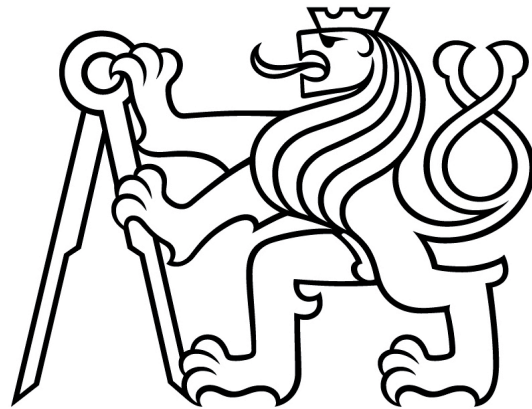


**ČESKÉ VYSOKÉ  
UČENÍ TECHNICKÉ  
V PRAZE**

**FAKULTA  
STROJNÍ**



**TEZE  
DISERTAČNÍ  
PRÁCE**



ČESKÉ VYSOKÉ UČENÍ TECHNICKÉ V PRAZE  
FAKULTA STROJNÍ  
ÚSTAV ENERGETIKY

TEZE DISERTAČNÍ PRÁCE

Dynamics of Heat Transfer During Cooling of Overheated Surfaces

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Doktorský studijní program: Strojní inženýrství

Studijní obor: Energetické stoje a zařízení

Školitel: doc.Ing. Václav Dostál, Sc.D.

Teze disertace k získání akademického titulu "doktor", ve zkratce "Ph.D."

Praha

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# Název česky: Dynamika přestupu tepla při vychlazování vysoce rozehřátých povrchů

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S disertací je možno se seznámit na oddělení vědy a výzkumu Fakulty strojní ČVUT v Praze, Technická 4, Praha 6.

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předseda oborové rady oboru Energetické stroje a zařízení

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# Current State of the Rewetting Research

## Analytical Studies

Many mathematical models were created in the last forty years in order to solve rewetting phenomenon for different geometries. These models are either one-dimensional or two-dimensional. Also, several three-dimensional models can be rarely found with advances in computer technology. [A1]

## Common Simplifications

Over time, many models have been created that contain common simplifications. These widely used assumptions are:

- Homogenous wall of infinite height
- Uniform wall thickness
- Neglect of precursory cooling (adiabatic dry region or its part)
- Constant quench front velocity
- Uniform heat transfer coefficient in the wet region. This assumption is used if the wet region is one-regional. Otherwise, if the wet region is two-regional, the region is split into two sub-regions: boiling and not boiling
- Only single fuel rod is included in the model. influence of surrounding rods is neglected.
- No heat generated within a wall

## State of the Art

The basic rewetting model includes the mentioned simplifications. The most basic model for the analytical solution of the rewetting phenomenon is model which includes only two regions - wet (rewetted) region and dry region. A constant heat transfer rate for the wet region and the adiabatic boundary condition for the dry region are assumed in this approach in many analytical efforts. The

model is then solved typically for infinite slab or infinite rod with the quasi-stationary approach. Most of these models were one-dimensional. Sometimes they included constant heat transfer coefficient also for the dry region instead of adiabatic one. However, the heat transfer rate in the dry region is much lower than the heat transfer coefficient in the wet region so the heat transfer rate in the dry region can be neglected. This simplification is present in most works.

However, it must be noted, that assuming adiabatic or uniform heat transfer coefficient in the dry region is not correct from the physical point of view and thus resulting data cannot be physically correct. In this case, the heat transfer rate rises along the geometry in front of the quench front. The rising heat transfer rate is caused by rising wall temperature from the quench front. This behavior is unreal.[1] With consideration of multi-regional dry region, we can get more acceptable results. In this case, the farthest dry part of geometry can be considered with adiabatic boundary condition. Such effort was published by Elias and Yadigaroglu (1977)[2].

The first model with precursory cooling was proposed by Edwards and Mather (1973)[3]. The model was two-dimensional. This model included exponential heat flux in front of the quench front as well behind it. The solution presented by Dua and Tien (1976)[4] included constant heat transfer coefficient in the wet region. Then analysis of precursory cooling of a tube with finite precursory length for the bottom and top flooding was proposed by Hirano and Asahi (1980)[5]. They solved the problem as one-dimensional with three-regional heat conduction model, where the first region included transition boiling, the second one included film boiling and the last one was characterized by adiabatic boundary condition (heat radiation was neglected). Other models by Olek (1988)[6] and (1990)[7] including precursory cooling were solved by separation of variables method or Wiener-Hopf technique in cartesian or cylindrical coordinate system.

Several analytical studies, which contained decay heat power, were published from the 1970s. One-dimensional unsteady analytical and numerical study was presented by Chan and Zang (1994)[8]. Their study was focused on grooved and smooth plates. [1] Another study on rewetting with the uniform heat source in infinite slab was presented by Platt (1993)[9]. An effort by Satapathy and Sahoo (2002)[10] was solved for the case of infinite cylindrical geometries using Wiener-Hopf technique. Then Sahu (2008)[11] proposed a solution for this model using the heat balance integral method technique. Other models included multi-regional assumptions of heat transfer coefficient and other variations such as thermal dependence of physical properties of the cooled wall, namely Elias and Yadigaroglu (1977)[2], Sawan (1979)[12] and Bera and Chakrabarti (1996)[13].

It can be expected, that temperature-dependent parameter will heavily influence the rewetting process due to rapid cooling of the wall. It is evident, that heat conduction included in Biot number is the most influencing parameter from all temperature-dependence parameters. Some analytical efforts include the temperature dependence into solution. The solution of the problem is then a

very non-linear set of equations. Works with incorporated temperature dependencies are for example solutions by Olek and Zvirin (1985)[14] and Sahu (2009)[15].

## Experimental Studies

Many experimental works have been performed in the last fifty years, especially in the 1970's and 1980's in order to deepen the knowledge of the rewetting phenomenon. These studies have been performed for various geometries such as an annulus, rod bundles, plates, etc. for a wide range of initial wall temperatures (generally  $300^{\circ}C - 700^{\circ}C$ ) and also under different pressures and for several flow rates. Some works even dealt with the effect of reduced gravity. Subsequently, a lot of analytical efforts has been based upon obtained data.

### Overview of Experimental Studies

Shires et al.[16] investigated the experimentally rewetting phenomenon in the year 1964. The resulting finding of the work was, that time needed for surface rewetting is prolonging with rising initial wall temperature. Another consequential result was, that a specific temperature level exists in the rewetting phenomenon and the temperature represents the limit for re-establishing solid-liquid contact.

Bennett (1966)[17] conducted a set of experiments with heated stainless steel tube in the steam environment. He proved the existence of the exact quenching temperature for every pressure level within many pressure level ranges with maximum pressure up to 1000psi (6.9MPa). Yamanouchi (1968)[18] presented quenching experiments and he described a conduction controlled rewetting phenomenon with axial heat conduction toward the quench front. This heat is then removed by fluid in the rewetted region due to high heat transfer coefficient. In this work the wet region heat transfer coefficient was taken as a constant, in the dry region he assumed the heat transfer rate as a function of actual wall temperature and saturated temperature of the coolant in the shape:

$$q'' = 1.5(T_w - T_s)^{1.5} \quad (1)$$

Later, Tothompson (1972)[19] introduced exponential heat transfer rate in the wet region as a function of the third power of saturation temperature. An important point of this work is the influence of wall heat capacity on the rewetting velocity. The observed effect of the capacity was that quenching of a zircalloy tube was twice as short than for stainless or Inconel tube. A similar model was introduced for the dry region by Karyampudi and Chon (1976)[20].

