Transient Heat Transfer in an Out-of-Pile SCWR Fuel Assembly Test at Near-Critical Pressure

While supercritical water is a perfect coolant with excellent heat transfer, a temporary decrease of the system pressure to subcritical conditions, either during intended transients or by accident, can easily cause a boiling crisis with significantly higher cladding temperatures of the fuel assemblies. These conditions have been tested in an out-of-pile experiment with a bundle of four heated rods in the supercritical water multipurpose loop (SWAMUP) facility coconstructed by CGNPC and SJTU in China. Some of the transient tests have been simulated at KIT with a one-dimensional (1D) MATLAB code, assuming quasi-steady-state flow conditions, but time dependent temperatures in the fuel rods.

Heat transfer at supercritical and at near-critical conditions was modeled with a recent look-up table of Zahlen (2015, “Derivation of a Look-Up Table for Trans-Critical Heat Transfer in Water Cooled Tubes,” Ph.D. dissertation, University of Ottawa, Ottawa, ON, Canada.), and subcritical film boiling was modeled with the look-up table of Groeneveld et al. (2003, “A Look-Up Table for Fully Developed Film Boiling Heat Transfer,” Nucl. Eng. Des., 225(1), pp. 83–97.). Moreover, a conduction controlled rewetting process was included in the analyses, which is based on an analytical solution of Schulenberg and Raqué (2014, “Transient Heat Transfer During Depressurization From Supercritical Pressure,” Int. J. Heat Mass Transfer, 79(12), pp. 233–240.). The method could well reproduce the boiling crisis during depressurization from supercritical to subcritical pressure, including rewetting of the hot zone within some minutes, but the peak temperature was somewhat under-predicted. Tests with a lower heat flux, which did not cause such phenomena, could be predicted as well. In another test with increasing pressure, however, a boiling crisis was also observed at a heat flux, which was significantly lower than the critical heat flux (CHF) predicted by the CHF look-up table of Groeneveld et al. (2007, “The 2006 CHF Look-Up Table,” Nucl. Eng. Des., 237(15–17), pp. 1909–1922.). The paper is summarizing the physical models and the numerical approach. Comparison with experimental data is used to discuss the applicability of the method for the design of supercritical water-cooled reactors (SCWR). [DOI: 10.1115/1.4038061]

Model Assumptions

The transient tests to be modeled were considered as slow compared with the residence time of the fluid in the test section. The transient temperatures and pressures to be expected under normal and accidental conditions in collaboration with the Chinese project SCRIPT. One of the major contributions of SCRIPT was an out-of-pile qualification test of the test section, using electrically heated rods instead of fissile power. The test section was scaled-up by a factor 1.25, providing the same mass flux and heat flux as the planned in-pile test. The coolant supply system, however, was different: instead of a recuperator designed for the in-pile test, the supercritical water was preheated electrically and was supplied to the downcomer of the test section.

Several steady-state and transient tests have been performed in the SWAMUP supercritical water loop, out of which the transient tests with decreasing and increasing pressure, passing the critical pressure, have been selected for this paper to validate a one-dimensional (1D) model for transient heat transfer at near-critical pressure. The model shall be applicable to system codes like RELAP, ATHLET, or APROS, which will be required later for safety analyses of the high performance light water reactor.
simulations could thus be performed with a quasi-steady-state model, in which the one-dimensional balance equations of the fluid are modeled as steady-state equations, and only the energy balance equation of the heated rods is considered as transient. The pressure drop $\frac{\partial p}{\partial z}$ of the fluid passing the test section is thus described by the steady-state momentum equation

$$\frac{\partial p}{\partial z} = \frac{1}{A} \frac{\partial}{\partial z} \left( \frac{M^2}{A} \right) + \frac{P_w}{A} \tau_w + g \rho$$

(1)

The mass flow $M$ is constant then in the entire test section, equal to the inlet mass flow. The cross section $A$ and the wetted perimeter $P_w$ are taken as constant rods, the wires, and the inside of the square box into account. The density $\rho$ is determined by the IAPWS 97 steam table as a function of pressure and temperature under supercritical pressure conditions, and as a homogeneous mixture of water and steam in the two-phase range. The wall shear stress $\tau_w$ is determined by the pressure drop correlation of Müller-Steinhagen and Heck [5] for single and two-phase flow in pipes. As the flow is vertically upward, the acceleration of gravity $g$ is negative.

