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Technical Note

Experimental investigation of turbulent heat transfer and fluid flow in internally finned tubes

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Nomenclature

A, b, c, d constants and exponents in equation (12)
A_{core} core flow area through an internally-finned tube ($=A_n(1-H)^2$) [m^2]
A_{fin} inner fin flow area through an internally-finned tube ($=A_{xs}-A_{core}$) [m^2]
A_n nominal flow area of an internally-finned tube ($=\pi d_{t,i}^2/4$) [m^2]
A_{xs} actual flow area of an internally-finned tube ($=A_n-Nes$) [m^2]
d tube diameter [m]
e fin height [m]
f fanning friction factor based on nominal inside diameter [equation (7)]
h heat transfer coefficient, based on nominal heat transfer area [W m^{-2} K^{-1}]
H non-dimensional fin height ($=2e/d_i$)
k thermal conductivity [W m^{-1} K^{-1}]
l_c characteristic length [m]
l_{csw} modified characteristic length for swirling flows [m]
n exponent in variable property correlation
N number of fins
Nu, Nu_h, Nu_{lc} Nusselt number based on nominal tube diameter ($=hd_i/k$), hydraulic diameter ($=hd_h/k$), or characteristic length ($=hd_{lc}/k$)
Pr Prandtl number
Q volumetric flowrate [m^3 s^{-1}]
Re, Re_h, Re_{lc} Reynolds number based on nominal tube diameter ($=Vd_i/v$), hydraulic diameter ($=Vd_h/v$), or characteristic length ($=Vd_{lc}/v$)
s mean fin thickness [m]
SA inside heat transfer area [m^2]
SW modified non-dimensional axial pitch ($=N\sin\gamma/\pi$)
V mean axial velocity based on nominal area ($=4Q/\pi d_i^2$) [m s^{-1}]
W non-dimensional inter-fin flow area [equation (11)].

Greek symbols
γ fin helix angle [°]
μ dynamic viscosity [Pa s^{-1}]
ρ fluid density [kg m^{-3}].

Subscripts
act actual
b bulk
c characteristic
cp constant property
h hydraulic
i inner
lc characteristic length
n nominal (based on tube inside diameter)
o outer
st smooth tube
w wall.

Other symbol
$\partial p/\partial z$ pressure gradient [Pa m^{-1}].

1. Introduction

A recent bibliography (Bergles et al. [1]) lists the literature on turbulent heat transfer in internally finned tubes. An excellent review on this topic is given by Shome [2]. In brief, using a variety of fluids (e.g., air, water, oil, R-113, ethylene glycol/water), a number of investigators (e.g., Edwards and Jensen [3], Vasil'chenko and Bar-

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baritskaya [4], Watkinson et al. [5], Carnavos [6], Chiou et al. [7], Kaushik and Azer [8], Brognaux et al. [9]) have studied this enhancement technique over a range of number of fins, fin heights, helix angles, and fin profiles. They obtained increases in the Nusselt numbers from 15–180% greater than that in a smooth tube. This benefit was counterbalanced by increases in friction factor of 50–500%.

Correlation of the data is difficult because of the many different parameters that can influence the Nusselt number and friction factor. The most widely cited correlations for friction factor and Nusselt number were proposed by Carnavos [6]:

$$f_{\mathrm{h}} = 0.046 Re_{\mathrm{h}}^{-0.2} (A_{\mathrm{n}}/A_{\mathrm{xs}})^{-0.5} (\sec\gamma)^{0.75} \tag{1}$$

$$Nu_{\mathrm{h}} = 0.023 Re_{\mathrm{h}}^{0.8}\, Pr^{0.4} (A_{\mathrm{xs}}/A_{\mathrm{core}})^{0.1} \times (SA_{\mathrm{act}}/SA_{\mathrm{n}})^{-0.5} (\sec\gamma)^{3}. \tag{2}$$

However, the general applicability of these correlations is questionable because of the many relatively tall fin tubes with low helix angles in their database and the correlation approach. Other correlations have been proposed (e.g., Chiou et al. [7], Kaushik and Azer [8], Brognaux et al. [9]) for friction factors and Nusselt numbers but these correlations have been developed with either very limited databases or have functions specific to the tubes used in the studies.

Because of these limitations, the objective of the current research is to develop physically based correlations for Nusselt number and friction factor for the finned tube geometry, which are generally applicable. To achieve this objective, a detailed experimental investigation of turbulent fluid flow in internally finned tubes was performed with a wide range of fin geometric and operating conditions. Length-averaged Nusselt numbers and friction factors were measured for both heating and cooling situations.

