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ABSTRACT

A new correlation is presented to predict the heat transfer coefficients (HTCs) of pure refrigerants and refrigerant mixtures condensing within horizontal microfin tubes. This is accomplished by first putting together a 1084 point experimental database from 21 sources. The data includes CO_2, R1234yf, R1234ze(E), R134a, R22, R407C, R404A, and R410A, 2.64–14.61 mm fin root diameter tubes, -25°C to 50°C saturation temperatures, vapor qualities from 0.02 to 0.98, reduced pressures from 0.16 to 0.81, and heat and mass fluxes ranging from 1.79 to 98.1 kW/m^2 and 49 to 872 kg/m^2.s respectively. The correlation was developed in two steps. One hundred fifteen unique dimensionless parameters pertinent to condensing flows in microfin tubes were first selected. Multi-variable regression analysis was then applied to identify the most significant variables influencing the flow condensation Nusselt number. First, the new correlation was evaluated and compared with six extant correlations on an overall basis. Comparisons were also conducted with the best among the extant correlations, Cavallini *et al.* (2009), for data sorted by refrigerant and fin root diameter. Overall evaluation for the entire database shows that the new correlation is significantly better than any of the extant correlations. In general, the new correlation shows reasonably good predictions, which are better than those of Cavallini *et al*. (2009), are for most parameter bins, with MAD values generally smaller than 20-25%. Based on the bin analysis, parameter ranges in which Cavallini *et al.* (2009) gives better predictions than the new correlation are also identified. Similarly, bins in which more data would be useful for further analysis are also identified. With these few exceptions in mind, the new correlation can be confidently used as a reasonably reliable predictive tool for a large variety of refrigerants under different operating conditions of practical interest.

1. INTRODUCTION

In 1977, Hitachi patented microfin tubes for the first time (Fujie *et al*., 1977). Such tubes are extensively used in HVAC&R condensing applications. The advantage these tubes offer is enhanced heat transfer coefficients (HTCs) with a marginal pressure drop penalty across the tube (Kim, 2016). A cross-sectional view of a typical microfin tube is shown in Figure 1. The important geometrical parameters are tube outer diameter d_o, fin root-to-root diameter d_r, fin tip-to-tip diameter d_t, fin height e, apex angle β, and helix angle γ. Apart from geometric parameters, the flow condensation HTC is affected by the type of manufacturing, operating conditions, refrigerant properties, and flow patterns. The possible variety of combinations of these factors makes the characterization of HTC for microfin condensation a challenging task. Many experimental studies have been conducted to investigate the effect of these factors. Each of these studies only covers a very small subset of all the physics involved in microfin condensation. For thermal-fluid engineers, it is important to determine the suitability of a given microfin tube for a particular application, and it is usually necessary to conduct experimental studies to characterize the tube's performance. However, such studies are difficult and expensive. An accurate condensing HTC correlation can lead to significant savings in experimental effort, cost, and time, while enabling the selection of optimal heat exchangers and thermal systems.
Various research groups have developed correlations to predict condensation HTCs and provided recommendations for and limitations of each correlation. This paper aims at developing a better and more accurate correlation using a new statistical approach as explained in (Mehendale, 2018). In section-2, the experimental data used for evaluating the existing correlations and developing the new correlation are discussed. Section 3 discusses the performance of the

Yu and Koyama (1998), Kedzierski and Goncalves (1999) (less and more accurate versions), Han and Lee (2005), Koyama and Yonemoto (2006) and Cavallini *et al.* (2009) correlations for the current database. The procedure used to develop the new correlation is briefly outlined in section 4. The performance of the new correlation and its comparison with Cavallini *et al*. (2009), the best of the extant correlations, is discussed in section 5.

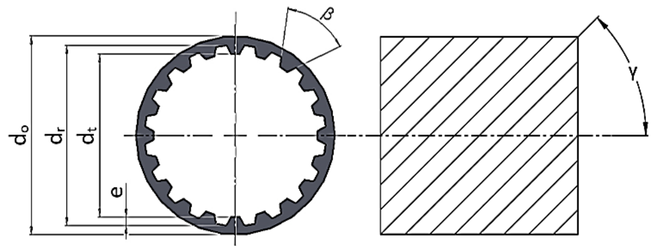


Figure 1. Cross-sectional view of typical microfin tube

2. EXPERIMENTAL DATABASE FOR MICROFIN TUBE CONDENSATION

Recently there has been a focus on using 3 to 5 mm diameter tubes to provide more compact heat transfer equipment (Mehendale, 2018). Low temperature applications like the rapid freezing and storage of frozen foods, as well as cascade refrigeration systems, require low mass flux in the condenser (Messineo, 2012), (Getu and Bansal, 2008). Again, due to more stringent environmental regulations, there has been an emphasis on developing and using low GWP refrigerants like R1234yf and R1234ze(E). To enhance the usefulness of the new correlation, points for these new refrigerants, low diameter tubes, and low mass fluxes have been included in the experimental database. The current database consists of 1084 data points extracted from 21 different sources. These points were collected using Webplot digitizer (Rohatgi, 2018). Since plot axes were carefully calibrated before collecting data, the error associated with data collection should be negligible. Table 1 gives a summary of the key geometric features of the tubes included in the current study. The corresponding refrigerants and operating conditions are provided in Table 2. The cooling fluid used, method of calculating the HTCs, uncertainty ranges for the HTCs, as well as the base areas for q, G, and h are summarized in Table 2.