Duffey and Porthouse (1973)[21] examined a wide range of materials with initial wall temperatures within the range  $300 - 800^{\circ}C$ , flow rates  $0.1 - 30 g/s$ . The length of the test section was  $20 cm$ .

The conclusion of their work was, that one-dimensional analytical solution is in good agreement with experimental data for low Pe and Bi numbers for the bottom and top flooding. On the other hand, the process begins to be more unpredictable in case of high velocities, thick walls and varying heat conduction of the wall. Another result was a designation of Leindenfrost temperature within the range of  $190 - 250^\circ C$ .

Experimental studies also showed a uniformity of quench front velocity along the cooled geometry. However, a majority of experimental studies was performed on geometries shorter than  $1.5\text{ m}$  whereas nuclear reactor fuel pins are over  $3\text{ m}$  high. With increasing distance to be cooled down also increases dry region that is cooled by vapor and droplets (precursory cooling). This means, that more heat will be removed during the process in front of the quench front a thus a lower temperature will be present at the specific point, while quench front reaches this place. Higher precooling of the dry region leads to higher quench front velocity and to unequal front advance. [A3]

One of the latest experimental efforts is work by Saxena et al. (2001)[22]. Their experimental loop consisted a test section  $3030\text{ mm}$  high for bottom flooding and  $2630\text{ mm}$  high for top flooding. Initial wall temperature range was  $200 - 500^\circ C$ . Two correlations based on obtained data were proposed for both directions of flooding.

- For top flooding:

$$u_T = \frac{5546}{\Theta_q^{0.15} \rho C} \left( \frac{Q}{2\pi r} \right)^{0.8} \quad (2)$$

- For bottom flooding:

$$u_B = \frac{7285}{\Theta_q \rho C} \left( \frac{Q}{2\pi r} \right)^{0.84} \quad (3)$$

However, this study also does not reflect the varying quench front velocity dependent on actual front position. Incorporation of some kind of position is necessary for the better description of the quench front velocity for high test sections.

Generally, we can conclude according to experimental efforts, that quench front velocity rises with:

- Coolant flow rate
- Coolant sub-cooling
- Thinner wall
- Lower internal heat source power

Selected experimental studies are graphically mapped in Figure 1. This map clearly shows the lack of bottom rewetting studies with heated length over  $1.5\text{ m}$  and simultaneously the lack of experiments



covering wide range of initial wall temperatures. The presented study is highlighted in the figure. It is also evident from this map, that lengths over 2 meters are very rare and only length of 3.6 *m* is covered by experimental studies with rod bundle test section configuration. The reason is, that the length of 3.6 *m* is the height of typical pressurized water reactor. However, this study focuses on basic research on rewetting behavior and for this purpose a basic geometry such as annulus is much more suitable. Unfortunately we can find only two experimental efforts above length of 1.5 *m*: Saxena (1998)[22] and Cho (2007)[23]. However, these studies do not cover a very wide temperature range. Despite the fact that this map clearly shows the scattering of experimental works, it does not take into account the mass flows and the channel cross-sections. This information is often difficult to obtain and properly compare each other's study.

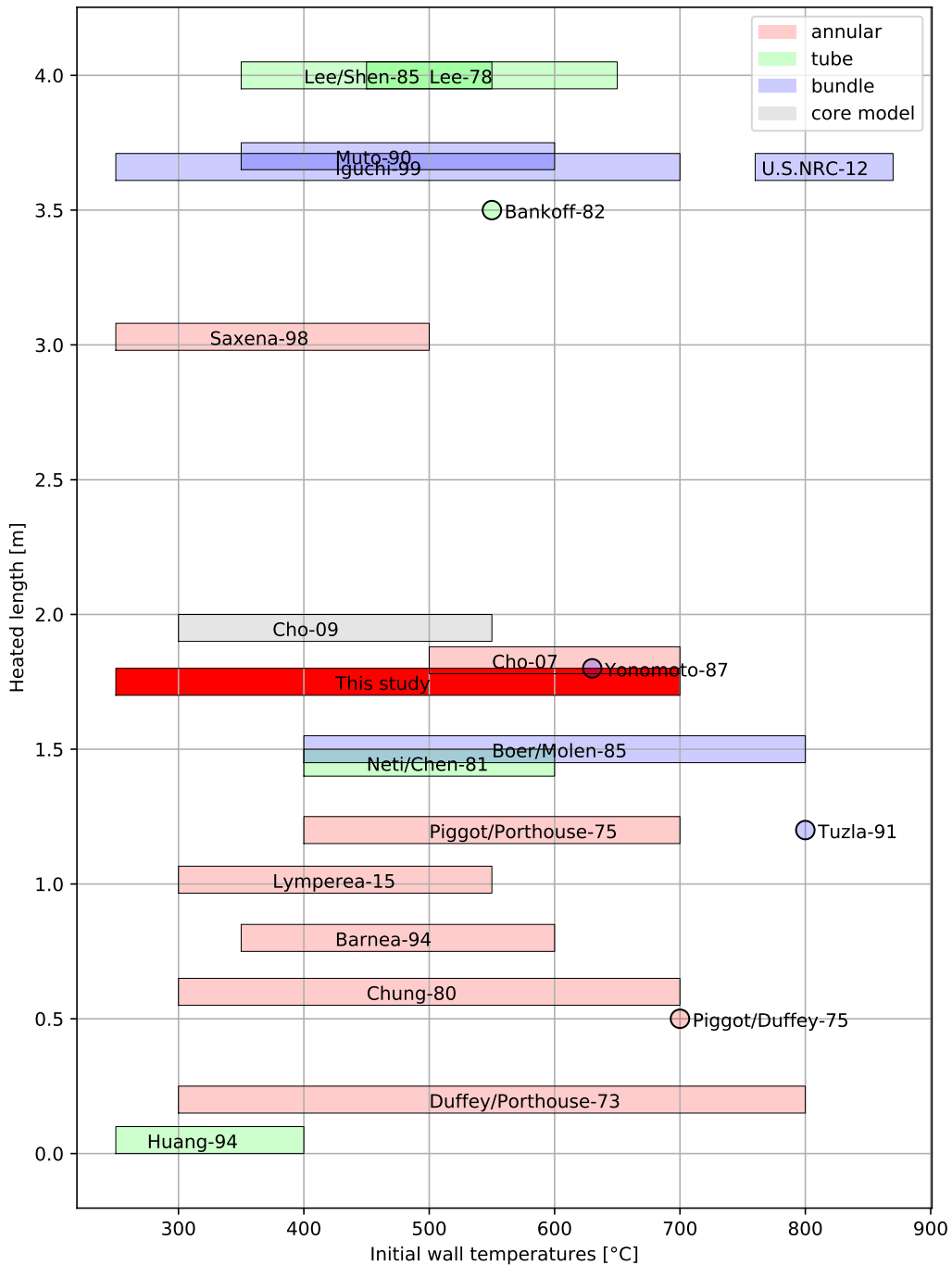


Figure 1: Map of experimental studies on bottom flooding ??

It can be concluded based on reviewed experimental efforts, that a lot of works includes test sections with the length under  $1.5\text{ m}$  and higher test sections are performed on complex geometries such as rod bundles. The annular channel is very good channel geometry for phenomenon description and moreover, these data can be also the input for analytical studies. This is very difficult for too complex geometries. Individual experiments can be slip into two groups: the first group takes initial wall temperatures within the range from  $300\text{ }^{\circ}\text{C}$  to  $500\text{ }^{\circ}\text{C}$  and the second group from  $500\text{ }^{\circ}\text{C}$  to  $700\text{ }^{\circ}\text{C}$ . Experimental studies covering a wide range of temperatures are quite rare, especially in terms of basic geometries such as annulus.