Neglecting heat which is transferred through the square box wall, the steady-state energy balance of the fluid yields a linear increase of specific enthalpy $h$ with height $z$ along the heated length, as the linear heat rate $q'$ is constant

$$\frac{\partial h}{\partial z} = \frac{q'}{M}$$

(2)

Gu et al. [4] estimated the heat losses through the box wall to be less than 8% in the test cases considered here, which changes the bulk temperature by less than 1.4 °C, justifying this simplification in a first-order approach.

The one-dimensional, time dependent heat balance equation of the heated rods can be written as

$$\rho_w c_{p,w} A_w \frac{\partial T_w}{\partial t} = \frac{\partial}{\partial z} \left( \frac{h_{w,A} A_w}{A} \frac{\partial T_w}{\partial z} \right) + q - q_w Ph$$

(3)

The four heated rods have been simplified here as solid rods with constant density $\rho_w$, specific heat $c_{p,w}$, total cross section $A_w$, and thermal conductivity $\lambda_w$. The wall temperature is averaged over the radius of the heated rods, changing with $z$ along the test section. In the one-dimensional approach taken here, $T_w$ equals the wall surface temperature. The radial temperature distribution inside rods is thus not yet included in the analyses. Heat is transferred to the fluid across the heated perimeter $P_h$ with a heat flux $q_w$ as

$$q_w = k(T_w - T_b)$$

(4)

The heat transfer coefficient $k$ is determined by different heat transfer correlations, depending on the flow regime, as will be described next. $T_b$ is the bulk fluid temperature.

Heat Transfer Correlations. The decision matrix for heat transfer correlations is shown in Fig. 3. At supercritical pressure, i.e., at a pressure $p > 22.064$ MPa, the heat transfer coefficient $k$ is determined by a recent transcritical look-up table (TCLT) of Zahn- lan [6], which is applicable in the pressure range from 19 to 30 MPa. At subcritical pressure, we need to differentiate between the following:

(a) Single-phase, subcooled heat transfer, if the wall temperature is less than the saturation temperature $T_{sat}$. The Dittus–Boelter equation is used in this case to determine the heat transfer coefficient.

(b) Two-phase nucleate boiling, if the wall temperature is greater than the saturation temperature, but less than the wall temperature $T_{CHF}$ at critical heat flux (CHF). It is modeled here using the Rohsenow equation [7], and a quadratic interpolation between single-phase flow and nucleate boiling is chosen to approximate subcooled boiling heat transfer.

(c) The critical heat flux $q_{CHF}$ is taken from the look-up table of Groeneveld et al. [8]. Knowing that the critical heat flux at critical pressure is zero, this look-up table can be extended to the critical pressure. The wall temperature at critical heat flux is determined as

$$T_{CHF} = T_{sat} + \frac{q_{CHF}}{k_{NB}}$$

(5)

d) The Leidenfrost temperature (also known as minimum film boiling temperature) is determined with a polynomial fit through measured data, as described by Schulenberg and Raqu [9].

e) If the wall temperature is greater than the Leidenfrost temperature, the heat transfer coefficient is taken from the

![Fig. 1 Planned test section of the in-pile SCWR fuel qualification test](image1)

![Fig. 2 Cross section of the electrically heated out-of-pile test in the SWAMUP facility](image2)
look-up table of Groeneveld et al. [10], if the pressure is less than 20 MPa, and from the transcritical look-up table of Zahlan [6], if the pressure is greater than 21.5 MPa. As both look-up tables do not match, a linear interpolation is used in case of film boiling at pressures between 20 and 21.5 MPa.

