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Experimental investigation on ammonia spray evaporator with triangular-pitch plain-tube bundle, Part II: evaporator performance

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Abstract

Spray evaporation of ammonia was tested using spray nozzles distributing liquid ammonia downward onto a horizontal, 3–2–3 triangular-pitch, 1.25 pitch ratio, plain-tube bundle composed of stainless-steel tubes of 19 mm (0.75 in.) diameter. The saturation temperature test range was from −23°C to 10°C, and the heat flux range from 3.2 to 35 kW/m^2. The effects of heat flux, saturation temperature, spray flow rate, nozzle height, and nozzle type (standard-angle or wide-angle) were investigated. The spray evaporation heat transfer performance was compared with that of pool boiling, as well as spray evaporation on square-pitch tube bundle and single tube.

1. Introduction

Very limited data of spray evaporation of ammonia on a tube bundle are available. Zeng et al. [1] investigated spray evaporation of ammonia on a 3×3 square-pitch plain-tube bundle using commercial spray nozzles. The heat transfer performance was found to be up to 48% higher than a flooded boiling bundle. The tube bundle effect and effects of heat flux, saturation temperature, spray flow rate, nozzle height, and nozzle type (standard- or wide-angle) were studied. Their results revealed that spray evaporation heat transfer coefficient primarily increases with heat flux and saturation temperature, and the effects of spray flow rate, nozzle height and nozzle type are insignificant in most cases.

An issue of interest is a comparison of spray evaporation performance between square-pitch and triangular-pitch tube bundles. Compared with square-pitch tube bundle, the triangular-pitch tube bundle provides the advantage of size reduction. In spray evaporation of saline water, triangular-pitch tube bundle was reported to demonstrate performance superior to square-pitch and rectangular-pitch bundles [2]. More data are needed before conclusion can be reached as to which tube pattern provides a better performance.

The present work reports the results of a nozzle-sprayed ammonia evaporation on a horizontal triangular-pitch plain-tube bundle. Tests were conducted under conditions similar to a typical ammonia refrigeration system using a small spray evaporator where liquid ammonia was sprayed by commercial nozzles downward onto a 3–2–3 triangular-pitch stainless-steel tube bundle consisting of tubes of 19.1 mm (0.75 in.) diameter heated by water/glycol solution. The saturation temperature test range was from −23°C to 10°C (−10–50°F), with the corresponding pressures from 164 to 615 kPa (23.7–89.2 psi). Heat transfer coefficient data were collected within the heat flux range from 3.2 to 35 kW/m^2 (1000–11,000 Btu/h ft^2). A systematic experimental investigation was undertaken to study the effects of heat flux, saturation temperature, spray flow rate, nozzle height, and nozzle type (standard-angle or wide-angle) on the tube bundle

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Nomenclature

D	tube diameter
d	nozzle height, viz., distance between the nozzle outlet and the top of the horizontal tube bundle
g	gravitational acceleration
h	heat transfer coefficient
$\bar{h}$	averaged heat transfer coefficient of tube bundle
k	thermal conductivity
$\dot{m}$	spray mass flow rate
Nu	Nusselt number
P_{cri}	critical pressure
P_{red}	reduced pressure
P	pressure
Pr	Prandtl number
q''	heat flux
$\bar{q}''$	average heat flux
Re	Reynolds number
T	temperature

Greek symbols

β	spray angle of nozzle
Γ_1	average mass flow rate of sprayed liquid reaching a tube bundle per unit tube length
μ	dynamic viscosity
ν	kinematic viscosity
Φ	non-dimensional heat flux

Subscripts

cri	critical
l	liquid
n	per nozzle
s	saturation

performance. The spray evaporation heat transfer performance was compared with pool boiling heat transfer performance of the same tube bundle, as well as the spray evaporation performance of square-pitch tube bundle and single tube reported by Zeng et al. [1,3]. A correlation was developed based on the present triangular-pitch bundle spray evaporation data in a form similar to the correlations for square-pitch tube bundle and single tube [1,3]. The experimental facility and procedure were the same as that described in Part I of the present two-paper set and will not be repeated here.

2. Results and discussion

As discussed in Part I, it was observed during the experiment that most of the interstices between tubes were filled with liquid and vapor mixture moving downward, and there were no clearly defined liquid films flowing on individual tube walls. During pool boiling experiment of the same tube bundle, the interstices between tubes were likewise filled with two-phase mixture. The fact that two-phase mixture fills the interstices between tubes in both spray evaporation and pool boiling suggests that the amount of refrigerant retained in the tube bundle under spray evaporation is comparable to that under pool boiling, and the anticipated reduction in the refrigerant inventory associated with a spray evaporator may be mainly in the volume between the bundle and the vessel wall. Similar result was reported concerning spray evaporation of ammonia in a square-pitch bundle [1].

In the present work, 18 ammonia spray evaporation tests ranging from −23°C to 10°C were conducted. In each test, the spray evaporation heat transfer coefficient data for individual tubes in the 3–2–3 triangular-pitch plain-tube bundles were collected. The average tube bundle coefficients based on data of all eight tubes, $\bar{h}$, were calculated to characterize the overall performance of the bundle. Compared with the average bundle coefficient of the same tube bundle under pool boiling at the same saturation temperature, the average spray evaporation coefficient can be up to 65% higher. The enhancement depends on the spray evaporation test condition. It generally increases with heat flux, and can be negligible at a low heat flux and a low saturation temperature.

The average tube bundle spray evaporation coefficient data are also used to study the effects of a number of parameters including heat flux, saturation temperature, spray flow rate, nozzle height, and nozzle type, discussed as follows.

2.1. Effect of heat flux

As displayed in all data figures in this work, all tube bundle spray evaporation coefficient data generally increase with heat flux, due to more active nucleate boiling at a higher heat flux. During the experiment, a larger quantity of vapor bubbles was observed at a higher heat flux. This result is in agreement with that of spray evaporation of ammonia on square-pitch tube bundle and single tube [1,3], as well as spray evaporation of R-134a on square-pitch and triangular-pitch tube bundles of plain and enhanced tubes [4]. The result of Moeykens and Pate [5] also demonstrate that for spray evaporation of R-134a on a single plain-tube, heat transfer coefficient increases with heat flux before dryout takes place. The trend also agrees with Conti [6] and Parizhskiy et al. [7] for ammonia, Parken et al. [8], Chyu and Bergles [9], Fletcher and Sernas [10] for water, and Fletcher et al. [11] for sea water.

