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Flow morphology and heat transfer analysis during high-pressure steam condensation in an inclined tube part I: Experimental investigations

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Abstract

In this paper, experimental investigations on the flow morphology and heat transfer in a single steam condenser tube are presented, which were performed at the thermal hydraulic test facility COSMEA (COndensation test rig for flow MOrchology and hEAt transfer studies). This facility has been setup to study the interrelation of condensation heat transfer with two-phase flow in an isolated single condenser tube that is cooled by forced convection. Studies have been performed for elevated pressures up to 65 bar at saturation conditions and for inlet steam mass flow of up to 1 kg/s and different inlet steam qualities. The wall heat flux is measured with distributed heat flux probe and global condensation rates have been obtained from integral heat and mass balances. As a unique feature the cross-sectional phase distribution has been studied via X-ray computed tomography. The data is going to be used for the validation of numerical simulations with 1D ATHLET and 3D CFD codes as presented in the second part of this paper.

Key words: condensation heat transfer, heat exchangers, two-phase flow, heat flux, X-ray tomography, passive safety systems

Nomenclature

A	Inner cross-sectional area [m ²]	\dot{m}	Mass flux [kg/s]
b	Length of the flume cross-section arc	N	Number of pixels
C	Contrast [-]	\dot{q}	Heat flux [W/m ²]
d	Diameter [m]	Q	Transferred heat [J]
g	Gravitational acceleration [m/s ²]	r	Radius [m]
h	Heat transfer [W/m ² /K]	Re	Reynolds number
H	Enthalpy [kJ/kg]	t	Time [s]
H'	Enthalpy of saturated water [kJ/kg]	T	Temperature [°C]
H''	Enthalpy of saturated steam [kJ/kg]	u	Velocity [m/s]
j	Superficial velocity [m/s]	\dot{V}	Volumetric flow rate [m ³ /h]
l	Length [m]	w	Weight function [-]
L	Level height (condensate) [m]	x	Steam mass fraction [-]

Subscripts

1	Large part of the separator vessel	ins	Thermal insulation
2	Small part of the separator vessel	l	Liquid
c	Condensate	o	Outer, outlet
cw	Cooling water	p	Primary side, partially
cy	Cylindrical part of the separator	r	flume
exp	Experiment	s	Steam
fw	Feed into the cooling water loop	sep	Separator vessel
g	Gas	th	Torospherical head of the separator
hl	Heat loss	w	Wall
Hu	Heat-up	x	Coordinate
hy	Hydraulic	y	Coordinate
i	Inner, inlet		

Greek symbols

α	Liquid fraction [-]	ν	Kinematic viscosity
δ	Wall thickness [m]	λ	Thermal conductivity [W/(mK)]
ε	Void fraction [-]	ρ	Density [kg/m ³]
μ	Linear attenuation coefficient [1/m]		

Abbreviations

ATHLET	Analysis of THERmal-hydraulics of LEaks and Transients
ATLAS	Advanced Thermal-Hydraulic Test Loop for Accident Simulation
APR	Advanced Power Reactor
CCC	Containment Cooling Condenser
CFD	Computational Fluid Dynamics
CFP	Core Flooding Pool
COSMEA	Condensation test rig for flow morphology and heat transfer study
CS	Cross-Section
CT	Computed Tomography
EC	Emergency Condenser
FI	Mass Flow Indication
GENEVA	GENERIC investigations on passive heat remoVAL systems
HFP	Heat Flux Probe
HUSTLE	Hitachi Utility Steam Test Leading facility
INKA	INtegral Test Facility KARlstein
INVEP	Invert Edward Pipe
KAERI	Korea Atomic Energy Research Institute
KONWAR	Ger.: KONdensation in WAagerechten Rohren (condensation inside horizontal tubes)

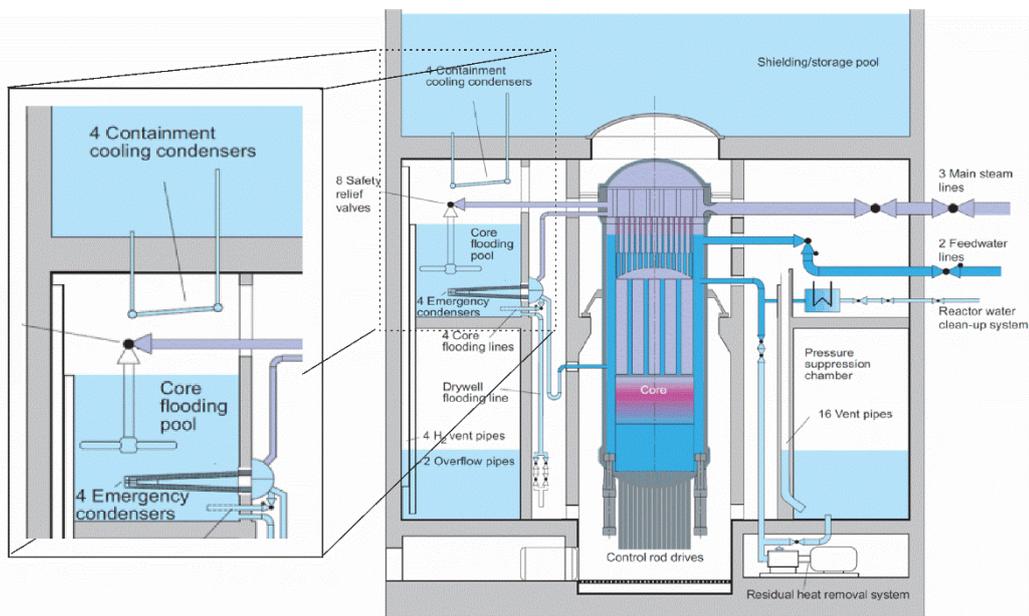
NOKO	Ger.: NOTKondensator (emergency condenser)
LI	Liquid Level Indication
LOCA	Loss Of Coolant Accident
MTF	Modulation Transfer Function
OPC	Open Platform Communications
PAFS	Passive Auxiliary Feedwater System
PASCAL	PAFS Condensing Heat Removal Assessment Loop
PI	Pressure Indication
PPPT	Passive Pressure Pulse Transmitter
RPV	Reactor Pressure Vessel
SCL	Stratified Condensate Level
SWR	Ger.: Siedewasser-Reaktor (Boiling Water Reactor)
SETCOM	Separate Effect Test for COndensation Modeling
TDI	Temperature Difference Indication
TE	Ger.: Thermoelement (Thermocouple)
TI	Temperature Indication
TOMO	Tomographic imaging plane
TOPFLOW	Transient two Phase FLOW test facility

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31 **1. Introduction**

32 Today, safety systems in nuclear power plants do mostly rely on active components. For the removal
33 of decay heat after a reactor shutdown, for instance, coolant circulation in the primary circuit is
34 sustained via pumps. This is, however, a potential safety risk in case of a station black-out as it was
35 experienced in 2011 at the Fukushima Daiichi Nuclear Power Plant. Therefore, future nuclear power
36 plant designs shall utilize passive safety systems which are independent from electrical power [1]-[7].
37 One of these improved reactor concepts is the boiling water reactor KERENA™ that combines a new
38 passive safety control strategy with the advantages of a Generation III+ nuclear reactor design [2] (see
39 Figure 1). Besides being a very economic design due to its simple operational concept and
40 configuration it comprises several passive safety systems such as a passive pressure pulse transmitter
41 for thermal-hydraulic actuation, core-flooding lines, drywell flooding lines, pressure suppression and
42 venting systems for pressure reduction and hydrogen blow off, and a fully passive heat removal chain
43 including an emergency condenser and a containment cooling condenser for transferring heat from
44 the reactor pressure vessel to the storage pool outside the containment. Additionally, large water
45 volumes are provided, such as a core flooding pool (CFP), a pressure suppression pool and a storage
46 pool actuating as passive heat sink for maximal three days.



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Figure 1: KERENA™ reactor concept with passive safety systems (Stosic et al, 2008) [2].

50 The basic design and function of the KERENA heat removal chain is illustrated in Figure 1. There are
51 four emergency condensers (EC) which are hydraulically connected to the reactor pressure vessel
52 (RPV) via a steam line (top) and a condensate return line (bottom). Note, that Figure 1 shows only one

53 of four ECs. Each EC is located inside the core flooding pool and is made of 61 lying U-shaped condenser
54 tubes [8]. As the EC is directly connected with the RPV it is filled with the primary circuit water during
55 reactor operation. As there is no circulation in this circuit the water temperature is close to that of the
56 CFP.

57 In a loss of coolant accident (LOCA), such as a break of a main steam line or any other leakage, the
58 resulting pressure loss leads to steam production in the primary circuit. The steam accumulates at the
59 top of the RPV and the liquid filling level in the RPV and the connected circuit decreases. At some point
60 high pressure steam enters the ECs and is there condensed. This starts a circulation in the EC loops,
61 which is driven by steam production in the RPV and the steam condensation in the EC. Thus, a passive
62 heat removal circuit is sustained. After some time the water in the CFP becomes saturated and starts
63 boiling. At that moment four so-called containment cooling condensers (CCCs) above the core flooding
64 pool start their action and transfer heat across the containment barrier. They are connected at their
65 secondary side to the storage pool, whose water is at room temperature at the beginning of the CFP
66 cooling and remains below 100 °C during emergency cooling. The subcooled liquid of the storage pool
67 enters the CCCs while steam rising from the flooding pool is condensed on the outside of the CCC
68 tubes. Hence, the water in the CCC evaporates and this drives another passive heat transfer to the
69 storage pool.

70 Nuclear safety assessments that involve thermal hydraulics are today carried out with system codes
71 such as AC² ATHLET, RELAP, and TRACE. While these codes have been well qualified for active hydraulic
72 circuits there is still a need to qualify them for passive decay heat removal systems. For that,
73 experimental data is needed. In this paper we report on an experimental studies on the condensation
74 process inside an inclined tube under operation condition of an emergency condensers. They have
75 been carried out at the COSMEA (COndenSation test rig for flow Morphology and hEA transfer studies)
76 which is part of the TOPFLOW facility at Helmholtz-Zentrum Dresden Rossendorf (HZDR). We studied
77 global and local heat transfer as well as flow morphology for different pressures up to 65 bar and
78 different inlet qualities.

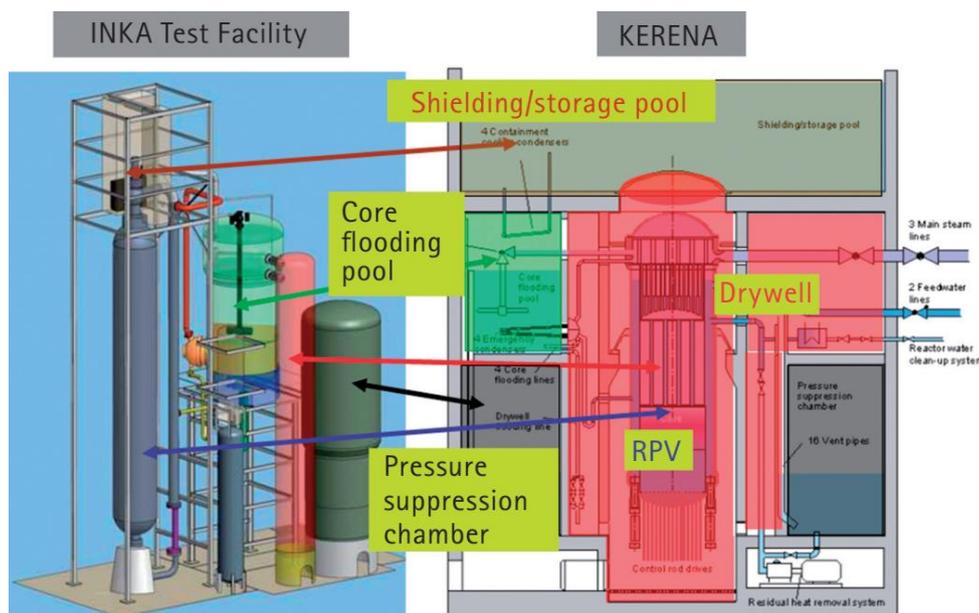
79 **2. Past experimental activities with respect to KERENA passive heat removal** 80 **systems**

81 In the recent past a number of experimental investigations have already been performed with respect
82 to the local and integral behaviour of the KERENA passive cooling circuit components. They will be
83 briefly discussed in the following. The INtegral test facility KArlstein (INKA) at Framatome GmbH [9]
84 was designed to experimentally investigate the passive safety systems of the KERENA reactor on an
85 integral scale (see Figure 2). Amongst others the facility comprises large vessels representing the

86 KERENA containment and the large water volumes of the storage pool, the core flooding pool and the
 87 pressure suppression chamber and one of the four heat removal systems (EC, CCC). The pressure vessel
 88 is designed for operating pressures up to 160 bar and a maximum power of 22 MW. The whole facility
 89 is equipped with more than 300 sensors for temperature, mass flow, absolute and differential pressure
 90 as well as two-phase distribution. INKA provides a 1:1 height and a 1:24 volumetric scaling. In INKA
 91 there is also a number of smaller KERENA components integrated for testing. These are the passive
 92 pressure pulse transmitter (PPPT), vent pipes and core flooding lines are modelled too. The PPPT is a
 93 passive switching device that is used to directly initiate reactor shut down, containment isolation at
 94 the main steam line penetrations and automatic depressurization of the reactor pressure vessel (RPV).
 95 The vent pipes are designed for limiting the pressure increase in the containment to maximal 4 bar by
 96 steam release to and condensation in the pressure suppression chamber. The core flooding lines are
 97 used to inject additional cooling water from the CFP into the RPV by means of gravity flow at low-
 98 pressure condition. The more interested reader may be referred to [9].

99 The INKA facility is used to assess the integral and component behaviour for different accident
 100 scenarios, such as steam-line breaks, RPV bottom leaks, or station black-out. From the extensive
 101 instrumentation it is possible to extract some information on single effects, such as steam flow and
 102 condensate level in EC single tubes, influence of non-condensable gases on the heat transfer inside the
 103 EC, two-phase instabilities in the CCC, as well as effects of natural convection on temperature
 104 stratification and gas composition in vessels. However, for a very detailed thermal hydraulic analysis
 105 that involves local flow and heat transfer conditions the instrumentation density is too low.

106



107

108

Figure 2 INKA test facility [9] regarding the KERENA reactor concept [2].

109 Single effect studies for the KERENA passive safety systems are being or were carried out at the
110 facilities described in the following. The GENEVA test facility (GENEric investigations of passive heat
111 remoVAI systems) is dedicated to single effect studies in the CCC [15]. It consists of four or fewer
112 condenser tubes in a steam chamber which are connected to an upper reservoir with a height scaling
113 equal to KERENA. The steam chamber emulates flooding pool conditions. Steam is fed into the
114 chamber from a 120 kW evaporator via eight equally distributed nozzles and there slowed down by
115 baffle plates. While the steam condenses at the condenser tubes, heat is being transferred to the inside
116 where boiling sets in. The resulting density changes create a natural upward flow in the riser tube. In
117 the downcomer tube sub-cooled liquid flows back into the condenser tubes what closes the natural
118 circulation. Experiments are being carried out to investigate natural convection and flow instabilities,
119 such as flashing and geysering, in detail. An upgrade of GENEVA was presented in 2017 by Viereckl et
120 al. [16] applying advanced measurement technique such as multipoint level sensors for a better
121 resolution of the two-phase flow structure inside the pipes as well as improved instrumentation to get
122 axial and circumferential temperature profiles and condensation rates in the steam chamber. The
123 whole experimental program is accompanied by system code analyses.

124 Until some years ago the so called NOKO facility (Ger.: NOtKOndensator) was operated at
125 Forschungszentrum Jülich. It has been designed to investigate the effectiveness of the emergency
126 condensers of the SWR 1000 reactor design [18], which is very similar to the KERENA one.
127 Investigations were carried out for an eight tube EC bundle having original materials and geometry of
128 the SWR 1000 design. The EC was submerged in a large tank and supplied with 10 MPa steam from an
129 electro boiler. Intensive instrumentation with thermocouples and void probes allowed transient
130 measurement of liquid distribution and condensation heat transfer on the primary side for different
131 experimental scenarios, among them such with non-condensable gases. The emergency condenser
132 power has been correlated to system pressure, condensation level and concentration of non-
133 condensable gases. Based on the results, the numerical system code ATHLET was expanded by the so-
134 called KONWAR heat transfer model, adding the ability to calculate condensation heat transfer in
135 inclined tubes [19]. In 2001, the NOKO facility was dismantled and parts of it transferred to the
136 TOPFLOW facility (Transient TwO Phase FLOW Test Facility) at Forschungszentrum Rossendorf (now
137 Helmholtz-Zentrum Dresden-Rossendorf) [20]-[22]. At TOFLOW further studies on the role of
138 secondary side boundary and flow conditions were carried out and accompanied by computational
139 fluid dynamics (CFD) modelling for large vessels [23]-[29]. The level of detail on the primary side
140 condensation obtained at NOKO and TOPFLOW was still rather low. Only information about axial
141 temperature profiles could be obtained with multiple thermocouples and axial phase indicator
142 distributions with multiple void probes. Therefore, we recently commissioned the thermal hydraulic

143 test facility COSMEA at TOPFLOW to study the high-pressure steam condensation in a single inclined
144 tube.

