

DEVELOPMENT OF CROSS FLOW AIR-COOLED PLATE HEAT EXCHANGERS

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ABSTRACT

Experimental study on a cross-flow air-cooled plate heat exchanger (PHE) was performed. The two types of PHEs were manufactured with single-wave plates and double-wave plates in parallel. Cooling air flows through the PHEs in a crosswise direction against internal cooling water. The heat exchanger aims to substitute open-loop cooling towers with closed-loop water circulation, which guarantees cleanliness and compactness. In this study, prototype single-wave and double-wave PHEs were designed and tested in a laboratory scale experiments. From the tests, the double-wave PHE shows approximately 50% enhanced heat transfer performance compared to the single-wave PHE. However, the double-wave PHE costs 30% additional pressure drop. For the commercialization, a wide channel design for air flow would be essential for reliable performance.

NOMENCLATURE

A	amplitude of the sinusoidal corrugation
b	average width between the gaskets (m)
C_p	constant pressure specific heat (kJ/kg°C)
D_e	hydraulic diameter (m)
h	heat transfer coefficient (kW/m ² °C)
k	thermal conductivity (kW/m°C)
$\dot{m}$	mass flow rate (kg/s)
n_{pl}	number of plate
Nu	Nusselt number
P	pressure (Pa)
$\dot{Q}$	heat transfer rate (kW)
t	plate thickness (m)

T	temperature (°C)
ΔT_{lm}	logarithmic mean temp. difference (°C)
U	overall heat transfer coeff. (kW/m ² °C)
$\dot{V}$	volume flow rate (m ³ /hr)
w	plate horizontal width (m)
β	corrugation inclination angel (rad)
ϕ	area enlargement factor
μ	viscosity (Pa·s)
ρ	density (kg/m ³)

INTRODUCTION

The large capacity coolers have been required in various fields of heating, ventilating and air-conditioning (HVAC) industry. Among them, an open-loop cooling tower is popular as a vapor-to-liquid heat exchanger which discharges large capacity heat to ambient. Since the open-loop cooling tower is easy to install and relatively efficient, many engineers preferred to install ones for plant application. However, the cooling tower requires large installation space, and involves circulating water pollution easily by being exposed to air. The contamination of cooling water degrades heat transfer performance, reduces water flow rate, and also arises sanitary problem.

As one of the alternatives of the open-loop cooling towers, an air-cooled PHE is introduced in the current research. The installation of an air-cooled PHE will keep the circulating water clean since the PHE protect the deterioration problems which comes from direct contact of the water and ambient air. With scale-up of the PHE, it can be

utilized as various applications in HVAC field which will even replace vapor-to-liquid heat exchangers.

In this study, single-wave and double-wave cross-flow air-cooled PHEs of 30° chevron angle were tested. Each PHE was manufactured with the single-wave and double-wave plates of the same external dimension, but has different plate number. The objective of this study is verification of the PHEs' performance and to find possibility to commercialize.

INTRODUCTION

Cross-flow air-cooled type PHE: A general PHE consists of two couples of inlet and outlet ports where each couple of ports connects either side of heat transfer fluid. The air-cooled cross-flow PHE has only one couple of inlet and outlet ports where hot water passes through. The one side edge of each waved plates was alternatively welded to the next plate in order to seal water channel, but the other side edge left unwelded where cooling air passes in between. Fig. 1 illustrates the cross flow architecture of the air-cooled PHE where the water flows crosswise and air flows longitudinal.

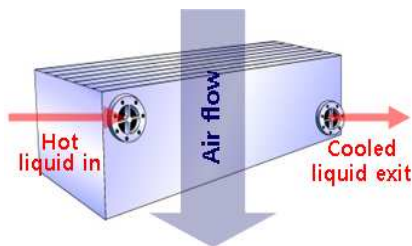
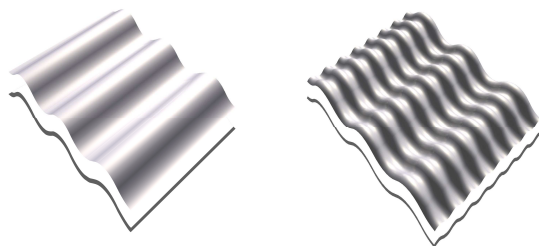


