A Review of Fin-and-Tube Heat Exchangers in Air-Conditioning Applications

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ABSTRACT: This study presents a short overview of the researches in connection to the fin-and-tube heat exchangers with and without the influence of dehumidification. Contents of this review article include the data reduction method, performance data, updated correlations, and the influence of hydrophilic coating for various enhanced fin patterns. This study emphasizes on the experimental researches. Performance of both sensible cooling and dehumidifying conditions are reported in this review article.

Nomenclature

A	: surface area [m^2]
A_t	: external tube surface area [m^2]
A_o	: total surface area [m^2]
h_o	: air-side heat transfer coefficient [W/m^2/K]
$h_{c,o}$	: sensible air-side heat transfer coefficient under dehumidifying condition [W/m^2/K]
h_d	: mass transfer coefficient
ε	: $\dot{Q}/\dot{Q}_{\max}$, heat exchanger effectiveness, dimensionless
c_p	: specific heat [J/kg/K]
C_{min}	: minimum heat capacity rate [W/K]
Dc	: fin collar outside diameter = $D_o+2\delta_f$ [m]
Do	: tube outside diameter [m]
f	: Fanning friction factor, dimensionless
Fp	: fin pitch [m]
i_s	: saturated air enthalpy evaluated at fin temperature [J/kg]
$i_{s,fb}$	: saturated air enthalpy evaluated at fin base temperature [J/kg]
$i_{a,in}$	: inlet air enthalpy [J/kg]
$i_{a,out}$	: outlet air enthalpy [J/kg]
j	: $Nu/RePr^{1/3}$, the Colburn factor, dimensionless
k_f	: thermal conductivity of fin [W/m/K]
Le	: Lewis number, dimensionless
N	: Number of longitudinal tube rows
NTU	: number of transfer unit, = UA/C_{min}
Nu	: Nusselt number, dimensionless
ΔP	: pressure drop [Pa]
P_l	: longitudinal tube pitch [m]
Pr	: Prandtl number, dimensionless
P_t	: transverse tube pitch [m]
$\dot{Q}$	: heat transfer rate [W]
$\dot{Q}_{\max}$	: maximum heat transfer rate [W]
r	: radius of tube [m]
Re	: Reynolds number, dimensionless (usually based on D_c)
Re_{Dh}	: Reynolds number based on hydraulic diameter, dimensionless
R_{eq}	: equilibrium radius for circular fin [m]
δ_f	: fin thickness [m]
η_f	: fin efficiency, dimensionless
$\eta_{f,wet}$	: wet fin efficiency, dimensionless
η_o	: surface efficiency, dimensionless

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μ : dynamic viscosity of fluid [Pa · s]
ρ : mass density of fluid [kg/m^3]

1. Introduction

Plate finned tube heat exchangers are widely used in HVAC, process and power industry as coolers, heaters, evaporators and condensers. To improve the overall heat exchanger performance, fin surface enhancement is critical because the air-side thermal resistance is generally over 80% of the total thermal resistance, and may be as high as 95%. As a consequence an enhanced fin surface provides opportunity for the reduction in heat exchanger size, weight, material cost, and increase in energy efficiency. Fig. 1 depicted the common fin patterns for plate fin-and-tube heat exchangers. The fin patterns include plain, wavy, louver, slit, and convex-louver. In addition to the conventional enhanced surfaces, there are still many enhanced fin designs that are conceptually feasible. Wang[1] reviewed the US patents from 1980~1999. Most of the patents are conceptual design without commercialized implementation, yet the patents are associated with the interrupted fin whereas only a few are related to the vortex generators. Despite the existence of many various designs, its corresponding air-side performance data is usually proprietary. Not much data were reported in the open literature. This is especially true for performance data in dehumidifying conditions. Notice that it is essential for the engineers and designers to correctly use appropriate performance data in relevant applications. Though there are some test data regarding the air-side performances are reported. However, as shown by Wang et al.,[2] some of the test results may be inconsistent with each other or lack of important details of their reduction process. The absence of reliable air-side performance data cause additional difficulties to the designers and engineers. As a result, the objective of this study is to summarize recent efforts in connection to the air-side performances of the fin-and-tube heat exchangers in both dry and wet conditions.

