MITSUBISHI ELECTRIC

Changes for the Better

# Development of Flat Tube Heat Exchanger for Heat Pump Air Conditioner

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# Contents

# 1. Introduction

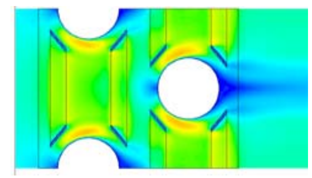
1) Air conditioner market trend

**Prevent Global warming, Enforce energy-saving regulations**

2) Air conditioner manufacture trend

**Improve the efficiency of the device: Comps, heat exchangers, and fans**

**Improve the refrigerant cycle control technology**



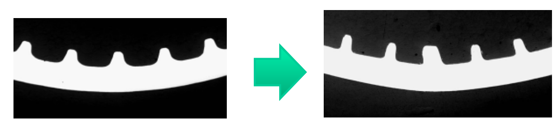
Optimize the fin pattern

3) Improvement of heat exchangers

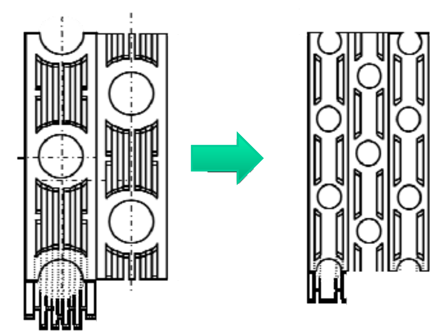
**>Optimize the fin pattern**

**>Optimize the inner fin pattern of the tubes**

**>Smaller tubes**



Optimize the inner fin of the tube



Smaller tube

# 1. Introduction



1) Characteristics of the Flat Tube Heat Exchanger

- All aluminum => **Save the weight**
- Reduce the air side pressure drop => **The # of the tube can be mounted in the high density.**
- Braze the contacting area between fin and the tube => **Reduce the contacting heat resistance**
- Increase the inner heat transfer area

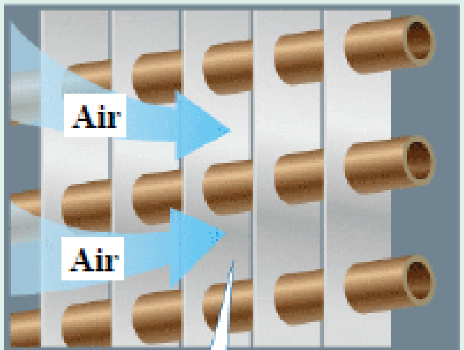
2) Challenge items

- Corrosion resistance
- Manufacturability
- Drainage performance
- Even refrigerant distribution by increasing the # of path

# 1. Introduction

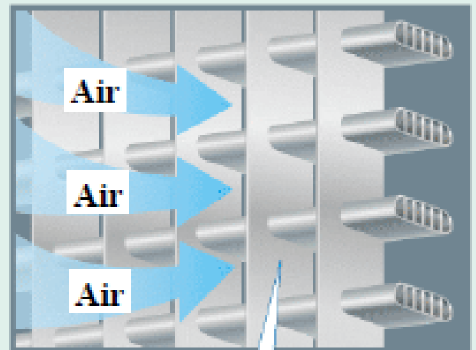


**Conventional Heat Exchanger**

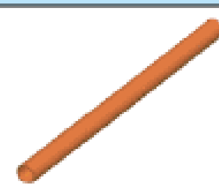
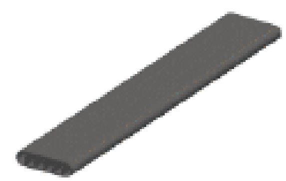
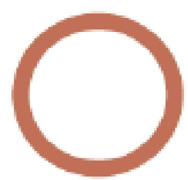
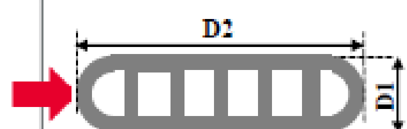



Fin + Circular Tube

**Flat Tube Heat Exchanger**




Fin + Flat tube pipe

| | Conventional | Flat tube |
|---|---|---|
| Material | Copper | Aluminum |
| Shape of the tube |  |  |
| Cross section of the tube |  |  |

