapsulated by an electrical mica-based insulation (*Cogetherm*[®]) and two pressure steel plates which maintain the channel gap and geometry reasonably constant during all tests. In fact, the gap size was proven to be quite constant along the channel, comparing the different pressure drop measured in the isothermal tests. On the external side, the test section is thermally insulated with 200 mm of rock wool so that heat losses could be reduced.

The dimensions of the test section with the associated nomenclature are reported in Table 2.

Table 2. Test section geometry (dimensions in mm)

	SE3	SE4
Gap size (e_{ch})	1.509 ± 0.040	2.161 ± 0.050
Heated height (H_{ch})	599.8 ± 0.1	599.7 ± 0.1
Adiabatic zone height (H_{ad})	70.0 ± 0.1	70.0 ± 0.1
Channel width without corners (l_{pl})	47.2 ± 0.1	47.15 ± 0.1
Length of the corners (l_{cor})	3.15 ± 0.1	2.85 ± 0.1
Thickness of the corners (e_{cor})	0.5	0.5
Averaged thickness of plate 1 (\bar{e}_{pl1})	1.0087 ± 0.006	1.003 ± 0.002
Averaged thickness of plate 2 (\bar{e}_{pl2})	0.9818 ± 0.018	1.004 ± 0.002

As can be seen in Fig. 2, the axial layout of the channel includes one heated and two adiabatic regions. The heated zone corresponds to the central part of the rectangular channel, where the largest part of the electrical power is transferred to the fluid. Along this zone, the heat flux is approximately uniform. The two adiabatic zones are at the entrance and at the exit of the test section. Each of them is 70 mm long. In these zones, the power (and consequently the heat flux) is much lower compared to the heated zone (approximately 2 % of the total electrical power). A smooth entrance in the test section was used in order to minimize the entrance effects on the measurements.

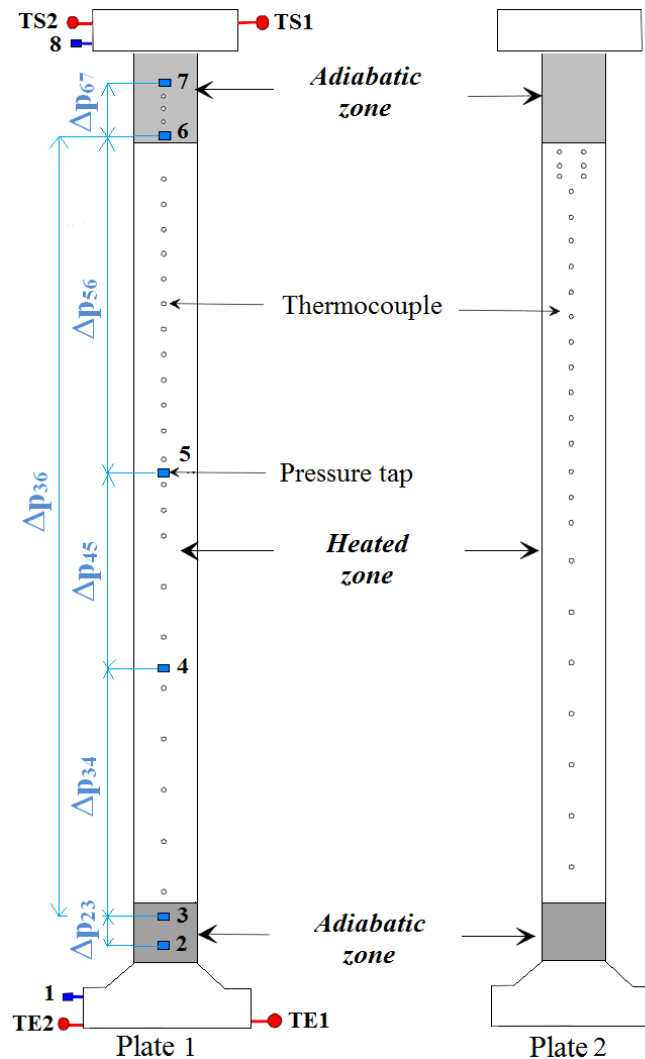


Fig. 2 SULTAN-JHR axial geometry and instrumentation layout.

2.2. Instrumentation

The arrangement of the thermocouples and pressure taps on the two Inconel-600 plates is displayed in Fig. 2. The 8 pressure taps (0.5 mm in diameter) were placed on plate 1 at different locations: at the entrance (PE1) and the exit (PS8) of the test section, in the adiabatic zones (P2, P3, P6 and P7) and in the heated zone (P3 and P4). The following pressure differentials were measured: P2-P3; P6-P7; P3-P6; P3-P4; P4-P5; P5-P6. The water temperature was measured at the entrance (TE1, TE2) and at the exit (TS1, TS2) of the test section with a platinum probe.