# Goals

Cooling of overheated surfaces also known as quenching is still not well-known. It is a very chaotic transition process where every new experimental study is an important piece of the puzzle on the way to fully understand the phenomenon. The motivation for this study is to propose a detailed view on the quenching phenomenon, specifically bottom reflooding of an annular channel with a heated model of nuclear fuel pin. The result can serve as an input for other analytical studies in the field and it can point out several side effects accompanying the process of reflooding. These outputs can improve predictions of processes where quenching takes place, especially processes related to LOCA accident in nuclear reactor safety analyses.

The main goal of this study is to deepen the knowledge of the quenching phenomenon. This overall goal was split into several individual goals:

- Build an experimental loop for investigation of quenching phenomenon in a flooded annular channel with three changeable tubes for consideration of accumulated heat and more real uneven heat generation
- Collect a large number of new experimental data with all relevant variables influencing the process
- Development of new approaches for experimental data evaluation and self-operational code for batch processing of individual quenching data with a built-in algorithm for correlation development
- Propose correlations for quenching and nucleate boiling temperatures, which are the main breakpoints in the process
- Propose formulas for calculations of heat transfer coefficients during the process
- Development of three-regional quenching model for further analytical studies
- Development of a correlation (or full set of correlations) for prediction of quench front velocity
- To point out the accompanying phenomena such as the influence of spacers and pressure peaks on the bottom flooding process

# Experimental Loop

For the purpose of detailed investigation of rewetting phenomenon an experimental loop has been built. Its component description, construction and parameters are in this chapter.

## Main Parts

The used experimental loop was originally constructed for critical heat flux experiments. After these experiments, the equipment was rebuilt for a purpose of rewetting research. [A1]

Main parts of the experimental loop are as follows:

- Variable hydraulic circuit
- Test section
- Power source
- Data acquisition system (DAS)

The loop scheme is shown in Figure 2.

Table 1: Main parameters of the presented study

Mass flux (G)	$80 - 270 \text{ kg.m}^{-2}.\text{s}^{-1}$
Coolant	Water
Coolant temperature ( $T_{cool}$ )	$\sim 20^\circ\text{C}$
Initial wall temperature ( $T_0$ )	$250 - 700^\circ\text{C}$
Pressure	Atmospheric
Internal heat source given by steady state at initial wall temperature	

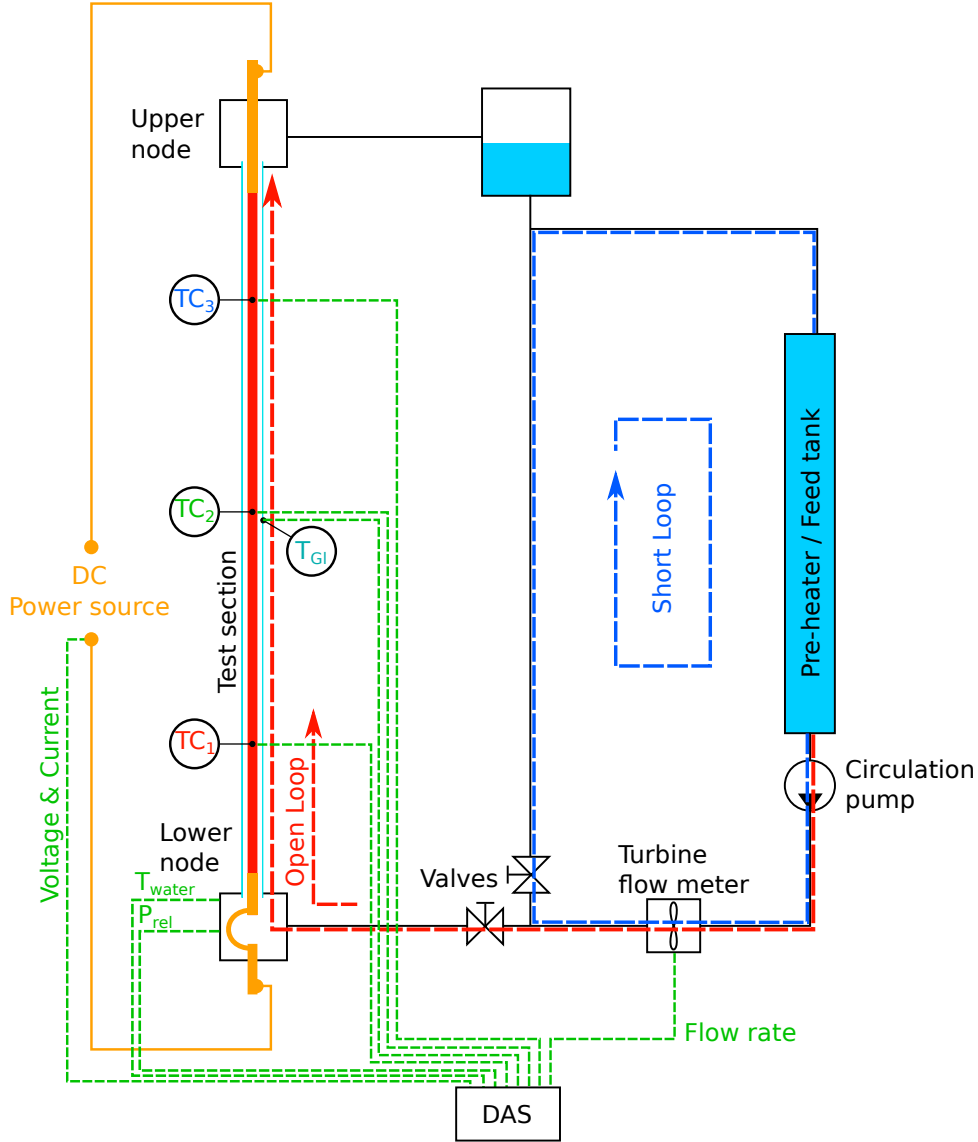


Figure 2: Simplified experimental loop scheme

## Test Section

The test section is the main part of the whole experimental equipment. It consists mainly of an annular channel and two nodes (chambers): upper and lower one. The flow channel is constructed from two parts. The first one is a transparent outer barrier made of silica glass and the second part is an inner electrically heated stainless steel tube equipped with the set of thermocouples.

In this study, annulus was chosen for several reasons. The first reason was technical restrictions such as maximum input power and dimensions of outer barrier. The second pragmatic and the more important reason is, that obtained experimental can be used for further analytical studies. Complex geometry such as rod bundle with complicated spacers are very good for disposable description of the phenomenon, but for analytical processing of data, a "simpler" approach is required. The main reason is, that mathematical solution techniques containing complex boundary conditions and its discontinuities are almost unable to solve the problem for very complicated geometries due to

resulting singularities in solution. Moreover, an annulus is clear, simple and uniform flow area and thus attention can be focused at just one advancing quench front and its behavior. The annulus is also a very good basis for further works in the field.

## Heated Tubes - Models

The inner tube in the test section is changeable. There were used three different tubes - models during experiments. These models have different wall thickness for heat capacity and profiled heat flux effect investigation. All models have the same outer diameter of  $9\text{ mm}$  and the wall thickness differs.

The tested models are ( $\delta$  is the wall thickness):

- Model A:  $\delta = 0.5\text{ mm}$ , steel *MONEL*<sup>®</sup> K500 (UNS N05500)
- Model B:  $\delta = 1.0\text{ mm}$ , steel X6CrNiTi18 - ČSN 17 248 / AISI-321
- Model C:  $\delta = f(z)$ , steel *MONEL*<sup>®</sup> K500 (UNS N05500)

The last tube has variable wall thickness for simulation of the variable heat source within the geometry. The models are heated by direct current. The thinner wall means smaller cross-sectional area and this leads to lower current density i.e. lower generated heat at given position. The profile of the thickness function corresponds to the cosine function.