Figure 1 Flow direction of a cross-flow PHE



(a) Single-wave plate (b) Double-wave plate
Figure 2 Magnified images of the plates

Most of all heat resistance resides at the air side in case of air-to-water heat exchangers. Fig. 2 illustrates magnified images of the single-wave and double-wave plates. Double-wave plates increase air-side heat transfer performance by the addition of vertical corrugated waves on single-wave plates. The additional corrugated waves are normally pressed

perpendicular to single waves. As the plates were shaped with double-wave, the heat capacity would be increased by the enlarged heat transfer area. Heat transfer coefficients would also be increased by the eddy motion in the sinusoidal double-waved inflection points. Fig. 3 shows the effect that small and large scale eddy wave motion of air flow reacts jointly between double-wave plates.

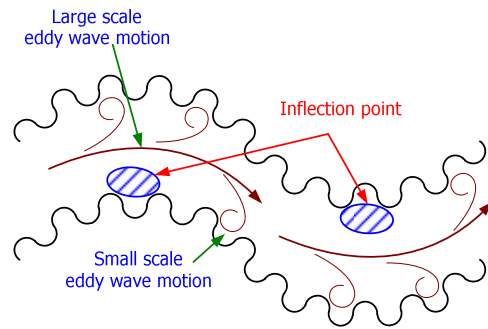
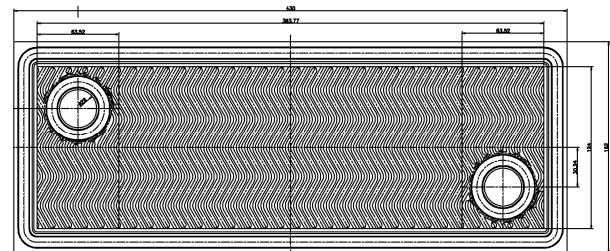
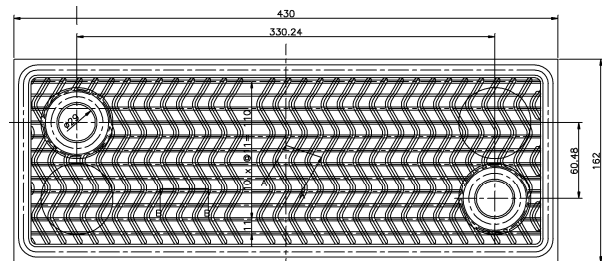


Figure 3 Motion of air flow in double-wave plates



(a) Single-wave plate



(b) Double-wave plate

Figure 4 Drawings of the plates

Fig. 4 is drawing of the plates of prototype PHEs. Each plate has the same projected area ($162 \times 430 \text{ mm}^2$). A single-wave plate is manufactured with 0.6 mm thickness of stainless-steel sheet. A double-wave plate was applied with 18% thinner Titanium plates of 0.5 mm thickness. Since double-wave shape is much more complicated than single-wave shape, it requires higher press force and deformation rate. Thus the plate thickness

was determined in order to avoid sheet fracture and provide deformability. The plate numbers of the single-wave and double-wave PHEs were 38 and 32, respectively.

