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COMPARISON OF 30 BOILING AND CONDENSATION CORRELATIONS FOR TWO-PHASE FLOWS IN COMPACT PLATE-FIN HEAT EXCHANGERS

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ABSTRACT

Over the years, empirical correlations have been developed for predicting saturated flow boiling [1-15] and condensation [16-30] heat transfer coefficients inside horizontal/vertical tubes or micro-channels. In the present work, we have examined 30 of these models, and modified many of them for use in compact plate-fin heat exchangers. However, the various correlations, which have been developed for pipes and ducts, have been modified in our work to make them applicable to extended fin surfaces. The various correlations have been used in a low-order, one-dimensional, finite-volume type numerical integration of the flow and heat transfer equations in heat exchangers. The NIST's REFPROP database [31] is used to account for the large variations in the fluid thermo-physical properties during phase change. The numerical results are compared with Yara's experimental data [32]. The validity of the various boiling and condensation models for a real plate-fin heat exchanger design is discussed. The results show that some of the modified boiling and condensation correlations can provide acceptable prediction of heat transfer coefficient for two-phase flows in compact plate-fin heat exchangers.

INTRODUCTION

In recent years, increased efforts have been devoted to the design of high performance compact heat exchangers due to the increasing heating and cooling requirements in various thermal systems. Among the many enhanced heat transfer techniques, working fluid with phase change by boiling or condensation can provide large heat fluxes, even with relatively small driving temperature differences. Therefore, two phase flows are widely used in the design of heat exchangers. Many efforts have been devoted to understanding the basic phase-change in the past and to develop models to predict heat transfer coefficients [33-36]. Multiple correlations for fluid boiling or condensation heat transfer have been proposed in the literature, most of them having been developed empirically. However, as for the design of the compact heat exchangers with two phase flows, the available boiling and condensation correlations cannot be used directly since these correlations were originally developed for two-phase flows in horizontal/vertical smooth tubes or, recently, for flows in micro-channels. The accuracy of these correlations for two-phase flows in compact heat exchangers need to be tested due to more complicated flow passage geometries and the existence of extended surfaces such as the fins in the plate-fin heat exchangers.

In this study, 15 boiling and 15 condensation correlations taken from the literature are chosen to predict the boiling and condensation heat transfer in compact plate-fin heat exchangers. The existing models developed have been modified to account for the effects of fins. The proposed modified models are then used to predict the heat transfer for R22 boiling and condensation inside a compact plate-fin heat exchanger with serrated fins. The numerical results are compared with Yara's experimental data, so that the validity of these modified models can be assessed.

BOILING CORRELATIONS

Many correlations have been proposed for predicting the heat transfer coefficient in situations where a liquid boils. The 15 boiling correlations [1-15] tested in this study are summarized in Table 1. For use in a plate-fin heat exchanger, the existing boiling correlations need to be modified so that the effects of the fins to both nucleate and convective boiling can be taken into account. Several studies [37, 38] have been carried out to study the boiling phenomena in a compact plate-fin evaporator. It was found that, on one hand, similar to single phase flow, the existence of the fins may lead to larger Reynolds number so that the convective boiling can be enhanced, On the other hand, nucleate boiling can also be suppressed due to larger shear stresses near the wall [37].

Therefore, a smooth tube-based boiling correlation with the form

$$\alpha_{TP,boil} = Fun\left(\alpha_{nb}, \alpha_{cb}\right) \tag{1}$$

can be modified to

$$\left(\alpha_{TP,boil}\right)_{Fin} = Fun\left(S_{nb}\alpha_{nb}, E_{cb}\alpha_{cb}\right) \tag{2}$$

for boiling in plate-fin heat exchangers. Here, $\alpha_{TP,boil}$ is the original boiling correlation, α_{nb} and α_{cb} are the nucleate boiling and convective boiling components for $\alpha_{TP,boil}$, respectively. $Fun(\ )$ is any boiling correlations listed in Table 1 (at the back of this paper). In the modified correlation, S_{nb} is the suppression factor for the nucleate boiling and E_{cb} is the augmentation factor for the convective boiling due to the existence of the fins. The unknown factors S_{nb} and E_{cb} depend on flow passage geometry and should be measured from experiments. However, very few experimental data is available for liquid boiling in plate-fin heat exchangers.

For the small flow passages in a compact plate-fin heat exchanger, convective boiling is enhanced by the presence of extended surfaces, on which a liquid film is formed. The heat is first transferred from the solid to liquid film by conduction and convection, followed by vaporization at the liquid/vapor interface. This is analogous to the case of pure single phase liquid flowing through the same passage but in which enhanced heat transfer is also due to the fact the fins provide extra surfaces for heat transfer between solid and liquid. On the foregoing basis, the assumption is made that

$$E_{cb} = \frac{\left(\alpha_l\right)_{Fin}}{\left(\alpha_l\right)_{Dittus-Boelter}}\ , \tag{3}$$

where $\left(\alpha_l\right)_{Fin}$ is the single phase boiling correlations for the liquid phase of fluid when it flows into a given plate-fin heat exchanger, and $\left(\alpha_l\right)_{Dittus-Boelter}$ is the Dittus-Boelter correlation commonly used for single phase flows in a smooth tube:

$$\alpha_{Dittus-Boelter} = \frac{\lambda}{D_h} 0.023 \,\mathrm{Re}_D^{0.8} \,\mathrm{Pr}^{0.4}\ . \tag{4}$$

Note that the assumption is made in Eqn. (3) that convective boiling is controlled by the heat transfer between the solid surface and the liquid film. The model is attractive for its simplicity and the fact that single phase empirical heat transfer correlations are available for many types of fins. The model assumes that even when a liquid is boiling in a plate-fin heat exchanger the majority of the heat transfer enhancement comes from the change of the flow patterns (turbulence, destruction of boundary layer, etc.) due to the existence of the fins. This assumption needs to be validated with experimental data.

The existence of fins may significantly suppress nucleate boiling. The physics behind this phenomenon needs to be understood. However, nucleate boiling may not significantly affect heat transfer when the flow Reynolds number is large, Robertson and Lovegrove [39] measured the boiling of R11 on a serrated fin and Kandlikar [37] suggests $S_{NB} = 0.77$ for it. In this work, we also assume this value.

