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Dennis Nasuta
Optimized Thermal Systems, 7040 Virginia Manor Road, Beltsville, MD, 20705, nasuta@optimizedthermalsystems.com

Song Li
Optimized Thermal Systems, Inc., United States of America, li@optimizedthermalsystems.com

Yunho Hwang
yhhwang@umd.edu

Cara Martin
Optimized Thermal Systems, Inc., Beltsville, MD, USA, cmartin@optimizedthermalsystems.com

Experimental Validation of CFD-Based Correlations for 5 mm Louver- and Slit-Fin Heat Exchangers: Lessons Learned

Dennis NASUTA*[1], Song LI[1], Yunho HWANG[2], Cara MARTIN[1]

[1]Optimized Thermal Systems, Inc.
Beltsville, MD, USA
nasuta@optimizedthermalsystems.com

[2]University of Maryland, Center for Environmental Energy Engineering
College Park, MD, USA

* Corresponding Author

ABSTRACT

This paper examines a recent effort to experimentally test copper-tube, aluminum-fin heat exchangers (HXs) to validate new CFD-based correlations for air-side thermal-hydraulic characteristics of enhanced fins with small-diameter tubes developed prior to this work (Sarpotdar et al., 2016a,b). Sample HXs with 5 mm diameter tubes were prepared from five manufacturers with a variety of slit- and louver-fin patterns and topologies. Experimental data was collected for HXs tested for dry air at 1-4m/s frontal velocity. Experimental data was reduced using a fluid-to-fluid HX solver based on the segment-level ε-NTU method as opposed to conventional ε-NTU relations for cross-flow HX's or Wilson plot method, which are typically used for data reduction. Correction factors were applied to the existing correlations based on experimental findings and the final correlations were able to predict 100% of all observed air-side pressure drops and 98% of observed heat transfer coefficients (HTCs) with 20% error or less. Some deviations between CFD predictions and experimental observations are evident and the work underscores the importance of experimental validation and tuning. It is hypothesized that factors which are not accounted for in the CFD model, including thermal contact resistances between fin and tube, may contribute toward the CFD correlation's consistent over-prediction of observed heat transfer. This method of combining numerical exploration of the design space with limited experimental testing and validation can be used to rapidly develop new, comprehensive correlations in a cost-effective manner.

1. INTRODUCTION

The design of heat transfer systems has been driven by desires to minimize environmental impacts while minimizing costs. Inefficiencies in heat exchanger (HX) performance comprise a significant portion of the total losses in heat transfer systems, so HX design plays a critical role in achieving these goals. In HVAC&R systems, HXs with small-diameter tubes have, in recent years, been implemented to improve the thermal-hydraulic performance and reduce costs. These small-diameter HXs, with tube diameters of 5 mm or less, have been shown to have the potential to improve air-side heat transfer performance, increase compactness, and reduce material consumption. Although these types of HXs have been successfully implemented in a vast array of HVAC&R applications, correlations to predict their performance, particularly on the air-side, have been limited. Experimentally-validated correlations covering a wide range of geometry and flow parameters have not been available for air-side heat transfer and pressure drop of small diameter HXs with louver and slit enhancements.

In 2016, this team published CFD-based correlations for air-side heat transfer and pressure drop of slit and louver fins with tube diameters of 3-5 mm (Sarpotdar, Nasuta, & Aute, 2016a, 2016b). These correlations were developed based on thousands of Computational Fluid Dynamics (CFD) simulations from a Design of Experiments (DOE) sampling of relevant geometry and flow parameters. The correlations were able to predict the results of CFD simulations of randomly-generated HX geometries and velocities with a high degree of accuracy.

Following the publication of these CFD-based correlations, the authors began testing sample 5 mm HXs to experimentally validate the proposed correlations. HXs were acquired from manufacturers in China, India, and the US and were tested in a closed-loop wind tunnel. This paper describes the experimental approach and findings, the performance of the proposed correlations, and presents new, tuned correlations based on the experimental results. Additionally, best practices for HX testing and data reduction are reviewed along with discussion of the validity of CFD-based air-side correlations. The correlations presented here will be useful for engineers to design and optimize HXs with small-diameter tubes.