2.2. Effect of saturation temperature

The effect of saturation temperature is demonstrated by the $\bar{h}$ data with standard-angle nozzles in Fig. 1. Curves 2 and 5 both obtained at $d = 10.2$ cm and $\Gamma_1 = 0.145 \pm 0.003$ kg/s m show that $\bar{h}$ increases with T_s, a well-known trend due to the fact that the wall superheat required to activate a given size of cavity decreases with saturation temperature. Chun and Seban [12] reported lower superheat required for incipient nucleate boiling in a liquid film at a higher superheat. During the present experiment, more vapor bubbles were observed at a higher saturation temperature, particularly at the bottom of the tube bundle. In spray evaporation of ammonia on a plain-tube [3] and a square-pitch plain-tube bundle [1], more vapor bubbles were also observed at a higher saturation temperature. However, the effect of saturation temperature on spray evaporation coefficient is less significant in the range of high saturation temperature, as revealed by comparing curves 1, 3, and 4 at $d = 5.08$ cm and $\Gamma_1 = 0.732 \pm 0.025$ kg/s m in Fig. 1. The data of −1.1°C are close to that of 10°C. A similar result is shown in Fig. 2 for wide-angle nozzles at $d = 10.2$ cm and $\Gamma_1 = 0.145 \pm 0.003$ kg/s m. Fig. 3 also demonstrates $\bar{h}$ increasing with T_s at $d = 5.08$ cm and $\Gamma_1 = 0.145 \pm 0.003$ kg/s m, but the effect is less significant in the range of high T_s and high $\bar{q}''$. All the data at −23.3°C are markedly lower than those at higher temperatures, and data at this low temperature are close to pool boiling (Fig. 3, Part I), a result similar to that of single tube [3] and square-pitch bundle [1].

2.3. Effect of spray flow rate

The average heat transfer coefficients with standard-angle nozzles at $T_s = -1.1$°C, $d = 5.08$ cm, and different spray flow rates are compared in Fig. 4. It is demonstrated that the increase in $\bar{h}$ is less than 10% for the spray flow rate to increase from 0.144 to 0.724 kg/s m. As discussed in Part I of the present paper, a larger spray flow rate results in higher heat transfer coefficients in the top row due to a stronger liquid droplet impingement effect, while the bottom row coefficients basically remain the same. Such high coefficients of tubes in the top row account for the minor increase in the average bundle coefficient.

A weak spray flow rate effect is also shown in Fig. 5 for wide-angle nozzles under the same T_s and d as in Fig. 4. In Fig. 6, results of standard-angle nozzles at $d = 5.08$ cm and different values of Γ_1 are compared. The effect of Γ_1 on $\bar{h}$ is negligible as shown by comparing curves 1 and 2 for −23.3°C and curves 3 and 4 for 10°C.

The present results mostly agree with that of Zeng et al. [1] that spray evaporation heat transfer coefficient of a square-pitch bundle does not vary significantly with spray flow rate except at a high saturation temperature when standard-angle nozzles are used. Zeng et al. [3] also reported that spray evaporation heat

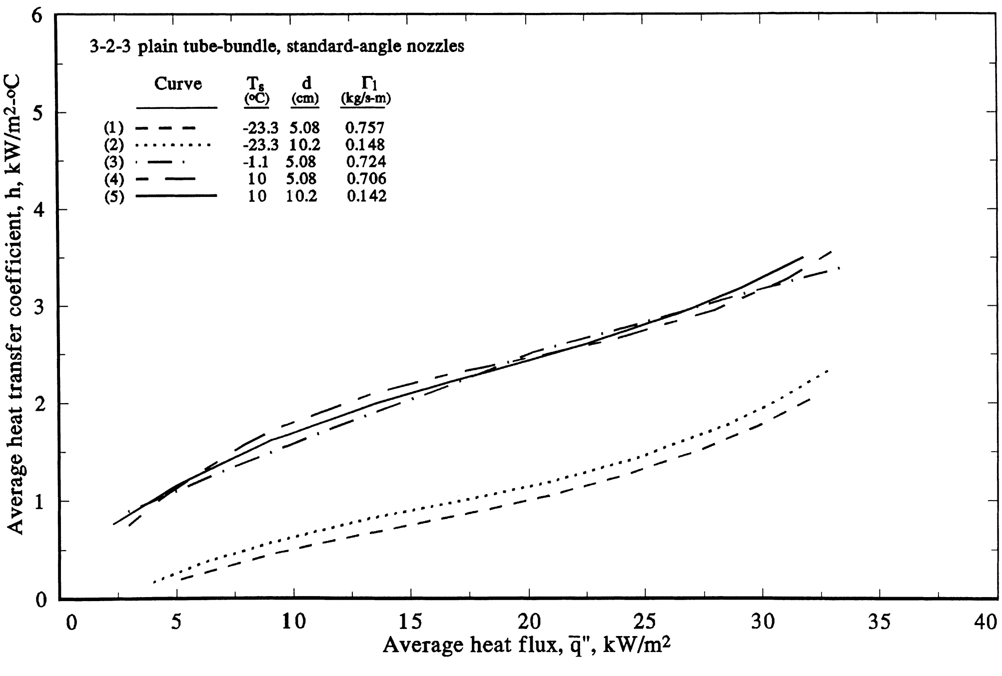

Fig. 1. Effect of saturation temperature on spray evaporation performance for standard-angle nozzles.