CONDENSATION CORRELATIONS

Similar to the boiling correlations, we have also chosen 15 condensation correlations originally developed for smooth tubes to investigate if they can be used for flows in a compact plate fin heat exchanger. These models are summarized in Table 2 (at the back of this paper). It can be seen that these models are mostly developed for annular or stratified flows. Two types of condensation modes are important: gravity controlled film condensation and shear controlled condensation. Therefore, these smooth tube-based condensation correlations can be written in the general form

$$\alpha_{TP,cond} = Fun\left(\alpha_{gra}, \alpha_{sh}\right), \tag{5}$$

where $\alpha_{TP,cond}$ is the original condensation correlation, α_{gra} and α_{sh} are the gravity controlled condensation and shear controlled condensation components of $\alpha_{TP,cond}$, respectively.

Gravity controlled film condensation will be enhanced/suppressed by the local surface profile. A concave surface tends to enhance condensation while a convex surface suppresses it [40]. The overall effects of the fins for the gravity controlled film condensation are not well understood. In this study, we assume there is no change in the gravity part of film condensation (α_{gra}) for finned surface:

$$\left(\alpha_{gra}\right)_{Fin} \approx \left(\alpha_{gra}\right)_{Smooth-Tube}\ . \tag{6}$$

For shear-controlled film condensation, based on the classical Nusselt film theory [41], we have

$$\alpha_{sh} \propto \phi_l = \sqrt{\frac{\left(dp/dL\right)_{TP}}{\left(dp/dL\right)_l}}\ , \tag{7}$$

where the parameter ϕ_l can be computed from the Lockhart and Martinelli model. The effects of the fins in the compact heat exchanger for the shear-controlled condensation may then be modeled as

$$\frac{\left(\alpha_{sh}\right)_{Fin}}{\left(\alpha_{sh}\right)_{Smooth-Tube}} = \frac{\left(\phi_l\right)_{Fin}}{\left(\phi_l\right)_{Smooth-Tube}}\ . \tag{8}$$

For smooth tubes, Blasius solutions can be used to calculate the single phase friction factor in $\left(\phi_l\right)_{Smooth-Tube}$:

$$f_{\text{Blasius}} = \begin{cases} 0.079\,\text{Re}_D^{-1/4} & \text{for } \text{Re}_D \le 10^4 \\ 0.046\,\text{Re}_D^{-1/5} & \text{for } \text{Re}_D > 10^4 \end{cases} \quad (9)$$

Substituting $(\alpha_{sh})_{Fin}$ and $(\alpha_{gra})_{Fin}$ into Eqn. (4), we can obtain the modified condensation correlations for condensation in a compact plate-fin heat exchanger as:

$$(\alpha_{TP,cond})_{Fin} = Fun\left((\alpha_{gra})_{Fin}, (\alpha_{sh})_{Fin}\right). \quad (10)$$

PROBLEM DESCRIPTION

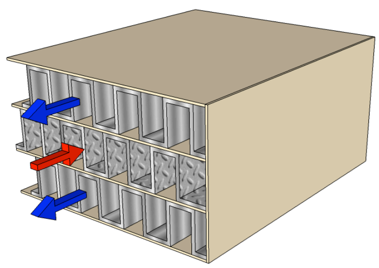

Figure 1Yara's Compact Plate-Fin Evaporator/Condensor Configuration.

The proposed models (Eqns. (2) and (10)) need to be tested. In this study, Yara's [32] compact plate-fin evaporator/condenser configurations are used for this purpose; this is shown in Fig. 1. The same geometry are used for both boiling and condensation measurements. This plate-fin heat exchanger has 1 refrigerant passage and 2 water passages. Rectangular serrated fin is used for the refrigerant flow and plain rectangular fins are used in water flows. The geometry of the heat exchanger is defined in Table 3.

Table 3 HEX Geometry

Geometry	R22	Water
Fin Type	Serrated	Plain
Fin Spacing [mm]:	1.478	2.297
Plate Spacing [mm]:	6.35	6.35
Fin Thickness [mm]:	0.203	0.305
Fin Offset Pitch [mm]	3.2	N/A
Plate Width/Passage Width [m]:	0.17	0.17
Plate Length/Passage Length [m]:	1.32	1.32
Number of Rows/Passages:	1	2
Number of Passes:	1	1
Fin Thermal Conductivity: [W/(m.K)]	168.0	168.0

In evaporator mode, the cold flow of R22 is heated by the hot water from both sides and the flow conditions are given in Table 4. In condenser mode, the hot flow of R22 is cooled by the cold water from both sides and the flow conditions are given in Table 5.

Table 4 Flow Conditions for Evaporator Test

Cold Fluid	R22
Inlet Temperature:	$T_{inlet} = 279.6\ K$
Inlet Pressure:	$P_{inlet} = 0.611\times10^6$ Pa
Inlet Quality:	$x_{inlet} = 0.2$
Mass Flux:	$G = 99.5$ kg/(m$^2\cdot$s)
Outlet Temperature:	$T_{outlet} \approx 286.5\ K$
Outlet Pressure:	$P_{outlet} = 0.575\times10^6$ Pa
Outlet Quality:	$x_{inlet} = 1.0$ (super-heating)
Hot Fluid	**Water**
Inlet Temperature:	$T_{inlet} = 296\ K$
Inlet Pressure:	$P_{inlet} = 1.01325\times10^5$ Pa
Inlet Quality:	$x_{inlet} = 0$
Mass Flux:	$G = 153.9$ kg/(m$^2\cdot$s)
Outlet Temperature:	$T_{outlet} \approx 286.5\ K$

Table 5. Flow Conditions for Condensor Test

Hot Fluid	R22
Inlet Temperature:	$T_{inlet} = 332.5\ K$
Inlet Pressure:	$P_{inlet} = 1.674\times10^6$ Pa
Inlet Quality:	$x_{inlet} = 1.0$
Mass Flux:	$G = 99.5$ kg/(m$^2\cdot$s)
Outlet Temperature:	$T_{outlet} \approx 316.9\ K$
Outlet Pressure:	$P_{outlet} = 1.664\times10^6$ Pa
Outlet Quality:	$x_{inlet} = 0.0$ (sub-cooling)
Cold Fluid	**Water**
Inlet Temperature:	$T_{inlet} = 301.5\ K$
Inlet Pressure:	$P_{inlet} = 1.01325\times10^5$ Pa
Inlet Quality:	$x_{inlet} = 0$
Mass Flux:	$G = 161.0$ kg/(m$^2\cdot$s)
Outlet Temperature:	$T_{outlet} \approx 314.1\ K$

