CHAPTER II

REVIEW OF LITERATURES

The wickless heat pipe or closed two-phase thermosyphon has the advantage of high heat transfer rate, simplicity in construction and low operating cost. So it attracts more attention than the conventional heat pipe when used in terrestrial applications.

In 1955 Cohen and Bayley constructed both rotating and static thermosyphons. Tests were conducted with various tube diameters, liquid fill quantities, and rotational speeds. They found that a maximum heat transfer rate depended on the quantity of the working fluid (3). Since then a large number of investigations on the thermosyphons have been carried out. Some of them studied the hydrodynamics of the inside phenomena. Numerous models were developed for predicting the limiting heat transfer and verified by experiments. By direct observation, the phenomena inside the thermosyphon is well-known now. The following literature survey concentrates on the heat transfer performance in relation to liquid fill quantity and inclination angle.

In 1976 Bezrodnyi, M.K. and Beloivan, A.I. (4) studied the dependence of maximum heat transfer capacity on the tube diameter, heated length, operating pressure, amount and physical properties of the heat carrier, and condenser size. Water, Ethanol and Freon-12 were used as the working fluids.

Bezrodnyi, M.K., Fainzil'berg, S.N. and Beloivan, A.I.,(5) also studied the maximum heat transfer of two-phase closed thermosyphons to explore their utilization in the evaporation-cooling systems of blast furnaces. The experimental apparatus had devices for applying heat and for cooling the condenser with flowing water. Chemically pure water was used in the tests. The relation of maximum heat flow density to the pressure for various lengths of pipe with 20 mm. inner diameter and to the level of it being filled with liquid, indicated that the maximum heat flow depended significantly on the pressure in the thermosyphon cavity and was independent of the heating zone length. According to the calculations, the average thickness of the liquid film washing the internal walls of the siphon was about 0.5 mm. With the level of filling over 25% in relation to the heating zone, the maximum heat flow decreased by 25%.

Bezrodnyi, M.K., and Alekseenko, D.V. (6) also carried out experiments to clarify the features of boiling heat transfer in thermosiphons having different geometrical, physical and regime parameters. The regime boundary was determined experimentally to correspond to the degree of filling at which the maximum heat flux no longer depended on the heat carrier amount. The siphons were built of stainless steel tubes and heat was supplied by passing electric current directly through the tube walls. The experiments with Freon-11 at 1 bar showed that the heat transfer coefficient did not depend on the thermosiphon geometry (diameter and length of the heating section) or the degree of filling. For the conditions of nondeveloped boiling, the data could be correlated using $Nu = 1.14\ Ra^{1/3}$, where Nu and Ra are the Nusselt and Rayleigh numbers, respectively. The data for the developed boiling region were given as a plot that indicated a certain intensification of the heat transfer compared to the corresponding value for boiling in a large volume.

In 1976 Gorbis, Z.R. and Savchenkore, G.A., (7) investigated the heat transfer performance of low temperature two-phase closed thermosyphon. Their thermosyphons were 5, 9, 10, 14 and 22 mm. in inside diameter, all 500 mm. long. The cooled lengths were 150, 200 and 250 mm. Water, Ethanol, Freon-113 and perfluorodibutyl ether were the working fluids. The filling fraction (2.9-60 %), inclination effect (4-90^{o}) and noncondensible gas fraction (0.6-100 %) were varied while the boundary condition temperatures and specific heat load were constant. The results were modelled by the thick-wall cylinder method. Their data and published results showed that boundary conditions affect boiling heat transfer in two phase closed thermosiphons. Condensation heat transfer at the cooling zone of the thermosiphon obeyed Nusselt's theory if the length of the noncondensable gas region was taken into account.

In 1977 Bezrodnyi, M.K., Alekseenko, D.V. (8) studied the boiling heat transfer of evaporative thermosiphons. The experiments with sufficient changes in heat flow showed two characteristics of heat transfer, that is underdeveloped and developed boiling. Visual observations of glass thermosiphons revealed that in the underdeveloped boiling region, individual vapor bubbles turbulized the region near to wall liquid layer. The heat transfer took place due to the evaporation of the superheated liquid from the near-to-wall layer (the vapor bubbles moving upward became large and became vapor missiles of considerable size). In the developed boiling region, vapor bubles were formed immediately in a superheated near-to-wall layer over the whole heating surface, the heat transfer intensity being affected by the turbulizing effects of the bubbles on the liquid.

