INITIATION OF WATER TURBULENCE IN CONCURRENT TWO-PHASE FLOW

Laurinavičius D.* and Šeporaitis M.
*Author for correspondence
Laboratory of Nuclear Installation Safety
Lithuanian Energy Institute
Breslaujos str. 3, LT-44403 Kaunas
Lithuania
E-mail: darius.laurinavicius@lei.lt

ABSTRACT

Stratified horizontal two-phase flow with condensation attracted attention in attempting to avoid water hammer problem during reactor LOCA/LOOP (loss of coolant/offsite power) accidents. Less complicated cases of direct steam condensation to stationary water were studied before. However, two-phase flow problem becomes even more difficult due to transfer of steam momentum to interface and its positive feedback to condensation intensity. Despite numerous experimental and analytical studies, the understanding level and modelling capabilities of condensation in two-phase flow are still moderate. Steam side heat and mass delivery to interface are limited by speed of sound. Waterside heat transfer depends on convection intensity and can vary by several orders of magnitude. That is why it is important to perceive the processes, which impede and accelerate the heat removal from interphase to water bulk. Therefore, in order to understand physics of the direct steam condensation better, the efforts are being made to investigate different two-phase flow conditions using modern measurement techniques. Water temperature field was investigated by using infrared camera at different conditions of condensing two-phase flow inside the rectangular short and narrow horizontal channel (1.00 m long, 0.02 m width and 0.10 m height). The velocities of water were 0.014 and 0.056 m/s ($1000<Re<2700$), while steam flowed at 8 and 12 m/s. As it was expected, increasing steam velocity accelerated the condensation intensity; also, a localized intensification of turbulence was observed. The location of this entire water cross-section penetrating turbulence depends on flow conditions. The possible explanation of this phenomenon is that it may be a consequence of down-flow building-up velocity and temperature gradients in the water. That means less viscous and faster flow of near surface water layer. It facilitates the surface renewal and accelerates condensation, and the local steam velocity near the interface becomes higher. These effects have positive feedback on each other, and turbulence spreads to water bulk.

INTRODUCTION

Instabilities of two-phase flows are met in different kinds of industries like electronics, chemicals, oil plants, space, thermal and nuclear energy. Dynamic instabilities of two-phase flow can be divided into four main types [2, 3]: density wave type oscillations (DWO), pressure-drop type oscillations (PDO), acoustic oscillation (AO) and thermal oscillation (TO). A simplified description of DWO is based on the delays induced by the transient distribution of pressure along the pipe [4]. PDOs are actually complex instabilities, since they are dynamic instabilities triggered by a static instability. These kinds of instabilities in order to occur need a compressible volume upstream the heated section [5]. AO instability is triggered by the amplitude film thickness variation, bubble collapse, which in turn induces a change in pressure. The amplitude of the acoustic oscillations is generally small with frequencies varying in the range 10–100 Hz [6]. TOs are characterized by large amplitude fluctuations in the heated wall temperature. The flow oscillates between annular flow, transition boiling and droplet flow at a given point, and thus produces large amplitude temperature oscillations [2]. However, these are just some of the many types of instabilities.

Liu [7] concluded that stratified/non-stratified flow transition during condensation is one of the most significant flow pattern transitions, which have been drawing great deal of attention for several decades. However, knowledge about two-phase flow instabilities in condensing systems is still insufficient. Therefore, it is important to conduct more analytical investigations in order to understand the different mechanisms related to instabilities in condensing systems [1].

Lim [8] studied condensation of steam on a subcooled water layer in a co-current horizontal channel flow at atmospheric pressure. The results showed that the heat transfer coefficients varied from 1.3 to 20.0 $kW/m^2 \cdot K$, which increased with the increase of steam flow rates and water flow rates. The correlation of the average heat transfer coefficient and condensation rate for wavy interface flow was obtained as a function of inlet conditions and downstream distance. Lee [9] investigated local thermal hydraulic two-phase flow characteristics. The measurements of the temperature and velocity distribution under co-current steam-water stratified flow conditions showed that the condensate is more effectively propagated into the lower region of the water layer when the steam flow velocity increased.