2. Background

2.1 CFD-Based Correlations

A detailed description of the CFD-based correlation development scope and process is available in the previously mentioned publications (Sarpotdar et al., 2016a, 2016b). Table 1 and Table 2 show the ranges for which the slit and louver correlations were developed; designs throughout this space were selected using DOE techniques and simulated with CFD.

Table 1: CFD-based slit fin correlation ranges (Sarpotdar et al., 2016a)

Design Variable	Unit	Min to Max
D_n	mm	3 to 5
$C_l = P_l / D_n$	--	2 to 4
$C_t = P_t/P_l$	--	1 to 2
N	--	1 to 6
N_s	--	2 to 6
FPI	in^{-1}	14 to 40
$C_h = S_h/F_{sp}$	--	0.3 to 0.7
S_w	mm	1
u	m/s	0.75 to 5
δ_f	mm	0.098

Table 2: CFD-based louver fin correlation ranges (Sarpotdar et al., 2016b)

Design Variable	unit	Min to Max
D_n	mm	3 to 5
$C_l = P_l / D_n$	--	2 to 4
$C_t = P_t/P_l$	--	1 to 2
N	--	1 to 6
N_l	--	2 to 8
FPI	in^{-1}	14 to 40
L_p	mm	0.8 to 1.8
θ	degrees	27
u	m/s	0.75 to 5
δ_f	mm	0.125

2.2 Sample Heat Exchangers

Several sample heat exchangers were acquired from manufacturers in China, India, and the US. These sample coils employed slit and louver fins with a range of fin densities and tube patterns. The major dimensions of these HXs are outlined in Table 3. All tested HXs had tubes with 5 mm nominal diameters; 3 and 4 mm tube HXs were still not widely-available at the time of publication. The sample HXs do not represent the extremes of the design space explored by the CFD-based correlations because many of these geometries are not presently manufacturable. Instead, these coils represent a typical sampling of 5 mm HXs that were commercially-available at the time of project commencement.

All tested HXs were constructed using smooth copper tubes. This ensured that the internal geometries were known, and the water-side performance could be well-predicted by existing empirical correlations. It should be noted that air-side pressure drop data was recorded for coil numbers 1-4, however, problems with test results and subsequent damage to the parts prevented the inclusion of heat transfer performance data for these four coils.

Table 3: Sample Heat Exchanger Characteristics

Coil No.	Mfg.	P_t [mm]	P_l [mm]	Tubes per Bank	Tube Banks	FPI [in^{-1}]	Enhancement Type
1	1	19.50	11.60	24	2	23.2	Slit
2	1	19.50	11.60	24	1	23.4	Slit
3	1	19.50	11.60	24	2	16.9	Slit
4	1	19.50	11.60	24	1	17.9	Slit
5	2	16.00	13.86	30	1	22.0	Slit
6	2	16.00	13.86	30	2	22.0	Slit
7	2	16.00	13.86	30	1	29.8	Slit
8	2	16.00	13.86	30	2	29.8	Slit
9	3	19.05	16.50	24	1	15.1	Louver
10	3	19.05	16.50	24	2	15.1	Louver
11	3	19.05	16.50	24	1	20.1	Louver
12	3	19.05	16.50	24	2	20.3	Louver
13	4	19.00	13.60	24	2	21.0	Louver
14	5	21.00	10.90	24	1	19.1	Louver
15	4	19.00	13.60	24	2	19.5	Louver
16	5	21.00	10.90	24	1	18.5	Louver

3. Experimental Testing

3.1 Experimental Facility

The heat exchangers were tested in a closed-loop wind tunnel facility (Figure 1) configured for air-side testing of HXs with hot water as the working fluid. The test apparatus was constructed in accordance with the ANSI/ASHRAE 33 Standard (2016) and a summary of instrumentation is included in Table 4. Test conditions require operating at a wide range of air flow rates and nozzle configurations, thus measurement uncertainties vary for all data points, however, air pressure drop uncertainty was typically less than 2 Pa and water-side capacity uncertainty was approximately 0.5%. Nasuta (2017) provides additional detail on the experimental procedure and results.

