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Influencing factors on flow boiling of carbon dioxide in enhanced tubes and comparison with correlations

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Abstract. Carbon dioxide two-phase flow characteristics are different from those of conventional refrigerants, due to the CO_2 particular thermodynamic and transport properties obtained by working at high reduced pressures. Moreover, the use of peculiar heat transfer surfaces such as grooves and internal fins are often preferred to enhance the boiling heat transfer performance. This paper collects CO_2 flow boiling heat transfer coefficient data from different independent databases available in scientific literature, regarding both smooth and enhanced geometries and a wide range of operative conditions, that are typical of refrigeration systems and heat pumps. The database for enhanced tubes covers internal diameters from 0.8 to 8.92 mm, saturation temperatures from -30 to +20 °C, imposed heat fluxes from 1.67 to 60 kW/m^2 and mass velocities from 75 to 800 kg/m^2s, collecting more than 800 points. Heat transfer data for smooth and enhanced surfaces under the same conditions are collected, in order to measure the enhancement and to correlate it to the geometry augmentation. The assessment of quoted prediction methods explicitly developed for carbon dioxide is finally carried out, with a proposal for a correction factor.

1. Introduction

Although the regular use of carbon dioxide as refrigerant (namely CO_2 or R744 according to the ASHRAE classification) is reported since the earliest era of vapor compression cycles, the subsequent production of synthetic CFCs, that granted chemical stability and lower operating pressures, temporarily replaced CO_2 among other natural fluids. In modern times, however, due to the increasing concern for the anthropological global warming, carbon dioxide is receiving a renewed attention as potential alternative working fluid for several low-temperature applications, such as commercial refrigeration systems, cascade cycles and transcritical ejector plants. Moreover, regarding its safety characteristics, it is non-toxic (up to moderate-high concentrations) and also non-combustible, whereas its accidental release is relatively easy to manage, since the liquid phase will either evaporate or become solid and left to evaporate in a short amount of time. However, one of its flaws is its high operating pressure (up to 120 bar in transcritical cycles), requiring a complete redesign of components and stronger materials, such as stainless-steel or reinforced K65 copper. The change of the thermal conductivity of the tubes may, in turn, significantly penalize the overall conductance UA of fin and tube heat exchangers, also due to an increased fin pitch for manufacturing demands. For this reason, enhanced surfaces (grooves and microfins) for boiling heat transfer are often proposed in order to provide for the general drop of performance.

Regarding the carbon dioxide thermodynamic and transport properties, they are substantially atypical when compared to other refrigerants. Its critical temperature and pressure are respectively 31.1 °C and 73.8 bar, thus CO_2 evaporates at much higher reduced pressures (0.61 at 10 °C) than other halogenated fluids (for instance 0.10 for R134a). As a consequence, carbon dioxide presents the lowest liquid-to-vapor density ratio, leading to smaller differences between the liquid and the vapor phase velocities than for other substances, implying less interface shear and a more homogeneous two-phase flow [1]. The high vapor density of CO_2 leads to an enhanced volumetric refrigeration capacity, whereas the very low surface tension (6.4 mN/m at -10 °C) influences two-phase flow and nucleate boiling characteristics: smaller surface tension means lower superheat required for bubble nucleation and growth, thus increasing the heat transfer efficiency. On the other hand, it tends to destabilize the liquid surface, with increased droplet formation and entrainment, which may negatively affect the heat transfer efficiency when convective boiling is dominant, possibly causing premature onset of dryout. As regards its transport properties, carbon dioxide has a liquid conductivity (0.13 W/mK at -10 °C) higher than that of typical refrigerants (with the exception of ammonia and R32), whereas it presents the lowest liquid viscosity and a very low liquid-to-vapor viscosity ratio (respectively 120 μPa s and 8.5 at -10 °C), thus having favourably small two-phase pressure drops during evaporation.

