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A complex but accurate correlation for predicting critical heat flux in a round tube for low and medium pressures under circumferentially non-uniform heating conditions

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Abstract

This paper presents an empirical correlation for predicting the critical heat flux (CHF) of vertical, upward, steam–water flows in round tubes for low and medium pressures under circumferentially non-uniform heating conditions. This correlation is based on experiments carried out with test sections having an inner diameter of 22 mm and heated lengths ranging from 1.8 to 3.5 m. The ratios between the maximum and the minimum heat fluxes were 1.0, 4.7 and 8.3. The experiments were carried out for outlet pressures and mass fluxes ranging from 10 to 40 bar and 300 to 1600 kg m^{-2} s^{-1}, respectively. The root mean square error of predicted CHF values by using the proposed correlation and applying the heat balance method is 3.7%.

Keywords: Critical heat flux; Non-uniformly heated tube; Boiling flow; Empirical correlation

1. Introduction

The boiling heat transfer plays an important role in large industrial heat transfer units such as nuclear and fossil fuel power plants, and in small-scale heat transfer devices such as heat pipes and microchannels used in electronic chip cooling systems. The use of a boiling process is, however, limited by flow conditions that can bring about a sudden deterioration of the heat transfer caused by the replacement of the liquid by the vapour on the heated surface. Such a situation, identified as critical heat flux (CHF), in surface-heat-flux-controlled systems can provoke the physical burnout of the materials of a heated element. It is obvious that the understanding of the CHF phenomenon and, in particular, the accurate prediction of flow conditions that can bring about CHF, are important for the safe and efficient operation of many heat transfer units including nuclear fission power reactors, fossil fuel boilers, fusion reactors, electronic chips, etc.

The crucial and practical importance of the CHF and the extreme physical complexity of the critical phenomenon justify the tremendous number of CHF studies that have been carried out during the last 50 years. Most of these studies were carried out under circumferentially uniform heating conditions, while, in reality, a uniform heat flux distribution of heated elements is quite rare.[1] In fossil-fired boilers, nuclear fusion reactors, compact heat exchangers and electronic chip cooling devices, the heat is transferred across one-side of the steam generator or cooling channels. In nuclear power reactors, the circumferential heat flux non-uniformity is lower, but it always occurs due to the

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[1] It must be pointed out that even in laboratory studied CHF in round tubes that are presumed to be uniformly heated, there exists a certain non-uniform heat flux distribution due to the inherent eccentricity of cold-drawn tubes and the use of direct electrical heating [1].

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Nomenclature

a	coefficient, Eq. (8)
b	coefficient, Eq. (8)
$C_1, C_2, \ldots C_7$	variable coefficients, Eq. (9)
D	diameter, m
E_m	exponential function, Eq. (19)
E_q	exponential function, Eq. (18)
G	mass flux, kg m^{-2} s^{-1}
h	enthalpy, kJ kg^{-1}
h_{lv}	latent heat of evaporation, kJ kg^{-1}
k	number of empirical coefficients (non-uniform heating)
L	length, m
m	number of empirical coefficients (uniform heating)
P	pressure, bar
P_r	reduced pressure, P/P_{cr}
R_m, R_q	reduced heat flux ratios, Eqs. (20) and (21)
q''	heat flux, W m^{-2}
S	root mean square relative prediction error
W	mass flow rate, kg s^{-1}
We	Weber number
x	thermodynamic quality
z	distance, m

Greek symbols

ρ	density, kg m^{-3}
σ	surface tension, N m^{-1}
Φ	characteristic function, Eq. (5)
Ψ	exponential function, Eq. (8)

Subscripts

cr	critical
dan	dispersed-annular flow
h	heated
in	inlet
l	saturated liquid
max	maximum
min	minimum
nun	non-uniform
osh	one-side heating
out	outlet
r	reduced (pressure)
un	uniform
v	saturated vapour

Superscript

i	intermediate

non-uniform distribution of the neutron flux over the core radius, the non-homogeneous concentration of fissile material in the fuel elements, the non-uniform contact resistance between the fuel rod and its cladding, etc.

With the exception of recent CHF research works carried out by using small inner diameter short channels for simulating plasma-facing components (e.g. [2]), or close to full scale nuclear reactor core experiments (e.g. [3]), only few papers related to CHF in round tubes under circumferentially non-uniform heating conditions can be found in the open literature [4–19]. In most of these papers, the scope is practically limited to a qualitative comparison of CHF values obtained in uniformly and non-uniformly heated tubes and to far from convincing discussions about which heat flux (i.e., maximum local or average) is better suitable for representing the critical phenomenon in non-uniformly heated channels. The evident lack of large-scale systematic investigations forces some authors to employ a disputable comparison of measured CHF values with those calculated from empirical correlations sometimes having uncertainties higher than 100%, as well as with data obtained from experiments conducted by using two-phase mixtures at the inlet of the test section. In certain cases such a comparison results in contradictory conclusions. For example, from the analysis of the studies carried out to date, it may be inferred that, in general, the peak heat flux q''_{max} (local maximum) under CHF conditions for non-uniformly heated tubes is greater than the CHF observed in uniformly heated tubes, with all other flow parameters being identical. On the contrary, the corresponding mean heat flux q''_{mean} over the tube perimeter is lower. The relationship between uniform CHF and non-uniform CHF nevertheless cannot be presented as a single heat flux ratio. With decreasing steam quality, the peak heat flux seems to approach the CHF values obtained under uniform heating conditions. On the other hand, with increasing the steam quality, q''_{mean} reaches the value measured in a uniformly heated tube. However, these observations are rather "snapshots" of the most frequently presented trends of the critical phenomenon in non-uniformly heated channels. According to Becker et al. [14], for the same exit quality ranging from 0 to 0.4, the dryout peak heat fluxes in tubes heated from one-side correspond well enough to the CHF values obtained in uniformly heated tubes. In turn, Belyakov [18] has reported that for a pressure of 150 bar the critical steam qualities $x_{cr_{nun}}$ in a non-uniformly heated tube having an inner diameter of 20 mm could be smaller as well as higher than $x_{cr_{un}}$. For a mass flux of 1500 kg m^{-2} s^{-1}, $x_{cr_{nun}}$ were approximately twice of those obtained in a uniformly heated one. This last observation indicates that under critical conditions q''_{mean} is roughly two times greater than the CHF for the uniformly heated tubes. A similar conclusion ($q''_{mean_{nun}} > q''_{mean_{un}}$) was also made by Umekawa et al. [19] for a tube with the same inner diameter of 20 mm, but for a much lower flow pressure ($P = 4$ bar) and an extremely low mass flux ($G < 20$ kg m^{-2} s^{-1}). In principle, such a behaviour may be explained by a higher droplet deposition rate, as compared to uniform heating, taking place on the low heat flux wall side in conjunction with a high circumferential redistribution (spreading) film flow rate. It is, however, doubtful that the same mechanism could take place simultaneously under radically different conditions.

In addition, some aspects associated with circumferential heat flux distributions (i.e., profile of the heat flux distribution) were scarcely studied, while they seem to be highly important. In fact, the occurrence of the critical phenomenon must, in general, depend on the following three factors or input variables: the profile of the heat flux distribution, the ratio between the peak (q''_{max}) and the average (q''_{mean}) heat fluxes and the ratio between the peak (q''_{max}) and the minimum (q''_{min}) heat fluxes. It is apparent that the relationship between these variables is rather complex. For example, for a similar profile of the heat flux distribution and for the same ratio q''_{max}/q''_{mean}, the relation between q''_{max} and q''_{min} can be quite different (Fig. 1). Thus, experiments carried out by fixing only one parameter or by simultaneously changing two parameters cannot provide a whole picture of the behaviour of the system under non-uniform heating conditions.