145 Eventually, we shall just briefly note that there are numerous other facilities in the context of nuclear
146 safety research which address similar problems of condensation heat transfer. Among them are the
147 LAOKOON facility for studying direct contact condensation on a sub-cooled water surface [30], the
148 SETCOM facility for investigating the wall condensation and effects of inclination angle on heat transfer
149 [31], the INVEP facility for investigation of condensation inside an inclined pipe which is immersed in a
150 tank of sub-cooled water and for pressures of up to 10 bar [34]. As another example, KAERI (Korea
151 Atomeneres Research Institute) operates the ATLAS facility for thermal hydraulic analyses for the
152 APR1400 reactor system. There, experiments on the Passive Auxiliary Feedwater System (PAFS) were
153 performed in the PASCAL and ATLAS-PAFS test setups, which included studies on condensation heat
154 transfer in a lying U-tube heat exchanger similar to the one in KERENA [35], [36].

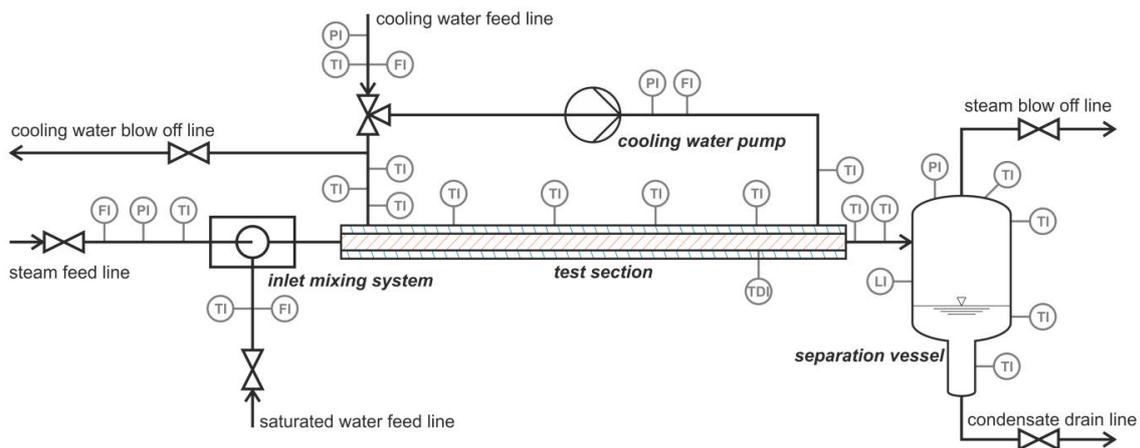
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156 **3. Experimental setup and conditions**

157 **3.1 Experimental facility**

158 Figure 3 shows a schematic representation of the COSMEA facility. It is essentially a single condenser
159 tube with 43.1 mm inner diameter, 48.3 mm outer diameter, 2.6 mm wall thickness, 2992 mm length
160 of heat transfer and a nominal inclination angle of $0.76 \pm 0.05^\circ$ to the horizontal. Note, that the latter
161 is only nominal as manufacturing tolerances and thermal expansion lead to slightly different
162 inclinations values as given in section 3.5.4 and 4.1. The tube material is stainless steel type 1.4571.
163 The test rig can be operated at up to 65 bar pressure and corresponding saturated steam
164 temperatures. The condenser tube is coaxially jacketed by a cooling tube with an outer diameter of
165 120 mm and a wall thickness of 2.0 mm made of grade 2 titanium alloy. The cooling circuit is operated
166 at max. 4 bar. The condenser tube is thus cooled via forced annular convective counter-current flow
167 which provides well-defined cooling conditions. During operation of COSMEA the condenser tube is
168 fed either with pure steam or a mixture of steam and saturated water via an in-house developed two-
169 phase mixer described in detail in the next chapter. Both fluids are provided by the steam generator
170 circuitry of the TOPFLOW facility. At the outlet of the condenser tube the two-phase mixture flows
171 tangentially into a properly dimensioned separation vessel (inner diameter 550 mm) where the
172 residual steam and the liquid are naturally separated. Both fluids are drained through separate tubes
173 into the TOPFLOW blow-off tank.

174



175

176 **Figure 3:** Scheme of the COSMEA facility (TI: temperature indication, PI: pressure indication, FI: mass flow
177 indication, LI: liquid level indication, TDI: temperature difference indication).

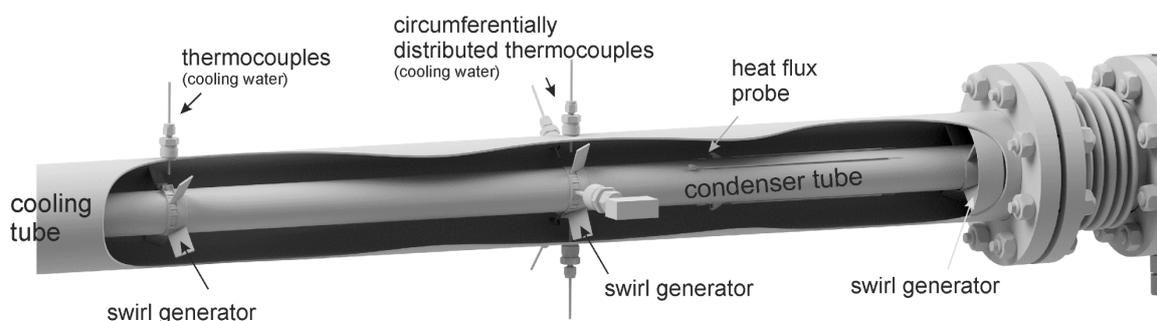
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179 The cooling water is circulated through the outer annulus with high mass flow. In this way, the heat-
180 up is minimized to a few Kelvin which allows controlling the heat transfer via the flow rate. The
181 temperature of the cooling water is controlled via the feed-and-split system shown in Figure 3. While
182 the major part of the cooling water is re-circulated through the annulus by the cooling water pump, a
183 minor part is extracted via the cooling water blow-off line and at the same time compensated by feed
184 from the cooling water feed-line. This way, the circulation loop is always in liquid single-phase
185 conditions. To achieve a high and homogenous heat flux, five swirl generators are installed along the
186 annular gap as shown in Figure 4 and Figure 5 (bottom part). The first one is positioned directly after
187 the cooling water inlet and the other ones in equal distances along the tube and near the temperature
188 sensors.

189 The cooling water circulation loop is equipped with several temperature sensors. As shown in Figure 3
190 there is one thermocouple at the inlet, three at the outlet and four more at intermediate positions in
191 equidistant places. Additionally, there are five thermocouples circumferentially distributed at an axial
192 position 2052 mm downstream the inlet, which are used to detect inhomogeneous temperature
193 distributions. Furthermore, there are absolute pressure sensors upstream the two-phase mixer and
194 inside the separation vessel and differential pressure sensors across the two-phase mixer. All sensors
195 were calibrated and their residual maximal measuring uncertainties determined as: temperature:
196 ± 0.3 K, pressure: $\pm 1\%$, mass flow steam: $\pm 2.2\%$ and liquid mass flow: $\pm 1.0\%$.

197 Two special measurement systems are additionally installed: a) an X-ray computed tomography (CT)
198 scanner that provides time-averaged cross-sectional images of the local flow morphology and b) a heat
199 flux probe (HFP) that allows a determination of the circumferential wall heat flux distribution at a given
200 axial position. Detailed information about both systems is given in the next chapters. In Figure 5, the
201 axial positions of the temperature sensors and special instrumentation are shown.

202



203

204 **Figure 4:** Cut-view of a part of the COSMEA facility discovering the swirl generators and instrumentation on
205 the secondary (cooling) side.

206

207 The operational data of COSMEA are recorded by a programmable logic controller that allows raw data
 208 receipt, conversion and calibration as well as process control and process synchronization. The
 209 operational data are sampled with a frequency of 1 Hz on an OPC (Open Platform Communications)
 210 server. The HFP data are managed by a separate controller with a sampling frequency of 3 Hz. Also the
 211 data of the X-ray CT scanner is stored on a separate computer. The synchronization of all measurement
 212 systems is realized by a common trigger signal that is provided by the tomographic system. In Table 1
 213 the range of experimental parameters as well as important geometric dimensions of the COSMEA
 214 facility are listed.

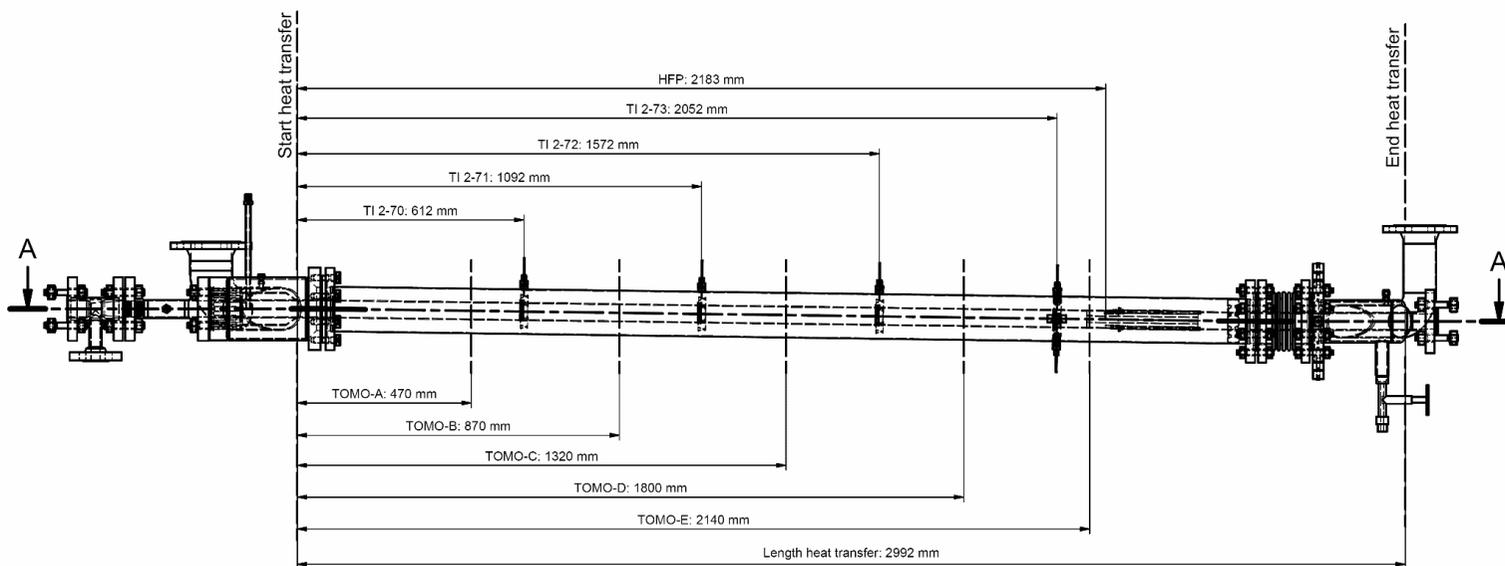
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216 **Table 1:** Compiled experimental parameters for condensation experiments
 217 at the COSMEA facility.

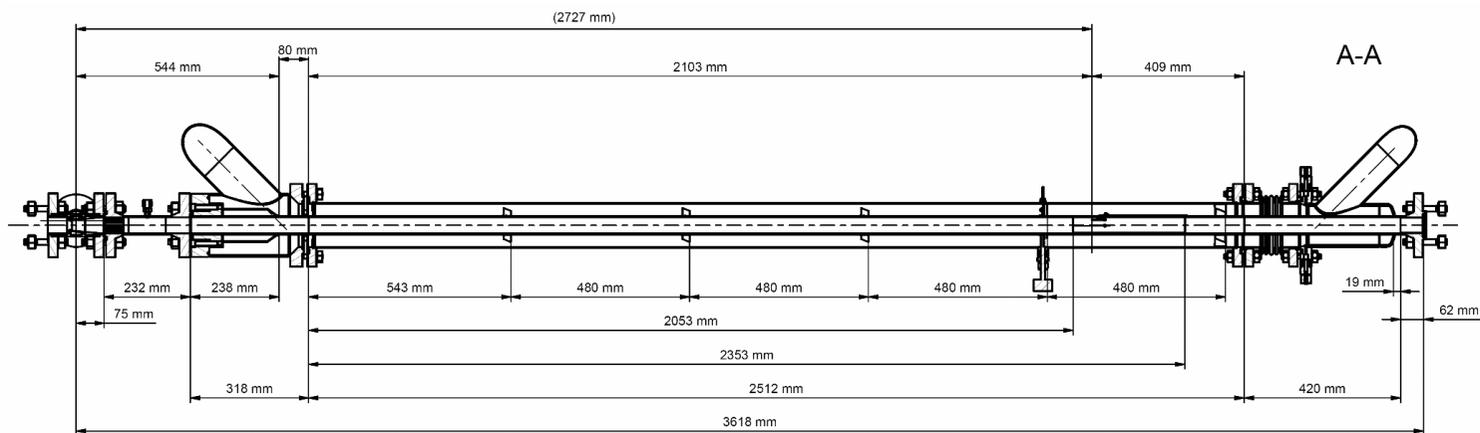
<i>Primary side parameters</i>	
Pressure	5 – 65 bar
Temperature	152 – 281 °C
Steam mass flow (inlet)	0.079 – 1.0 kg/s
Water mass flow (inlet)	0 – 0.751 kg/s
<i>Cooling water parameters</i>	
Pressure	3.0 - 4.0 bar
Water mass flow	13 - 24 kg/s
Temperature (mean)	nominal 45.5 °C
<i>Condenser tube parameters</i>	
Material	stainless steel (1.4571)
Wall thickness	2.6 mm
Heat flux length	2992 mm
Inclination	nominal 0.76 ±0.05° measured 0.88 ±0.035°
Inner diameter	43.1 mm

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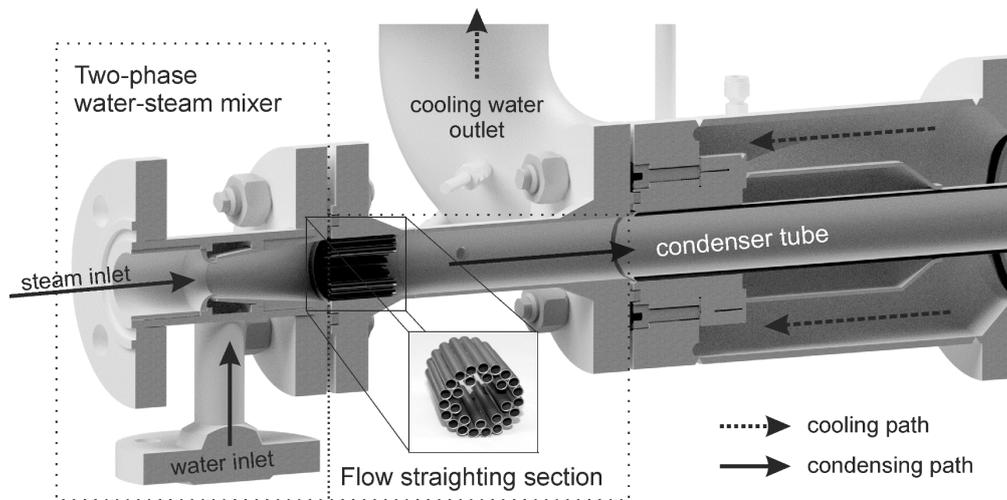
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222 **Figure 5:** Longitudinal cut of the COSMEA test section: top: with locations of the instrumentation (TI – temperature measurement, HFP – heat flux probe, TOMO – tomographic
223 imaging plane); bottom: with additional dimensions.

224 3.2 Inlet mixing system

225 The condensation process strongly depends on the flow regime inside the condenser tube. To
226 investigate steam condensation over a wide range of void fraction either a long condenser tube is
227 needed or alternatively, a well-defined two-phase flow must be fed in at the inlet. Due to limited space
228 in the laboratory we reverted to the second option. Hence, a two-phase steam-water mixer has been
229 designed and installed at the inlet of the horizontal test section (see Figure 6).

230



231 **Figure 6:** Two-phase steam-water flow mixer at the COSMEA facility.

232

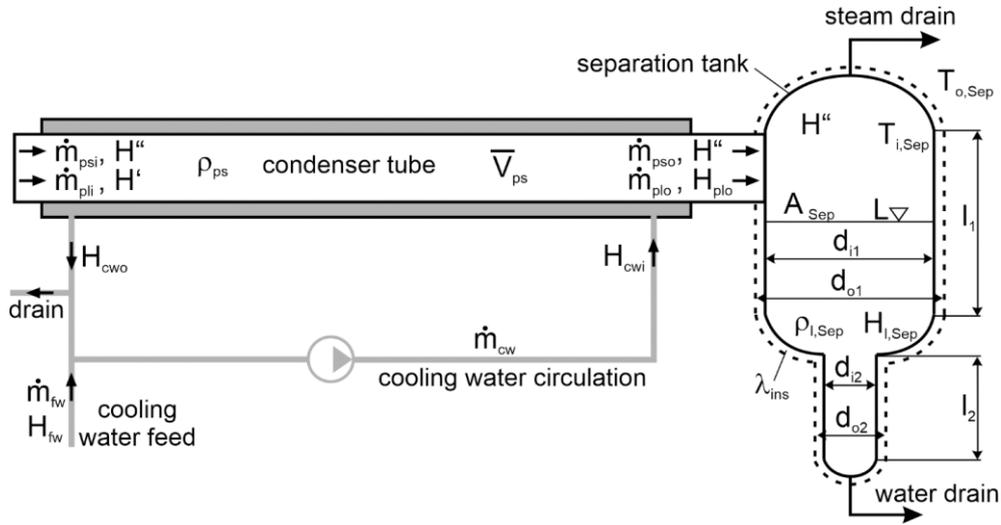
233 It provides an annular liquid injection (annular gap width: 1.8 mm) into the condenser tube, which is
234 close to an expected partially developed condensate film after some distance. However, as the
235 tangential liquid injection causes an undesired swirling flow we additionally provided a so-called flow
236 straightener downstream the mixer, which almost eliminates the swirling motion of the liquid. The
237 flow straightener is fixed on the flange pair directly downstream the mixer by stainless steel clips.
238 Between the mixer and the test section inlet, a plain tube segment with a total length of 225 mm is
239 flanged. The flow straightener and the cooling water outlet module provide an adiabatic two-phase
240 flow inlet length of 10 length-to-diameter ratio.