To measure the heat transfer enhancement of internally finned tubes, two geometrically identical double pipe heat exchangers were used. The test fluid flowed through the tube side of each of the heat exchangers in counterflow with hot water in one test section and cold water in the other. Each test tube was 4.72 m long with the first 1.52 m of tube preceding the heat exchangers used as an adiabatic calming length. Two different test fluids were used, water and ethylene glycol (for low Reynolds friction factor tests).

Sixteen pairs of tubes (15 finned and one smooth tube) were tested. The smooth tube was used to validate the test rig, data reduction, and methodology, and to serve as a baseline reference case for the finned tube results. The details of the internally finned tubes are shown in Table 1. Tubes numbered 1–11 were manufactured by Wolverine Tube and have the prefix 'WO'. Tubes numbered 12–15 with the prefix 'WI' were manufactured by Wieland-Werke AG of Germany.

The data were reduced in terms of tube nominal inside diameter and area. This approach was recommended by Marner et al. [10]. The main advantage of this technique is to facilitate direct comparison of finned and smooth tube performance. See Vlakancic [11] for details.

2. Experimental results and discussion

Smooth tube isothermal friction factors were measured and compared to the smooth tube correlation by Filonenko (see Kakac [12]):

$$f_{\mathrm{iso}} = [1.58 \ln Re - 3.28]^{-2}. \tag{3}$$

Agreement between the measured and predicted values was within $\pm 5\%$. Smooth tube diabatic friction factors were obtained with bulk-to-wall viscosity ratios of $0.62 \leqslant \mu_{\mathrm{b}}/\mu_{\mathrm{w}} \leqslant 1.35$. The measured friction factors were corrected with Petukhov's [13] variable viscosity correction factor.

$$\frac{f}{f_{\mathrm{cp}}} = \begin{cases} (7-\mu_{\mathrm{b}}/\mu_{\mathrm{w}})/6 \text{ heating} \\ (\mu_{\mathrm{w}}/\mu_{\mathrm{b}})^{0.24} \text{ cooling} \end{cases}. \tag{4}$$

The corrected friction factors agreed with the predictions to within $\pm 5\%$.

The smooth tube Nu were corrected with Petukhov's [13] variable viscosity corrections:

$$Nu/Nu_{\mathrm{cp}} = (\mu_{\mathrm{b}}/\mu_{\mathrm{w}})^{n} \tag{5}$$

where $n = 0.11$ for heating and $n = 0.25$ for cooling. The corrected Nu were compared to the Gnielinski [14] correlation:

$$Nu = 0.012 Pr^{0.4} [Re^{0.87} - 280]. \tag{6}$$

The data and predictions were in good agreement with each other and the Gnielinski correlation.

Typical results for the effect of the number of fins, N, on the friction factors are shown in Figs 1(a) (for tall fins) and 2(a) (for short fins). As can be seen in Fig. 1(a), these tubes show friction factors curves similar to those of a smooth tube, only displaced higher. This is typical of the tubes tested with $H > 0.06$. The flow below $Re \approx 10\,000$ seems to be in transition, and the flow beyond 10 000 to be fully turbulent, where the friction factor increases with increasing N.

Tubes with short fins ('micro fins') display much different friction factor trends than their higher finned counterparts (see Fig. 2(a)). For instance, in the region, $Re < 10\,000$, the friction factor is insensitive Re and the fin height has no appreciable effect on friction factor for the tubes with $H = 0.02$ and 0.03. This insensitivity to Re continues until about $Re \cong 20\,000$, when the friction factor begins to decrease with increasing Re. For the tube with $H = 0.04$, the friction factor steadily increased from $10\,000 < Re < 20\,000$, reached a plateau, and then decreased with increasing Re at rate similar to that of a

Table 1
Geometry of smooth and internally finned tubes tested

Tube	Outside diameter d_o (mm)	Inside diameter d_i (mm)	Number of fins N	Fin height e (mm)	Fin thickness s (mm)	Fin helix angle γ (°)	H	Group
ST	25.40	21.18	—	—	—	—	—	—
WO01	26.57	24.23	14	2.06	0.64	30	0.17	High
WO02	25.48	23.72	14	1.18	0.92	15	0.1	High
WO03	25.42	23.70	30	1.30	0.82	30	0.1	High
WO04	25.44	23.64	8	1.16	1.00	30	0.1	High
WO05	25.50	23.78	14	1.20	1.02	30	0.1	High
WO06	25.40	24.18	30	0.36	0.64	30	0.03	Micro
WO07	25.40	24.03	54	0.18	0.40	30	0.015	Micro
WO08	25.43	24.41	54	0.36	0.70	30	0.03	Micro
WO09	25.43	24.05	54	0.36	0.62	15	0.03	Micro
WO10	25.40	24.13	54	0.33	0.90	45	0.03	Micro
WO11	25.44	23.64	14	1.18	0.70	0	0.1	High
WI12	25.00	23.00	36	0.68	1.14	25	0.06	Micro
WI13	24.30	21.86	22	0.62	1.84	25	0.06	High
WI14	24.90	22.10	54	0.22	0.58	45	0.02	Micro
WI15	24.70	22.08	54	0.44	0.54	45	0.04	Micro

smooth tube. Al-Fahed et al. [15], Chiou et al. [7], and Brognaux et al. [9] also have observed this behavior.