Regarding the non-uniform CHF predictive tools, only few existent methods can be mentioned. The empirical correlations proposed by Miropolskii and Mostinskii [4] as well as by Styrikovich and Mostinskii [5] were developed from experiments carried out by using very short channels ($L/D \approx 25$) and, in most of the cases by using steam–water mixtures at the inlet of the test sections. Chojnowski and Wilson [9] examined the validity of the Styrikovich–Mostinskii correlation. It was found that their correlation produces "pessimistic results". For predicting dryout in a tube with circumferential variation of heat flux, Butterworth [8] proposed a film flow model by using, according to the author himself, "a number of sweeping assumptions". This model was developed from the two-dimensional continuity equation for a liquid film with a secondary circumferential flow on the tube perimeter. Butterworth postulated that, for a given tube diameter, the value of the circumferential flow rate is simply proportional to its angular gradient. The value of the proportionality coefficient, called a spreading coefficient, based upon the data of Miropolskii and Mostinskii [4], and Alekseev et al. [6], was assumed by the author to be constant and equal to 0.9 mm. However, this assumption is not confirmed by the analysis carried out later [9,12,19]. These latter investigations show that the spreading coefficient is a clear function of the steam quality and the mass flux, and its value can vary from 0.15 mm to infinity (the last value corresponds to the case where there is an equality between the non-uniform average and uniform heat fluxes under dryout conditions). Unfortunately, no fundamental methods for determining the spreading coefficient were developed. Therefore, Butterworth's model still remains an abstract framework.

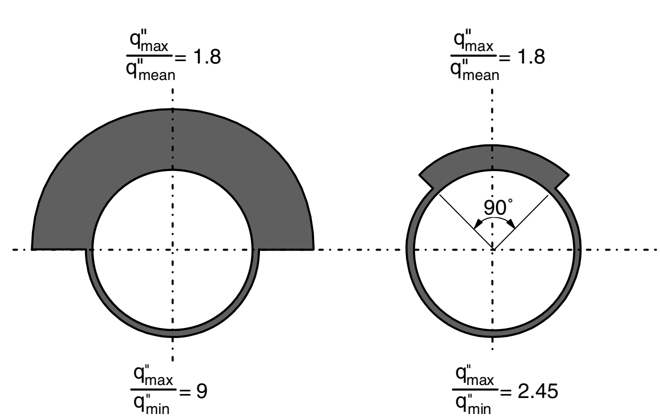

Fig. 1. Interrelationship between the non-uniform heat flux distribution parameters.

A different approach, based on the boundary or limiting quality Doroshchuk's concept[2] (e.g. [20]), is given in [11,13,15,16,18]. As stated by these authors, within a range of parameters close to those adopted for designing power boilers in Russia (tubes with inner diameter of 20–50 mm; pressures from 120 to 200 bar; mass fluxes from 500 to 1500 kg m^{-2} s^{-1}), the heat transfer deterioration occurs just when the boundary quality conditions are reached. However, an in-depth analysis of these works shows that this statement is really sustained only by the authors of references [15,16] (these studies were partly carried out by using steam–water mixtures having inlet steam qualities of up to 0.2). For a one-side heated tube having an inner diameter of 48 mm, Galetskii et al. [16] reported that the boundary quality is approximately equal to 0.23, independently of the mass flux and flow pressure. In contrast, Belyakov [18] indicates that for a tube having an inner diameter of 40 mm, the boundary quality decreases from 0.41 to 0.22 with increasing the mass flux from 500 to 1000 kg m^{-2} s^{-1}. Moreover, the results presented by Belyakov and his colleagues [11,13,18] show that the critical quality depends, in a certain manner, on the heat flux.

Even though CHF under circumferentially non-uniform heating conditions better reflects the behaviour of boiling systems used by the power industry, this brief literature survey clearly shows that it has been barely studied. Therefore, the need for additional research on this subject is of prime relevance. The present study is focused towards investigating the critical phenomenon in small-capacity, low-pressure boilers. According to a recently developed concept [21,22], the use of such boilers could permit to achieve a substantial reduction of greenhouse gas emissions. Furthermore, operating large-scale boilers during off-peak hours results in producing large amounts of waste energy. Their replacement by multiple compact boilers that

[2] Following this concept, the dryout in a developed dispersed-annular flow occurs at a constant thermodynamic quality; its value does not depend on the heat flux. In the q''_{cr}, x_{cr} plane, this "boiling crisis of the second kind" represents a boundary between the "CHF of the first kind" (mainly the departure from nucleate boiling in subcooled and low quality flows) and the "CHF of the second kind with droplet deposition" for which the CHF value depends strongly on the critical quality.

can be shut down during off-peak hours can provide considerable energy consumption savings.

2. Experimental setup and procedures

The experiments were carried out for upward, vertical, inlet subcooled water flows by using a steam–water thermohydraulic loop previously described in [23]. The same flow conditions were used for conducting experiments with uniformly heated and non-uniformly heated test sections having the same inside diameter (ID = 22 mm). In all the cases studied, the axial heat flux distribution was uniform. The test sections were heated by Joule effect with a controlled direct current passing through the tube wall. The circumferentially non-uniform heat distribution was obtained by designing and machining appropriate tube wall thickness profiles. The experimental conditions were: outlet pressures 10–40 bar; mass fluxes 300–1600 kg m^{-2} s^{-1}; inlet subcoolings 5–40 °C and heated lengths 1.8, 2.5, 3.55 m. The power applied to the test section was simultaneously determined by two different methods. One method consists of an analog multiplier that gives the power as the product of the voltage drop multiplied by the electric current measured with a 50 mV–5000 A, 1% calibrated shunt (Bach–Simpson, Model H5000-50). The second method consists of a numerical sampling and multiplication of the same parameters carried out by the computer through a data acquisition system. In this case the electric current passing through the test section was measured with a high accuracy LEM (Model LA-5000T) unit. Before starting the experiments heat balance tests were carried out; the maximum observed uncertainty in the applied power was rarely greater than ±2% of the collected values. In addition, the inlet and outlet flow temperatures were measured with thermocouples calibrated to ±1 °C of the readings; the water flow rate was measured by using "Flow Technology" turbine flow meters with an accuracy better than ±1% of the reading. The internal diameters of the tubes were determined with a precision of ±0.3%; the inaccuracy in the heated length, even if the thermal expansion is taken into account, cannot provoke an error greater than ±0.2%.

The CHF was approached by slowly increasing the power applied to the test section under steady mass flux, inlet temperature and outlet pressure conditions. The boiling crisis was identified by either a fast wall temperature increase or by the onset of wall temperature fluctuations. It must be pointed out that for all the cases studied, the CHF was always observed at the end of the heated length. Furthermore, for the experiments carried out under circumferentially non-uniform heating conditions, CHF was always detected on the high heat flux side of the tubes' wall.

Three different test sections were used in this study; they were manufactured from stainless steel SS 316/316L (X5CrNiMo17-12-2/X2CrNiMo17-12-2) cold-drawn, annealed, solution treated, seamless tubes. One of these test sections was made of a 22 mm ID and 1.65 mm wall thickness (1″ × 0.065″) tube; it was intended for studying CHF under uniform heating conditions. The others two were machined from thick wall tubes in such a way that the heat