PLATE-FIN MODELING

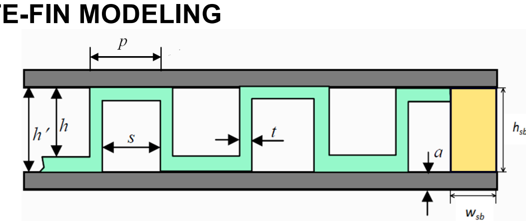

Figure 2 Illustration of Serrated Fin Geometry

The geometry of serrated fin is illustrated in Fig. 2. Manglik and Bergles' correlations [41] are used to predict the single phase heat transfer coefficient and pressure drop:

$$j_{MB}(Re) = 0.6522 Re^{-0.5403} \alpha^{-0.1541} \delta^{0.1499} \gamma^{-0.0678} \times \left(1 + 5.269 \times 10^{-5} Re^{1.34} \alpha^{0.504} \delta^{0.456} \gamma^{-1.055}\right)^{0.1} \quad (11)$$

$$f_{MB}(Re) = 9.6243 Re^{-0.7422} \alpha^{-0.1856} \delta^{0.3053} \gamma^{-0.2659} \times \left(1 + 7.699 \times 10^{-8} Re^{4.429} \alpha^{0.92} \delta^{3.767} \gamma^{0.236}\right)^{0.1} \quad (12)$$

$$\alpha = \frac{s}{h}, \quad \delta = \frac{t}{l}, \quad \gamma = \frac{t}{s}, \quad (13)$$

where s is fin width, h is fin height, t is fin thickness and l is the fin offset pitch. The hydraulic diameter is defined as

$$D_h = \frac{4hsl}{2(sl + hl + th) + ts}. \quad (14)$$

$(\alpha_l)_{Fin}$ in Eqn. (3) is then calculated by

$$(\alpha_l)_{Fin} = \frac{\lambda_l}{D_h} j_{MB}(\mathrm{Re}_l) \mathrm{Re}_l \mathrm{Pr}_l^{1/3}. \quad (15)$$

$(\phi_L)_{Fin}$ in Eqn. (8) is calculated by

$$(\phi_L^2)_{Fin} = 1 + \frac{C_{tt}}{(X_{tt})_{Fin}} + \frac{1}{(X_{tt})_{Fin}^2}, \quad (16)$$

$$(X_{tt})_{Fin} = \sqrt{\left(\frac{dP}{dL}\right)_{l,FIN} \Big/ \left(\frac{dP}{dL}\right)_{g,FIN}}, \quad (17)$$

$$\left(\frac{dP}{dL}\right)_{l,\mathrm{Fin}} = 2 f_{MB}(\mathrm{Re}_l) \frac{G^2}{\rho_l D_h} (1-x)^2, \quad (18)$$

$$\left(\frac{dP}{dL}\right)_{g,\mathrm{Fin}} = 2 f_{MB}(\mathrm{Re}_g) \frac{G^2}{\rho_g D_h} x^2, \quad (19)$$

where

$$\mathrm{Re}_l = \frac{G D_h (1-x)}{\mu_l}, \quad (20)$$

$$\mathrm{Re}_g = \frac{G D_h x}{\mu_g}. \quad (21)$$

The mean temperature difference between the fin surfaces and saturated stream can be estimated as $\eta_F \Delta T_{sat}$. The mean heat transfer coefficient across the overall heat transfer area can be calculated as [37]

$$\overline{\alpha_{TP,\eta}} = \left[\alpha_{TP,P} + \alpha_{TP,F}(A_F / A_P)\eta_F\right] / (1 + A_F / A_P) \quad (22)$$

where A_F is the fin surface area, A_P is the prime surface area (heat transfer area not covered by fins), $\alpha_{TP,P}$ is calculated from boiling/condensation models by using ΔT_{sat} as the driving temperature difference for phase change, and $\alpha_{TP,F}$ is calculated from two phase models by using $\eta_F \Delta T_{sat}$ as the driving temperature difference for phase change. The fin efficiency for the serrated fin is calculated by

$$\eta_F = \frac{\tanh\left(\sqrt{\frac{2\overline{\alpha_{TP,\eta}}}{\lambda_s} \frac{(t+l)}{tl}} \frac{h}{2}\right)}{\sqrt{\frac{2\alpha_f}{\lambda_s} \frac{(t+l)}{tl}} \frac{h}{2}} \quad (23)$$

It can be seen that $\overline{\alpha_{TP,\eta}}$ requires η_F in Eqn. (22) while η_F requires $\overline{\alpha_{TP,\eta}}$ in Eqn. (23). Therefore, an iterative scheme is required to solve Eqns. (22) and (23). Our study shows that the values of $\overline{\alpha_{TP,\eta}}$ and η_F will converge in less than 10 steps of iterations with the initial guess of η_F set to 0.5. With $\overline{\alpha_{TP,\eta}}$ solved, the averaged heat flux based on the total heat transfer area can be calculated as

$$q'' = \overline{\alpha_{TP,\eta}} \Delta T_{sat}. \quad (24)$$

NUMERICAL PROCEDURE

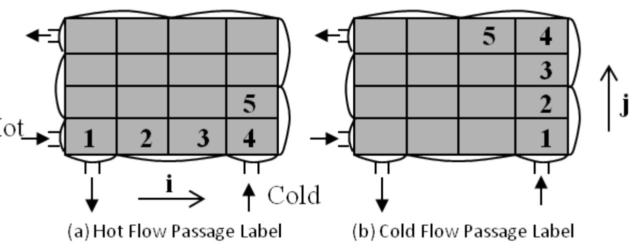

Figure 3. Illustration of fluid segments discretized along flow path

The modified boiling and condensation models have been implemented in the Thermal Analysis Software, INSTED. In this program, the flow passages are divided into small sections (Fig. 3). The calculation tracks the flow from the inlet to the outlet of a fluid stream as the fluid goes through the passage. For instance, for every section along the cold fluid passage, there is an exchange of heat with the hot fluid. However, to exchange the heat between the streams, the temperature of the fluid in the other stream must be determined. This can only be done when the equations are solved on the segments of the other stream. Therefore, an iterative scheme is needed to solve the both streams. The procedure called "Incremental Method" is implemented and summarized in Fig. 4. Details of the numerical procedure can be found in our previous work [42, 46].