Generally, whenever one parameter N, H, or γ is increased, the friction factor increases. However, notice on Fig. 2(a) the tube with $H = 0.02$ has a higher friction than the tube with $H = 0.03$ at higher Re, contrary to what is expected. The tube with $H = 0.03$ had much thicker fins. The smaller flow area between those fins would be more affected by the viscous sublayer and give lower velocities than tubes with greater fin spacing. The lower velocity would in turn account for the lower friction factor. The effect of N, H, or γ are not linear. For example, for tubes with relatively tall fins and low number of fins, the effect of helix angle becomes more pronounced for $\gamma \geqslant 30^\circ$, while at lower γ, there is only a marginal increase in friction factor. The diabatic friction factor when heating and cooling were corrected for variable viscosity using the smooth tube correction factors recommended by Petukhov [13]. The trends are generally consistent with those of the isothermal friction factor data, and the smooth tube correction factors bring the diabatic friction factors into reasonable agreement.

The Nusselt numbers, Nu, were corrected for variable viscosity effects using the smooth tube correction factors of Petukhov [13]. A Pr dependency exponent of 0.4, which is typical for smooth tubes, was used, as did Carnavos [6]. All data were compared to the Gnielinski smooth tube correlation. Some data show slight differences between the heating and cooling Nu.

As shown on Fig. 1(b) for tall fins Nu increases with N in a straightforward progression; with $N = 30$, the enhancement is nearly three times that of the smooth tube. For tubes with micro fins (Fig. 2(b)), while there still was substantial enhancement observed (Nu was nearly doubled), the number of fins, at least in the range tested, appeared to have only a small effect. For both types of tubes, the slope of the Nu curves generally follows that of the smooth tube. Note, however, that the increases in Nu for high finned tubes are slightly more than that of a smooth tube at lower Re and slightly less at the higher Re. This slope difference might be attributed to the benefits of swirling flow at lower Re as mentioned by Manglik and Bergles [16] for twisted tape inserts. Consistent with the friction factor data on Fig. 2(b), the tube with $H = 0.02$ has Nu greater than the tube with $H = 0.03$. Again, a possible explanation of this seemingly contradictory trend is that the fin profiles were different for these two tubes. (These tubes were retested to confirm this trend.)

Generally, the effects of N, H, and γ on Nu are consistent with their effects on friction factor. An increase in the parameter results in an increase in Nu; their influence is nonlinear. For example, the helix angle has a much stronger effect for the tall fin tubes than what was observed for micro fin tubes. For tall fin tubes, changing the helix angle from 0–30° increased Nu by 30–60%. For micro fin tubes, a change in γ from 15–45° increased Nu by only 20–40%.

Extensive comparisons were made against the Carnavos correlations for friction factor and Nu. While data from several of the present tubes fall within $\pm 10\%$ of Carnavos' correlations, a significant number do not.