Table 1. Microfin tube geometries for refrigerant condensation experimental studies

Source	**Data points**	**d_r (mm)**	**n_f**	**e (mm)**	**γ (deg)**	**β (deg)**
Colombo *et al*. (2012)	68	8.62 – 8.92	54 – 82	0.185 – 0.195	18	40
Diani *et al*. (2017)	100	3.64	40	0.12	18	43
Diani *et al*. (2018)	180	2.64	40	0.12	7	43
Honda *et al*. (2005)	82	5.38	45	0.164	19	45
Huang *et al*. (2010)	35	3.56 – 4.6	40	0.12 – 0.14	12 – 18	40
Jang and Hrnjak (2004)	44	6.32	54	0.18	14	24
Jung *et al*. (2004)	102	8.92	60	0.2	18	53
Kim (2016)	20	4.6	40	0.15	18	40
Kim and Shin (2005)	56	8.8 – 8.82	54 – 65	0.12 – 0.25	15.5 – 30	0 – 55
Kim *et al*. (2009)	28	3.74	57	0.22	6	30
Kim *et al*. (2018)	8	6.44	65	0.1	18	40
Koyama *et al*. (2008)	116	5.8	50	0.23	15	30
Lee *et al*. (2014)	15	6.50	65	0.12	15	40
Li *et al*. (2012)	22	4.54 – 4.6	35 – 58	0.1 – 0.15	18	21 – 42
Olivier *et al*. (2007)	15	8.94	60	0.209	18	40
Vu *et al*. (2017)	77	6.61	65	0.12	17	67
Wu *et al*. (2013)	14	6.06 – 6.56	58 – 65	0.06 – 0.15	0 – 30	13 – 125
Wu *et al*. (2014)	30	4.54 – 8.98	35 – 70	0.1 – 0.16	18	21 – 42
Bogart and Thors (1999)	24	8.87	60	0.203	18	59.53
Kim *et al*. (2013)	12	6.49 – 6.58	55 – 60	0.15 – 0.2	10 – 18	18.3 – 59
Eckels and Tesene (1999)	36	7.34 – 14.61	50 – 60	0.203 – 0.305	18	45 – 57

An important consideration in analyzing the data is that different base areas have been used for reporting the heat flux, mass flux, and HTCs in the various experimental studies, as shown in Table 3. Additionally, each of the existing correlations considered in this study requires a specific basic area for q, G, and h. Therefore, all the data points have been converted into the base areas required by the particular correlation using equations (1), (2) & (3):

$$q_{exp}.A_{exp}=q_{corr}.A_{corr}\ ,\ G_{exp}.A_{c,exp}=G_{corr}.A_{c,corr}\ ,\ h_{exp}.A_{exp}=h_{corr}.A_{corr} \qquad (1), (2), (3)$$