The dry wall temperatures were measured with insulated K-thermocouples (1 mm in diameter) centrally located on both the plates at 42 axial locations along the heated channel. The thermocouples were placed in the insulation layer, according to the simplified scheme in Fig. 3. The glue was introduced to guarantee the direct contact between the thermocouple and the insulation layer.

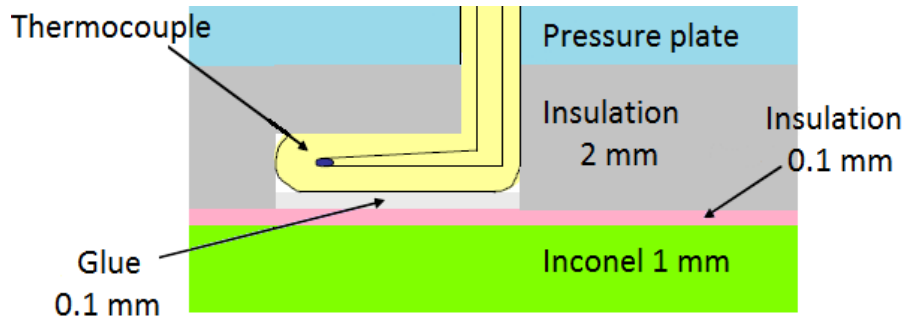


Fig. 3 Simplified thermocouple layout.

Furthermore the mass flow rate, the voltage ΔV between the pressure taps P3 and P6 and the direct electrical current I were measured. The electrical power released in the test section was then computed as:

$$P = \Delta VI \quad (1)$$

The measurement uncertainties [4] are shown in Table 3.

Table 3. Measurement uncertainties.

Flow rate	$\pm 1 \%$	Electrical power	$\pm 1.4 \%$
Absolute pressure	$\pm 0.8 \%$	Differential pressure	$\pm 0.8 \%$
Fluid temperature	$\pm 0.25 \text{ }^\circ\text{C}$	Dry wall temperature	$\pm 1.5 \text{ }^\circ\text{C}$

3. EXPERIMENTAL DATA REDUCTION AND MODELING

For the assessment of the heat transfer correlations for single-phase turbulent flow with respect to the SULTAN-JHR data, the experimental and the predicted Nusselt numbers are compared. The experimental Nusselt number can be calculated according to its standard definition:

$$Nu = \frac{hD_h}{k} \quad (2)$$

where h is the experimental heat transfer coefficient, D_h is the hydraulic diameter and k is the thermal conductivity.

The hydraulic diameter is equal to:

$$D_h = \frac{4A}{P_w} = \frac{2 e_{ch} l_{ch}}{e_{ch} + l_{ch}} \quad (3)$$

where A indicates the flow area, P_w is the wetted perimeter, e_{ch} is the gap size and l_{ch} is the

channel width. The latter reads:

$$l_{ch} = l_{pl} + 2(l_{cor} - e_{cor}) \quad (4)$$

being the notation consistent with the one used in Table 2.

The experimental heat transfer coefficient is defined as the ratio between the imposed local heat flux ϕ and the difference between the wet wall temperature T_w and the bulk temperature T_b :

$$h = \frac{\phi}{(T_w - T_b)} \quad (5)$$

A relative uncertainty of $\pm 20\%$ was estimated on the heat transfer coefficient.

The estimation of the bulk temperature is based on the heat balance for the fluid (neglecting the axial conduction), which gives the fluid enthalpy i as a function of the axial distance z :

$$i(z) = i_{z=0} + \frac{\phi P_w}{\dot{m}} z \quad (6)$$

where \dot{m} indicates the mass flow rate.

The wet wall temperatures T_w are derived from the dry wall temperatures T_{dw} , which are measured quantities. The one-dimensional Fourier's law of conduction can be applied to calculate the wet wall temperatures as described in [4]:

$$T_w = \frac{1}{a} \left(\sqrt{b^2 - a(e_{pl}(\phi - 2\phi_{loss}) - aT_{iw}^2 - 2bT_{iw})} - b \right) \quad (7)$$

where e_{pl} stands for the thickness of the Inconel-600 plates, ϕ_{loss} takes into account the heat losses experimentally evaluated, T_{iw} is the temperature at the interface between the mica-based insulation *Cogetherm*[®] and the Inconel-600 plate and a , b are the coefficients coming from the linear correlation for the Inconel conductivity, according to the formulation $k_{inconel} = b + a \times T$. The temperature T_{iw} reads:

$$T_{iw} = T_{dw} + \phi_{loss} \left(\frac{e_{co}}{k_{co}} + \frac{e_{gl}}{k_{gl}} \right) \quad (8)$$

where e_{co} , e_{gl} are the thicknesses of the *Cogetherm*[®] and glue layer respectively and k_{co} , k_{gl} are the thermal conductivities. The errors on the wet wall temperatures were obtained with a propagation of the experimental uncertainties and vary between ± 1.6 and ± 6.3 °C, depending on the value of the heat flux and of the wall temperature.