# Contents

# 2. Short Length D1 of Flat Tube

Changes for the Better

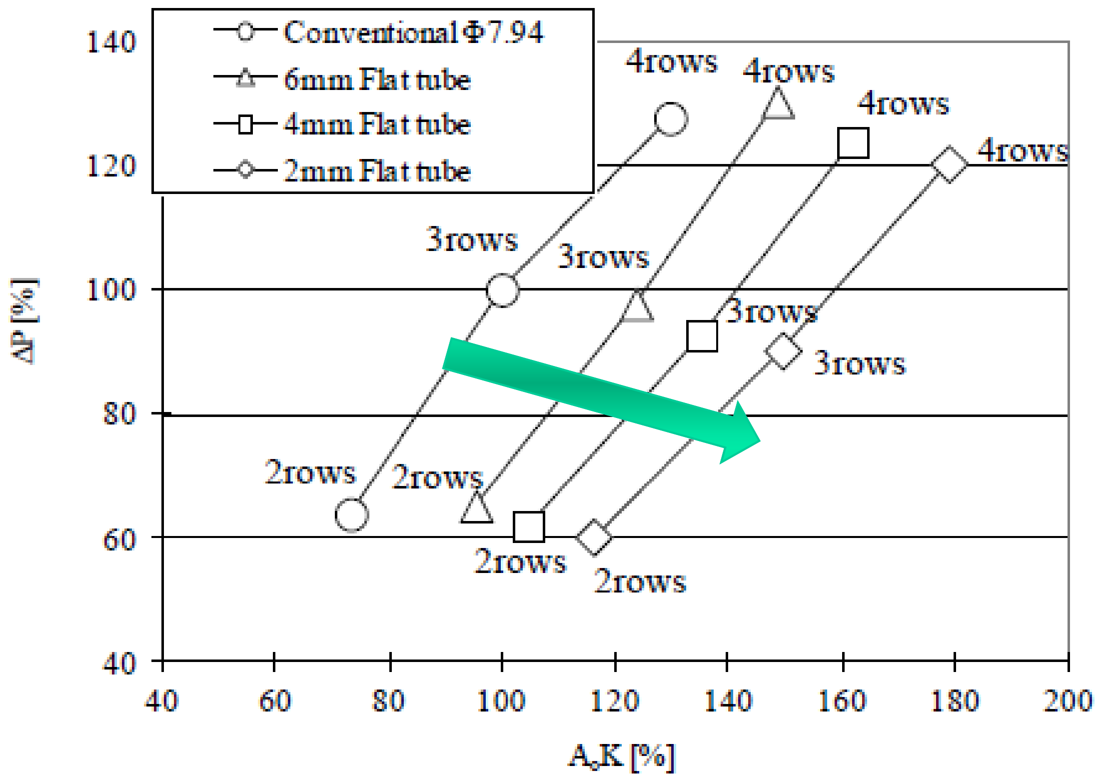



Fig. 2 Relationship between $A_oK$ and ΔP

AoK becomes larger and ⊿P_air decreases as the short length D1 becomes smaller.