Table 2. Operating conditions for refrigerant condensation experimental studies

Source	**Refrigerant(s)**	**P_{sat} (kPa, absolute)**	**q (W/m^2)**	**G (kg/m^{2}s)**	**X**
Colombo *et al*. (2012)	R134a	887.5	5833 – 82071	88.80 – 436	0.15 – 0.8
Diani *et al*. (2017)	R1234yf	783.5 – 1018	20413 – 89289	100 – 1000	0.16 – 0.93
Diani *et al*. (2018)	R1234yf, R1234ze(E), R134a	579.6 – 1017	11707 – 74025	300 – 1000	0.16 – 0.95
Honda *et al*. (2005)	R407C	1549 – 1726	1790– 42183	50 – 300	0.11 – 0.96
Huang *et al*. (2010)	R410A	2420 – 2424	4210 – 8420	200 – 500	0.2 – 0.82
Jang and Hrnjak (2004)	CO_2	1683 – 2291	16304 – 91663	200 – 400	0.09 – 0.94
Jung *et al*. (2004)	R22, R134a, R407C, R410A	1017 – 2425	7800	100 – 300	0.05 – 0.93
Kim (2016)	R410A	2728 – 2732	4000	49.38 – 250	0.21 – 0.80
Kim and Shin (2005)	R22, R410A	1534 – 2425	11000	280	0.12 – 0.88
Kim *et al*. (2009)	CO_2	1683 – 2291	8092 – 72190	200 – 800	0.1 – 0.91
Kim *et al*. (2018)	R410A	2048 – 2055	6000	80 – 200	0.3 – 0.8
Koyama *et al*. (2008)	CO_2	5002 – 6003	6009 – 24054	200 – 350	0.04 – 0.98
Lee *et al*. (2014)	R410A	2727 – 2733	10000	100 – 400	0.15 – 0.9
Li *et al*. (2012)	R22	1813	6814 – 25023	206.51 – 627.89	0.45
Olivier *et al*. (2007)	R22, R134a, R407C	1017 – 1650	7575 – 15467	400 – 800	0.475
Vu *et al*. (2017)	R410A, R22	1856 – 2932	6000 – 12000	50 – 380	0.02 – 0.95
Wu *et al*. (2013)	R410A	2852	4102– 39858	98.54 – 573.80	0.45
Wu *et al*. (2014)	R410A	2852	6118– 31080	99.64 – 584.87	0.45
Bogart and Thors (1999)	R410A, R22	1557 – 2458	13398 – 57619	205 – 839	0.45
Kim *et al*. (2013)	R410A	2926 – 2931	10000	590	0.21 – 0.78
Eckels andTesene (1999)	R410A	2422 – 3070	9379 – 59217	120 – 608	0.12 – 0.8

Table 3. HTC calculation method, method of cooling, uncertainties, and base areas for q, G and h

Source	**Cooling fluid**	**HTC calculation method**	**Uncertainty in h**	**Base area for q**	**Base area for G**	**Base area for h**
Colombo *et al*. (2012)	Water	Wall temperature	± 3 – 8.1 %	A_r	$A_{c,a}$	A_r
Diani *et al*. (2017)	Water	Wall temperature	± 2.1 – 7.3 %	A_o*	$A_{c,t}$	A_t
Diani *et al*. (2018)	Water	Wall temperature	± 2.1 – 7.3 %	A_o	$A_{c,t}$	A_t
Honda *et al*. (2005)	Water	Wall temperature	± 4.8 %	A_r	$A_{c,r}$	A_r
Huang *et al*. (2010)	Water	Wall temperature	± 12.5 %	A_a	$A_{c,a}$	A_a
Jang and Hrnjak (2004)	HFE**	Wall temperature	± 1.19 – 20.34 %	A_{mi}*	$A_{c,mi}$*	A_{mi}
Jung *et al*. (2004)	Water	Wall temperature	± 7 %	A_a*	$A_{c,a}$*	A_a*
Kim (2016)	Water	Wilson plot	± 11.1 – 13 %	A_{mi}	$A_{c,mi}$*	A_{mi}
Kim and Shin (2005)	Water	Wilson plot and wall temperature	± 12.5 %	A_m	$A_{c,m}$	A_m
Kim *et al*. (2009)	HFE**	Wall temperature	± 6.3 – 25.6 %	A_{mi}*	$A_{c,mi}$*	A_{mi}
Kim *et al*. (2018)	Water	Wilson plot	± 17.30 %	A_{mi}*	$A_{c,mi}$*	A_{mi}
Koyama *et al*. (2008)	Water	Wall temperature	Not reported	A_a	$A_{c,a}$*	A_a
Lee *et al*. (2014)	Water	Wilson plot	± 17.30 %	A_{mi}	$A_{c,mi}$*	A_{mi}
Li *et al*. (2012)	Water	Wall temperature	± 9.6 %	A_a	$A_{c,a}$	A_r
Olivier *et al*. (2007)	Water	Wilson plot	± 9 – 11.3 %	A_a	$A_{c,a}$	A_a
Vu *et al*. (2017)	Water	Wall temperature	Not reported	A_a*	$A_{c,a}$*	A_a*
Wu *et al*. (2013)	Water	Wall temperature	± 15.40 %	A_a	$A_{c,a}$	A_r
Wu *et al*. (2014)	Water	Wall temperature	± 15.40 %	A_a	$A_{c,a}$	A_r
Bogart and Thors (1999)	Water	Wilson plot	± 10 %	A_{r*}	$A_{c,a*}$	A_r
Kim *et al*. (2013)	Water	Wilson plot	± 17.3 %	A_t	$A_{c,a}$	A_{mi}
Eckels andTesene (1999)	Water	Wall temperature	± 8 – 18 %	A_{r*}	$A_{c,a*}$	A_r