By discretizing the flow path into sections, the local heat transfer coefficient can be calculated by Eqn. (2) for boiling or Eqn. (8) for condensation. Also, to account for the large variations in thermos-physical properties, NIST's REFPROP database has been integrated into the INSTED program.

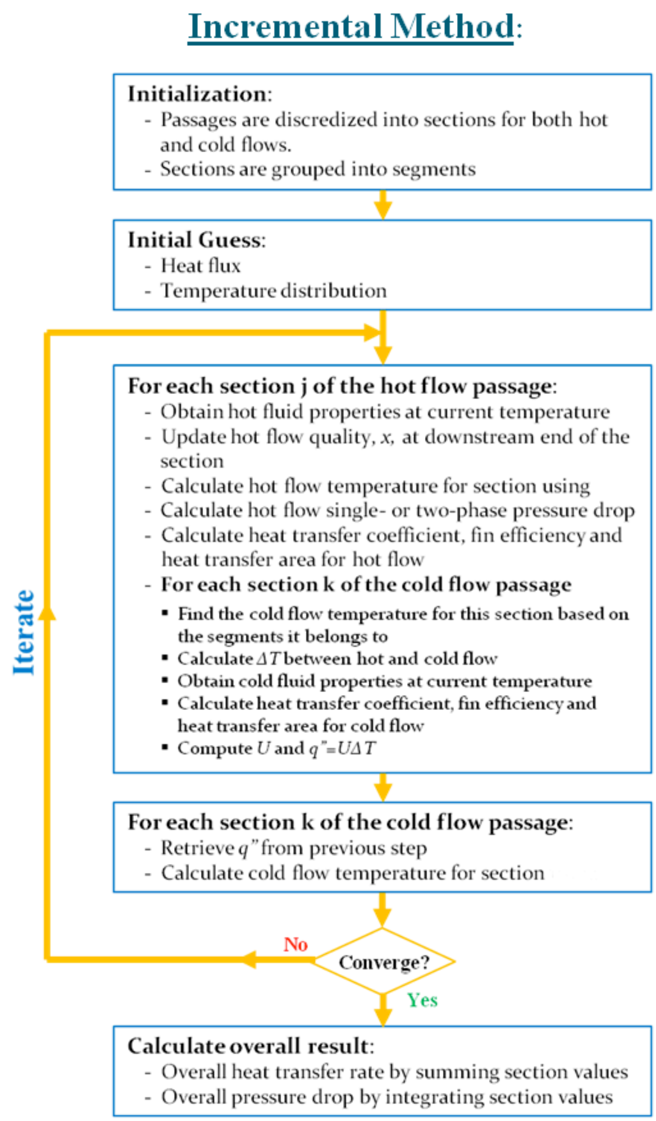

Figure 4 Flow chart of the incremental iteration procedure in INSTED Thermal Analysis Software [42]

RESULTS & DISSCUSSION

The boiling calculation results are shown in Fig.5 where the flow quality (x) versus the heat transfer coefficient (α_{TP}) curves are plotted for all the 15 modified boiling correlations. Results from both the original and proposed modified boiling models are shown. The numerical predictions are compared with experimental data. The mean absolute relative errors (MARE) are also given in Table 1. The results show that the "nucleate boiling only" models (Rohsenow [9], Cooper [10], Tran [12], Kew and Cornwell [13], Warrier [14]) give the most error. This agrees with the fact that for this test, convective boiling is dominant, which favors the models which can predict convective boiling: Chen [1], Gungor and Winterton [3-4], Kandlikar [5], Liu and Winterton [6], and Steiner and Taborek [7]. It can be seen that the modified models provide more accurate results compared to the original models. The MAREs of the modified models are around 20%, which is acceptable considering the possible errors coming from the original single phase correlations. Therefore, Eqn. (3) seems to give a good estimation of E_{CB} in this case. Note that these models appear to over-predict the two-phase heat transfer coefficient when $x > 0.6$, which is due to the fact that the assumption made in Eqn. (3) becomes invalid when the fluid approaches the gaseous state (quality approaching 1).

Similarly, the calculated results for the 15 modified condensation correlations are shown in Fig.6, together with the results from the original models and Yara's experimental data. The mean absolute errors (MARE) of these models are also given in Table 2. For this calculation, as stated in Yara's paper, the condensation is gravity-controlled. It can be observed that the "stratified-flow only" model (Jaster-Kosky [19]), "gravity + shear" (Fujii [22], Haraguchi [21], and Yu [23]) models, and "multi-regime" models (Dobson [25], Thome [28], Cavallini [27, 29], Shah [30]) gives relatively smaller errors compared with "annular flow only" models (Kosky and Staub [17], Cavallini and Zechin [18], Shah [20], Moser [24]). Since the condensation is gravity-controlled, the scaling factor proposed in Eqn. (8) needs to be assessed. To validate the proposed scaling factor on the shear terms in these models, the numerical results are again compared with the experimental data, where shear- controlled condensation dominates. In Fig. 6, we can see that, with the exception of the models by Carpenter and Colburn [16] and Webb [26], most of the original condensation models can still provide fair agreement with the experimental data, suggesting that the assumption made on the gravity-controlled condensation in the proposed models - where we assume that the effects of the fin surface on the gravity-controlled condensation is negligible - is probably reasonable.

CONCLUSION

In this study, 30 boiling and condensation correlations developed for smooth tubes have been modified and tested for use in a compact plate-fin heat exchanger. The test results show that the modified correlations can provide acceptable results. However, more experimental data is needed to further validate the proposed models.

REFERENCES

1. Chen, John C. "Correlation for boiling heat transfer to saturated fluids in convective flow." Industrial & engineering chemistry process design and development 5.3 (1966): 322-329.
2. Shah, M. Mohammed. "Chart correlation for saturated boiling heat transfer: equations and further study." ASHRAE Trans.;(United States) 88.CONF-820112- (1982).