Due to these peculiar characteristics, the heat transfer coefficients of carbon dioxide during flow boiling are not so easily predictable if compared to conventional refrigerants at the same conditions, especially in case of enhanced surfaces that may affect the heat transfer performance. In fact, the presence of fins increases from one side the heat transfer surface, at the same time promoting the liquid turbulence and avoiding the interface stripping, thus possibly delaying the dryout occurrence. For these reasons, one of the main objectives of this work is to collect CO_2 flow boiling data for smooth and microfin tubes, by paying particular attention to the values obtained at the same operating conditions, in order to isolate the effect of the heat transfer surface enhancement ratio from the other contributions. Specific carbon dioxide prediction methods for two-phase heat transfer coefficient are then implemented and modified by taking into account the geometric enhancement ratio, providing a statistical analysis.

2. Experimental data for smooth and enhanced surfaces

2.1. Presentation of the database

A summary of the collected data for smooth and enhanced surfaces is shown in Table 1. This research has highlighted 9 independent works in microfinned surfaces using pure carbon dioxide as refrigerant, covering circular and horizontal channels having internal diameters from 0.8 to 8.92 mm, with mass fluxes from 75 to 800 kg/m^{2}s, saturation temperatures from -30 to +20 °C and heat fluxes from 1.67 to 60 kW/m^2. By excluding the works of Jeong and Park [2] and of Wu et al. [3] performed in multi-minichannel geometries, all the collected data refer to single circular and horizontal channels. For most of the papers, the authors presented also the corresponding heat transfer coefficient data in smooth tubes at the same operating conditions, whose number is shown in the last column.

2.2. Effect of the microfin structure on convective heat transfer

The flow boiling heat transfer coefficients in a microfin surface are generally higher than those evaluated in a smooth surface under the same operating conditions. Cho and Kim [4] stated that this behavior can be attributed to the increased turbulence of the liquid phase, that is forced to follow the path of the grooves. This usually avoids stratified flows and the early appearance of dryout; at the same time, it increases the convective contribution to the heat transfer mechanism, by anticipating the onset of the annular flow regime. Carbon dioxide is particularly sensitive to the presence of fins, since its very low surface tension implies an unstable liquid film and therefore an earlier occurrence of dryout with respect to other fluids working at lower reduced pressures. Figure 1 (a) shows the flow boiling heat transfer coefficients of Dang et al. [5] for their microfin and smooth 2.0 mm tubes, at a mass flux of 720 kg/m^{2}s, an imposed heat flux of 9 kW/m^2 and a saturation temperature of 15 °C.

Table 1. Collected data of flow boiling of CO_2 in smooth and enhanced tubes.

	Geometry*	**d [mm]**	**G [kg m^{-2} s^{-1}]**	**q [kWm^{-2}]**	**T_{sat} [°C]**	**Collected data for microfin/ smooth tubes**
Schael and Kind [6]	S,C,H H_{fin}=0.25mm Fin angle=18° Fins=60	8.62	75/500	3.9/61	5	30/26
Cho et al. [4] [7]	S,C,H H_{fin}=0.15mm Fin angle=18° Fins=60	4.4/8.92	212/656	6/30	-5/20	200/200
Gao et al. [8]	S,C,H H_{fin}=0.11mm Fin angle=12° Fins=40	3.04	190/770	10/30	10	71/13
Jeong and Park [2]	M,C,H H_{fin}=0.1mm width=0.2	0.8	400/800	12/18	0/10	51/51
Dang et al. [5]	S,C,H H_{fin}=0.12mm Fin angle=6.3° Fins=40	2	360/720	4.5/18	15	185/0
Ono et al. [9]	S,C,H H_{fin}=0.11mm Fin angle=12° Fins=50	3.75	190/380	10/30	10	58/61
Zhao and Bansal [10]	S,C,H H_{fin}=0.20mm Fin angle=18° Fins=50	7.31	100/250	9.9/30	-30	82/0
Wu et al. [3]	M,C,H H_{fin}=0.16mm Fin angle=0° Fins=13	1.7	100/600	1.67/8.33	1/15	195/62
Overall		**0.8/8.92**	**75/800**	**1.67/61**	**-30/20**	**872/413**