2. Data reduction of test results

2.1 Data reduction for completely dry surface

Many experimental investigations to determine the air-side performance of fin-and-tube heat exchangers were reported in the literature. Unfortunately, most researchers did not clearly address the related details. Furthermore, many controversies still exist in the literatures. Wang et al.[2] postulated several inconsistencies existed in the published data. Deviations in the test results include (1) contact resistance, (2) reduction method, (3) the split of thermal resistances on the air and water sides, and (4) experimental uncertainties. Most of the investigators claimed a negligible contact resistance and acceptable uncertainties in their investigations. However, their basis for claiming small contact resistance is not justified, nor is the method of tube expansion described.

In summary, Wang, et al.[2] recommend the following procedures be checked in the practice of reduction of air-side performances.

(1) The energy balance in the experiments for the air- and tube side should be less than 5%. For better accuracy of the water side heat transfer rate, the temperature drop in the tube side should be higher than 2 ℃.

(2) Appropriate ε-NTU (ε is effectiveness and NTU represents number of transfer unit) relationship should be carefully examined before applying to the reduction of heat transfer coefficients. This is especially important for the 1-row configurations having larger NTU. The researcher should check the applicability of the ε-NTU relationship to their circuitry design. Table 5 shows the typical ε-NTU relationship cross-flow arrangement having N = 1~4.

(3) For fin-and-tube heat exchangers having

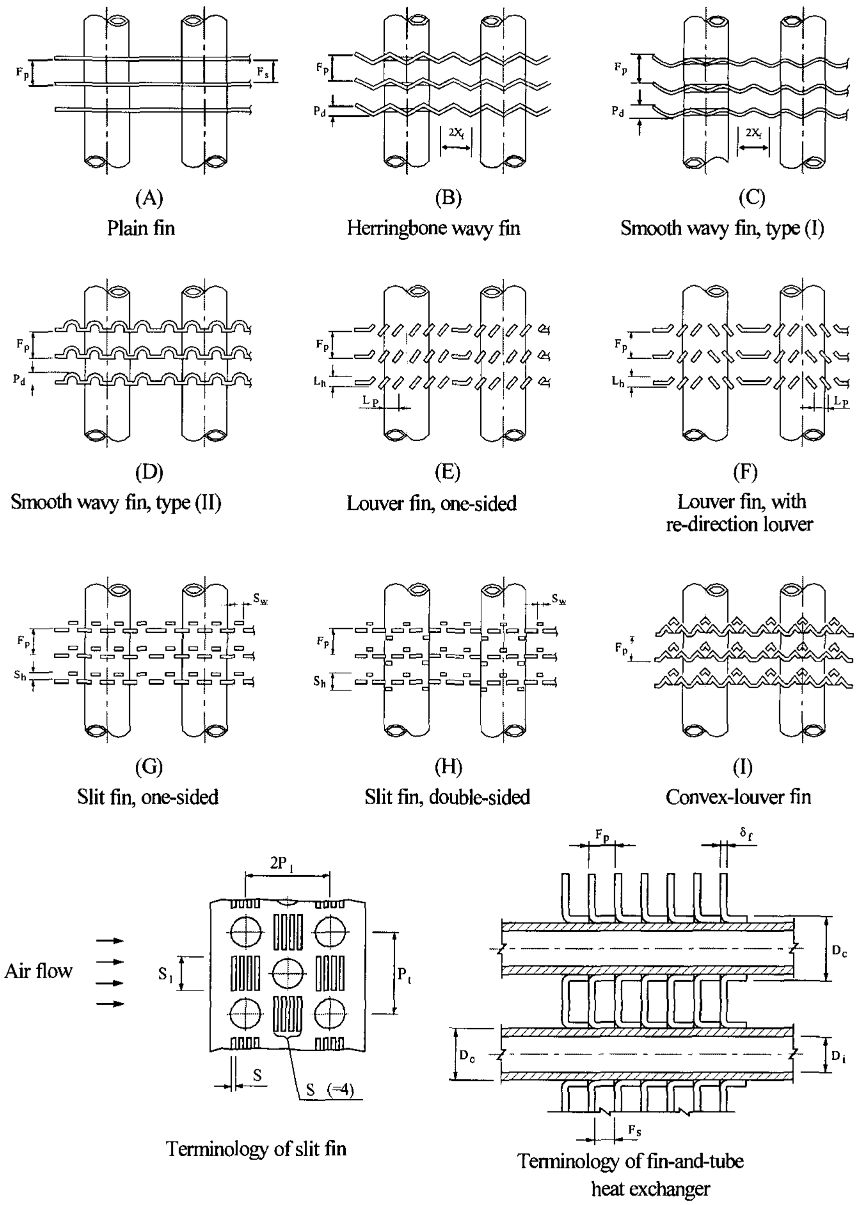

Fig. 1 Schematic of various fin patterns and terminology of fin-and-tube heat exchanger.

round tube configuration, it is suggested that the entrance and exit loss be included in the friction factor, which negates the need for calculating entrance and exit losses and would result in nearly independent of the number of tube rows for the friction factors.