The study of steam-water interaction is important, because the unstable nature of the interface may result into the formation of hydrodynamic instabilities and the rapid condensation of steam across the interface on macro scale may lead to condensation induced water hammer inside the piping system of nuclear power plants [11].

The aim of this experimental investigation is to examine steam velocity and condensation dependent turbulence in a water layer of stratified two-phase flow. This paper presents and analyses 2D temperature field results obtained by infrared camera. Temperature fields and their sequences allow evaluating heat removal process from interphase area to water bulk and herewith identifying turbulence in water.

NOMENCLATURE

u_s	[m/s]	Superficial velocity of a given phase
V	[m^3/s]	Volume flow rate of the phase
S	[m^2]	Cross sectional area
ΔH	[kJ/kg]	Difference of enthalpy
h	[kJ/kg]	Enthalpy
A	[m^2]	Nominal area of steam and water interface
W	[kg/s]	Mass flow rate
H	[J/kg]	Enthalpy
$\dot{m}$	kg/s	Mass flow rate
T	[K]	Bulk temperature
b	[m]	Width of channel
Δ	[-]	Difference
Q	[W]	Heat flux to water layer
x/h		Axial distance over water level

Subscripts

cond	Condensation
g	Steam (gas)
l	Water (liquid)
s	Saturation
env	Environment

METHODOLOGY

The special experimental facility is used in order to investigate turbulent effects inside water layer by observing heat removal from water interface to bulk in stratified concurrent two-phase flow (Figure 1). The length of rectangular channel is 1.19 m, width 0.02 m and height 0.1 m. The sustaining of the test section geometry is very important, and it is made from 10 mm thick stainless steel to be stiff enough. In order to limit heat loses and decrease steam condensation on the inner walls, they were thoroughly insulated. The infrared (IR) optical windows (Spinel $MgAl_2O_4$) were installed in the channel sidewall at four different positions with distance of 0, 140, 280 and 760 mm from the beginning of steam-water interface. Heights of the windows are 20 mm, widths – 1000 mm.

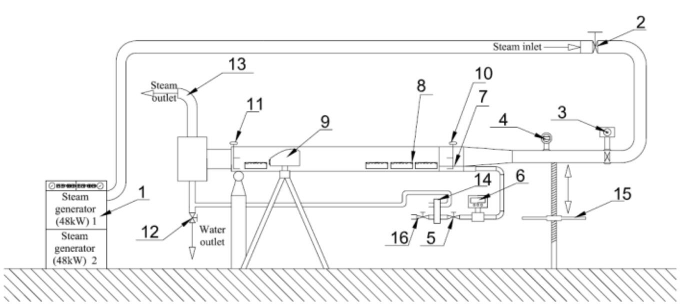


Figure 1. Experimental facility. 1 – steam generator (96 kW), 2 – steam inlet ball valve; 3 – steam flow control valve; 4 – steam flow rate meter; 5 – water flow control manual valve; 6 – water flow rate meter; 7 – separation plate; 8 – optical IR windows (Spinel $MgAl_2O_4$); 9 – infrared camera SC-5000; 10, 11 – K type thermocouples; 12 – water outlet; 13 – steam outlet; 14 – heat exchanger, 15 – regulator of channel inclination, 16 – cooling water flow control valve

At the entrance of rectangular channel, a separation plate is installed, which makes start steam and water interface smooth (Figure 1, 7). Water and steam inflows before entering the test section are stabilized in installed comb-like inlet structures. Water level is maintained by using weir (Figure 1) and changing the inclination angle of the channel (Figure1, 15). Steam and water inflow rates were measured using PRO-Wirl W vortex type and ISOMAG ML 201 electromagnetic flow meters (Figure 1, 4, 6); the inlet and outlet temperatures were measured by K type thermocouples (Figure 1., 10, 11). Water temperature fields measured and recorded using IR camera SC-5000 (Figure 1, 9). Water emission coefficient in water is very high (~0.97 in 2.5-5.1 µm band). Therefore, by IR method, water temperature field can be measured, and heat carrying turbulence is observed only within 30 µm distance from internal wall of optical window (Figure 2). However, 25 points/mm^2 spatial resolution and 50Hz frame rate of temperature field measurement allow examining dynamics of the heat conduction in water by convection. Water temperature profiles were taken at four different channel locations (x/h = 7.6; 13.2; 18.8 and 38.0). Each IR measurement sequence conducted as 30 s film consisting of 1500 frames, which were averaged for each corresponding temperature profile.