As shown in [4], the choice of the thermal conductivity for the Inconel-600 plates plays an important role in the determination of the temperatures. For the current study, an extended literature review combined with the analysis of two sets of measurements for the conductivity led to the decision of using the Corsan conductivity [11]:

$$k_{inconel} \left[\frac{W}{m \cdot ^\circ C} \right] = 12.12 + 0.0178 \times T [^\circ C] \quad (9)$$

This conductivity was obtained with a direct method based on the measurements of the heat flux and the wall temperatures in a sample. It was found to be consistent with the conductivity evaluated at CEA with a flash laser indirect method [4]. A maximum experimental uncertainty of 2.8 % and a standard deviation of 0.214 were reported in [11].

As regards the prediction of the Nusselt number by making use of correlations (see next section), the Reynolds and Prandtl number are required:

$$Re = \frac{GD_h}{\mu} \quad (10)$$

$$Pr = \frac{\mu c_p}{k} \quad (11)$$

where G stands for the mass flux, μ is the fluid dynamic viscosity and c_p is the specific heat. In these calculations, the physical properties of the fluid are evaluated at the bulk temperature and pressure of the fluid.

The pressure drops are estimated according to the following expression:

$$\Delta p = \Delta p_{grav} + \Delta p_{fric} + \Delta p_{acc} \quad (12)$$

The acceleration term was neglected since the analysis focuses on single-phase flow (in the SULTAN-JHR tests, $\Delta p_{acc} < 10^{-4} \cdot \Delta p$). The gravity and the friction term read respectively:

$$\Delta p_{grav} = \rho g \Delta z \quad (13)$$

$$\Delta p_{fric} = f \frac{\Delta z}{D_h} \frac{G^2}{2\rho} \quad (14)$$

The friction factor is modeled as:

$$f = F_{cor} \times f_{iso} \quad (15)$$

In the equation above, the friction factor f_{iso} is optimized over the isothermal SULTAN-JHR experiments:

$$f_{iso} = 0.202 Re^{-0.196} \quad (16)$$

F_{cor} is the corrective factor for the diabatic tests taking into account the heat flux influence on the friction. It is given by:

$$F_{cor} = 1 - \frac{P_h}{P_w} \frac{0.0085(T_w - T_b)}{1 + 2 \left[\frac{T_w + T_b}{200} \right]^{1.5}} \quad (17)$$

where P_h and P_w indicates the heated and the wetted perimeter respectively.

4. RESULTS

In this study, the analysis focuses on the SULTAN-JHR experiments which are characterized by single-phase turbulent flow. The Reynolds number varies between 5.5×10^3 and 2.4×10^5 ; the Prandtl number is between 1.5 and 6 and the wet wall temperatures T_w are well below the saturation temperature (by at least 5 °C).

Only the central part of the test section was taken into account, thus the possible effect of the channel entrance over the results could be ruled out. A conservative criterion was used such that the ratio between the length and the hydraulic diameter of the channel is larger than 60 (i.e. $L/D_h > 60$) [12].

A total number of 1824 and 525 experimental points are then available for the analysis of SE3 and SE4, respectively.

4.1. Selected correlations from the literature

The experimental values of the Nusselt number were compared with the ones estimated with existing correlations that are valid for circular or rectangular channels.

The Dittus-Boelter correlation [5] was derived for circular pipes and its application may be extended for other geometries with the use of the hydraulic diameter. This equation, as introduced by McAdams [13], can be written as:

$$Nu = 0.023Re^{0.8}Pr^{0.4} \quad (18)$$

The validity range is such that: $Re > 10^4$ and $0.7 < Pr < 160$. In the literature, several works can be found where the Dittus-Boelter correlation was suggested for and applied to rectangular channels (e.g. [14] and [15]). Nevertheless, the comparison with the SULTAN-JHR experimental data shows that the Dittus-Boelter correlation significantly under-estimates the heat transfer coefficient, especially at high Reynolds number (see Fig. 4 and 10). It should be pointed out that a minor fraction of experimental points has Reynolds number below 10^4 .