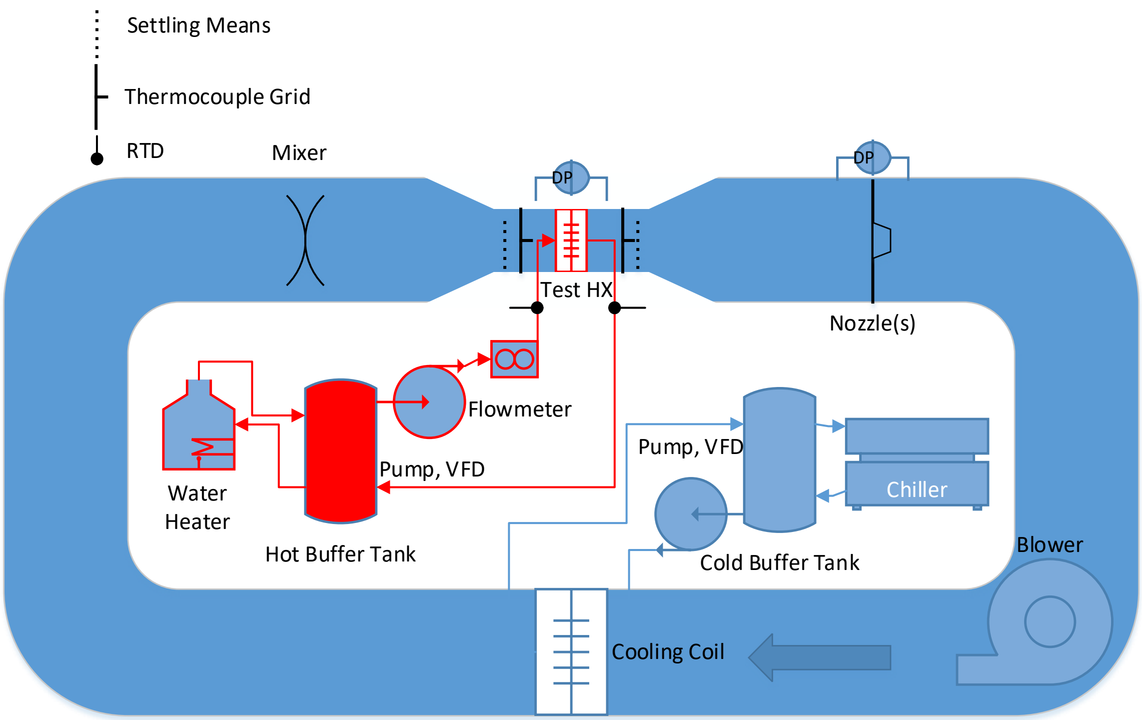

Figure 1: Heat Exchanger Test Facility

Table 4: Summary of Instrumentation

Measurement	Type	Range	Accuracy
Air Temperature	Thermocouple	-200 - 200ºC	±0.5 K (0.17 K for 9-point grid)
Air Relative Humidity	Relative Humidity	0-100%	±2%
Air Pressure	Barometric Pressure	800-1100 mbar	±0.55 mbar
Nozzle Pressure Drop	Differential Pressure	0-1245 Pa	±3.1 Pa
Coil Pressure Drop	Differential Pressure	0-249 Pa	±0.6 Pa
Water Temperature	RTD	-100ºC – 400ºC	±0.03 K
Water Mass Flow Rate	Coriolis Flow Meter	0-150 g/s	±0.5% of reading
Water Pressure	Absolute Pressure	0-5170 kPa	±4.1 kPa

3.2 Test Conditions

Experimental test conditions were established in order to provide data suitable for determination of air-side HTCs. This required testing at conditions where uncertainty was low and where air-side thermal resistance was dominant over water-side and tube-conduction thermal resistance. Water flow rates must be sufficiently high to ensure that flow is fully-turbulent, and water-side resistance should typically make up 15% or less of the total thermal resistance while still maintaining a water-side inlet/outlet temperature difference greater than 2°C so water temperature uncertainty is small. Each HX was tested under nine test points, consisting of the combination of three air flow rates and three water flow rates. Typical conditions are indicated in Table 5, but some two-row HXs were re-circuited and tested with flow rates over 1 kg/s as will be discussed in Section 5. All results used for HTC data reduction had energy balance error less than 5%.