In 1978 Semena, M.G. and Kiselev, Yu.F., (9) examined heat transfer of Cu-thermosiphons with inside diameters 6-24 mm., and filled with 2-50 % of Freon-11, Freon-113, Freon-142 and ethanol as the working fluid. The heat flux density was $(0.3\text{-}40)\times10^3$ W/m^2. Air and water cooling were used in the experiment. The inclination angle to the horizontal was 5-90°. The deviation of the calculated data from the experimental results were ±10 and 10-20 % for inclination angles 15-90° and 5-15° respectively. The heat transfer processes in the heat supply zone was also studied. The dependence of the heat transfer coefficient on the heat flux density at various cooling conditions and saturation pressure was given. The heat transfer coefficient is not affected by the length of heat conductor zone and the bed height, but by the inside diameter. The experiments of this studies were carried out with heat flux $(0.16\text{-}50) \times 10^3$ W/m^2, degree of filling of heat input zone 1-40%, the data are correlated by

$$Nu = 3050\ Re^{-0.33} Pr^{-0.9} (l/d)^{0.75}$$

in the evaporation regime and

$$Nu = 0.0096\ Pe^{0.7} Pr^{-1} Kp^{0.6}$$

in the boiling regime

where Nu = Nusselt number, Re = Reynolds number, Pr = Prandtl number, Pe = Peclet number, l = heat-input-zone length, d = diameter, Kp = the pressure number.

In 1978 Kamotani, Yasuhiro (10) studied theoretically the performance of a gravity assisted heat pipe operated at small tilt angles, including the vapor-shear effect. The vapor shear created a backflow region at the surface of the puddle and, thereby, degraded the pipe performance. The performance of the pipe was strongly affected when the fluid charge exceeded a certain amount; the pipe operation became unstable at a relatively small heat transport.

In 1979 Feldman, K., Thomas Jr. and Munje, S. (11) carried out experiments on gravity assisted heat pipes with and without circumferential grooves. Their heat pipes were 91.4 cm. long with 1.27 cm. outside diameter. For a heat pipe with 40 grooves/cm., the maximum thermal conductance measured was 22.7 W/degree, and the heat transfer coefficients in the evaporator and condenser were 4,700 and 11,992 W/m^2-degree K respectively. For a heat pipe without grooves, the thermal conductance was 4.2 W/degree and the respective evaporator and condenser heat-transfer coefficients were 1,000 and 1,500 W/m^2 K.

In 1979 Imura, H. and et.al., (12) studied the heat transfer in a closed thermosiphon using ethanol and water as the working fluid. The heat-transfer coefficient in the heating zone was independent of the volume ratio (V^+) of liquid/tube and of the length ratio of heating zone/cooling zone. At $V^+ < 0.10$, the heat-transfer coefficient decreased with relatively low heat flux; at $V^+ > 0.25$, the liquid was subcooled in the cooling zone and vibration occured.

In 1980 Bezrodnyi, M.K., and et.al. (13) investigated the maximum heat transfer of a thermosyphon using various heat carriers. The maximum heat flux at a given pressure depended on the vapor velocity in the adiabatic zone and the type of heat carrier, but was independent of the thermosiphon dimensions. The dependence of the critical heat flux on pressure was analogous to that of the critical heat flux during boiling in large liquid volumes.

In 1980 Andros, F.E. (3) studied the heat transfer characteristics of thermosyphon using Freon-113, ethanol and water as the working fluid. Visual observations were used to interprete the heat transfer characteristics as determined from temperature and pressure measurements for various liquid fill quantities, heat rates, condenser temperatures and heated to cooled length ratios. The flow of condensate and dry-out phenomena were observed. He found that condenser heat transfer was independent of liquid fill charge. The accepted condensation correlations agree with Nusselt's theory. Evaporator results were indicative of nucleate pool boiling for various quantity and geometric parameters.

In 1980 Harada, Kazuo and et.al., (14) studied the heat transfer of large thermosyphons for use as a heat exchanger for a blast furnace. Their thermosyphons had outside diameter of 50.8 mm. and length of 6 m. The experimental results showed that a slight inclination from the vertical did not affect the heat-transfer characteristics. The optimum liquid (water) content was 25-30%. The evaporative heat transfer coefficient was not much affected by the liquid content of the working fluid. The condensation heat transfer coefficient was slightly lower than that predicted from the Nusselt theory. The threshold heat-transfer rates agreed fairly with Diehl's eqution.