# 2. Short Length D1 of Flat Tube



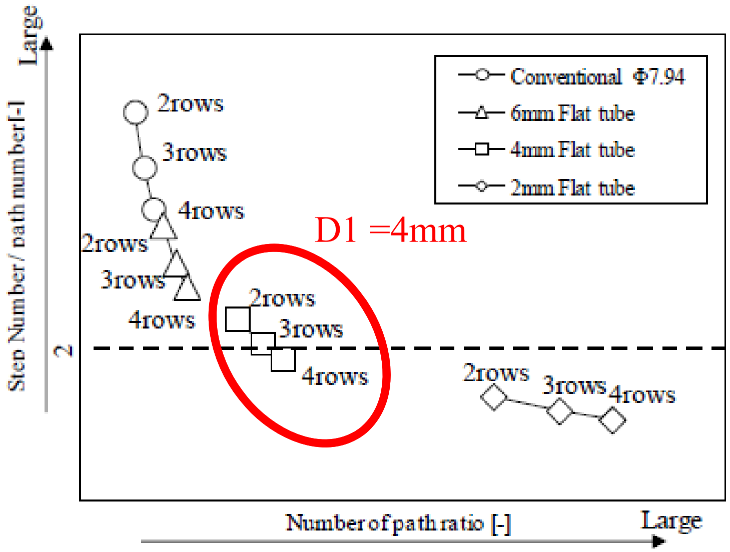



Fig.3 Step number divided by path number

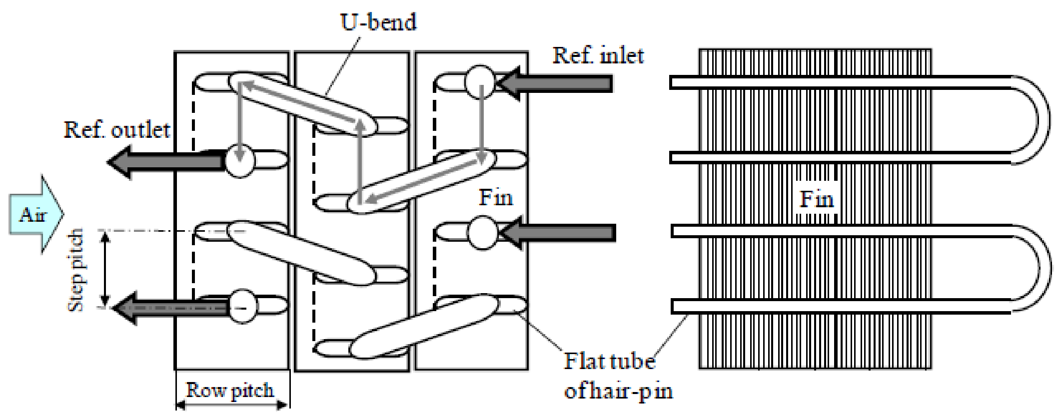



Fig.4 Path pattern of "counter-flow type" and Hair-pin tube

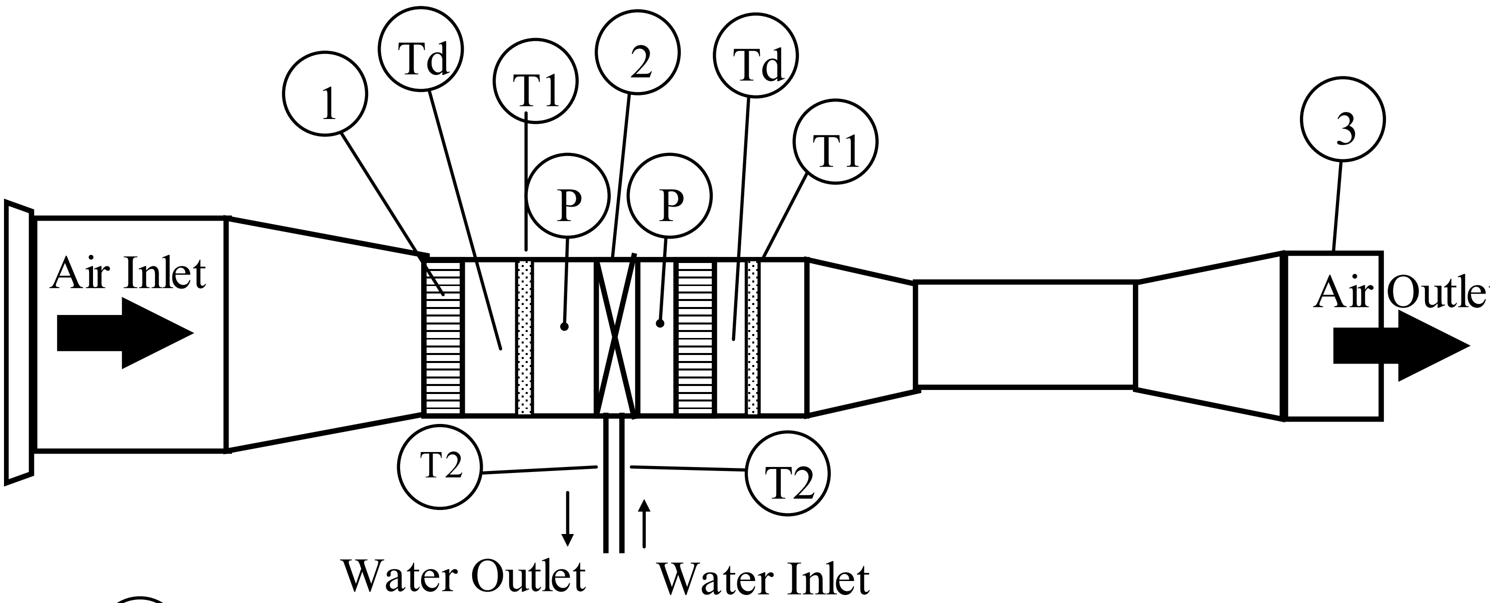
Required item

1) Use flat tube of hair-pin type
2) Use more than 3 rows
3) "Counter flow type" of refrigerant flow

➡ We chose D1 of 4mm because it had the largest AoK.

# Contents

## ◆ Experimental apparatus and test heat exchanger

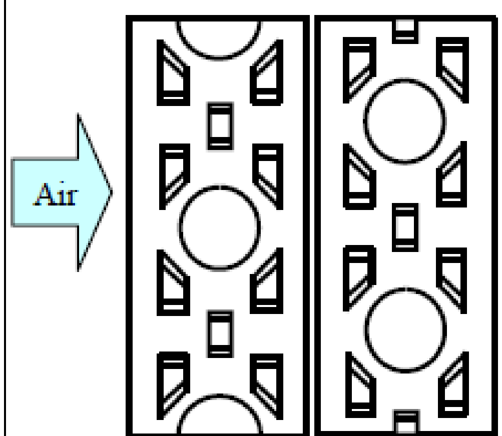
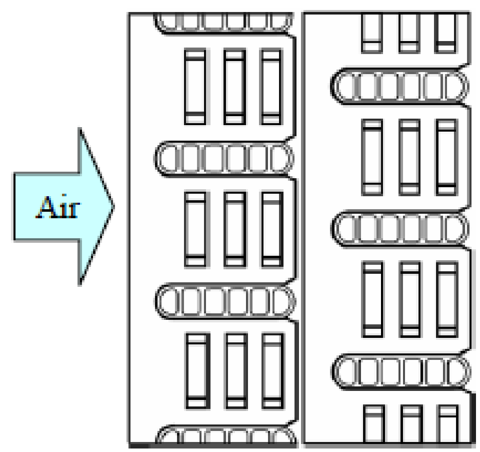