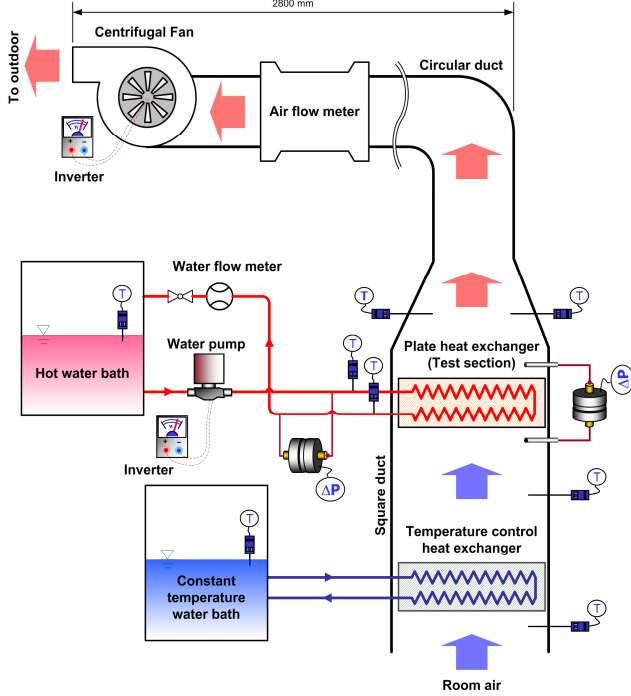


Figure 5 Schematic diagram of experimental setup

Experimental setup: Fig. 5 is a schematic diagram of experimental setup for the performance test of cross-flow air-cooled PHEs. Air flow ductwork was built in a series of a pre-heat exchanger, test section, an air-flow meter (0~7 LPM), and an exit centrifugal blower. To control inlet air temperature, water circulates from the constant temperature water bath to fin-tube type pre-heat exchanger. For the test section, hot water bath circuits that passes prototype air-cooled PHEs was set with circulation water pumps and water flow meter (0~6,000 m³/h). The exit air blower and circulation pump are controlled by inverters. RTDs (resistance temperature detector) and differential pressure transducer were installed at the inlet and outlet of water and air side, and the measurements were averaged to calculate the performance. Hot water is supplied from 200 liter bath with 5 kW heaters in order to set operating load to the test section.

Experimental conditions: Equations (1) and (2) present heat transfer rates for water and air. The measured volume flow rates were converted into mass flow rates using packaged property software

PROPATH Version 11.1 (1990). Water density was calculated from flow meter exit temperature which is PHE inlet temperature of water loop. Specific heat of water and air were calculated based on mean temperature of inlet and outlet. Because of air flow meter located at the rear of air flow path, the spot pressure is reduced by the pressure drop, where the pressure was compensated by the differential pressure of inlet and outlet at PHE.

$$\dot{Q}_w = \rho_w \big|_{T=T_{w,i}} \dot{V}_w c_{pw} (T_{w,i} - T_{w,o}) \quad (1)$$

$$\dot{Q}_a = \frac{P_{atm} \dot{V}_w}{RT_{a,o}} c_{pa} (T_{a,o} - T_{a,i}) \quad (2)$$

If the value of heat transfer rate calculated from Eqs. (1) and (2) were within 4%, the system was regarded to be steady and data were collected. Table 1 is the performance test condition.

Table 1 Performance test conditions

Parameters	Unit	Range
Air inlet temp., $T_{a,i}$	°C	20±1.0
Air side flow rate, $\dot{V}_a$	m ³ /hr	400 ~ 600
Water inlet temp., $T_{w,i}$	°C	40±0.5, 50±0.5
Water side flow rate, $\dot{V}_w$	m ³ /hr	0.8 ~ 1.1

RESULTS AND DISCUSSIONS

Data reduction: The heat transfer coefficient of air side was indirectly estimated by averaged thermal resistance of water. First, we determined overall heat transfer coefficient, U , from averaged heat transfer of air and water side. Then the heat transfer coefficient of air side was calculated by Eq. (3). Surface area, A , can be neglected since contact of air and water is the same. Eq. (4) shows logarithmic mean temperature difference for cross-flow heat exchangers.

$$\frac{1}{U} = \frac{1}{h_w} + \left(\frac{t}{k} \right)_{wall} + \frac{1}{h_a} \quad (3)$$

$$\frac{T_{a,o} - T_{a,i}}{\Delta T_{lm}} = \ln \left[\frac{\frac{T_{w,i} - T_{w,o}}{T_{a,o} - T_{a,i}}}{\frac{T_{w,i} - T_{w,o}}{T_{a,o} - T_{a,i}} + \ln \frac{T_{w,o} - T_{a,i}}{T_{w,i} - T_{a,i}}} \right] \quad (4)$$

Eq. (5) presents the equivalent diameter of a PHE, where b and w mean length and width, respectively.