In 1981 Prenger, F.C., Jr., and Kemme, J.E. (15) studied the performance limits of gravity assisted heat pipes using volume of the working fluid and tilt angle as the independent variables. Two limits were identified : an entrainment limit which results in overheating of the evaporator wall near its midpoint and a pressure drop limit resulting in overheating at the base of the evaporator. The former is a function of the vapor flow characteristics and was observed to be independent of the liquid volume and tilt angle, whereas the latter is related to both the tilt angle and the liquid inventory in the heat pipe.

In 1982 Bilegan, I.C. and Fetcu, D. (16) investigated the heat transfer characteristics of gravity-assisted aluminum extruded heat pipes with Freon-12 (R-12) as working fluid. The experiment was carried out on two sets of heat pipes : 1 m. and 1.5 m. long (16 mm. outer diameter). The inclination angle, operating temperature and the length of heat pipe were used as parameters. The results of the experiments indicated that the transferred axial heat flow rate increased rapidly with the tilt angle up to values of about 30° and then it remained constant. The 1.5 m. long heat pipes can transfer 900 W, in the range of operating temperatures between $20^{\circ}C$ and $60^{\circ}C$. Finally they concluded that axial grooved aluminum heat pipes can be used to achieve some inexpensive and compact heat exchangers in waste heat recovery systems at low temperature.

In 1982 Larkin, B.S. (17) studied the heat transfer performance of a heat pipe to be used as component in an air-to-air heat recovery. His heat pipe was made from a thick-walled copper tube; both smooth tubes and circumferential groove tubes were used. The internal dimensions were: length 1.15 m., diameter 24.2 mm. The two fluid used were water and refrigerant 22 , over a temperature range from $20^{\circ}C$ to $100^{\circ}C$. The tests were conducted by varying the fill quantity of the working fluid and heat flux. The heat pipe was inclined at 5° to the horizontal. Temperature profiles and heat transfer coefficients were given for heat fluxes from 500 watts to 1,100 watts. The results showed that the condensing coefficient decreased as the fluid temperature and heat flux increased, while the evaporating coefficient increased as heat flux and fluid temperature increased. The fill quantity did not have a significant effect on the heat transfer performance. The grooved surface improve the overall heat transfer. In the case of refrigerant 22, grooved surface gave better overall heat transfer below 40^{o}C while water was superior at temperatures above 50^{o}C.

Negishi, K. and Sawada, T. (18) studied the interactive influence of the inclination angle and the amount of working fluid on the heat transfer performance of an inclined two-phase closed thermosyphon. The thermosyphon was made of 15 mm. standard copper tube. The tube was 300 mm. long and had a 13 mm. I.D., with a wall thickness of 1 mm. In order to observe the detailed phenomena concerned with the heat transfer mechanism of this device, two glass windows were fitted at both ends of the thermosyphon. In addition, a glass thermosyphon made to the same dimensions was prepared. Two jackets were set on the tube, one being used as a heating jacket for the evaporator and the other as cooling jacket for the condenser. Distilled water and ethanol were used as the working fluid. The mean temperature of cooling water was kept at 25^{o}C. The heating water temperature was varied. The fill ratio (based on inner volume of the evaporator) of the working fluid was varied from 5-100%. The angle of the thermosyphon was in the range between 90^{o} and -10^{o} from the horizontal. The overall heat transfer coefficient was found to vary from 2.4 x 10^{3} W/m^{2}.K to 3.0 x 10^{3} W/m^{2}.K in the case of water and from 0.9 x 10^{3} W/m^{2}.K and 1.1 x 10^{3} W/m^{2}.K for ethanol. To obtain a high heat transfer rate, the fill ratio should be between 25 and 60 % for water and between 40 and 75 % for ethanol. Also the inclination angle should be between 20^{o} and 40^{o} for water, and more than 5^{o} for ethanol.