3. Gungor, Kernal Ersin, and R. H. S. Winterton. "A general correlation for flow boiling in tubes and annuli." International Journal of Heat and Mass Transfer 29.3 (1986): 351-358.
4. Gungor, K. E., and R. H. S. Winterton. "Simplified general correlation for saturated flow boiling and comparisons of correlations with data." Chemical engineering research & design 65.2 (1987): 148-156.
5. Kandlikar, Satish G. "A general correlation for saturated two-phase flow boiling heat transfer inside horizontal and vertical tubes." Journal of heat transfer 112.1 (1990): 219-228.
6. Liu, Z., and R. H. S. Winterton. "A general correlation for saturated and subcooled flow boiling in tubes and annuli, based on a nucleate pool boiling equation." International journal of heat and mass transfer 34.11 (1991): 2759-2766.
7. Steiner, Dieter, and Jerry Taborek. "Flow boiling heat transfer in vertical tubes correlated by an asymptotic model." Heat transfer engineering 13.2 (1992): 43-69.
8. Kattan, Nakhlé, J. R. Thome, and D. Favrat. "Flow boiling in horizontal tubes: Part 1—Development of a diabatic two-phase flow pattern map." Journal of Heat Transfer 120.1 (1998): 140-147.
9. Rohsenow, Warren M. A method of correlating heat transfer data for surface boiling of liquids. Cambridge, Mass.: MIT Division of Industrial Corporation,[1951], 1951
10. Cooper, M. G. "Saturation nucleate pool boiling—a simple correlation." Inst. Chem. Eng. Symp. Ser. Vol. 86. No. 2. 1984.
11. Koyama, Sh, et al. "Forced convective flow boiling heat transfer of pure refrigerants inside a horizontal microfin tube." Proceedings of the convective flow boiling (1995): 137-142.
12. Tran, T. N., M. W. Wambsganss, and D. M. France. "Small circular-and rectangular-channel boiling with two refrigerants." International Journal of Multiphase Flow 22.3 (1996): 485-498.
13. Kew, Peter A., and Keith Cornwell. "Correlations for the prediction of boiling heat transfer in small-diameter channels." Applied Thermal Engineering 17.8 (1997): 705-715.
14. Warrier, Gopinath R., Vijay K. Dhir, and Leslie A. Momoda. "Heat transfer and pressure drop in narrow rectangular channels." Experimental Thermal and Fluid Science 26.1 (2002): 53-64.
15. Yu, W., et al. "Two-phase pressure drop, boiling heat transfer, and critical heat flux to water in a small-diameter horizontal tube." International Journal of Multiphase Flow 28.6 (2002): 927-941.
16. Carpenter, Frank G., and Allan Philip Colburn. The effect of vapor velocity on condensation inside tubes. University of Delaware, 1951.
17. Kosky, Philip G., and Fred W. Staub. "Local condensing heat transfer coefficients in the annular flow regime." AIChE Journal 17.5 (1971): 1037-1043.
18. Cavallini, Alberto, and Roberto Zecchin. "A dimensionless correlation for heat transfer in forced convection condensation." Proceedings of the Sixth International Heat Transfer Conference. Vol. 3. 1974.
19. Jaster, H., and P. G. Kosky. "Condensation heat transfer in a mixed flow regime." International Journal of Heat and Mass Transfer 19.1 (1976): 95-99.
20. Shah, M. M. "A general correlation for heat transfer during film condensation inside pipes." International Journal of heat and mass transfer 22.4 (1979): 547-556.
21. Haraguchi, H., Koyama, S., and Fujii, T., "Condensation of Refrigerants HCFC22, HFC134a, and HCFC123 in a Horizontal Smooth Tube (2nd Report, Proposal of Empirical Expressions for Local Heat Transfer Coefficient)," Transactions JSME, vol. 60, no. 574, pp. 245–252, 1994. In Japanese.
22. Fujii, Tetsu. "Enhancement to condensing heat transfer-new developments." Journal of Enhanced Heat Transfer 2.1-2 (1995).
23. Yu, J., and S. Koyama. "Condensation heat transfer of pure refrigerants in microfin tubes." (1998).
24. Moser, K. W., R. L. Webb, and B. Na. "A new equivalent Reynolds number model for condensation in smooth tubes." Journal of Heat Transfer 120.2 (1998): 410-417.
25. Dobson, M. K., and J. C. Chato. "Condensation in smooth horizontal tubes." Journal of Heat Transfer 120.1 (1998): 193-213.
26. Webb, Ralph L., Ming Zhang, and R. Narayanamurthy. "Condensation heat transfer in small diameter tubes." Heat Transfer 6 (1998): 403-408.
27. Cavallini, Alberto, et al. "Condensation of halogenated refrigerants inside smooth tubes." HVAC&R Research 8.4 (2002): 429-451.
28. Thome, John R., J. El Hajal, and A. Cavallini. "Condensation in horizontal tubes, part 2: new heat transfer model based on flow regimes." International Journal of Heat and Mass Transfer 46.18 (2003): 3365-3387.
29. Cavallini, Alberto, et al. "Condensation in horizontal smooth tubes: a new heat transfer model for heat exchanger design." Heat Transfer Engineering 27.8 (2006): 31-38.
30. Shah, M. Mohammed. "An improved and extended general correlation for heat transfer during condensation in plain tubes." Hvac&R Research 15.5 (2009): 889-913.
31. Lemmon, Eric W., Marcia L. Huber, and Mark O. McLinden. "NIST reference fluid thermodynamic and transport properties–REFPROP." (2002).
32. Yara, Tomoyasu, Shigeru Koyama, and Tetsu Fujii. "Condensation and Evaporation of Refrigerants in a Plate Fin Heat Exchanger." NASA 19990008999 (1994): 61-69.
33. Qu, Weilin, and Issam Mudawar. "Flow boiling heat transfer in two-phase micro-channel heat sinks—I. Experimental investigation and assessment of correlation methods." International Journal of Heat and Mass Transfer 46.15 (2003): 2755-2771.