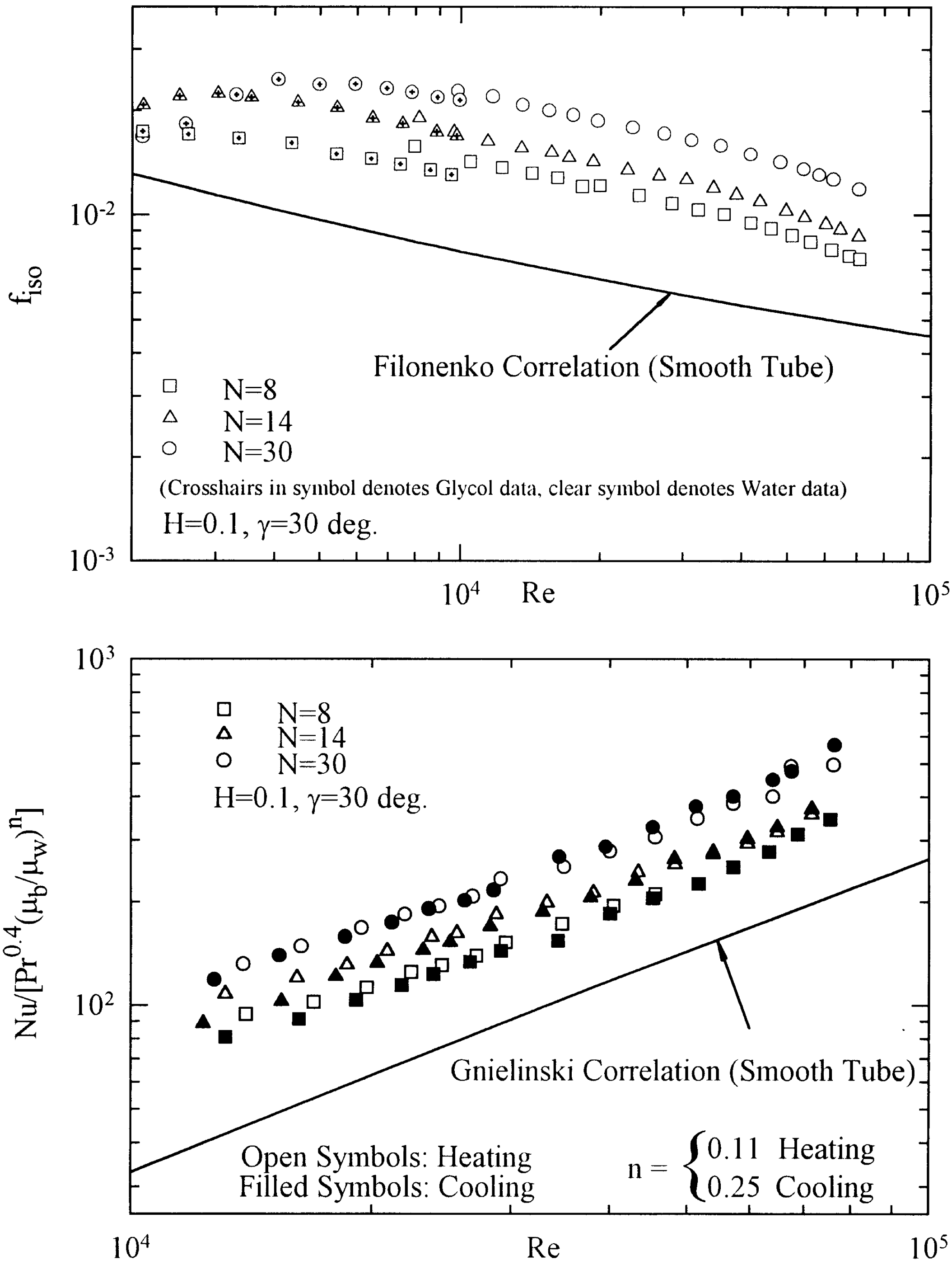

Fig. 1. Effect of N on (a) isothermal friction factors and (b) Nusselt numbers.

Data from neither micro fin tubes ($H \leqslant 0.04$) nor the taller fin tubes were accurately correlated by the Carnavos correlation. For friction factor, the comparisons ranged from $+75$ to -25%. For Nu the agreement ranged from $+200$ to -30%. Generally, there was poorer agreement when $\gamma > 30°$.

Data obtained from the nine low fin tubes ($H \leqslant 0.06$) were also compared to the Chiou et al. [7] correlations for