**M=multi, S=single; C=circular, R=rectangular; H=horizontal, V=vertical*

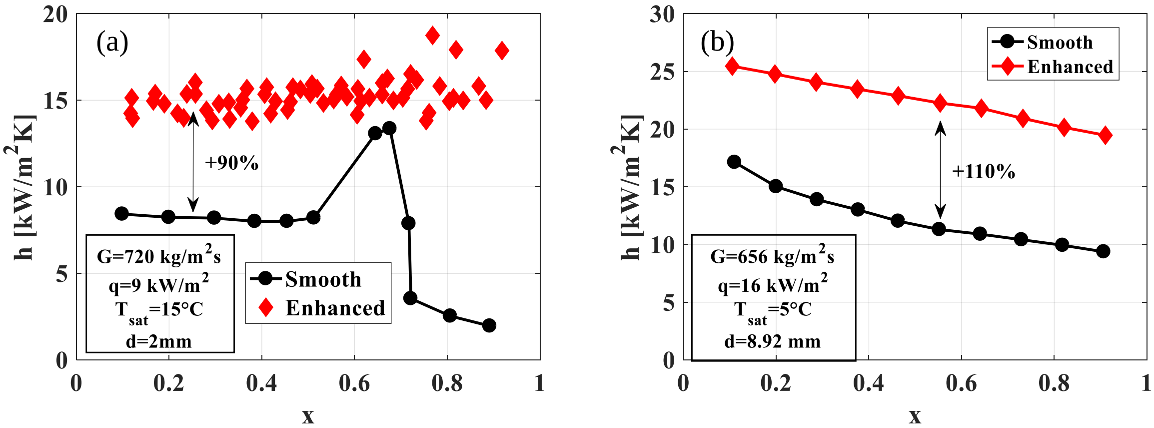

Figure 1. Effect of the microfin structure on the flow boiling heat transfer coefficient of carbon dioxide. (a) Data from Dang et al. [5]; (b) Data from Cho et al. [4].

Besides higher values of the heat transfer coefficients, the smooth tube approaches the dryout condition at a vapor quality of approximately 0.65, whereas the microfin tube still shows a solid constant trend. For a larger diameter of 8.92 mm, Figure 1 (b) presents the heat transfer coefficient data of Cho and Kim [4], in which the trend with vapor quality is not affected by the presence of fins, but the values are approximately 110% higher for the enhanced surface.

It is worth noting that the original data reduction proposed by the authors leads to an inevitable error on the estimation of the effective heat transfer performance. In fact, the experimental heat flux is usually referred to an equivalent smooth tube, whereas a fair comparison when calculating the heat transfer coefficient should consider the enhancement of the heat transfer surface and the fin efficiency. The geometric enhancement ratio can be calculated as a function of the fin tip angle γ, the fin helix angle β, the fin height H_{fin}, the number of fins N_{fin} and the equivalent internal diameter of the smooth tube d_0 as follows:

$$\frac{A}{A_0} \cong 1 + \frac{2N_{fin}H_{fin}}{\pi d_0 \cos\beta}\left(\frac{1}{\cos\gamma} + \tan\gamma\right) \tag{1}$$

The fin efficiency can also be expressed as a function of the fin height H_{fin} and the fin width B_{fin}. For a rectangular fin, it can be evaluated as follows:

$$\eta_{fin} = \left(1 + \frac{1}{3}\left(\sqrt{\frac{2h}{k_{fin}B_{fin}}} \cdot H_{fin}\right)^2\right)^{-1} \tag{2}$$