(f) Transition boiling is expected, if the wall temperature is greater than the temperature at critical heat flux, but less than the Leidenfrost temperature. The heat flux in this range is linearly interpolated between nucleate boiling at critical heat flux and film boiling at Leidenfrost temperature in a log–log Nukiyama diagram, Fig. 4.

A Nukiyama diagram at near-critical pressure, which can be constructed from these heat transfer correlations, is shown exemplarily in Fig. 4 for a mass flux $G$ of 1410 kg/m$^2$ s, a bulk temperature $T_b$ of 350°C at supercritical pressure, and assuming saturated liquid conditions at subcritical pressure. Three different pressure levels have been assumed, demonstrating that the continuous increase of heat flux with wall superheat temperature at supercritical pressure turns into a step function at subcritical pressure. Thus, depending on the wall temperature at supercritical pressure, a transition to subcritical pressure can either produce local film boiling or local nucleate boiling.

The difference between data of the transcritical look-up table and the film boiling look-up table (FBLT) is illustrated in Fig. 5 for a wall superheat of 100°C. At 20 MPa, where both look-up tables should be applicable, the heat transfer coefficient of the film boiling look-up table increases strongly with decreasing subcooled bulk enthalpy, which is neither supported by experimental data nor by the transcritical look-up table at 20 MPa. As such increase is doubtful, the heat transfer at subcooled film boiling ($x < 0$) is modeled here taking the heat transfer coefficient at saturated film boiling ($x = 0$) as constant. More experimental data at subcooled bulk temperature and near-critical pressure will be needed, however, to correct the film boiling look-up table reliably.

![Diagram](https://example.com/diagram.png)

**Fig. 3** Decision matrix for heat transfer correlations

**Fig. 4** Nukiyama diagram of heat transfer at near-critical pressure

**Fig. 5** Film boiling heat transfer predicted with the FBLT [10] and with the TCLT [6]; mass flux 1500 kg/m$^2$ s, tube diameter 8 mm
Conduction Controlled Rewetting. If parts of the heater rods are wetted, transferring heat by subcooled or saturated nucleate boiling, and other parts are dry, transferring heat by film boiling, we can observe a slow rewetting process of the dry zone, which is known as conduction controlled rewetting. Heat is transferred axially inside the rods from the hotter, dry zone to the colder, wetted zone, and, as soon as part of the dry zone is cooled down below the Leidenfrost temperature, this part is rewetted. The quench front, i.e., the point which separates the wetted zone from the dry zone, is thus moving slowly downstream until the entire heated rod is rewetted.

The process can be simulated by solving Eq. (3), using piecewise constant heat transfer coefficients in the dry and in the wetted zone, respectively. Schulenberg and Raqué [9] were solving this equation analytically. As a result, the velocity $U$ of the quench front can be determined as

$$ U = \kappa \sqrt{\frac{k_{wet} P_r}{\rho W A_W} U^*} $$

(6)

Here, $\kappa$ is the thermal diffusivity of the rod and $k_{wet}$ is the heat transfer coefficient in the wetted zone. The dimensionless quench front velocity $U^*$ is predicted as

$$ U^* = \frac{\theta^2 - \varepsilon}{\sqrt{\theta(1 + \theta)(\theta + \varepsilon)}} $$

(7)

The temperature ratio $\theta$ is given by the temperature differences far upstream and downstream of the quench front to the Leidenfrost temperature, as illustrated in Fig. 6

$$ \theta = \frac{\Delta T_{wet}}{\Delta T_{dry}} $$

(8)

The heat transfer ratio $\varepsilon$ is defined by the piecewise constant heat transfer coefficients $k_{dry}$ and $k_{wet}$ downstream and upstream of the quench front, respectively,

$$ \varepsilon = \frac{k_{dry}}{k_{wet}} $$

(9)

The method has been validated by Schulenberg and Raqué using transient heat transfer data of Köhler and Hein [11], measured at a uniformly heated tube of 14 mm inner diameter.

