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DETERMINATION OF R134A'S CONVECTIVE HEAT TRANSFER COEFFICIENT IN HORIZONTAL EVAPORATORS HAVING SMOOTH AND CORRUGATED TUBES

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Abstract- Boiling heat transfer of R134a flowing in the inner tube and hot water flowing in the annulus were examined inside smooth and corrugated tube-in-tube heat exchangers in authors' previous studies experimentally. The apparatus has 0.85 m long double tube for horizontal orientation as its test section whose inner ones are smooth and corrugated copper tubing having inner diameters of 0.0081 m. The range of mass fluxes are 300, 400 and 500 kg m^{-2}s^{-1}, the average vapor qualities vary from 0.09-0.81, and saturation pressure interval is between 4.5 and 5.7 bar. In this existing study, the predictability of almost whole well-known empirical correlations in literature suggested for convective boiling flows in smooth and corrugated tubes are experienced by using authors' a large number of experimental database (227 data points for horizontal tubes). Characteristics of trend lines plotted for the change of vapor quality with experimental boiling heat transfer coefficients are examined as well in relation to various saturation temperatures and mass fluxes. Furthermore, the most successful correlations are validated their predictabilities for horizontal oriented evaporators having smooth and corrugated tubes.

Keywords: Boiling, Evaporation, R134a, Heat transfer coefficient, Corrugated tube

I. Introduction

Two-phase enhanced tubes and channels have been shown up recently as a leading way out for thermal managing of related applications. This idea brought many advantages such as large thermal capacity and low thermal resistance between the heat source and the cooler. Additionally, boiling flows demand a smaller amount of coolant flow rate and pumping power to remove a heat load comparing to single-phase flows. Thus, cooling devices having these kind of enhanced geometries can enable enhancements in temperatures' uniformity, and take up a little volume possibly.

Many studies about fundamental issues and experiments have been performed to determine the characteristics of convective boiling flow in enhanced tubes and channels. According to the previous studies in the literature, boiling phenomena in these geometries has different characteristics regarding in boiling transition, flow regimes and heat transfer features. Some of the leading studies on the subject are summarized in the following paragraphs. It should be noted that empirical correlations of these researchers have been used for the comparison part of the current study.

Liu and Winterton [1] emphasized the importance of generation of vapor in the boundary layer next to the wall regarding with the forced convective heat transfer mechanisms. According to their approach, the new correlation should have Prandtl number in order to have a success in nucleate boiling, because they stated that existing boiling number prevents application to subcooled boiling. They were also able to show the independency from the boiling number on the predictability issue.

Gungor and Winterton [2] have used 4300 data points of 28 researchers to develop a reliable correlation for forced convection boiling of water, refrigerants and ethylene glycol in vertical and horizontal tubes, and also annuli. They showed that their correlation has a better predictability than others in the literature. Their mean deviation value was 21.4% for saturated boiling and 25.0% for subcooled boiling. They continued to their previous study and prepared a new one [3]. They made a comparison using existing correlations with their ones and they indicated the necessity on a new correlation for low flow rates in horizontal tubes.

Shah [4] developed almost the most compared correlations in papers on in-tube boiling in open sources. He tested his correlation's reliability with his data and others in the study. Most of them can be considered as agreeable as a result of his work. He also stated that his one of previous correlations had some limitations on the predictability regarding with annuli' gap size. He added the parametric effects of density ratio, quality and boiling number into his previously published analyses. He was able to test his correlation with 3000 data points for 12 fluids.

Kandlikar [5] developed a correlation on the flow boiling of 10 refrigerants inside horizontal and vertical tubes. He focused on nucleate boiling and convective mechanisms. He used 5246 data points from 24 experimental studies for the validation of his correlation. His average success rates were 15.9% for water data and 18.8% for all refrigerants. R-22 and R-113 data were the most incompatible ones among others.