In the U.S., the air-side performance is typically reported as h_o (surface efficiency being removed, h_o denotes the heat transfer coefficient). However, in some Asian countries such as Japan or Korea, it is rather common for researchers to report the $\eta_o h_o$ product as the

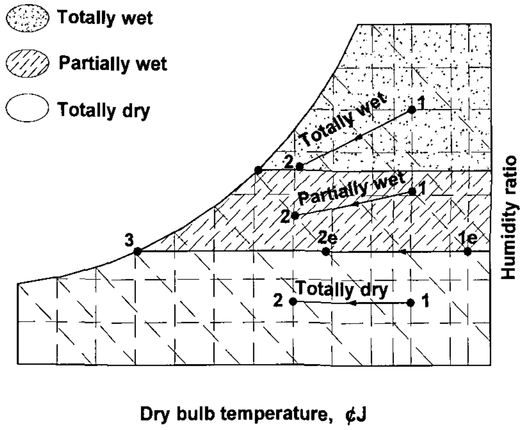

Fig. 2 Type of cooling process for a typical fin-and-tube heat exchanger.

"air-side heat transfer coefficient". Such examples of this were provided by Nakayama and Xu[3] and Kang and Kim.[4] Therefore, one should be careful about the terminologies used by different investigators before applying their developed correlations. For the correlations developed by the present author, the heat transfer coefficients reported are for h_o only. The surface efficiency, η_o, is related to the fin surface area, total surface area, and fin efficiency as $\eta_o = 1-A_f/A_o(1-\eta_f)$.

For calculation of fin efficiency, η_f, two methods are recommended for consideration. The most accurate one is the "sector method," as described by Rich,[5] or an adoption of the circular fin efficiency equation. Schmidt[6] presented an easier way to approximate the fin efficiency. Hong and Webb[7] have shown that this equation will yield errors greater than 5% if $R_{eq}/r > 3$ and $m(R_{eq}-r) > 2.0$ (Where R_{eq} is equilibrium radius for circular fin, r is radius of tube, including collar fin thickness, and $m=\sqrt{\frac{2h_o}{k_f g_f}}$, k_f : thermal conductivity of fin, δ_f : fin thickness). However, most practical applications are within the range of the above limits. Therefore, for faster and easier manipulation, the Schmidt[6] approximation to the fin efficiency is recommended.

In practice, test results of the air-side performance are usually in terms of plots of dimensionless Coburn j factors and Fanning friction factors f vs. the Reynolds number. A great diversity of characteristics length was used in the definition of the Reynolds number in open literature. The use of either tube collar diameter or hydraulic diameter as the characteristics length is arbitrary as discussed by Webb.[8] However, use of D_c is much more appropriate to correlate the experimental data. As a result, most of the successful correlations of fin-and-tube heat exchangers were based on fixed dimensions like tube diameter. Thus, it is recommended to use fixed dimension as the characteristics length in the definition of the Reynolds number.

2.2 Data reduction for completely wet surface

For wet surface data reduction, Hong and Webb[8] pointed out there are two different driving potentials used to define the wet sensible heat transfer coefficient. These are specific humidity and enthalpy. For dehumidifying conditions, the heat transport consists of sensible and latent heat (Threlkeld[10]). In the approach of specific humidity, the reduction process is basically the same as those of dry coils. Detailed description of this reduction method can be also found in Myer[11] and Wang et al.[12]