241

242 3.3 Condensation rate measurement

243 The condensation rate \dot{m}_c within the condenser tube can be derived using three different balancing
244 approaches. They will be described in detail below. Enthalpies are denoted as H' and H'' for saturated
245 water and saturated steam and as H_x for any specific stream x at absolute temperature T_x . To keep

246 the notation as simple as possible we do not explicitly denote the pressure dependence of enthalpies.
 247 Hence enthalpies of saturated water H' and saturated steam H'' are to be taken at the actual system
 248 pressure. For practical calculations the enthalpies and further properties of water and steam were
 249 taken from FluidExcel[®]. For a better understanding of the following equations Figure 7 provides an
 250 overview of the physical parameters and their relation to local heat and mass fluxes in the facility. An
 251 explanation of the abbreviations is given after the equations.



252
 253 **Figure 7:** Scheme of the COSMEA test rig including the relevant geometry, heat and mass flux quantities used
 254 in equations (1) - (13).

255
 256 Approach 1: The first approach considers the increase of enthalpy of the cooling water. The (rate of)
 257 heat transferred into the circulating cooling water on the secondary side is

$$\left(\frac{\Delta Q}{\Delta t}\right)_{cw} = \dot{m}_{cw}[H_{cwo} - H_{cwi}] \quad (1)$$

258 with the cooling water mass flux \dot{m}_{cw} and the enthalpy difference of the cooling water across the test
 259 section. On the primary side, the heat extraction leads to condensation as well as sub-cooling of
 260 condensate and injected saturated water, that is,

$$\left(\frac{\Delta Q}{\Delta t}\right)_p = \dot{m}_c[H'' - H_{plo}] + \dot{m}_{pli}[H' - H_{plo}]. \quad (2)$$

261 Here, \dot{m}_c is the condensate mass flux, \dot{m}_{pli} the mass flux of the liquid injected into the condenser tube,
 262 and H_{plo} the enthalpy of the sub-cooled water leaving the condenser tube. Balancing (1) and (2) one
 263 gets

$$\dot{m}_c^{(1)} = \frac{\dot{m}_{cw}[H_{cwo} - H_{cwi}] - \dot{m}_{pli}[H' - H_{plo}]}{H'' - H_{plo}} \quad (3)$$

264 The superscript “(1)” denotes, that this is the first out of three possible balancing approaches.

265 Approach 2: The temperature in the cooling circuit is controlled by the split-and-mix procedure as
 266 described above. This procedure gives way to a second approach of condensate rate quantification.
 267 Instead of using the secondary side circulation mass flux and enthalpies one may use the feed water
 268 mass flux \dot{m}_{fw} and the feed water enthalpy H_{fw} , that is,

$$\left(\frac{\Delta Q}{\Delta t}\right)_{fw} = \dot{m}_{fw}[H_{cwo} - H_{fw}]. \quad (4)$$

269 With that one obtains

$$\dot{m}_c^{(2)} = \frac{\dot{m}_{fw}[H_{cwo} - H_{fw}] - \dot{m}_{pli}[H' - H_{plo}]}{H'' - H_{plo}}. \quad (5)$$

270 Approach 3: As a third approach we consider the level increase rate dL/dt in the large cylindrical part
 271 (indicated as 1 in Figure 7) of the separation vessel downstream the condenser tube after an
 272 intermediate liquid drain line closure. For that, the condensation rate is

$$\dot{m}_c^{(3)} = \frac{dL}{dt} A_{Sep} \rho_{l,Sep} - \dot{m}_{pli} - \dot{m}_{hl} - \dot{m}_{hu} \quad (6)$$

273 with the cross-sectional area A_{Sep} of the separator vessel and the liquid density in the vessel $\rho_{l,Sep}$.
 274 The mass flux term \dot{m}_{hl} accounts for condensation in the separator vessel due to heat losses and the
 275 term \dot{m}_{hu} accounts for steam condensation due to heat-up of sub-cooled water from the test section.

276 For the determination of the heat losses from the separation vessel we can start from the assumption
 277 that the 150 mm thick thermal insulation with thermal conductivity $\lambda_{ins} = 0.047 - 0.059 \text{ Wm}^{-1}\text{K}^{-1}$
 278 (indeed it is a function of the mean insulation temperature) is the dominating thermal resistance.
 279 Hence, with reference to Figure 7, we can calculate the heat flux through the cylindrical parts
 280 (subscripts **cy,1** and **cy,2** for the large and small part respectively) and both torospherical heads
 281 (subscript **th**) by combining the heat conduction equations for cylindrical walls and hemispheric walls
 282 [39] as

283

$$\left(\frac{\Delta Q}{\Delta t}\right)_{hl} = \lambda_{ins} \left[\left(\frac{2\pi l_1}{\ln \frac{d_{o1}}{d_{i1}}} \right)_{cy,1} + \left(\frac{2\pi l_2}{\ln \frac{d_{o2}}{d_{i2}}} \right)_{cy,2} + \left(\frac{2\pi}{\frac{1}{d_{i1}} - \frac{1}{d_{o1}}} \right)_{th} \right] (T_{i,Sep} - T_{o,Sep}). \quad (7)$$

284

285 Here, d_i and d_o are the inner and outer diameter of the insulation shell (subscripts 1 – large and 2 –
 286 small respectively), $T_{i,sep} - T_{o,sep}$ is the temperature difference between the inner vessel atmosphere
 287 and the lab environment as well as l_1 and l_2 are the length of the large and small cylindrical part of the
 288 separation vessel respectively. In a further step we conservatively estimated the uncertainties due to
 289 unconsidered parts of the separation vessel, like e.g. instrumentation feed-throughs and steam blow-
 290 off pipe. From that we doubled the calculated value of the separation vessel heat losses. The last one
 291 can then be recalculated into a mass flux of condensed steam

$$\dot{m}_{hl} = \frac{\left(\frac{\Delta Q}{\Delta t}\right)_{hl}}{H'' - H_{l,sep}} \quad (8)$$

292 with $H_{l,sep}$ as the enthalpy of water at averaged liquid temperature inside the separator.

293 The last term in Eq. (6) accounts for the steam condensation in the separation tank due to heat-up of
 294 sub-cooled liquid from the test section, that is

$$\dot{m}_{hu} = \frac{\left(\frac{\Delta Q}{\Delta t}\right)_{hu}}{H'' - H_{l,sep}}. \quad (9)$$

295 and

$$\left(\frac{\Delta Q}{\Delta t}\right)_{hu} = \dot{m}_{plo}[H_{l,sep} - H_{plo}]. \quad (10)$$

296 As

$$\dot{m}_{plo} = \dot{m}_c^{(3)} + \dot{m}_{pli}, \quad (11)$$

297 we get

$$\dot{m}_{hu} = \left(\dot{m}_c^{(3)} + \dot{m}_{pli}\right) \frac{H_{l,sep} - H_{plo}}{H'' - H_{l,sep}}. \quad (12)$$

298 Inserting (12) into (6) and rearranging for $\dot{m}_c^{(3)}$ gives

$$\dot{m}_c^{(3)} = \frac{\frac{dL}{dt} A_{sep} \rho_{l,sep} - \dot{m}_{hl}}{\left(1 + \frac{H_{l,sep} - H_{plo}}{H'' - H_{l,sep}}\right)} - \dot{m}_{pli}. \quad (13)$$

299 While the 1st and 2nd approaches result in an exact calculation of the condensation rate, the 3rd one
 300 includes some assumption and corrections about heat losses and is therefore prone to slightly higher
 301 uncertainties. Hence, the 3rd method was used for plausibility cross-comparison only. To examine the

302 quality of both energy balance methods their uncertainties were calculated, applying the law of
 303 uncertainty propagation for Eq. (3) and Eq. (5). The structure of both equations is similar, so only the
 304 calculation of the 1st approach is presented here:

$$\Delta\dot{m}_c^{(1)} = \sqrt{\left(\frac{H_{cwo} - H_{cwi}}{H'' - H_{plo}} \cdot \Delta\dot{m}_{cw}\right)^2 + \left(\frac{\dot{m}_{cw}}{H'' - H_{plo}} \cdot \Delta H_{cwo}\right)^2 + \left(\frac{-\dot{m}_{cw}}{H'' - H_{plo}} \cdot \Delta H_{cwi}\right)^2 + \left(-\frac{H' - H_{plo}}{H'' - H_{plo}} \cdot \Delta\dot{m}_{pli}\right)^2 + \left(\frac{-\dot{m}_{pli}}{H'' - H_{plo}} \cdot \Delta H'\right)^2 + \left(\frac{\dot{m}_{pli}}{H'' - H_{plo}} \cdot \Delta H_{plo}\right)^2 + \left(\frac{-[\dot{m}_{cw} \cdot (H_{cwo} - H_{cwi}) - \dot{m}_{pli} \cdot (H' - H_{plo})]}{(H'' - H_{plo})^2} \cdot \Delta H''\right)^2 + \left(\frac{[\dot{m}_{cw} \cdot (H_{cwo} - H_{cwi}) - \dot{m}_{pli} \cdot (H' - H_{plo})]}{(H'' - H_{plo})^2} \cdot \Delta H_{plo}\right)^2} \quad (14)$$

305 Both uncertainty calculations are based on individual uncertainties of single parameters listed in Table
 306 2.

307 **Table 2:** Individual uncertainties of all parameters used for condensation rate and heat flux calculation.

Parameter	Pressure [bar]	Uncertainty	Reference
$\dot{m}_{cw}, \dot{m}_{fw}$		±1%	Swirl flow meter, TOPFLOW documentation
\dot{m}_{pli}		±0.2%	Coriolis flow meter, TOPFLOW documentation
\dot{m}_{psi} (FIC4-05)		±2.1%	Orifice plate, TOPFLOW documentation
\dot{m}_{psi} (FIC4-04)		±2.2%	Orifice plate, TOPFLOW documentation
H_{cwo}, H_{cwi}, H_{fw}		±1.24%	Individual uncertainties of pressure and temperature measurement and of IAPWS IF97
H', H''	5 12 25 45 65	±0.95% ±0.85% ±0.76% ±0.67% ±0.58%	Individual uncertainties of pressure measurement depending on test conditions and of IAPWS IF97
H_{plo}	5 12 25 45 65	±0.97% ±0.87% ±0.77% ±0.68% ±0.59%	Individual uncertainties of pressure and temperature measurement depending on test conditions and of IAPWS IF97
T_{wi}, T_{wo}		±0.3 K	After thermal calibration and polynomial correction
λ		± 7%	VDI-Wärmeatlas 2013, section D6, page 630
δ		±0.15 mm	0.1 mm from ultrasonic device and 0,05 mm from confocal white-light microscopy
ρ_{ps}	5 12 25 45 65	0.9% 0.8% 0.7% 0.6% 0.5%	Individual uncertainty of pressure measurement depending on test conditions and of IAPWS IF97

308

309 The uncertainty analysis showed that the 1st approach has a significantly higher uncertainty due to the
 310 fact that the mass flow of circulating cooling water is relatively high. Hence we consider the 2nd

311 approach as best for condensation rate estimation. Results including uncertainties are presented in
312 chapter 4.2.

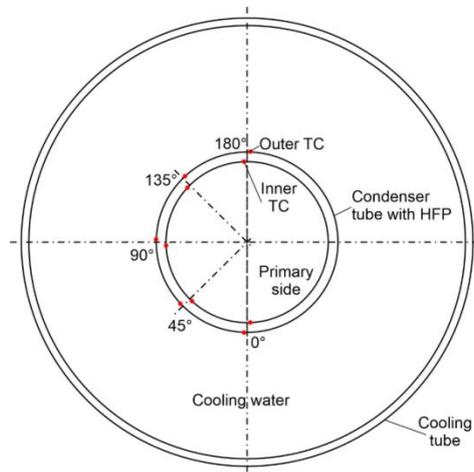
313

314 3.4 Wall heat flux measurement

315 The wall heat flux \dot{q}_w through the condenser tube wall is determined from the temperature difference
316 $T_{wi} - T_{wo}$ measured by pairs of thermocouples at the inner (primary side) and outer (secondary side)
317 condenser tube wall according to

$$\dot{q}_w = \lambda(\bar{T}) \frac{T_{wi} - T_{wo}}{\delta} \quad (15)$$

318 Here, δ is the distance between the thermocouples and $\lambda(\bar{T})$ is the thermal conductivity of the wall
319 material (stainless steel) at mean wall temperature [39]. Thermocouple pairs are arranged in five
320 circumferential positions (0° , 45° , 90° , 135° , 180°) as shown in Figure 8. This arrangement provides the
321 circumferential heat flux with some angular resolution and particularly allows discrimination of heat
322 flux from the steam and the condensate (bottom part) through the wall. As the inlet flow straightener
323 provides a good axis-symmetric inlet flow we considered a thermocouple pair arrangement in only one
324 half of the cross-section as sufficient. This assumption of flow symmetry has been confirmed by the X-
325 ray measurements.



326

327 **Figure 8:** Schema of TC arrangement on the heat flux probe (HFP).

328

329 We employed thermocouples of Type K class 1 with a sensor tip diameter of 0.5 mm. They are spot-
330 welded in eroded grooves with a depth of 250 μm to minimize the influence of the near-wall fluid
331 temperature. For reasons of mechanical stability the grooves on the outer wall are 4° displaced against
332 the grooves at the inner wall. As the true distance δ of each thermocouple pair is important for

333 accurate heat flux measurement it was determined before the test section assembly by measuring the
 334 condenser tube wall thickness at each position using an ultrasonic inspection technique and measuring
 335 the thermocouple immersion depth using confocal white light microscopy. To assess the quality of the
 336 heat flux measurement, their uncertainty was determined using the law of error propagation to Eq.
 337 (14). From this one gets

$$\Delta \dot{q}_w = \sqrt{\left(\frac{T_{wi} - T_{wo}}{\delta} \cdot \Delta \lambda\right)^2 + \left(\frac{\lambda}{\delta} \cdot \Delta T_{wi}\right)^2 + \left(-\frac{\lambda}{\delta} \cdot \Delta T_{wo}\right)^2 + \left(-\frac{\lambda \cdot (T_{wi} - T_{wo})}{\delta^2} \cdot \Delta \delta\right)^2}. \quad (16)$$

338 The individual uncertainties and their references are taken from Table 2. The uncertainty of the heat
 339 flux is dominated by the 1st and 4th term in Eq. (16), respectively the individual uncertainties of the
 340 thermal conductivity and the wall thickness. Since both terms in Eq. (16) mainly depend on the
 341 temperature difference between the primary and secondary wall side and the heat flux itself shows
 342 the same dependency, the heat flux relative uncertainty is practically independent of the operational
 343 boundary condition. It was found approximately at 9%. Similar to the previous section the results of
 344 the circumferentially distributed heat flux and their uncertainties are presented in section 4.3.

345

346 3.5 X-ray tomography

347 3.5.1 CT setup and data processing

348 The COSMEA facility is equipped with a proprietary X-ray computed tomography (CT) system that
 349 enables non-invasive imaging of the cross-sectional liquid fraction distribution along the entire
 350 condenser tube section (see Figure 9). The CT scanner comprises a rotating frame with an X-ray source,
 351 a radiation flat panel detector (1024×1024 pixels of 400×400 μm² active area) and a control unit. The
 352 X-ray source is collimated by means of adjustable lead plates in front of the beam exit window to
 353 suppress scattered radiation by at least one order of magnitude. Moreover, the whole CT system can
 354 be automatically traversed along the whole test section. For cross-sectional scanning a servo-motor
 355 drives the source-detector assembly around the condenser tube section in an angular range of about
 356 230°. Every 0.36° an X-ray image is taken with an exposure time of 100 ms. From a set of 640 projection
 357 images (total scanning time approx. 10 min) a computer program reconstructs the cross-sectional
 358 images. For that we implemented a numerical inverse Radon transformation in GNU OCTAVE v4.2.1.
 359 The whole data processing and image reconstruction comprises the following single steps:

- 360 - Dark field subtraction to compensate for the detector's dark current
- 361 - Attenuation value calculation using a reference CT scan without object
- 362 - Defective pixel correction by linear interpolation from non-defective neighbour pixels

- 363 - Scattered radiation correction using an approximation of the scattered radiation intensity
- 364 profile fitted to measured intensities behind the X-ray source collimator edges
- 365 - Correction of electrical crosstalk between the detector module panels
- 366 - Averaging of projection data over 61 detector lines corresponding to an averaging along
- 367 an effective axial distance of 8.5 mm in the test section
- 368 - Correction of the condenser tube position within the fan beam projection data
- 369 - Interpolation from fan beam to parallel beam projection data
- 370 - Reduction of the projection data to the essential 180° parallel beam projection
- 371 - Clipping of the parallel beam projection data so that it contains the condenser tube only
- 372

373 As the CT scanning interval is rather long it was decided to limit tomography to five equally spaced
 374 axial positions along the test section. The exact positions are selected in places, where images are least
 375 distorted by extra materials, such as thermocouples or swirling elements in the cooling tube. For that
 376 we initially performed a frontal radiographic scan of the inactive test section and determined the five
 377 axial CT scanning positions denoted as "A"- "E" shown in Figure 10 (A: 470 mm, B: 870 mm, C: 1320 mm,
 378 D: 1800 mm, E: 2140 mm).