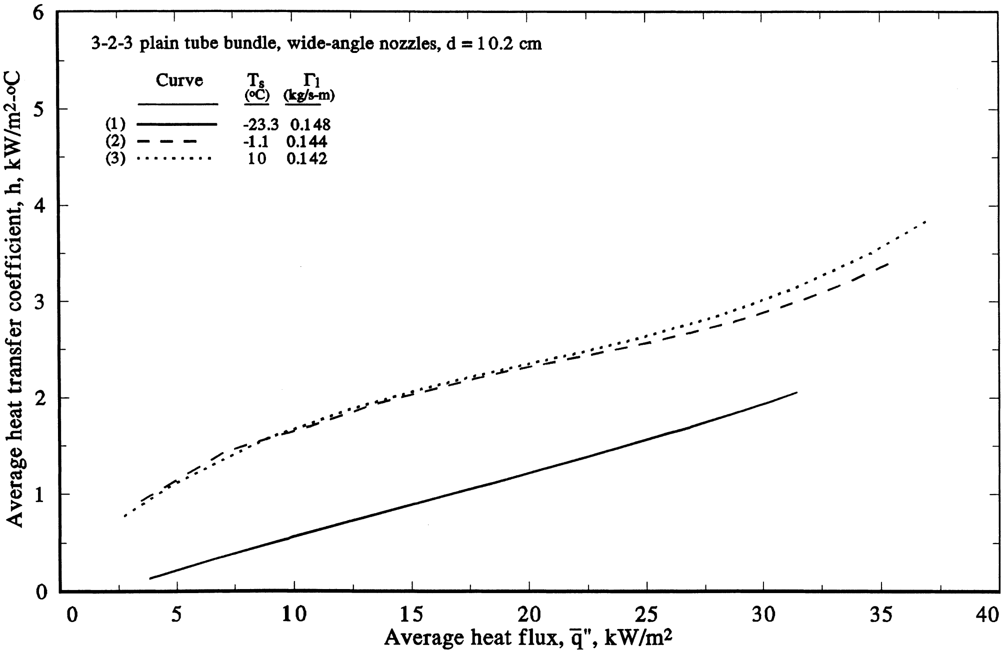

Fig. 2. Effect of saturation temperature on spray evaporation performance for wide-angle nozzles, $d = 10.2$ cm.

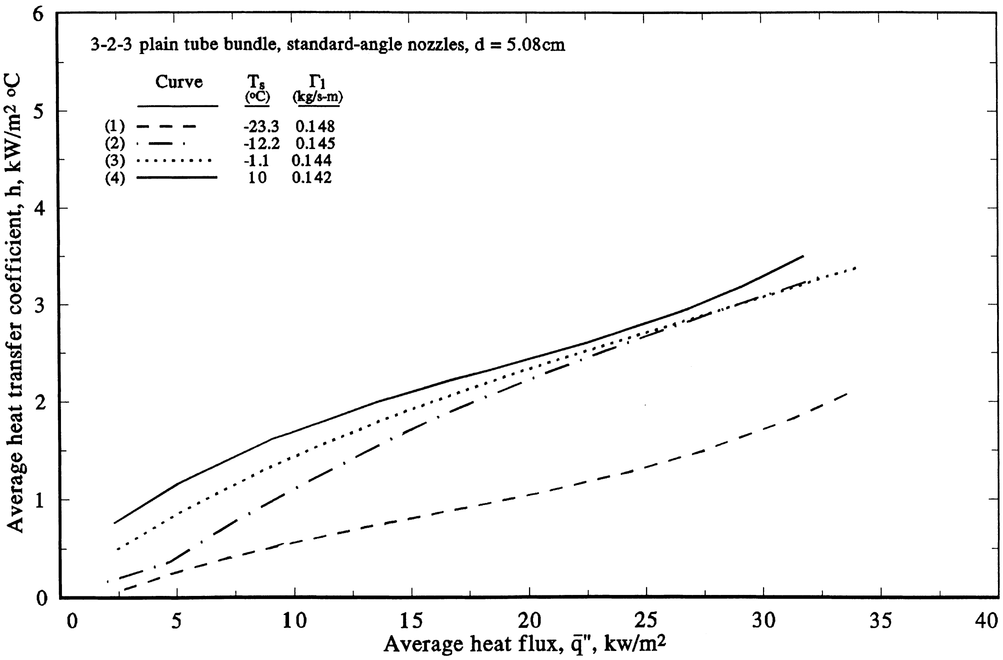

Fig. 3. Effect of saturation temperature on spray evaporation performance for standard-angle nozzles, $d = 5.08$ cm.

transfer coefficient on a single plain-tube does not vary significantly with spray flow rate at low saturation temperatures up to $T_s = -1.1$°C, a result consistent with that of Chyu and Bergles [9], and Conti [6]. Moeykens and Pate's [5] data indicated that spray evaporation performance of R-134a on a plain-tube is weakly dependent on liquid feed rate until dryout occurs. In their spray evaporation test with R-123 on a

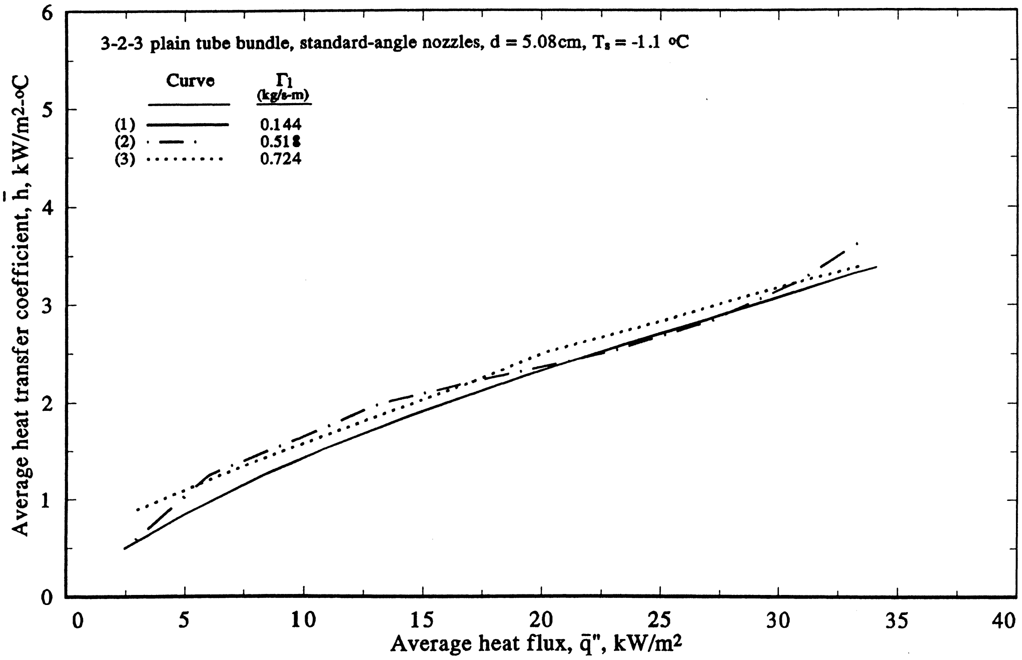

Fig. 4. Effect of spray flow rate on spray evaporation performance for standard-angle nozzles at −1.1°C.

