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Effect of Inlet Direction on the Refrigerant Distribution in an Aluminum Flat-Tube Heat Exchanger

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Abstract

The refrigerant R-134a flow distributions are experimentally studied for a round header/ten flat tube test section simulating a brazed aluminum heat exchanger. Three different inlet orientations (parallel, normal, vertical) were investigated. Tests were conducted with downward flow for the mass flux from 70 to 130 kg/m^{2}s and quality from 0.2 to 0.6. In the test section, tubes were flush-mounted with no protrusion into the header. It is shown that normal and vertical inlet yielded approximately similar flow distribution. At high mass fluxes or high qualities, however, slightly better results were obtained for normal inlet configuration. The flow distribution was worst for the parallel inlet configuration. Possible explanation is provided based on flow visualization results.

Key words: Parallel flow heat exchanger, Header, Two-phase distribution, R-134a, Inlet direction

Nomenclature

c_p : specific heat [kJ/kgK]
D : header inner diameter [m]
G : mass flux [kg/m^{2}s]
h : protrusion depth [m], enthalpy [kJ/kg]
m : mass flow rate [kg/s]
T : temperature [K]
x : quality

Lower character

f : saturated liquid
g : saturated vapor
i : inlet
o : outlet
r : refrigerant
w : cooling water

1. Introduction

Brazed aluminium heat exchangers, which consist of flat tubes on the refrigerant-side and louver fins on the air-side, are seriously considered as evaporators of residential air conditioners, due to superior thermal performance as compared with conventional fin-tube heat exchangers. Typical hydraulic diameter of the flat tube is 1 ~ 2 mm. To manage the excessive tube-side pressure drop by the small channel size, a number of tubes are grouped to one pass using a header. In an evaporator, it is very important to distribute the two-phase refrigerant (especially the liquid) evenly into each tube. Otherwise, the thermal performance is significantly deteriorated. According to Kulkarni et al.[1], the performance reduction by flow mal-distribution could be as large as 20%.

For an evaporator, vertical flat tube configuration is preferred (with headers in horizontal position), because it facilitates the air-side condensate drainage. In such a case, refrigerant may be supplied from three different directions as shown in Fig. 1. The refrigerant may be supplied parallel to the header, normal to the header and vertical to the header. The outlet may be located at the same side of the heat exchanger with the inlet or it may be located at the opposite side of the heat exchanger. In Fig. 1, the outlet is located at the same side of the heat exchanger. In addition to the inlet direction, many parameters, both flow and geometric, will affect the flow distribution in a parallel flow heat exchanger. Webb and Chung[2], Hrnjak[3] and Lee[4] provided recent reviews on this subject.

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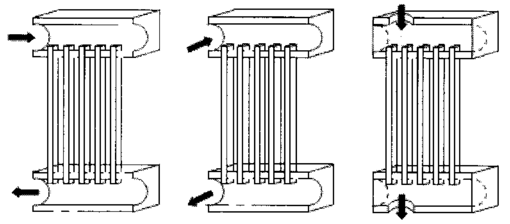
The literature reveals several studies on the two-phase distribution in a header/branch tube configuration. Watanabe et al.[5] conducted a flow distribution study for a round header/four round branch tube upward flow configuration using R-11. At the inlet, flow was supplied parallel to the header. The flow distribution was highly dependent on the mass flux and the quality.

Vist and Pettersen[6] investigated a round header/ten round branch tube configuration using R-134a. Both upward and downward flow were tested. The flow was supplied parallel to the header. For the downward flow configuration, most of the liquid flowed through frontal part of the header. For the upward configuration, on the contrary, most of the liquid flowed through the rear part of the header. The liquid distribution improved as the vapor quality decreased. The mass flux had negligible effect on the flow distribution.