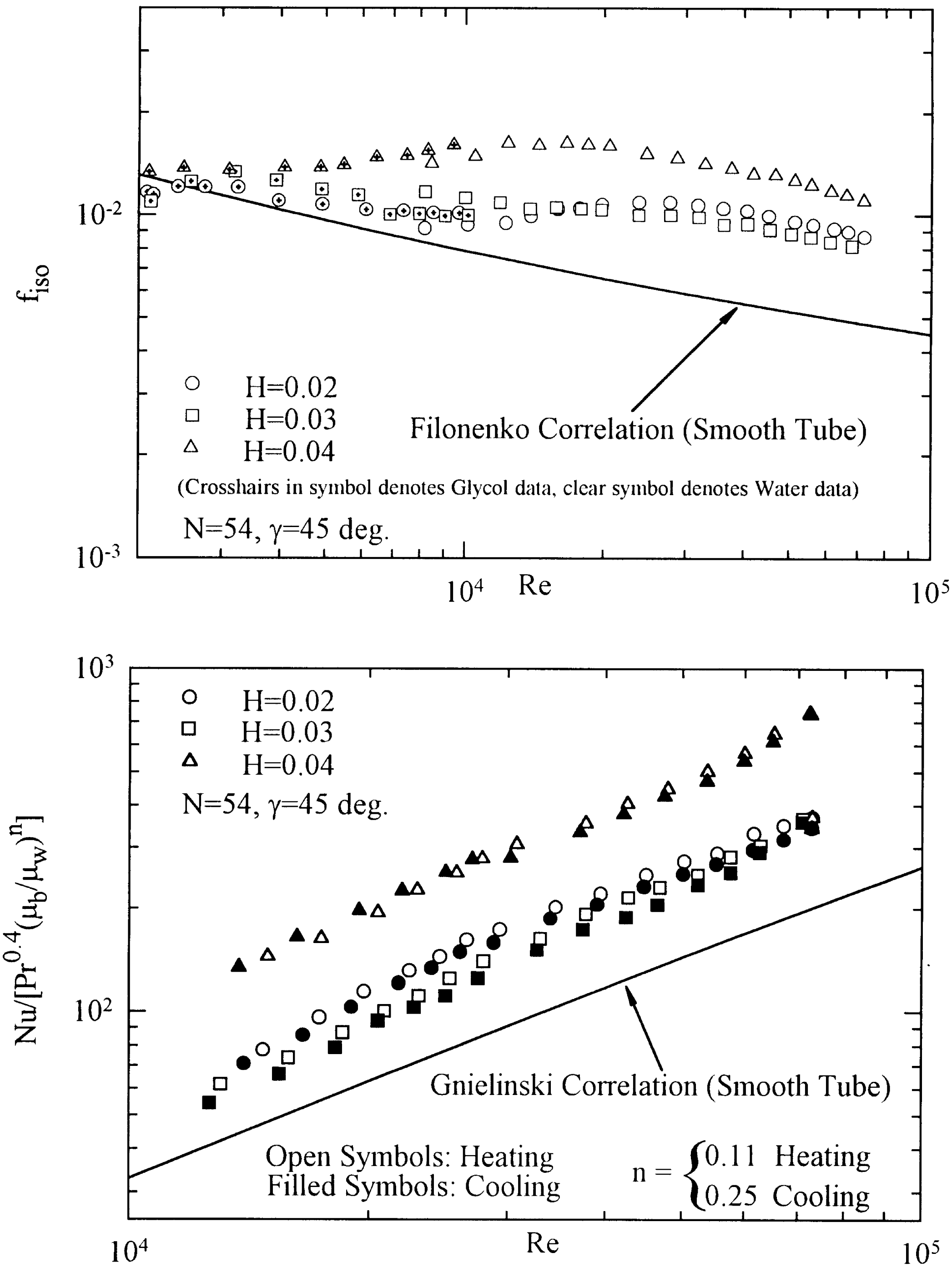

Fig. 2. Effect of H on (a) isothermal friction factors and (b) Nusselt numbers.

micro-fin tubes. The friction factor correlation produced unacceptable overprediction (as high as 2400%, with seven tubes consistently overpredicted by a minimum of 100%). The Chiou et al. correlation for Nu also has poor accuracy. Only one tube falls within $\pm 20\%$ for the entire Re range. The accuracy ranges from $+160$ to -60%.

The isothermal friction factor data reveal two distinct types of tubes, those with relatively tall fins and those with small or 'micro' fins. The micro-fin tubes display a long transitional period before becoming fully turbulent at $Re \cong 20\,000$. During the transitional period the friction factors are insensitive to Re. Tall fin tubes have friction factor curves with the same slope as smooth tubes, only the curve is displaced higher and reached fully turbulent flow at $Re \cong 10\,000$. These two types of tubes have been classified (see Table 1) into 'high-fin' and 'micro-fin'. Note that two of the tubes, WI12 and WI13, have the same nondimensional fin height ($H = 0.06$), yet are classified in different groups. These tubes had different friction factor characteristics due to the different number of fins (36 compared to 22). Hence, for this study, any tube with $H \leqslant 0.06$ and 30 or more fins were classified as a micro-fin tube. Overall, the increase in friction factor for the high finned tubes ranged from 40–170% over that of a smooth tube. Micro-fin tubes showed at 40–140% increase over smooth tubes.

Nu trends for tall finned tubes and micro-fin tubes reveal a different slope at lower Re for the two types of tubes. This characteristic was attributed to the greater capacity of swirling flow for higher finned tubes. However, the trends with geometry were similar to those noted for the friction factors. Overall, the tubes with high fins experienced a 50–150% increase in Nu over that of a smooth tube. Micro-fin tubes experienced even greater enhancement that ranged from 20–220% over smooth tubes. More extensive discussion can be found in Vlakancic [11].

3. Correlation development

Most of the early correlations developed for turbulent flow in finned tubes utilize the hydraulic diameter. While this method is accurate for a wide variety of noncircular ducts, it is not a good performance predictor for ducts of complex cross section, necessitating the use of additional parameters to adjust for the effects of geometry (e.g., Carnavos [7]).