# Experiment Setup and Methodology

Experimental methodology is described in this chapter. The measuring process can be summed up in the following three steps:

- Flow rate setup
- Test section heating
- Flooding

Each experimental point, i.e. given initial wall temperature and flow rate, was measured at least three times ideally five times. The number of minimum three repetitions was chosen due to degradation of silver solder at high temperatures. This precaution ensured the correct surface temperature measurement.

All initial wall temperature level are in Table 2 and initial flow rates in Table 3. These conditions were set for each model in the same way. This gives 140 experiments per model. In total, 420 experiments were performed and evaluated consequently.

Table 2: Initial wall temperatures

Initial wall temperature levels $T_0$ [ $^{\circ}C$ ]									
250	300	350	400	450	500	550	600	650	700

Table 3: Initial mass flow rates

Initial mass flow rates $G$ [ $kg.m^{-2}.s^{-1}$ ] (reps.)			
80 (3)	110 (5)	190 (3)	270 (3)

# Data Processing

Before the full-scale data evaluation, each data set passed through individual evaluation. This step contains a calculation of average quench front velocities, surface heat fluxes, evaluation of temperature profiles and its derivatives, identification of quenching and rewetting temperatures, calculation of heat transfer coefficients, pressure drops and etc.

Full datasets of each experiment are converted and stored on a hard drive with corresponding full-scale plots for further possible analyses. Subsequently, the flooding phase of the experiment is extracted from these sets. The extraction process is done automatically by the specialized script. The beginning of the flooding phase is detected through the first order derivative of the pressure. The rapid pressure increase occurs while the coolant enters annular flow channel. This rapid change can be easily found in the pressure array through set pressure gradient threshold. If pressure gradient is higher than this threshold, the script searches for its local minimum backward in time.

The best-proved interpolation method for given data was a Piecewise Cubic Hermite Interpolation Polynomial (PCHIP). The main difference between the spline and PCHIP curve is that PCHIP does not overestimate temperatures near the edges in temperature profile and the curve looks more natural and credible. [A3] As it turned out, spline and PCHIP can be used for identification of the position of critical heat flux and second derivative extremes. But the main lack of spline and similar functions is an underestimation of the maximum first-order derivative value, which is used for heat flux and heat transfer coefficient calculation.

Some experimental data needs to be filtered in order to get more consistent and clear flow of the specific variable. The filtered data in the presented experiment were variables, which show very frequent small chaotic changes around mean value. The filtered variable was pressure, flow rate, and electric current. For this purpose, a Savitzky-Golay filter was used. Savitzky-Golay filtering can be thought of as a generalized moving average. You derive the filter coefficients by performing an unweighted linear least-squares fit using a polynomial of a given degree. For this reason, a Savitzky-Golay filter is also called a digital smoothing polynomial filter or a least-squares smoothing filter. Note that a higher degree polynomial makes it possible to achieve a high level of smoothing without attenuation of data features. [24]



## Quench Front Velocity

Quench front velocity is defined as a known distance overcome in a measured time interval. The known distance is, in this case, the pitch of two thermocouples. The chosen easily recognizable moment is the position of the minimum first order derivative of temperature profile. Each temperature has assigned a time point, for example  $T_{1chf}$  is located at  $t_{1chf}$ . The average quench front velocity between thermocouples TC1 and TC2 is then calculated as:

$$u_{12} = \frac{L_{h2} - L_{h1}}{\Delta t_{12}} = \frac{0.5}{t_{2chf} - t_{1chf}} \quad [m.s^{-1}] \quad (4)$$

## Quench Front Position

For further calculations and data evaluations, a simple quench front position script was created. The purpose of the script is to convert time axis into spatial coordinates. Spatially dependent variables can be aligned and they can be compared to each other. This conversion aligns all measured and calculated parameters with respect to known time and spatial points. These sets of points are sent in the PCHIP interpolator and a smooth set of spatial points is obtained as a solution.

## Heat Fluxes

Heat fluxes within the channel must be solved in order to solve heat transfer coefficients. The source of these fluxes is electrically generated heat within the steel wall, which is a function of electrical current and resistance of the wall. The generated heat is removed by a combination of heat radiation, conduction, and convection through the gap of the annulus. It can be assumed, that radiation escapes directly from the wall through the gap and the silica glass barrier due to its high translucency. Rest of the heat is conducted through the silica glass barrier by conduction and then it escapes to the surroundings. The generated heat is accumulated within the wall up to the initial wall temperature which corresponds to the steady state where heat flux from the wall is equal to the heat loss escaping to the surroundings. In front of the quench front and at the front, saturation temperature is assumed as coolant temperature. Beyond the quench point, the coolant temperature has to be calculated from the water inlet temperature and its enthalpy increase.

## Tube Nodalization

The heat balance was calculated in nodalized geometry at three thermocouples. Evaluation of the heat fluxes was performed for all time points within the interpolated arrays of temperatures and velocities. The essential assumption here is, that time-temperature profile is the same for two spatial

nodes at a very short distance. The finite distance named  $\Delta z [m]$  is calculated from actual quench front velocity and the time step of interpolated time array. This obtaining of node height respects rapid changes of temperatures when the quench front passes the thermocouple. The nodalization is shown in Figure 3. The total sum of all heat fluxes (axial-top, axial-bottom, Joule's heat, heat removed by coolant and accumulated heat change) must be equal to zero. The internal tube surface is assumed as adiabatic.

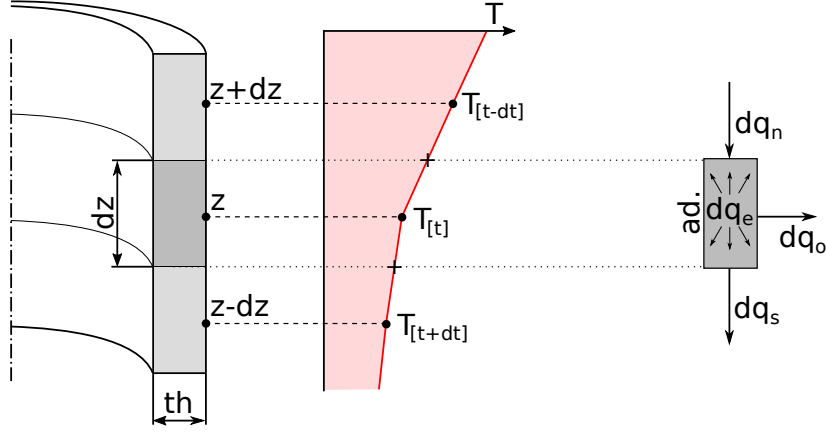


Figure 3: Axial nodalization and heat fluxes

$$q_c = \frac{\Delta T_t}{\Delta t} c(T_t) \rho(T_t) A_z \Delta z - q_n - q_s - q_e \quad [W] \quad (5)$$

Heat balance in full form:

$$q_c = \frac{\Delta T_t}{\Delta t} c(T_t) \rho(T_t) A_z \Delta z - k_n \frac{\Delta T_n}{\Delta z} A_z - k_s \frac{\Delta T_s}{\Delta z} A_z - \frac{r(T_t)}{A_z} I^2 \Delta z \quad [W] \quad (6)$$

## Heat Transfer Coefficients

Calculation of heat transfer coefficient is one of the most tricky part of quenching data evaluation and thus it is very rare in experimental works in the field. When we have calculated areal heat flux into coolant the only thing it needs to be calculated is the coolant temperature. Any direct measuring of temperature in the flow is impossible due to the small area for thermocouple installation and due to the undesirable influence of the thermocouple on the quenching process. The thermocouple within the coolant flow will be cooled down significantly earlier and it generates secondary quench fronts spreading further along the geometry.