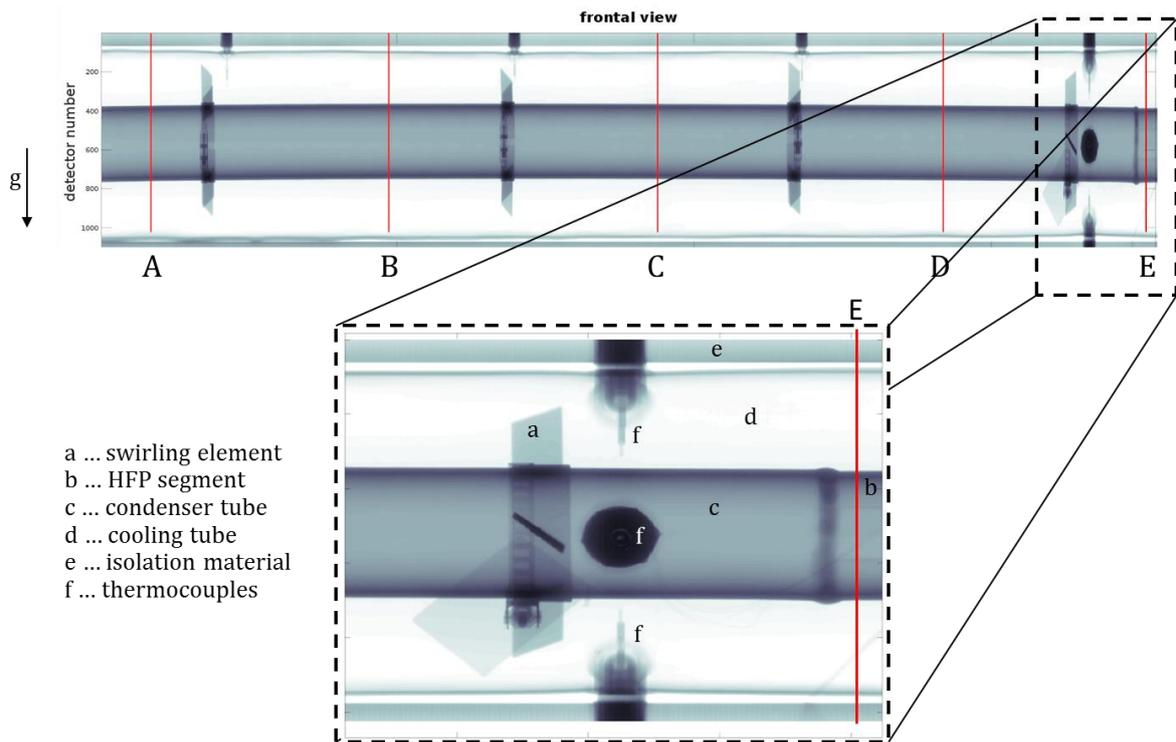
379



380

381 **Figure 9:** COSMEA facility at HZDR with the X-ray CT scanner.

382



383

384 **Figure 10:** Frontal radiographic scan of the inactive COSMEA facility with determined CT scanning positions “A”-
 385 “E” (top) and detailed information (bottom).

386

387 3.5.2 Assessment of the CT measurement uncertainty

388 As our X-ray CT imaging system is custom-made we initially determined the measuring uncertainty by
 389 means of a phantom experiment. The phantom (Figure 11a) resembles a water film of increasing
 390 thickness at the inner tube wall. It comprises of two hollow steel cylinders resembling the outer and
 391 inner tube walls with the annulus being completely filled with casting resin to model the cooling water.
 392 A stack of four silicone stripes, representing static condensate films of thickness between 1 mm and
 393 4 mm is placed on the inner wall of the condenser tube section. Finally, the phantom was jacketed
 394 with mineral wool insulation material. The following CT scanning parameters have been experimentally
 395 determined as optimal with respect to detector exposure and total scanning time:

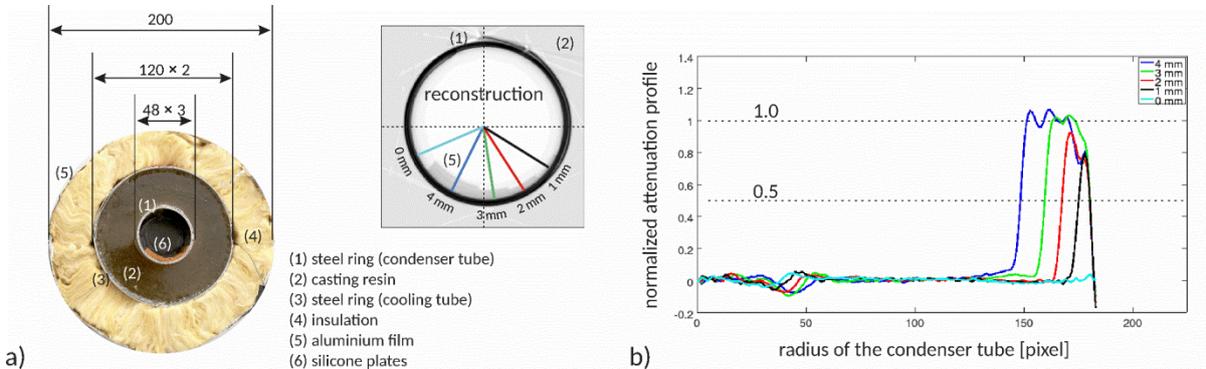
- 396 - Tube voltage: 150 keV
- 397 - Tube power: 3.75 kW
- 398 - Focal spot size: 0.6 mm
- 399 - Exposure interval: 100 ms (power controlled)

400

401 The resulting reconstructed cross-sectional image of the phantom is shown in Figure 11. As it can be
 402 seen in the corresponding extracted averaged and normalized attenuation profiles in Figure 11b, the

403 1 mm silicon stripe can still be visualized with approx. 80% of its contrast. It indicates that the
 404 resolution ability for a condensation film near the condenser inner wall is better than 1 mm.

405

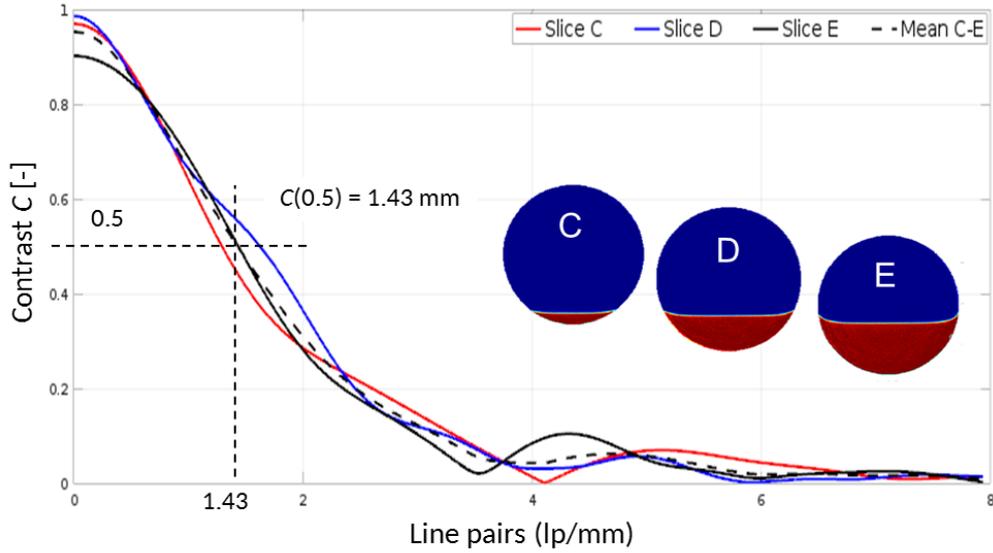


406

407 **Figure 11:** Assessment of the uncertainty for X-ray imaging based liquid film thickness measurement. a)
 408 Phantom that resembles a section of the concentric tubes with a liquid film of different thickness
 409 inside. b) Extracted normalized attenuation profiles of the silicone stacks.

410

411 A more appropriate measure for the contrast resolution is the modulation transfer function (MTF) of
 412 the imaging system. It has been determined from a CT scan of a quiescent liquid level in the tube.
 413 Therefore, the condenser tube was filled with deionized water and all valves were closed. The trapped
 414 stratified deionized water develops a sharp static interface and the resulting edge at the interface in
 415 the cross-sectional images can be used to calculate the MTF for the present setup from the attenuation
 416 profiles along the vertical diameter, which were evaluated for different filling levels denoted at cross-
 417 section "C" (1320 mm), "D" (1800 mm) and "E" (2140 mm) (see Figure 12). The filling level represents
 418 an edge and the CT scanner obtains an image which contains the edge response function. From that
 419 the point spread function can be de-convolved, whose Fourier transform is the MTF. The resulting MTF
 420 gives a resolution of 1.43 line pairs per millimeter at 50% contrast $C(0.5)$, which in turn indicates a
 421 detectability of an interface for a film of a thickness down to 0.5 mm. From the evolution of the
 422 quiescent gas-liquid interface along the tube axis in the X-ray images we were able to measure the real
 423 inclination angle of the condenser tube as $0.88 \pm 0.035^\circ$. The uncertainty comes from the spatial
 424 resolution (~ 0.5 mm). Note, that the inclination difference between the nominal value (0.76°) and the
 425 measured mean value (0.88°) corresponds to about 7.2 mm difference in height of the rightmost point
 426 of the condenser tube. The difference is due to both manufacturing tolerances and the loose-fit
 427 fixation of the downstream part of the condenser tube. Further note, that thermal expansion of the
 428 tube during experiments does give another inclination offset of up to 0.06° or 1.8 mm (section 3.5.6,
 429 Figure 14).



430

431 **Figure 12:** Modulation transfer function of the X-ray CT imaging system (for details see text).

432

433 3.5.3 CT data analysis

434 Ideally, the image reconstruction delivers cross-sectional images, i.e. 2D maps, of the linear
 435 attenuation coefficient of the materials in the cross-section. They will be further denoted by $\mu(x, y)$
 436 with an additional index for a particular data set from an experiment (*exp*) or a reference
 437 measurement. In principle, the linear attenuation coefficient is linearly proportional to the material
 438 density. This allows a quantitative analysis of the liquid distribution. However, the reconstruction of
 439 absolute linear attenuation coefficients is prone to a number of uncertainties associated with the X-
 440 ray propagation and image reconstruction. Therefore, it is common practice to employ differential
 441 measurements, that is, scaling the reconstructed values $\mu_{\text{exp}}(x, y)$ of a given experiment with
 442 reference values for the tube being filled once completely with gas $\mu_{\text{g}}(x, y)$ and once completely with
 443 liquid $\mu_{\text{l}}(x, y)$. If such references are available, the quantitative liquid fraction distribution can be
 444 calculated

$$\alpha(x, y) = \frac{\mu_{\text{exp}}(x, y) - \mu_{\text{g}}(x, y)}{\mu_{\text{l}}(x, y) - \mu_{\text{g}}(x, y)} \cdot \frac{\rho_{\text{l}} - \rho_{\text{g}}}{\rho_{\text{exp}} - \rho_{\text{g}}}. \quad (17)$$

445 The second term at the right-hand side accounts for density differences between experiment and
 446 reference scan due to temperature differences. This procedure is e.g. fully described in [40].

447 It has been found, however, that the procedure described above is not straightforwardly applicable to
 448 COSMEA experiments. Thus, the geometrical displacements of the tube between images taken at
 449 different temperatures leads to strong artefacts in the images. Even when applying displacement
 450 correction by means of image processing (e.g. by geometrical shifting of the tube in the images) such

451 artefacts cannot be fully removed, especially in the near-wall regions, which are important for analysis.
 452 Moreover, for technical reasons it is very difficult to fill the tube completely with saturated water and
 453 keep the water on saturation temperature during X-ray scans. Hence, we reverted to a modified scaling
 454 approach. First of all we performed reference scans for the gas-filled tube (μ_g) and the liquid-filled
 455 tube (μ_l) at ambient conditions (1 bar, 20°C). Next we performed a third reference scan for the
 456 maximum possible steam flow at given experimental conditions (pressure, saturation temperature).
 457 The resulting image is referred to as μ_s^* . The asterisk indicates the reference to the values given in Table
 458 3. Note, that the difference images, $\mu_{\text{exp}}(x, y) - \mu_s^*(x, y)$ and $\mu_l(x, y) - \mu_g(x, y)$, are free of
 459 displacement artifacts, as the individual scans have been performed at the same temperature and
 460 pressure stages. However, they still have a displacement against each other. When we now perform a
 461 shift correction by automatic image processing, the resulting liquid fraction image (see Eq. (18)) is no
 462 more corrupted. Thereby, $\mu_s^*(x, y)$ can be considered as a reference for a fully steam-filled tube.
 463 However, as the tube wall is slightly sub-cooled we may expect a small condensate film at the inner
 464 wall side. To ensure that this does not introduce additional uncertainties we assessed all reconstructed
 465 cross-sectional images with a visible bottom condensate flow, if there is a negative gray value gradient
 466 towards the wall in the condensate region. We could confirm that this is not the case, from which it
 467 follows that a) the condensate film is well below the resolution limit (~ 0.5 mm) of the X-ray
 468 tomography and b) does therefore not introduce uncertainty in the near-wall region.

469 Eventually, we obtain the liquid fraction by further correcting with the density differences as

$$\alpha_{\text{CT}}(x, y) = \frac{\mu_{\text{exp}}(x, y) - \mu_s^*(x, y)}{\mu_l(x, y) - \mu_g(x, y)} \cdot \frac{\rho_l - \rho_g}{\rho_{\text{exp}} - \rho_s} \quad (18)$$

470 and correspondingly the steam fraction as

$$\varepsilon_{\text{CT}}(x, y) = 1 - \alpha(x, y). \quad (19)$$

471 The condensate height L_c can be directly taken from the central vertical attenuation profile of the
 472 reconstructed slice as shown in Figure 13a. To improve accuracy a centrally placed profile thickness of
 473 $\delta_{\text{profil}} = 1$ mm is used. As liquid fraction threshold value $\alpha = 0.5$ is used for this parameter. As the
 474 steam-liquid interface is agitated by the gas-liquid shear and turbulent structures wave structures are
 475 developed. Though those cannot be resolved in time the wave amplitude can be quantified via the
 476 liquid fraction transition zone at the interface in the reconstructed images. For that the transition zone
 477 width ΔL is extracted by defining lower and upper thresholds $\alpha = 0.1$ and $\alpha = 0.9$. The corresponding
 478 condensate level heights L_c^- and L_c^+ are then used to compute the transition zone $\Delta L = L_c^- - L_c^+$, as
 479 shown in Figure 13b. The accuracy has been determined as ± 0.13 mm.

480 The total liquid fraction in the cross-section can directly be computed from the CT image according to

$$\bar{\alpha}_{CT} = \frac{1}{N_x N_y} \sum_{x=1}^{N_x} \sum_{y=1}^{N_y} w(x, y) \alpha(x, y). \quad (20)$$

481 and the total steam fraction as well

$$\bar{\epsilon}_{CT} = 1 - \bar{\alpha}_{CT}. \quad (21)$$

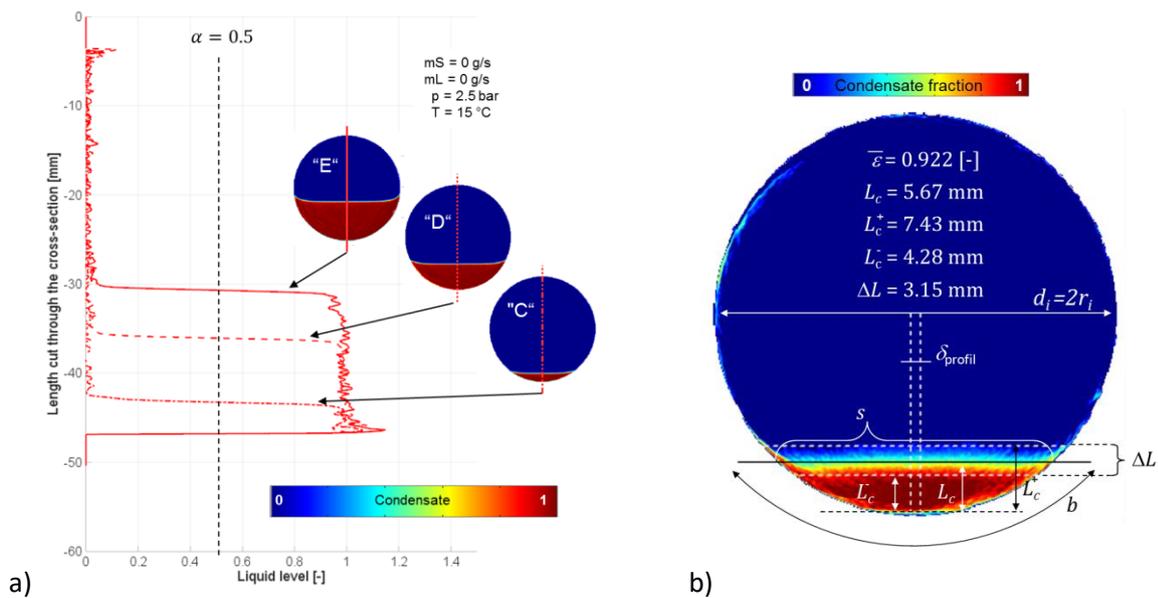
482 The weight function $w(x, y)$ defines the share of the pixel with the internal cross-section of the tube,
 483 that is, $w(x, y) = 0$ for pixels outside the tube cross-section, $w(x, y) = 1$ for pixels inside and $0 <$
 484 $w(x, y) < 1$ for pixels on the boundary.