The collected heat transfer coefficients for microfin tubes can be therefore normalized to the same heat transfer surface of the corresponding smooth tube:

$$h_{enh,norm} = \frac{h_{enh}}{\eta_{fin} A/A_0} \tag{3}$$

Figure 2 (a) shows the ratio between microfin and smooth heat transfer coefficients obtained at the same operating conditions and with the original data reduction implemented by the authors, as a function of the vapor quality. By normalizing the enhanced values according to equation (3), the updated ratio is presented in Figure 2 (b). It is clear that most of the gap in the heat transfer coefficients can be compensated (in a ±50% band) by only adjusting the data reduction with the effective heat transfer surface. For vapor qualities higher than 0.2, it can be seen still a positive effect of the microfin structure, which is independent on the surface enhancement ratio and can be attributed to a possible dryout occurrence in the smooth tube, which is instead delayed in the presence of fins. From this analysis, it seems that besides a greater heat transfer surface, there are not clear further effects of the microfin structure on the convective heat transfer, whose normalized efficiency prior to dryout is somehow similar to that obtained in conventional smooth tubes. This is probably due to the strong nucleate boiling contribution of CO_2 with respect to the convective heat transfer for the operating conditions explored in this paper (reduced pressures from 0.19 to 0.78, mass fluxes from 75 to 800 kg/m^{2}s and diameters from 0.8 to 8.92 mm). Further hydrodynamic effects, such as the increase of turbulence for the liquid film may therefore have a secondary role on the heat transfer mechanism.

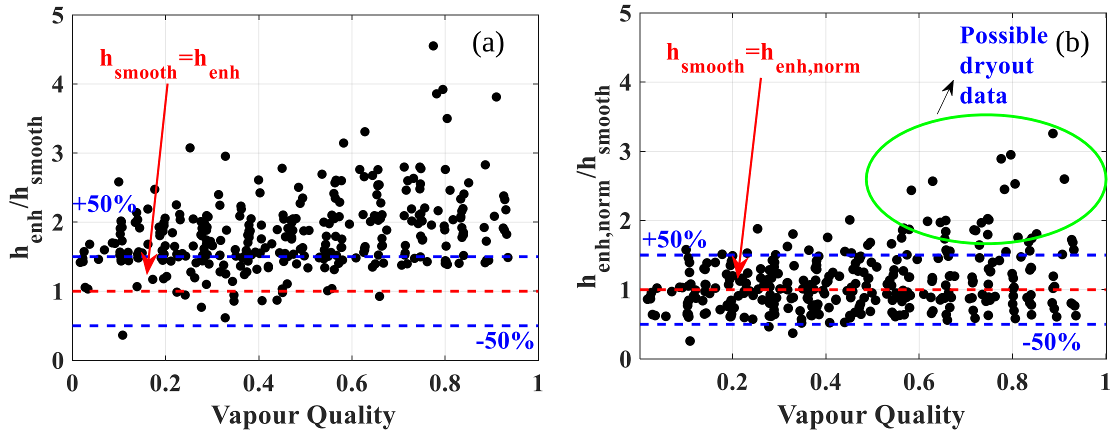

Figure 2. Comparison between smooth and enhanced heat transfer coefficients vs vapor quality (a) as they are; (b) by taking into account the heat transfer surface enhancement ratio.

3. Assessment of predictive methods

From our research, the only flow boiling heat transfer prediction method developed for microfin tubes and carbon dioxide as working fluid is that of Wu et al. [3], conceived for a multi-minichannel geometry and 0° helix angle. Before the dryout region, the heat transfer coefficient is a function of the fin height and reads as:

$$h_{Wu,x<x_{cr}} = \left[C_1 Bo^{C_2} \left(\frac{P_{sat} d}{\sigma} \right)^{C_3} + C_4 \left(1/X_{tt} \right)^{C_5} \left(\frac{G H_{fin}}{\mu_L} \right)^{C_6} \right] \cdot Re_{LO}^{C_7} Pr_L^{C_8} \left(\frac{\delta}{H_{fin}} \right)^{C_9} \cdot h_L \tag{4}$$