Kenning and Cooper [6] investigated the flow boiling of water under convection or by nucleate boiling regimes inside bore vertical tubes. They stated the agreeable correlations with specific flow regimes such as annular flow, plug/chum flow, and nucleate boiling, they also discussed the sensitives of the flow regimes related with the parameters. Besides, they also tested the general flow boiling correlations' suitable features.
Kew and Cornwell [7] studied R141b's flow boiling in single, small diameter tubes to reveal the energy-saving potential of compact heat exchangers. His developed correlation has been found to be successful for the largest diameter tube and unsuccessful for the smaller diameter tubes. They stated that heat transfer coefficient increased with increasing quality, and gave some suggestions for the intermittent dry-out regime.

Lazarek and Black [8] carried out some boiling experiments using R-l 13 flowing inside vertical, co-current upflow and downflow configured tubes in order to calculate the local heat transfer coefficient, pressure drop, and critical heat flux. Their correlation was developed based on Nusselt number having the dimensionless numbers of liquid Reynolds and boiling. They also predicted the pressure drop related with frictional, spatial acceleration and 180 bends successfully. Moreover, they discussed the effect of mass velocity, inlet subcooling and heated length on their new critical heat flux correlation's vapor quality result at dryout regime.

Sun and Mishima [9] tested 13 well-known correlations' validity range using 2505 boiling data points for 11 refrigerants. They focused on the flow boiling heat transfer in mini-channels and proposed a modified correlation based on another one published previously. They reported that some of the correlations could not reflect the expected trend line drawn for the variation of heat transfer coefficient with vapor quality.
Akhavan-Behabadi and Esmailpour [10] performed experimental works of R-134a's boiling flow inside a corrugated tube for 7 different tube inclination angles. They showed that boiling heat transfer coefficient at low vapor quality and specific mass fluxes has been affected by the tube inclination angle importantly. They stated that there was an increase on heat transfer coefficient at low vapor qualities and all mass fluxes for the +90° inclined tube as 62% more than that of the −90° inclined tube. In addition, they also proposed an empirical correlation depending on the tube inclination angle for the boiling inside corrugated tubes.

Wongsa-ngam et al. [11] researched evaporation of R-134a in smooth and micro-fin tubes in order to calculate two-phase heat transfer coefficient and pressure drop at high mass flux conditions. According to their experimental work, heat transfer coefficients of smooth and micro-fin tubes increase with increasing quality, mass flux, and evaporating temperature. Besides, pressure drops of both tubes increase with increasing quality and mass flux, but they decrease with increasing evaporating temperature. Moreover, they also proposed new correlations belonging to the tested smooth and micro-fin tubes.

Laohalertdecha et al. [12] correlated new equations on boiling heat transfer coefficient and two-phase friction factor of R-134a flowing inside horizontal corrugated tubes having various corrugation pitches, corrugation depths and similar inner diameters with the smooth tube. They obtained 200 data points for five different corrugated tube geometries from their experimental investigation. Their proposed correlation has been developed on Nusselt number including equivalent Reynolds number, Prandtl number, and corrugated tube's parameters regarding its geometry.

The local fluid pressure is considered as one of the most significant parameters characterizing the two-phase flow in enhanced tubes and channels. There aren't enough empirical correlations for two-phase flows' heat transfer coefficients and friction factors due to the generalization problem. In order to determine the thermal characteristics of the two-phase flow in these geometries, an appropriate usage of the models and correlations regarding with the heat transfer coefficients and friction factors are necessary due to the fact that an important alteration in properties is occurred during two-phase flows in channels.

In the current study, well-known in-tube boiling correlations, which are 10 for smooth tubes and 2 for corrugated tubes, are compared with the experimental data derived through the studies of authors' previous publications [12, 13] on the prediction of the boiling heat transfer coefficient. The examined empirical equations' success are also examined by their change with vapor quality, saturation temperature, mass flux and heat flux. Consequently, the main aim of this investigation is to show the consistency of the authors' experimental database with well-known correlations having the boiling of R134a flow inside horizontal smooth and corrugated tubes, as shown in Table 1, at the high mass flux region. It should be realized that this work should be evaluated as an extension of authors' previous studies [12, 13] and the experienced correlations' numbers have been increased from 3 for the tubes [12, 13] to 12 investigators, as they can be seen in open sources, with this study by the authors in this work.