## Downstream Heat Transfer Coefficient

Common approach, that can be found in several studies [25], is assumption of saturated steam downstream the quench front i.e.  $T_{cool} = 100^\circ C$ . At high initial wall temperatures, this assumption will be less accurate. In this case, overheated steam will be present in the channel. However the overheating can be assumed to be relatively small and thus negligible. The saturated steam approach is adopted in this section for the dry region. The main reason is a too difficult prediction of the steam velocity in the channel due to the unknown exact evaporated mass of water. Based on the heat transfer version of Newton's law of cooling, the heat transfer coefficient can be obtained as:

$$h_{sat} = \frac{Q_c}{\Delta T_{sat}} = \frac{Q_c}{T_w - T_{sat}} \quad [W.m^{-2}.K^{-1}] \quad (7)$$

The only way how to obtain coolant temperature behind the quench front is to calculate enthalpy increase caused by known areal heat flux behind the front in combination with measured inlet coolant temperature. Because the heat flux behind the front is relatively low, the temperature increase of water is also insignificant except the region right behind the front. The coolant temperature calculation is based on the idea, heat flux from the surface to water is very similar along the geometry, or it morphs linearly in the case of varying wall thickness. The upstream HTC was calculated using "cascade" algorithm. An example of resulting heat transfer coefficient map is in Figure 4.

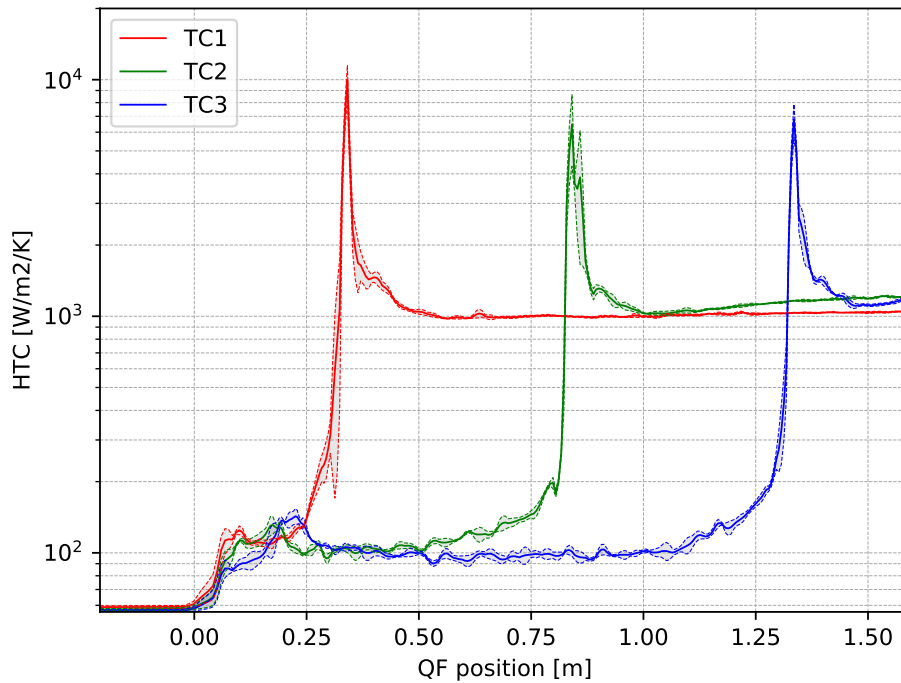


Figure 4: Map of HTC vs front position ( $T_0 = 600^\circ C$ , Model B)

# Results

## Pressure Peaks

The reflood process goes with significant pressure peaks while the coolant meets the hot surface. This sudden contact generates an enormous amount of vapor due to the momentum of the coolant. The effect is clearly detectable at initial wall temperature levels above  $400\text{ }^{\circ}\text{C}$ . This effect was already published and described by Stepanek (2016) [A3] for model A. In this study, the pressure peak rate is described also for models B and C.

The pressure peaks significantly influence pressure drop at the beginning of the reflood process and its value can easily exceed total pressure drop of fully reflooded test section. This can be a challenge for parallel channels, e.g. nuclear reactor core. The unevenness can lead to cooling water flowing around the central area of the core, where the surface temperature is lower, and the central area of the core can be cooled down distinctly later.

## Quenching Temperature

Quenching temperatures have been found via second order derivatives on the time-dependent temperature data. The quenching temperature represents a state at which re-establishment of solid-liquid contact is possible. After this point a high heat transfer rate takes place and thus the quenching temperature determination is the crucial part of the study. From this point of view, a higher quenching temperature (relative to initial one) means, that less accumulated heat needs to be removed in front of the quench front and thus the front can move faster along the surface.

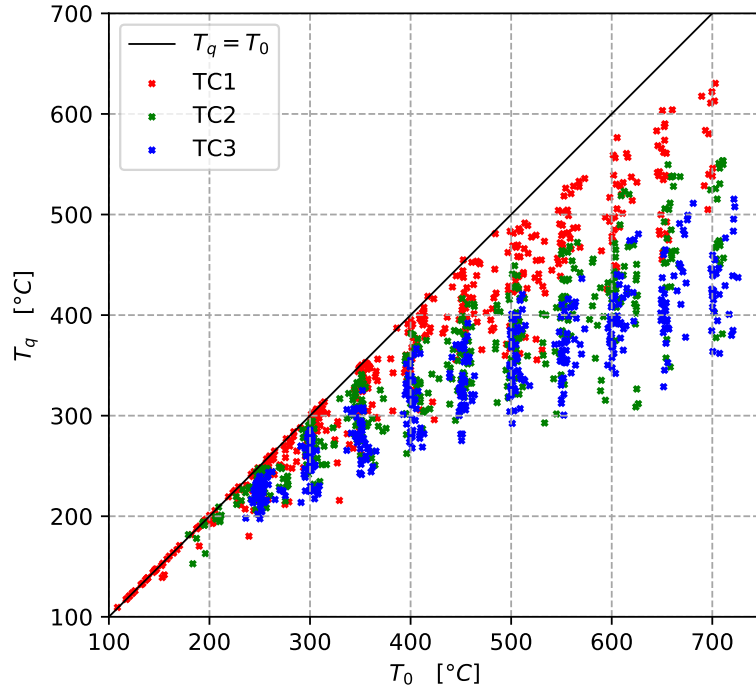


Figure 5: Quenching temperatures for all models and flow rates

All quenching temperature points are shown in Figure 5. Based just on this scatter plot, it can be concluded (regardless of flow rate), that with rising initial wall temperature the quenching temperatures are more deflected from the line, which represents the state where quenching temperature equals the initial wall temperature. Another insight is, that the deflection rate is also dependent on thermocouple's position which is obvious at first glance. It can be caused by more pre-cooling time further from the inlet. Points below temperature of  $T_0 = 200$  °C represent states for the model C, where no boiling at the advancing water surface was visible. In this case, the surface was rewetted instantly.

Quenching temperature was checked for dependency on material and global experimental parameters. The dependence on initial global parameters proved to be the reliable way how to correlate experimental data. Variables, which are not investigated e.g. inlet coolant temperature (remains almost constant through all experiments), are not present in correlation from this reason. Figure 6 shows the linear dependency of quenching temperature on resulting combination of initial experiment parameters.