(1) Honeycomb
(2) Test Heat Exchanger
(3) Fan
(P) Pressure Taps
(T1) Thermocouple Grid
(T2) Platinum Thermo resistance
(Td) Dew Point Meter

Experimental apparatus

## ◆ Specifications of test heat exchangers

Table1. Specifications of Test Heat Exchanger

| | **Conventional heat exchanger** | **Flat tube heat exchanger** |
|---|---|---|
| Tube diameter [mm] | 7.94 | 4 x 15 |
| Number of channels [-] | 1 | 6 |
| Step pitch [mm] | 20.4 | 15.3 |
| Row pitch [mm] | 17.7 | 17.7 |
| Number of rows [-] | 2 | 2 |
| Number of path number [-] | 2 | 6 |
| Fin pitch [mm] | 1.6 | 1.6 |
| Step number [-] | 14 | 18 |
| Fin height [mm] | 285.6 | 275.4 |
| Surface treatment | Hydrophilic or Non-Hydrophilic coating | Non-Hydrophilic coating |
| Fin pattern | Cross stair slit   | 3-slit   |

## ◆Experimental condition

Table 2 Experimental Conditions

| Surface Condition of Fin | Dry | Wet | Frost |
|---|---|---|---|
| Inlet Temperature of Air [°C] | 20 | 27 | 2 |
| Inlet Absolute Humidity of Air [g/kg] | - | 10.5 | 3.7 |
| Air Velocity [m/s] | 1, 1.5, 2 | 1, 1.5, 2 | 1.5 |
| Fluid | Water | Water | Brine |
| Flow Rate [liters/min] | 18-24 | 18-24 | 6 |
| Inlet Temperature of Fluid [°C] | 50 | 10 | -5 |

## ◆Experimental method

By changing the water flow rate, air side heat transfer coefficient $\alpha_o$ is measured by assuming the water side heat resistance.

$$\frac{1}{A_o K} = \frac{1}{A_o \alpha_o} + \frac{1}{A_i \alpha_i} \quad , A_o K = \frac{Q}{LMTD}$$

Overall heat resistance | Air side heat resistance | Water side heat resistance

## ◆Experimental result
## On dry surface condition

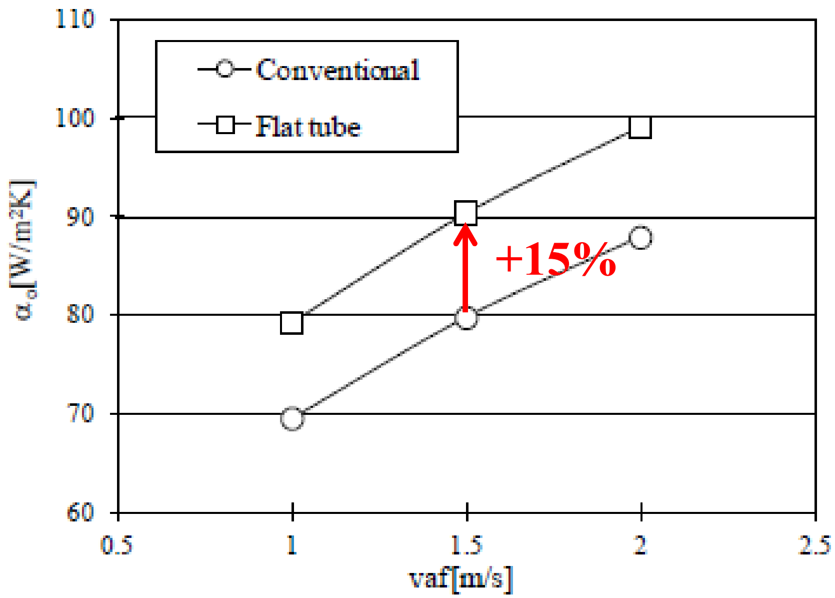



Fig. 6 Air Side Heat Transfer Coefficient $\alpha_o$
(On dry surface condition)

