

AN EXPERIMENTAL STUDY ON STEAM CONDENSATION

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Introduction

The condensation mode of heat transfer plays an important role for the passive heat removal applications in the current nuclear power plants and advanced water-cooled reactor systems. It is well established that the presence of noncondensable gases can greatly inhibit the condensation process due to build-up of noncondensable gas concentration at the liquid/gas interface. The isolation condenser of passive containment cooling system of the simplified boiling water reactors is a typical application area of in-tube condensation in the presence of noncondensable gases. Air, as a noncondensable gas, existing in a heat exchanging system leads to a significant reduction in the heat transfer rate during condensation. This effect is due to the boundary layer of air and vapor forming next to the condensate layer and through which a gradient of the pressure of air and vapor develops. The build-up of noncondensable gas near the condensate film reduces the mass and energy transfer rate on the structure where condensation takes place.

An experimental study for the fundamental investigation of condensation in the presence of air was undertaken at the Middle East Technical University (METU), Ankara Turkey, in the frame of a project supported by the Turkish Atomic Energy Authority and the International Atomic Energy Agency. The test matrix consists of steady state and transient conditions with different operating parameters. In this paper, only the steady-state case is discussed.

Description of the Metu Condensation Test Facility

The test facility, named as METU Condensation Test Facility (METU-CTF), was installed at the Mechanical Engineering Department of METU [1]. The experimental apparatus consisting of an open steam or steam/gas system and an open cooling water system is depicted in the flow diagram of Fig. 1.

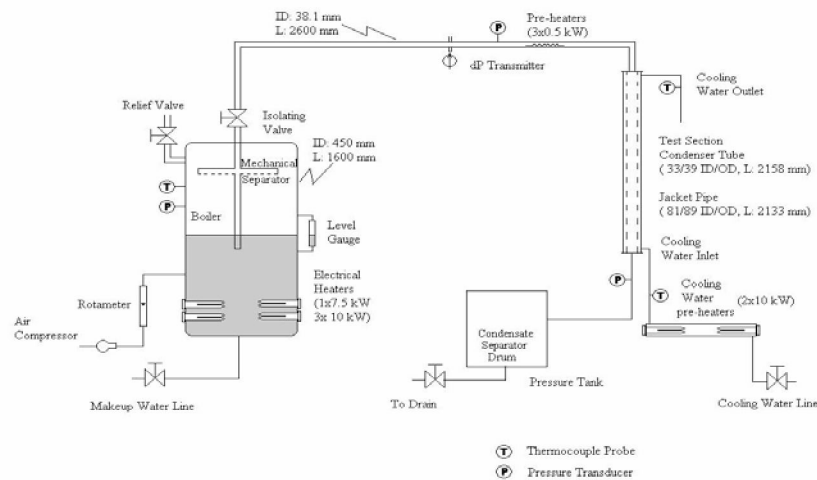


Figure 1. Flow Diagram of METU-CTF

Steam is generated in a boiler (1.6 m high, 0.45 m ID) by using four immersion type sheathed electrical heaters. Three of these heaters have a nominal power of 10 kW each and the fourth one has a power of 7.5 kW, at 380 V. All heaters can be individually controlled by switching on or off. To ensure dry steam at the exit of the boiler, a mechanical separator directly connected to the exit nozzle was installed. However, electrical pre-heating with three heaters is also available at the entrance of the test section to increase the temperature of steam, so that steam is guaranteed to be 100% dry. The boiler tank was thermally insulated to reduce environmental heat loss. Compressed air can be supplied either to the boiler tank (directly to the water) or to the steam line via a nozzle (after the orifice meter) on the horizontal part of the pipe, which connects the boiler and the test section. The pipe connecting the boiler tank and the test section has a length of approximately 2 m and an ID of 38.1 mm. The pipe was connected to the boiler tank via an isolation valve. This isolation valve is used to isolate the boiler until inside pressure of the tank is increased to a pre-determined level. The measurements performed on this part of the experimental facility are: mass flow rate via a differential pressure transmitter and temperature. The pipe connecting the boiler and the test section was thermally insulated.