*Assumed base area ** Hydrofluoroether 7100

3. PERFORMANCE OF EXISTING CORRELATIONS

The six correlations selected for the current study are Yu and Koyama (1998), Kedzierski and Goncalves (1999) (less and more accurate versions), Han and Lee (2005), Koyama and Yonemoto (2006), and Cavallini *et al*. (2009). These correlations were selected based on the number of citations, which is a measure of their popularity. In this section, the experimental HTCs in the 1084 point database have been compared with the HTCs predicted by the above correlations. The statistical metrics used for this comparison are ξ_{30}, Mean Deviation (MD), and Mean Absolute Deviation (MAD). ξ_{30} gives the percentage of points predicted by a given correlation within ± 30% of the experimental HTC. As seen from Table 3, leaving out the sources with unreported experimental uncertainties, the maximum uncertainty in the experimental HTC is 25.6%. For this reason, $\xi_{30}$was considered a reasonable metric to assess correlation performance. These six selected correlations were used to predict the HTCs corresponding to the 1084 points in the database. The scatter plots comparing the experimental and predicted HTCs appear in Figures 2(a)-(f). It should be noted that Figure 3(b) in section-5.1provides the legend used for Figures 2(a)-(f).

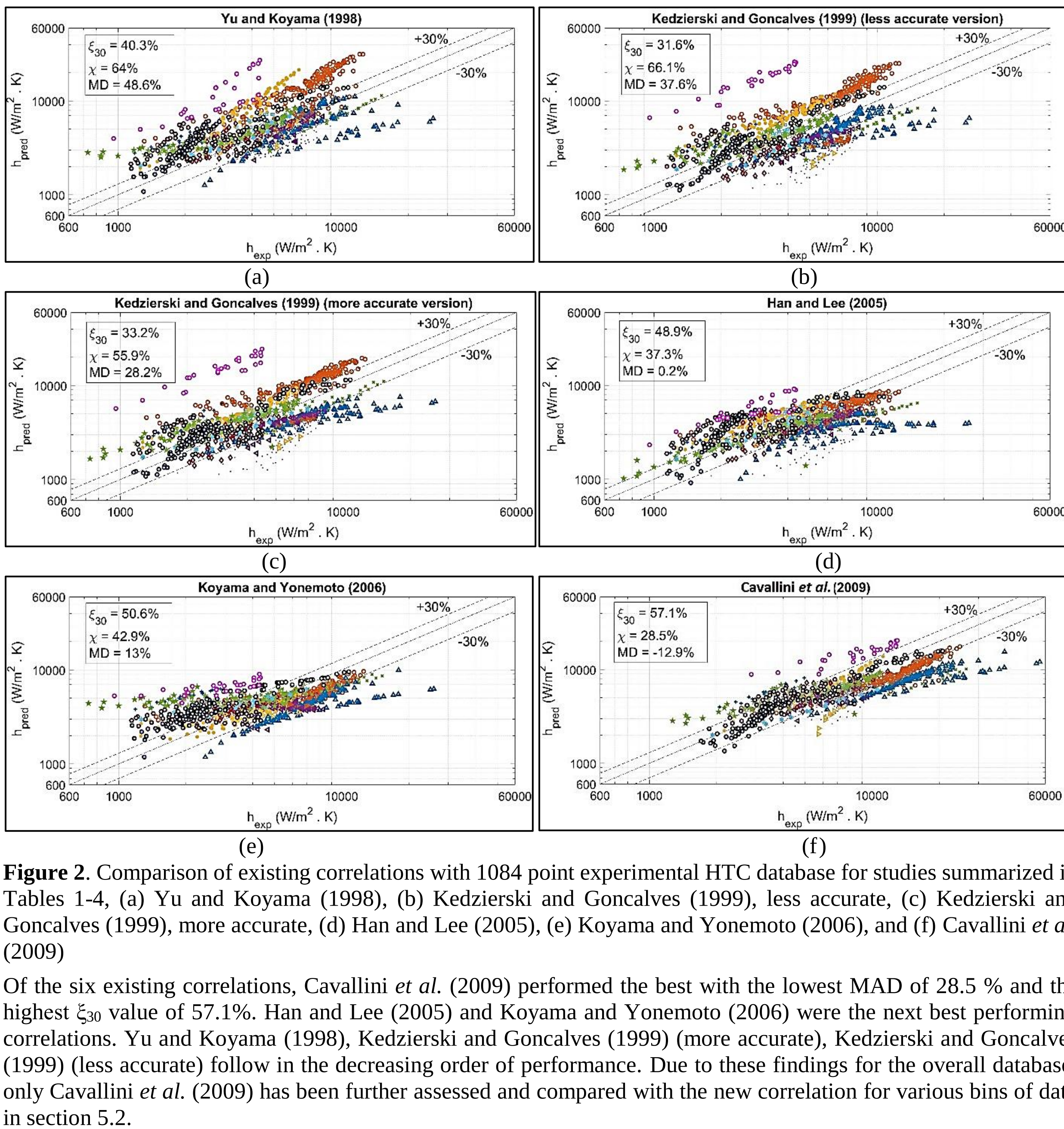


Figure 2. Comparison of existing correlations with 1084 point experimental HTC database for studies summarized in Tables 1-4, (a) Yu and Koyama (1998), (b) Kedzierski and Goncalves (1999), less accurate, (c) Kedzierski and Goncalves (1999), more accurate, (d) Han and Lee (2005), (e) Koyama and Yonemoto (2006), and (f) Cavallini *et al.* (2009)