In deriving the fully wet fin efficiency by specific humidity driving potential, many analytical approaches were performed to derive the wet fin efficiency. However, some of the analytical approaches may be inappropriate. Lin et al.[13] commented on many of the controversies in their formulations. Among the formulations, Hong and Webb[9] recommended fin efficiency formulation by Wu and Bong[13] be adopted. The calculated results by Wu and Bong[13] indicated that the difference of wet fin efficiency formulated by the specific humidity driving potential and enthalpy potential (Threlkeld[10]) is negligible. Note that the formulation of the wet fin efficiency by enthalpy potential is $\eta_{f,wet} = (i-i_{s,fin})/(i-i_{s,fb})$ (Where i denotes enthalpy, subscripts fin and fb represent evaluations are made at fin and base surface). Despite the enthalpy-based reduction method by Threlkeld is well accepted, the method is actually a lumped based method and may suffer from the influence of local humidity. This is especially severe because part of the fin-and-tube heat exchanger may be in dry condition. Hence a discrete tube-by-tube or tiny element approach were recently developed by Pirompugd et al.[14,15] to overcome the influence of partially dry condition. An alternative reduction method termed as equilibrium dry-bulb temperature method (EDT) capable of handling both partially wet and fully wet surface was proposed by Wang and Hihara.[16] In this method, an equivalent dry process, as depicted in the partially wet process of Fig. 2 characterizes an identical cooling capacity of the actual process is used in the analysis. By doing this, the heat transfer characteristics of the coil were analyzed based on the temperature difference of inlet temperature and tube surface temperature. Huzayyin et al.[17] claimed that the EDT method is slightly better than that of enthalpy based method. However, the judgment is based on comparison of some test data of their own and without any detailed physical insight to differentiate these two methods. Hence, further investigations must be made and, for the time being, the enthalpy based method is still recommended for its wide-spread acceptance. Xia and Jacobi[18] also compared the data reduction methods of logarithmic-mean temperature difference (LMTD) and logarithmic-mean enthalpy difference (LMED) methods for wet-a nd frosted-surface heat exchangers. For all conditions considered, they argued that the LMED method is based on Le~1 which cause an error as large as 8% in heat transfer rate. Based on their data, they claimed LMTD method may be more accurate than LMED method.

3. Performance of Continuous fin geometry

3.1 Performance in dry condition-plain fin geometry

There are many enhanced surfaces available. However, the continuous fin geometry is still by far the most popular design owing to its long-term superiority and reliability. The continuous fin geometry is the first generation of fin-and-tube heat exchangers. The most commonly used continuous fin geometry takes the form of plain and wavy. For airside performance in fully dry conditions, the most informative studies were those carried out by Rich,[19,20] who investigated a total of fourteen coils, in which the tube size was 13.34 mm, and the longitudinal and transverse tube pitches were 27.5 and 31.75 mm, respectively. He concluded that the heat transfer coefficient was essentially independent of the fin spacing and the pressure drop per row is independent of the number of tube rows. McQuiston[21] also presented test results for five heat exchangers (F_p = 1.81~6.35 mm, D_o = 9.96 mm, P_l = 22 mm, P_t = 25.4 mm, and N = 4). McQuiston[22] then proposed the first well known correlation by employing a "finning factor," defined as A_o/A_t, to correlate his data along with those by Rich.[18,19] His correlation shows a strong dependence of heat transfer performance with the finning factor; i.e. $j \sim (A_o/A_t)^{-0.15}$. For the friction factor correlation, McQuistion[22] claimed the accuracy is about ±35%.

Gray and Webb[23] proposed an updated correlation of the plain fin geometry from five investigators. Note that the database by Gray and Webb was for $N \geq 4$ and the database was filtered in advance to prevent "overweighting" for certain data sources. The root-mean-square error of the resulting correlation was 7.3% for heat transfer coefficients, and 7.8% for friction factors.

The Gray and Webb[23] correlation gave reasonably predictive ability for those having larger tube diameter, larger longitudinal tube pitch, transverse tube pitch, and multiple number of tube row. A significant improvement of the Gray and Webb[23] correlation over the McQuiston[22] correlation is its superior predictive ability of friction factors. One of the important results concluded from the McQuiston correlation and the Gray and Webb correlation is that the heat transfer performance is relatively independent of fin spacing. The result is in fact based on the test results of the 4-row coils that reveal negligible effect of fin spacing (Rich[19]; Wang et al.[24]). However, some investigations show a detectable effect of fin spacing on the heat transfer performance for $N = 1$ and $N = 2$ (e.g. Seshimo and Fujii,[25] Wang and Chang,[26] and Wang and Chi[27]). Therefore, updated correlation was developed by Wang et al.[28] that can take into account the effect of fin spacing more accurately. The correlation is based on test results of consistent reduction process from seven data sources. A total of 74 samples were used to develop the correlation. Apart from its considerable amount of samples of database, one should notice that the database did not contain test results of larger diameter tube. In typical applications like fan-coil or ventilator, use of larger diameter like 15.88 mm is also very common. Hence, a very recent study by Liu et al.[29] had included additional test results from nine new test samples and developed an modified correlation from original Wang et al's correlation.