Cho et al.[7] investigated the effect of the header orientation (vertical and horizontal) and the refrigerant inlet direction (parallel, normal, vertical) for a round header/fifteen flat tube configuration using R-22. The header mass flux was fixed at 60 kg/m^{2}s, and the quality varied up to 0.3. For a vertical header configuration, most of the liquid flowed through the frontal part of the header, and the effect of the flow inlet direction was not significant. For a horizontal header, the flow distribution was highly dependent on the inlet direction, and better distribution was obtained for the vertical or the normal flow configuration.

Kim et al.[8] also investigated the effect of flow inlet direction (parallel, normal, vertical) for a round header/ten flat tube configuration. Using air and water, downward flow configuration was tested. Best flow distribution was obtained for the vertical inlet configuration. It was observed that vertically supplied water hits bottom of the header, and is forced to downstream of the header, yielding more uniform distribution. Normal inlet was less effective in distributing the water than the vertical inlet, although it was much more effective than the parallel inlet.

This study is a continuing effort succeeding Kim et al.[8], who investigated the effect of inlet direction using air and water. In this study, R-134 was tested at a room temperature (20°C). Tests were conducted for three inlet directions shown in Fig. 1. The normal direction is 90° facing the test section from the front, and vertical direction is 90° facing the test section from the top. The inlets were located 10 mm upstream of the first branch.



(a) Parallel inlet (b) Normal inlet (c) Vertical inlet

Fig. 1. Flow inlet orientations.

The header mass flux and the quality were varied for $70 \leq G \leq 130$ kg/m^{2}s and $0.2 \leq x \leq 0.6$. For the test range, stratified flow was observed at the inlet of the header. At the same mass flux and quality range, annular flow was observed in the previous air-water study. The vapor-liquid density ratio of R-134a at 20°C is 0.023, which is approximately nineteen times larger than that of air-water. The increased density ratio appears to result in a stratified flow. For all the test samples, tubes were flush-mounted with no protrusion into the header. The inlet and outlet were located at the same side of the heat exchanger as illustrated in Fig. 1. The cross-section of the present flat tube is shown in Fig. 6. The hydraulic diameter of the present flat tube is 1.32 mm, and the flow cross-sectional area is 12.24 mm^2.

2. Experimental apparatus

A schematic drawing of the experimental apparatus is shown in Fig. 2. Detailed drawing of the test section is illustrated in Fig. 3. The test section consists of the 17 mm ID round upper and lower headers, which are 91 cm apart, and 10 flat tubes inserted at 9.8 mm pitches. This configuration was chosen to simulate the actual parallel flow heat exchanger. The headers were made from transparent PVC rods for flow visualization. A 17 mm round hole was machined longitudinally in a square PVC rod (25 mm x 25 mm x 150 mm). Ten flat holes were machined at the bottom of the rod for insertion of flat tubes. An aluminum plate, which had matching flat holes, was installed underneath the header. Flat tubes were secured, and the protrusion depth was adjusted using O-rings between the header and the aluminum plate. Transition blocks were installed at the center of the test section to connect the flat tubes and the 6.0 mm ID round tubes. The round tubes served as flow measurement lines.

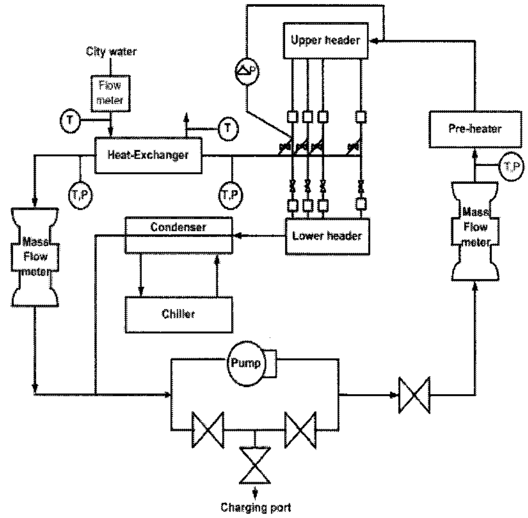


Fig. 2. Schematic drawing of the apparatus.