II. Experimental setup

A schematic diagram of the test apparatus is shown in Fig. 1. The experimental apparatus was designed to determine the heat transfer coefficient and pressure drop of pure R-134a over the length of the test tube. The test loop consists of a test section, refrigeration loop, cooling water flow loops, sub-cooling loop and relevant instrumentation. The objective of the water loop before the test section is to provide a controlled inlet vapor quality. The second water loop located in the test section provides a controlled heat output from the test section. The sub-cooling loop is used to prevent any two-phase flow condition of the refrigerant occurring before it enters the refrigerant pump.

For the refrigeration circulating loop, liquid refrigerant is discharged by a gear pump regulated by an inverter. The refrigerant passes in series through a filter/dryer, a sight glass tube, flow meter, pre-heater and enters the test section. The inlet vapor quality before entering the test section is controlled by the pre-heater. The preheater is a spiral counter flow heat exchanger that supplies energy to provide a controlled inlet vapor quality for vaporization of the refrigerant. On leaving the test section, the refrigerant is then condensed and sub-cooled by a chilling loop that removes the heat input from the pre-heater and test section and then it returns from the two-phase refrigerant to a sub-cooled state, which later collects in a receiver and eventually returns to the refrigerant pump to complete the cycle.
The test section is a horizontal counter-flow double tube heat exchanger. The dimensions of the heat exchangers is given in Table 1.