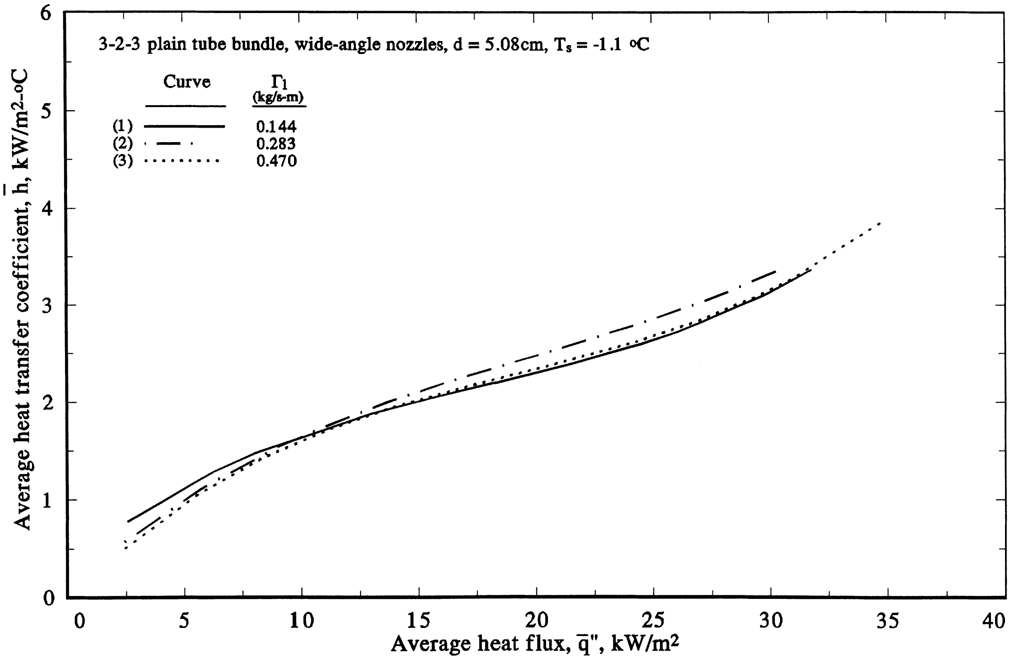

Fig. 5. Effect of spray flow rate on spray evaporation performance for wide-angle nozzles.

plain-tube bundle, Moeykens et al. [13] reported up to 9% variation in the heat transfer coefficients due to variation of spray flow rate; however, the uncertainty of their data can be as larger as 11%. The spray evaporation data of R-134a on triangular-pitch bundles of plain-tube and enhanced tubes by Moeykens et al. [4] demonstrated a moderate dependence on film flow rate in the upper heat flux range. They also indicated

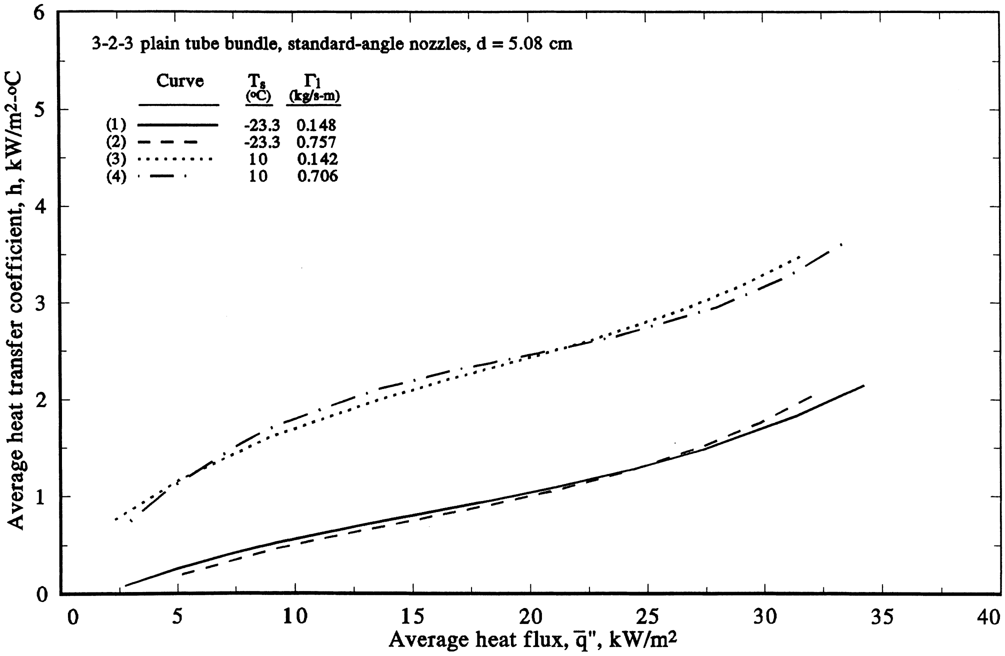

Fig. 6. Effect of spray flow rate on spray evaporation performance for standard-angle nozzles at $d = 5.08$ cm.

that increased heat transfer performance does not always occur with increased spray flow rate. Similar result is shown in the present Figs. 4–6 that $\bar{h}$ may moderately increase or decrease with spray flow rate depending upon the heat flux range.