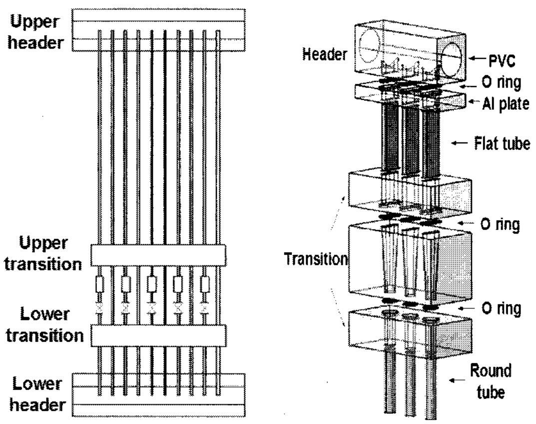


Fig. 3. Detailed drawing of the test section.

At the inlet of the header, 1.0 m long copper tube having the same inner diameter as the header was attached. The tube served as a flow development section. One thing to note is that no heat was applied to flat tubes during tests. Tests were conducted under adiabatic condition.

Before the refrigerant flows into the test section, the quality of the refrigerant is controlled at the preheater (5 kW capacity) section. The refrigerant out of the test section is directed to a condenser, and the condensed refrigerant is pumped to the pre-heater. The flow rate was controlled by by-passing an appropriate amount of liquid from the magnetically coupled gear pump. A mass flow meter (0 - 400 kg/hr) is installed before the pre-heater. The branch channel flow rate was measured for every other channel, by directing the refrigerant to the flow measurement section. As shown in Fig. 4, two valves – one at the main stream, the other at the bypass stream – are installed at every other channel. Normally, main stream valves are open, and bypass stream valves are closed. To measure the flow rate at a certain channel, the main stream valve is closed, and the bypass valve is open. To prevent possible flow pattern change before and during the measurement, the differential pressure between the inlet of the upper header and the transition section was maintained the same by controlling the valve in the transition section. The pressure fluctuations during measurement were within 10% of the average value.

The liquid and vapor fraction (quality) of the flow was thermally determined using a double tube heat exchanger. The measurement section is illustrated in Fig. 5. The tube-side refrigerant was condensed by a city water flowing at the annular side. The double tube heat exchanger was divided into five subsections to cover a wide range of thermal loads from different channels. After branching the refrigerant to the measurement section, the total flow rate slightly changed. The flow rate was adjusted to the original value controlling the valve in front of the pump.

Temperatures were measured at four locations; refrigerant temperatures at inlet and outlet of the tube-side, cooling water temperatures at inlet and outlet of the annular-side. Thermo-wells having five thermocouples each were used to measure local temperatures. Two refrigerant pressures were also measured - one at

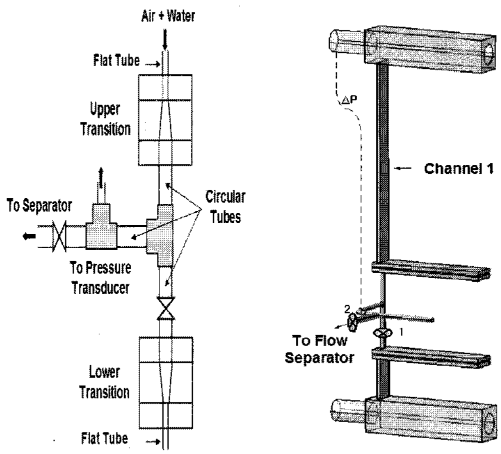


Fig. 4. Schematic drawing illustrating the flow measurement method.

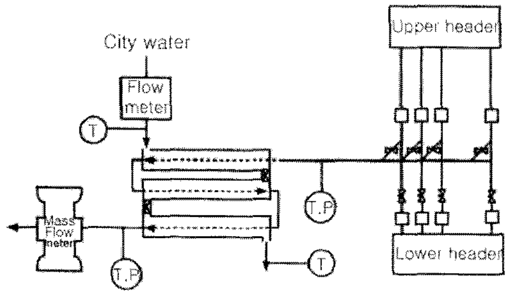


Fig. 5. Schematic drawing of the channel flow rate measurement section.