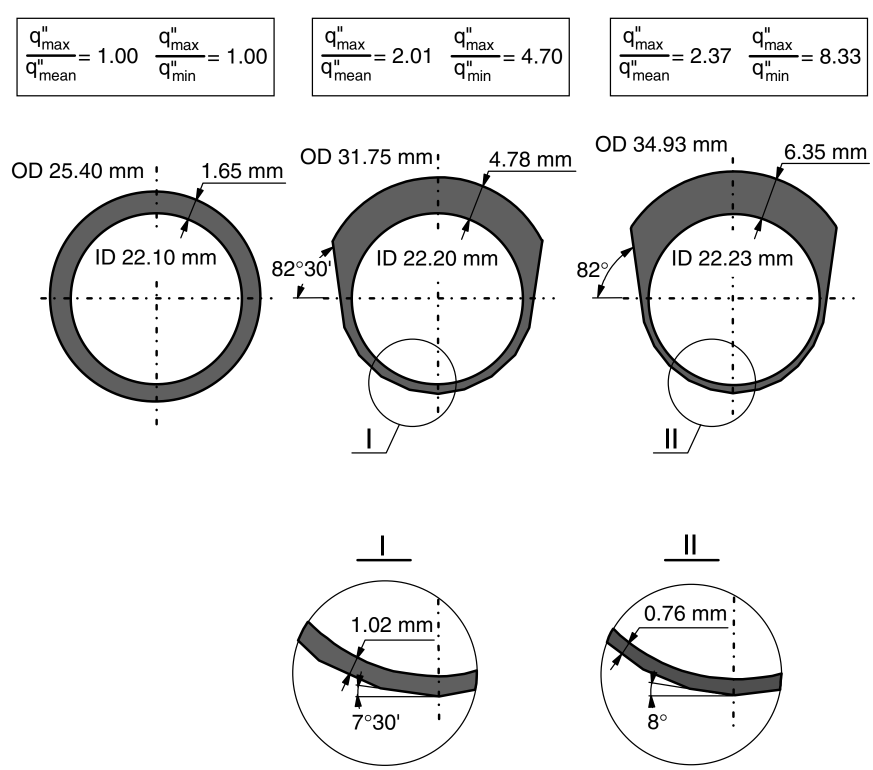

Fig. 2. Cross-sectional view, dimensions and heat flux distribution parameters of the test sections used for carrying out the experiments.

flux distribution satisfied the following criteria: (a) it should be as close as possible to a typical heat flux distribution encountered in the furnaces of water-wall boilers; (b) in order to determine the effect of q''_{max}/q''_{mean} on the CHF, this ratio must slightly differ from one test section to the other and (c) the values of q''_{max}/q''_{min} should be quite different and high enough to better approach the one-side heating condition.

The cross-sectional view as well as the geometrical dimensions of the test sections are shown in Fig. 2. It must be pointed out that due to the relatively low heat conduction of the stainless steel, combined with the extremely high forced convection heat transfer that characterizes boiling flows, the heat flux distribution corresponds well enough to the wall thickness profile of the tube. The wall temperatures were measured with 0.5 mm OD type-K ungrounded thermocouples spot-welded on the external surface of the test sections. The instrumented test sections are shown in Fig. 3. The heated length was varied by the displacement of the lower electrical clamp having a geometry that matched the shape of the test sections. Both, the lower and the upper electrical clamps were manufactured from copper bars electroplated with nickel in order to avoid oxidation. In addition, thin, easily compressed, silver sheets were placed in between the clamps and the test section surface. Note that for $q''_{max}/q''_{min} = 8.3$ and for $L_h \leqslant 2.5$ m, the external wall temperature on the high heat flux side of the tube became very high even before CHF occurred. In order to protect the test sections, the applied power was shut down when the wall temperature reached 800 °C. The test sections were covered with a 40 mm thick layer of ceramic fiber insulation (Calsimag CSM Blanket). Even in the worst scenario, i.e., critical power of 350 kW, the heat losses were estimated to be less than 1.5 kW. These heat

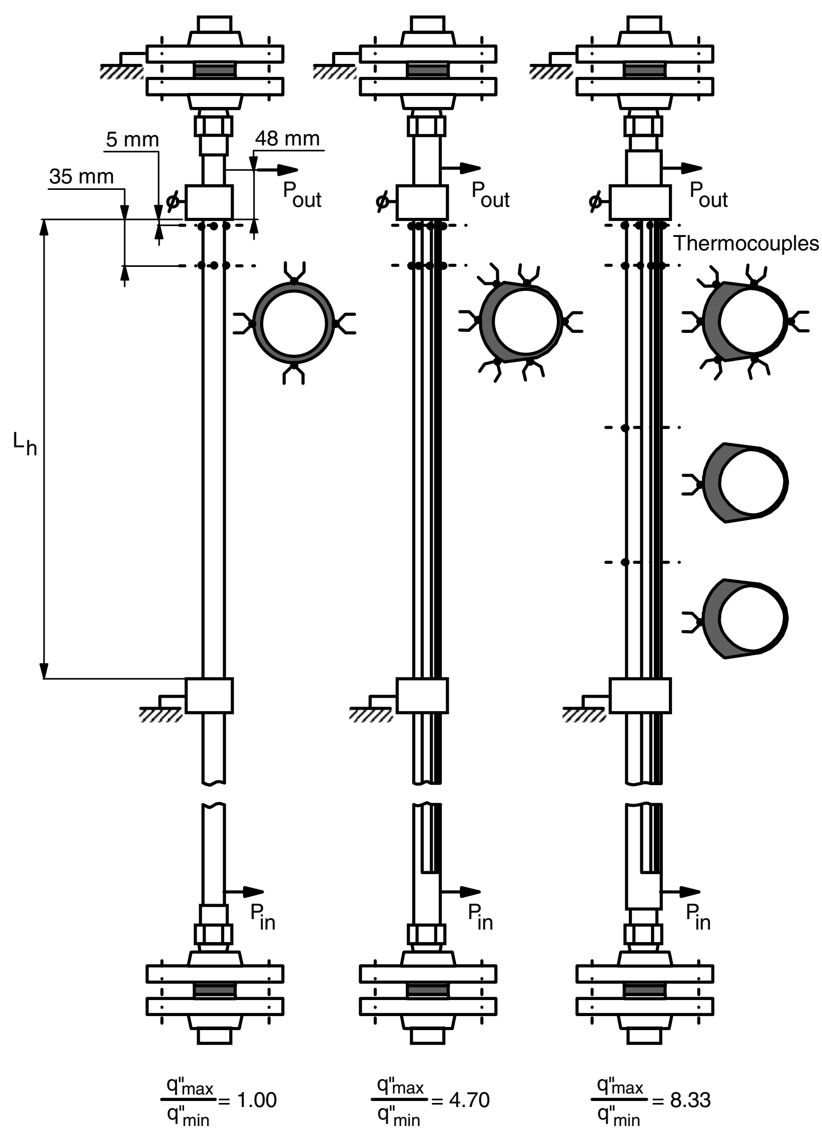

Fig. 3. Instrumented test sections.

losses were lower than the uncertainties of the CHF detection; thus, the heat losses were considered as negligible.

2.1. Particular experimental observed features

For the experiments carried out with circumferentially non-uniform heat flux distributions, some particular features were observed. A discussion about the effects that these features may have on the CHF is presented in the following sections.

2.1.1. Bowing of tubes under circumferentially non-uniform heating conditions

The test sections used for carrying out experiments under non-uniform heating conditions can only be nominally considered as vertical. Even though during endmilling, they were carefully clamped to a special manufactured jig and the tubes were cut in multiple passes, a slight but visible bowing due to residual mechanical stresses of the material resulted at the end of the manufacturing process. Fig. 4a shows an exaggerated sketch of the residual stress effect on the test section. This bowing was easily corrected[3] by applying a force of about 600 N to the test section before starting the experiments (Fig. 4b). However, due to a non-uniform tube dilatation during the heating, an excessive bowing, with a chordal deviation of up to 0.15 m, was often observed (Fig. 4c). A substantial reduction in the applied power led the straightening of the tube to take place. It must also be pointed out that after removing the non-uniformly heated test sections from the loop, they showed a very pronounced bowing in the opposite direction to that observed when heat was applied to them (Fig. 4d). In spite of this particular feature, any other tube deformation was not detected after the experiments. This uncontrolled bowing behaviour can be considered as a supplementary factor that may affect the values of CHF obtained in non-uniformly heated channels.

2.1.2. Mass flow rate pseudo-pulsations

For $q''_{max}/q''_{min} = 4.7$ and for $L_h = 1.8$ m, and for $q''_{max}/q''_{min} = 8.3$ and for $L_h \leqslant 2.5$ m, a phenomenon never previously observed, was revealed. As soon as the applied power reached a value close enough to the critical one, the recorded mass flow rate started to pulsate while all other flow parameters, the applied power and the wall temperatures remained stable. The amplitude of these quite sharp pulsations could reach up to 20–30% of the nominal value. Besides the fact that the occurrence of such pulsations could alter the CHF, its asymmetrical character might also have lead to an average mass flow rate reduction due to the feedback of the flow rate control system.