Table 5: Original Test Matrix

Test no.	Air velocity [m/s]	Water flow rate [g/s]	Air Inlet Temp [°C]	Water Inlet Temp [°C]
1	1	70~80	16	50
2	1	70~90	16	50
3	1	90~140	16	50
4	2.5	70~80	16	50
5	2.5	70~90	16	50
6	2.5	90~140	16	50
7	4	70~80	16	50
8	4	70~90	16	50
9	4	90~140	16	50

3.3 Data Reduction

Rather than employing more conventional approaches, such as the Wilson plot method (Wilson, 1915), or the procedure using ε-NTU relations outlined by Wang (2000), HTCs were reduced from the test data by iteratively solving with a numerical HX model (Jiang, Aute, & Radermacher, 2006). Using this approach requires back-calculating the HTC in the software until the model performance matches the experimental observations; this approach is advantageous because it ensures that the HX modeling software will always reproduce the observed performance when the reduced HTC is used in forward calculation. While the approach proposed by Wang relies on ε-NTU relations for parallel- or counter-flow geometries, the iterative approach employed here can account for the actual HX circuitry from within the finite-control-volume model. The Wilson plot method is capable of determining air-side thermal resistance from test data with or without the use of empirical correlations to predict refrigerant-side heat transfer; however, the approach introduces uncertainty in the linear fitting of test data. The implemented approach relies on empirical correlations to predict the water-side HTC. Since these values are well-predicted in smooth tubes by the selected correlations (AHRI, 2001), and the refrigerant-side thermal resistance represents only a small portion of the total resistance, the air-side HTCs can be determined with high confidence.

4. Results

4.1 Air-Side Pressure Drop

Nine test points were recorded for each HX to characterize performance across a range of air velocities. Figure 2 shows the observed louver and slit fin air-side pressure drops, plotted against predictions from the previously-published correlations (Sarpotdar et al., 2016a, b). The same figures show the predicted values directly from the publication, alongside the values after the application of a single correction factor calculated from least-squares regression. For louver fins, the results predicted by the correlation match well with the observations, and a near-negligible 0.984 correction factor can be applied to fit 100% of the data points with ±20% error or less and 68% of the points with ±10% error or less. For the slit fin coils, a 0.822 correction factor can be applied to ensure 100% of the observed values match the predictions with ±20% error or less and 86% of the observations within ±10% error.

4.2 Air-Side Heat Transfer Coefficients

The data reduction approach discussed in Section 3.3 allowed for the consideration of actual HX circuitries, which in this study included full counter-flow and cross-flow type patterns. Furthermore, as will be elaborated on in Section 5.1.2, this methodology allows for the inclusion of fin conduction effects in the HX model. As will be shown, this can have a significant impact on HX performance in some cases and should be considered as part of the data reduction process. Figure 3 shows the predicted and reduced HTCs for louver and slit fins.

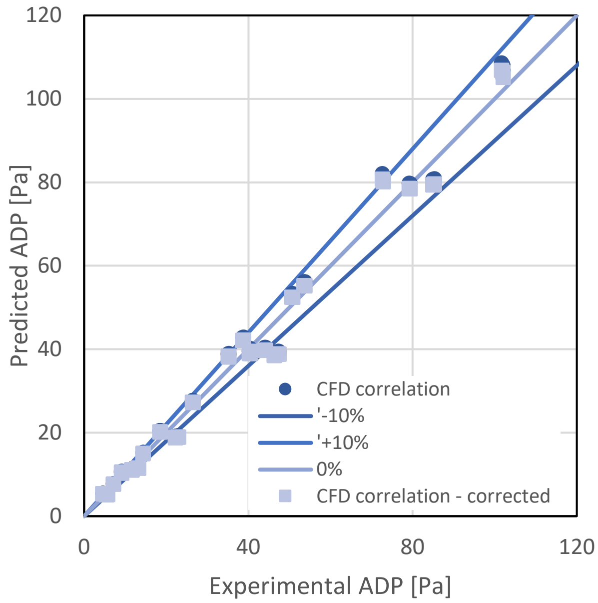

Figure 2: Predicted versus observed air-side pressure drop, Left: louver fin (coils #9-16), Right: slit fin (coils #1-8)