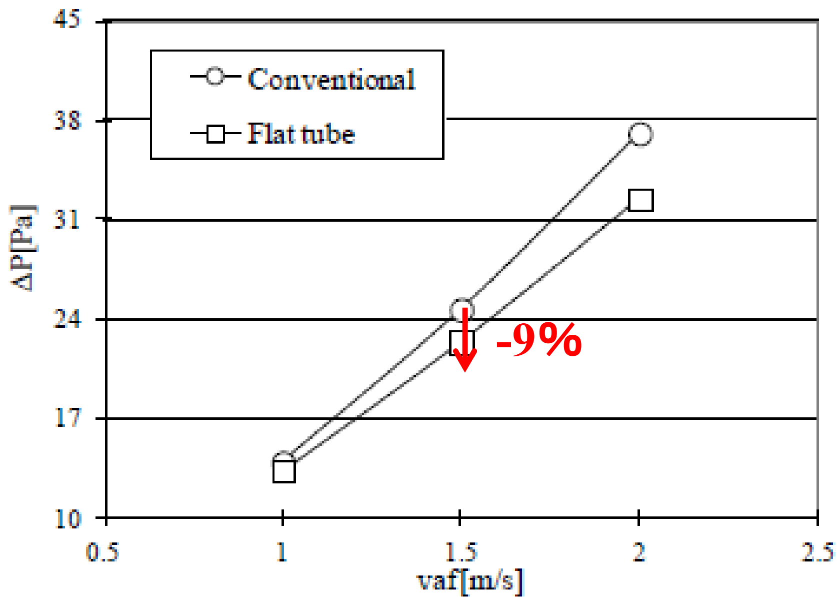



Fig. 7 Air Side Pressure Drop ΔP
(On dry surface condition)

## ◆Experimental result
## Increasing ratio of ⊿Pwet / ⊿Pdry

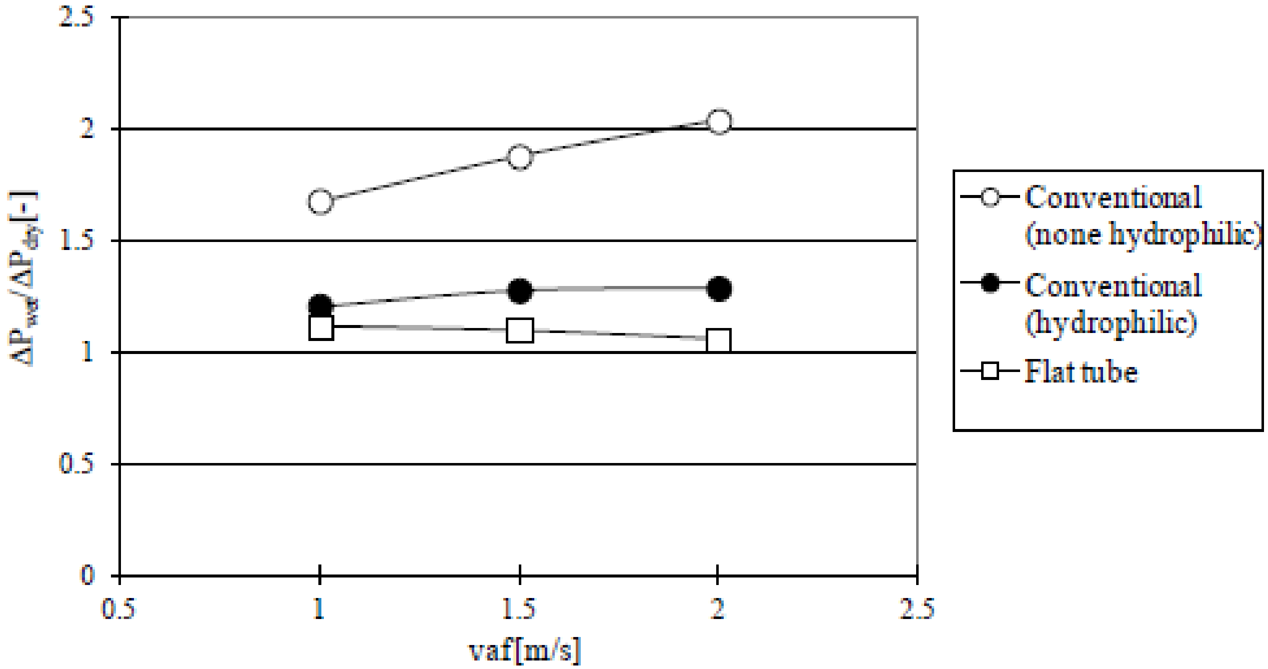



Fig.8 Increasing Ratio of $\Delta P_{wet}/\Delta P_{dry}$

# ◆Experimental result

## On frost condition

The frost weight on a time interval △M is given by equation (1).
The total amount of frost weight M is calculated by equation (2).

$$\Delta M = G_a \cdot (X_{a1} - X_{a2}) \cdot \Delta t \quad \cdots (1)$$

$$M = \sum_{i=1}^{n(P=150)} \Delta M \quad \cdots (2)$$

$G_a$ [kg/s]: Air mass flow rate
($X_{a1}$-$X_{a2}$) [kg/kg]: Absolute humidity difference between the inlet and the outlet.