It was established that these pulsations were generated by the appearance of electromagnetic pulses which were captured by the turbine flow meter pick-off coil. The installation of an appropriate electromagnetic shield permitted to completely eliminate these mass flow rate pseudo-pulsations.[4] It must be pointed out, however, that the process that caused these pulsations was real and in some way or another could affect the CHF. In order to prevent the occurrence of this process to happen, a number of precautions were seriously taken. Since at very high wall surface temperature, intermittent loss of electrical contact between the clamps and the test section could occur; the experiments with $q''_{max}/q''_{min} = 8.3$ and for $L_h \leqslant 2.5$ m were carried out by using brass-soldered electrical clamps. In addition, different electrical and electrochemical processes that could take place between the grounded loop pipeline and the high voltage side of the test section, supplementary precautions were considered. In order to reduce the strength of the electric field in this test section region, a special insulator insert fabricated from Ketron® PEEK material was manufactured and employed during the experiments; this device is shown in Fig. 5.

It must be pointed out that none of these modifications produced an acceptable effect; as a matter of fact without the electromagnetic shield, the flow pulsations were still present just before the CHF occurrence takes place. It is possible that this observed phenomenon is triggered by very high electric currents (of up to 5.5 kA) during an aus-

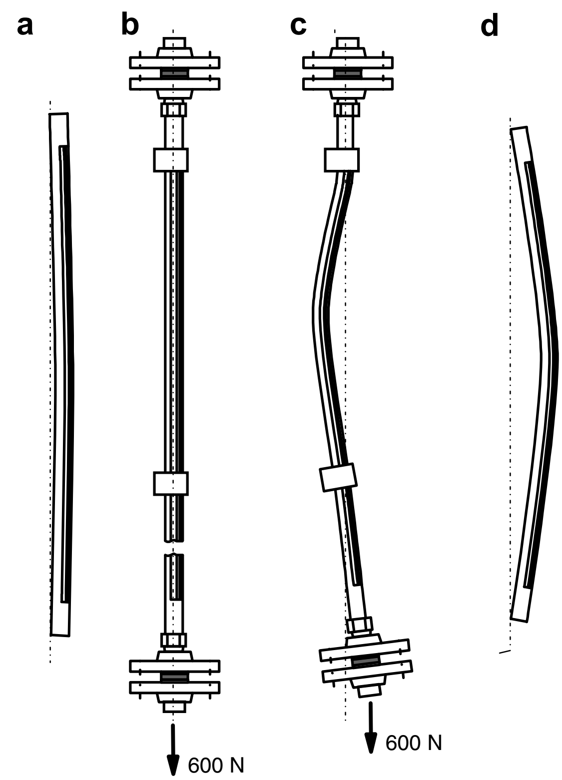

Fig. 4. Bowing observed on non-uniformly heated tubes.

[3] In order not to damage a thin tube wall, no supporting guides along the test section were used.

[4] The experiments initially carried out under "pulsating" flow conditions were repeated.

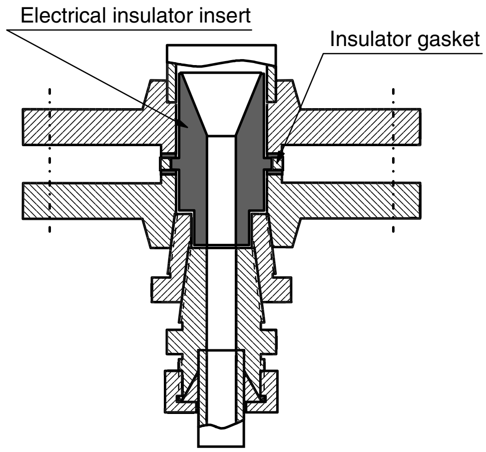

Fig. 5. Test section gasket and electrical insulation device.

tenite–ferrite transformation, which normally takes place for 300 series austenitic stainless steels at a temperature of about 700 °C. Such a hypothesis, however, could neither be proven, nor refuted in the framework of the present study.

3. Experimental results

A total of 343 CHF data points were collected under uniform heating conditions and 598 data points were collected under non-uniform heating conditions. Hereafter, only general observed trends will be discussed.

The CHF is not a separate physical phenomenon but rather the result of a large number of processes which take place in boiling fluids. While only some of these processes have a local character, others depend on all the thermohydraulic parameters that vary along the heated channel. For instance, consider the dryout type CHF where the depletion of the liquid film is related to: the vaporization, the circumferential film spreading and to the rate of droplet entrainment and deposition. These processes are developed and modified within the entire length over which the dispersed-annular flow pattern takes place. In turn, dryout is also determined by the liquid film flow rate at the position, or rather in the local flow area where the dispersed-annular flow is formed, which does not corresponds to the end of the heated length where CHF normally occurs. In addition, CHF can also be caused by a local dynamic rupture of a thin liquid film at the end of the heated region. Moreover, the parameters which are usually employed for describing or predicting CHF have a dual interpretation. For example, the steam quality (frequently used for correlating CHF data) at a given location can be considered either as a local steam mass concentration or as the relative mass of steam generated downstream of the same location.

The former discussion shows that, even though for axially uniform heat fluxes CHF always occurs at the end of the heated length, on the high heat flux wall side during circumferentially non-uniform heating of the channel, CHF cannot be identified as being a local or a global phenomenon. Thus, it is almost impossible to decide a priori if the maximum local or the average heat flux is better suited for representing the critical phenomenon in non-uniformly heated channels. However, for the particular purpose of development of an empirical correlation for predicting the deterioration of heat transfer conditions, this dilemma does not exist at all. For a given heat flux distribution characteristics, the ratio between the peak and the average heat fluxes is expressed by a single value. Hence, the discrepancy between two corresponding correlations can only consist of two coefficients that will differ by the same value. Following this approach and for the sake of simplicity, only the average critical heat flux will be considered in this paper. The data are analyzed by using two different CHF representations: (i) as a function of the exit thermodynamic quality x and (ii) as a function of the length over which the dispersed-annular flow takes place L_{dan}. The first representation constitutes the basis of a local hypothesis, and the most used method for predicting CHF in uniformly heated channels. The second one [24,25] is based on the relationship between the thermal power and the channel length over which this power must be applied to completely evaporate or to considerably thin the wall liquid film (in the case of the CHF caused by a dynamic rupture of the liquid film) at the heated channel exit, that is:

$$q''_{\mathrm{cr}} = \frac{h_{\mathrm{lv}}}{\pi D L_{\mathrm{dan}}}(W_{\mathrm{f0}} + \Delta W_{\mathrm{f}}), \tag{1}$$

where W_{f0} is the mass flow rate in the liquid film at the point where the dispersed-annular flow is formed; ΔW_{f} is the total change in the film flow rate over the same length, caused by various mechanisms such as the droplet entrainment from the surface of the film by the vapour core, the deposition of droplets entrained in the vapour core back to the liquid film, the ejection of droplets from the liquid film into the vapour core due to the droplet deposition, the ejection of droplets from the liquid film into the vapour core due to boiling if it takes place in the liquid film. The length L_{dan} can be determined from a heat balance as follows:

$$L_{\mathrm{dan}} = \frac{h_{\mathrm{lv}} W (x_{\mathrm{out}} - x_{\mathrm{dan}})}{\pi D q''} \tag{2}$$

or, taking into account the steam flashing due to the pressure drop along the heated channel, it can be calculated from the following approximate equation:

$$L_{\mathrm{dan}}(P_{\mathrm{in}}) = \frac{W}{\pi D q''}\{x_{\mathrm{out}} h_{\mathrm{lv}}(P_{\mathrm{out}}) - x_{\mathrm{dan}} h_{\mathrm{lv}}(P_{\mathrm{in}}) - [h_{\mathrm{l}}(P_{\mathrm{in}}) - h_{\mathrm{l}}(P_{\mathrm{out}})]\}, \tag{3}$$

where x_{dan} is the thermodynamic quality corresponding to the onset of the dispersed-annular flow. This quality can be determined, for example, from the correlation proposed by Levitan and Borevskiy [26] given by:

$$x_{\mathrm{dan}} = 2.7We_{\mathrm{f}}^{-0.25}\left(\frac{\rho_{\mathrm{l}}}{\rho_{\mathrm{v}}}\right)^{-0.33}, \tag{4}$$

where $We_{\mathrm{f}} = G^2D/\rho_{\mathrm{v}}\sigma$ is the film Weber number.