485 Moreover, from geometrical considerations of a well separated flow with a flat interface the following
 486 relationship between total liquid fraction $\bar{\alpha}_L$ and liquid level L_c can be derived by

$$\bar{\alpha}_L = \frac{A_r}{A_i} = \frac{\frac{1}{2} \cdot (r_i \cdot b - s \cdot (r_i - L_c))}{\pi \cdot r_i^2}. \quad (22)$$

487 Here, A_r denotes the flume cross-sectional area and A_i the inner cross-sectional area of the tube. The
 488 other geometrical parameters are given in Figure 13. Although, the total liquid fraction $\bar{\alpha}_{CT}$ that is
 489 directly computed from the CT image is more appropriate for a non-flat interface, this second method
 490 that considers the stratified condensate level (SCL) has been used for cross-comparison.

491

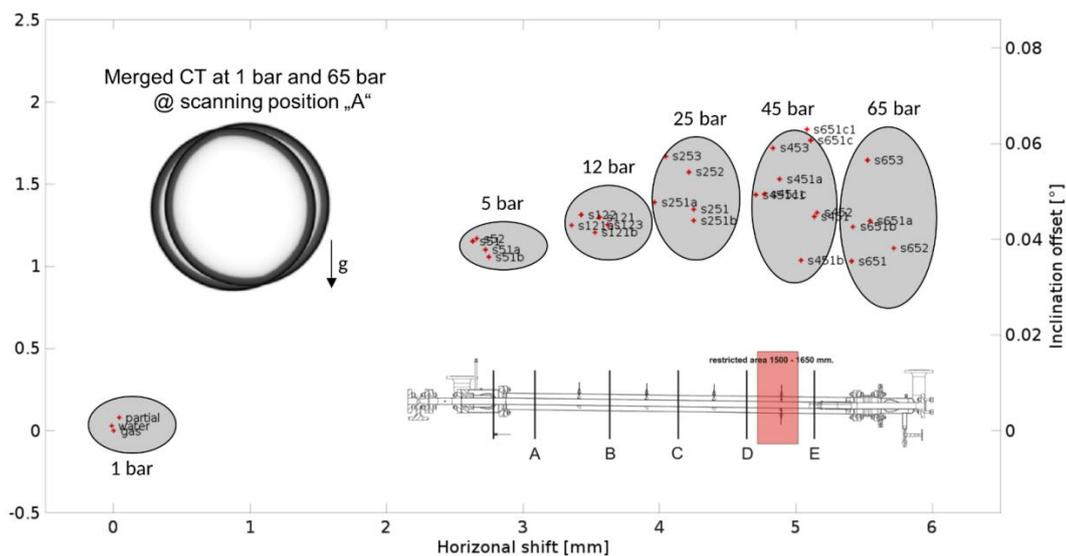


492 **Figure 13:** Procedure to investigate a) the height and b) the transition zone of the stratified condensate (flume)
 493 from the cross-sectional X-ray CT images.

494 **3.5.4 Displacement of the condenser tube due to thermal expansion**

495 Due to thermal expansion the inclination angle of the test section changes slightly at different
 496 operating conditions, which may have to be considered in forthcoming numerical simulations. The
 497 displacement increases with temperature and axial distance from the separation vessel, as the latter
 498 is a fix-point of the condenser tube. Thus, we used the tomographic images to determine the vertical
 499 and horizontal displacement by determining the centre position of the condenser tube in Section “A”
 500 and “E”, calculating the centre shift offset against the centre positions given in chapter 3.5. As expected
 501 the largest displacement was found at CT position “A” at maximal pressure difference of 65 bar, and is
 502 exemplarily shown as an inset in Figure 14. Moreover the displacement is fortunately stronger in
 503 horizontal direction and therefore, the inclination angle change is less affected. The results are
 504 summarized graphically in Figure 14 and quantitatively in Table 3.

505



506 **Figure 14:** Vertical and horizontal displacement of the condenser tube at different operating points (pressure
 507 stages) as obtained from the reconstructed cross-sectional images between CT scanning position “A”
 508 and “E”.
 509

510

511

512 **4. Results and discussion**

513 **4.1 Test matrix and experimental procedure**

514 Experiments were performed for the conditions given in Table 3. Grey cells indicate experiments with
 515 additional injection of saturated water (two-phase mixture at inlet).

516 **Table 3:** Summary of the basic experimental conditions (*indicates reference CT scan for corresponding
 517 pressure stage, grey cells indicate experiments with additional injection of saturated water).

Test #	p_p [bar(a)]	\dot{m}_{total} [kg/s]	\dot{m}_{psi} [kg/s]	\dot{m}_{pli} [kg/s]	T_{pli} [°C]	p_{cw} [bar(a)]	\dot{m}_{cw} [kg/s]	T_{cwi} [°C]	T_{cwo} [°C]	Inclination offset [°]
s51	5.004	0.0795	0.0795	0		3.110	13.38	44.3	46.6	+0.040
s51a	5.010	0.1195	0.1195	0		3.132	13.45	43.5	45.9	+0.038
s51b*	5.015	0.1510	0.1510	0		3.120	13.56	44.2	46.5	+0.036
s52	5.008	0.0985	0.0585	0.040	151.9	3.187	13.33	44.9	47.0	+0.040
s121	12.008	0.1834	0.1834	0		3.138	13.62	43.9	47.2	+0.045
s121a	12.007	0.0996	0.0996	0		3.139	13.58	44.2	47.1	+0.043
s121b	12.028	0.2797	0.2797	0		3.188	13.65	43.4	47.0	+0.041
s122	12.006	0.1850	0.1400	0.045	183.0	3.173	13.57	44.2	47.4	+0.045
s123	12.006	0.1821	0.1021	0.080	187.6	3.145	13.58	44.0	47.0	+0.043
s251	25.058	0.3723	0.3723	0		3.127	13.77	43.3	47.8	+0.046
s251a	25.040	0.1707	0.1707	0		3.149	13.57	42.9	46.5	+0.048
s251b	25.093	0.5536	0.5536	0		3.218	15.86	43.7	48.0	+0.044
s252	25.053	0.3711	0.2331	0.138	223.7	3.169	13.74	43.3	47.4	+0.054
s253	25.050	0.3720	0.1240	0.248	224.5	3.158	13.64	43.6	47.3	+0.057
s451	45.094	0.6739	0.6739	0		3.340	19.44	43.5	47.8	+0.045
s451a	45.056	0.2492	0.2492	0		3.249	15.38	43.2	47.5	+0.053
s451b	45.289	1.0039	1.0039	0		3.480	20.95	42.9	47.2	+0.036
s451c	45.056	0.1528	0.1528	0		3.187	13.64	43.8	47.8	+0.049
s452	45.064	0.6727	0.4197	0.253	257.7 ⁺	3.319	18.19	43.4	47.7	+0.046
s453	45.048	0.6742	0.2042	0.470	259.2 ⁺	3.278	16.13	43.5	47.8	+0.059
s651*	65.161	1.0038	1.0038	0		3.590	23.40	43.0	47.3	+0.035
s651a	65.107	0.3493	0.3493	0		3.355	18.73	43.3	47.6	+0.044
s651b	65.121	0.6785	0.6785	0		3.462	21.94	43.8	48.1	+0.043
s651c	65.104	0.1965	0.1965	0		3.208	14.41	43.3	47.6	+0.061
s652	65.126	1.0006	0.6226	0.378	281.3 ⁺	3.500	22.18	43.3	47.6	+0.038
s653	65.164	0.9937	0.2427	0.751	284.3 ⁺	3.347	19.54	43.4	47.6	+0.056

518

519 The description of the parameters in Table 3 corresponds to the nomenclature introduced in Figure 7
 520 and in Eq. (1) - (15). The liquid temperatures at condensation tube inlet (T_{pli}) marked with a (*) are
 521 slightly higher than the saturation temperature at primary pressure (p_p). This is due to pressure drop
 522 over the annular gap of the mixer at condensation tube inlet (see Figure 6) that causes a low
 523 overpressure at temperature measurement position T_{pli} . Saturated steam mass flows up to 1.00 kg/s

524 and water mass flows up to 0.75 kg/s resulting in inlet mass steam fractions between 0.244 and 1.000
525 that have been studied at steady-state conditions.

526 At the beginning of each experimental campaign the COSMEA test rig was heated up by starting and
527 regulating the cooling water circulation loop. Then steam from the TOPFLOW steam generator was
528 injected into the condensation tube with a mass flow higher than the condensation rate. Thereby, the
529 drain-line of the separation tank was opened to blow-off the residing mixture of air, steam and
530 condensate. This procedure allows an effective way to degas the test section and the separation vessel,
531 which is a basic requirement for high quality condensation tests without non-condensable gases. The
532 degassing was concluded when the gas temperature inside the separation vessel was at saturation
533 temperature for the given pressure. During steam injection the primary side of the test rig was heated
534 up until the pressure set value was slightly exceeded. This operation accelerates the temperature rise
535 in the COSMEA steel components. Afterwards, the drain line was partly closed and a low constant level
536 in the large part of the separation vessel was adjusted. Besides, the experimental conditions (Table 3)
537 were adjusted too and stabilized over 15 min to give stable steady-state conditions. The pressure in
538 the test rig was controlled automatically by regulation of the blow-off steam flow. When all parameters
539 were properly adjusted the CT scans were conducted at constant level in the separation vessel. After
540 this the condensate drain line of the separation tank was completely closed to determine the level
541 gradient for the condensation rate calculation.

542 As aforementioned, all tests described here were run in steady-state conditions. For the perpetuation
543 of constant thermal hydraulic conditions we used two control mechanisms. In the cooling circuit we
544 applied the above described feed-and-split procedure to adjust the circulating flow at almost constant
545 temperature, independent of the transferred heat. On the primary side we had to control the pressure
546 under condensation conditions. Note that beside the investigated condensation of steam in the
547 condenser tube, there is also “parasitic” condensation of steam in the separation vessel due to heat
548 losses through its wall and heat-up of sub-cooled condensate from the condenser tube in the
549 separation tank. Both effects were explained and quantified in section 3.3. Furthermore, a small
550 amount of additional steam has to be continuously fed into the separation tank to replace steam losses
551 from pressure-controlling steam blow-offs.

552

553 **4.2 Condensation rate**

554 The results of the condensation rate \dot{m}_c calculation regarding to section 3.3 for all experiments
555 including their uncertainties are presented in Table 4. For a better overview \dot{m}_c was plotted in two
556 diagrams, at the one hand as a function of mean steam mass fraction \bar{x} (Figure 15a) and at the other

557 hand subject to the mean steam volumetric flow rate, averaged over the condensation tube length
 558 (Figure 15b). Mean values were chosen because the condensation rate represents an integral
 559 characteristic over the entire condensation domain. The inlet steam fraction was calculated as

$$\bar{x} = \frac{2 \cdot \dot{m}_{\text{psi}} - \dot{m}_c^{(2)}}{2 \cdot (\dot{m}_{\text{psi}} + \dot{m}_{\text{pli}})} \quad (23)$$

560 In Eq. (23), \dot{m}_{psi} is the steam mass flow injected into the primary side of the condensation tube. The
 561 mean steam volumetric flow rate was determined as

$$\bar{V}_{\text{ps}} = \frac{2 \cdot \dot{m}_{\text{psi}} - \dot{m}_c^{(2)}}{2 \cdot \rho_{\text{ps}}} \quad (24)$$

562 Here ρ_{ps} stands for the primary side steam density. For a correct visualization an uncertainty analysis
 563 for both steam mass fraction and steam mass flow rate was performed (Table 4). Similar to section 3.3
 564 the law of uncertainty propagation was applied to Eq. (23) and Eq. (24). The individual uncertainties of
 565 the raw data were taken from Table 2.

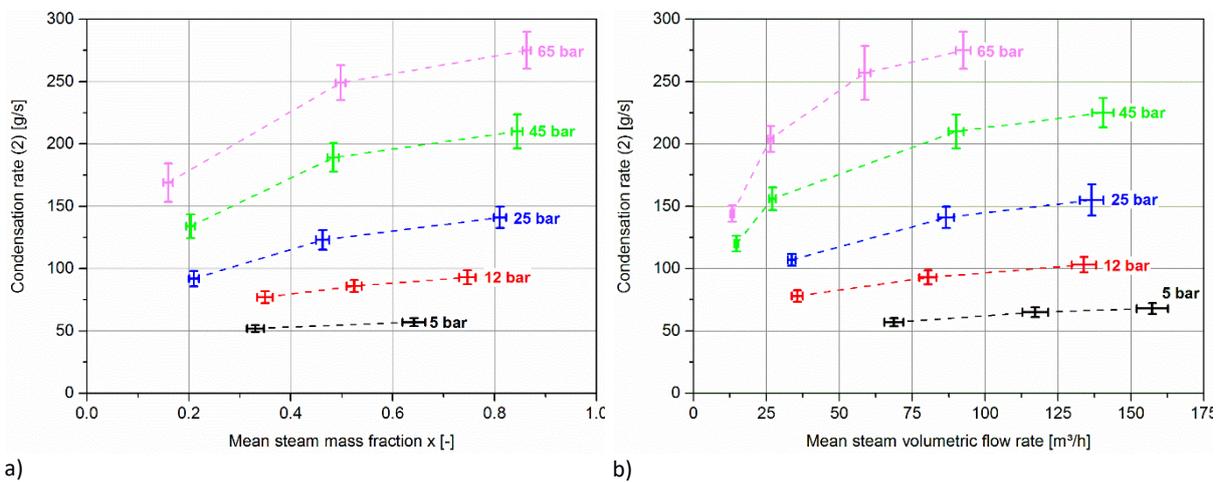
566 **Table 4:** Condensation rates (calculated by the 2nd approach explained in section 3.3).

Test #	\bar{x} [-]	\bar{V}_{ps} [m ³ /h]	$\dot{m}_c^{(2)}$ [g/s]	Test #	\bar{x} [-]	\bar{V}_{ps} [m ³ /h]	$\dot{m}_c^{(2)}$ [g/s]
s51	0.642 ±0.022	68.8 ±3.2	57 ±3.3	s451	0.844 ±0.011	90.0 ±2.6	210 ±13.5
s51a	0.728 ±0.017	117.2 ±4.4	65 ±3.9	s451a	0.687 ±0.019	27.1 ±1.1	156 ±9.1
s51b*	0.775 ±0.016	157.3 ±5.4	68 ±4.4	s451b*	0.888 ±0.006	140.4 ±3.7	225 ±11.7
s52	0.331 ±0.017	43.9 ±2.5	52 ±2.8	s451c	0.607 ±0.023	14.7 ±0.7	120 ±6.4
				s451c1	0.608 ±0.021	14.9 ±0.7	121 ±6.1
s121	0.746 ±0.016	80.4 ±3.0	93 ±5.6	s452	0.483 ±0.011	51.5 ±1.7	189 ±11.5
s121a	0.608 ±0.025	35.6 ±1.9	78 ±4.7	s453	0.203 ±0.009	21.7 ±1.0	134 ±9.5
s121b*	0.816 ±0.012	133.9 ±4.2	103 ±6.1				
s122	0.524 ±0.015	57.0 ±2.3	86 ±4.8	s651*	0.863 ±0.008	92.5 ±2.5	275 ±14.8
s123	0.349 ±0.015	37.4 ±1.9	77 ±4.8	s651a	0.708 ±0.016	26.4 ±1.0	204 ±10.3
				s651b	0.811 ±0.016	58.7 ±2.0	257 ±21.6
s251	0.811 ±0.012	86.7 ±2.7	141 ±8.6	s651c	0.634 ±0.018	13.3 ±0.6	144 ±6.5
s251a	0.687 ±0.016	33.7 ±1.3	107 ±4.8	s651c1	0.633 ±0.020	13.3 ±0.6	144 ±6.9
s251b*	0.860 ±0.012	136.5 ±4.0	155 ±12.5	s652	0.498 ±0.010	53.2 ±1.7	249 ±14.1
s252	0.462 ±0.013	49.3 ±1.9	123 ±7.9	s653	0.159 ±0.009	16.9 ±1.0	169 ±15.4
s253	0.210 ±0.010	22.4 ±1.2	92 ±6.1				

567
 568 Figure 15a shows the influence of the liquid content in the horizontal tube on the condensation rate.
 569 The condensation rate decreases with decreasing mean steam mass fraction. This effect is attributed

570 to the fact that liquid blocks direct steam contact to the bottom part of the inner tube wall and
571 therefore lowers an effective heat transfer.