In 1983 Gross U.(19) studied the heat transfer in the heating zone and cooling zone of a closed thermosiphon tube. Refrigerant R115 was used as working fluid. Fluid pressure, heat flow rate and inclination angle of the tube were varied. The fill quantity of fluid was varied so that the critical specific volume was obtained. In subcritical states there were boiling in the heating zone and condensation in the cooling zone. Depending on the pressure, there were 3 regimes of boiling, pure nucleate boiling, nucleate boiling in a transition region, and film boiling. Small tilt, large heat flow rates, and high pressure were best conditions for a large heat transfer coefficient; these, however, also promote transition to film boiling. In the thermosiphon there existed laminar and turbulent film condensation, respectively. The largest heat transfer coefficients were obtained in the laminar film conditions at small heat flow rates, with inclination angles of 20 and 40^{o} and high pressures, caused by a growing disturbance of smooth film surface with increasing pressure. In the supercritical state heat was transferred by single-phase natural convection. The heat transfer coefficients were largest near critical point, especially for low heat flow rates and large inclination angles though smaller than subcritical ones.

Fetcu, D. (20) performed experimental investigations to find the optimum heat pipe orientation and fluid inventory in the case of using methanol and acetone as working fluid. He found that the maximum heat transfer rate was obtained at the inclination angle (with respect to horizontal) between 40 - 60^{o} for methanol and about 40^{o} for acetone. The optimum fill ratio (compared with total volume) for the methanol heat pipe was about 0.03 for temperature differences less than 50°C and 0.09 for greater temperature differences. In the case of acetone heat pipe the optimum fill ratio was 0.12 for low temperature difference and 0.09 for high temperature difference. From these experimental results, some fairly large heat pipes were constructed and tested for use in the manufacture of heat exchangers. The heat pipes were tested in liquid-liquid, gas-liquid or gas-gas systems. The results show that these type of heat pipes can successfully be employed in the manufacture of heat exchangers.

Singh, G. and Tathgir, R.G. (21) experimentally studied the heat transport capability of a stainless steel gravity-assisted axially grooved heat pipe with water as working fluid. The experiments were performed on the 425 mm. long, 25.5 mm. in diameter heat pipe using the heat input to the evaporator, the inclination angle and the mass of the working fluid (m) as parameters. The heat pipe offered maximum heat transfer rate between the input values of 60 W while operating at 18° from the horizontal with m = 4 mg. and at 12° with m = 5 mg.

Feldman, K.T. and Srinivasan, Jr. R. (22) investigated the influence of tilt angle, liquid fill charge and operating temperature on the maximum heat transfer. The experiments were carried out on a 21 ft long, 3/4 inch in diameter thermosyphon.

They concluded that the optimum liquid fill was in the range of 18 to 22 percent of the thermosyphon volume. The heat transfer performance improved with the increase in operating temperature and tilt angle. The optimum tilt was not obtained because the tilt angle was varied only up to 20° from the horizontal. A performance correlation for the heat transfer limit of the thermosyphon was obtained from the experimental results. The experimental values deviated less than ± 20% from the correlation.

Yaopu, Wen and Shun, Guo (23) studied the heat transfer performance with emphasis on the effects of heat input and saturated pressure. The tests were conducted on Cu thermosyphon of three inner diameters (10, 13, 20 mm.) using water, acetone and Freon-12 as working fluid. The thermosyphons were all 1 m. long. Heat input supplied by an electrical heater was varied from 0.5-16 W/m^2, saturated pressure from 0.1-16 bar and the tilt angle from 1^{o} to 90^{o} from the horizontal. In the tests, heat input increased gradually while the coolant was maintained at a constant temperature, and to observe the effect of saturated pressure on the heat transfer coefficients another coolant temperatures were examined at the same conditions. The working fluid in each thermosyphon was about 30% of the evaporator volume.

From the experiments, they concluded that the heat flux had a more significant effect on the evaporation coefficient than the saturated pressure and temperature, the evaporation coefficient being proportional to about 0.7 power of the heat flux, but the heat flux was not so important in the condensation zone. The condensation coefficient depended largely on the tilt angle. The coefficient rose quickly as the tilt angle varied from 0^{o} to 30^{o} and reached the maximum in the range of 15^{o}-30^{o} and had the lowest values at 90^{o}. Empirical correlations for determining the evaporation and condensation coefficient were presented. The two correlations were in good agreement with the experimental data for acetone and Freon-12, but had a significant deviation in the case of water.