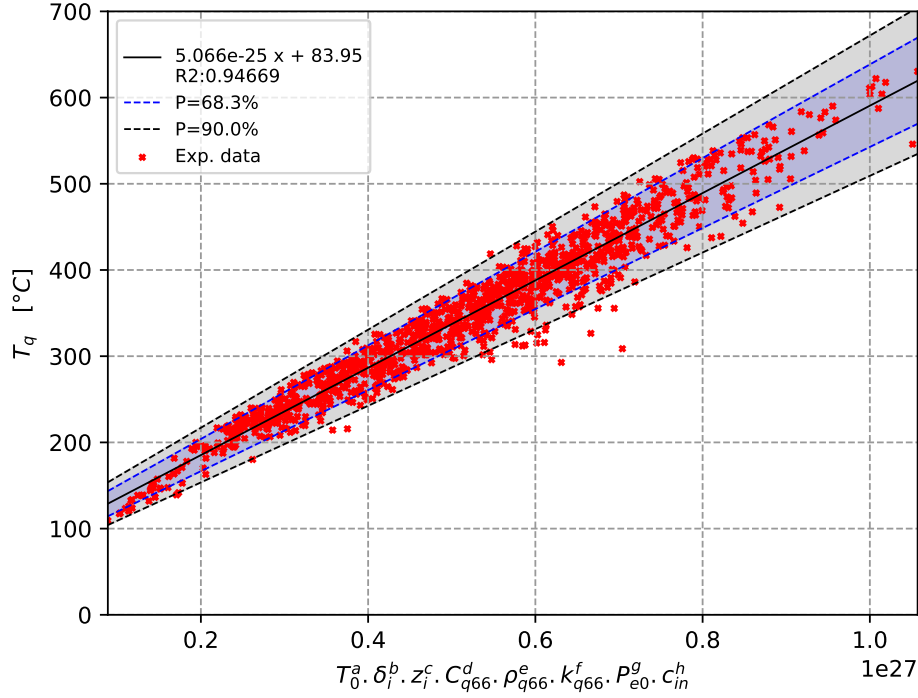


Figure 6: Quenching temperature data correlation

Material properties were included in the simplified form as parameters at 66% of initial wall temperature in order to avoid further temperature iteration and reflect its temperature dependency in reasonable form.

Equation ?? bellow solves quenching temperature for all measured models, all flow rates and all initial wall temperatures:

$$T_q = T_0^{2.01} \cdot \frac{[(5426e - 7)\rho_{q66}]^{7.44}}{C_{q66}^{1.73} k_{q66}^{2.48}} \cdot \frac{C_{in0}^{0.15}}{\delta_i^{0.15} z_i^{0.19} P_{e0i}^{0.024}} + 84 \quad [^{\circ}C] \quad (8)$$

## Heat Transfer Coefficients

Results from heat transfer calculation described in Section are presented in this section. These results contain dependence of heat transfer coefficient on given position, flow rate and initial wall temperature.

### Semi-Spatial View on HTC

Semi-spatial heat transfer coefficients were defined as a value of HTC, which depends on relative distance to quench front. The positive value of QF distance in presented results means a position in front of the front and negative otherwise. As it was mentioned in Section the coolant temperature up to the nucleate boiling point was taken as 100 °C and behind this point, the coolant temperature was calculated through "cascade" algorithm.

The thermocouple TC2 (position  $0.835 \text{ mm}$ ) for all three models with mass flux of  $80 \text{ kg.m}^{-2}.\text{s}^{-1}$  was taken as a representative sample of resulting data. At this low flow rate are clearly visible all common features of the time-spatial dependence of heat transfer coefficient.

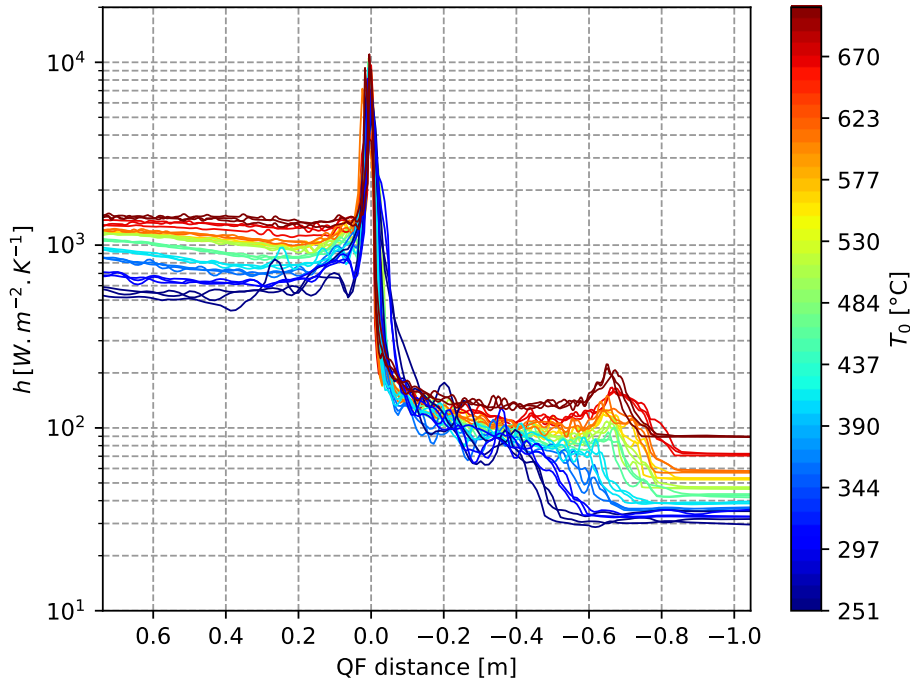


Figure 7: TC2: HTC vs distance to quench front (Model B,  $G = 80 \text{ kg.m}^{-2}.\text{s}^{-1}$ )

Figure 7 shows the dependence of HTC on distance to quench front (negative values represent positions in front of the QF and positive values otherwise) for middle thermocouple TC2.

## Relative Velocity

The first step was a definition of dimensionless quench front velocity. Very basic idea was to check quench front velocity as a portion of coolant inlet velocity as:

$$U_{13} = \frac{u_{13}}{c_{in13}} \quad [-] \quad (9)$$

where

$$c_{in13} = \frac{\int_{t_1}^{t_3} c_{in}(t) dt}{\Delta t_{13}} = \frac{\int_{t_1}^{t_3} \frac{G(t)}{\rho(t)} dt}{\Delta t_{13}} \quad [m.s^{-1}] \quad (10)$$

In Figure 8, there are plotted relative velocities for all coolant mass fluxes for Model B. A surprising finding is, that consideration of relative front velocity is not dependent on coolant flow rate.