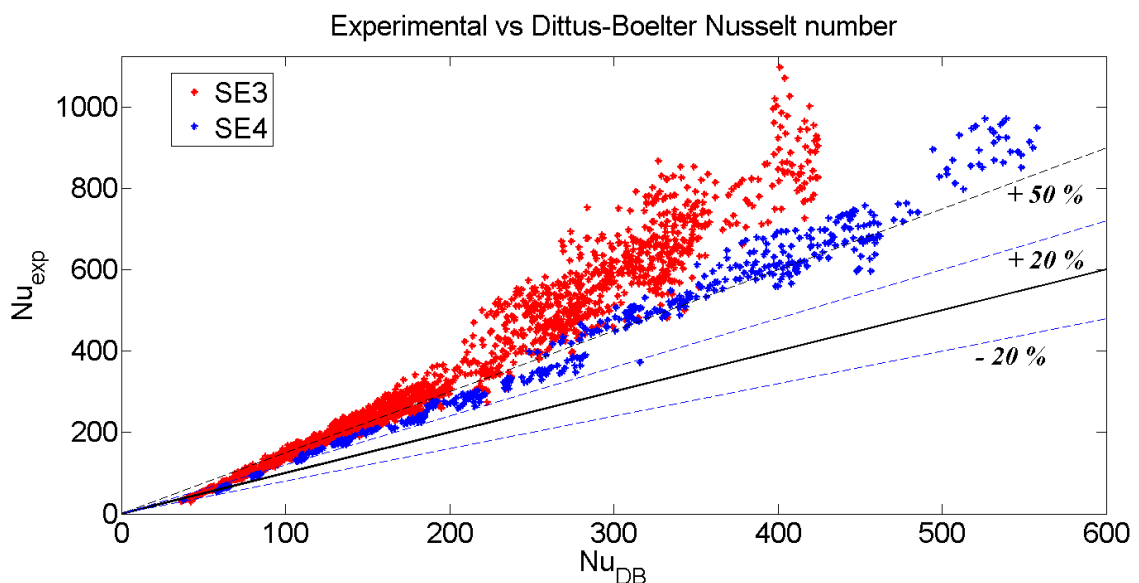


Fig. 4 Comparison of the experimental data with Dittus-Boelter correlation.

Similar results are obtained when other standard correlations are employed, e.g. Sieder-Tate [6], Gnielinski [7] and Popov-Petukhov correlation [8] with Siman-Tov correction for rectangular geometry [16], which reads:

$$Nu = \frac{\frac{f_P Re Pr \left(\frac{\mu}{\mu_w}\right)^{0.11}}{8}}{(1+3.4f_P) + \left[11.7 + \frac{1.8}{Pr^{\frac{1}{3}}}\right] \left(\frac{f_P}{8}\right)^{0.5} \left(Pr^{\frac{2}{3}} - 1\right)} \quad (19)$$

$$f_P = \frac{\left[1.0875 - 0.1125 \left(\frac{e_{ch}}{l_{ch}}\right)\right]}{(1.82 \log_{10} Re - 1.64)^2} \quad (20)$$

The validity range for Popov-Petukhov correlation is: $10^4 < Re < 5 \cdot 10^6$ and $0.5 < Pr < 2000$. In particular, the Popov-Petukhov correlation still underestimates the experimental points, but provides some improvement in the predictions, as shown in Fig. 5.

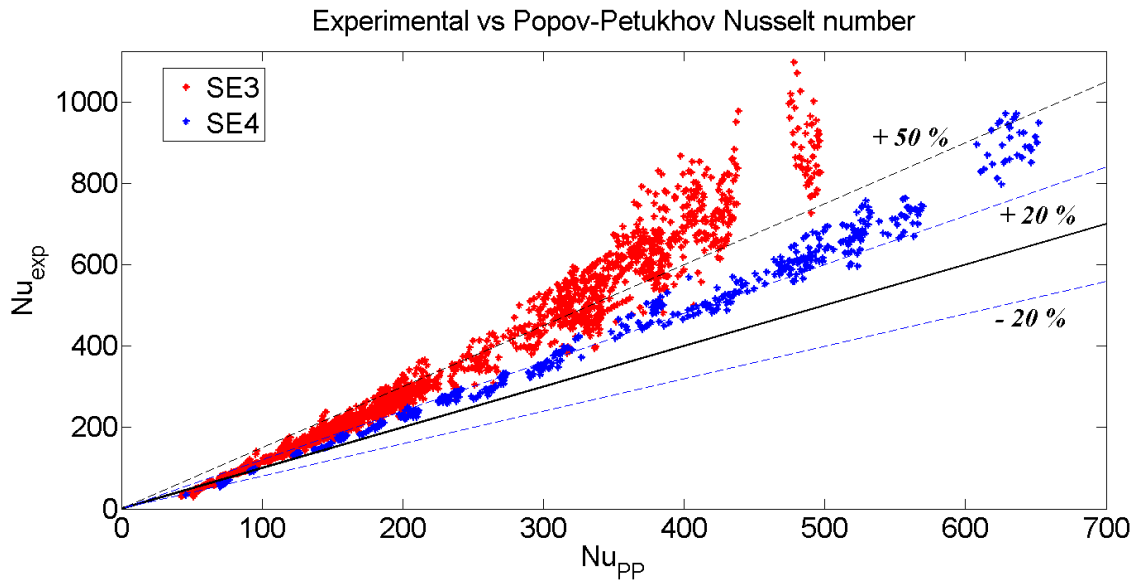


Fig. 5 Comparison of the experimental data with Popov-Petukhov correlation.