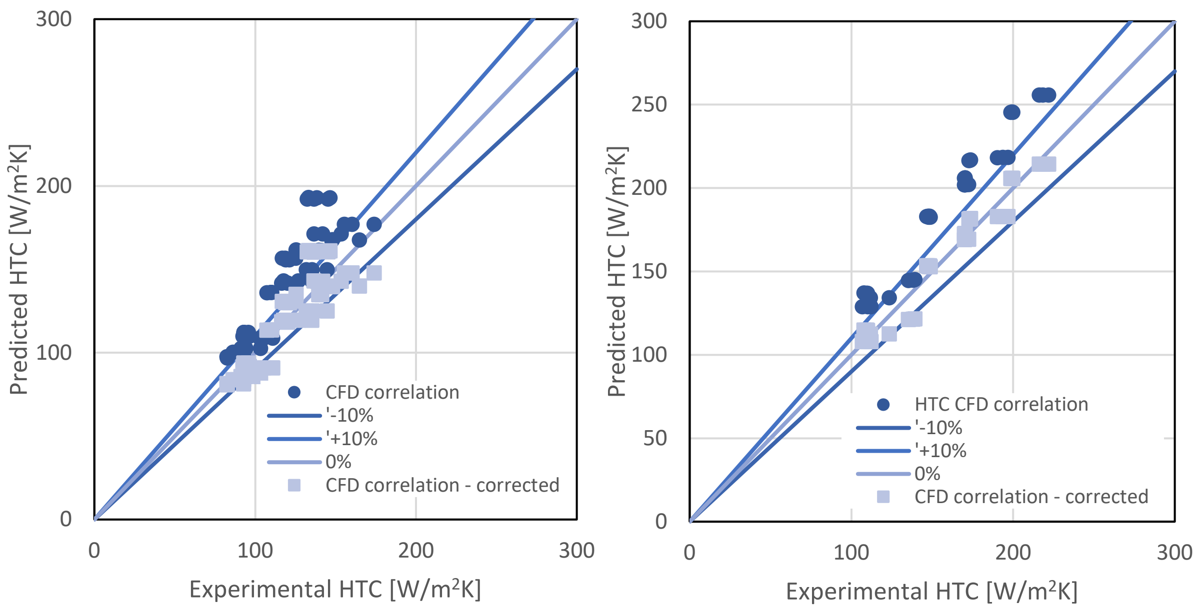

Figure 3: Predicted versus observed heat transfer coefficients, Left: louver fin heat exchangers (coils #9-16) and Right: slit fins (coils #5-8)

For the louver fin HXs, a correction factor of 0.835 was applied to the previously-published correlation (Sarpotdar et al., 2016a). The resulting correlation was able to predict 69% and 96% of all observations within ±10% and ±20% error or less, respectively. Similarly, a correction factor of 0.838 minimized the prediction errors of the existing slit fin HTC correlation (Sarpotdar et al., 2016b). The corrected model was able to predict 92% and 100% of all observations within ±10% and ±20% error or less, respectively. These results for air-side pressure drop and HTC correlations are summarized in Table 6.

Table 6: Corrected correlation performance (Sarpotdar et al., 2016a, 2016b)

Correlation type	Proposed correction factor	Portion of data predicted with ±10% error or less	Portion of data predicted with ±20% error or less
Louver fin air pressure drop	0.984	68.1%	100.0%
Slit fin air pressure drop	0.821	86.1%	100.0%
Louver fin air HTC	0.835	68.5%	96.3%
Slit fin air HTC	0.838	91.7%	100.0%

The observations were also compared against results from existing correlations developed for larger diameter HXs (Chi-Chuan Wang, Lee, & Sheu, 2001, C. -C. Wang, Lee, Chang, & Lin, 1999). Although these correlations were not developed for 5 mm tubes, many designers may still be interested to know if existing correlations can be applied to predict the performance of small-diameter tube HXs. Figure 4 summarizes the predictions made by these existing correlations when compared to the experimental observations from this study. The results of this study, as outlined in Table 7, indicate that these existing correlations may result in high errors when predicting the performance of the sample 5 mm HXs.