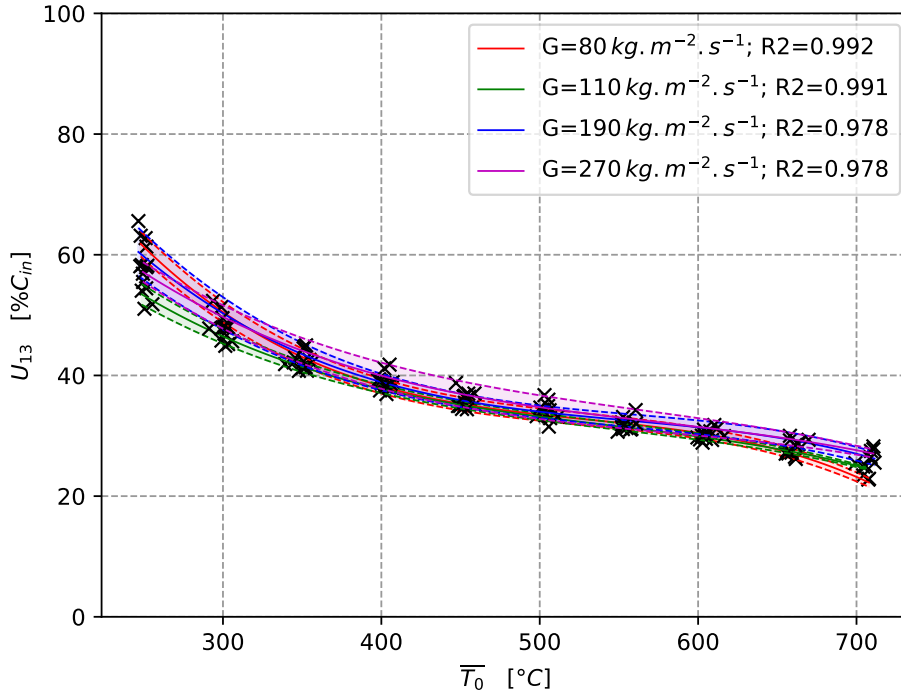


Figure 8: Mean relative quench front velocity for Model B

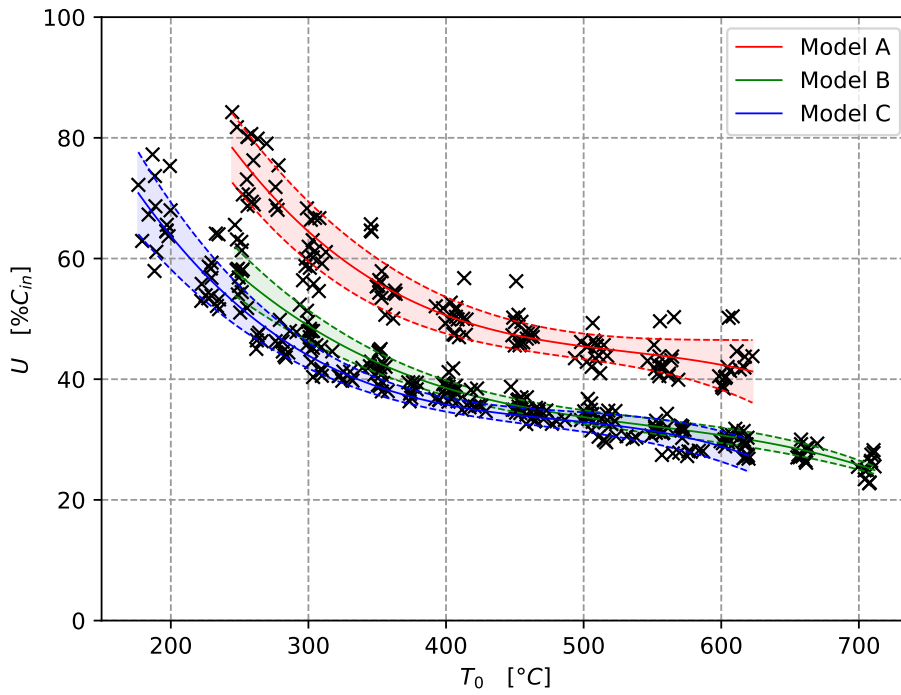


Figure 9: Mean relative quench front velocity for all models (TC1 to TC3)

## Quench Front Velocity Correlation

A quench front velocity correlation was developed based on presented experimental quench front velocity data from Section . Data shown in Figure 9 were correlated incorporating selected variables which are listed below. These variables were chosen through sensitivity analysis of quench front



velocity on various variables. The quench front velocity proved to be dependent on these three variables:

- Mean initial wall temperature  $\overline{T}_0$  [ $^{\circ}C$ ]
- Mean quenching temperature  $\overline{T}_q$  [ $^{\circ}C$ ]
- Mean characteristic length - Wall thickness  $\overline{\delta}$  [ $m$ ]

These variables were chosen based on "All-in-One" correlation test, where other variables were excluded as variables without influence or with negligible influence on the result. Moreover, these variables were chosen as averaged values from all three thermocouples. It must be noted, that most of the material properties are included in the quenching temperature definition which is one of the chosen variables and it is solved by Equation 8.

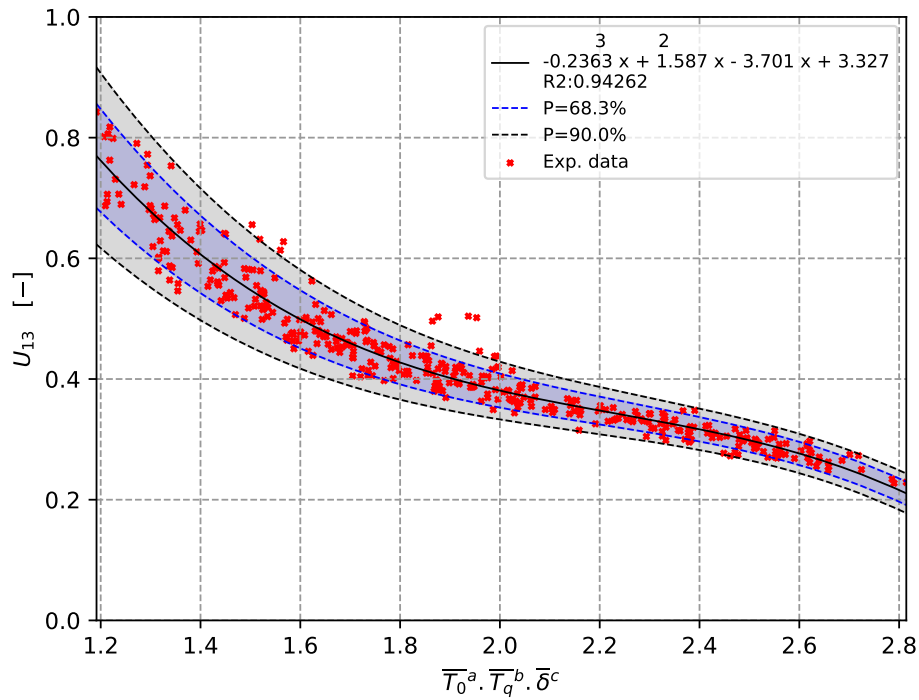


Figure 10: Correlated relative quench front velocity for all models (TC1 to TC3)

Figure 10 shows the resulting correlation between the chosen variables and relative quenching velocity. The correlation is shown in Equation 11 below:

$$U_{13} = -0.2363A^3 + 1.587A^2 - 3.701A + 3.327 \quad \text{where} \quad A = \overline{T}_0^{0.83} \left(\frac{\overline{\delta}}{\overline{T}_q}\right)^{1/3} \quad [-] \quad (11)$$

## Quench Front Velocity Validation

Quench front velocity correlation in final form was validated against experimental data. From this comparison an overall uncertainty is calculated and the result is compared with correlation by Saxena,

(2001)[22] (Equation 3, p. 7). The mentioned main equations for current study are recapitulated below:

$$\begin{aligned}
 u_{ab} &= U_{ab} \cdot c_{inab} \\
 U_{ab} &= -0.2363A^3 + 1.587A^2 - 3.701A + 3.327 \\
 A &= \bar{T}_0^{-0.83} \left( \frac{\bar{\delta}}{\bar{T}_q} \right)^{1/3} \\
 \bar{T}_q &= \frac{1}{n} \cdot \sum_{i=1}^n (T_0^{2.01} \cdot \frac{[(5426e - 7)\rho_{q66}]^{7.44}}{C_{q66}^{1.73} k_{q66}^{2.48}} \cdot \frac{c_{in0}^{0.15}}{\delta_i^{0.15} z_i^{0.19} P_{\epsilon 0i}^{0.024}} + 84)
 \end{aligned}$$

Results are plotted against experimental data and correlation by Saxena, (2001)[22] applied to acquired experimental data. The study by Saxena was evaluated on the experimental loop with annular flow channel with the internal heated tube with 1.5 mm wall thickness and with the outer diameter of 15 mm. The inner diameter of the outer barrier was 19 mm. The temperature range of the study was 200 – 500 °C. Its values were calculated using Equation 3 on page 7. As it is evident, the experimental loop has slightly different geometrical and thus hydraulic parameters, but it is one of several studies close enough with the experimental setup to the current study.