3.2 Performance in wet conditions - Plain Fin geometry

For test results of the plain fin in fully wet conditions, the most influential studies were those carried out by McQuiston.[21,22] He proposed separate correlations for film type and drop type condensation. In spite of its popularity, it is questionable to extrapolate the correlation outside their database. In addition, their correlation shows no influence of the inlet relative humidity. Also based on test results from nine samples but only one plain fin pattern (P_t = 25.4 mm and P_l = 22 mm, D_c = 10.2 mm, N = 2, 4, and 6, F_p = 1.7~3.1 mm), Wang et al.[12] proposed a correlation to describe their test results in the range of $300 < Re_{Dc} < 5500$. Wang et al.[12] claimed their correlation can describe 92% of the j factors within 10% and 91% of the friction factors were correlated within 10%. Again, their correlation shows no effect of the inlet relative humidity on the frictional performance. This is because their fin pitch is quite large ($F_p > 1.7$ mm) and the condensate retention phenomenon is rather small.

Wang et al.[30] showed that the influence of the inlet relative humidity may become pronounced as the fin pitch is further reduced (eg. $F_p < 1.7$ mm). This is because the effect of condensate retention is rather pronounced. As a consequence, the frictional performance may be influenced by the presence of water condensate. Thus Wang et al.[31] implicitly included the effect of relative humidity by introducing the condensate Reynolds number $Re_{film} = 2\Gamma/\mu_{water}$. However, their correlations may not be easy to employ because the involvement of condensate Reynolds number ($2\Gamma/\mu$). One has to guess an outlet state (outlet humidity) by employing heat and mass transfer analogy first for estimation of the heat transfer coefficient or friction factor, and then check iteratively about the assumption. In addition, the previous correlations are applicable in fully wet conditions, extrapolation of the correlations to the partially wet region is not recommended. Note that the partially wet conditions are frequently encountered when the airflow velocity is high or the inlet relative humidity is low. Furthermore, in practice, it is easier to have the "completely dry" data rather than "completely wet" data. Therefore Wang et al.[32] developed the equations that are capable of handling the "completely wet" and "partially wet" conditions by use of "completely dry" data.

3.3 Performance in dry condition-Herringbone Wavy Fin geometry

The first comprehensive study related to the herringbone wavy fin pattern was done by Beecher and Fagan.[33] They presented test results for twenty-one herringbone fin-and-tube heat exchangers having $N = 3$. Data were presented in terms of Nusselt number, Nu_a, based on the arithmetic mean temperature difference (AMTD) vs. Graetz number. However, the wavy fin geometry tested by Beecher and Fagan[33] were rather uncommon when compared to practical design. Their fins were electrically heated, and thermocouples were embedded in the plates to determine the plate surface temperature. The power to the several electric heaters was adjusted to maintain a constant temperature over the air-flow length. This simulated a fin-and-tube heat exchanger having 100% fin efficiency and zero contact resistance between the tube and fin. Webb[34] recasts the investigation of Beecher and Fagan,[33] and developed a correlation. Based on test results by Beecher and Fagan[33] and Wang et al.,[35] Kim et al.[36] also proposed a correlation.