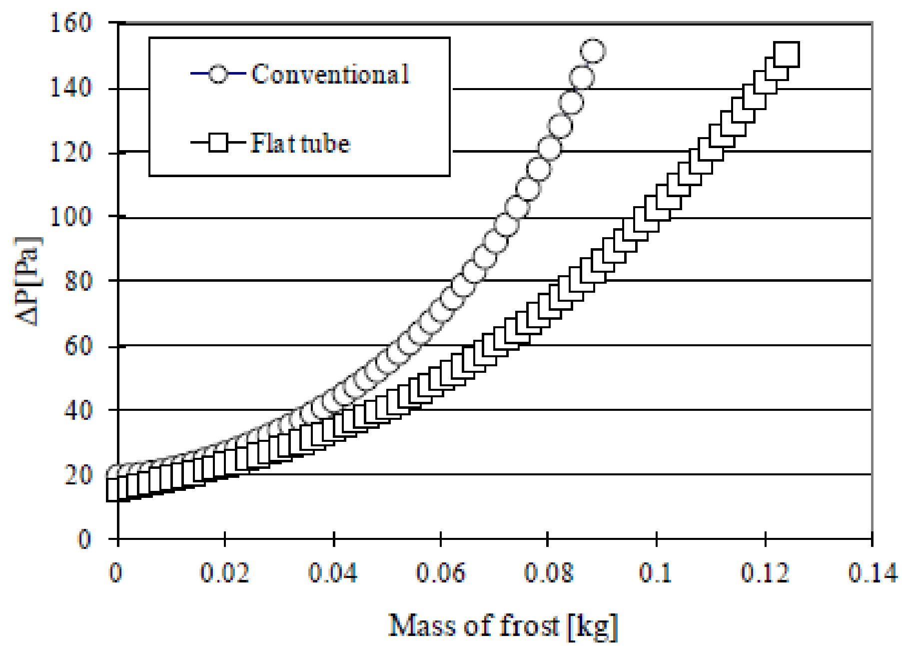



Fig.9 Air Side Pressure Drop ΔP
(On frost surface condition)