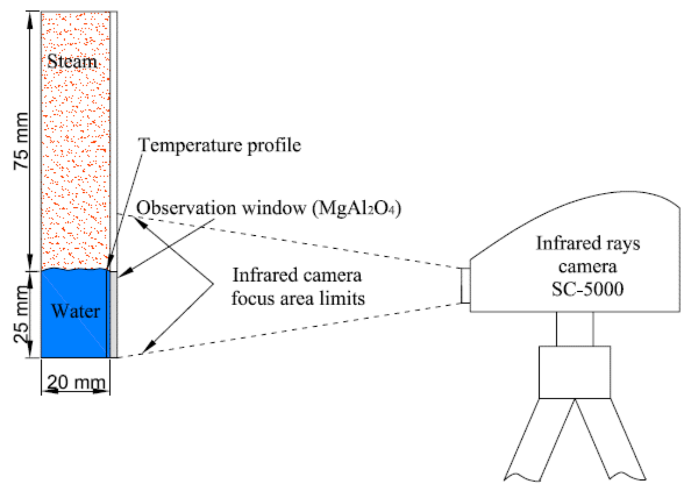


Figure 2. Scheme of measurement of water temperature fields

Measurements were made in different combination of steam and water inflow velocities. Steam velocity was 8 and 12 m/s, while that of water was 0.014 and 0.056 m/s. Inlet temperature was set to 383 K (110°C) and 298 K (25°C) for steam and water, respectively. The pressure inside channel was close to the atmospheric ~ 0.1 MPa.

PROCESSING OF RESULTS

For further analysis of experimental data, the steam and water superficial velocities were calculated (Table 1):

$$u_s = \frac{V}{A} \qquad (1)$$

Table 1. Steam and water superficial velocities

Velocity, m/s	Superficial velocity, m/s
Water	
0.014	0.0035
0.056	0.0140
Steam	
8	6.0
12	9.0

Flow regimes were classified according to the superficial steam and water velocities (Figure 3). Stratified smooth and wavy flow regimes were observed in the beginning of the channel test section (marked as yellow zone). Moving further in axial direction, flow regime toggles depending on local near interfacial velocities.

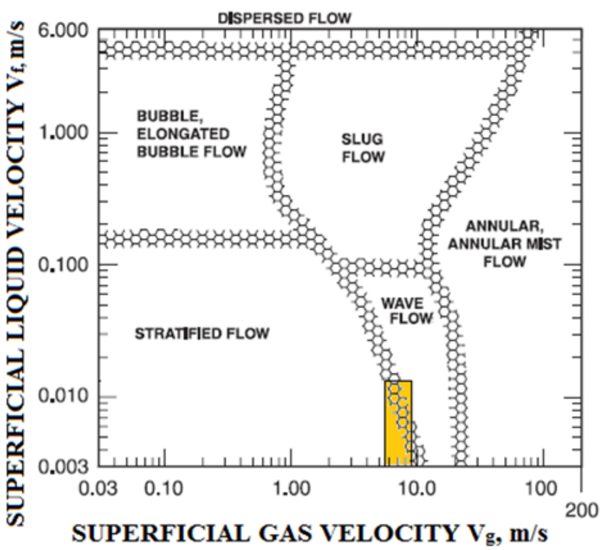


Figure 3. Flow pattern map [10]

In order to evaluate the efficiency of heat removal from interface to water bulk, the condensing steam flux was calculated:

$$\Phi_{m,cond} = \frac{\dot{m}_{cond}}{A}\left[{}^{kg}\!/_{s \cdot m^2}\right], \qquad (2)$$

where condensate rate:

$$\dot{m}_{cond} = \frac{Q_{cond}}{\Delta H_{cond}}[\text{kg/s}], \qquad (3)$$

and heat flux from steam to water:

$$Q_{cond} = \Delta H_l \cdot \dot{m}_l + Q_{env}[W], \qquad (4)$$

where Q_{env} is heat loss to environment estimated to be 150 W (by test).