However, it should be pointed out that either the Webb correlation[34] or the Kim et al.'s correlation[36] may over-predict the experimental data. Part of the explanations to their over-prediction may be attributed to the data-source of Beecher and Fagan.[33] Some possible explanations of this result can be seen from Wang.[37] The first one is due the presence of contact resistance inherited in the actual fin-and-tube heat exchangers that apparently did not appear in the Beecher and Fagan's[33] data. Depending on the manufacturers, the contact resistance usually comprises 2~7% of the total resistance. Secondly, the published data usually absorbed the contact resistance into the air-side resistance since it is very difficult to differentiate it from the air-side resistance. In addition, their constant temperature approach eliminates the presence of fin efficiency which also exists in the actual reduction process of the heat transfer performance of the fin-and-tube heat exchangers. The third possible reason may be related to their reduction process. Beecher and Fagan[33] used AMTD method instead of conventional LMTD or ε-NTU approach, they also derived a relation in connection to the LMTD approach. However, it is not fully aware about the derivation of this equation. Meanwhile, Beecher and Fagan[33] used Graetz number to present their test results and argued their approach may be more appropriate than the conventional Reynolds number presentation. However, Webb[34] pointed out that the presentation by Graetz number is no better than those using Reynolds number. Apart from the forgoing explanations, there is another problem arising from the database of Wang et al.[35] Note that the original circuitry of Wang et al.'s sample is cross-counter flow arrangement. However, they used typical cross- flow ε-NTU relationship for data reduction, leading to an additional uncertainty.

A series of investigations to the herringbone wavy fin patterns based on commercially available samples were conducted by Wang et al.[35, 38 42] Effects of fin spacing, the number of tube row, wave height, and edge corrugation were systematically examined. Wang et al.[42] developed a correlation from 61 samples.

3.4 Performance in wet conditions-Herringbone Wavy Fin geometry

Two wavy fin patterns were commonly used in commercial applications, namely the herringbone wavy and smooth wavy fin pattern (Fig. 1B~Fig. 1C). For herringbone wavy fin pattern, Wang et al.[43,44] had investigated the influence of wavy height, the number of tube row, and fin pitch in wet condition. They found that the effect of wavy height on the frictional performance is much more pronounced than that in fully dry conditions. However, an unexpected dependence of the frictional factor with the number of tube row is observed. Based on the flow visualization results of a scale up wavy channel in dehumidifying condition by Yoshii et al.,[45] Wang et al.[43] argued this phenomenon is associated with the interaction of the airflow pattern and the condensate. Analogously, based on test results from 24 samples, Wang et al.[32] developed a correlation for herringbone wavy fin geometry.

3.5 Performance in dry condition - Smooth wavy fin

As shown in Figs. 1B-1D, basically, there are two variants in the wavy fin patterns (namely the herringbone and smooth wavy fin patterns). As pointed out by Wang et al.,[39] for smaller fin spacing like 1.6 mm, the herringbone wavy fin pattern would produce significantly increase of the pressure drops when comparing to that of the plain fin while only reveals 5~10% increase of heat transfer performance when the corrugation angle is less than 20°. Accordingly this is related to the protruding edges of herringbone fin geometry. Sparrow and Hossfeld[46] had examined the effect of rounding protruding edges of a wavy channel. They reported that the rounding of the protruding edges would result in a significant decrease of friction factor while show only small loss in the heat transfer performance. Thus, smooth wavy fin may be more beneficial if pressure drops are the major concerns. Unfortunately, researches of this fin pattern are extremely rate. Based on the test results of two smooth wavy fin patterns (Figs. 1C and 1D) from five samples, Mirth and Ramadhyani[47] proposed a correlation applicable for four and eight row coils. A very recent work by Youn and Kim[48] had tested twenty-nine samples having different waffle heights (1.5 and 2.0 mm), fin pitches (1.3 - 1.7 mm) and tube rows (1 - 3) with sinusoidal wave fins. One of the special features of this fin is its much higher pressure drop than the herringbone fin pattern. It is not quite understood about this feature.

For type II smooth wavy fin (Fig. 1D), the friction factor correlation gives an unexpected strong length dependency. Similar results are also encountered for herringbone wavy fin patterns tested in dehumidifying conditions by Wang et al.[43] However, it is not clear about this phenomenon for the type II smooth wavy fin.

3.6 Performance in wet conditions - Smooth wavy fin

The only investigation related to this subject was done by Mirth and Ramadhyani.[47,49] They developed a heat transfer correlation for their test results in fully dry conditions based on five samples of heat exchangers. However, Mirth and Ramadhani[47] failed to develop the correlation in wet conditions because of considerable scattering of their test results. In contrast to the heat transfer correlation, they proposed a frictional correlation in wet conditions related to that in dry condition. However, the frictional performance of type II smooth fin shown implies a dependence of the friction factors with the number of tube row (since L_{coil} is proportional to the number of tube row). However, type I smooth wavy fin shows no dependence of the fin friction factor with the number of tube row. Note that type II smooth wavy fin is a combination of smooth wavy (Fig. 1C) and plain fin pattern which resembles the previous herringbone fin pattern to some extent. Accordingly, a dependence of the number of tube row is seen.