# Contents

# 4. Refrigerant Distribution Measurement



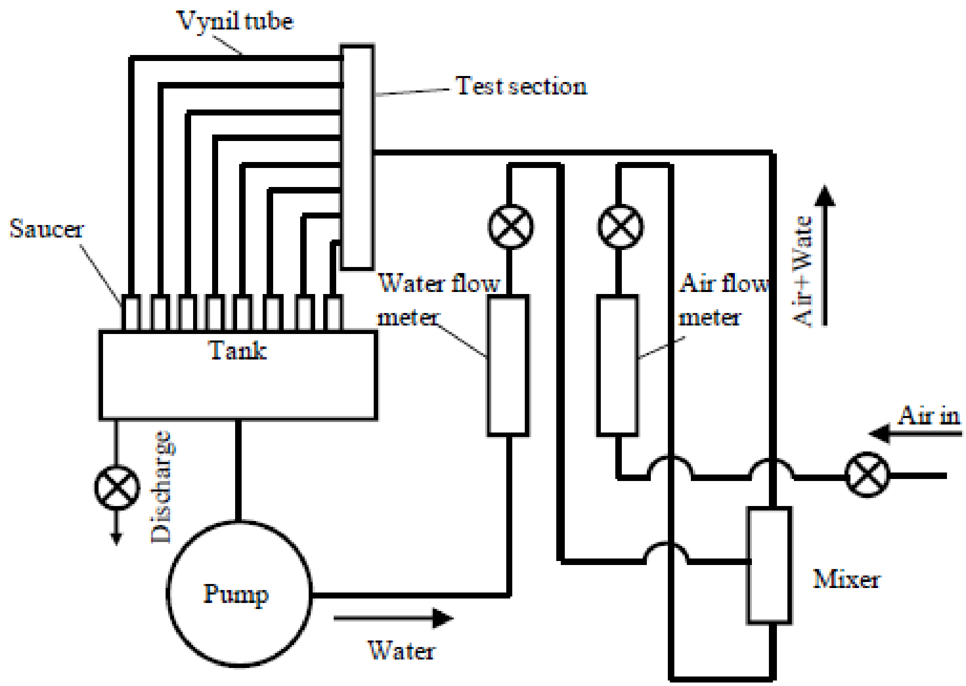



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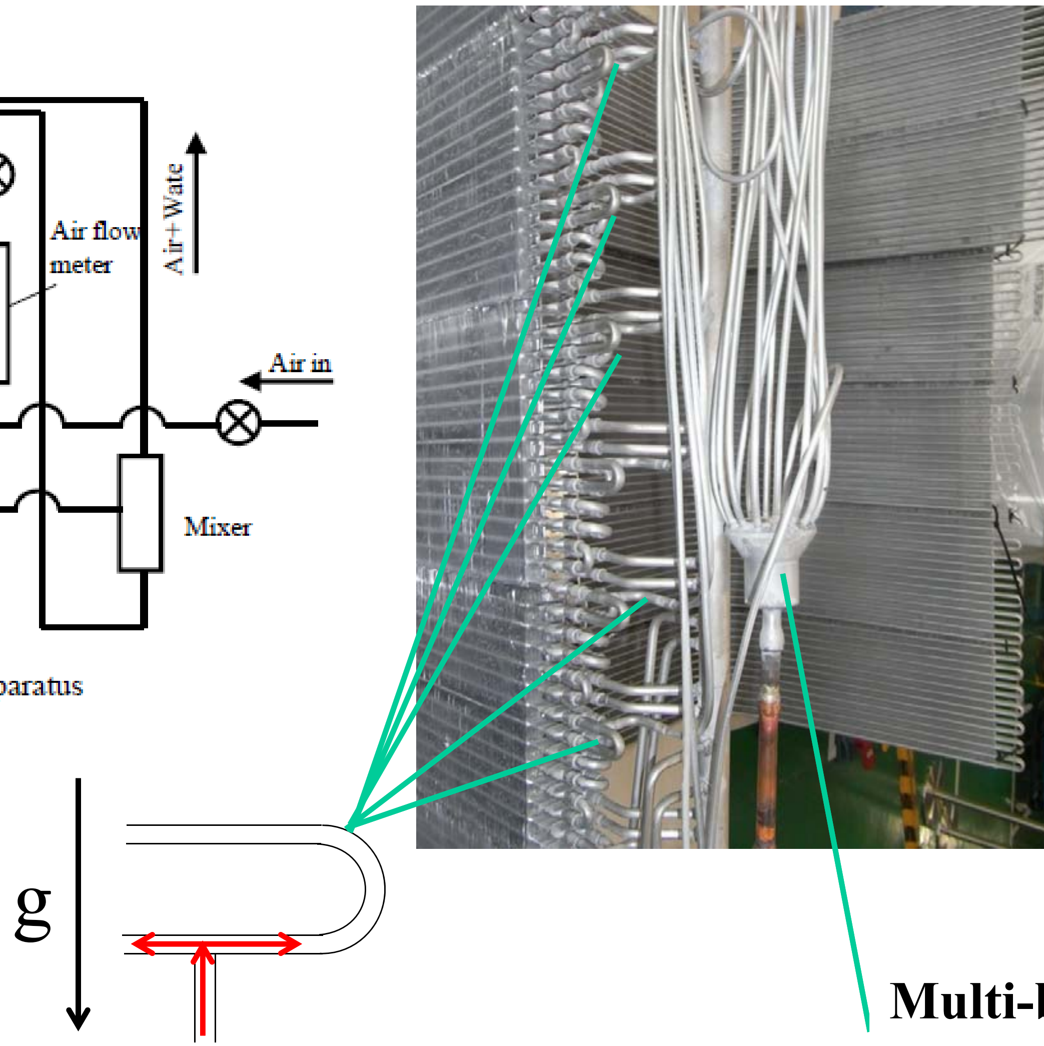
Fig.11 Experimental Apparatus




3-way pipe

Multi-branched distributer

## ◆ Experimental condition

The two-phase flow pattern of water-air was correlated to the flow pattern of R410A by the modified Baker chart (1983, Hashizume)

Table 3 Experimental conditions

| | R410A (Target) | | Water – Air | |
|---|---|---|---|---|
| | Mass flow rate: Gr [kg/h] | Quality: x [-] | Water flow rate [l/min] | Air flow rate [l/min] |
| Case 1 | 125 | 0.2 | 1.48 | 56 |
| Case 2 | 200 | 0.2 | 2.36 | 89.6 |