572 In Figure 15b the condensation rates (2nd approach) for experiments with pure steam injection are
573 plotted against the mean steam volumetric flow rate. The strong influence of steam flow, respectively
574 the averaged steam velocity, on the condensation rate is clearly recognizable. As for example, reducing
575 the steam flow from 92.5 m³/h to 13.3 m³/h at 65 bar halves the condensation rate at no change of
576 any other operational boundary condition. This effect bases on a decrease of the primary heat transfer
577 coefficient due to a decrease of the steam velocity. Both diagrams in Figure 15 clearly show the
578 influence of the pressure in the condensation. As expected from the increasing temperature difference
579 between cooling water and steam, the condensation rates increase with system pressure at
580 comparable boundary condition.



581 **Figure 15:** Condensation rates as a function of inlet steam fraction (a) and mean steam volumetric flow rate (b).

582

583 4.3 Flow morphology

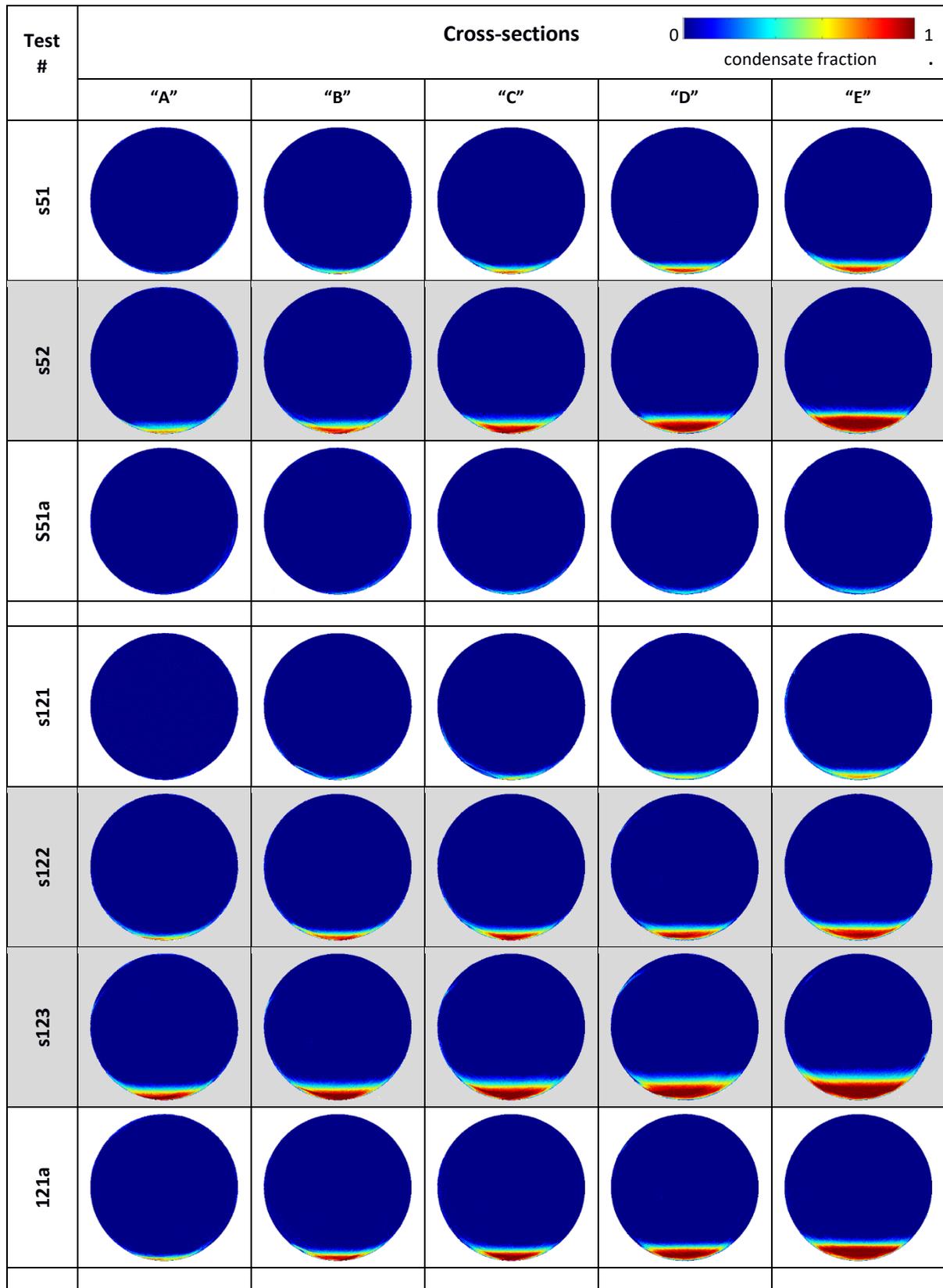
584 Eventually, a selection of reconstructed condensate fraction distributions is shown in Table 5. A
585 summary of all results and data processing procedures can be found in Bieberle et al. (2019) [41], [42].

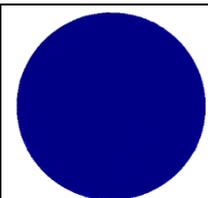
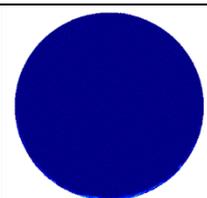
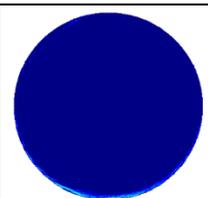
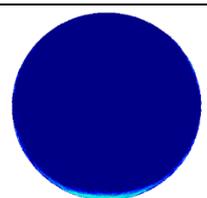
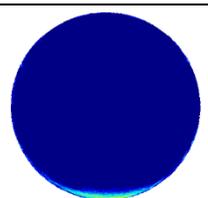
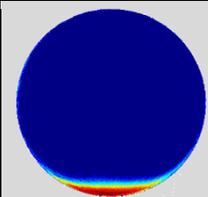
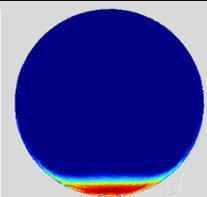
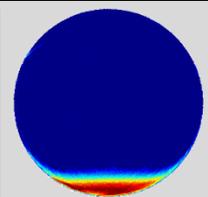
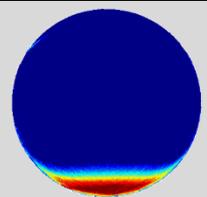
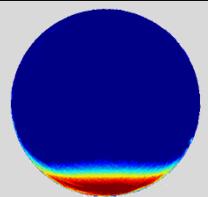
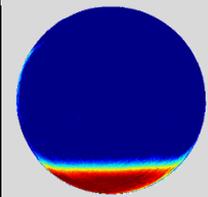
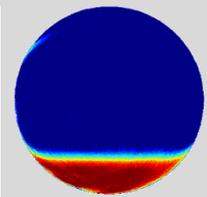
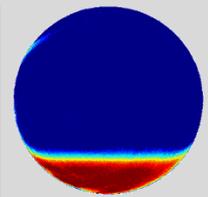
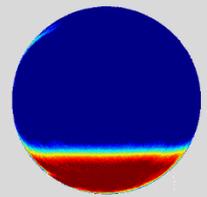
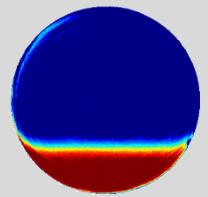
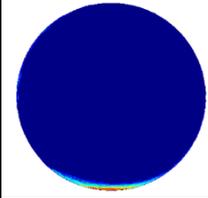
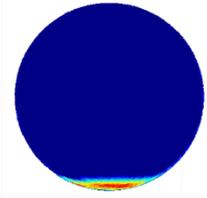
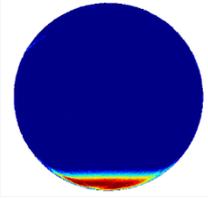
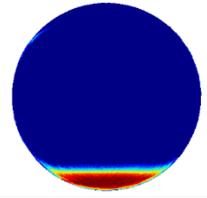
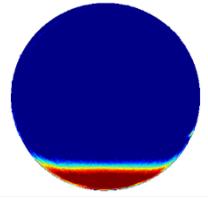
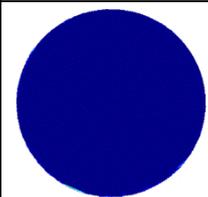
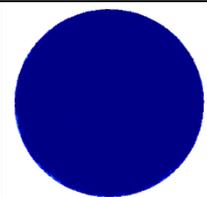
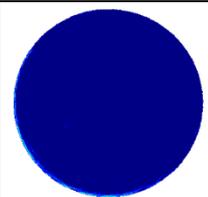
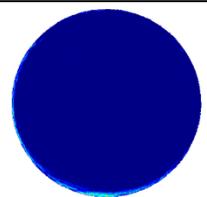
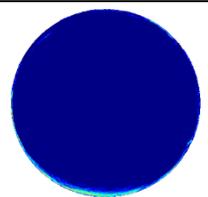
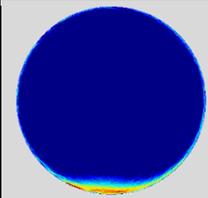
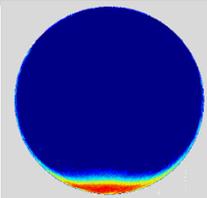
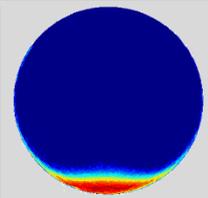
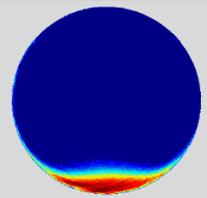
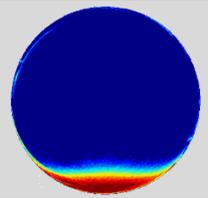
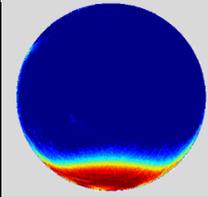
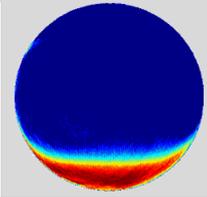
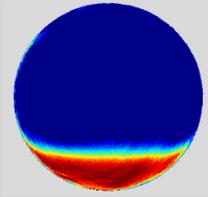
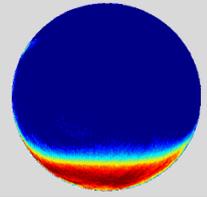
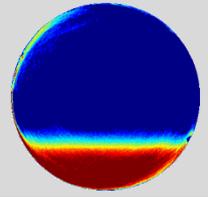
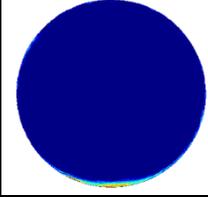
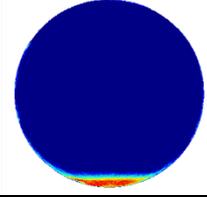
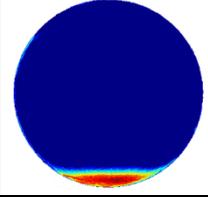
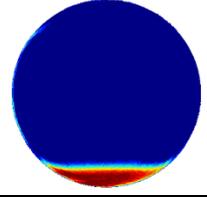
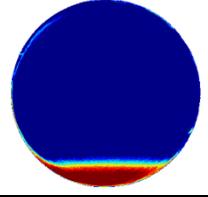
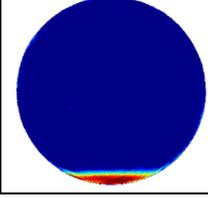
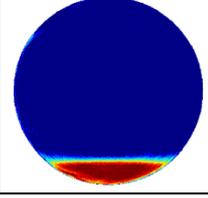
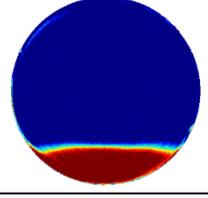
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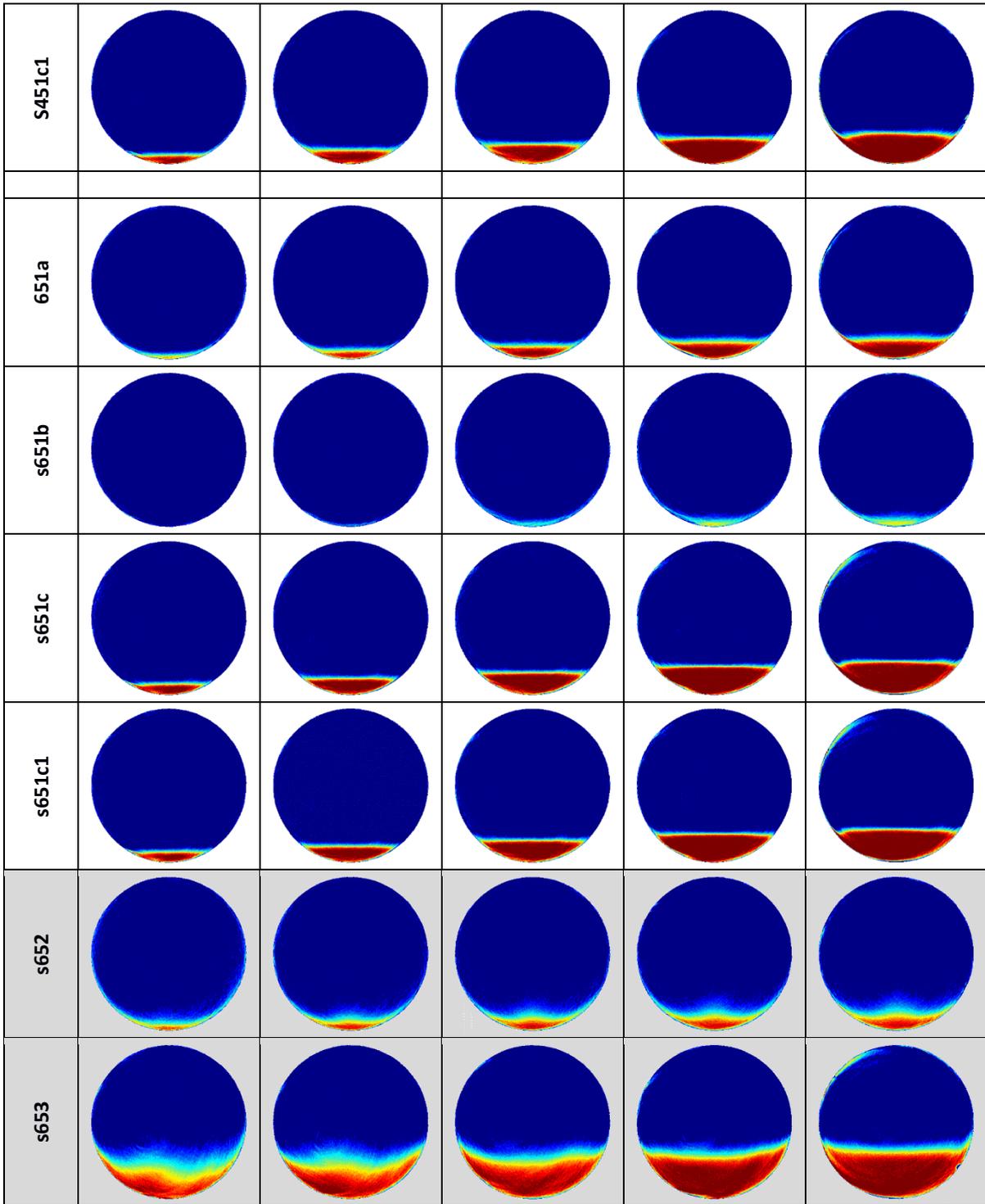
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Table 5: Compilation of reconstructed cross-sectional water-steam distributions within the condenser tube at different pressures and inlet flow rates for saturated steam and water injections.



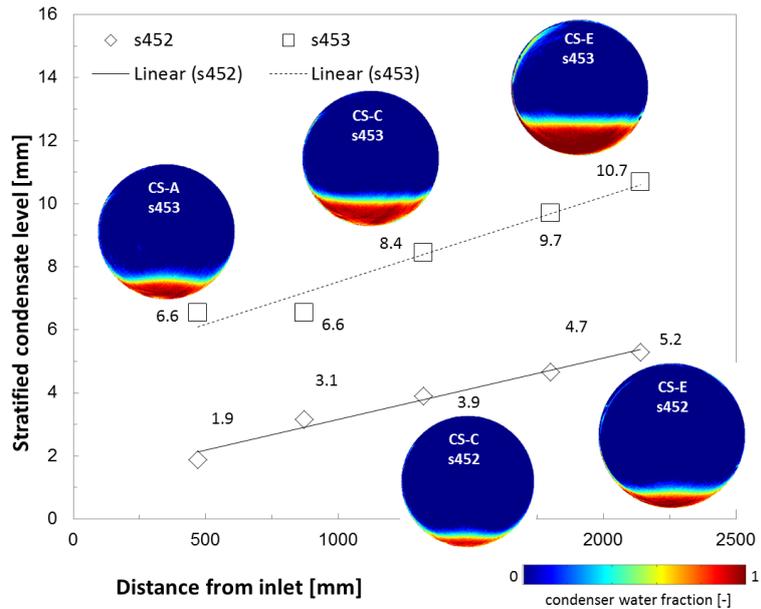
s251					
s252					
s253					
S251a					
s451					
s452					
s453					
s451a					
S451c		n.a.		n.a.	