Groll, M. and Spendel, Th. (24) studied the effect of inclination angle on the maximum performance and effective thermal conductance of four different types of thermosyphons, i.e., smooth surface, grooved inner surface, smooth tube with concentric perforated tube as flow separator, grooved surface with flow separator. The tubes were made of copper and had 34 mm. inner diameter, wall thickness of 3 mm., and evaporator and condenser lengths of 2.1 m. each.

They found that the influence of the roughened surface was not so significant when compared to the flow separator effect. For thermosyphons without flow separator, the maximum heat flow reached maximum values at the angle of 40^{o} from the horizontal but for thermosyphons with flow separator the maximum heat flow increased as the tilt angle increase and had the shape like a sine function. Finally, they concluded that the closed two-phase thermosyphons operated in near horizontal orientation should have an adequate capillary structure, and that the thermosyphons operated at very high heat flows should be supplied with a flow separator.

In 1983 Tokuma, Masao and et.al.(25) studied the effects of shear stress acting on the liquid-vapor interface at the condenser of the two-phase closed thermosiphon. An analysis based on the Nusselt theory of flimwise condensation as well as heat transfer experiments were carried out. The analysis showed that the interfacial shear stress caused by rising vapor made the condensate film thicker and that the condensation heat transfer coefficient and heat transfer rate decreased, though they were larger than analytical results.

In 1983 Suriyashay, C. (26) investigated the fabrication method of heat pipes in a laboratory. The successfully developed method was one in which heat pipes were heated by immersion in an oil bath kept at 110^{o}. The heat pipes were made of pyrex glass, with inner diameter of 8 and 10 mm., respectively and approximately 1 ft long. The wicks were made of 3-layer copper screen (80 mesh) and the working fluid was distilled water.

To test the performance of these heat pipes, the experiments were carried out in the anti-gravity direction at various angles: 14, 16, 18 and 20 degrees with respect to the horizontal. From the test results, it was found that the limiting heat transfer was of the capillary action type, and the measured limits agreed with their theoretical values to within $\pm$ 10% the effective thermal conductivity of the heat pipes was highest at the angle of 14 degrees and at the axial heat rate of 2.93 watts. The highest thermal conductivity was about 10 times higher than that of copper rod.

In 1983 Tanthapanichakoon, W and Pichiansophon, S (27) investigated the heat transfer performance and temperature distribution on the heat pipe. Their heat pipes were made of copper tube with water as the working fluid. The wicks were made of 3 layers of brass screen (150 mesh). The experiments were carried out in the anti-gravity direction at the tilt angle of 0-10 degree from the horizontal.

The experimental temperature distribution on the heat pipe surface was agree very well with the calaulated values, but the heat rate is significantly deviated from the experiments. Because the equation for calculated the wick resistance is suitable for only the liquid metal working fluid. To corrected the wick resistance, the n factor were presented. The new calculated values agreed very well with the experiments within $\pm$ 16% error. In addition, they also investigated the limiting heat transfer on the same type of heat pipe, and found that the controlling heat transfer limit was the limit of the capillary action.

In 1984 Dikii, N.A. and et.al. (28) studied the limiting heat flux as a function of the size, degree of filling (ϵ), pressure, and the type of intermediate heat-transfer agent. The dependence of the heat flux density on ϵ, the pressure, and the ratio of the internal conderser area to that of the heating zone is presented graphically. The optimum ϵ and ratio of condenser to the heating zone area are 65.75 % and 1.3-1.6, respectively.

In 1985 Gross, U. and Hahne, E. (29) studied the effects of the working fluid (R13B1) pressure (p = 16-39.85 bar; p/p_c = 0.4-1.0) and the pipe inclination angle (α = 0-80°) on the heat transfer capacity of a wickless heat pipe. The average heat-transfer coefficient in the evaporator depended strongly on p and not on α, but it depended on p and α in the condenser. Favorable operating conditions are $p/p_c \approx 0.9$ with avoidance of film boiling, small inclination angle when a laminar condensate film is to be expected and large inclination angle with a turbulent condensate film.

In 1985 Kobayashi, K. and et.al.(30) studied the flow pattern of the working fluid in a glass wickless heat pipe via direct observation while changing the inclination angle. The heat transfer rate was estimated through classification of the heat-transfer region on the basis of heat transfer mechanism. The calculated heat-transfer rates agreed with the experimental data. The heat-transfer resistance of the wickless heat pipe decreased with increasing inclination angle and the heat transfer rate had a maximum at inclination angle 30-60°.