2.4. Effect of nozzle height

The effect of nozzle height on spray evaporation heat transfer performance using standard-angle nozzles is demonstrated in Fig. 7 where curves 1 and 2 show that at a low temperature (−23.3°C), heat transfer coefficient slightly increases with nozzle height. Even though the effect is not significant considering the 13% uncertainty of the experimental data, the data demonstrated very good repeatability in terms of such slight increase. Curves 3 and 4 demonstrate that heat transfer coefficient slightly decreases with nozzle height at a high temperature (10°C) while the effect was again moderate. For both temperatures, $\bar{h}$ is less dependent on d at a low heat flux.

The effect of nozzle height on wide-angle nozzle performance is similar to the standard-angle nozzle, as shown in Fig. 8. Curves 1 and 2 show that heat transfer coefficient slightly increases with nozzle height at −23.3°C, while curves 5 and 6 demonstrate that $\bar{h}$ slightly decreases with d at 10°C. Further, curves 3 and 4 show that $\bar{h}$ varies little with d at −1.1°C. It is thus concluded that the effect of nozzle height on spray evaporation coefficient is not significant, and the weak effect varies from positive to negative as saturation temperature increases. Spray evaporation heat transfer coefficient of ammonia in a square-pitch bundle also slightly decreases with nozzle height at high T_s [1]. In addition, a weak effect of nozzle height was reported on the spray evaporation heat transfer performance of R-134a on a low-finned, triangular-pitch tube bundle [14]. In contrast, spray evaporation coefficient of ammonia on a single tube increases with d at a high T_s [3].

2.5. Effect of wide-angle nozzle

During the experiment, it was observed that liquid droplets generated by the wide-angle nozzles were smaller in size and moving at a lower velocity compared with the standard-angle nozzles. The results with standard-angle nozzle and wide-angle nozzle at $d = 5.08$ cm are compared in Fig. 9. Curves 1 and 2 are both based on data at −23.3°C, while curves 3 and 4 are both at 10°C. For both high and low saturation temperatures, heat transfer coefficients of standard-angle and wide-angle nozzles are very close. Similar behavior is observed for $d = 10.2$ cm by comparing curves 1 and 2 at −23.3°C, and curves 3 and 4 at 10°C in Fig. 10. The present triangular-pitch bundle result is similar to that of a square-pitch tube bundle at low T_s that the coefficients of the standard-angle nozzle and wide-angle nozzle are about the same [1]. However, for a square-pitch tube bundle at high T_s, the wide-angle nozzle coefficient is slightly higher than the standard-angle nozzle coefficient. In the spray evaporation test with R-134a on a plain-tube

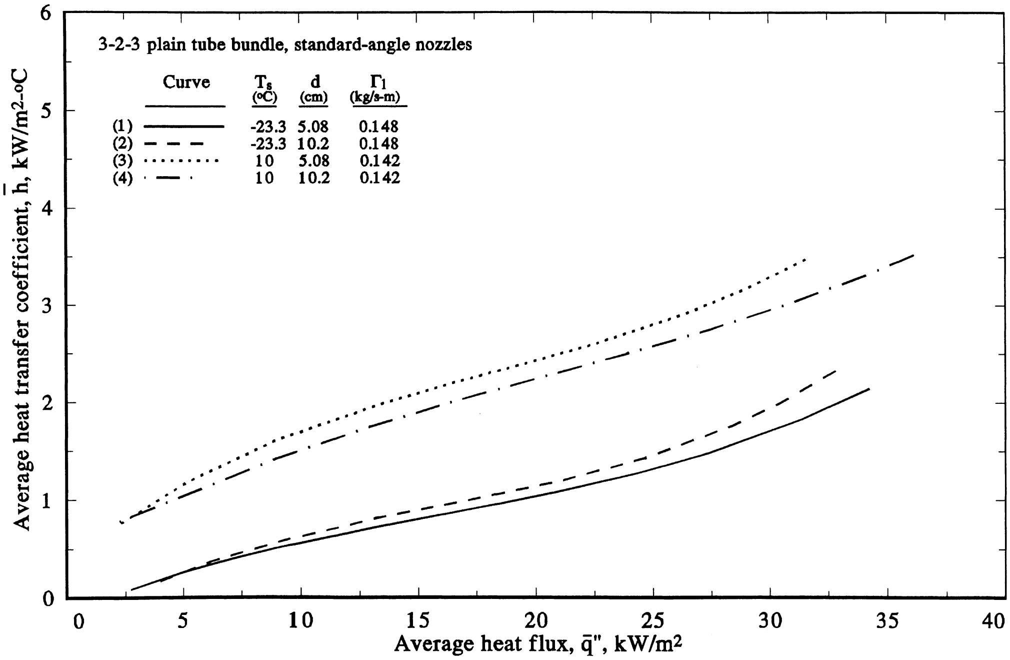

Fig. 7. Effect of nozzle height on spray evaporation performance for standard-angle nozzles.

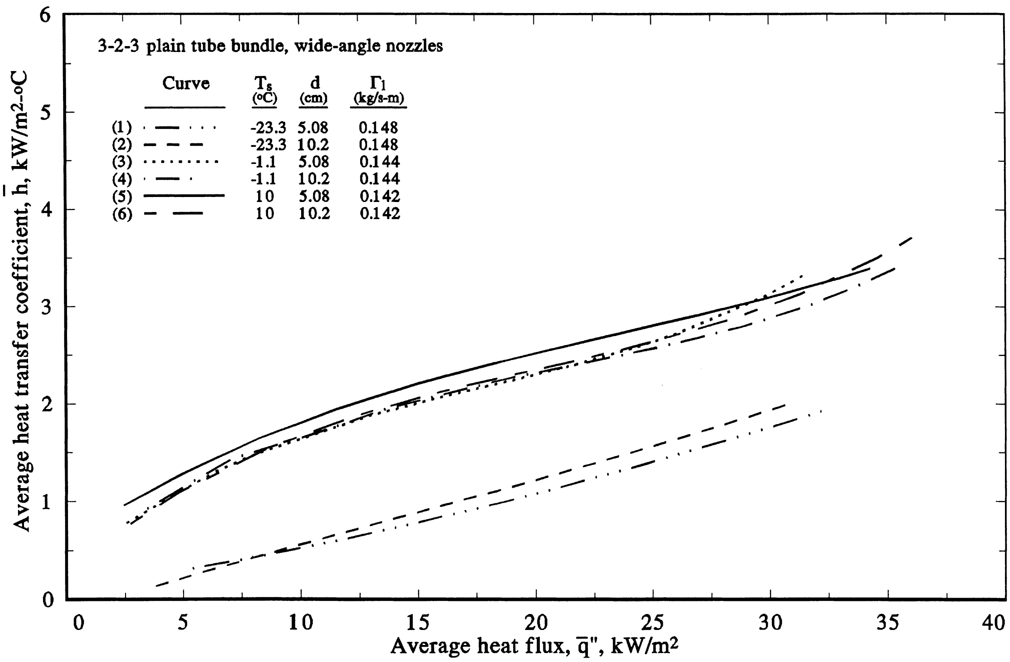

Fig. 8. Effect of nozzle height on spray evaporation performance for wide-angle nozzles.