TRENDS AND RESULTS

From the beginning of channel (x/h = 0) to x/h = 6.4, axial position phases were separated by plate. Direct interaction of steam and water has begun only from x/h = 6.4. During condensation of steam on subcooled water surface, phase transition heat and hot condensate enter the interface. Due to rising water temperature, its viscosity and density decrease. Buoyant force keeps denser water at the interface and builds steep temperature gradient. Saturated temperature of the surface water restricts condensation by thermal diffusion in water layer. However, the axial steam flow drags interface, and steep velocity gradient builds in the water, too. Hot and less viscous interfacial water accelerates even better until flow regime transits from laminar to turbulent. Turbulence assists condensation by removing heat from interface. Condensation, by thinning steam velocity boundary layer, increases local steam velocity at the interface in the axial direction. This positive feedback between condensation and momentum transfer to the interface spreads turbulence deeper into water bulk. The effect was called self-initiation of water turbulence in condensing two-phase flow.

Thermal pictures of water flow at four different positions (x/h = 7.6; 13.2; 18.8; 38.0) and different steam (8, 12 m/s) and water (0.014, 0.056 m/s) flow velocities are demonstrated in Figures 4, 5, 7, 8. Water flows from the left to the right side, temperature of the colour scale differs in all cases because of the better pattern identification.

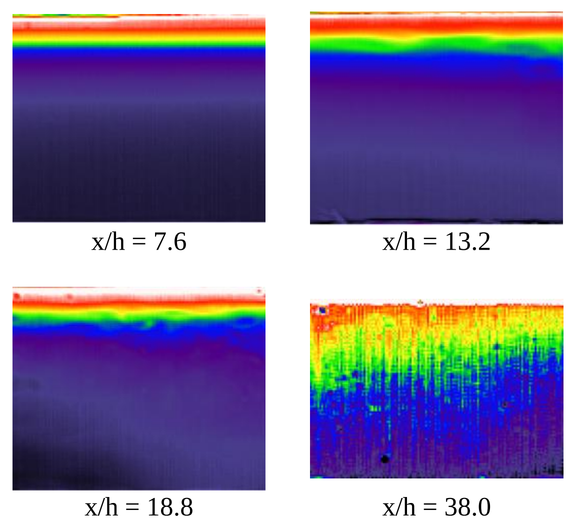


Figure 4. Water thermal pictures. Velocities: steam 8 m/s, water 0.014 m/s

From thermal pictures, it is seen that turbulent self-initiation effect develops gradually. At x/h = 7.6 (Figure 4), the flow is laminar. Further at x/h = 13.2, water in near surface area begins to fluctuate, and it tends to propagate deeper. Complete self-initiation of turbulence occurs at x/h = 18.8 position. This proves water temperature profiles (Figure 6). Intensified heat transport from interphase to water bulk enhanced the steam condensation flux twice – from 0.018 to 0.036 kg/s·m^2 (Figure 10). While water flow is laminar, the temperature profiles are almost the same (Figure 6. 8 m/s, x/h = 7.6 and 13.2); therefore, when flow transits from laminar to turbulent (self-initiation of turbulence), water temperature increases rapidly (Figure 6. 8 m/s, x/h = 18.8). The most effective heat transport zone is between 0.85 and 1.00 relative height of water level, where the temperature profiles are close to linear.