4. Interrupted Fin Geometry

4.1 Performance in dry condition - Louver fin geometry

The objective of the interrupted surface is the exploitation of boundary layer renewal, self-sustained flow unsteadiness, and thermal wake arrangement (Jacobi and Shah[50]). The most common interrupted surfaces are offset strip and louver fin. The louvered fin pattern is more beneficial when produced in large quantities since it can be manufactured by high-speed production techniques.

Systematic information related to the louver fin was reported by Chang et al.,[51] Chi et al.,[52] Wang et al.[53 55] An interesting feature of the louver fin geometry is, when compared to plain or wavy fin geometry, the effect of fin pitch and tube row on the air-side performance is comparatively small. Based on the test results of Wang and his co-workers, Wang et al.[56] developed a louver fin correlation for six different kinds of louver fin geometry.

4.2 Performance in wet condition - Louver fin geometry

Despite the popularity of louver fin geometry, very few data were concerning the airside performance in wet conditions, the only reported data are by Wang and Chang,[25] Hong and Webb,[9] and Wang et al.[57] Wang et al.[32] had proposed a correlation from 17 samples.

4.3 Fully dry results - Slit fin geometry

The first paper related to the slit fin geometry was made by Nakayama and Xu.[3] They presented test results for three samples, and proposed a correlation based on their test results. Notice that the air-side heat transfer correlation developed by Nakayama and Xu[3] had already contained surface efficiency η_o, users should be very careful about this.

Applicability of the correlation by Nakayama and Xu[3] as pointed out by Garimella et. al.[58] is very limited. Extrapolation of the correlations is inadvisable due to very strong dependence of the j-factor on the ratio of fin thickness to fin spacing.

An updated correlation for the slit fin geometry which covers the test results of one-sided slit (Fig. 1G) by Wang et al.,[59] Du and Wang,[60] and Nakayama and Xu[3] and new test results for two-sided slit (Fig. 1G) was developed by Wang et al.[61]

4.4 Performance in wet condition - Slit fin geometry

Similarly, test results for slit fin geometry in wet condition is very rare, the only test results were by Wang et al.[30] and Wang et al.[62]. They also provided correlations for their test results, however, as discussed previously, their correlation incorporated the condensate film Reynolds number that makes the correlations difficult to use. Therefore Wang et al.[32] re-correlated the data from 34 samples, and developed the related correlation.

5. Vortex generator

Despite of the fact that interrupted surfaces can significantly improve the heat transfer performance, the associated penalty of the pressure drop is also tremendous. Thus, in contrast to the commonly interrupted surfaces, the third generation of enhanced surfaces is now receiving a lot of attentions. The third generation of the enhanced surfaces employing longitudinal vortex generator that provides swirling motion to the flow filed (see Fig. 1F of US patent 4817709[63] that superposed a delta wing on conventional wavy fin surface). Usually the swirling motions may be classified as transverse and longitudinal vortices. The rotation axis of a transverse vortex lies perpendicular to the flow direction while the longitudinal vortices have their axes parallel to the main flow direction. The longitudinal vortex flow may swirl around the primary flow and exhibits three-dimensional characteristics. In general, longitudinal vortices are more effective than transverse vortices from the heat transfer perspective (Fiebig[64]). The imposed vortical motions provide enhanced heat transfer performance with relatively low penalty of pressure drop. This is because wall friction is related to the derivative of streamwise velocity but spanwise and normal velocities. The vortex generators characterized the secondary flow pattern from the vortical motions which are caused by the spanwise and normal velocities. Thus, heat transfer enhancement is associated with the secondary flow but with small penalty of wall friction (Jacobi and Shah[65]).

