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Optimal nozzle spray cone angle for triangular-pitch shell-and-tube interior spray evaporator

Tong-Bou Chang *, Li-Yang Yu

Department of Mechanical and Energy Engineering, National Chiayi University, Chiayi City, Taiwan

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ABSTRACT

The present study proposed a triangular-pitch shell-and-tube spray evaporator featuring an interior spray technique. In the proposed approach, the nozzle tubes are positioned within the tube bundle in such a way that the surface of each heater tube is sprayed simultaneously by two cooling sprays. As a result, the dry-out phenomenon is prevented, and thus the heat transfer performance is improved. An analytical expression is derived for the optimal spray cone angle for the proposed evaporator. It is shown that the optimal spray cone angle increases with an increasing diameter of the heater tubes, but decreases with an increasing distance between the heater tubes and nozzle tubes. The analytical expression for the optimal spray cone angle was validated by experimental spray tests using spray cone angles of 0°, 30°, 40°, 45° and 60°, respectively. It is shown that a good agreement exists between the experimental results for the optimal spray cone angle and the theoretical results. Furthermore, the experimental results confirm that the shell-side heat transfer coefficient obtained using the interior spray technique is significantly higher than that achieved in a conventional flooded-type evaporator over a wide range of surface heat fluxes.

1. Introduction

The performance of many engineering systems is reliant on an efficient removal of the generated heat. In general, the heat transfer performance of a system can be improved by either increasing the heat transfer area or improving the efficiency of the heat transfer process itself. However, in many cases, increasing the heat transfer area is impractical for either cost or space reasons. As a result, the problem of developing more efficient heat transfer systems has attracted significant interest in recent decades. Spray evaporation heat exchangers, first proposed by Nakayama et al. [1] in 1982, have two major advantages over conventional flooded evaporators, namely an improved heat transfer performance and a 20–90% reduction in the chiller refrigerant inventory. Consequently, spray evaporation systems are now widely used throughout industry and agriculture.

Hodgson and Sutherland [2] showed that in spray cooling systems in which the surface temperature is higher than the Leidenfrost temperature, a thick vapor film is formed between the droplets and the wall, and thus film boiling conditions generally occur. Pais et al. [3] showed that for very smooth surfaces and a water coolant, the maximum spray cooling heat flux is around 1000 W/cm^2 at surface temperatures less than 105 °C. Several studies have shown that the critical heat flux (CHF) for spray cooling is much higher than that of pool boiling for the same working fluid. For example, Chow et al. [4] showed that for water coolant, the CHF values for pool boiling and spray cooling are around 120 W/cm^2 and 1000 W/cm^2 , respectively. Similarly, for liquid nitrogen, the CHF of spray cooling is around 160 W/cm^2 , while that of pool boiling is 16 W/cm^2 [5,6]. Finally, for FC-72 refrigerant, the CHF values of pool boiling and spray cooling are 20–30 W/cm^2 [7,8] and 80 W/cm^2 [9], respectively. Deb and Yao [10] found that the droplet size, velocity and spray pattern (conical or square) play a major role in determining the heat transfer performance of dilute spray heat transfer systems, but play only a minor role in dense spray systems.

Hsieh et al. [11] performed spray cooling experiments using R-134a and water as working fluids, and found that the Weber number and the degree of liquid subcooling both have a significant effect on the heat transfer performance. Tartarini et al. [12] showed that the ability to predict the onset of boiling depends strongly on an accurate measurement of the solid–liquid interfacial temperature, and is more easily achieved for solids with a higher conductivity. Sehmbey et al. [13] conducted spray cooling tests

* Corresponding author at: No. 300, Syuefu Rd., Chiayi 600, Taiwan. Tel.: +886 5 2717565; fax: +886 5 2717561.

E-mail address: tbchang@mail.ncyu.edu.tw (T.-B. Chang).

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Nomenclature

A	area, m^2
d	distance between heater tube surface and nozzle tube surface, mm
D	diameter of heater tube, mm
h	heat transfer coefficient, W/m^2 °C
L	nozzle pitch in axial direction of spray tube, mm
M	molecular weight
P	pressure, MPa
P_c	critical pressure, MPa
P_{cr}	P/P_c
q''	wall heat flux, W/m^2
Q_t	total power of heater, W
r	radius of spray coverage area, m
R_p	surface roughness, μm
T	temperature, °C
Greek symbols	
α	segment angle of A_1, degrees
β	spray cone angle, degrees
Superscript	
$\bar{}$	average quantity
Subscripts	
c	critical point value
g	vapor phase
h	heated surface
l	liquid phase
nb	nuclear boiling
opt	optimal value
sat	saturation state value
w	wall