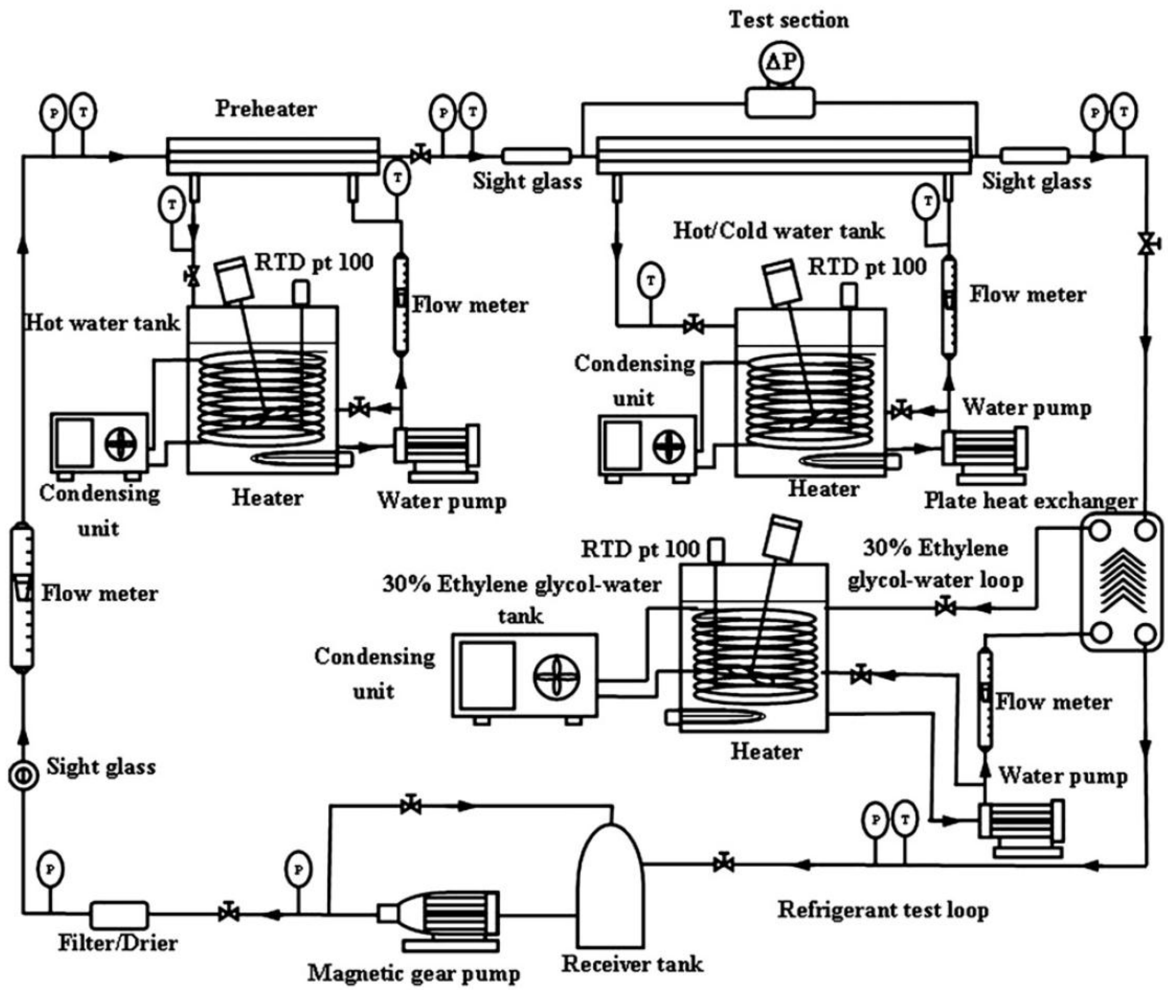


Fig. 1 Schematic view of the experimental setup

The inlet temperature of the water is controlled by a thermostat. A differential pressure transducer and thermocouples are installed in the test section to measure the pressure drop and temperature across the test section. In the present study, the pressure of the refrigerant is mainly controlled by a high-pressure limit switch. High-pressure limit switches (Danfoss Type KP 5) are devices for regulating the pressure of a system so that the pressure is maintained near a desired set point pressure in order to prevent the damage to the apparatus.

The refrigerant temperature and tube wall temperature in the test section are measured by Type T thermocouples. A total of 15 thermocouples are soldered to the top, bottom and at five points along side of the tube. All the temperature measuring devices are well calibrated in a controlled temperature bath using standard precision mercury glass thermometers. The uncertainty of the temperature measurements as recorded by the data acquisition system is ±0.1 °C. All static pressure taps are mounted on the tube wall. The refrigerant flow meter is a variable area type and specially calibrated in the range 0.2-3.4 LPM for R-134a by the manufacturer. Also, the differential pressure transducer is calibrated by the manufacturer.

Experiments were conducted with various flow rates of refrigerant, vapor qualities of refrigerant entering the test section, heat flows and temperatures of refrigerant condensing in the test section. In the experiments, the refrigerant flow rate in the test section was controlled by adjusting the speed of the magnetic gear pump. To vary the vapor quality at the inlet of the test section, the heating water flow and the cooling water flow rates were varied by small increments while the refrigerant flow rate was kept constant. The hot water in the test section was re-circulated by a centrifugal pump to supply heat from the water to the refrigerant. During each experiment, the heat transferred from the test section was kept at a desired value. This might be obtained by simultaneously adjusting and controlling the temperature and flowrate of the hot water entering the test section. The system was allowed to approach a steady state before any data was recorded. After stabilization, the temperature on the tube wall, the temperature of the refrigerant at the locations mentioned above, the inlet and outlet temperatures of the heating water and flow rates of heating water and refrigerant were all recorded. The pressure drop was measured by a differential pressure transducer installed between the inlet and outlet of the test section. The experiments could be carried out by increasing the refrigerant flow rate while the saturation temperature in the test section was kept constant.

All details of experimental setups regarding with explanations, measurement devices, test sections, uncertainties etc. having horizontal [12] and vertical [13] tube test sections, and also calculation of experimental heat transfer coefficient can be seen from authors' previous publications in open sources.