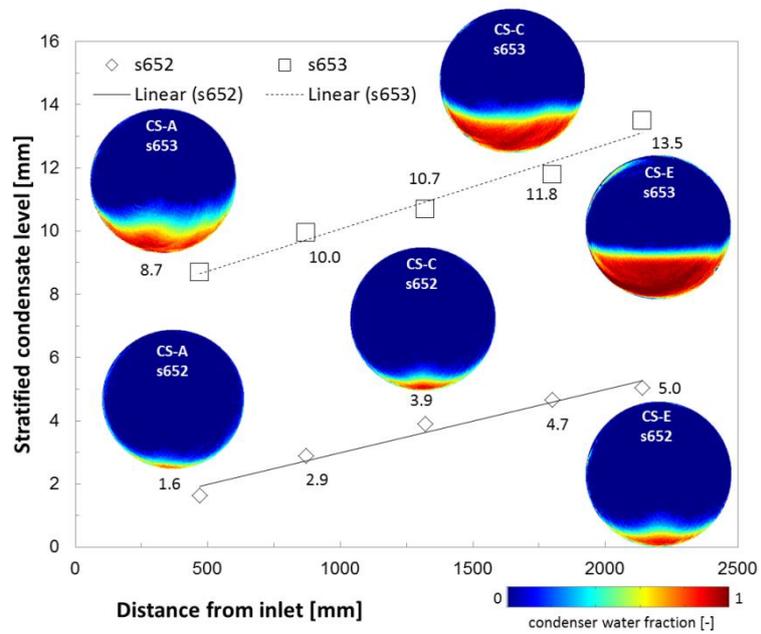
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591 From the obtained condensate/liquid fraction distributions, the stratified condensate levels and their
 592 corresponding transition zone were calculated. As an example the condensation level evolution along
 593 the condenser tube is shown for 45 bar (Figure 16a) and 65 bar (Figure 16b) each for two different inlet
 594 mass flow mixtures. As can be seen, the stratified condensate height follows a linear trend along the
 595 condenser tube length.

596

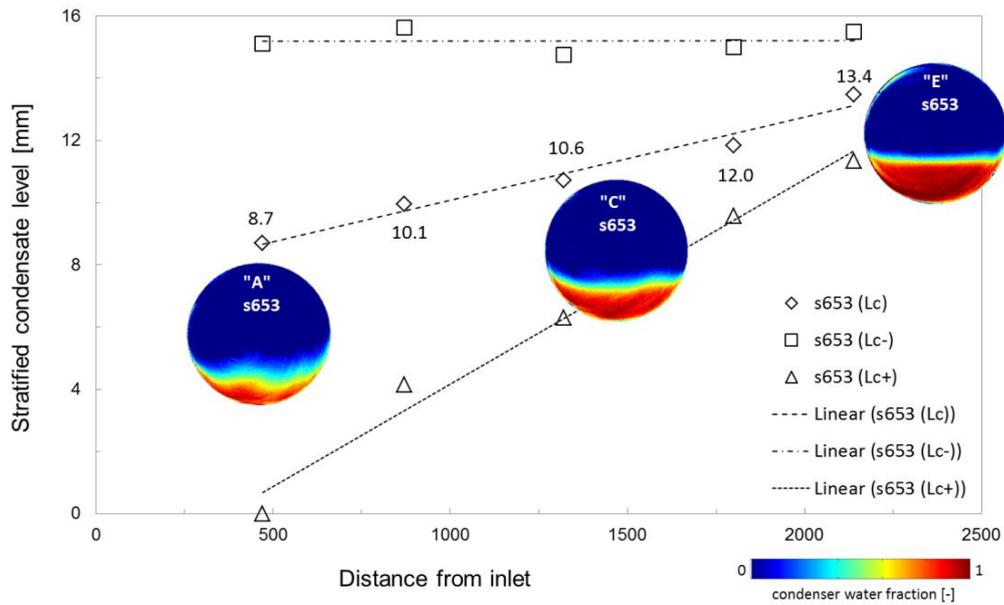


a)



b)

597 **Figure 16:** Determined stratified condensate level L_c in the condenser tube at a) 45 bar and b) 65 bar. ("CS" –
 598 cross-section).



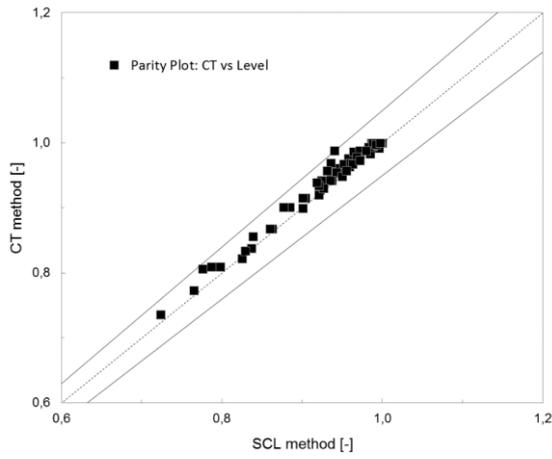
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601 **Figure 17:** Stratified condensate level L_c in the condenser tube at 65 bar (s653) with transition zone (L_c^- and
602 L_c^+) indicators.

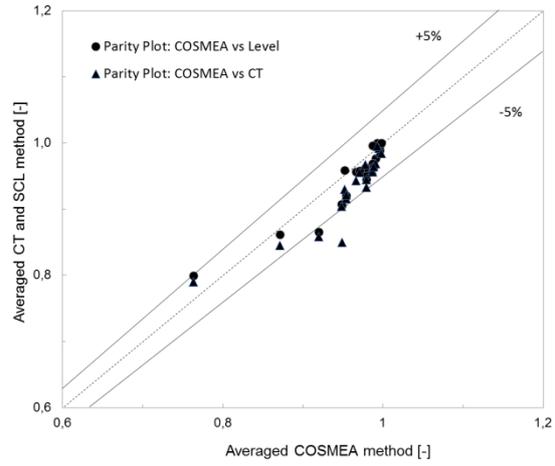
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604 Besides, Figure 16b and Table 5 reveal in most cases a reduction of the steam-liquid interface thickness
605 with the length of the condenser tube which reflects the expected physics, i.e. along the condenser
606 tube the steam volume flow is reduced and so it is its agitating action on the stratified condensate
607 surface. A detailed analysis is shown in Figure 17 for a measurement at 65 bar. All quantitatively
608 investigated values of L_c , L_c^- and L_c^+ are listed in Table 6.

609 Eventually, the total steam fraction from the stratified condensate level (SCL) method (Eq. (22)) are
610 compared with the values directly obtained from the reconstructions (Eq. (20) and Eq. (21)) for each
611 cross-section "A"- "E". As shown in the parity plot in Figure 18a, their deviation is less than 5% which
612 means a good cross-validation for the obtained experimental data. Moreover, the mean total steam
613 fraction ("A"- "E") of both approaches from the CT scans are compared with the mean total steam
614 fraction taken from the three COSMEA facility approaches, as both represent the integral value of the
615 entire test section. The parity plot in Figure 18b shows again a deviation of about 5%.



a)



b)

616 **Figure 18:** Parity plot of the a) mean steam fraction obtained from SCL (stratified condensate level) and CT
 617 method and b) averaged mean steam fraction values obtained from the SCL and the CT method
 618 versus averaged COSMEA method.

619

620
621

Table 6: Measured total steam fractions from CT scans $\bar{\epsilon}_{CT}$ [-] and stratified condensate levels $\bar{\epsilon}_L$ [-] as well as determined condensate levels $L_c^{(-,+)}$ [mm] as determined from the CT images at cross-sections “A”–“E”.

Test #	Cross-section: A					Cross-section: B					Cross-section: C					Cross-section: D					Cross-section: E				
	$\bar{\epsilon}_{CT}$	$\bar{\epsilon}_L$	L_c	L_c^-	L_c^+	$\bar{\epsilon}_{CT}$	$\bar{\epsilon}_L$	L_c	L_c^-	L_c^+	$\bar{\epsilon}_{CT}$	$\bar{\epsilon}_L$	L_c	L_c^-	L_c^+	$\bar{\epsilon}_{CT}$	$\bar{\epsilon}_L$	L_c	L_c^-	L_c^+	$\bar{\epsilon}_{CT}$	$\bar{\epsilon}_L$	L_c	L_c^-	L_c^+
s51	1.000	1.000	x	0.6	x	0.991	1.000	x	1.6	x	0.985	0.992	1.3	2.6	x	0.985	0.983	2.0	3.5	x	0.972	0.973	2.8	4.5	1.3
s52	0.996	0.990	1.4	3.4	x	0.972	0.980	2.3	4.3	x	0.960	0.969	3.0	5.4	1.6	0.950	0.948	4.3	6.2	2.9	0.921	0.920	5.8	7.6	4.5
s51a	0.986	1.000	x	x	x	0.981	1.000	x	0.5	x	0.981	1.000	x	1.3	x	0.986	1.000	x	1.6	x	0.987	1.000	x	2.0	x
s121	0.997	1.000	x	x	x	0.986	1.000	x	1.3	x	0.988	0.996	0.8	2.0	x	0.983	0.992	1.3	2.4	x	0.980	0.985	1.9	3.5	x
s122	0.988	0.995	0.9	2.0	x	0.972	0.988	1.6	3.1	x	0.968	0.978	2.4	4.2	1.1	0.963	0.967	3.1	4.9	1.6	0.955	0.955	3.9	5.8	2.1
s123	0.959	0.975	2.6	4.7	1.3	0.946	0.961	3.5	5.7	2.1	0.936	0.948	4.3	6.8	2.8	0.922	0.927	5.4	7.7	3.7	0.901	0.915	6.0	9.1	4.4
s121a	0.978	0.996	0.8	1.8	x	0.962	0.985	1.9	3.7	1.1	0.956	0.971	2.9	4.4	1.9	0.950	0.955	3.9	5.3	2.8	0.936	0.937	4.9	6.4	4.0
s251	0.998	1.000	x	x	x	0.998	1.000	x	x	x	0.987	1.000	x	1.1	x	0.988	1.000	x	1.6	x	0.992	0.998	0.5	2.1	x
s252	0.958	0.975	2.6	4.3	1.1	0.959	0.963	3.4	5.2	1.8	0.944	0.953	4.0	6.2	2.4	0.937	0.942	4.7	6.9	2.9	0.927	0.927	5.4	7.9	3.4
s253	0.904	0.915	6.0	7.9	4.4	0.901	0.899	6.8	8.9	5.2	0.863	0.868	8.2	10.1	6.6	0.837	0.841	9.3	11.2	7.7	0.787	0.809	10.6	12.6	8.8
s251a	0.993	0.996	0.8	1.6	x	0.989	0.983	2.0	3.1	x	0.965	0.967	3.1	4.5	2.3	0.945	0.946	4.4	5.7	3.4	0.928	0.927	5.4	6.6	4.4
s451	0.998	1.000	x	x	x	0.997	1.000	x	x	x	0.993	1.000	x	0.5	x	0.992	1.000	x	1.1	x	0.997	1.000	x	1.4	x
s452	0.964	0.985	1.9	3.9	x	0.952	0.967	3.1	5.5	x	0.943	0.955	3.9	6.6	1.9	0.935	0.942	4.7	7.4	2.5	0.921	0.932	5.2	8.1	2.8
s453	0.885	0.904	6.6	10.1	4.2	0.877	0.904	6.6	10.0	3.8	0.860	0.862	8.4	11.1	5.7	0.829	0.831	9.7	12.2	7.7	0.776	0.806	10.7	13.4	8.4
s451a	0.996	0.996	0.8	1.4	x	0.982	0.981	2.1	3.3	x	0.972	0.965	3.3	4.4	2.0	0.953	0.944	4.5	5.8	3.4	0.933	0.927	5.4	6.8	4.2
s451c	0.967	0.976	2.5	3.5	1.6	x	x	x	x	x	0.923	0.925	5.5	6.6	4.8	x	x	x	x	x	0.858	0.856	8.7	9.6	8.1
s451c1	0.972	0.976	2.5	3.5	1.6	0.950	0.955	3.9	4.9	3.1	0.931	0.925	5.5	6.4	4.8	0.888	0.888	7.3	8.1	6.6	0.865	0.856	8.7	9.4	7.9
s652	0.940	0.988	1.6	5.3	x	0.936	0.971	2.9	6.7	1.1	0.931	0.955	3.9	8.2	1.5	0.924	0.942	4.7	9.2	1.8	0.918	0.934	5.0	11.1	1.9
s653	0.839	0.856	8.7	15.1	x	0.825	0.825	10.0	15.6	4.2	0.798	0.806	10.7	14.7	6.3	0.765	0.777	11.8	15.0	9.6	0.724	0.732	13.5	15.5	11.3
s651a	0.978	0.995	0.9	2.1	x	0.975	0.978	2.4	3.8	1.4	0.959	0.957	3.8	5.3	2.5	0.940	0.937	4.9	6.8	3.7	0.925	0.922	5.7	7.4	4.3
s651b	0.993	1.000	x	x	x	0.995	1.000	x	0.8	x	0.986	1.000	x	2.0	x	0.981	0.993	1.1	3.7	x	0.985	0.989	1.5	4.2	x
s651c	0.956	0.969	3.0	3.9	2.3	0.934	0.942	4.7	5.5	4.0	0.906	0.909	6.3	7.2	5.7	0.864	0.870	8.1	8.7	7.4	0.836	0.841	9.3	10.0	8.7
s651c1	0.961	0.969	3.0	3.9	2.1	0.941	0.944	4.5	5.4	3.9	0.909	0.912	6.2	7.1	5.5	0.872	0.873	7.9	8.7	7.4	0.838	0.844	9.2	10.0	8.6

622

623 **4.4 Wall heat flux**

624 Beside the condensation rate, the heat flux through the tube wall is a further important parameter for
 625 condensation analysis. For that, we processed the data of the heat flux probe, which gives
 626 circumferentially distributed inner and outer wall temperatures at one axial position near the
 627 condensation tube outlet (Figure 5). Table 7 summarizes the heat fluxes for all condensation tests
 628 (Table 3) together with uncertainties.

629 For further analysis, we included the flume Reynolds number

$$Re_r = \frac{u_{plo} \cdot d_{hy}}{\nu} \quad (25)$$

630 in the table. Here, d_{hy} denotes the hydraulic diameter and ν the kinematic viscosity of the liquid. The
 631 mean velocity of the liquid in the flume is

$$u_{plo} = \frac{\dot{m}_{pli} + \dot{m}_c}{\rho_{plo} \cdot A_r}, \quad (26)$$

632 with the cross-sectional flume area A_r (see Eq. (22)). The hydraulic diameter is

$$d_{hy} = 4 \cdot \frac{A_r}{b}. \quad (27)$$

633 with b denoting the wetted perimeter, i.e. the total length of peripheral liquid-wall contact.

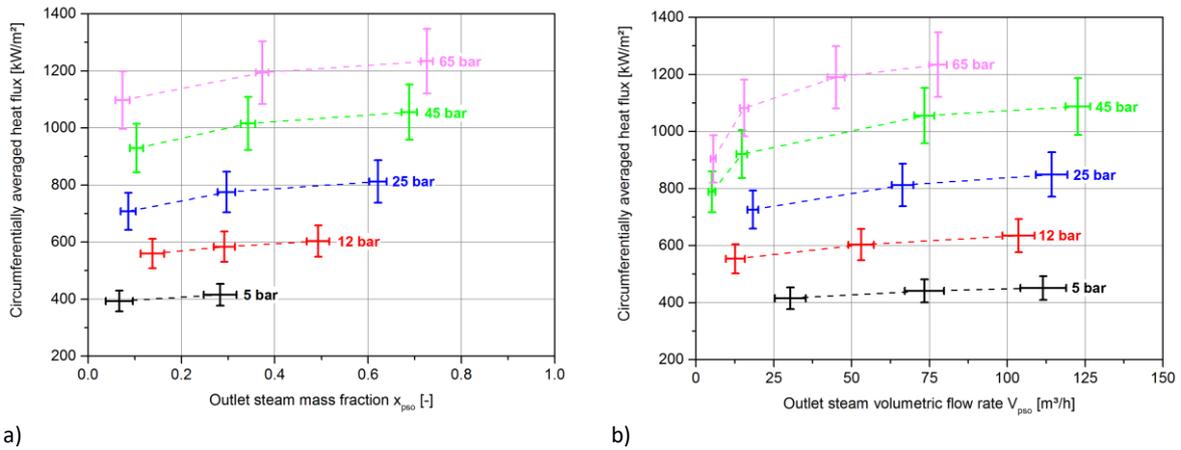
634 **Table 7:** Heat flux along the tube circumference as determined from the heat flux probe data. The second and
 635 the third columns give calculated liquid outlet velocities and Reynolds numbers for experimental points where
 636 the flume could be reconstructed with sufficient confidence, that is, with a height of $L > 1.5$ mm.