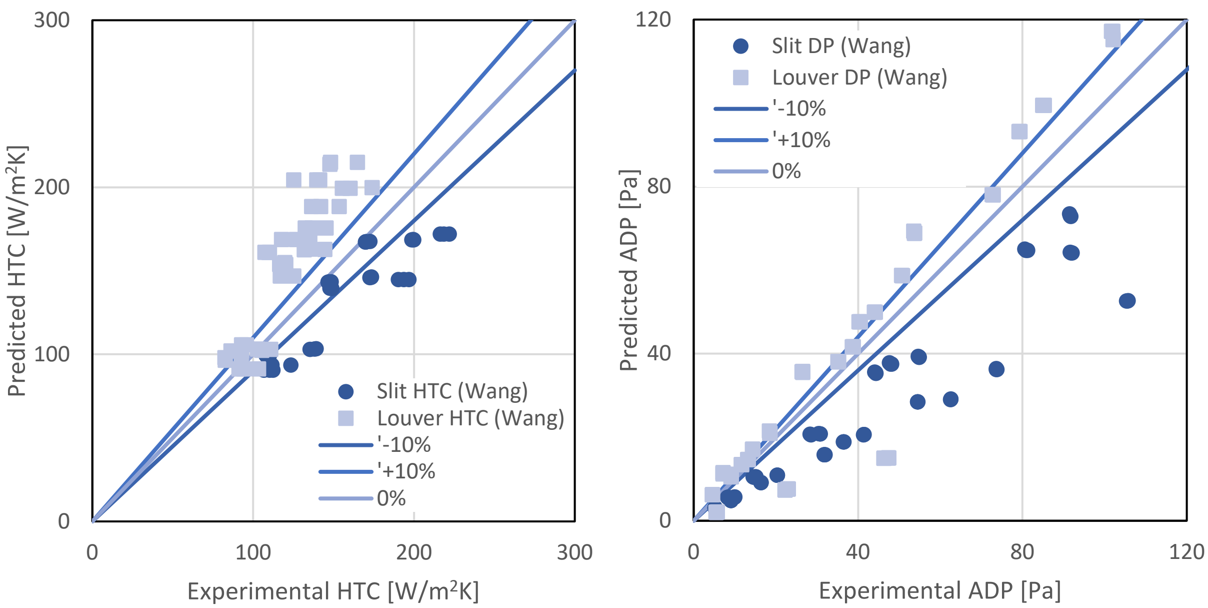

Figure 4: Left: predicted (Wang et al.) versus experimental HTCs; Right: predicted (Wang, et al.) versus experimental pressure drop

Table 7: Predictive ability of existing large-diameter correlations

Correlation type	Portion of data predicted with ±10% error or less	Portion of data predicted with ±20% error or less
Louver fin air pressure drop (Wang et al., 1999)	15.3%	56.9%
Slit fin air pressure drop (Wang et al., 2001)	0.0%	4.2%
Louver fin air HTC (Wang et al., 1999)	13.9%	34.7%
Slit fin air HTC (Wang et al., 2001)	41.7%	72.2%

5. Discussion

5.1 Lessons Learned

5.1.1 Heat Exchanger Test Conditions: In order to acquire measurements suitable to calculate air-side HTCs, it is necessary to perform tests under appropriate conditions. Other authors, including Wang (2000) have provided recommendations for testing conditions. This experimental effort followed these recommendations to maintain at least 2°C water-side temperature difference (ΔT) and operate at water flow rates that minimize water-side thermal resistance, typically 15% or less. Because data reduction is done using CoilDesigner® rather than the ε-NTU relations used by Wang, the recommendation that C* should be less than 0.2 is not necessary, so long as the water-side resistance is appropriately small. As will be discussed in the next section, it is also important to consider potential conduction effects within the HX when selecting water flow rates for a test. Water flow rate should be maximized to reduce water-side thermal resistance and fin conduction effects but must be kept above 2°C so temperature uncertainty is not too large.

5.1.2 Heat Exchanger Fin Conduction: In some cases, HX samples with counter-flow circuitries and a small number of circuits conducted a significant quantity of heat between the tube banks through the fin material. In two-phase HXs, where refrigerant often operates at near-constant saturation temperature, this phenomenon is not often viewed as significant, however, in the case of testing HXs with hot water as the working fluid, significant temperature differences can exist along each water circuit between inlet and outlet.

An extreme example occurred during initial testing, where the two-row coil #8 with 30 FPI and only three circuits developed a >15°C temperature difference between the inlet and outlet of each water circuit. In such a situation, where conduction occurs between two tube banks, the heating of the first row by the second row is detrimental to the overall performance of the HX. As such, when data is reduced to determine the effective air-side HTC, the calculated value can be significantly lower than what may be possible in more favorable conditions. In this example of coil #8 when the coil with three circuits and 0.1 kg/s water flow rate was modified and re-tested with 15 circuits and water flow rates on the order of 0.7 kg/s, the calculated air-side HTCs were as much as 100% greater. In this work, extreme cases like this were resolved by increasing the water flow rate and the number of circuits to minimize these conduction and pinching effects.