Of the six existing correlations, Cavallini *et al.* (2009) performed the best with the lowest MAD of 28.5 % and the highest ξ_{30} value of 57.1%. Han and Lee (2005) and Koyama and Yonemoto (2006) were the next best performing correlations. Yu and Koyama (1998), Kedzierski and Goncalves (1999) (more accurate), Kedzierski and Goncalves (1999) (less accurate) follow in the decreasing order of performance. Due to these findings for the overall database, only Cavallini *et al.* (2009) has been further assessed and compared with the new correlation for various bins of data in section 5.2.

4. NEW FLOW CONDENSATION HTC CORRELATION

A new correlation has been developed as part of this research to predict HTCs for pure refrigerants and refrigerant mixtures condensing within horizontal microfin tubes. The method of deriving the correlation is very similar to that developed in an earlier work on flow boiling (see Mehendale (2018) for the details), the only difference being that the dimensionless variables considered in the present work pertain to condensing, and not boiling, flows. Multi-variable regression analysis was applied to the entire 1084 point database to derive a correlation between the dependent variable (the condensing flow Nusselt number Nu_{cond}) and key dimensionless variables that significantly influence the physical processes involved in the condensation within the microfin tube. This new correlation is given below in its entirety in Table 4.

Table 4. Equations and constants of the new correlation

$Nu_{cond} = \frac{h_{cond} d_r}{k_l} = C_0 \cdot \Pi_1^{C_1} \cdot \Pi_2^{C_2} \cdot \Pi_3^{C_3} \cdot \Pi_4^{C_4} \cdot \Pi_5^{C_5} \cdot \Pi_6^{C_6} \cdot \Pi_7^{C_7} \cdot \Pi_8^{C_8} \cdot \Pi_9^{C_9} \cdot \Pi_{10}^{C_{10}} \cdot \Pi_{11}^{C_{11}} \cdot \Pi_{12}^{C_{12}} \cdot \Pi_{13}^{C_{13}} \cdot \Pi_{14}^{C_{14}}$	(4)
$\Pi_1 = \mathrm{Nu}_{\mathrm{Gn,\,homo}}$, $\Pi_2 = \left(4064.4 \cdot (d_r - 2 \cdot e) + 23.257\right)/n_f$, $\Pi_3 = \frac{G \cdot X_{lv}}{\sqrt{g \cdot d_r \cdot \bar{\rho} \cdot \Delta\rho}}$, $\Pi_4 = \frac{d_r}{n_f \cdot e}$	(5) – (8)
$\Pi_5 = g \cdot d_r^3 \cdot \Delta\rho^2 \left(\frac{x}{\mu_v} + \frac{1-x}{\mu_l}\right)^2$, $\Pi_6 = Su_v = \frac{\rho_v \cdot \sigma \cdot d_r}{\mu_v^2}$, $\Pi_7 = Ka_h = \frac{\left(x \cdot \mu_v + (1-x) \cdot \mu_l\right)^4 \cdot g}{\bar{\rho} \cdot \sigma^3}$	(9) – (11)
$\Pi_8 = Bond = \frac{g \cdot \Delta\rho \cdot e \cdot d_r}{\sigma \cdot n_f}$, $\Pi_9 = Ga_h = \frac{\bar{\rho} \cdot g \cdot \Delta\rho \cdot d_r^3}{\left(x \cdot \mu_v + (1-x) \cdot \mu_l\right)^2}$, $\Pi_{10} = Pr_h$, $\Pi_{11} = \frac{\Delta\rho}{\rho_l}$, $\Pi_{12} = M / M_{H_2}$	(12) – (16)
$\Pi_{13} = \frac{q}{h_{lv}^{1.5} \cdot \Delta\rho}$, $\Pi_{14} = E_a = \left(\frac{2 \cdot e \cdot n_f}{\pi \cdot d_r}\right) \cdot \left(\sqrt{\left(\frac{1}{\cos^2\beta} + tan^2\left(\frac{\gamma}{2}\right)\right)} - tan\left(\frac{\gamma}{2}\right)\right) + 1$	(17), (18)
$C_0 = 0.955\mathrm{e}^{-12.38}$, $C_1 = 0.6075$, $C_2 = 6.5254$, $C_3 = -0.2201$, $C_4 = -3.1063$, $C_5 = 0.2005$, $C_6 = 0.2290$, $C_7 = 0.5184$, $C_8 = -2.4510$, $C_9 = 0.6167$, $C_{10} = -0.8982$, $C_{11} = -9.3991$, $C_{12} = 0.5056$, $C_{13} = 0.0447$, $C_{14} = -0.3720$	(19) – (33)