Using this analytical solution allows a significantly larger mesh size in system codes. The nodal point downstream of the quench front is taken to determine $k_{dry}$ and $\Delta T_{dry}$, and the nodal point upstream the quench front is taken to determine $k_{wet}$ and $\Delta T_{wet}$. The initial appearance of dry and wetted heat transfer on the rod surface is taken to determine the initial quench front position, which is moved to the next downstream mesh as soon as the quench front was running over the downstream nodal point. The analytically determined exponential wall temperature functions, shown in Fig. 6, are simplified thus as a step function of wall temperature.

Numerical Procedure. These equations have been implemented into a simple MATLAB code at KIT, using the software XSTEAM for MATLAB [12] to calculate water and steam properties according to the IAPWS IF 97 standard. The heated length of 75 cm has been discretized with 25 equidistant nodal points with a specified pressure and specified temperature at the inlet and a specified mass flow at the outlet. Upwind differences have been used to approximate the space derivatives in Eqs. (1) and (2) and central differences in Eq. (3). The transient tests were simulated with a time-step of 0.02 s.

The initial pressure, coolant, and wall temperature distribution was calculated with a transient analysis, assuming a constant pressure and temperature distribution at the beginning. Once pressures and temperatures had reached a steady-state, the pressure and temperature distribution were stored and the simulation was started with these data as initial conditions. Fluid properties were updated each time-step.

Transient Heat Transfer Tests

The SWAMUP test facility is sketched in Fig. 7. Two plunger pumps produce the intended mass flow through the test section, which is measured with two venturi flow meters. After passing a recuperator (reheater), the water to the test section is heated-up in an electrical preheater, controlled by a thermocouple at the inlet of the test section. Downstream the test section, the hot fluid is first cooled down in the recuperator and then in a cooler, before a pressure control valve at the outlet of the loop is reducing the pressure again. It is controlled by the pressure at the inlet of the test section. More details of the supercritical water loop have been described by Gu et al. [4] and Li et al. [13].

The four rods of the test section are designed as tubes with an outer diameter of 10 mm and a wall thickness of 1 mm, made of Inconel 718. They are heated internally with DC power over a length of 750 mm. A wire with 1.65 mm diameter is wrapped (clockwise in flow direction) around the rods with an axial pitch of 250 mm, serving as spacer and mixer. Each rod has been equipped with eight thermocouples at 31.25 mm, 62.5 mm, 125 mm, 250 mm, 375 mm, 500 mm, 625 mm, and 750 mm, measured from the inlet of the heated section. Measured wall temperatures are corrected to the tube surface. The square box around the rod bundle is made of SS 304 with a wall thickness of 2 mm. More details of the test section have been described by Gu et al. [4].

Four transient test with decreasing (D) or increasing (I) pressure between 17 and 25 MPa, which had been performed with this facility, have been selected to validate the model described here. As listed in Table 1, case 1-D and case 1-I were performed with a higher heat flux of around 640 kW/m² and with a slower pressure gradient of 1 MPa/min, whereas case 2-D and case 2-I were performed with a lower heat flux of around 430 kW/m² but with a faster pressure gradient of 2 MPa/min. A constant mass flux of 1410 kg/m² s was used in all cases. It was measured with an accuracy of ±0.5% and the mass flux data scattered accordingly during the tests. As the inlet temperature of the test section can only be controlled under subcooled conditions, the inlet temperature setpoint, shown in Table 1, was always lower than the saturation temperature at 17 MPa. During the tests, the temperature deviation from this setpoint was up to ±2°C.
Wall temperatures were measured with sheathed N-type thermocouples with an accuracy of ±0.2 °C. The DC power applied to the test section was measured by a voltmeter shunt with an accuracy of ±0.1% and a current shunt with an accuracy of ±0.2%. The pressure at the inlet of the test section was measured by a Yokogawa EJA-150A capacitance-type pressure transducer with an accuracy of ±0.2%.

Results

The predicted rod surface temperatures for the transient test 1-D, with decreasing pressure and higher heat flux, is shown in Figure 8. Table 1 summarizes the parameters of the transient tests performed in the SWAMUP test facility.