III. Results and Discussion

In this current work, the predictability of practically complete well-known empirical correlations in open sources recommended for convective boiling flows in smooth and corrugated tubes are tested by using authors' experimental database having 227 data points for horizontal tubes.

Fig. 2 represents the variation of experimental convective heat transfer coefficients with average quality for horizontal tubes. Because lower liquid film thickness and higher vapor velocity, heat transfer coefficient increases with increasing average quality.

Table 1 Dimensions of tubes used in the experiments

Tube type	Diameter (m)	Length (m)	Tube depth (mm)	Tube pitch (mm)
Vertical smooth tube	0.0087	0.85	-	-
Vertical corrugated tube 1	0.0087	0.85	0.5	12.7
Vertical corrugated tube 2	0.0087	0.85	0.75	12.7
Vertical corrugated tube 3	0.0087	0.85	1	12.7
Vertical corrugated tube 4	0.0087	0.85	1	8.46
Vertical corrugated tube 5	0.0087	0.85	1	6.35
Horizontal smooth tube	0.0081	2.5	-	-
Horizontal corrugated tube 1	0.0087	2.5	1.5	8.46
Horizontal corrugated tube 2	0.0087	2.5	1.5	6.35
Horizontal corrugated tube 3	0.0087	2.5	1.5	5.08
Horizontal corrugated tube 4	0.0087	2.5	1.25	5.08
Horizontal corrugated tube 5	0.0087	2.5	1	5.08

Table 2 Average proportional errors for smooth vertical and horizontal tubes

Smooth Horizontal Tube									
Gungor and Winterton [2]	Gungor and Winterton [3]	Liu and Winterton [1]	Kandlikar [5]	Shah [4]	Sun and Mishima [9]	Kew and Cornwell [7]	Lazarek and Black [8]	Kenning and Cooper [6]	Wongsagnam et al. [11]
17,06	22,07	36,46	46,04	25,25	36,03	51,31	54,80	21,02	20,88

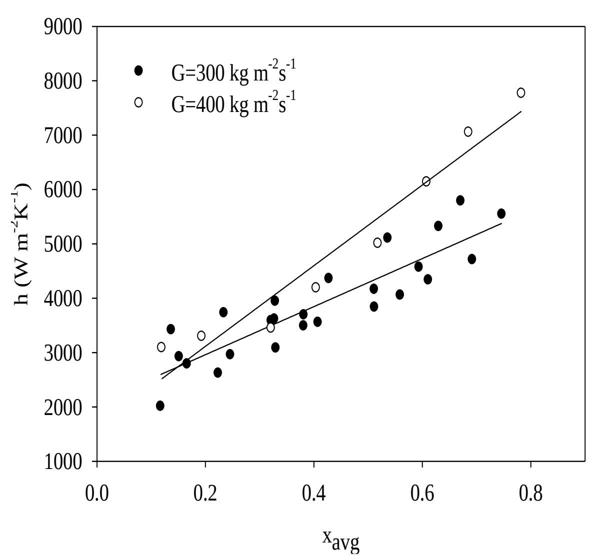
Table 3 Average proportional errors for vertical and horizontal corrugated tubes

Gungor and Winterton [2]	Gungor and Winterton [3]	Liu and Winterton [1]	Kandlikar [5]	Shah [4]	Sun and Mishima [9]	Kew and Cornwell [7]	Lazarek and Black [8]	Kenning and Cooper [6]	Wongsagnam et al. [11]	Laohalertdecha et al. [12]	Akhavan-Behabadi and Esmailpour [10]
Horizontal corrugated tube 1											
24,19	33,57	45,64	53,50	33,83	48,82	64,10	66,74	31,88	33,88	13,46	29,77
Horizontal corrugated tube 2											
26,77	36,70	46,12	52,54	38,33	50,28	65,57	67,71	36,95	37,45	12,63	34,95
Horizontal corrugated tube 3											
29,46	40,10	52,44	59,93	39,98	53,31	66,84	69,56	37,67	39,41	10,92	34,92
Horizontal corrugated tube 4											
18,96	30,12	42,02	50,79	29,75	46,82	62,85	65,46	27,96	31,68	10,86	25,50
Horizontal corrugated tube 5											
18,34	28,91	43,03	51,77	28,46	45,71	61,66	64,65	26,26	29,27	11,39	24,14