The significant Π variables in Table 4 are now briefly discussed. $Nu_{Gn,\,homo}$ is the Nusselt number calculated using the Gnielinski (2013) correlation. It should be noted that the turbulent $Nu_{Gn,\,homo}$ is based on the homogeneous Reynolds numbers and Prandtl numbers, Re_{homo} and Pr_{homo}, respectively. In the turbulent regime ($Re_{homo} \geq 4000$), the Nusselt number is calculated using:

$$\mathrm{Nu_{turb}} = \frac{(\mathrm{f_{turb}}/8)(\mathrm{Re_{homo}}\text{-}1000)\mathrm{Pr_{homo}}}{1+12.7\sqrt{\mathrm{f_{turb}}/8}\left(\mathrm{Pr_{homo}}^{2/3}\text{-}1\right)} \quad (34)$$

Where, the friction factor is calculated using: $f_{turb} = \left(1.8\log_{10}\left(Re_{\mathrm{homo}}\right) - 1.5\right)^{-2}$ (35)

In the laminar regime, ($Re_{homo} \leq 2300$): $Nu_{lam} = 3.66$ (36)

In the transition region ($2300 < Re_{homo} < 4000$), $Nu_{Gn,\,homo}$ is linearly interpolated between 3.66 and Nu_{turb} ($Re_{homo} = 4000$). Π_2 is the ratio of the optimum number of microfins (similar to that in Tsuchida *et al*., 1993) to maximize the condensing HTC to the actual number of microfins. Π_3 represents the dimensionless vapor velocity, given by equation (7). X_{lv} used in equation (7) is the variant of the Lockhart-Martinelli parameter and is given by:

$$X_{lv} = \sqrt{(f_l / f_v) \cdot (\rho_v / \rho_l)} \cdot \left(\frac{1-x}{x}\right) \quad (37)$$

In equation (37), the Darcy liquid and vapor flow friction factors f_l and f_v have been calculated using the Churchill (1977) correlation for a smooth tube that spans the laminar, turbulent, and transition regimes. When calculating f_v, Re_v is to be used instead of Re_l in the Churchill (1977) correlation. Dimensionless variables $\Pi_6 - \Pi_{12}$ depend exclusively on fluid properties (and d_r in some cases), and are well known from the two-phase flow literature. Further discussion of these variables can be found in Mehendale (2018). Π_{13} depends on the heat flux as well as fluid properties, and was shown to be important in flow boiling phenomena as well (Mehendale (2018)). Finally, Π_{14} is also a purely geometric factor, representing the ratio of the actual surface area of the microfin tube to the surface area of a smooth tube with the same fin root diameter.

5. COMPARISON OF THE NEW CORRELATION AND CAVALLINI *et al*. (2009)

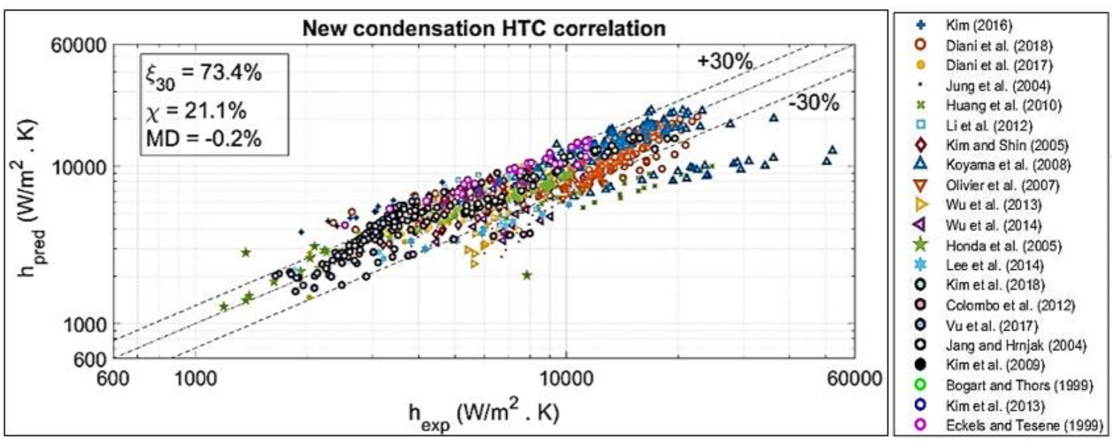
In this section, the predictions of the new correlation are presented from various perspectives, and compared with the performance of Cavallini *et al.* (2009), which, as shown in section 3, is significantly better compared to the other five extant correlations. First, the overall performance of the new correlation for the entire 1084 point database is discussed. After this, the performance of the new correlation is assessed by dividing the data by refrigerant, as well by various ranges of fin root diameter and mass flux.