In the above equation, Bo is the Boiling number, X_{tt} is the Martinelli parameter, δ is the liquid film thickness (evaluated in the hypothesis of symmetric annular flow and employing the void fraction model of Rohuani and Axelsson [11]), h_L is the liquid Dittus-Boelter heat transfer coefficient and H_{fin} is the fin height. The parameters from C_1 to C_9 represent fitting coefficients for the authors' own experimental data. The dryout vapor quality was correlated instead to the liquid Reynolds, Boiling and Bond numbers and to the ratios of liquid and vapor density and viscosity as follows, by using from E_1 to E_9:

$$x_{cr} = E_1 Re_L^{E_2} Bo^{E_3} Bd^{E_4} \left(\frac{\rho_V}{\rho_L} \right)^{E_5} \left(\frac{\mu_L}{\mu_V} \right)^{E_6} \tag{5}$$

Different flow boiling heat transfer coefficient prediction methods are instead available for carbon dioxide in smooth tubes and for this paper we proposed the assessment of three independent models. The flow pattern based method of Thome and El Hajal [12] has been developed for a carbon dioxide database including 404 points and covering five tube diameters from 0.79 to 10.06 mm, mass velocities from 85 to 1440 kg/m^{2}s, heat fluxes from 5 to 36 kW/m^2 and saturation temperatures from -25 to 25 °C. The authors kept the same structure of the previous flow pattern based model of Kattan et al. [13] and isolated the nucleate boiling contribution from the overall heat transfer performance of carbon dioxide, observing that it was larger than expected, due to the high operating reduced pressures. On this regard, the predicted nucleate boiling heat transfer coefficient was modified as follows, in which the original nucleate boiling expression is taken from Cooper correlation [14]:

$$h_{NB,Thome} = 0.71 h_{NB} + 3970 \tag{6}$$

Later on, Cheng et al. [15] also proposed a similar flow pattern based method for the evaluation of the flow boiling heat transfer coefficient of carbon dioxide, by covering all flow regimes within a wide range of operating conditions. According to the authors collected database, the model is applicable to tube diameters from 0.6 to 10 mm, mass velocities from 50 to 1500 kg/m^{2}s, heat fluxes from 1.8 to 46 kW/m^2 and saturation temperatures from -28 to 25 °C. For the sake of simplicity and briefness, the complete mathematical expression of this model is not provided here and can be instead obtained from the reference mentioned [15]. Equation (7) shows the updated nucleate boiling contribution, for which also in this case a modification of the Cooper expression was proposed:

$$h_{NB,Cheng} = 131 P_{red}^{-0.0063} \left(-\log_{10} P_{red}\right)^{-0.55} M^{-0.5} q^{0.58} \tag{7}$$

Finally, Oh et al. [16] developed a superposition model as shown in equation (8):

$$h_{Oh} = S h_{nb} + E h_L \tag{8}$$

in which the pool boiling heat transfer coefficient was taken from the Cooper expression and the liquid heat transfer coefficient was instead calculated with different models according to the magnitude of the liquid Reynolds number. The suppression factor S was related to the Boiling number and a two-phase multiplier Φ, which is in turn a function of the Martinelli parameter X and the Chisholm parameter C, set to 20, 12, 10 and 5 in case of liquid-vapor flow condition of turbulent-turbulent, laminar-turbulent, turbulent-laminar and laminar-laminar respectively:

$$S = 0.279 \left(\phi^2\right)^{-0.029} Bo^{-0.098} \tag{9}$$

$$E = \max\left[\left(0.023\phi^{2.2} + 0.76\right), 1\right] \tag{10}$$

$$\phi^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{11}$$

The authors calibrated their model with their own database including smooth tubes with internal diameters from 0.5 to 3.0 mm, mass fluxes from 200 to 600 kg/m^{2}s, heat fluxes from 5 to 40 kW/m^2, saturation temperatures from 0 to 20 °C. Besides carbon dioxide, this correlation was also originally developed with R22, R410A, R134a and propane.