## ◆ Experimental result

**Standard divitation σ**

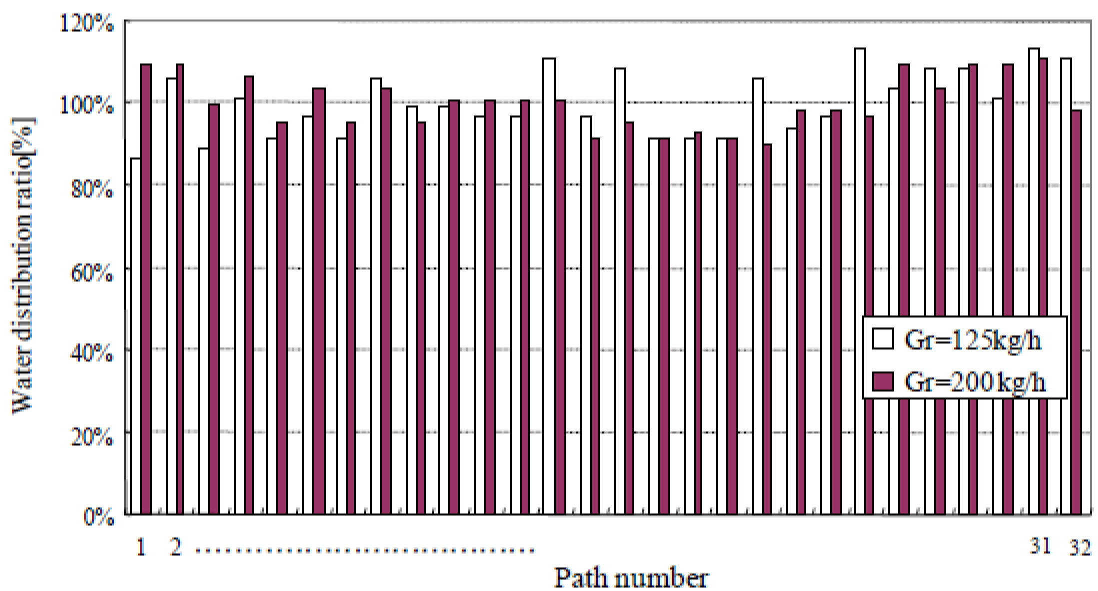



Fig.12 Refrigerant Distribution Characteristic

# Contents

# 5. Measurement of Overall Heat Transfer Performance AoK



Table 4 Specifications of Heat Exchanger

| | Conventional heat exchanger | Flat tube heat exchanger |
|---|---|---|
| Row number [-] | 2.5 | 2.33 |
| Height [mm] | 1224 | 1224 |
| Width [mm] | 1770 | 1770 |
| Path number [-] | 14 | 32 |

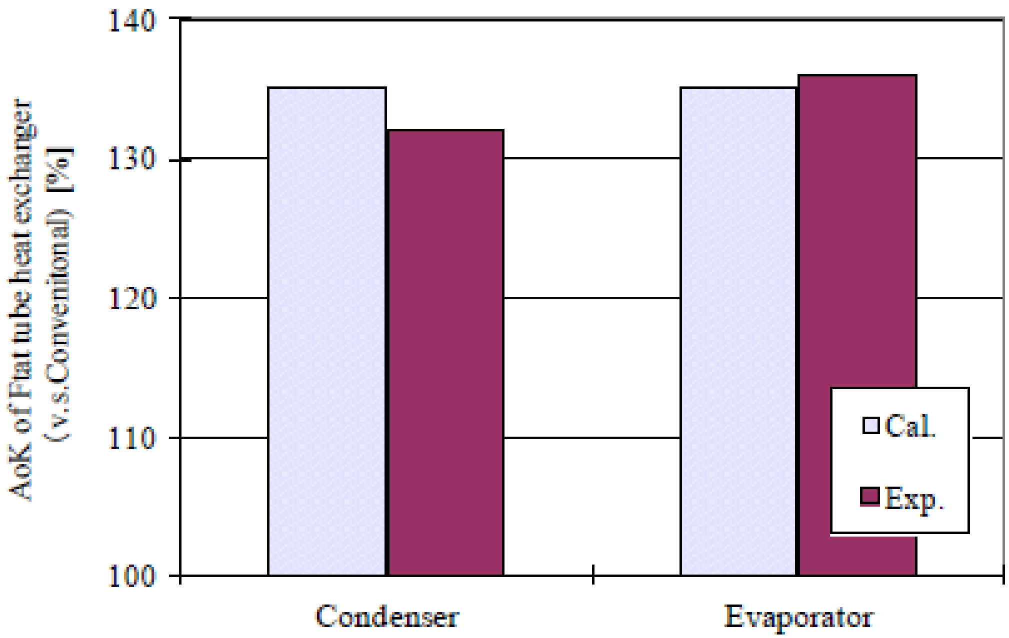
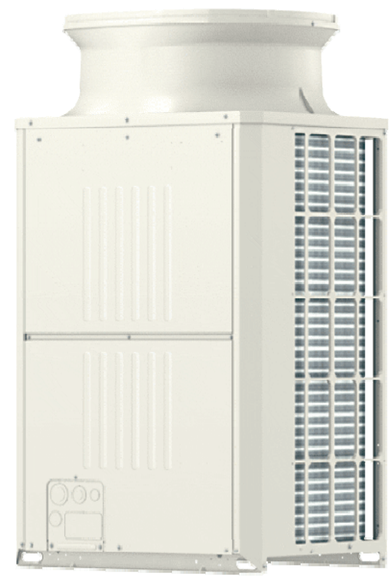



Fig.13 Overall Heat Transfer Performance $A_oK$



**10HP outdoor unit**

# 6. Conclusions



We developed a new type of an aluminum flat tube heat exchanger for the heat pump outdoor unit. Conclusions are as follows:

1) The air side heat transfer coefficient $\alpha_o$ of the flat tube heat exchanger was 15% larger and the $\Delta P$ of the flat tube heat exchanger was 9% smaller than those of the conventional heat exchanger for the dry condition.

2) The ratio $\Delta P_{wet}/\Delta P_{dry}$ of the flat pipe heat exchanger was smaller than the conventional heat exchanger with non-hydrophilic fins, and equivalent to the conventional heat exchanger with hydrophilic fins.

3) The air side pressure drop $\Delta P$ of the flat tube heat exchanger was smaller than that of the conventional heat exchanger for the frost condition.

4) The coefficient of variation of the refrigerant distribution for the flat tube heat exchanger was 7% and was not influenced by changing the mass flow rate.

5) The condensing $A_oK$ of the flat tube heat exchanger was 32% larger than that of the conventional heat exchanger. The evaporating $A_oK$ of the flat tube heat exchanger was 36% larger.