It must be noted that this method of CHF representation has proven itself to be exceptionally suited for predicting the CHF in uniformly heated tubes [24,25], as well as in channels subjected to axially non-uniform heating conditions [27]. Fig. 6 shows the CHF as a function of the exit quality for different pressures and mass fluxes. These data reveal the following trends:

- independently of the type of heat flux distribution, for a given pressure and mass flux, in general minor critical quality variations are observed. However, these quality variations do not allow Doroshchuk's boundary quality

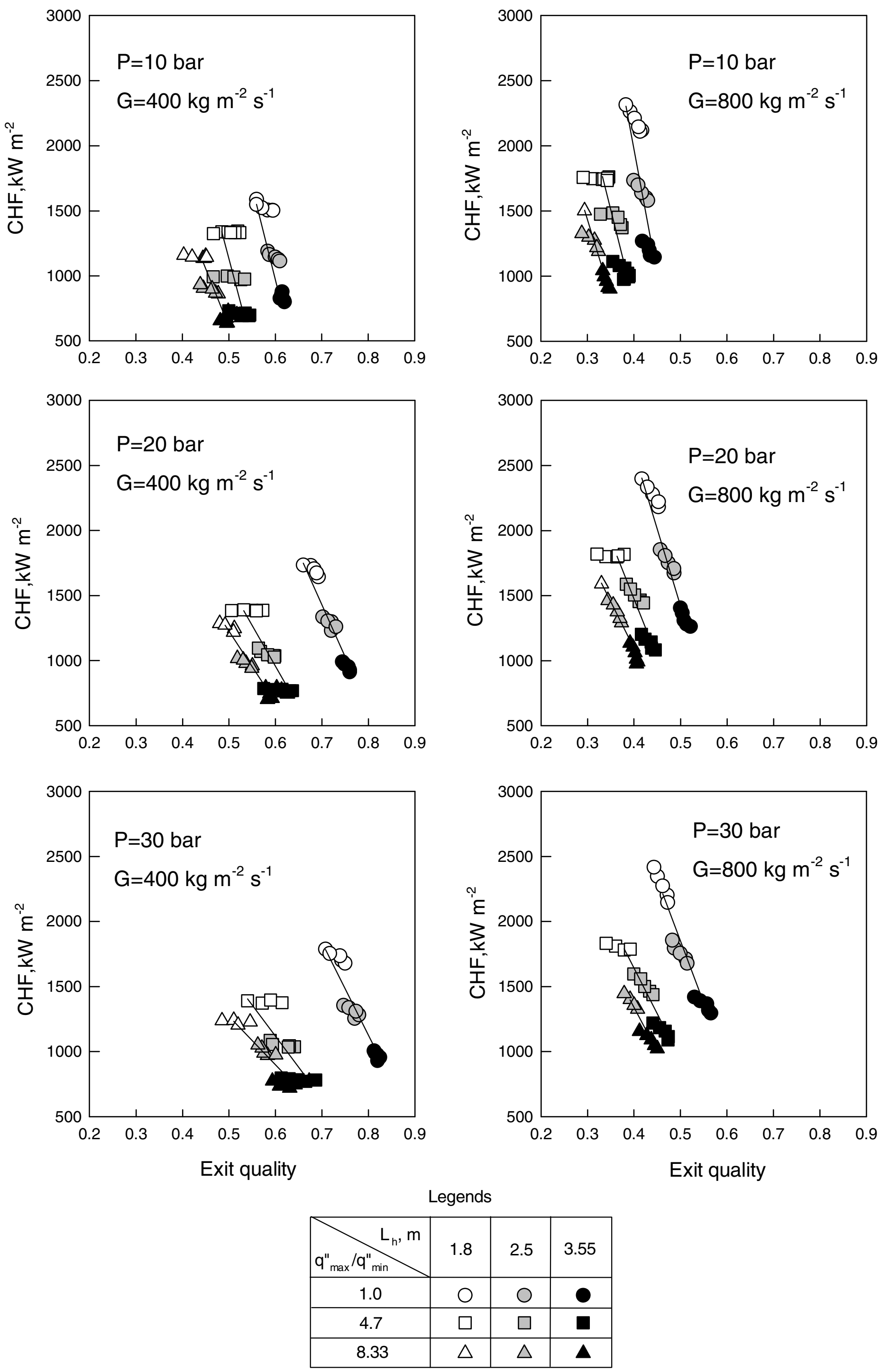

Fig. 6. CHF as a function of the exit thermodynamic quality.

concept to be corroborated. On one hand, this quality variation can reach up to 0.2. On the other hand, by examining the data obtained for each heated length, it can be observed that for certain cases, $q''_{cr}(x)$ is almost parallel to the quality axis. Thus, for these cases and for a given heated length, it could be also concluded that there is a unique value of the CHF. Therefore, Doroshchuk's boundary quality concept can only be regarded as a simplification based on a narrow range of thermodynamic qualities for several number of the heated lengths and do not necessarily reflect all of the actual mechanisms governing the critical phenomenon;

- dissimilar relationships between the CHF and the exit quality for each and all of the heated lengths bring about a substantial data scattering around of the regression lines;
- the CHF values under non-uniform heating conditions are always lower than those for the uniformly heated tube;