Table 1 Parameters of transient tests performed in the SWAMUP test facility

<table>
<thead>
<tr>
<th>Case</th>
<th>Initial pressure (MPa)</th>
<th>Final pressure (MPa)</th>
<th>Pressure gradient (MPa/min)</th>
<th>Mass flux (kg/m² s)</th>
<th>Heat flux (kW/m²)</th>
<th>Inlet temp. (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-D</td>
<td>25</td>
<td>17</td>
<td>−1</td>
<td>1410</td>
<td>639</td>
<td>345</td>
</tr>
<tr>
<td>1-I</td>
<td>17</td>
<td>25</td>
<td>+1</td>
<td>1410</td>
<td>640</td>
<td>350</td>
</tr>
<tr>
<td>2-D</td>
<td>25</td>
<td>17</td>
<td>−2</td>
<td>1410</td>
<td>429</td>
<td>345</td>
</tr>
<tr>
<td>2-I</td>
<td>17</td>
<td>25</td>
<td>+2</td>
<td>1410</td>
<td>427</td>
<td>345</td>
</tr>
</tbody>
</table>

Fig. 7 Sketch of the SWAMUP supercritical water loop [13]

Fig. 8 Predicted rod surface temperatures for test case 1-D

Fig. 9 Comparison of predicted with measured rod surface temperatures, case 1-D

by a Yokogawa EJA-150A capacitance-type pressure transducer with an accuracy of ±0.2%.
The test starts with a supercritical pressure of 25 MPa and the pressure is reduced linearly with \(-1\) MPa/min. While the pressure is supercritical, the rod wall is well cooled, staying just \(\sim 10^\circ C\) hotter than the bulk temperature. Once the critical pressure has been passed, however, the inlet of the test section, which was colder than the Leidenfrost temperature of 374 \(^\circ C\) at critical pressure, is wetted, whereas the rear part of the test section is hotter than the Leidenfrost temperature. It can thus not be wetted and changes to subcooled film boiling at subcritical pressure. With further decreasing pressure, the film boiling heat transfer deteriorates. The quench front between subcooled nucleate boiling and subcooled film boiling moves slowly downstream, driven by the conduction controlled rewetting process described above, until the entire surface is quenched after 300 s.

A comparison of predicted data (solid lines) with the average of the four measured surface temperatures at three axial positions, dashed lines in Fig. 9, shows that the surface temperature at supercritical pressure is somewhat over-predicted, which is also due to the missing heat transfer to the downcomer to a small extent. The surface temperature increase during the first 100 s after passing the critical pressure agrees well with the measured data, as well as the time needed for the rewetting process. The peak temperature of 550 \(^\circ C\), which was measured at around 500 s, is still underpredicted, which means that the film boiling heat transfer below 20 MPa is over-predicted.

The measured temperature drop during quenching appears by far not as abrupt as predicted. This is not a weakness of the model, but simply due to averaging the measured data. As Fig. 10 shows exemplarily for the axial position at 0.5 m, the four rods did not quench simultaneously, but with a time delay up to 40 s. The average temperature, shown with a dashed line in Fig. 10, is thus decreasing stepwise in the time interval between 490 and 530 s. The one-dimensional model prediction (bold line), which does not differ between individual rods, is quenching abruptly instead.

At the lower heat flux of case 2-D, all rods have been colder than the Leidenfrost temperature on the entire rod surface, and film boiling is neither predicted nor measured, shown in Fig. 11. A small but unimportant temperature peak predicted right before reaching the critical pressure is due to a weakness of the transcritical look-up table of Zahlan [6]. Again, the surface temperature is mostly somewhat over-predicted.

In case of increasing pressure, starting from subcritical pressure, film boiling can only occur if the CHF has been exceeded. Test case 1-I shows surprisingly in Fig. 12, however, a temperature peak between 20.5 and 22 MPa, which is obviously due to departure from nucleate boiling. Such effect is not predicted at 640 kW/m\(^2\) by the CHF look-up table of Groeneveld et al. [8] and thus also not predicted by our model. Instead, a small peak right before reaching the critical pressure is predicted here, which is due to the linear interpolation between CHF data at 21 MPa, the highest tabulated data points, and zero at critical pressure.
should never be depressurized to subcritical pressure as long as surface temperatures, instead, show a smooth transition from subcritical to supercritical conditions.