$$D_e = \frac{4A_c}{P} = \lim_{W \rightarrow \infty} \frac{4bw}{2b + 2W} = 2b \quad (5)$$

P means the projected area of a plate. The distance between the plates is 5.2 mm in common. We assumed water flows equally along the longitudinal direction between the plate channels. When total plate numbers are n_{pl} , Reynolds number can be presented by,

$$Re_w = \frac{\rho_w \dot{V} D_e}{\mu_w} = \frac{D_e}{\mu_w A} \dot{m}_w = \frac{2}{\mu_w W} \frac{\dot{m}_w}{n_{pl}} \quad (6)$$

Generally, if the heat transfer coefficients of both fluid sides were unknown, Wilson plot technique (Shah, 1990) is recommended. However for solid Wilson plot approximation, it requires thermal resistance of both fluids to be similar like the cases of water-to-water or air-to-air combination. In this study, practical difficulties existed to employ the same phase fluids at PHE. Therefore, alternative analysis for air-to-water PHEs was carried out with the existing correlations.

The up-to-date single phase heat transfer correlations developed for PHE are mostly for a standard PHE with longitudinal flow direction. Also, the correlations have great difficulties in application since they accept limited geometrical information such as a chevron angle and the distance between plates at most. Actually, there are a number of the shapes and structures of a plate which affects flow patterns differently, so empirically measured heat transfer coefficients are varied depending on the shape of test PHE very much.

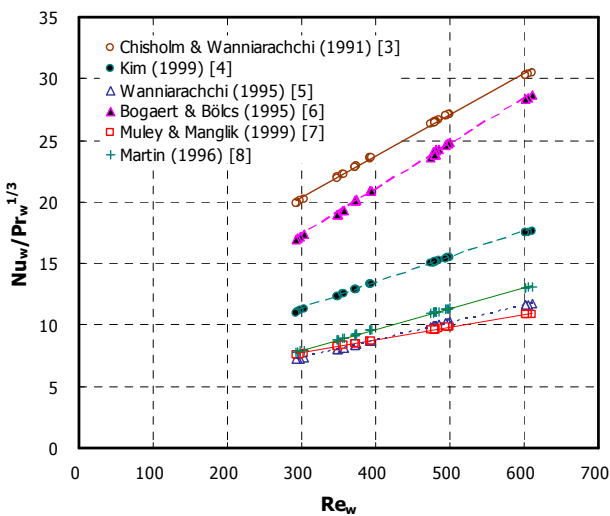


Figure 6 Comparison of water-side single phase heat transfer coefficient for a PHE with 30° chevron angle.

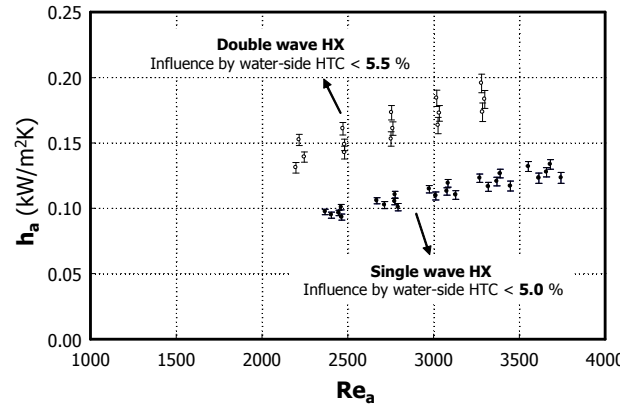


Figure 7 Uncertainty of air side heat transfer coefficients by the water side heat transfer correlation.

Fig. 6 showed the comparison of water side heat transfer coefficients with 6 different single phase correlations. Since the applicable flow patterns are from laminar to turbulent for the correlations, the deviation shows the maximum 300%, which is very large. Since the correlations of Chisholm and Wanniarachchi (1991) and Bogaert and Bölcs (1995) in Fig. 6 were built from the database of turbulent transition range, their correlations predict heat transfer coefficient higher than other correlations.