The test section is a heat exchanger of countercurrent type that is steam or steam/gas mixture flows downward inside the condenser tube (inner tube) and cooling water flows upward inside the jacket pipe (outer pipe). The condenser tube consists of a 2.15 m long seamless stainless steel tube with 33/39 mm ID/OD and is flanged at both ends with sealing materials. A pressure measurement port was located at the vertical part of the inlet pipe flanged to the condenser tube. A total of 13 holes (1.5 mm diameter) were drilled with an angle of 30° at different elevations along the condenser tube length to fix the thermocouples for inner wall temperature measurements. The distance between the inner wall and the tips of the thermocouples is approximately 0.5 mm. The jacket pipe surrounding the condenser tube is made of sheet iron and has a length of 2.133 m and 81.2/89 mm ID/OD. The cooling water is supplied via a nozzle, which has been welded on the jacket pipe. A total of 15 holes (1.5 mm diameter) were drilled in radial direction at different elevations for installation of the thermocouples to be used for cooling water temperature measurements. Ten thermocouples were fixed to a 2 mm diameter Inconel guide wire and installed at the central position of the condenser tube for the central temperature measurements.

Experimental Results and Discussion

The heat flux distributions for experimental runs corresponding to the system pressures of 3 and 4 bars are presented in Figs. 2–3 [2]. These figures include the data of pure vapor and air/vapor mixture cases with different inlet air mass fractions. There are two major conclusions that can be drawn from these figures: First, the local heat flux drastically decreases as inlet air mass fraction increases for the same pressure setting and second, the axial local heat flux trend reveals the fact that the performance of the condenser is considerably decreased towards the exit of the condenser due to increase of local air mass fraction. The aforementioned first concluding remark is the evidence for how some amount of air, mixed with vapor, degrades the overall performance of the condenser while the second shows the axial dependency of the condenser performance on local air mass fraction. The suppression of local heat flux (at the axial location of 1 m from top) for 3 bars is calculated as; 23%, 24%, 30% and 48%, for inlet air mass fraction of 10%, 19%, 28% and 42%, respectively. The suppression of local heat flux (at the axial location of 1 m from top) for 4 bars is calculated as; 22%, 24%, 28%, 37% and 44% for inlet air mass fraction of 10%, 19%, 27%, 37% and 52%, respectively. The case with $X_a=52\%$ is a limiting case in the test matrix since vapor Reynolds number is lowest amongst other runs in the test matrix. The vapor Reynolds number for the case of $X_i=52\%$ is calculated to be 45,000 whereas it is between 76,000

and 86,000 for other runs, including the pure vapor case. Another point to be emphasized from the observation made for Figs. 2–3 is that the heat flux values corresponding to the air/vapor mixture get closer to those of pure vapor towards the bottom of the condenser tube due to diminishing condensation rate as the result of dominating film and diffusion resistance. In Fig. 4, the local heat flux for pure vapor case as the function of system pressure is presented. The local heat flux (at 1 m from top) increases from 100 kW/m² to 175 kW/m² as the system pressure increases from 1.8 bars to 5.5 bars. The case with P=1.9 bars and Re_v=77,000, in Fig. 4, was performed by taking the Re_v of the case with P=4 bars and Re_v=77,000 as reference to see the effect of pressure only. The net effect of pressure increase by 2.1 bars is an increase of the local heat flux of about 100% at the entrance region of the tube and 75% at mid elevation.

The local heat flux trends for air/vapor mixture case as the function of system pressure are shown in Fig. 5. This figure includes the runs for nominal air mass fraction of 20% with 2, 2.9 and 3.9 bars, respectively. As in the case of pure vapor, the vapor mass flow rate increases with system pressure so that Re_v is given in the figure for each run. The local heat flux (at 1 m from top) increases from 75 kW/m² to 105 kW/m² as the system pressure increase from 2 bars to 3.9 bars along with an increase of Re_v about 39%. Two results of other experiments performed at the MIT and UCB are also shown in Fig. 4.