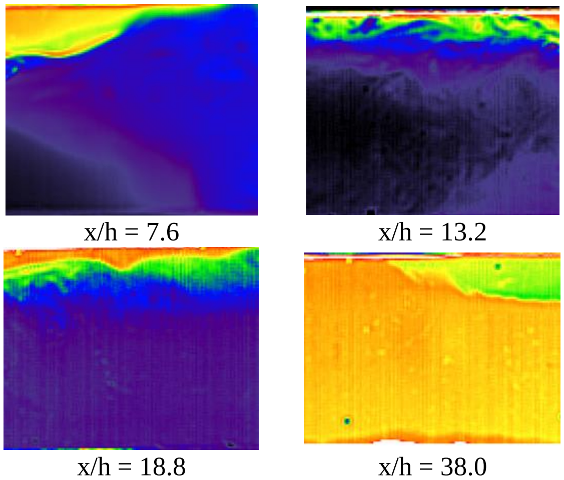


Figure 5. Water thermal pictures. Velocities: steam 12 m/s, water 0.014 m/s

By increasing velocity of steam to 12 m/s, the interface becomes wavy. The beginning of turbulence self-initiation flow moves close to the beginning of the channel. In Figure 5, x/h = 7.6 turbulence penetrates the entire water layer except the dark zone in the left corner, where water flow is laminar and very slow. The substantially higher velocity of water at the interface forms large vortex that entrains stagnant cold water from the corner.

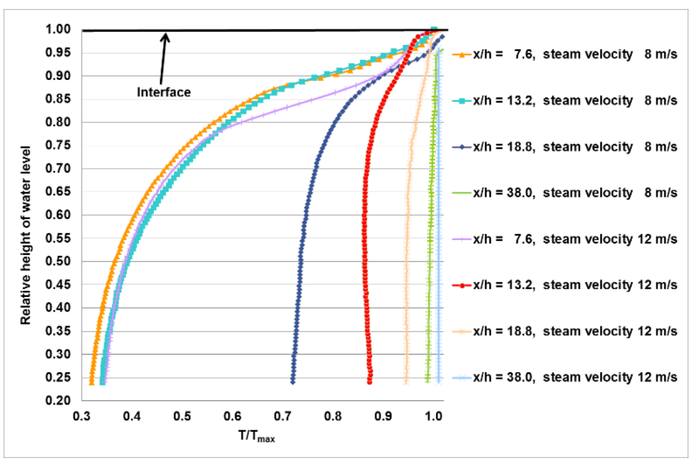


Figure 6. Temperature profiles in different axial positions (inflow: water 0.014 m/s, steam 8 and 12 m/s)

When steam velocity is 12 m/s, at x/h = 7.6 position (Figure 6), the temperature profile shows that heat removal from interface is going much more effectively than in 6 m/s (x/h = 7.6 and 13.2) case. However, deeper zone temperature remains the same. At the next testing position (x/h = 13.2, 12 m/s), the temperature profile seems counterintuitive because of the higher temperature in the middle than at the bottom. However, in Figure 5, it is shown that a large vortex results in a backflow at the depth. The vortex transports hot water down from the interface. Intensified heat transport from the interface results in more than twice increased steam condensation flux between intervals x/h 0-13.2 and 0-7.6 (Figure 10. 0.014 m/s, 12 m/s). At following x/h = 18.8 and 38.0 positions, all the water has already reached near saturation temperature; thus, aggregated steam condensation flux increases only slightly (from 0.042 to 0.046 and 0.049 kg/s·m^2). It can be stated that the main heat load to the water took place mainly in this self-induced turbulence region.

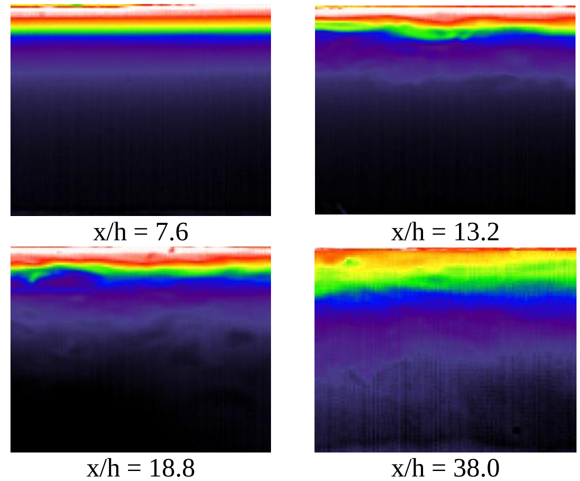
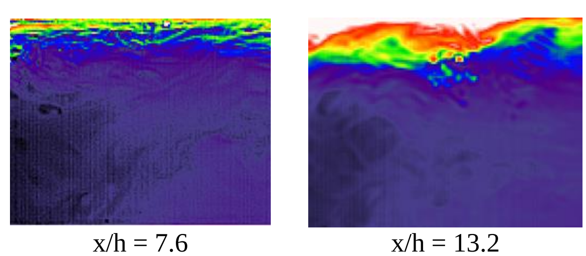
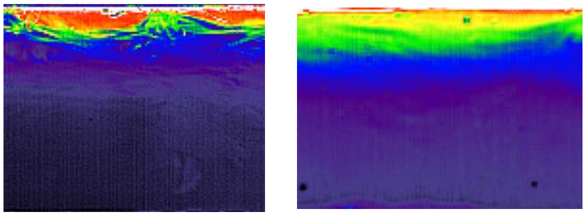