A different length scale has been proposed by Malak et al. [17] and developed by Edwards and Jensen [3]. The latter authors used a characteristic length based on the largest turbulent structures in the flow. Their measurements showed that two turbulence scales, proportional to the core and interfin regions of flow, governed the overall turbulence structure. The two scales were combined into a single, averaged value for the characteristic length. Edwards and Jensen defined the friction factor and Re in terms of this length scale and geometry as:

$$f = \left(\frac{\partial p}{\partial z} l_c\right) \Big/ \frac{1}{2}\rho V_c^2 = f\left(Re = \frac{\rho V_c l_c}{\mu}, \text{geometry}\right). \tag{7}$$

The characteristic velocity, V_c, is an axial velocity based on the nominal flow area. The characteristic length, l_c, is the inside tube diameter for a circular tube, while for internally finned tubes, the characteristic length is given by:

$$\frac{l_c}{d_i} = \frac{A_{\text{core}}}{A_{\text{xs}}}(1-H) + \frac{A_{\text{fin}}}{A_{\text{xs}}}\left(\frac{\pi}{N}\left(1-\frac{H}{2}\right) - \frac{s}{d_i}\right). \tag{8}$$

The pressure drop data were put into a Blasius form for friction factor:

$$\frac{f}{f_{\text{st}}} = \left(\frac{l_c}{d_i}\right)^{-1.25}\left(\frac{A_n}{A_{\text{xs}}}\right)^{1.75}. \tag{9}$$

The flow area ratio accounts for in tube flow velocity calculated using actual flow area rather than nominal flow area as was used in this study. Equation (9) reduces to the Filonenko correlation [equation (3)] when $H = 0$.

Equation (8) was developed for longitudinally finned tubes. For the present data, the presence of a helix angle, and its spiralling flow, modifies l_c by interfering with the development of turbulent eddies inside the tube. Flow in the core region of the tube would strike peaks of fins at some angle rather than merely travelling along the tops of fins lengthwise as in the case of longitudinal fins. In addition, the flow in inner fin regions would be subject to increased flow path. Hence, l_c must be modified to include axial pitch and fin height to take into account spiraling flow, and the two types of tubes noted in the previous section.

To correlate the present data, the characteristic length is modified by H and axial pitch $SW = N \sin\gamma/\pi$. Note that $\sin\gamma$ is used instead of $\tan\gamma$. This approximation is used because in the range of $\gamma = 0$–$30°$ both functions are approximately similar, and for tubes with $\gamma > \cong 45°$, $\sin\gamma$ does not approach infinity as does $\tan\gamma$, thus resulting in a better behaved correlation.

For the high-fin tubes, the friction factor data were correlated as:

$$l_{\text{csw}}/d_i = (l_c/d_i)[1 - 0.203(SW)^{0.65}(H)^{0.20}]. \tag{10}$$

Equation (10) is used in equation (9) in conjunction with the original length scale [equation (8)]. About 95% of the data fall within $\pm 15\%$ with a standard deviation of 4.8% (see Fig. 3(a)). The correlation is accurate to $Re \cong 5000$ and reduces to the original Edwards and Jensen correlation for $\gamma = 0$.

Micro-fin tube data indicated that beyond $Re \cong 20\,000$ flows can be considered fully turbulent. The interfin flow area affects friction factor behavior since, in one case, the tube with the large γ and significantly lower interfin flow area had lower pressure drop than a geometrically similar tube with lower γ. Thus, the following parameter was developed:

$$W = (\pi/N - s/d_i)\cos\gamma \tag{11}$$

which is the nondimensional interfin flow area, with the cosine term accounting for the actual helical path length.