5.1 Performance of the new correlation for the entire database

Figure 3(a) shows a scatter plot comparing the new correlation's HTC predictions to the experimental HTCs for the entire database of 1084 points summarized in Tables 1–3. The legend for the sources of data in Figure 3(a) is given in Figure 3(b), and is identical to that for Figures 2(a)–(f). Referring to Figures 2(a)–(f) and Figure 3(a), the following conclusions can be drawn:

a. For the entire database, the new correlation gives ξ_{30} which is 16.3% higher than that for the best extant correlation, Cavallini *et al.* (2009).
b. For the entire database, the new correlation gives χ which is 7.4% lower than the corresponding value for the best extant correlation, Cavallini *et al.* (2009).

Thus, the new correlation additionally predicts 177 out of the 1084 points within ± 30% error bands compared to the Cavallini *et al.* (2009) correlation.



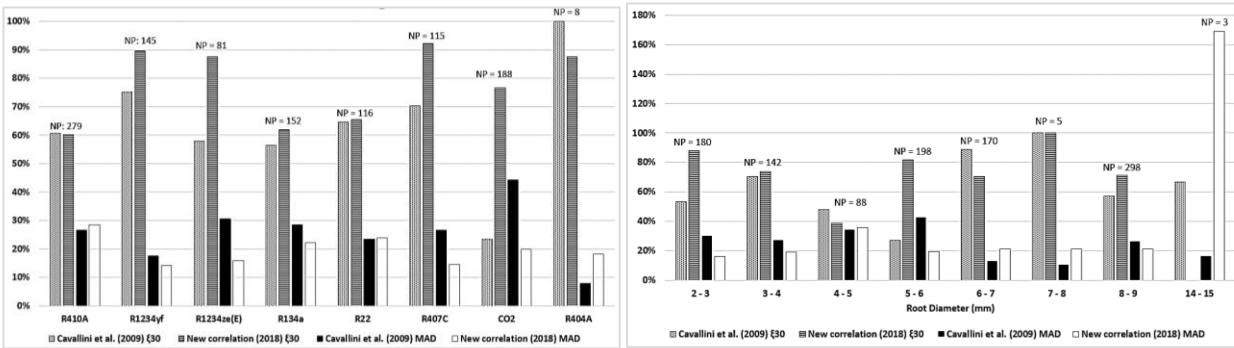
(a) (b)

Figure 3.(a) Scatter plot for new HTC correlation for the entire database (b) legend for figures 2 (a)–(f), and 3(a)

5.2 Performance of the new correlation for various distributions of data

Some valuable insights can be gained by studying microfin performance from several different perspectives, taken one at a time. Following the approach of Mehendale (2018), Figures 4(a), (b) show, for two important parameters (refrigerant and fin root diameter), bar chart distributions of ξ_{30} and the corresponding χ value for the new correlation and the Cavallini *et al.* (2009) correlation. The number of data points in each bin are also shown in Figures 4 (a), (b) above the respective groups of bars. Overall, the new correlation shows reasonably good predictions for most parameter bins, with MAD values generally smaller than 20–25%.

From Figure 4(a) it can be seen that the new correlation gives more accurate predictions than Cavallini *et al*. (2009) for refrigerants R1234yf, R1234ze(E), R134a, R22, R407C, and CO_2. For R410A, the two correlations have similar accuracy, while the Cavallini *et al*. (2009) correlation is significantly better for R404A. However, it should be pointed out that there are only 8 R404A points in the database, and the conclusions drawn based on such a small sample space might are likely not statistically significant.



(a) (b)

Figure 4. Comparison of ξ_{30} and MAD values for the Cavallini *et al.* (2009) and the new correlation for data sorted by (a) refrigerant and (b) fin root diameter

From Figure 4(b), it is seen that the new correlation performs better than Cavallini *et al.* (2009) for tubes in the 2-4, 5-6, and 8-9 mm range of fin root diameters. On the other hand, Cavallini *et al.* (2009) gives better predictions for data in the 4-5 and 6-7 mm range. Cavallini *et al.* (2009) also performs better for 7-8 and 14-15 mm data. However, these data bins only contain 5 and 3 points respectively, and therefore comparisons for these ranges of tube diameter are likely not statistically meaningful.