conducted by Moeykens and Pate [5], the wide-angle nozzle data (Fig. 8 of [5]) are also close to the high-pressure nozzle data (Fig. 10 of [5]). However, the saturation temperatures of the two sets of data are different.

2.6. Optimal operating condition

The above results revealed that in order to achieve a high overall spray evaporation heat transfer performance with a triangular-pitch tube bundle, the

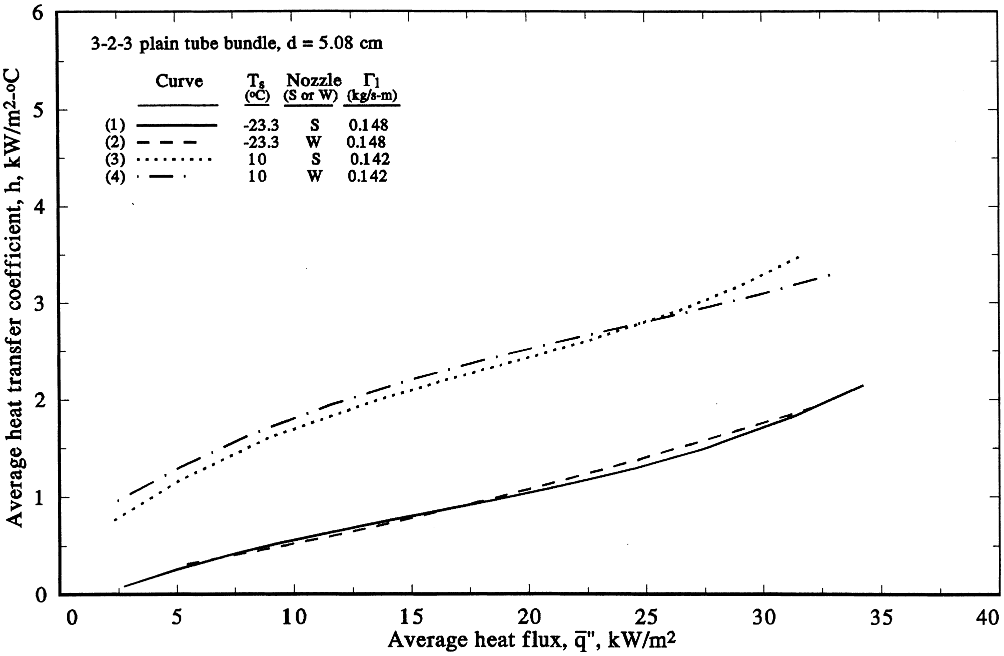

Fig. 9. Comparison of spray evaporation performance between standard-angle and wide-angle nozzles at $d = 5.08$ cm.

operating heat flux and saturation temperature need to be high, while spray flow rate is not important. The nozzle height is also not important, although a small gain may be possible by employing a large *d* at a low saturation temperature and a small *d* at a high saturation temperature insofar as the entire bundle is well covered by the spray. There is no significant difference in terms of heat transfer performance between the

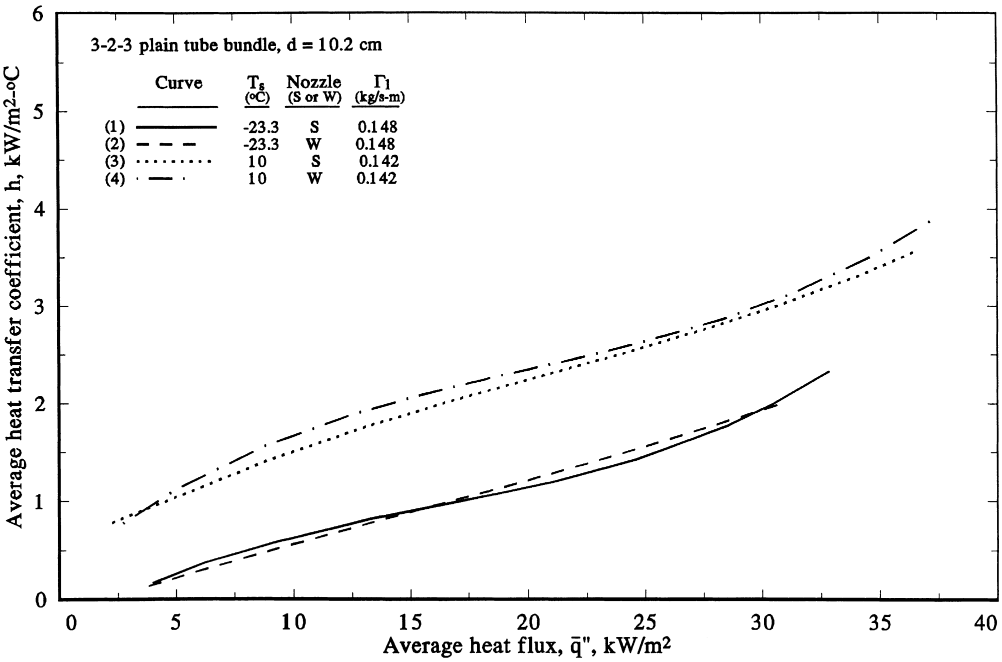

Fig. 10. Comparison of spray evaporation performance between standard-angle and wide-angle nozzles at $d = 10.2$ cm.

standard-angle nozzle and wide-angle nozzle. The number of wide-angle nozzles required will be smaller than standard-angle nozzles because the former can cover a larger area. However, wide-angle nozzles are usually more expensive than standard-angle nozzles of similar capacity.