The correlations that were specifically developed for narrow rectangular channels by Liang et al. [9] and by Ma et al. [10] were also tested.

The correlation of Liang et al. is based on experimental data for a rectangular test section of 1.8 x 50 x 800 mm, with a pressure range between 5 and 50 bar; relatively low heat fluxes between 4 and 50 kW/m² and Reynolds number between 2300 and 6150. It reads:

$$Nu = 0.00666 Re^{0.933} Pr^{0.4} \quad (21)$$

The correlation still underestimates the SULTAN-JHR experimental data, even though it performs relatively better for test section SE4 (see Fig. 6).

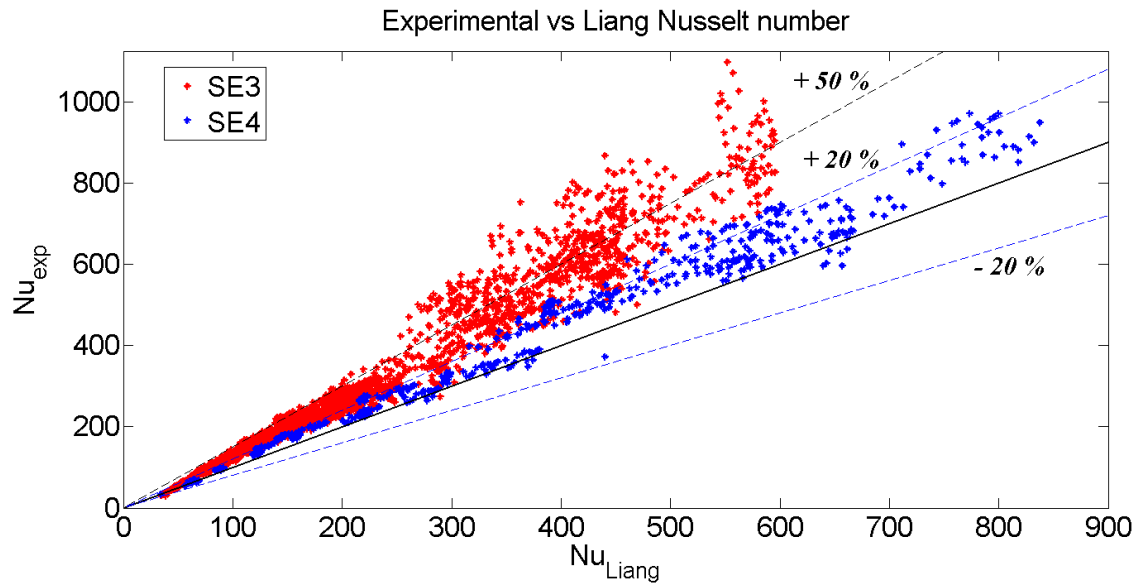


Fig. 6 Comparison of the experimental data with Liang correlation.

The correlation of Ma et al. was developed from experiments performed in a rectangular test section of 2 x 40 x 1092 mm, at atmospheric pressure, with heat fluxes between 14 and 214 kW/m² and Reynolds numbers between 4000 and 13000. It reads:

$$Nu = 0.00354 Re^{1.0} Pr^{0.4} \quad (22)$$

The geometry of the test section is close to the one in SE4, although the heat fluxes are lower and the range of variation of the Reynolds number is smaller in comparison with the SULTAN-JHR tests. The results for this correlation are shown in Fig. 7. A good agreement was reached with the experimental points from test section SE4 (i.e. the test section similar to the one used by Ma et al.): the error is within $\pm 20\%$ and no significant bias could be observed. On the other hand, a large underestimation is again obtained for SE3.