using plates with different materials and temperatures ranging from 60–220 °C. It was shown that for surface temperatures greater than 100 °C, the droplet contact angle increased with an increasing surface heat flux. Grissom and Wierum [14] presented a conduction-controlled analytical model of droplet evaporation based on experimental measurements obtained at atmospheric pressure. The analytical results showed that the initiation of the Leidenfrost state provides the upper bound on the surface temperature required for spray evaporative cooling. Choi and Yao [15] showed that when liquid coolant is sprayed from a high-pressure nozzle, it atomizes due to an instability between the liquid droplets and the surrounding air. In addition, it was shown that the high-speed droplets penetrate the liquid film on the heated surface, and therefore improve the heat transfer performance by facilitating the release of the evaporated vapor bubbles. Ayub et al. [16] designed a low charge ammonia spray-type shell-and-tube evaporator for process cooling applications in chemical plants. The evaporator was shown to have a water outlet temperature of 0.6 °C and a refrigerant charge of just 0.06 kg/kW.

In a refrigeration cycle, an expansion process is required to reduce the refrigerant pressure to a level at which evaporation can take place. In practice, this pressure drop can be used to drive a liquid spray. However, even though spray evaporation has a high heat transfer performance, it is seldom used in compact heat exchangers with tube bundles since the overhead sprays used in such systems cannot impinge upon the lower tubes directly. In other words, the surfaces of the lower tubes are simply covered with a liquid film falling from the tubes above them (see Fig. 1). Moeykens and Pate [17] reported that the heat transfer performance of a spray evaporator system with horizontal plain tubes is lower than that of a pool boiling system under high heat flux conditions due to the occurrence of a dry-out phenomenon on the lower surface of the tubes. However, the authors showed that the occurrence of this dry-out phenomenon could be suppressed by increasing the supply rate of the refrigerant. The same authors [18] used four overhead nozzles to spray coolant on a triangular-pitch tube bundle comprising four rows of tubes. It was found that the heat transferred from the second and fourth rows was significantly lower than that from the first (i.e., top) and third rows. This finding was attributed to the fact that a greater volume of liquid dripped from the tubes in the top row onto the tubes in the third row, positioned directly below them. By contrast, relatively less liquid dripped on the tubes in the second and fourth rows (see Fig. 1), and hence the cooling effect was reduced. Chang and Chiou [19] devised a technique for enhancing the heat transfer performance of spray evaporators by fitting a liquid collector to the underside of each tube in the bundle. The experimental results showed that the liquid collectors prevented the onset of the dry-out phenomenon and therefore resulted in a significant improvement in the heat transfer coefficients of the lower tubes. Chang and Chiou [20] investigated the heat transfer characteristics within the gap between the heater surface and the liquid catchers, and found that a smaller gap size increased the heat transfer coefficient under low surface heat flux conditions. Chang et al. [21] showed that the dry-out problem in compact shell-and-tube evaporators can also be prevented by placing the spray nozzles within the bundle interior rather than above the upper row of tubes.

Many previous studies have investigated the effects of the nozzle configuration on the heat transfer performance of spray cooling systems. Shedd and Pautsch [22] proposed a full-coverage spray drainage system in which the sprays were inclined in such a way that the coolant flowed across the heated surface and then exited

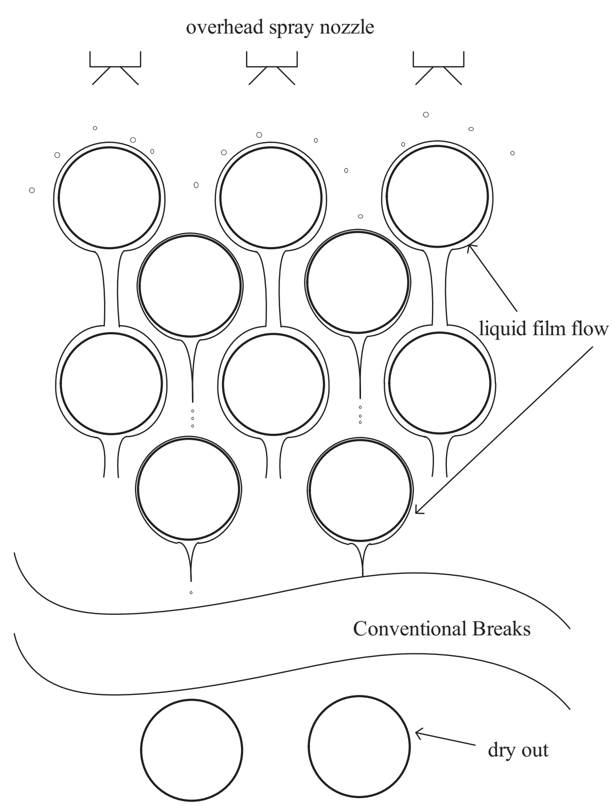

Fig. 1. Liquid film distribution in triangular-pitch tube bundle with overhead spray.