Table 2 represents average proportional errors for vertical and horizontal smooth tubes. Gungor and Winterton [2] correlation showed the best prediction performance among the tested correlations for both smooth vertical and horizontal tubes. It can be seen that average proportional errors of for smooth vertical and horizontal tubes are 4.28% and 17.06%, respectively.

Average proportional errors for vertical and horizontal corrugated tubes are given in Table 3. Gungor and Winterton [2] having maximum average proportional errors of 6.52% and Laohalertdecha et al. [12] correlations having maximum average proportional errors of 13.46% have the best predictability of experimental values for all tested corrugated tubes.

The experimental heat transfer coefficient of horizontal smooth and corrugated tubes compared in Fig. 3. The correlations of Gungor and Winterton [2], Wongsagnam et al. [11], Kenning and Cooper [6] and Gungor and Winterton [3] predicted most of experimental data in the range of ± 30% for smooth tube. Also, the heat transfer coefficients calculated by means of Laohalertdecha et al. [12], Gungor and Winterton [2], Akhavan-Behabadi and Esmailpour [10] and Kenning and Cooper [6] correlations predicted most of experimental data in the range of ± 30% for corrugated tubes.



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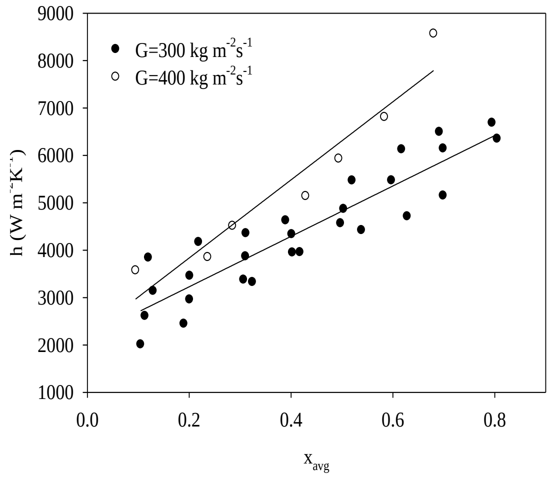

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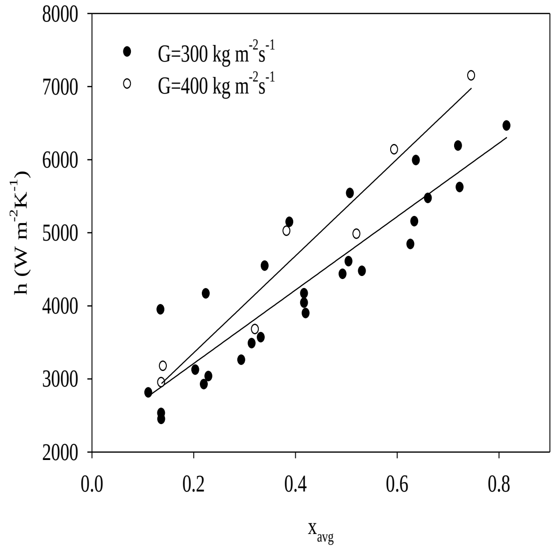