The first literature reporting the heat transfer performance improvements is by Edwards and Alker.[66] They reported the local average heat transfer coefficient was about 40% higher than that of a plain fin surface. For applications to the compact heat exchangers with vortex generators, Fiebig and his coworkers[67 71] had conducted very comprehensive studies in association with the influence of wing type, delta-wing type, delta winglet, and rectangular winglet vortex generators. Very detailed information was reported about these kinds of vortex generators. Fiebig[69] then concluded that the delta winglet shows the best performance. Similar results was also reported by the numerical simulations by Biswas et al.,[72] they showed that the flow loss of the winglet-pair is less than that of the wing type. Many investigations were performed to examine the thermofluids characteristics in compact heat exchanger surface. However, until now, there are no reported data for the actual fin-and-tube heat exchangers yet. This is because the cost of fin die is very expensive. Wang et al.[73,74] had reported flow visualization of the enlarged fin-and-tube heat exchangers having inline and staggered arrangements. The flow visualization results show significant improvements of mixing characteristics by the presence of longitudinal vortex. The corresponding frictional penalty of the annular and delta winglet vortex generators is about 10%~65% higher than that of the plain fin geometry. The good mixing flow characteristics of the vortex generator with only moderately increase of frictional penalty shows promising aspect of the vortex generator. It is highly recommended that future works should be focused on the implementation of the advanced vortex generators. For implementation of vortex generator in an oval finned tube heat exchanger, O'brien et al.[75] found that the corresponding heat transfer results with the single winglet pair to the oval-tube geometry yielded significant heat transfer enhancement, averaging 38% higher than the oval-tube, no-winglet case. The corresponding increase in friction factor associated with the addition of the single winglet pair to the oval-tube geometry was very modest, less than 10% at Re_{Dh} = 5500 and less than 5% at Re_{Dh} = 55000.

6. Influence of hydrophilic coating

For compact fin-and-tube heat exchangers, hydrophilic coating was often employed to effectively improve the condensate drainage problem. Detailed investigations of the influence of hydrophilic coating were performed by Min et al.[76] and Min and Webb[77]. Though very useful information such as long-tern characteristics and contact angle effect was given in their investigation, however, there was no quantitative correlation to describe the influence of hydrophilic coating. Previous investigations by Mimaki,[78] Wang and Chang,[25] Hong and Webb,[9] and Wang et al.[79] had clearly shown that the influence of hydrophilic coating on the heat transfer coefficients is rather small. In this regard, Wang et al.[32] had developed the following correlation to account the influence of hydrophilic coating on the pressure drop. Test results from eleven samples were used to generate the correlation. The proposed correlation is independent of fin pattern.

7. Heat and Mass analogy

The dehumidifying process involves heat and mass transfer simultaneously, if mass transfer data are unavailable, it is convenient to employ the analogy between heat and mass transfer. The existence of the heat and mass analogy is because the fact that conduction and diffusion in a liquid are governed by physical laws of identical mathematical form. Therefore, for air-water vapor mixture, the ratio of $h_{c,o}/h_d c_p$ is generally around unity (Where $h_{c,o}$ denotes sensible heat transfer coefficient under dehumidification and h_d denotes mass transfer coefficient). This analogy holds for dilute mixtures like water vapor in air near the atmospheric pressure (temperature well-below corresponding boiling point). The validity relies heavily on the mass transfer rate. The experimental data of Hong and Webb[9] indicated that this value is between 0.7~1.1, Seshimo et al.[80] gave a value of 1.1. Eckels and Rabas[81] also reported a similar value of 1.1~1.2 for their test results of fin-and-tube heat exchangers having plain fin geometry. The aforementioned studies all showed a rough the applicability of the analogy. A recent examination by Wang[82] also substantiates the results and showed the results are generally between 0.6~1.1. In addition, it is noted that this ratio is not a constant throughout the test range. A slight drop of the ratio of vs. Reynolds number is seen with the decrease of fin spacing and with the rise of the Reynolds number. This is associated with the more pronounced influence during condensate removal. Moreover, during the dehumidifying process, the temperature gradient is directly responsible for establishing the concentration gradient, suggesting the heat transfer and mass transfer are not independent. Based on a simple analysis, Wang[82] also supported the test results analytically.

8. Conclusions

This study presents a short overview on the air-side performance of the fin-and-tube heat exchangers in both dry and wet conditions. Subjects discussed in this study include reduction method for the air-side performance and updated correlations for the commercially available fin patterns such as plain, herringbone, smooth wavy, louver, and slit fin geometry. The discussions are mainly focused on the experimental study. The interrupted surfaces are the most popular fin pattern existing in the market as well as in researches. Some recent studies clearly highlighted the superior feature of vortex generator, hence it is highly recommended that future efforts should be made towards the realization of this fin concept.

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