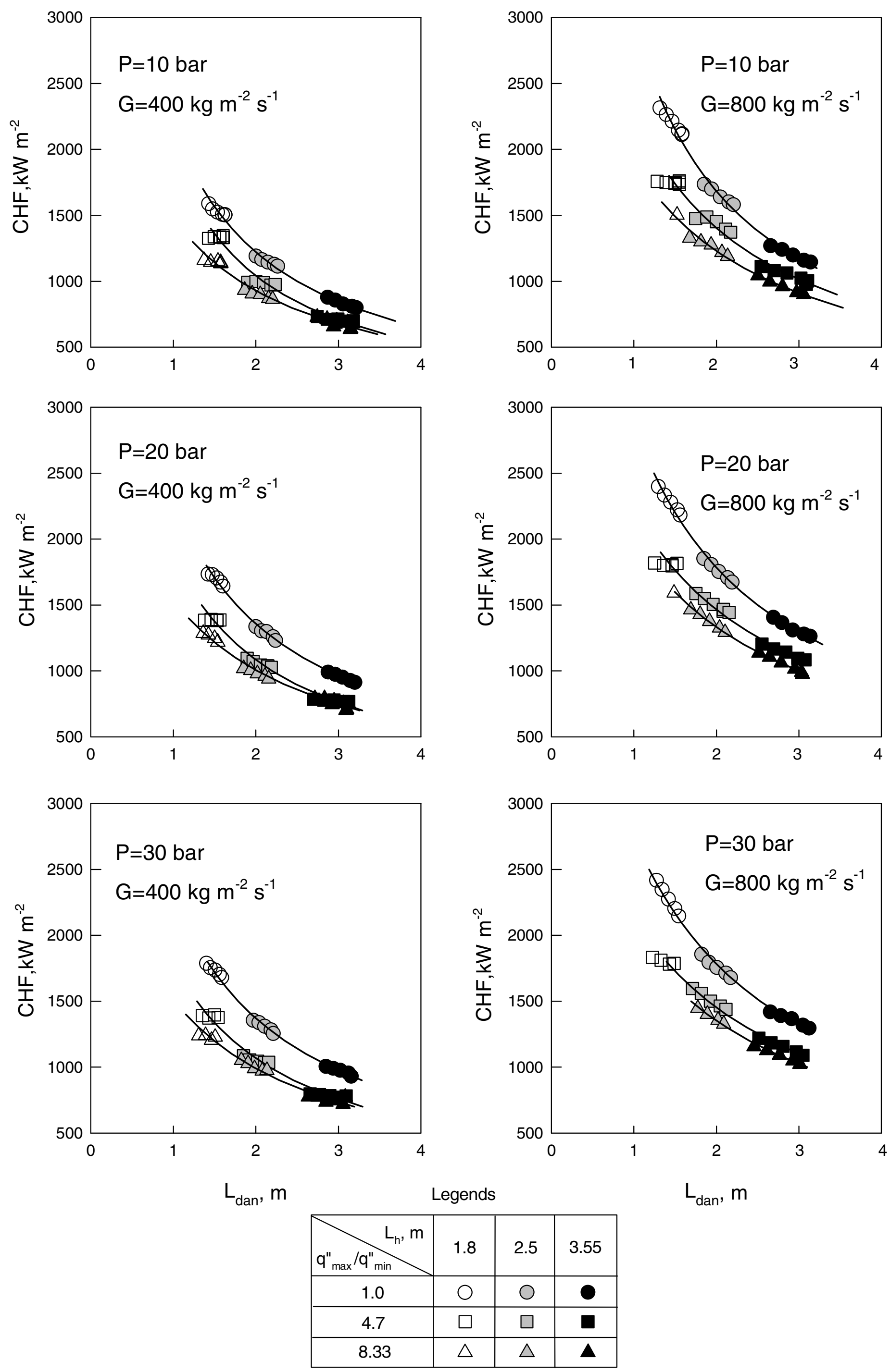

Fig. 7. CHF as a function of $L_{dan}(P_{in})$.

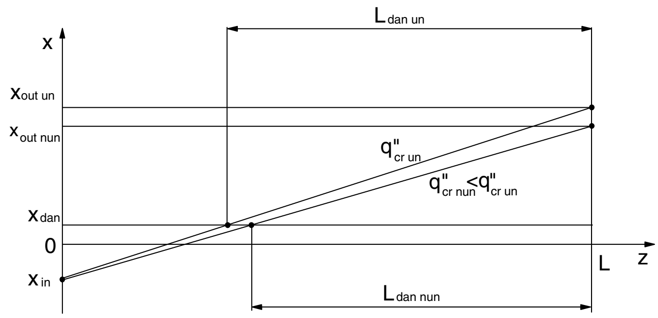

Fig. 8. Length of the dispersed-annular region as a function of thermo-hydraulic parameters.

- despite the close values of the ratios q''_{max}/q''_{mean} for both non-uniformly heated test sections, the CHF values for the case with $q''_{max}/q''_{min} = 8.3$ are in general considerably lower as compared with the corresponding values for $q''_{max}/q''_{min} = 4.7$.

Fig. 7 shows the values of CHF as a function of $L_{dan}(P_{in})$ for the same outlet pressures and mass fluxes. It can be seen that the use of the coordinate system (L_{dan}, q''_{cr}) results in a more compact data grouping with a much lower dispersion. Note that only for the heated length of 1.8 m the relationship between q''_{cr} and L_{dan} is different. The scatter in the data points is not very high and this behaviour can be easily explained by the tube bowing observed on non-uniformly heated tubes. Moreover, it is also possible that such a particular behaviour could be related to the lack of precision in determining the length over witch the dispersed-annular flow takes place or, more exactly, to the use of correlation for x_{dan} (Eq. (4)) which was developed for adiabatic two-phase flows. Taking into account that the maximum deviation is observed at high subcoolings (i.e., low dryout qualities), which correspond to non-typical flow conditions encountered in power boilers, no rectification of the proposed correlation for x_{dan} (e.g. [24]) was carried out. According to Fig. 7, it must also be noted that L_{dan} does not depend on the heat flux distribution. This is due to the fact that this length is only related to the exit quality and to the quality at the position where the onset of dispersed-annular flow occurs (Eq. (2)). Fig. 8 shows that for a given subcooling and a small change of CHF conditions, this position reminds nearly the same. Hence, for identical flow parameters, the variation of the ratio q''_{max}/q''_{min} or q''_{max}/q''_{mean} provokes a change of a unique variable, i.e., the CHF. This circumstance as well as the low data dispersion demonstrates a clear advantage of using the representation of the CHF as a function of L_{dan} for the development of an empirical correlation.

4. Empirical correlation

Even though there is a very low data dispersion about the best-fit curves in terms of the coordinate system (L_{dan}, q''_{cr}), the influence of other system parameters (mass flux, pressure, heat flux distribution characteristics) is quite complex. For instance, a data analysis has shown that within the range of flow parameters used for this study, the values of CHF initially increase with increasing the pressure and then start decreasing with pressure. Under such circumstances the correlation which might provide an accurate prediction of CHF cannot be "simple" [28], even for uniformly heated channels.

4.1. Uniformly heated tubes

Firstly, only the CHF data for the uniformly heated tube have been considered. Without describing the whole process used for developing the proposed correlation, let us present its final form that is given by the following relationship:

$$q''_{cr} = \frac{h_{lv} W}{\pi D L_{dan}} \left(\frac{W_{f0}}{W} + \frac{\Delta W_f}{W} \right) = \frac{h_{lv} W}{\pi D L_{dan}} \cdot \Phi, \qquad (5)$$

$$\Phi = \frac{35.48 \cdot (1 + 2.5e^{-5 \cdot W}) \cdot L_{dan}^{1+P_r W^{0.275}}}{L_{dan} + 12.05 \cdot P_r W} \cdot P_r^{0.635} \cdot (1 - P_r^{0.0172}). \qquad (6)$$

It must be noted the use of the mass flow rate W instead of the mass flux G; according to Olekhnovitch et al. [24] this leads to a considerable reduction of the influence of the channel inside diameter on the CHF value. Therefore, the proposed correlation, in principle can be applied to tubes having slightly different diameters. A comparison between the measured and predicted CHF values obtained using the aforementioned correlation is given in Fig. 9. The root mean square (rms) prediction error presented in this figure is calculated as follows:

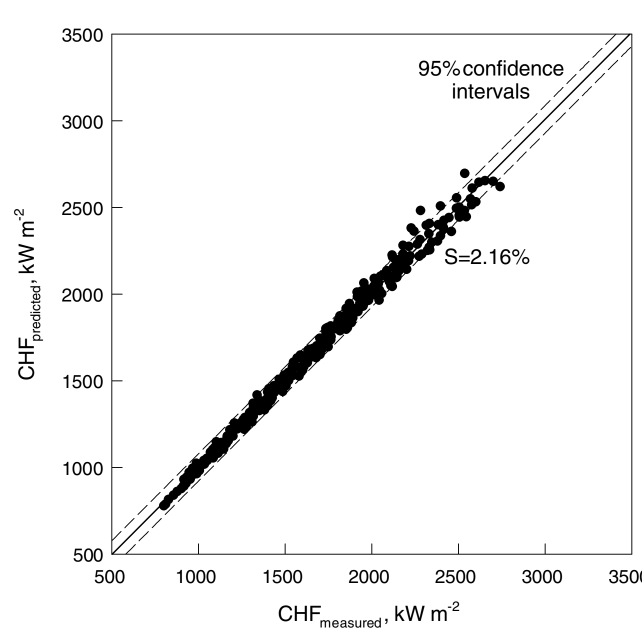

Fig. 9. Comparison of predicted and measured CHF values under uniform heating conditions.

$$S = \sqrt{\frac{1}{N-m}\sum_{i=1}^{N}\left(q''_{\mathrm{cr}_{\mathrm{predicted}}}/q''_{\mathrm{cr}_{\mathrm{measured}}} - 1\right)^2}, \tag{7}$$

where $(N - m)$ is the correlation degree of freedom (N is the total number of data points and m is the number of empirical coefficients used in the correlation; in the present case $N = 343$ and $m = 7$).