34. Qu, Weilin, and Issam Mudawar. "Flow boiling heat transfer in two-phase micro-channel heat sinks—II. Annular two-phase flow model." International Journal of Heat and Mass Transfer 46.15 (2003): 2773-2784.
35. García-Valladares, O. "Review of in-tube condensation heat transfer correlations for smooth and microfin tubes." Heat Transfer Engineering 24.4 (2003): 6-24.
36. Awad, M. M., A. S. Dalkilic, and S. Wongwises. "A Critical Review on Condensation Heat Transfer in Microchannels and Minichannels." Journal of Nanotechnology in Engineering and Medicine 5.1 (2014): 010904.
37. Kandlikar, S. G. "A model for correlating flow boiling heat transfer in augmented tubes and compact evaporators." Journal of heat transfer 113.4 (1991): 966-972.
38. Cohen, Mo, and V. P. Carry. "A comparison of the flow boiling performance characteristics of partially-heated cross-ribbed channels with different rib geometries." International journal of heat and mass transfer 32.12 (1989): 2459-2474.
39. Robertson, J. M., and P. C. Lovegrove. "Boiling heat transfer with Freon 11 (R11) in brazed aluminum, plate-fin heat exchangers." J. Heat Transfer 105 (1983): 605-610.
40. Kandlikar, Satish G. Handbook of phase change: boiling and condensation. CRC Press, 1999.
41. Bergles, RM Manglik AE. "The thermal-hydraulic design of the rectangular offset-strip-fin compact heat exchanger." Compact Heat Exchangers: A Festschrift for AL London (1990): 123.
42. Ladeinde, Foluso, and Kehinde Alabi. "A new procedure for two-phase thermal analysis of multi-pass industrial plate-fin heat exchangers." Compact Heat Exchangers and Enhancement Technology for the Process Industries, Davos, Switzerland, July (2001): 1-6.
43. Kandlikar, Satish, et al. Heat transfer and fluid flow in minichannels and microchannels. elsevier, 2005
44. Panitsidis, Haralampus, R. D. Gresham, and J. W. Westwater. "Boiling of liquids in a compact plate-fin heat exchanger." International Journal of Heat and Mass Transfer 18, no. 1 (1975): 37-42.
45. Watel, Barbara. "Review of saturated flow boiling in small passages of compact heat-exchangers." International Journal of Thermal Sciences 42, no. 2 (2003): 107-140.
46 Ladeinde, F., & Alabi, K. 2003. A New Procedure for Two-Phase Analysis of Industrial Heat Exchangers. *Int. J. Heat Exchangers* Vol. IV (1), pp. 71-90.

NOMENCLATURE

A	=	area, m^2
Bo	=	boiling number, $\equiv \dfrac{q''}{h_{lg}G}$
c_p	=	specific heat, $J/(kg\cdot K)$
Co	=	convection number, $\equiv \left[\dfrac{1-x}{x}\right]^{0.8}\left(\dfrac{\rho_g}{\rho_l}\right)^{0.5}$
D_h	=	hydraulic diameter, m
E	=	augmentaion factor
f	=	friction factor
Fr	=	Froude number, $\equiv \dfrac{G^2}{\rho^2 g D_h}$
g	=	gravitational constant, $m^3/(kg\cdot s^2)$
G	=	mass flux, $kg/(m^2\cdot s)$
Ga_l	=	Galileo number, $\equiv \dfrac{g\rho_l(\rho_l-\rho_g)D_h^3}{\mu_l^2}$
h_{lg}	=	latent heat, J/kg
j	=	Colburn factor, $\equiv StPr^{2/3}$
Ja_l	=	Jakob number, $\equiv \dfrac{c_{p,l}(T_{sat}-T_w)}{h_{lg}}$
M	=	molecular weight, g/mol
MAE	=	mean absolute error
N_{conf}	=	confinement number, $\equiv \left[\dfrac{\sigma}{g(\rho_l-\rho_g)}\right]^{0.5}/D_h$
Nu	=	Nusselt number, $\equiv \dfrac{\alpha D}{\lambda}$
P	=	pressure, Pa
P_{red}	=	reduced pressure, Pa, $\equiv P/P_{cr}$
ΔP_{sat}	=	change of saturated fluid pressure due to ΔT_{sat}, Pa
Pr	=	Prandtal number
Q	=	heat transfer rate, W
q''	=	heat flux, W/m^2
Re	=	Reynolds number
S	=	suppression factor
St	=	Stanton number, $\equiv \dfrac{\alpha}{Gc_p}$
T	=	temperature, K
T^+	=	dimensionless temperature
ΔT_{sat}	=	temperature difference between wall and saturate fluid temperature, K $\equiv \begin{cases} T_w - T_{sat} & \text{for boiling} \\ T_{sat} - T_w & \text{for condensation} \end{cases}$
We	=	Webber number, $\equiv \dfrac{G^2 D_h}{\rho\sigma}$
x	=	quality
X	=	Martinelli parameter
Y	=	Chishom parameter
α	=	heat transfer coefficient, $W/(m^2\cdot K)$
η_F	=	fin efficiency
μ	=	viscosity, $kg/(m\cdot s)$
ρ	=	density, kg/m^3
λ	=	thermal conductivity, $W/(m\cdot K)$
σ	=	surface tension coefficient, N/m
ϕ	=	two phase frictional multiplier
ε_g	=	void fraction
τ_w	=	wall shear stress, N/m^2
A	=	area, m^2
A	=	area, m^2

Subscript

an	=	annular
cb	=	convective boiling
cr	=	critical state
f	=	fin
g	=	vapor
go	=	with all fluid as vapor
gra	=	gravity controlled
l	=	liquid
lo	=	with all fluid as liquid
nb	=	nucleate boiling
sat	=	saturation
sh	=	shear controlled
str	=	stratified
TP	=	two phase flow
w	=	wall