Figure 7. Water thermal pictures. Velocities: steam 8 m/s, water 0.056 m/s

Comparing Figures 4 and 7, it can be seen that at x/h = 7.6, water flow seems similar, and condensed steam mass flux grows almost directly proportional to water inflow increase (Figure 10. 8 m/s). The water velocity is four times higher and flow regime is early transitional. Even weak water mixing greatly assists condensation. At x/h = 18.8, the turbulence spreads slightly deeper into water flow (Figure 7). As it is shown in Figure 10: steam condensation flux increases from 0.063 to 0.070 kg/s·m^2. It can certainly be stated that self-initiation of turbulence occurs between x/h 18.8 and 38.0, because the condensation flux jumps from 0.070 to 0.132 kg/s·m^2. Consequently, by increasing water velocity, the position of turbulence self-initiation is moved farther. However, the temperature in Figure 9 shows that at x/h = 38.0, water remains significantly subcooled.

When steam and water inflow velocities are highest (12 m/s and 0.056 m/s), self-initiation of water turbulence begins a little farther, too. Temperature profiles at x/h = 13.2 in Figures 6 and 9 are similar in shape, but differ in magnitude. It is mainly because of the different water mass inflow to the channel. But at 12 m/s steam inflow, the condensing flux grows only by two times, while at 8 m/s, quadrupling of water inflow results in 3.5 times bigger condensing flux. In general, the condensation flux, at highest steam and water inflows, grows gradually over all cross-sections. Small condensation boost can be noticed only between x/h = 13.2 and 18.8.





x/h = 18.8 x/h = 38.0

Figure 8. Water thermal pictures. Velocities: steam 12 m/s, water 0.056 m/s

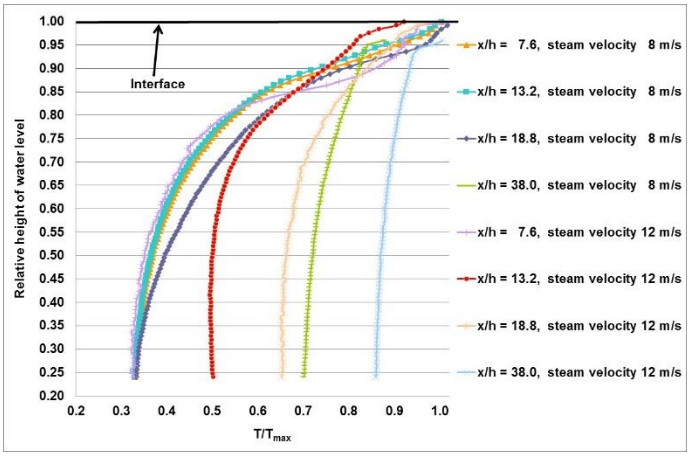


Figure 9. Dependence of water temperature profiles on different steam inflow velocities and distances from the entrance of the channel, when water velocity is 0.056 m/s (100 l/h)