In Fig. 7, the air side heat transfer coefficients are shown. From Eq. (3) and six correlations in Fig. 6, the heat transfer coefficients are calculated. The upper and lower error bars present the deviation between the maximum and minimum values when using six correlations. From Fig. 7, even if the water side heat transfer coefficients of the six correlations deviate over 300% in Fig. 6, the air side heat transfer coefficients were deviates within the range of 5~5.5%. Air side heat transfer coefficient was affected less than water because thermal resistance of air side is much larger than the water side, which implies that the air side heat transfer coefficient can be estimated with less than 5.5% cost of uncertainty without applying Wilson plot technique. As the purpose of this study is to observe air side heat transfer performance for single-wave and double-wave plate shape, it is reasonable to estimate indirectly from existing water side correlation.

Most of the correlations for the PHE were expressed as a function of chevron angle, and it was rare other geometric parameters reflecting the characteristic shapes of the double-wave PHE. Eq. (7) defines the enlargement factor, ϕ , of a plate sheet which is in the ratio of the effective area to the projected area.

$$\phi = \frac{(\text{effective area})}{(\text{projected area})} \quad (7)$$

Through this, double-wave shape effect can be partially considered into an applicable correlation. Generally, ϕ of a single-wave plate is known between 1.1 and 1.3, depending on the chevron shape (Fernandes et al., 2005).

Martin's correlation (1996) was applied here to calculate water side heat transfer coefficient. This correlation covers the wide range of flow regime with ϕ in the formulation. In a single-wave plate, ϕ is defined by the ratio of the vertical surface length by projected length. In a double-wave plate, ϕ is calculated by the multiplication of the vertical surface length and horizontal surface length over its projected area. The ϕ of the double-wave and single-wave plates in this study is calculated as 1.37 and 1.17, respectively.

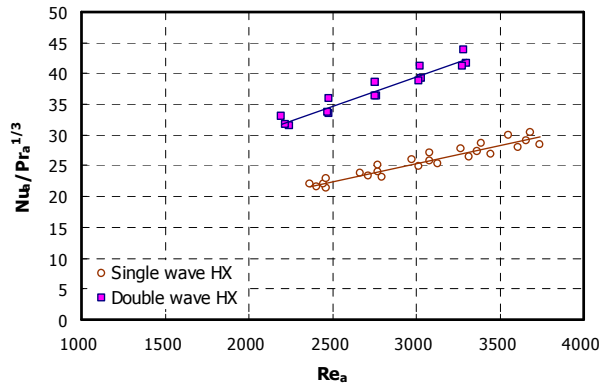


Figure 8 Comparison of Nusselt numbers for single-wave and double-wave PHEs

Test results and discussion: Fig. 8 shows Nusselt number with regard to air side Reynolds number. The Nusselt number for a double-wave PHE shows approximately 50% heat transfer enhancement than single-wave PHE in Fig. 8. Since the air flow in single-wave plates is moved along the single-waved chevron, the little flow mixing occurs. But for double-wave PHE, the eddy motion in air flow by the double-waved shape against flow direction as in Fig. 3 enhances the heat transfer performance.

Considering fin-tube heat exchanger, wavy-finned heat exchanger shows performance enhancement about 30% than flat-finned heat exchanger (Jung et al., 1997). But double-wave PHE showed much higher progression than single-wave PHE. This result does not present that double-wave PHE have superior performance than wavy-finned heat exchanger.

Reversely, single-wave PHE shows worse performance than flat-finned heat exchanger which may mislead into exaggeration of the performance enhancement of double-wave PHE. Since the tube arrangement of a flat-finned heat exchanger across the air flow enhances the air mixing at the back of the tube body, the flat-finned heat exchanger has better performance than single-wave PHE.

$$f = \frac{2\rho_a \Delta P_a D_h}{G_a^2 L} \quad (8)$$

Fig. 9 shows a Darcy-Weisbach friction factor in Eq. (8). Obviously, the friction factor of the double-wave PHE is bigger about 30% than the single-wave PHE, since flow resistance increases by additional corrugation by double-wave plate. Fig. 10 presents a Chilton and Colburn's j -factor by Eq. (9) for an air-cooled heat transfer performance. The j -factor is the value of the heat transfer coefficient indexed the heat transfer performance as dimensionless form.