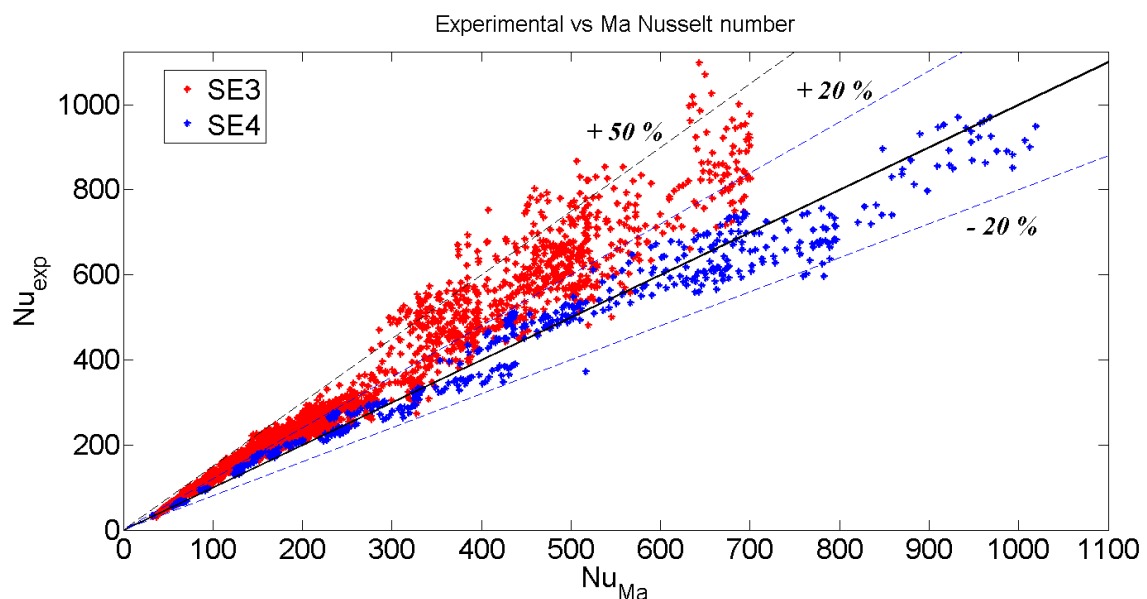


Fig. 7 Comparison of the experimental data with Ma correlation.

4.2. Development of a correlation based on the SULTAN-JHR data

To the authors' knowledge, no correlation is available in the open literature for highly turbulent flow in narrow rectangular channels and with relatively high heat fluxes. Therefore, a new correlation was developed from a best-fitting of the SULTAN-JHR data. The Dittus-Boelter form of the correlation was assumed, i.e.:

$$Nu = aRe^{b_{Re}}Pr^{c_{Pr}} \quad (23)$$

The coefficients of the correlation were simultaneously optimized using a multiple linear regression approach. This procedure was applied to the data for SE3 and SE4 separately, since an effect of the channel geometry was observed as discussed in the next subsection.

The correlation for the test section SE4 reads:

$$Nu = 0.0044Re^{0.967}Pr^{0.557} \quad (24)$$

From the statistical analysis of this best-fitting, the coefficient of determination R^2 is equal to 0.9947 and the standard deviation is 0.0528.

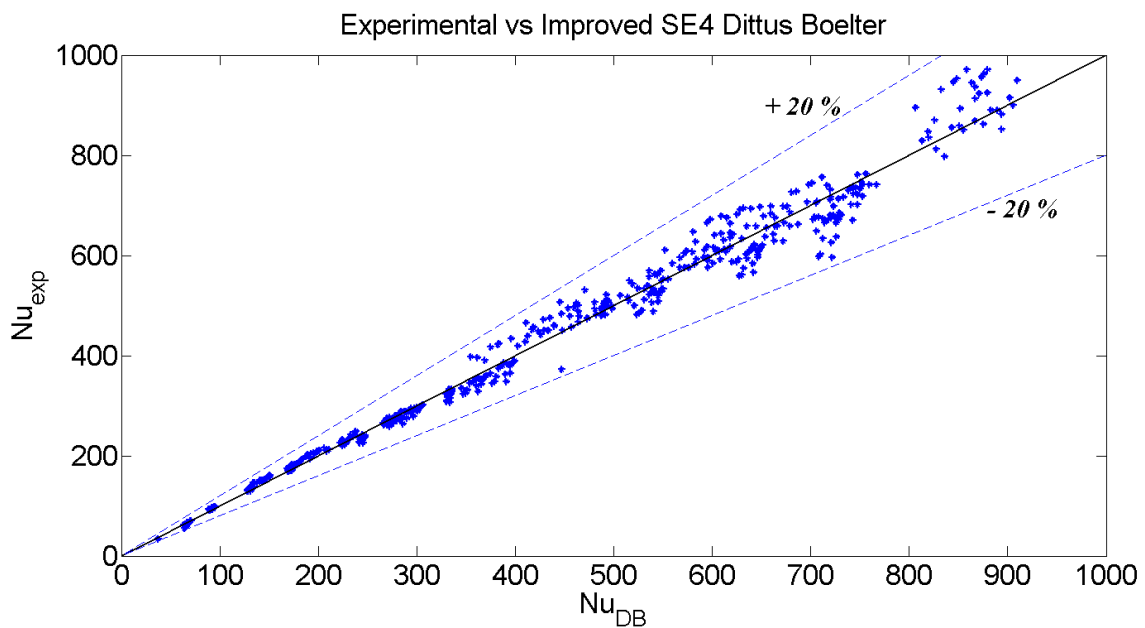


Fig. 8 Fitting of the new heat transfer correlation over the SE4 experimental data.