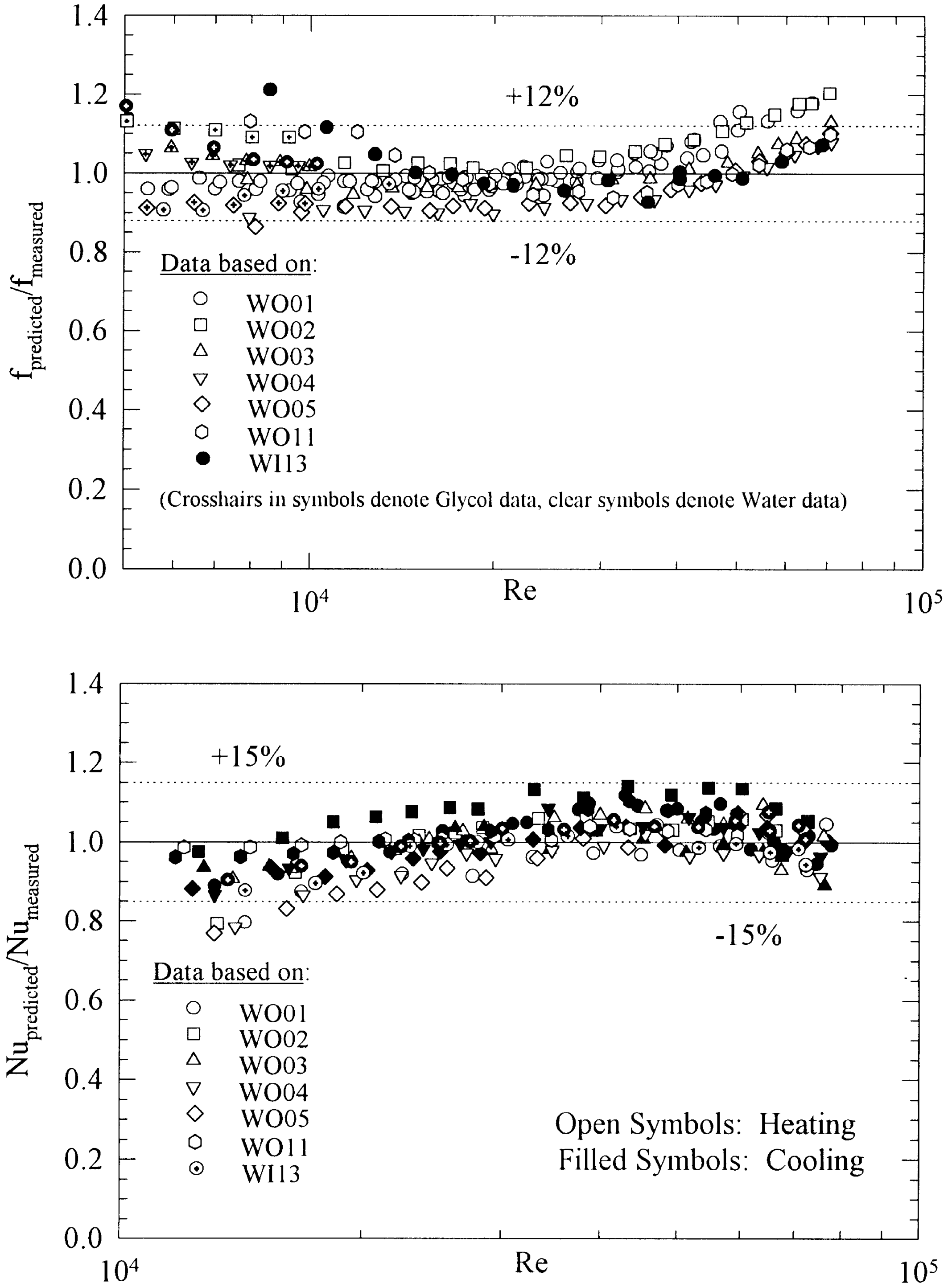

Fig. 3. Comparison of experimental data with proposed correlations for high-fin tubes. (a) Friction factors. (b) Nusselt numbers.

The characteristic length for flows above $Re \cong 20\,000$ is correlated as:

$$l_{csw}/d_i = [1 - A(SW)^b (H)^c (W)^d]$$
$$A = 1.577,\quad b = 0.64,\quad c = 0.53,\quad d = 0.28 \quad \text{for } H \leqslant 0.04$$
$$A = 0.994,\quad b = 0.89,\quad c = 0.44,\quad d = 0.41 \quad \text{for } 0.04 < H \leqslant 0.06. \qquad (12)$$

Different regression curves were fitted to tubes with $H \leqslant 0.04$ and $0.04 < H \leqslant 0.06$. An explanation is that tubes with $0.04 < H \leqslant 0.06$ is a third 'transitional group' which has characteristics of both high-fin and micro-fin tubes. Note that equation (12) does not include the characteristic length as developed by Edwards and Jensen. Reasons for this include that in micro-fin tubes H is a much stronger correlating parameter in micro-fin tubes and the original characteristic length does not vary much in micro-fin tubes, since H and the interfin area are so small.

Adaptation of the correlation to transitional flow would be useful for low Re flows. As seen in the data, at approximately $Re \cong 2500$, the friction factors for all the micro-fin tubes are similar to that of a smooth tube. What remains is to connect the friction factor correlation at $Re \cong 20\,000$ to that of the smooth tube at $Re \cong 2500$. This can be done by adopting the form:

$$\frac{f}{f_{st}} = \left(\frac{l_{csw}}{d_i}\right)^{-1.25}\left(\frac{A_n}{A_{xs}}\right)^{1.75} - \frac{0.0151}{f_{st}} \times \left\{\left(\frac{l_{csw}}{d_i}\right)^{-1.25}\left(\frac{A_n}{A_{xs}}\right)^{1.75} - 1\right\}\exp^{(-Re/6780)}. \qquad (13)$$

This equation is designed to reduce to equation (9) for $Re > 20\,000$. About 95% of the data fall within $\pm 12\%$ of the correlation, even as low as $Re \cong 3000$, with a standard deviation of 6.4%. (The data scatter for the micro fin tubes is about the same Fig. 3(a).)