$$j = \frac{\bar{h}_a}{G_a C_{p,a}} \text{Pr}^{2/3} \quad (9)$$

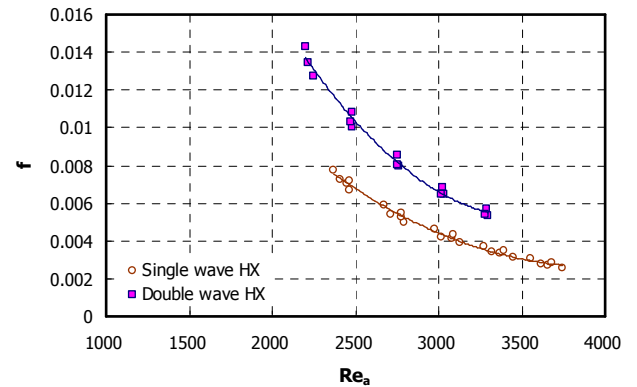


Figure 9 Comparison of friction factor

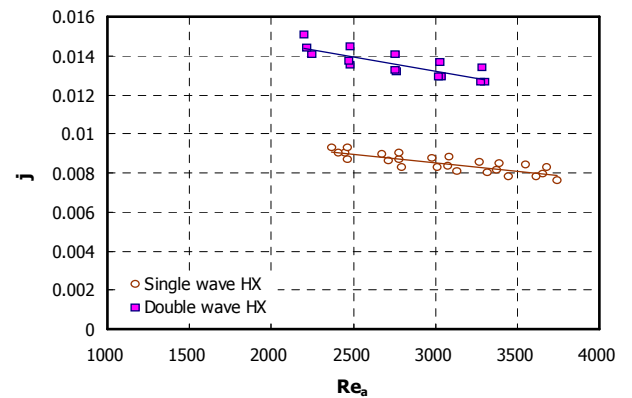


Figure 10 Comparison of j -factor

The performance of the double-wave PHE is greater about 50% in the heat transfer performance than the single-wave PHE from Fig. 10. As the increase of Reynolds number, the heat transfer coefficient increases in Fig. 8, but the j -factor decreases. It means that the increase of the air flow more affects the j -factor value than the increase of heat transfer coefficient. If the air flow increased, it means that total quantity of heat transfer increases, while the heat transfer effect of the unit air flow decreases. But the performance decrease rate of the double-wave PHE for the j -factor along increasing Reynolds number corresponding with the slope of the linear regression curve in Fig. 10 is about 15% bigger than that of the single-wave PHE. This means, if the air flow increased for reducing thermal resistance of air side, the heat transfer efficiency reduction of the double-wave PHE for unit air flow is relatively bigger than that of the single-wave PHE. Accordingly, corrugation across air flow is essential for the increased heat capacity when designing air-cooled PHE. Also, wider air flow path between plates would be considered at designing the plate shape of the double-wave plate for continuing the reduction rate of the j -factor.

According to above results, the double-wave plate is useful and good choice for the development of the air-side heat transfer in an air-cooled cross type PHE. And the advanced design of the double-wave plate which increases the air flow path between the plates as wide as possible is essential to guarantee the heat exchanger performance in the future.

CONCLUSION

In this study, we investigated the performance of an air-cooled cross type PHE to replace large capacity of open-loop cooling towers. This PHE can be designed more compact and provides cleanliness of cooling water loop. As a fundamental test, prototype air-cooled single-wave PHE and double-wave PHE were installed and tested in the ductwork in a laboratory scale. To calculate the performance of air side heat transfer, the water side heat transfer coefficient was chosen among the existing single-phase heat transfer correlation. The double-wave PHE showed the air-side heat transfer performance approximately 50% higher than the single-wave PHE, but requires 30% more pressure drop. For a field application of the air-cooled PHE, double-wave plate shape is desirable but need to be modified to keep wider gap between the plates for less pressure drop.

ACKNOWLEDGEMENTS

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