The CHF uncertainties presented in Fig. 9 were determined in accordance with the so-called direct substitution method (DSM) which consists of comparing measured CHF values with those calculated from the correlation by using the L_{dan} determined from measured parameters. However, another approach known as a heat balance method (HBM) can be employed as well. In this case, the length L_{dan} must be determined from measured inlet parameters and by increasing the heat flux until it reaches the predicted value. From a statistics view point, this method is rather incorrect [29], but its application results in a certain (sometimes considerable) reduction of prediction error. Therefore, this method is frequently used for the evaluation of the CHF correlation goodness. Fig. 10 presents the prediction error frequency distribution obtained using the HBM approach. It can be seen that this distribution corresponds to the normal probability distribution with a standard deviation $\sigma = 1.8\%$. Taking into account that: (i) the critical phenomenon has a stochastic nature, (ii) a rather subjective method is used for detecting CHF and (iii) there are relatively high statistic measurement uncertainties essentially related to variable electromagnetic and temperature fields, it can be stated that this value is very close to the minimum feasible dispersion of the CHF data measurement.

Usually, a new correlation is compared with some recognized correlations. The authors of the present work do not believe that such a comparison is necessary for the proposed correlation, because it was developed for a specific range of system parameters by using a rather limited data base. Thus, its precision could be a priori higher than the precision of any corresponding more general correlation. Nevertheless, it must be pointed out that an outstandingly low rms prediction error of the proposed correlation shows clearly that the representation of the CHF as a function of the length over which dispersed-annular flow takes place has a distinct advantage over other dryout-type CHF prediction methods. In addition, note that this correlation is based on $L_{\mathrm{dan}}(P_{\mathrm{in}})$; hence its appropriate application necessitates knowing or calculating the inlet pressure.

4.2. Circumferentially non-uniform heating

It is clear that the goodness of a predicting tool for any complex physical process is directly related to the capability of this tool to reflect all mechanisms that determine the process itself. In the case of the CHF and, in particular, the CHF in non-uniformly heated channels, the number of these mechanisms is quite large and they were barely studied.

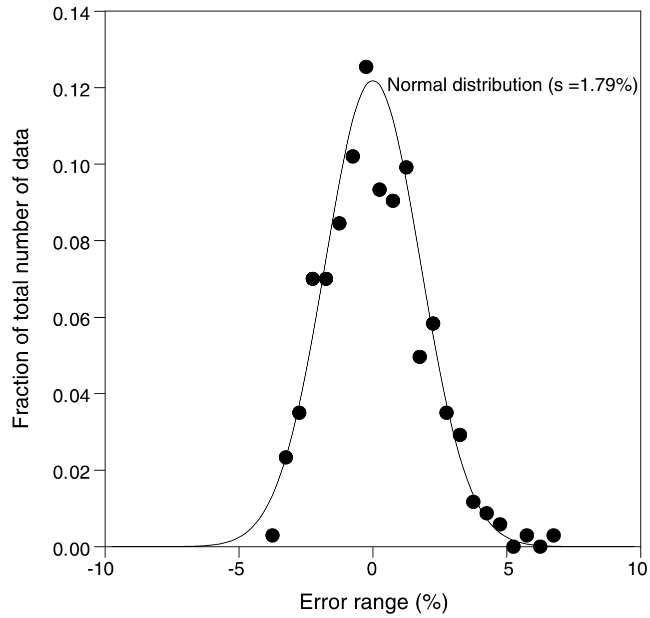

Fig. 10. Frequency distribution of prediction errors (uniform heating; HBM).

Thus, presently a more or less precise CHF prediction method can be improved either by employing a direct data fitting (e.g. the creation of CHF look up tables [30], the application of artificial neural network methods [31], etc.), or by developing classic correlations applying the regression analysis to a set of functions chosen on the basis of mathematical considerations. Even though the first approach gives satisfactory results for predicting CHF under uniform heating conditions, it is doubtful that it can provide a convenient accuracy for the case of non-uniformly heated channels. Indeed, the creation of multiple tables for different heat flux distributions is hardly feasible. Further, since the influence of the heat flux non-uniformity is so complex, singular correction functions should de developed for multiple regions with different combinations of CHF determining parameters (P, G, x or L_{dan}). Taking into account the size of CHF table, the development of such correction functions presents a very difficult computational task. Moreover, in order to fulfill this requirement it is necessary to have at least a several times much larger non-uniform CHF database than the CHF look-up table itself. An empirical correlation, even if it usually has a larger prediction error, especially near the limits of its application, at least can be developed more easily. Fig. 11 presents the results of a virtual (thought) experiment on the effect of the non-uniform heat flux distribution on the value of CHF. It can be seen that, in the region of interest, the influence of the both heat flux ratios $q''_{\max}/q''_{\min}$ and $q''_{\max}/q''_{\mathrm{mean}}$ can be adequately represented by an exponential function given by the following form:

$$\Psi = q''_{cr_{\mathrm{un}}}\{1 - a[1 - \mathrm{e}^{-b(q''_{\max}/q''_{\min(\mathrm{mean})}-1)}]\}. \tag{8}$$

In reality the coefficients a and b should depend on all system parameters which could be very difficult to adjust. Consequently and accordingly to this analysis no

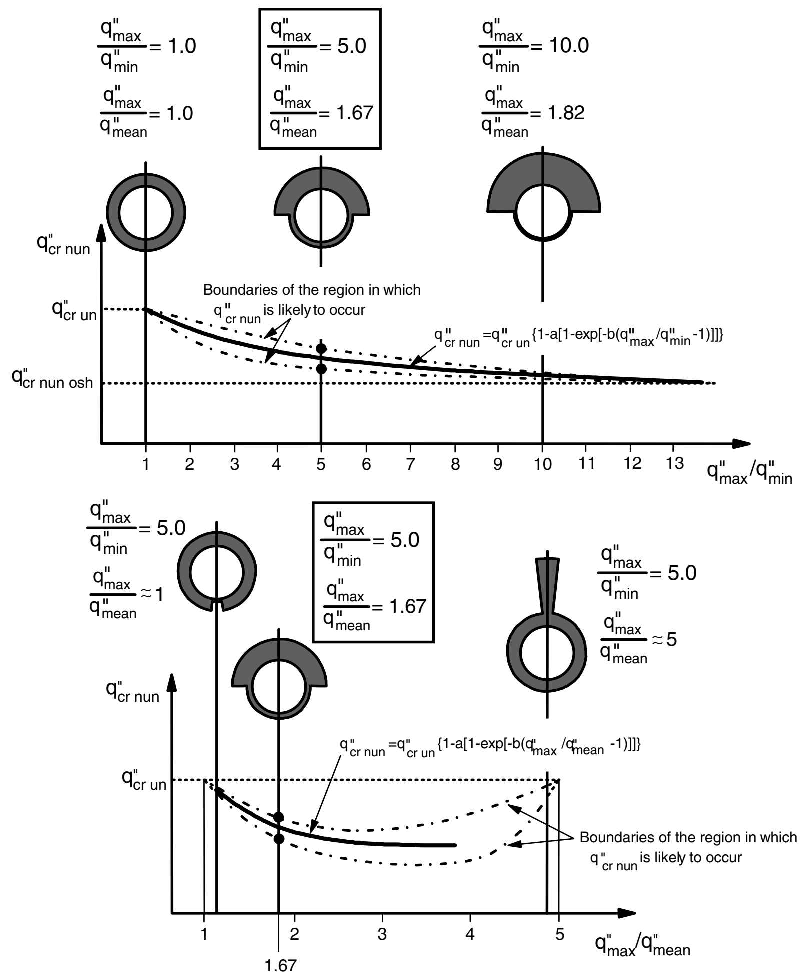

Fig. 11. Influence of heat flux distribution parameters on the values of CHF.