Test #	u_{plo} [m/s]	Re_r	$\dot{q}_w \pm \Delta\dot{q}_w$ [kW/m ²]				
			180° (top)	135°	90°	45°	0° (bottom)
s51	1.55	53174	444.4 ±40.4	430.0 ±39.7	413.4 ±37.6	417.7 ±38.6	357.3 ±33.0
s51a			465.3 ±42.3	435.1 ±40.2	430.4 ±39.1	445.6 ±41.1	441.6 ±40.8
s51b			470.4 ±42.7	440.4 ±40.7	437.9 ±39.8	460.6 ±42.5	462.8 ±42.7
s52	0.86	56633	438.4 ±39.9	416.4 ±38.5	409.1 ±37.2	364.2 ±33.7	327.8 ±30.3
s121	4.67	138357	640.8 ±58.2	598.6 ±55.3	591.5 ±53.7	609.7 ±56.3	585.5 ±54.1
s121a	0.97	66643	603.0 ±54.7	573.3 ±52.9	563.5 ±51.2	543.0 ±50.1	467.8 ±43.2
s121b			660.3 ±59.9	617.2 ±57.0	614.4 ±55.8	654.7 ±60.4	648.9 ±59.9
s122	2.27	131324	627.6 ±56.9	589.4 ±54.4	576.6 ±52.3	584.9 ±54.0	539.3 ±49.8
s123	1.45	120657	607.3 ±55.2	580.2 ±53.6	566.4 ±51.5	541.1 ±50.0	495.5 ±45.7
s251			847.8 ±76.9	796.1 ±73.4	786.4 ±71.4	834.6 ±77.0	815.2 ±75.2
s251a	1.21	105188	798.4 ±72.5	748.4 ±69.0	732.6 ±66.5	713.3 ±65.7	621.5 ±57.4
s251b			869.5 ±78.9	829.5 ±76.6	819.8 ±74.4	878.9 ±81.1	865.8 ±79.9
s252	2.95	269560	829.5 ±75.3	775.0 ±71.5	764.0 ±69.3	780.2 ±72.0	732.8 ±67.6

s253	1.46	230284	788.7 ±71.6	750.4 ±69.2	731.0 ±66.4	632.2 ±58.3	648.3 ±59.9
s451			1081.2 ±98.1	1038.0 ±95.7	1024.0 ±92.9	1086.6 ±100.2	1065.2 ±98.3
s451a	1.87	183305	1010.5 ±91.7	936.9 ±86.5	924.3 ±93.9	907.5 ±83.7	816.0 ±75.3
s451b			1097.3 ±99.6	1070.1 ±98.7	1049.0 ±95.2	1130.1 ±104.3	1107.5 ±102.1
s451c	0.72	103294	948.0 ±86.0	879.1 ±81.1	881.5 ±80.0	618.2 ±57.1	602.2 ±55.6
s451c1	0.73	104605	950.0 ±86.2	880.1 ±81.2	882.2 ±80.1	630.1 ±58.1	614.3 ±56.7
s452	5.60	551656	1066.6 ±96.8	1003.4 ±92.5	996.1 ±90.3	1036.0 ±95.5	987.3 ±91.1
s453	2.70	492255	1015.3 ±92.1	940.6 ±86.8	921.9 ±83.7	897.1 ±82.8	899.4 ±83.0
s651			1249.1 ±113.3	1218.3 ±112.3	1193.5 ±108.3	1274.7 ±117.5	1250.1 ±115.3
s651a	2.38	260990	1176.7 ±106.8	1090.4 ±100.6	1080.6 ±98.1	1074.0 ±99.0	989.0 ±91.18
s651b			1228.4 ±111.5	1176.8 ±108.6	1162.9 ±105.5	1219.3 ±112.4	1175.4 ±108.4
s651c	0.83	131147	1095.9 ±99.5	1015.0 ±93.7	1005.2 ±91.2	722.8 ±66.7	651.8 ±60.2
s651c1	0.84	132107	1096.3 ±99.5	1014.9 ±93.6	1004.9 ±91.2	736.7 ±67.9	660.2 ±60.9
s652	8.85	889174	1233.2 ±111.9	1181.4 ±109.0	1172.1 ±106.4	1221.5 ±112.6	1169.1 ±107.8
s653	3.14	740063	1173.8 ±106.5	1085.3 ±100.1	1075.4 ±97.6	1099.3 ±101.4	1086.2 ±100.2

637

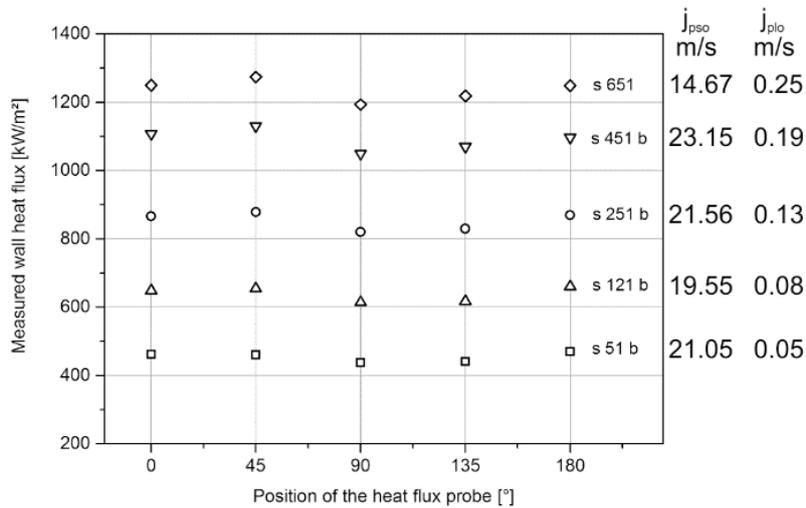
638 To give an overview about the variation of heat flux regarding to the operational parameters the five
639 circumferential heat flux values per test are averaged using weighting factors which account for
640 assumed symmetry of the flow. Thus, top (180°) and bottom (0°) values are weighted with 1/8 while
641 the other values are double-weighted with 1/4. These averaged heat fluxes are shown in Figure 19 as
642 a function of the outlet steam mass fraction x_{ps0} and subject to the outlet steam volumetric flow rate
643 \dot{V}_{ps0} respectively. Other than the condensation rate we relate the heat fluxes to outlet parameters since
644 the local measurement position was near the tube outlet. Similar to the condensation rates also in
645 these plots the influence of the system pressure is clearly visible. The higher the pressure the higher
646 the heat fluxes due to the increasing temperature difference between the inner and outer tube wall.
647 Furthermore, with a higher amount of liquid in the tube the heat flux decreases too. Eventually, a
648 decreasing steam volumetric flow causes a decreasing averaged heat flux due to a significant reduction
649 of the heat transfer coefficient at the inner tube wall.



650 **Figure 19:** Circumferentially averaged wall heat flux as function (a) of the outlet steam mass fraction and (b) of
 651 the outlet steam volumetric flow.

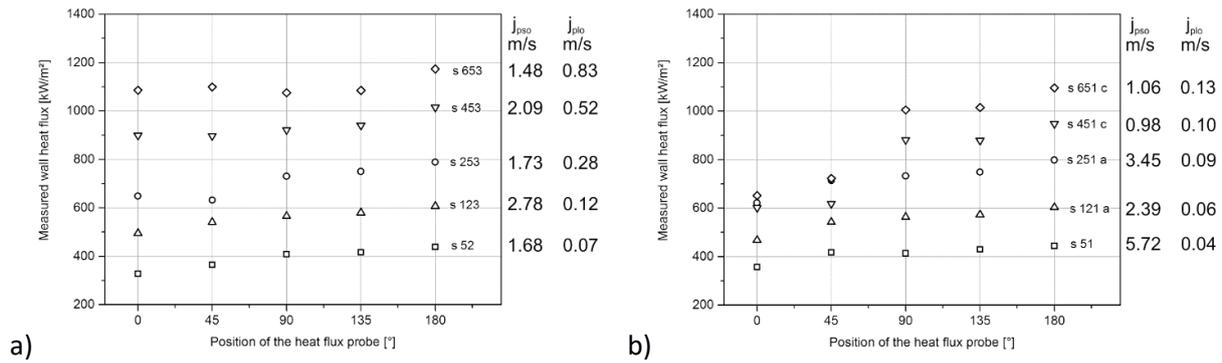
652

653 As a representative example the circumferential heat flux distributions of the tests at highest outlet
 654 steam flow are shown in Figure 20. Due to the high condensation rates a condensate flume forms at
 655 the bottom of the tube. According to the common opinion the heat flux through the flume should be
 656 significantly lower than in regions of film condensation. However, we can see an almost equal value
 657 along the circumference. At the bottom position (0°), where definitely a liquid flume exists, the heat
 658 flux is similar to or even higher than in the top position. In addition, the high values at 45° are
 659 remarkable. A plausible explanation is the high steam velocity in the tube (e.g. j_{pso}) that agitates the
 660 liquid in the flume by strong interfacial shear. The resulting turbulence in the liquid then promotes
 661 heat transfer between the steam-liquid interface and the tube wall.



662

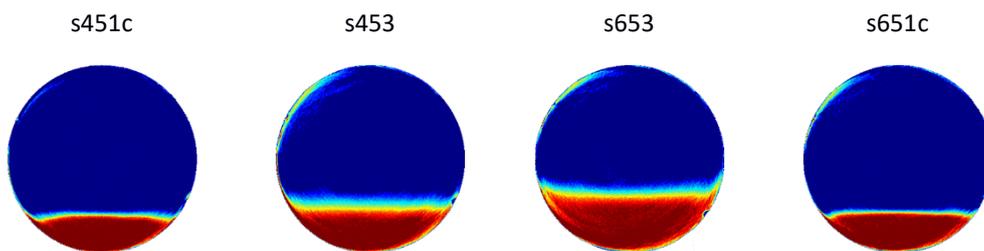
663 **Figure 20:** Circumferential wall heat flux at five positions; tests at the highest outlet steam flow and pressures
 664 5-65 bar.



665 **Figure 21:** Circumferential wall heat flux a) at high liquid and low steam outlet superficial velocities and b) at
 666 low liquid and steam outlet superficial velocities.

667

668 The comparison in Figure 21 gives a further indication of this. Figure 21a depicts the tests with the
 669 highest amount of liquid injection and Figure 21b tests at the same pressure levels but without liquid
 670 flow at the tube inlet. That is, the outlet liquid flow is much higher for tests in Figure 21a as the
 671 condensation rate is similar. For the 45 bar and 65 bar tests the heat flux is almost equal at all angles
 672 at high outlet liquid velocity (tests s653 and s453) while it is considerably lower in positions 0° and 45°
 673 at low outlet liquid velocity (tests s651c and s451c). The Reynolds numbers in Table 7 indicate, that
 674 the flume flow is never laminar ($2500 \leq Re_{r,crit} \leq 4000$). There are three thermal hydraulic effects
 675 that can explain this behavior. 1) At higher Reynolds numbers the turbulent heat transfer within the
 676 liquid intensifies in such a way that the circumferential heat flux is equalized. Such an explanation is
 677 in-line with e.g. surface renewal theory [47], where the surface renewal rate is related to turbulence
 678 intensity. 2) With higher interfacial shear interfacial waves build up towards a slug flow which lead to
 679 increased heat transfer. According to the flow map of Tandon (Figure 23), case s651c starts as pure
 680 steam and ends up in a stratified flow. On the other hand, case s653 starts as annular flow and ends
 681 on the boundary of transition region to slug flow. A similar trend is observed for cases s451c and s453.
 682 This observation has consistency with the CT data. The CT profiles of these two experiments at cross
 683 section E reveal that case s651c is fully stratified with a low interface thickness while case s653 shows
 684 a broader and hence wavier interface (Figure 22). Thus, interfacial waves (case s453 and s653) may
 685 contribute to a homogenization of the circumferential heat flux by intermittent wall rewetting.



686 **Figure 22:** CT profiles at cross-section "E" for 45 bar and 65 bar experiments.

687 3) The different outlet liquid velocities (listed in Table 7) lead to different residence times of the
 688 condensate in the cooled tube. Longer residence times then may lead to a stronger sub-cooling at 0°
 689 and 45°. A deeper insight can only be gained by denser instrumentation, higher time resolution of e.g.
 690 X-ray tomography or validated CFD simulations.

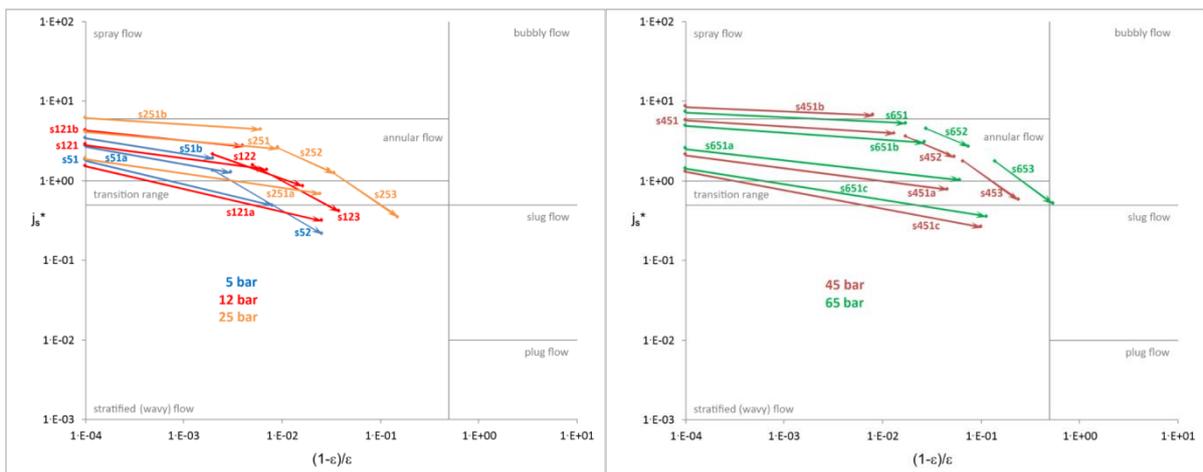
691 4.5 Flow classification

692 The observed flow patterns were compared with theoretical correlations for condensation in
 693 horizontal tubes using the established flow map of Tandon et al. [37]. For clarity, steady-state tests are
 694 divided in two groups: measurements at low pressure of 5, 12 and 25 bar (see Figure 23, left) and
 695 measurements at high pressure of 45 bar and 65 bar (see Figure 23, right). To compare all performed
 696 measurements the dimensionless superficial steam velocity

$$j_s^* = \frac{j_s}{\sqrt{g \cdot d_i}} \sqrt{\frac{\rho_s}{\rho_l - \rho_s}} \quad (28)$$

697 as introduced by Wallis & Dobson [38] is plotted against the ratio of the volumetric liquid and steam
 698 fraction. Thereby, j_s is the superficial steam velocity, g the gravitational acceleration, ρ_s and ρ_l the
 699 steam and liquid density respectively and d_i the inner tube diameter. Using these parameters gives
 700 an almost linear classification of the flow regimes in the logarithmic plot.

701



702 **Figure 23:** Steady-state condensation experiments within the horizontal flow map of Tandon et al. [37].

703

704 Each experimental point is plotted as an arrow with the starting point (left) that corresponds to the
 705 inlet conditions and an end point (right) that relates to the outlet conditions. As tests with an initial
 706 void fraction of 1 (pure steam) cannot be fit into the logarithmical scale we started the abscissa at 10^{-4} .

707 The outlet conditions have been calculated from the inlet condition and the condensation rate
708 calculated by the 2nd approach.

709 Figure 23 shows that most of the tests fall into the annular and transition flow regime, which is due to
710 operational limits of the COSMEA test rig. As we control the pressure by steam feed, as described
711 above, even at low steam mass flow rates we have annular or transition flow due to the high density
712 ratio of liquid and gas, even at 65 bar pressure. Future experiments will be run with a facility
713 modification allows also regimes with low steam fraction. However, as the largest contribution to total
714 condensation is at high steam quality, these experiments are a good basis for heat transfer analyses.
715 Nonetheless, some tests were found to end in stratified flow with a high liquid level as it was confirmed
716 by the X-ray CT water level determination. Examples for that are the tests s52, s121a, s253, s451c and
717 s653.

718

719 **5. Conclusions**

720 In this paper, the thermal hydraulic test facility COSMEA was introduced that allows single effect
721 studies for high-pressure steam condensation in an inclined tube at up to 65 bar pressure and
722 saturation conditions, i.e. 281 °C. Boundary conditions are given by adjustable steam and water feed
723 rates and a forced convective secondary cooling with water in counter-current flow. We performed
724 experiments in steady-state conditions at pressures of 5, 12, 25, 45 and 65 bar and varying inlet steam
725 qualities. The wall heat flux was measured at five circumferential positions in a given axial position
726 using a custom-made heat flux probe. The averaged cross-sectional flow morphology was non-
727 intrusively investigated by proprietary X-ray CT system at a spatial resolution of approx. 0.5 mm. None
728 of the operating scenarios revealed detectable condensate films, meaning, that their thickness must
729 be less than 0.5 mm. The data has been used for validation of CFD models and simulations, which is
730 subject of the second part of this paper.

731

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