2.7. Correlation of experimental data

In their work on square-pitch plain-tube bundle, Zeng et al. [1] developed a correlation for spray evaporation coefficient of liquid distributed by nozzles over a wide range of saturation temperature/pressure. Their correlation was based on the following equation proposed by Chun and Seban [12] for turbulent falling film evaporation on a vertical plain-tube:

$$h\left(\frac{v^2}{gk^3}\right)^{1/3} = a_1 Re^{a_2} Pr^{a_3} \tag{1}$$

with a_1, a_2, and a_3 being coefficients determined based on experimental data. This equation was adopted to correlate horizontal-tube falling-film evaporation data in several studies, including that of Owens [15] for non-boiling turbulent ammonia film evaporation on horizontal tube, and that of Parken et al. [8] for similar water film. However, the above equation does not take into account the heat flux effect which was evidently important as shown by the present and published experimental data. For the heat transfer coefficient of a boiling liquid film, Parken et al. [8] proposed the following equation that included heat flux:

$$h\left(\frac{v^2}{gk^3}\right)^{1/3} = a_1 Re^{a_2} Pr^{a_3} q'' a_4. \tag{2}$$

In order to make the above equation non-dimensional, instead of using heat flux, Zeng et al. [1] proposed a dimensionless heat flux, Φ. In addition, reduced pressure (P_{red}) was employed to account for the dependence on saturation temperature. The result was the following equation [1]:

$$Nu = a_1 Re^{a_2} Pr^{a_3} P_{\text{red}}^{a_4} \Phi^{a_5}, \tag{3}$$

where

$$Nu = \frac{h}{k}\left(\frac{v^2}{g}\right)^{1/3}, \tag{4}$$

$$Re = \frac{2\Gamma_1}{\mu}, \tag{5}$$

$$P_{\text{red}} = \frac{P_s}{P_{\text{cri}}}, \tag{6}$$

$$\Phi = \frac{q''D}{(T_{\text{crit}} - T_s)k}. \tag{7}$$

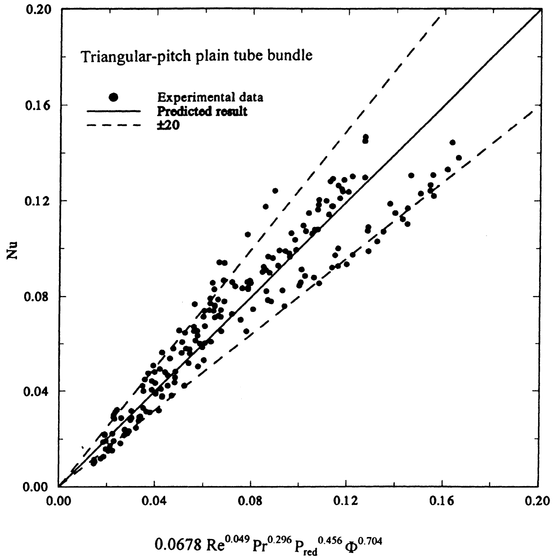

Fig. 11. Correlation of triangular-pitch plain tube bundle spray evaporation data.

The above form of equation was employed to correlate ammonia spray evaporation data for both single plain-tube and square-pitch plain-tube bundle [1], and single low-fin tube [16]. In the present work, the equation was applied to correlate the ammonia spray evaporation data of triangular-pitch plain-tube bundle, and the following values of coefficients were obtained:

$$a_1 = 0.0678, \quad a_2 = 0.049, \quad a_3 = 0.296,$$
$$a_4 = 0.456, \quad a_5 = 0.704.$$

It is noted that the small value of a_1 reflects the weak influence of spray flow rate on the heat transfer coefficient. The large values of a_4 and a_5 indicate the strong dependence of heat transfer coefficient on saturation temperature and heat flux. The spray flow rate Γ_1 in Re should be determined following the method of Chyu et al. [17]. The above correlation is compared with the experimental data in Fig. 11. It is shown that the correlation can cover the experimental data within ±20%.

2.8. Comparison with square-pitch bundle and single tube

The spray evaporation heat transfer coefficients of ammonia with triangular-pitch bundle and square-pitch bundle [1] were compared, and the result is summarized in Table 1. The comparison is based on two-tube bundles of identical diameter, geometry of the tubes, and pitch ratio. In other words, the two-tube bundles are identical except the bundle pattern. In Table 1, the coefficients of 14 tests of the two-bundle configurations are compared, with the test parameters (T_s, Γ_1, d, and nozzle type) listed in four left columns. The next column

Table 1
Comparison of spray evaporation performance between square-pitch and triangular-pitch plain tube bundles

Test condition				Configuration of higher h, S, T, or E[a]	Maximum difference in h (%)
T_s (°C)	Γ_1 (kg/s m)	d (cm)	Nozzles S or W[b]		
−23.3	0.148	5.08	S	S	30
−23.3	0.148	5.08	W	S	55
−23.3	0.148	10.2	S	E	<3
−23.3	0.148	10.2	W	S	28
−23.3	0.757	5.08	S	S	25
−12.2	0.145	5.08	S	T	20
−1.1	0.144	5.08	S	T	18
−1.1	0.47	5.08	W	E	<3
−1.1	0.724	5.08	S	T	28
10	0.142	5.08	S	T	34
10	0.142	5.08	W	T	14
10	0.142	10.2	S	E	<3
10	0.142	10.2	W	T	6
10	0.706	5.08	S	E	<3

[a] S – square-pitch tube bundle; T – triangular-pitch tube bundle; E – equal performance.
[b] S – standard-angle nozzles; W – wide-angle nozzles.

indicates the configuration that provides a higher coefficient under a particular test condition. The right column lists the maximum difference in h between two-bundle configurations. For example, the first entry indicates that at $T_s = -23.3$°C, $\Gamma_1 = 0.148$ kg/s m, $d = 5.08$ cm, with standard-angle nozzles, the square-pitch bundle provides an average heat transfer coefficient up to 30%, depending on heat flux, higher than that of triangular-pitch bundle. Table 1 suggests that a square-pitch bundle tends to provide a higher heat transfer performance than a triangular-pitch bundle at a low saturation temperature, while at a high saturation temperature, the performance of a triangular-pitch bundle is more likely to be higher than a square-pitch bundle. At a low saturation temperature (−23.3°C), the average spray evaporation coefficient of a square-pitch bundle can be as much as 55% higher than a triangular-pitch bundle, while at a high saturation temperature (10°C), the coefficient of a triangular-pitch bundle can be as much as 34% higher than square-pitch bundle.