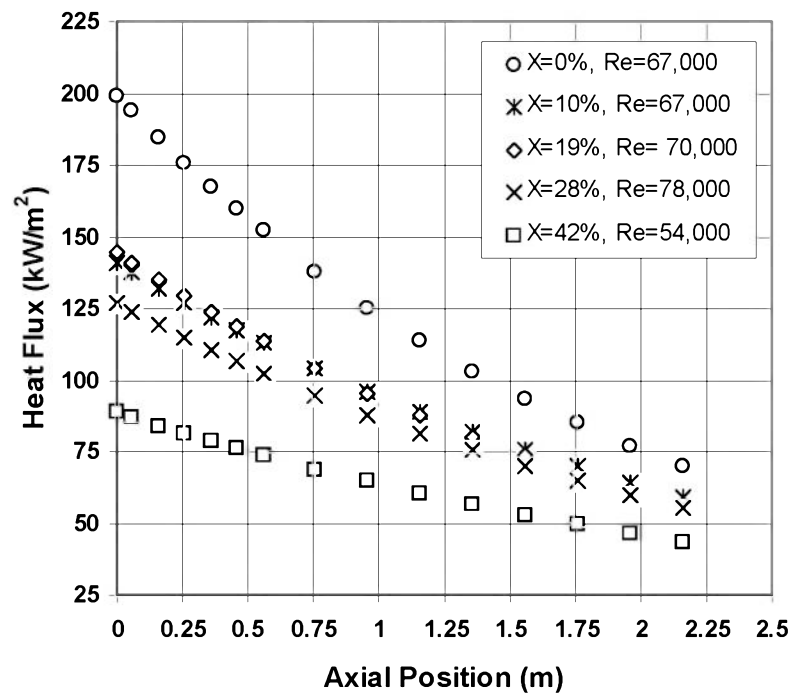


Figure 2. Heat Flux Distribution along the Condenser Tube (Pn= 3 bars)

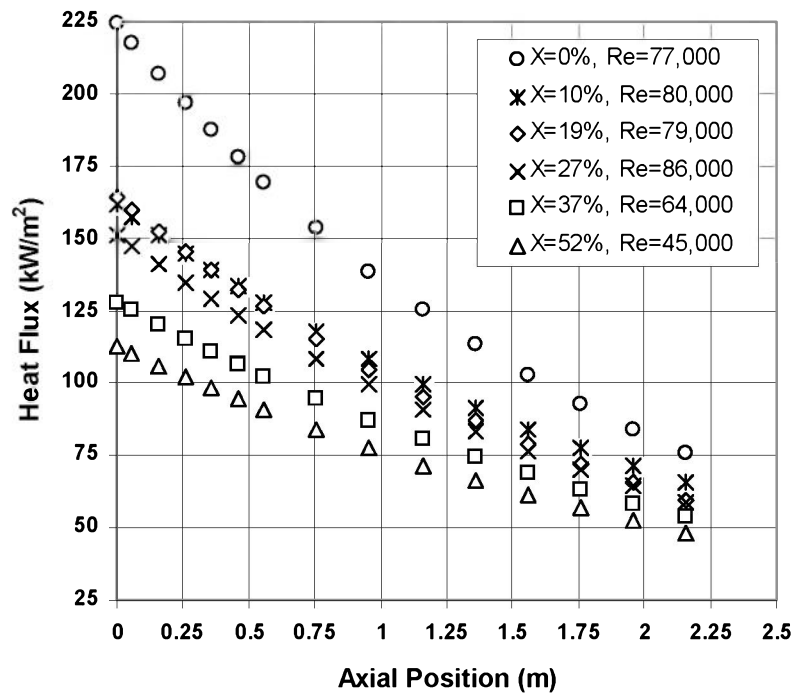


Figure 3. Heat Flux Distribution along the Condenser Tube ($P_n = 4$ bars)