As regards test section SE3, the correlation that fits best the experimental points is:

$$Nu = 0.0021Re^{1.053}Pr^{0.606} \quad (25)$$

with a coefficient of determination equal to 0.9845 and a standard deviation of 0.0913. Equations (24) and (25) can then lead to predictions whose error is in a range of $\pm 20\%$ with respect to the experiments, as shown in Fig. 8 and 9.

A Sieder-Tate form of the correlation was also tried, but no significant advantage was found. Thus the Dittus-Boelter form is suggested for simplicity.

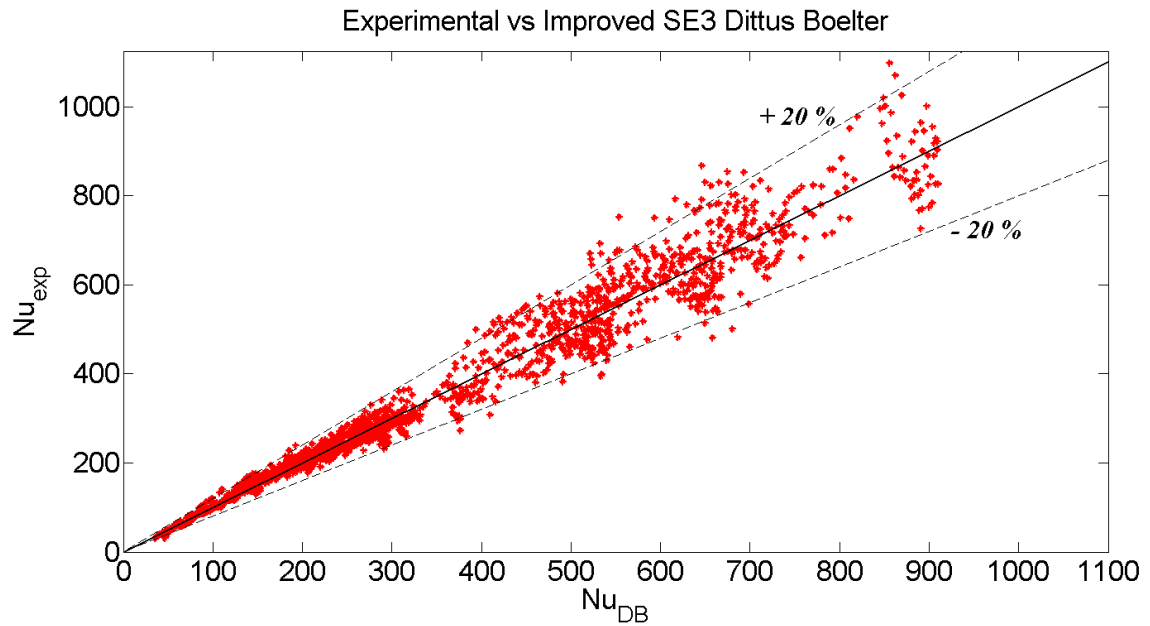


Fig. 9 Fitting of the new heat transfer correlation over the SE3 experimental data.

4.3. Influence of the Channel Geometry

The comparison between the two test sections SE3 and SE4 points out that the channel geometry can affect the heat transfer coefficient, as displayed in Fig. 10. In fact, the heat transfer coefficient becomes higher in SE3 which has a smaller channel gap. Therefore an enhancement of the heat transfer occurs with the decrease of the gap.