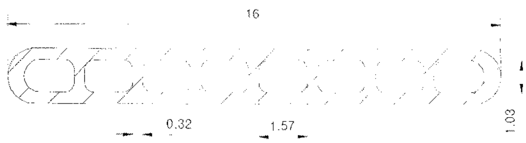


Fig. 6. Cross-sectional view of the flat tube used in this study (unit: mm)

the inlet of the heat exchanger, and the other at the outlet of the heat exchanger. The outlet refrigerant was maintained sub-cooled to insure sub-cooled flow into the flow meter. The quality of each channel is determined from the following equation.

$$x = (h_{r,i} - h_f)/(h_g - h_f) \tag{1}$$

$$h_{r,i} = h_{r,o} + m_w c_{pw}(T_{w,o} - T_{w,i})/m_r \tag{2}$$

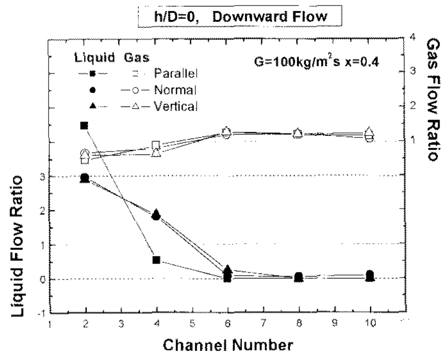
Here, the refrigerant inlet and outlet enthalpy is determined from the measured temperatures and pressures. The refrigerant flow rate was measured using a precision mass flow meter (0 - 100 kg/hr). The water flow rate was measured by weighing the drained water. During the whole series of tests, several runs were made to check the repeatability of the data. The data were repeatable within ± 10%. When the channel flow rates were added and compared with the supplied flow rate (for the channels where flow rates were not measured, the average values of the upstream and downstream channel flow rates were used), they agreed within 10%. The experimental uncertainties are summarized in Table 1.

Table 1. Experimental uncertainties.

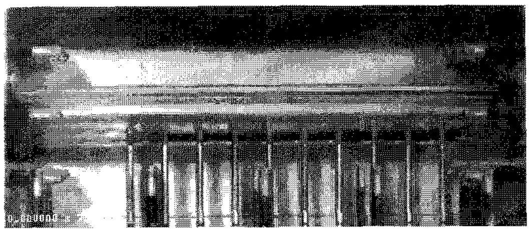
Measurement	Uncertainty
heater power	± 0.2%
mass flow rate	± 0.2%
differential pressure	± 0.3%
liquid flow rate	± 12%
gas flow rate	± 12%

3. Results and discussions

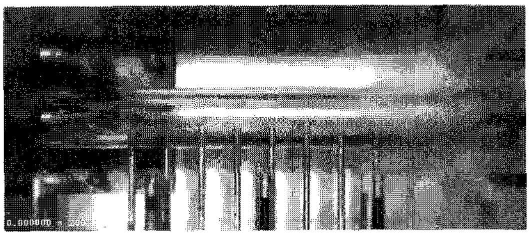
Typical flow distribution data along with flow pattern photos are shown in Fig. 7. The ordinate of the Fig. 7 is the ratio of liquid or gas flow rate in each tube to the average values. For example, assume that