As with pressure drop, Edwards and Jensen developed a correlation for heat transfer based on the characteristic length concept. The correlation has the form:

$$\frac{Nu}{Nu_{st}} = \left(\frac{l_c}{d_i}\right)^{-1/2}\left(\frac{A_n}{A_{xs}}\right)^{0.8}[f(\text{geometry})] \qquad (14)$$

where the Gnielinski correlation was used for Nu_{st} with the variable viscosity correction factors of Petukhov [13]. For the present data, in equation (14) l_{csw} replaces l_c for use with spirally finned tubes.

The heat transfer characteristics of several high-finned tubes display a slight but noticeably different slope than that of a smooth tube, particularly for heating data at $Re < 20\,000$. This was attributed to the increased capacity for swirling flow in the high-fin tubes. Including this additional Re dependency, the present data were correlated with:

$$f(\text{geometry}) = (SA_{act}/SA_n)^{0.29} \times [1 - 1.792(SW)^{0.64}(H)^{2.76}(Re)^{0.27}] \qquad (15)$$

SA_{act} and SA_n represent the actual surface area of the tube and the nominal surface area of the tube, respectively. As shown on Fig. 3(b), for $Re > 10\,000$ approximately 97% of the data were predicted within $\pm 15\%$, with a standard deviation of 5.6%.

For micro-fin tubes their data trends for heat transfer did not significantly deviate from the slope of a smooth tube. Hence, no additional Re dependence was needed, and the correlation is:

$$f(\text{geometry}) = (SA_{act}/SA_n)^{1.0} \times [1 - 0.059(SW)^{-0.31}(W)^{-0.66}]. \qquad (16)$$

Several features of this geometric function include the linear dependence on the total surface area of the tubes, the opposite effect of SW on the geometric function than on a high-fin tube, and the absence of H. The absence of H is attributed to observations that the geometric function increases with increasing H from $0.015 \leqslant H \leqslant 0.04$, and then decreases from $0.04 \leqslant H \leqslant 0.06$. This non-linear behavior might be due to a different physical mechanism becoming dominant near the tube wall at low fin heights, which is not present at higher H. For $Re > 10\,000$ about 94% of the data fall within $\pm 20\%$ of the correlation with a standard deviation of 10.2%.

The current correlations were compared to data from nine of Carnavos' [6] high fin tubes. Over the range $10\,000 < Re < 70\,000$, the present correlation predicted all the Carnavos friction factors within $\pm 12\%$ except for two tubes. One was predicted about 15% low and the other about 25–30% high. We attribute this difference to the fact that this latter tube had fins which were significantly taller than the tallest used in this study ($H = 0.31$ vs $H = 0.17$). The present Nu correlation for $10\,000 < Re < 70\,000$ predicted most of Carnavos' data within $\pm 15\%$.

The present correlations also were compared against the data ($4000 < Re < 30\,000$) from the one micro-fin tube tested by Al-Fahed et al. [15] and the two micro-fin tubes tested by Chiou et al. [7]. The friction factor data of Al-Fahed et al. were underpredicted by about 25%; those of Chiou et al. were underpredicted by about 16% and 4%. These differences might be explained by their fin's triangular profiles being different than the generally rectangular profiles used in the present study as well as difficulty estimating the fin widths since they were not given. The Nu numbers from both of those studies were predicted to within $\pm 10\%$.

4. Conclusions

The parametric effects of fin geometry on turbulent friction factors and Nusselt numbers in internally finned tubes have been described. Trends in the data are different depending on whether the tube is a 'high' fin or a 'micro' fin tube. The criterion for labeling a tube a micro-fin tube is characterized by its peculiar pressure drop behavior with long lasting transitional flow up to $Re \cong 20\,000$. Correlations from the literature poorly predict the present data.

New correlations for friction factors and Nusselt numbers have been developed for these two types of tubes. For high-fin tubes, the friction factor is calculated with equations (9) and (10) [with equation (3) for the smooth tube f], and Nu is calculated with equation (14) and (15) [with equation (6) for the smooth tube Nu]. For the micro-fin tubes, the friction factor is calculated with equations (9), (12) and (13), and Nu is calculated with equation (14) and (16). These correlations are applicable to a wide range of geometric and flow conditions in high- and micro-fin tubes and estimate well the present data and data from the literature. Because of their format, designers should be able to use these correlations to more easily select internally finned tubes for use in heat exchangers.

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