Table 1 Selected boiling correlations

No.	Correlation	Channel Geometry	Boiling Mechanism	Fluids	MARE
1	Chen (1966)	Horizontal tubes	Nucleate boiling and forced convective boiling	Water, Methanol, Pentane, Heptane, Benzene, etc.	17.77%
2	Shah (1982)	Horizontal and vertical tubes D_h=5.0-15.8 mm	Nucleate boiling and forced convective boiling	R11, R12, R22, R502, etc.	21.77%
3	Gungor and Winterton (1986)	Horizontal and vertical tubes D_h=2.95-32 mm	Nucleate boiling and forced convective boiling	Water, R11, R12, R113, etc.	19.40%
4	Gungor and Winterton (1987)	Horizontal and vertical tubes D_h=2.95-32 mm	Nucleate boiling and forced convective boiling	Water, R11, R12, R113, etc.	18.28%
5	Kandlikar (1990)	Horizontal and vertical tubes D_h=4.6-32 mm	Nucleate boiling and forced convective boiling	Water, R11, R12, R22, R113, Nitrogen, etc.	23.08%
6	Liu and Winterton (1991)	Horizontal and vertical tubes D_h=2.95-32 mm	Nucleate boiling and forced convective boiling	Water and refrigerants	23.80%
7	Steiner and Taborek (1992)	Horizontal tubes D_h=1-32 mm	Nucleate boiling and forced convective boiling	Water, refrigerants, cryogenics	15.30%
8	Kattan (1998)	Microfin tube	Nucleate boiling and forced convective boiling	R134a, R123, R402a, R404a, R502	36.51%
9	Rohsenow (1951)	Horizontal tubes	Nucleate boiling	Water, CCl4, Benzene, n-Pentane, Ethyl alcohol, etc.	49.93%
10	Cooper (1984)	Pool boiling	Nucleate boiling	Water, refrigerants, organic fluids, cryogens	92.71%
11	Koyama (1995)	Microfin tube	Nucleate boiling	Refrigerants	25.00%
12	Tran (1996)	Horizontal tubes D_h=2.4-2.92 mm	Nucleate boiling	R12, R113	90.90%
13	Kew and Cornwell (1997)	Horizontal tubes D_h=1.39-3.69 mm	Nucleate boiling	R141b	92.57%
14	Warrier (2002)	Horizontal tubes D_h=0.75 mm	Nucleate boiling	FC-84	> 95%
15	Yu (2002)	Horizontal tubes D_h=2.98 mm	Nucleate boiling (moderate convective boiling maybe included)	Water	38.64%

Table 2 Selected condensation correlations

No.	Correlation	Channel Geometry	Condensation Regime	Fluids	MARE
1	Carpenter and Colburn (1951)	Horizontal tubes	Annular flow	Steam	18.62%
2	Kosky and Staub (1971)	Horizontal tubes	Annular flow	Steam	31.98%
3	Cavallini and Zechin (1974)	Horizontal tubes	Annular flow	Steam	47.24%
4	Jaster and Kosky (1976)	Horizontal tubes D_h=12.5 mm	Stratified flow	Steam	21.27%
5	Shah (1979)	Horizontal tubes D_h=7-40 mm	Annular flow	Water, R11, R12, R22, R113, methanol, ethanol, benzene, etc.	33.72%
6	Haraguchi (1994)	Horizontal tubes D_h=8.4 mm	Annular flow	R22, R134a, R123	17.53%
7	Fujii (1995)	Horizontal tubes D_h=8.4 mm	Gravity and shear flows	R22, R134a, R123	14.75%
8	Yu and Koyama (1998)	Microfin tubes	Gravity and shear flows	R22, R134a, R123	20.66%
9	Moser (1998)	Horizontal tubes D_h=3.14-20 mm	Annular flow	Steam	30.45%
10	Dobson and Chato (1998)	Horizontal tubes D_h=3.14-7.04 mm	Annular and stratified-wavy flows	R12, R22, R134a, etc.	25.13%
11	Webb (1998)	Horizontal tubes D_h=1-7 mm	Annular flow	R12	18.33%
12	Cavallini (2002)	Horizontal tubes D_h=8 mm	Annular, annular-stratified, and stratified-slug flows	R22, R134a, R125, R236ea, R32, R410A	19.99%
13	Thome (2003)	Horizontal tubes D_h=8 mm	Annular, stratified-wavy, and wavy flows	R22, R134a, R125, R236ea, R32, R410A	8.95%
14	Cavallini (2006)	Horizontal tubes D_h=8 mm	ΔT-dependent and ΔT-independent flows	R22, R134a, R125, R236ea, R32, R410A	19.99%
15	Shah (2009)	Horizontal/vertical tubes	Laminar, transitional, and turbulent flows	Water, halocarbon refrigerants, hydrocarbon refrigerants, and organics	26.29%