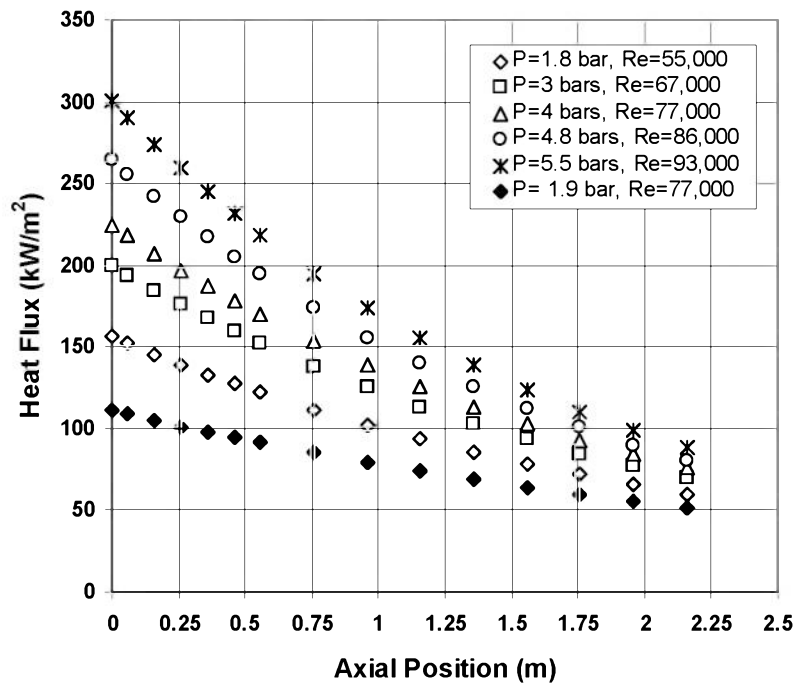


Figure 4. Effect of System Pressure and Vapor Mass Flow Rate for Pure Vapor Case

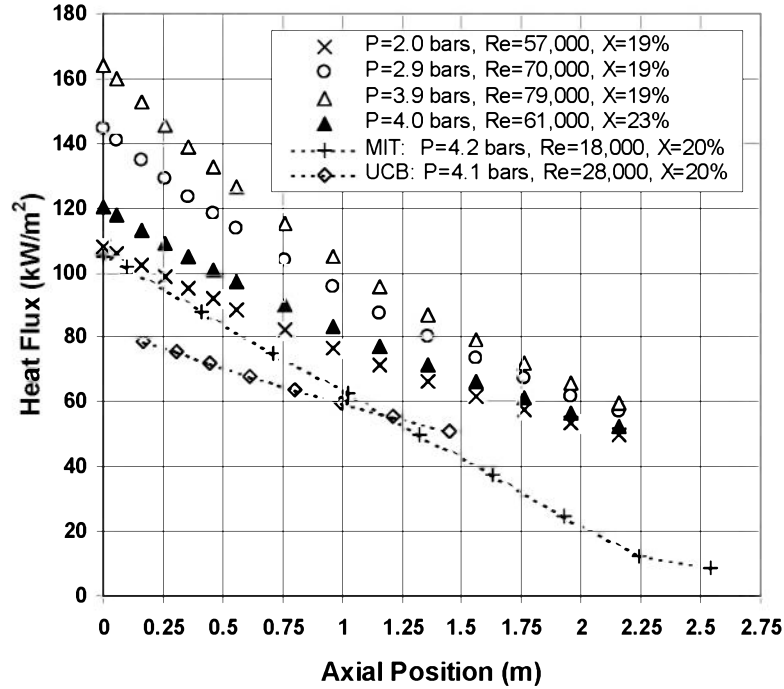


Figure 5. Effect of System Pressure and Vapor Mass Flow Rate for Air/Vapor Mixture Case

Conclusion

Even at the lowest inlet mass fraction, 10%, the decrease in heat flux is as high as ~20% while it reaches up to about 48% when the air mass fraction is increased to about 42%. The heat flux is strongly dependent on system pressure for both pure vapor and air/vapor mixture cases: The pressure increase of 2.1 bars and 0.8 bar yields an increase of local heat flux of about 43% (pure vapor case) and 14% (air/vapor mixture of X_i of 20%), respectively. Thus, the presence of noncondensable gases can greatly degrade the performance of heat exchangers when the main heat transfer mechanism is steam condensation while the system pressure has a reverse effect. Since the heat transfer performance of heat exchangers used for passive cooling applications in advanced nuclear reactors mainly relies on condensation, intrusion of any noncondensable gas should be well analyzed in design phase of such heat exchangers.

References

1. A.Tanrikut, In-tube Condensation in the Presence of Air, Ph.D. thesis, Dept. of Mechanical Eng., Middle East Technical University, Ankara, 1998.
2. A.Tanrikut and O. Yeşin, "In-Tube Steam Condensation in the Presence of Air," NUREG/IA-0184, US-NRC, 2000.