The higher spray evaporation coefficient provided by a square-pitch bundle at a low temperature is considered to be due to strong single-phase flow convection. At a low saturation temperature, single-phase flow convection is an important mode of heat transfer although nucleate boiling may be also important. The geometry of a square-pitch bundle allows fluid to flow freely between tube columns, therefore achieving higher single-phase convective heat transfer. On the other hand, the zigzag flow passages available in a triangular-pitch bundle tend to slow down fluid flow between tubes, and making the convective heat transfer weaker. Therefore, a square-pitch bundle is more likely to provide higher heat transfer coefficients than triangular-pitch bundle at a low saturation temperature.

In addition, the higher heat transfer coefficients of a square-pitch bundle at a low saturation temperature can be also due to a favorable thermal environment for nucleate boiling between tubes. Low-velocity wake regions exist between tubes in both square-pitch and triangular-pitch tube bundles. However, in a square-pitch tube bundle, a larger portion of the tube surface area is exposed to the low-velocity regions than a triangular-pitch bundle. Low flow velocity provides a more favorable thermal environment for nucleate boiling than high flow velocity. Therefore, more active nucleate boiling, and thus higher boiling heat transfer coefficient is anticipated with a square-pitch tube bundle at a low saturation temperature.

At a high saturation temperature, more active nucleate boiling takes place, and the heat transfer performance of a triangular-pitch bundle tends to be higher than that of a square-pitch bundle due to a more effective two-phase flow convective heat transfer. As discussed in Part I of the present work, it is a geometrical fact that the space between tubes in a triangular-pitch bundle is narrower than that in a square-pitch bundle of the same pitch ratio. The narrower flow passages make it more likely for vapor bubbles to be in contact with tube walls. The zigzag passages between tubes in a triangular-pitch bundle also increase the chance for bubbles to impinge onto and to slide across tube walls. In addition, the distance that a bubble has to travel across a triangular-pitch bundle is longer than that of a square-pitch bundle. The more contact between tubes and flowing bubbles, the better the overall heat transfer performance because of the increased turbulence induced by bubbles impinging onto and sliding over the tube walls, as well as thin film evaporation on tube walls as bubbles slide over. Therefore, the heat transfer coef-

Table 2
Summary of effects on spray evaporation performance on plain single tube and tube bundles[a]

Configuration	q''	T_s	Γ_1	d	β
Square-pitch bundle	I	I	N–S, low T_s I–S, high T_s N–W	N, low T_s D, high T_s	N, low T_s I, high T_s
Triangular-pitch bundle	I	I	N	I, low T_s D, high T_s	N
Single tube	I	I	N, low T_s I, high T_s	N, low T_s I, high T_s	N, low T_s D, high T_s

[a] I – heat transfer coefficient increases with parameter; D – heat transfer coefficient decreases with parameter; N – heat transfer coefficient does not change with parameter, or effect is not clear; S – standard-angle nozzle; W – wide-angle nozzle.

ficient of a triangular-pitch bundle is higher than that of a square-pitch bundle at a high saturation temperature.

Bundle geometry effect was also investigated by Moeykens et al. [14] by comparing results of R-134a spray evaporation on square-pitch and triangular-pitch bundles of an enhanced boiling tubing (Turbo-B). The triangular-pitch bundle was found to provide higher heat transfer coefficients at high heat fluxes and lower heat transfer coefficients at low heat fluxes than the square-pitch bundle. However, in a spray evaporation study of R-123, Moeykens et al. [13] found the square-pitch tube bundle yielded heat transfer coefficients 6–17% higher than the triangular-pitch tube bundle. No comparison can be made with the current result because in each study, Moeykens et al. [13,14] only conducted tests at one saturation temperature, and their tube surface geometry was totally different than the present plain test tubes.

The effects of various parameters including heat flux, saturation temperature, spray flow rate, nozzle height, and nozzle type on spray evaporation heat transfer coefficients of square-pitch tube bundle, triangular-pitch tube bundle, and single tube are compared in Table 2. The comparison is based on the square-pitch bundle data of the same pitch ratio and the same tube size as the present triangular-pitch bundle [1], and single tube data of the same size [3]. It is concluded that for all configurations, spray evaporation coefficient increases with heat flux and saturation temperature. The coefficient also slightly increases with spray flow rate except for triangular-pitch tube bundle. Wide-angle nozzle either has no significant effect or slightly increases bundle coefficient, but it may decrease that of single tube.

3. Conclusions

In the present study on spray evaporation heat transfer of ammonia distributed by spray nozzles onto a horizontal 3–2–3 triangular-pitch plain-tube bundle, the effects of heat flux, saturation temperature, spray flow rate, nozzle height, and nozzle type were investigated. The results reveal that spray evaporation heat transfer coefficient primarily increases with heat flux and saturation temperature. Spray flow rate has little influence on spray evaporation coefficient. The effect of nozzle height on spray evaporation coefficient is weak, and varies from positive to negative as saturation temperature increases. It is also shown that coefficients with wide-angle nozzles and standard-angle nozzles are close. The spray evaporation coefficient can be up to 65% higher than pool boiling coefficient. The present triangular-pitch spray evaporation data can be correlated by an equation similar to those for single tube and square-pitch tube bundle. Comparison showed that square-pitch bundle tends to provide a higher spray evaporation coefficient than triangular-pitch bundle at a low saturation temperature, and triangular-pitch bundle is more likely to provide a higher coefficient at a high saturation temperature. Ammonia spray evaporation performance of single tube, square-pitch tube bundle and triangular-pitch tube bundle were also compared in this study.

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