Both smooth and enhanced heat transfer coefficient collected data are compared with the predicted values from the chosen correlations. The model of Wu et al. [3] can only be applied to microfin structures, since the position H_{fin}=0 leads to an inconsistent physics. Although this method includes the effect of the enhanced channel characteristics through the fin height, it largely overestimates the collected data, working reasonably well only with the authors' own database. The remaining three correlations developed for smooth tubes are instead also considered for the enhanced channels. Firstly, they are implemented as they are (see Figure 3 (a) and (c)), and then they are modified by multiplying the predicted heat transfer coefficient to the parameters η_{fin} and A_{enh}/A_0 obtained in equations (1) and (2), thus including the geometric enhancement ratio (see Figure 3 (b) and (d)). The complete assessment with the evaluation of Mean Absolute Error (*MAE*) and the percentage of data points falling in a ±30% (δ_{30}) and ±50% (δ_{50}) error range is provided in Table 2. The heat transfer coefficients for smooth tubes are fairly predicted with all the chosen three correlations. The flow pattern based model of Thome and El-Hajal [12] provides a Mean Absolute Error (*MAE*) of 44% when used with the smooth database, whereas the second flow pattern based method of Cheng et al. [15] gives an overall *MAE* of 42%. Better results are surprisingly obtained with the simpler superposition correlation of Oh et al. [16], with almost 52% of the data included in a ±30% error range. The heat transfer coefficients of the microfin tubes database are generally underpredicted by all the three correlations for smooth tubes when used as they are. When the geometry enhancement ratio is taken into account, the agreement substantially improves, becoming similar to that of the smooth tubes. On this regard, the best results are obtained with the modified flow pattern based method of Cheng et al., for which the updated *MAE* is 40%, with 51% of the collected heat transfer coefficients in enhanced tubes falling in a ±30% error range. In this method, the recognized dryout and mist flow data are treated as annular flow, according to the observed trends in Figure 1 (a).