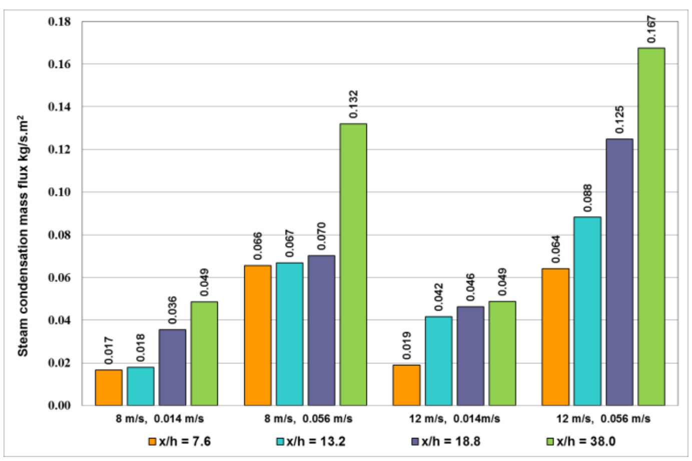


Figure 10. Steam condensation mass flux at different places of the rectangular channel and different velocities of steam and water

CONCLUSION

The heat transport from water interface to bulk enhancing turbulence, initiated by steam-water interaction and accelerated by condensation, was investigated in horizontal condensing two-phase flow.

The analysis of the results shows that steam flow of much higher velocity heats up and drags thin near surface water layer. The steep gradients of viscosity and velocity build up at water surface. Then, the laminar water flow near the interface transits to turbulent, and positive feedback between condensation and momentum to interface amplifies interphase interaction. This mechanism leads to self-initiation and spread of turbulence in the water. Increasing velocity of steam moves this process closer to the channel entrance, while increasing velocity of water – further away.

The knowledge related to dynamics of water turbulence self-initiation in condensing two-phase flow allows lowering the risk of water hammer inside the piping system of nuclear power plants.

References

[1] Ruspini, L. C., Marcel, P. M. and Clausse A. Two phase flow instabilities: A review, *International Journal of heat and mass transfer*. Vol. 71. 2014. pp. 521-548

[2] Kakac, S. and Bon, B. A review of two-phase flow dynamic instabilities in tube boiling systems. *International Journal of heat and mass transfer.* Vol 51. 2008. pp. 399-433

[3] Mosdorf, R. and Grybowski, H. Dynamics of pressure oscillations in flow boiling and condensation in the minichannel, *International Journal of heat and mass transfer*. Vol. 73. 2014. pp. 500-510.

[4] Dorao, A. A. Effect of inlet pressure and temperature on density wave oscillations in a horizontal channel, *Chemical engineering science.* Vol. 134. 2015. pp. 767-773

[5] Chiapero, E. M., Fernandino, M. and Dorao, C. A. Experimental study of pressure drop oscillations in parallel horizontal channels, *International Journal of heat and fluid flow*. Vol. 50. 2014. pp. 123-133

[6] Tadrist, L. Review on two phase flow instabilities in narrow spaces. *International journal of heat and fluid flow*. Vol. 28. Issue 1. 2007. pp. 54-62

[7] Liu G., Wang Y., Zang G. and Zhao H., Viscous Kelvin–Helmholtz instability analysis of liquid–vapor two-phase stratified flow for condensation in horizontal tubes, *International Journal of Heat and Mass Transfer*. Vol. 84, 2015. pp. 592-599

[8] Lim. I. S., Tankin R. S., Yuen M. C. Condensation measurement of horizontal cocurrent steam/water flow. *Journal of heat transfer*.1984. Vol. 106/425

[9] Lee S., Euh D. J., Kim S. and H. S. Chul. Quantitative observation of co-current stratified two-phase flow in a horizontal rectangular channel. *Nuclear Engineering and Technology*, Vol. 47, Issue 3. 2015. pp. 267-283.

[10] Mandhane J. M., Gregory G. A. and Aziz K. A flow pattern map for gas-liquid flow in horizontal pipes. *International journal multiphase flow*. Vol. 1, 1974. pp. 537-553

[11] Khan A., Haq N. U., Chughtai I. R. Shah A. and K. Sanaullah. Experimental investigations of the interface between steam and water two phase flows. *International Journal of Heat and mass transfer*. Vol. 73, 2014. pp. 521-532