Conclusions

As a direct conclusion from these transient tests, an SCWR should never be depressurized to subcritical pressure as long as the peak heat flux is greater than 400 kW/m². Otherwise, local film boiling can become a concern for the fuel rods.

The physical phenomena occurring during pressure variations across the critical pressure have been modeled here primarily based on look-up tables. This method is fast and usually quite accurate, but look-up tables can only be as accurate as their physical database. Beyond 20 MPa up to the critical pressure, however, this database is still very poor, and we recommend further transient and steady-state tests to update the CHF look-up table as well as the film boiling look-up table.

The new transcritical look-up table turned out to be quite successful at supercritical pressure, but it interpolates between wetted and dry surfaces at subcritical pressure, whereas both cases produce quite different heat transfer coefficients in reality. Therefore, this look-up table should only be taken at subcritical pressure to run smoothly over the critical pressure, and we need to differentiate clearly between wetted and dry surfaces otherwise, as described earlier.

The conduction controlled rewetting process at subcritical pressure can be modeled numerically with a transient rod temperature analysis. An analytical solution of this heat balance equation of the rod at the quench front, however, can help to avoid fine discretization of the rod, saving a lot run time of the code.

We recommend to imply the method described here in system codes like RELAP, ATHLET, or APROS to model the phenomena during depressurization from supercritical to subcritical pressure in more detail. Such codes will enable them to simulate the tests more realistically, including radial temperature gradients inside the rods and heat transfer to the downcomer. The method should also be applicable to subchannel codes such that temperature differences inside the flow cross section of the bundle become predictable.

Nomenclature

- \( A \) = tube cross section, m²
- \( A_w \) = wall cross section, m²
- \( c_{p,w} \) = wall specific heat, J/kg K
- \( d_H \) = hydraulic diameter, m
- \( g \) = acceleration of gravity, m/s²
- \( G \) = mass flux, kg/m² s
- \( h \) = spec. enthalpy, J/kg K
- \( k \) = heat transfer coefficient, W/m² K
- \( k_{fb} \) = heat transfer coeff. at film boiling, W/m² K
- \( k_{nb} \) = heat transfer coefficient at nucleate boiling, W/m² K
- \( k_{wet} \) = heat transfer coeff. at wetted surface, W/m² K
- \( M \) = mass flow rate, kg/s
- \( p \) = pressure, Pa
- \( P_H \) = heated perimeter, m
- \( P_W \) = wetted perimeter, m
- \( q_W \) = heat flux to fluid, W/m²
- \( q_{CHF} \) = critical heat flux, W/m²
- \( q \) = linear heat rate, W/m
- \( t \) = time, s
- \( T_b \) = bulk temperature, °C
- \( T_w \) = wall surface temperature, °C
- \( T_{CHF} \) = wall temperature approaching critical heat flux, °C
- \( T_{LF} \) = Leidenfrost temperature, °C
- \( T_{sat} \) = saturation temperature, °C
- \( U \) = quench front velocity, m/s
- \( U^* \) = dimensionless quench front velocity
- \( x \) = steam quality
- \( z \) = axial coordinate, m
- \( \Delta T_{dry} \) = temp. diff. to \( T_{LF} \) far downstream the quench front, °C
- \( \Delta T_{wet} \) = temp. diff. to \( T_{LF} \) far upstream the quench front, °C
- \( \varepsilon \) = ratio of heat transfer coefficients
- \( \theta \) = ratio of temperature differences
- \( \lambda_W \) = wall thermal conductivity, W/m K
- \( \rho \) = density, kg/m³
- \( \rho_W \) = wall density, kg/m³
- \( \tau_W \) = wall shear stress, N/m²

References