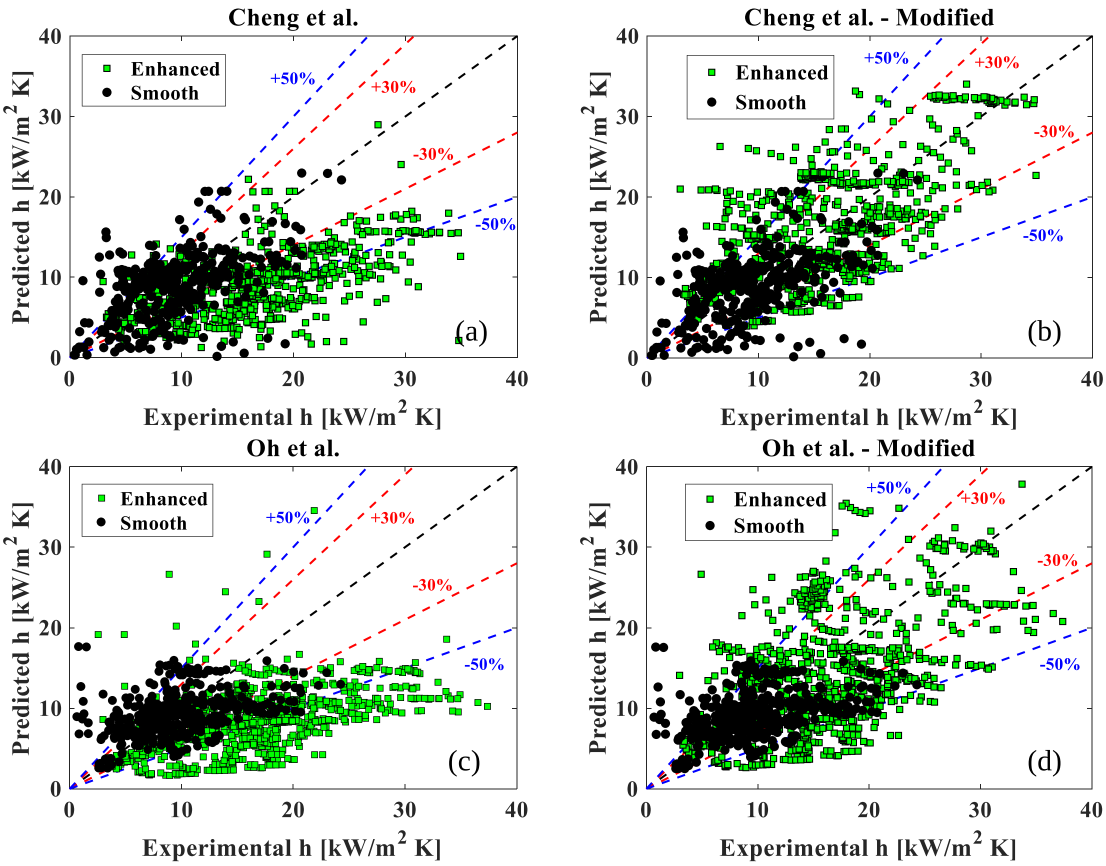

Figure 3. Assessment of the chosen prediction methods with the smooth and the enhanced CO_2 flow boiling database. (a) Correlation of Cheng et al. [15] as it is; (b) Modified Cheng et al.; (c) Correlation of Oh et al. [16] as it is; (d) Modified Oh et al.

Table 2. Assessment of original and modified correlations with the smooth and enhanced database.

	Wu et al.	**Thome-El Hajal**	**Modified Thome-El Hajal**	**Cheng et al.**	**Modified Cheng et al.**	**Oh et al.**	**Modified Oh et al.**
Smooth database	-	MAE=44% δ_{30}=38% δ_{50}=81%	-	MAE=42% δ_{30}=58% δ_{50}=81%	-	MAE=18% δ_{30}=52% δ_{50}=80%	-
Enhanced database	MAE=163% δ_{30}=22% δ_{50}=32%	MAE=57% δ_{30}=10% δ_{50}=31%	MAE=46% δ_{30}=37% δ_{50}=68%	MAE=45% δ_{30}=23% δ_{50}=58%	MAE=40% δ_{30}=51% δ_{50}=74%	MAE=55% δ_{30}=22% δ_{50}=48%	MAE=45% δ_{30}=41% δ_{50}=69%

4. Conclusions

A series of data for flow boiling heat transfer of carbon dioxide in smooth and microfin tubes having different geometries and operating conditions is reviewed and the assessment of some existing correlations explicitly developed for CO_2 is carried out. The main outcomes of this study can be summarized as follows:

- A total amount of 872 heat transfer coefficients in microfin tubes were collected. Approximately the half of the entire database is also provided in the same operating conditions for the corresponding smooth tubes for comparison purposes.
- The presence of fins delays the onset of dryout, which usually occurs earlier for carbon dioxide due to its very low surface tension effect on the top of the tube. Moreover, the heat transfer coefficient is significantly increased with respect to a smooth tube, and this behavior is justified by only considering the effective heat transfer surface in the data reduction process.
- The superposition correlation of Oh et al. best fits the smooth tubes data (*MAE*=18%), whereas the heat transfer coefficients for enhanced surfaces, as they are, are significantly underpredicted by the three prediction methods for smooth tubes implemented (*MAE*>45%).
- Within the range of operating conditions investigated for this paper (reduced pressures from 0.19 to 0.78, mass fluxes from 75 to 800 kg/m^{2}s and diameters from 0.8 to 8.92 mm), by modifying the chosen methods with the introduction of the effective heat transfer surface, their accuracy increases considerably. The corrected flow pattern based correlation of Cheng et al. best fits all the enhanced collected database, with a calculated *MAE* of 40% and more than half of the experimental points falling in an error range of ±30%.

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