# Conclusion

The presented experimental study was focused on the rewetting phenomenon in the annular channel with bottom flooding configuration. Over 400 experiments were performed in order to obtain sufficient information about the process. Results showed the complexity of the process and helped to deepen knowledge of the investigated phenomenon.

Experiments were performed on three electrically heated models (tubes) at ten initial wall temperature levels (from 250 to 700 °C), moreover, four flow rate levels were included in these experiments.

The main goal was to collect experimental data and using this data to develop a set of correlations for the description of the flooding process. Based on individual results it can be concluded, that the goal was completely fulfilled with many additional findings.

Experimental data were processed completely with custom-built scripts. These scripts have several key features such as "cascade" algorithm for calculation of heat transfer coefficient behind the quench front, locating the quench front via the second-order derivative of temperature with node heat balance sub-routine. As an output of this basic data processing is converted raw data and over three thousands individual plots describing all relevant measured and calculated variable. On top of that, a superior autonomous batch processing script is capable to manage individually processed data and search through variables and its influence on the investigated parameter for the purpose of correlation development.

The most important results are solutions for quenching and nucleate boiling temperature, which is accompanied by correlations for heat transfer coefficients in the main individual points and regions along the flooded geometry.

Quenching temperature correlation is also main input for the solution of quench front velocity prediction. A surprising finding related to quench front velocity is, that relative front velocity to inlet velocity of the coolant is not dependent on actual flow rate. It follows that the percentage inlet velocity value can be calculated only through initial wall temperature, quenching temperature, and wall thickness. The quenching temperature is, therefore, most important and the most complex output of the study. The set of correlations developed in this study showed good agreement with acquired experimental data through all models and initial parameters.

In general, quench front velocity decreases with rising initial wall temperature and with more

accumulated heat (thicker wall) regardless of flow rate from the relative velocity point of view

Moreover increasing initial wall temperature and flow rate goes with significant pressure pikes. These pressure pikes strongly influence the hydraulic characteristic of the heated channel and this leads to a big pressure drop in the channel. These effects hand in hand with higher quench front velocity at low initial wall temperatures can lead to unevenness of water flow through the core during LOCA. The unevenness can lead to coolant flowing around the central area of the core, where the surface temperature is higher, and the central area of the core can be cooled down distinctly later. On the other hand, geometrical elements, such as spacer grids, create secondary quench fronts moving along the fuel rod. Due to these secondary quench fronts, the last rewetted point can't be clearly determined. In other words, the water presence above the core doesn't mean that the rest of the core is rewetted and cooled down. [A2]

As a side product of these results is the three-regional model which can be used as an input for further analytical studies in the field.

# Anotace

Jevy provázející smáčení vysoce rozehrátých povrchů jsou jedněmi z nejméně prozkoumaných jevů z oblasti přenosu tepla. Tyto procesy jsou jako takové známé již dlouhou dobu především díky tepelnému zpracování kovů (angl. quenching). V posledních několika dekádách se tento jev dostal do popředí zájmu především ve spojení s bezpečností jaderných reaktorů a kryogenní technikou. Smáčení je definované jako počátek prudkého poklesu teploty chlazeného objektu, který je způsobený kontaktem s relativně chladným chladivem, např. vodou. Přímý kontakt mezi rozehrátým povrchem a chladivem však není přímočarý proces. Pokud je teplota povrchu dostatečně vysoká, je kapalina oddělena od povrchu parní vrstvou, která brání přímému kontaktu povrch-chladivo. Tato parní vrstva se chová jako tepelně izolační vrstva a přestup tepla je tak značně snížen až do okamžiku, kdy teplota povrchu poklesne pod tzv. Leidenfrostovu teplotu, při které dojde ke kolapsu parní vrstvy. Vlastní hodnota této teploty je závislá na mnoha faktorech, jako například dynamice chladiva, počáteční teplotě povrchu, vlastnostech chlazeného objektu, geometrii, atd.

Za účelem prohloubení znalostí o tomto jevu bylo postaveno experimentální zařízení s anulárním průtočným kanálem. Testovací kanál má výšku přes  $1.7\text{ m}$  a je vybaven měnitelnými průchodem elektrického proudu vyhřívanými modely (trubkami) s vnějším průměrem  $9\text{ mm}$ . Tyto modely byly použity ve třech variantách: Model A s tloušťkou stěny ( $\delta$ )  $0.5\text{ mm}$ , Model B -  $\delta = 1.0\text{ mm}$  a Model C s proměnnou tloušťkou stěny po výšce. Každý model byl vystaven sérii experimentů se zaplavováním zdola na počátečních tepelných hladinách povrchu od  $250\text{ }^\circ\text{C}$  do  $700\text{ }^\circ\text{C}$  se čtyřmi různými průtoky chladiva  $80, 110, 190, 270\text{ kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$ .

Výsledky ukazují komplexní charakter daného jevu pro danou konfiguraci a parametry. Byly vytvořeny korelace pro všechny důležité body v procesu smáčení, tj. korelace pro teplotu smáčení, teplotu kritického tepelného toku a teplotu bublinkového varu. Další částí studie je návrh součinitelů přestupu tepla pro tří-oblastní model smáčení. Navíc jsou zde předloženy korelace pro lokální součinitele přestupu tepla. Poukázáno je ve studii také na efekt tlakových pulzací během zaplavování a jejich vliv na daný proces. Všechny tyto výsledky společně s naměřenými daty mohou sloužit jako vstup pro další studie zaměřené na tento jev, popřípadě pro bezpečnostní analýzy jaderných reaktorů.

# Summary

Quenching phenomenon is one of the most unexplored phenomena in the field of heat transfer. However, quenching is known for a long time especially due to metal hardening. In the several last decades, the phenomenon has come to the fore of interest in connection with nuclear reactor safety and cryogenic technologies. Quenching is defined as an onset of rapid temperature decrease within cooled geometry by relatively cold liquid. The contact between cooled surface and coolant is not a straightforward process. If the surface temperature is high enough, liquid can't touch the surface directly, but it is separated by stable vapor layer. The layer acts as a thermal insulation barrier and the resulting heat transfer is very limited until surface temperature falls below so-called Leidenfrost temperature. The actual value of quenching temperature is influenced by many factors such as dynamics of the coolant, initial wall temperature, material properties of cooled geometry, etc.

An experimental loop with an annular flow channel has been built in order to deepen the knowledge on the quenching phenomenon. The flow channel is vertical and it is equipped with changeable electrically heated tubes (models) with the outer diameter of  $9\text{ mm}$  and length over  $1.7\text{ m}$  in three different geometrical configurations: Model A - wall thickness ( $\delta$ )  $0.5\text{ mm}$ , Model B -  $\delta = 1.0\text{ mm}$  and Model C with variable wall thickness. Each model was exposed to bottom flooding at initial wall temperature levels from  $250^\circ\text{C}$  to  $700^\circ\text{C}$  and with four different coolant mass fluxes for each temperature level ( $80, 110, 190, 270\text{ kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$ ).

Results show the complexity of the phenomenon for given configurations and initial experiment parameters. Correlations for all important points in the process were developed i.e. correlations for quenching, critical and nucleate boiling temperature. Another part of the study is suggestion of heat transfer coefficients for the three-regional rewetting model. On top of that correlations for local heat transfer coefficients were proposed. There is also pointed out the effect of pressure pulsation during the flooding and its influence on the process. All these results together with raw experimental results can serve as an input for further investigations of the phenomena and for nuclear reactor safety analyses.

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