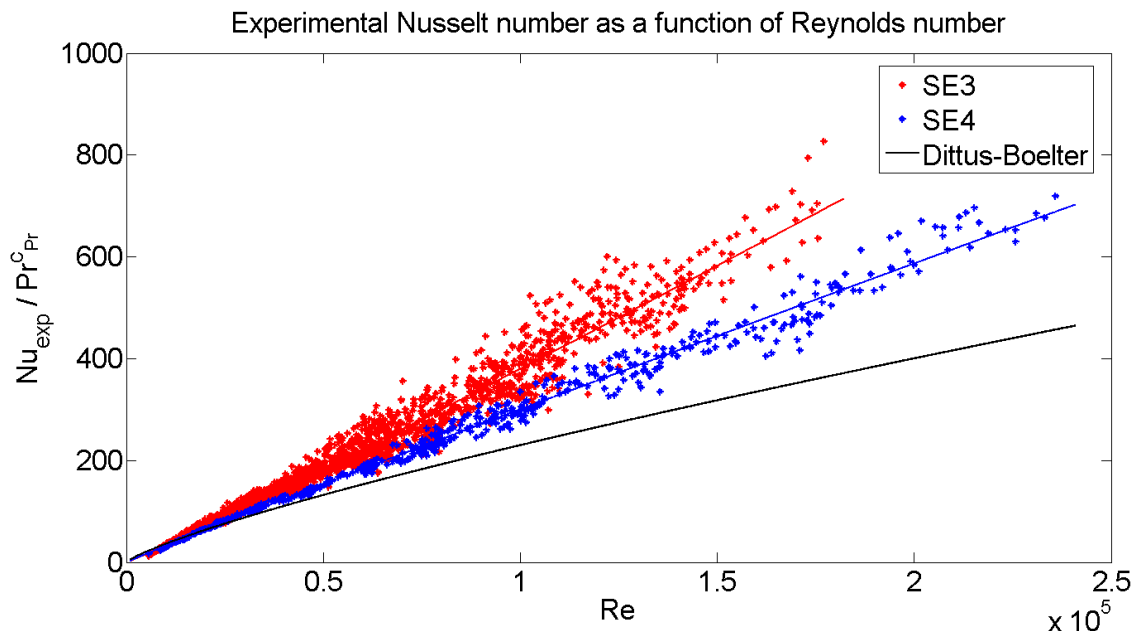


Fig. 10 Nusselt number as a function of the Reynolds number.

In the figure, the experimental Nusselt number is divided by the Prandtl number to the power of the corresponding c_{Pr} coefficient (see Table 4) in order to visualize the effect of the Reynolds number on the heat transfer and compare the two sets of data.

It is then showed that the discrepancies between the two test sections grow bigger with the increase of the Reynolds number. This is confirmed by the examination of Eqn. (24) and (25). In fact, the change of the channel gap seems to impact mainly the proportionality coefficient a , as well as the power coefficient b_{Re} associated to the Reynolds number, rather than the contribution related to the Prandtl number (see also Table 4).

In order to deal with the influence of the geometry in rectangular channels, the aspect ratio is usually employed (e.g. in [4], [14] and [16]):

$$AR = \frac{e_{ch}}{l_{ch}} \quad (26)$$

This parameter is dimensionless and takes into account both the influence of the channel gap and width. For instance, the correlation of Popov-Petukhov includes the aspect ratio (see Eqn. (19) and (20)), even if it gives unsatisfactory results when applied to the SULTAN-JHR experiments. Thus, future work will be addressed to derive a single correlation for SE3 and SE4 with the eventual introduction of the aspect ratio.

Table 4. Influence of the gap and of the aspect ratio on the heat transfer correlation

	gap[mm]	AR	a	b_{Re}	c_{Pr}
SE3	1.51	0.0287	0.0021	1.053	0.606
SE4	2.16	0.0417	0.0044	0.967	0.557
Dittus-Boelter	-	1	0.023	0.8	0.4

5. CONCLUSIONS

The SULTAN-JHR experimental database was used for the evaluation of selected heat transfer correlations for single-phase forced convection in narrow vertical rectangular channels. The data are related to two test sections with different gap sizes of 1.51 and 2.16 mm. The experimental conditions range between: 2 and 9 bar for the pressure; 0.5 and 18 m/s for the coolant velocity and 0.5 and 7.5 MW/m² for the heat flux. Therefore, relatively high values of Reynolds number and heat flux are included.

The analysis shows that the standard correlations of Dittus-Boelter, Sieder-Tate, Gnielinski and Popov-Petukhov significantly under-estimate the heat transfer coefficient when applied to the SULTAN-JHR experiments. Moreover, the correlations of Liang et al. and Ma et al., which were specifically designed for narrow rectangular channels, have a limited range of validity and could not predict accurately the experimental data.

Therefore, a correlation for each of the two test sections was developed with a best-fitting approach. These correlations have the following range of validity: the Reynolds number is between 5.5×10^3 and 2.4×10^5 and the Prandtl number is between 1.5 and 6.

It is also pointed out that the geometry of the narrow rectangular channel plays an important role in the determination of the heat transfer coefficient. In particular, a reduction of the channel gap leads to an improvement of the heat transfer coefficient. Further research will be conducted in order to investigate the possibility of having an unique heat transfer correlation that can be valid for both test sections.

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