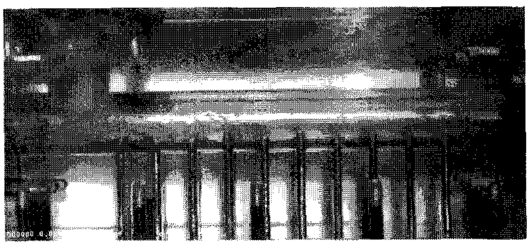
(a) flow distribution data



parallel



normal



vertical

(b) corresponding photos

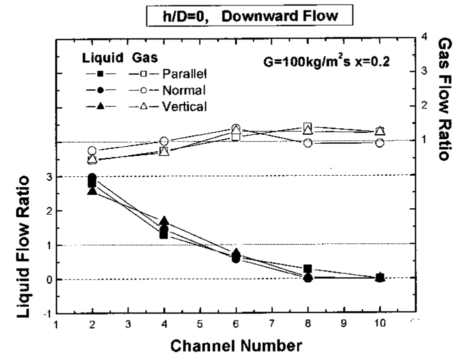
Fig. 7. Flow distribution data and photos at G = 100 kg/m^{2}s, x = 0.4, h/D = 0.5

the total liquid and gas flow rate in the header is 1.0 kg/s and 0.1 kg/s. Then, the average liquid and gas flow rate in each channel (for 10 channels in the header) is 0.1 kg/s and 0.01 kg/s. If the measured liquid and gas flow rate in the branch tube is 0.2 kg/s and 0.01 kg/s, then the liquid flow ratio become 2.0 and gas flow ratio becomes 1.0.

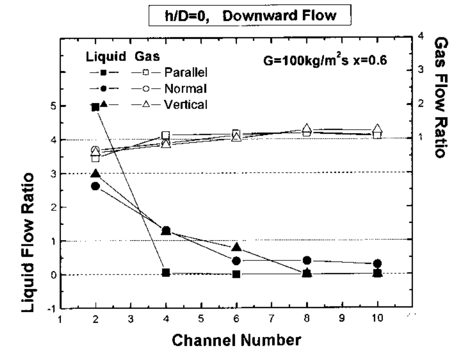
Fig. 7 shows the liquid and gas flow ratio of the three different inlet directions at G = 100 kg/m^{2}s, x = 0.4. For all configurations, significant amount of liquid flows to frontal part of the header. The trend is the most severe for parallel inlet configuration. The liquid flow ratio of second channel is 4.5, decreases to 0.5 at fourth channel and almost no liquid is supplied after sixth channel. The corresponding photo in the same figure confirms that most of the liquid flows into frontal channels, with almost no liquid at latter part of the header.

Fig. 7 shows that the situation improves for normal or vertical inlet configuration, especially at frontal part of the header. For vertical inlet configuration, the liquid flow ratio of second channel is 2.9, decreases to 1.8 at fourth channel and 0.3 at sixth channel. Liquid at front-most part of the header is forced to latter part of the header. The accompanying photos reveal the clue for the improved distribution. For normal inlet configuration, part of the incoming liquid hits rear part of the header, spreads around the header and is delivered downstream. For vertical inlet configuration, vertically supplied liquid hits bottom of the header, spreads around the header, and is forced downstream. Fig. 7 shows that no significant difference in liquid distribution is found between normal and vertical inlet configuration. Calculation of the standard deviation of liquid flow ratio yielded 1.75 for the parallel inlet, 1.19 for the normal inlet and 1.17 for the vertical inlet, confirming better liquid distribution for the normal or vertical inlet configuration.

In Fig. 8, flow distribution data taken at different qualities (x = 0.2 and 0.6) are shown. The mass flux was 100 kg/m^{2}s. Fig. 8 shows that, at a low quality of x = 0.2, the effect of inlet direction on flow distribution is insignificant. At a low vapor quality, the strength of impinging flow caused by normal or vertical inlet will be weak, and the flow distributions become similar irrespective of the inlet direction. At a high quality of 0.6, however, Fig. 8 clearly shows the effect of inlet direction. Standard deviations of the liquid flow distribution are 1.98 for parallel inlet, 0.89 for normal inlet and 1.10 for vertical inlet, suggesting