correction functions were developed; instead, a different method was used. The correlation for the CHF under uniform heating conditions in its generic form was formally applied to the case of the non-uniform heating, that is

$$q''_{\mathrm{cr_{nun}}} = \frac{h_{\mathrm{lv}} W}{\pi D L_{\mathrm{dan_{nun}}}} \cdot \frac{C_1 \cdot (1 + C_2 \cdot \mathrm{e}^{-C_3 \cdot W}) \cdot L_{\mathrm{dan_{nun}}}^{1+P_{\mathrm{r}} W^{C_4}}}{L_{\mathrm{dan_{nun}}} + C_7 \cdot P_{\mathrm{r}} W} \cdot P_{\mathrm{r}}^{C_5} \cdot (1 - P_{\mathrm{r}}^{C_6}) \tag{9}$$

and all the empirical coefficients as well as the length over which the dispersed-annular flow takes place were adjusted using a similar ψ function (i.e., Eq. (8)). The results are listed below:

$$C_1 = C_{1_{\mathrm{un}}} \cdot (1 - 0.532 \cdot E_{\mathrm{q}} \cdot E_{\mathrm{m}}) = 35.48 \cdot (1 - 0.532 \cdot E_{\mathrm{q}} \cdot E_{\mathrm{m}}), \tag{10}$$

$$C_2 = C_{2_{\mathrm{un}}} \cdot (1 - 0.0806 \cdot E_{\mathrm{q}} \cdot E_{\mathrm{m}}) = 2.5 \cdot (1 - 0.0806 \cdot E_{\mathrm{q}} \cdot E_{\mathrm{m}}), \tag{11}$$

$$C_3 = C_{3_{\mathrm{un}}} \cdot (1 - 0.385 \cdot E_{\mathrm{q}} \cdot E_{\mathrm{m}}) = 5.0 \cdot (1 - 0.385 \cdot E_{\mathrm{q}} \cdot E_{\mathrm{m}}), \tag{12}$$

$$C_4 = C_{4_{\mathrm{un}}} \cdot \mathrm{e}^{-0.444 \cdot R_{\mathrm{q}} \cdot E_{\mathrm{m}}} = 0.275 \cdot \mathrm{e}^{-0.444 \cdot R_{\mathrm{q}} \cdot E_{\mathrm{m}}}, \tag{13}$$

$$C_5 = C_{5_{\mathrm{un}}} \cdot [1 + 0.507 \cdot E_{\mathrm{m}} \cdot (1 - \mathrm{e}^{-0.0195 \cdot R_{\mathrm{q}}})] = 0.635 \cdot [1 + 0.507 \cdot E_{\mathrm{m}} \cdot (1 - \mathrm{e}^{-0.0195 \cdot R_{\mathrm{q}}})], \tag{14}$$

$$C_6 = C_{6_{\mathrm{un}}} = 0.0172, \tag{15}$$

$$C_7 = C_{7_{\mathrm{un}}} \cdot (1 + 0.0927 \cdot E_{\mathrm{q}} \cdot E_{\mathrm{m}}) = 12.05 \cdot (1 + 0.0927 \cdot E_{\mathrm{q}} \cdot E_{\mathrm{m}}), \tag{16}$$

$$L_{\mathrm{dan_{nun}}} = L_{\mathrm{dan_{un}}} \cdot (1 - 0.307 \cdot E_{\mathrm{q}} \cdot E_{\mathrm{m}}), \tag{17}$$

$$E_{\mathrm{q}} = 1 - \mathrm{e}^{-R_{\mathrm{q}}}, \tag{18}$$

$$E_{\mathrm{m}} = 1 - \mathrm{e}^{-1.31 \cdot R_{\mathrm{m}}}, \tag{19}$$

$$R_{\mathrm{q}} = \frac{q''_{\mathrm{max}}}{q''_{\mathrm{min}}} - 1, \tag{20}$$

$$R_{\mathrm{m}} = \frac{q''_{\mathrm{max}}}{q''_{\mathrm{mean}}} - 1. \tag{21}$$

Fig. 12 shows a comparison between the measured and predicted non-uniform CHF values obtained by using this correlation. The rms prediction error, $S = 4.48\%$, was calculated with the degree of freedom $N - m - k = 598 - 7 - 9 = 582$. Fig. 13 presents the prediction error frequency distribution obtained by applying the HBM approach. Even though the standard deviation $\sigma = 3.72\%$ is two times greater than the corresponding value of the

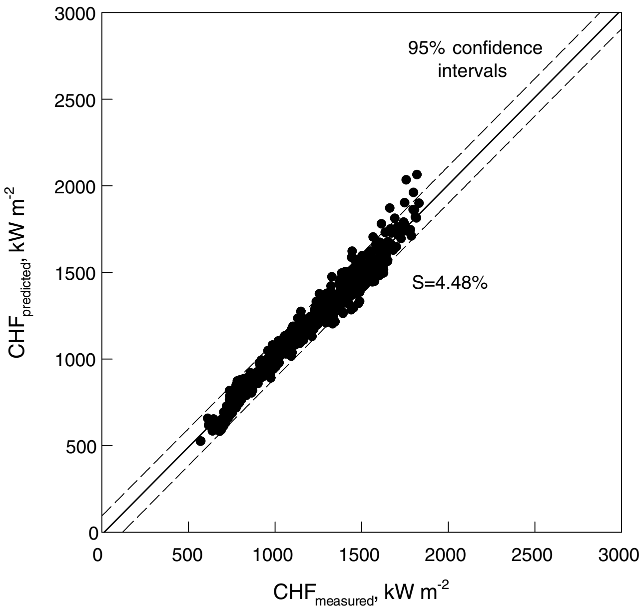

Fig. 12. Comparison of predicted and measured CHF under non-uniform heating conditions.

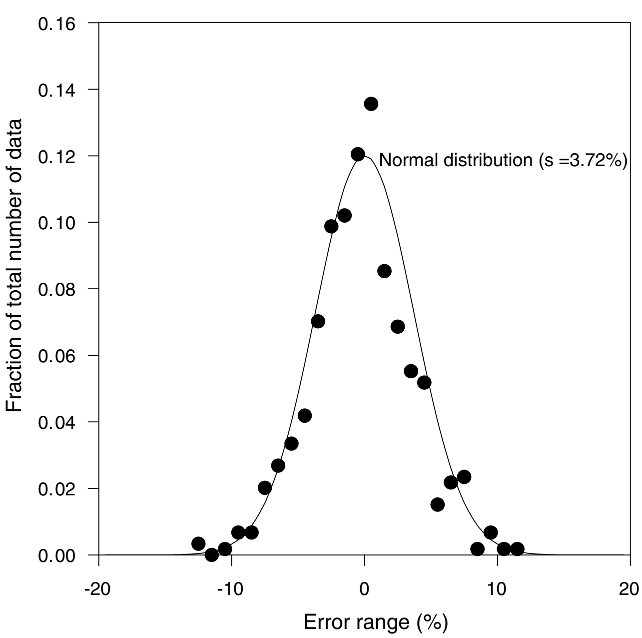

Fig. 13. Frequency distribution of prediction errors (non-uniform heating; HBM).

correlation (5), it is nevertheless almost two times smaller than that given by the authors of “The 2005 CHF look-up table” [30] for the uniformly heated round tubes.

5. Conclusions

The main objective of this work was to collect CHF data for a wide range of flow parameters under circumferentially non-uniform heating conditions at low and medium pressures. The experiments were carried out in 22 mm ID vertical test sections for the three heat flux ratios q''_{max}/q''_{min}: 1, 4.7 and 8.3 as well as for the three heated lengths: 1.8, 2.5 and 3.55 m. An analysis of the experimental results using a number of different critical heat flux representations was carried out with the goal of creating a correlation for predicting CHF under the above-mentioned conditions. It has been shown that the traditional representation of the CHF as a function of the exit quality demonstrates considerable data dispersion and has no promising prospects towards the stated goal. On the other hand, the presentation of the CHF data in terms of the coordinate system (L_{dan}, q''_{cr}) has revealed that the scatter of the data about the resulting regressions is substantially lower. On the base of this representation an empirical correlation was developed. The root mean square prediction error of the proposed correlation when it is used in conjunction with a heat balance method is 3.7%.

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