6. CONCLUSIONS AND FUTURE WORK

A new correlation for predicting the heat transfer coefficients of pure refrigerants and refrigerant mixtures flow condensing in horizontal microfin tubes has been developed. First, a data bank consisting of 1084 experimental points collected from 21 sources was compiled as part of this research. The database covers the following range of parameters:

1. Refrigerants: CO_2, R1234yf, R1234ze(E), R134a, R22, R404A, R407C, and R410A
2. Fin root diameter: 2.64–14.61 mm
3. Saturation temperature: -25°C–50°C and reduced pressures: 0.16–0.81
4. Mass flux: 49–872 kg/m^{2}s. (Mass flux is based on the smooth tube flow area $A_{c,r}$).
5. Heat flux: 1.79–98.1 kW/m^2. (Heat flux is based on the fin root surface area, A_r).
6. Vapor quality: 0.02–0.98.
7. All-liquid Reynolds number: 2009–65582 and all-vapor Reynolds number: 14371–545989

The main findings of the study are summarized below:

i. For the entire database, the new correlation gives the best overall predictions with a MAD of 21.1%, and 73.4% of the data falling within ± 30% error bands. Among the six extant correlations, Cavallini *et al*. (2009) was the best performing, with 57.1% of the points predicted within ± 30%, while it gave a MAD of 28.5%.

ii. Apart from these overall assessments, the distribution of ξ_{30}, χ, and number of data points for the new correlation and that for Cavallini *et al*. (2009) was examined relative to refrigerant, and for various bins of fin root diameter, d_r. In general, the new correlation shows reasonably good predictions for most parameter bins, with MAD values generally smaller than 20-25%. However, the following exceptions are noteworthy:
   a. The Cavallini *et al.* (2009) correlation is significantly better for R404A. However, there are only eight R404A points in the database, and the conclusions drawn based on such a small sample space might are likely not statistically meaningful. More R404A experimental data would be desirable to improve the predictions of the new correlation.
   b. Cavallini *et al.* (2009) gives better predictions for data in the 4-5 and 6-7 mm range. Cavallini *et al.* (2009) also performs better for 7-8 and 14-15 mm data. However, these data bins only contain 5 and 3 points respectively, and therefore these comparisons are likely not statistically meaningful for these two data bins. More data in the 7-8 and 14-15 mm ranges would be desirable to improve the predictions of the new correlation.

Based on these comparative assessments, with the exception of the ranges listed under ii. (a), (b) above, the new correlation can be confidently used as a reliable tool to predict the microfin condensation HTCs of a large variety of refrigerants under different operating conditions of practical interest.

NOMENCLATURE

A	Heat transfer surface area	m^2
A_c	Cross sectional area	m^2
d	Diameter	
e	Fin height	m
G	Mass flux	$kg\ m^{-2}\ s^{-1}$
h	Heat transfer coefficient	$W\ m^{-2}\ K^{-1}$
h_{lv}	Latent heat of vaporization	$J\ kg^{-1}$
k	Thermal conductivity	$W\ m^{-1}\ K^{-1}$
n_f	Number of microfins	
Nu	Nusselt number	
P_{red}	Reduced pressure P_{sat}/P_c	
Pr	Prandtl number $\mu C_p/k$	
q	Heat flux	$W\ m^{-2}$
Re_{al}	All-liquid Reynolds number $G\ d_r / \mu_l$	
Re_{av}	All-vapor Reynolds number $G\ d_r / \mu_v$	
Re_l	Liquid Reynolds number $(1 - x)\ G\ d_r / \mu_l$	
Re_v	Vapor Reynolds number $x\ G d_r / \mu_l$	
x	Vapor quality	
ΔT	$T_{sat} - T_{wall}$	K
$\bar{\rho}^{-1}$	Homogeneous specific volume $x\rho_v^{-1} + (1-x)\rho_l^{-1}$	$m^3 kg^{-1}$
Δx	Change in quality	
P_{sat}	Saturation pressure	Pa
g	Acceleration due to gravity	$m\ s^{-2}$
$\Delta\rho$	$\rho_l - \rho_v$	$kg\ m^{-3}$

Subscripts

a	Actual
h	Hydraulic
l	Liquid
r	Fin root
mi	Melt down
t	Fin tip
v	Vapor
eq	Equivalent
o	Outside
sat	saturation
exp	experimental
corr	correlation
basis, q	Base area for q
basis, h	Base area for h
basis, G	Base cross sectional area for G
lam	Laminar
turb	Turbulent

Greek symbols

γ	Microfin helix angle	deg
β	Microfin apex angle	deg
ρ	Density	$kg\ m^{-3}$
σ	Surface tension	N/m
μ	Absolute viscosity	$kg\ m^{-1}\ s^{-1}$
ξ_{30}	Percentage of points within ±30% error bands	%
Π	Dimensionless variables used to develop new correlation	
χ	Mean absolute deviation (MAD)	%

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