(a) x = 0.2



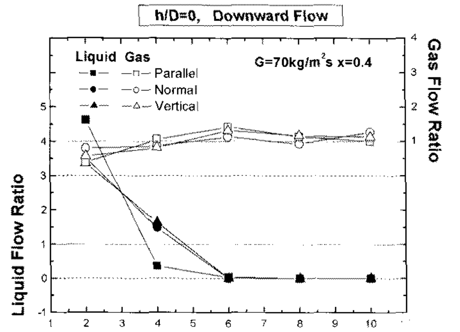
(b) x = 0.6

Fig. 8 Flow distribution data taken at different qualities (x = 0.2 and 0.6)

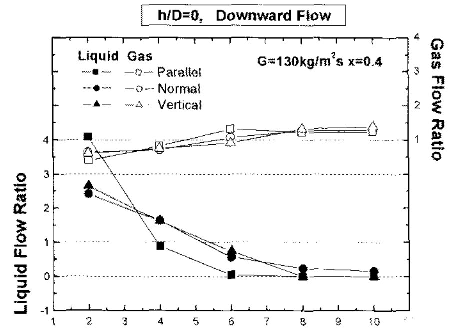
the best distribution for normal inlet. The previous x = 0.4 results shown in Fig. 7 revealed insignificant difference between normal and vertical inlet configurations. Thus, it appears that, as the quality increases, normal inlet become superior to vertical inlet.

Fig. 9 shows the flow distribution data taken at different mass fluxes (G = 70 and 130 kg/m^{2}s). The quality was 0.4. At a low mass flux of 70 kg/m^{2}s, the liquid distributions of the normal and vertical inlet are approximately the same, and they are better than that of parallel inlet. At a high mass flux of 130 kg/m^{2}s, however, normal inlet (with 0.88 standard deviation) is better than vertical inlet (1.02 standard deviation) or parallel inlet (1.58 standard deviation). These results are in contrast to the air-water results by Kim et al.[8], They reported the best distribution for vertical inlet irrespective of mass fluxes or qualities. The difference in the gas velocity, where that of air-water is much larger than that of R-134a, might be responsible for the contrasting trend.

Standard deviations of the liquid flow ratio are summarized in Table 2. Table 2 shows that standard



(a) G = 70 kg/m^{2}s



(b) G = 130 kg/m^{2}s

Fig. 9. Flow distribution data taken at different mass fluxes (G = 70 and 130 kg/m^{2}s).

Table 2. Standard deviations of the liquid flow ratio.

G (kg/m^{2}s)	x	Standard Deviation (Liquid)		
		Parallel	Normal	Vertical
70	0.4	1.81	1.38	1.34
100	0.2	0.99	1.12	0.98
100	0.4	1.75	1.19	1.17
100	0.6	1.98	0.89	1.10
130	0.4	1.58	0.88	1.02

deviations of the parallel inlet are the largest. Standard deviations of normal and vertical inlet are approximately the same, except for the largest mass flux or quality, where those of the normal inlet are smaller. One thing to note is that, for all configurations, standard deviation decreases as mass flux increases. The effect of quality on standard deviation is dependent on inlet configuration. For parallel inlet, standard deviation increases as quality increases. For normal and vertical inlet, however, the effect of quality is not significant.

4. Conclusions

The refrigerant R-134a flow distributions are experimentally studied for a round header/ten flat tube test section simulating a brazed aluminum heat exchanger. Three different inlet orientations (parallel, normal, vertical) were investigated. Tests were conducted with downward flow for the mass flux from 70 to 130 kg/m^{2}s and quality from 0.2 to 0.6. Tubes were flush-mounted in the header with no protrusion. It is shown that normal and vertical inlet yielded approximately similar flow distribution, although slightly better results were obtained for normal inlet at high mass fluxes or high qualities. The flow distribution was worst for the parallel inlet. For parallel inlet, most of the liquid is sucked into frontal channels, with almost no liquid at latter part of the header. For normal inlet configuration, part of the incoming liquid hits rear part of the header, spreads around the header and is delivered downstream. For vertical inlet configuration, vertically supplied liquid hits bottom of the header, spreads around the header, and is forced downstream. For all configurations, flow distribution improves as the mass flux increases.

Acknowledgement

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