5.1.3 Data Reduction Technique: Even in much less extreme cases with lower ΔTs between tube banks, some conduction always occurs across fins. To address this, it is appropriate to use a data reduction technique that accounts for these physics. By utilizing a HX model with a solver that includes the detailed circuitry and fin conduction effects for data reduction, the calculated HTCs are more representative of the actual average air HTCs of the fin surfaces. In geometries where fin conduction is not significant, such as 1-row coils, the difference in calculated HTC can be near-negligible when including or excluding fin conduction physics from the data reduction technique. However, in other geometries, and test conditions, where the effect of conduction is significant, this difference in data reduction procedure can result in calculated HTCs which are frequently more than 10% different from the original result. As such, this procedure for determining HTCs from test data is recommended.

5.2 Applicability of CFD-Based Correlations

The findings presented here suggest that the CFD-based correlations can predict the experimental trends well, however some adjustments are necessary to match the absolute values observed. Some of the predicted errors can be explained by the nature of the original correlations and their inability to exactly predict all of the source data. However, the experimental results reveal a consistent over-prediction of air-side HTC. This observation is expected because the simulated, ideal model, does not account for all imperfections that may be present in a real HX. In fact, the CFD simulations neglect the thermal contact resistance between fin and tube, which other authors have suggested may account for 12.5-55% of total thermal resistance in some fin-tube HXs (ElSherbini, Jacobi, & Hrnjak, 2003).

When conventional air-side HTCs are developed experimentally, authors typically do not isolate the fin-tube contact resistance, and instead the effect is included in the effective air-side HTC. Since the actual contact resistance is not the same for all manufacturers and manufacturing processes, the impact of this effect on a correlation can vary based on the selection of sample HXs used in its development.

This work indicates that the CFD-based approach for developing air-side correlations is appropriate and cost-effective. However, at present, it is still necessary to conduct some experimental tests to validate the simulation results and tune correlations. This tuning is required to resolve potential errors in the simulation results and account for the impact of details that are not included in the CFD model, such as manufacturing imperfections and tube-fin contact resistance. Despite this limitation, the CFD-based approach is highly desirable for developing comprehensive correlations rapidly and at low cost, even for geometries which cannot yet be manufactured.

6. CONCLUSIONS

This effort provided the following contributions:

- Experimental measurements of air-side pressure drop and heat transfer coefficients were provided for a diverse set of 5 mm tube-fin geometries with slit and louver fins
- Comparison of experimental results with existing correlations showed that correlations developed for larger diameter tubes were not able to accurately predict the performance of these small-diameter samples
- New correction factors were proposed to tune the CFD-based air-side correlations to better correlate with experimental data
- CFD simulation errors, fitting errors, manufacturing defects/inconsistencies, and the effect of fin-tube contact resistance were identified as possible reasons for the deviations between the predicted and observed values
- Best practices were reviewed for experimental testing and data reduction to determine heat transfer coefficients
- The impact of fin conduction in hot water heat exchanger tests was shown to be non-negligible in some tests
- A data reduction procedure using a finite-control-volume heat exchanger model, including fin conduction effects, was recommended to determine heat transfer coefficients from experimental data

NOMENCLATURE

C^*	ratio of heat capacity rates C_{min}/C_{max}	(-)
C_h	S_h/F_{sp}	(-)
C_l	P_l/D_n	(-)
C_t	P_t/P_l	(-)
D_n	Nominal Diameter	(mm)
FPI	Fins per Inch	(in^{-1})
F_{sp}	Fin Spacing	(mm)
L_p	Louver Pitch	(mm)
N	Number of banks (in airflow direction)	(#)
N_l	Number of louvers	(#)
N_s	Number of slits	(#)
P_l	Longitudinal (airflow direction) tube pitch	(mm)
P_t	Transverse tube pitch	(mm)
S_h	Slit height	(mm)
S_w	Slit width	(mm)
u	Velocity	(m/s)
δ_f	Fin thickness	(mm)
ΔT	Temperature Difference	(K)
θ	Louver angle	(°)

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