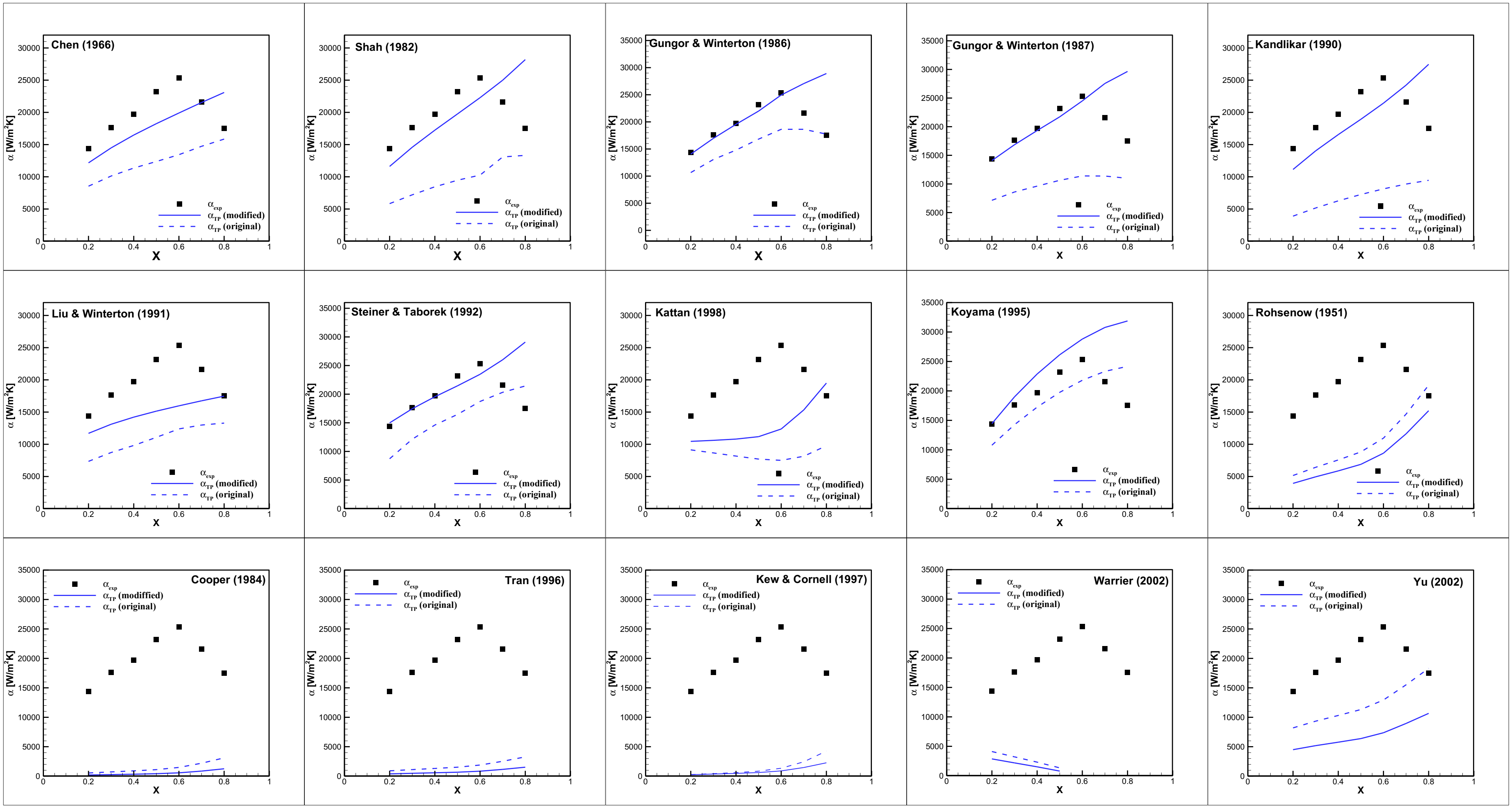

Figure 5. Numerical quality v.s. heat transfer coefficient plots for various boiling correlations with comparison of Yara's experimental data.

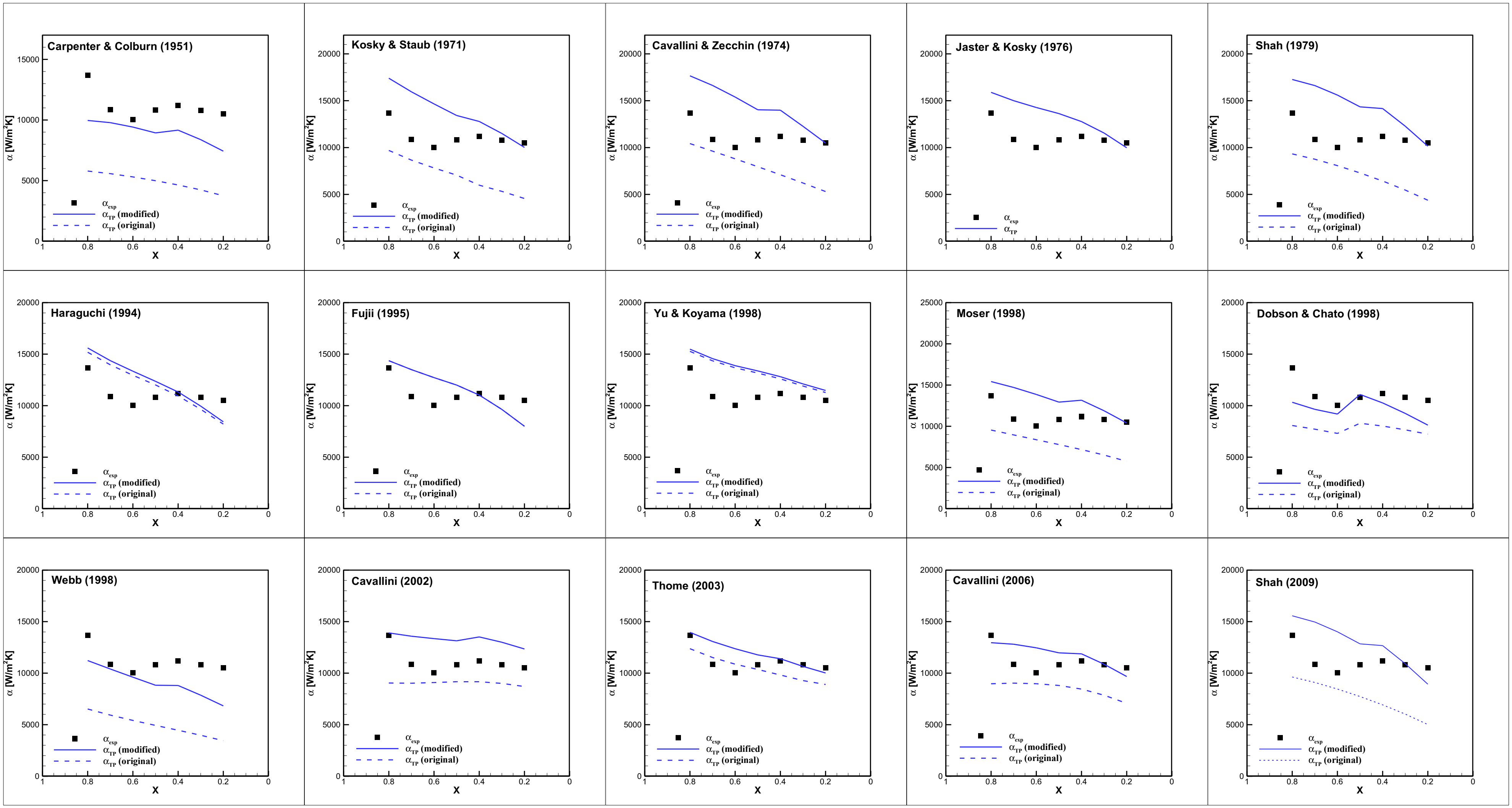

Figure 6. Numerical quality v.s. heat transfer coefficient plots for various condensation correlations with comparison of Yara's experimental data