In 1986 Hlavacka, V. and et.al.(31) presented the heat transfer characteristics of Freons, ammonia, methanol , ethanol, dimethylketone, water, ethylbenzene, phynylbenzene, and a mixture of 26.5:73.5 diphynyl-diphynyloxide. The heat pipes were positioned at 75-90° to the horizontal plane. Grooves improved the flow distribution and increased the heat transfer by $\leq$ 30 %

In 1986 Takuma, Masao and et.al.(32) conducted visual observation of flow patterns at the condenser and made heat transfer measurement for heat input rate of 18-800 W using a vertical annular device with R113 as working fluid. Ripples were generated on the condensate film surface when the condensate film Reynolds number was over 20 and condensation heat transfer was promoted. A theoretical analysis was presented, in which the effects of interfacial waves and vapor drag were considered. The analysis agreed very well with experimental results. When the working fluid quantity was not large enough, the two phase mixture generated by boiling did not reach the condenser.

In 1986 Tangsathapornphanich, P. (33) investigated the heat transfer performance of a wickless heat pipe. The heat pipe was made of pyrex glass, 10 mm. in diameter, and 370 mm. long, using distilled water as working fluid. The data obtained was used to design the heat pipe heat exchanger. A prototype liquid-liquid heat exchanger was also constructed and tested its performance . The average values of UA obtained from the tests were 0.12-0.43 W/$^{\circ}$C per heat pipe and the average heat transfer rates were 3-16 W per heat pipe.

In 1986 Tanthapanichakoon, W. and Komolpamorn, W. (34, 35) studied the thermal performance of closed-loop type thermosyphons. Two loop type thermosyphons, one without and one with special packings in the evaporator section were designed and constructed. Both thermosyphons were made of pyrex glass. In order to investigated these thermosyphons, both transient response test and steady state heat flow test were carried out. In the transient response test, both thermosyphons had the fastest response time when their evaporator sections were placed horizontally, and the conventional thermosyphon without packings had quicker times because its total mass was less, and the optimal filled volume was approximately 30% of the evaporator volume. In steady state test, the result showed that the heat flow rate and the proportion of heat loss to the environment was increased as the heat input increased. The special packings were also found effective in enhancing the heat flow rate by about 22% when the two thermosyphons were compared under the same conditions. The closed-loop thermosyphons can transfer heat approximately 100% better than the conventional thermosyphons with the same material. In particular, the copper closed-loop thermosyphon can transport axial heat at a rate above 464 watts, and had effective thermal conductivity about 157 times of copper rod.

In 1987 Fukano, Tohru and et. al., (36) studied the relation between the flow patterns inside a vertical thermosyphon and the operating limit by visual observation while varying the diameter of the thermosyphon, the kind and amount of working fluid and the system pressure. A correlation was obtained based on a large amount of critical heat flux data obtained for many kinds of thermosyphons with the height of liquid pool and the bubble rising velocity as variables. The agreement of the correlation with experimental data was better than that of R.K. Sakhuja for flooding correlation and that of H. Imura.

In 1987 Ogushi, T. and Yamanaka, G. (37) studied the effects of fluid inventory and tilt angle on the capillary pumping limit of axial grooved aluminium heat pipe using R-11 as working fluid. A model predicting the temperature rise in the evaporator and the heat transfer capacity was developed. The flooding limit and the evaporator and condenser film coefficients in case of gravity-supported operation were also investigated and compared with the performance of the closed two-phase thermosyphons.

In 1987 Gross, U. and Hahne, E. (38) studied the condensation heat transfer of a thermosyphon by varying the heat flux (200-3500 W.), inclination angle (0-80^{o} to the vertical) and reduced pressure (0.4-0.98). The thermosyphon was made of steel tube with 40 mm. diameter and 765 mm. cooling length. The heat transfer was favorable when the pressure was low, the heat flux small, the inclination large and the condensate film in laminar flow, and when the pressure is high, the heat flux large, the inclination angle small, and the condensate film wavy or turbulent.

In 1987 Roesler, S. and et.al. (39) experimentally and theoretically examined a thermosyphon with R-113 as working fluid and fill ratio of 2-40%. The critical heat flux at which dry-out occurred increased from about 10^4 at 5% fill ratio to about 5×10^4 W/m^2 at 40% fill ratio.