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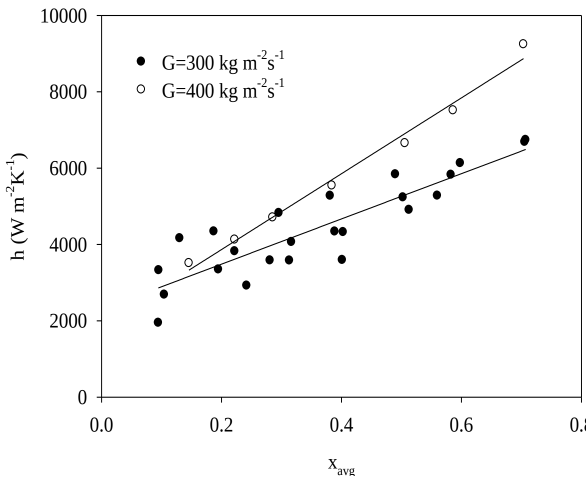

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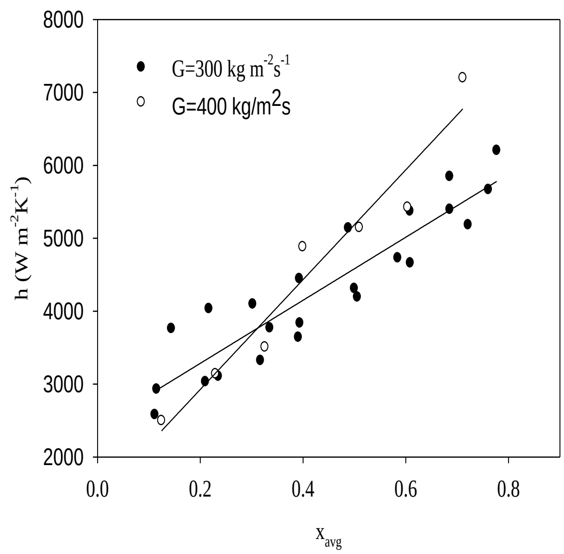

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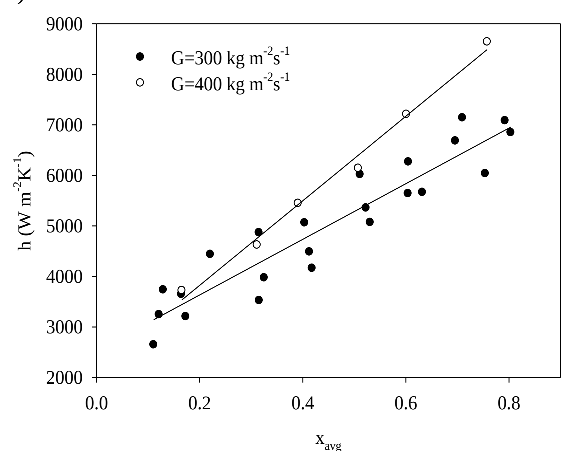
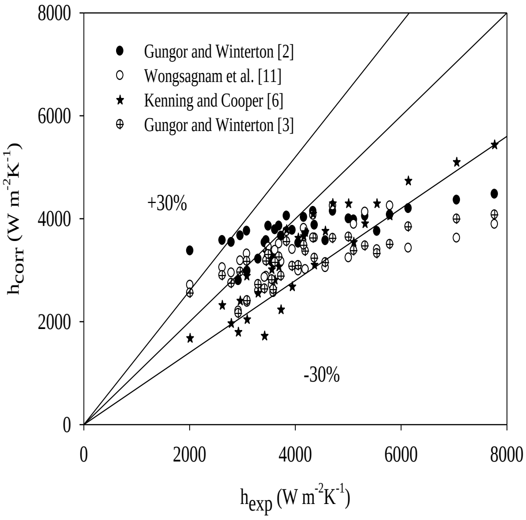


d)

Fig. 2. The variation of experimental convective heat transfer coefficients with average quality for mass fluxes of 300 and 400 kg/m^{2}s (a) horizontal smooth tube, (b) horizontal corrugated tube 1, (c) horizontal corrugated tube 2, (d) horizontal corrugated tube 3, (e) horizontal corrugated tube 4, (f) horizontal corrugated tube 5


Gungor and Winterton [2]
Wongsagnam et al. [11]
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h_corr (W m^-2K^-1)
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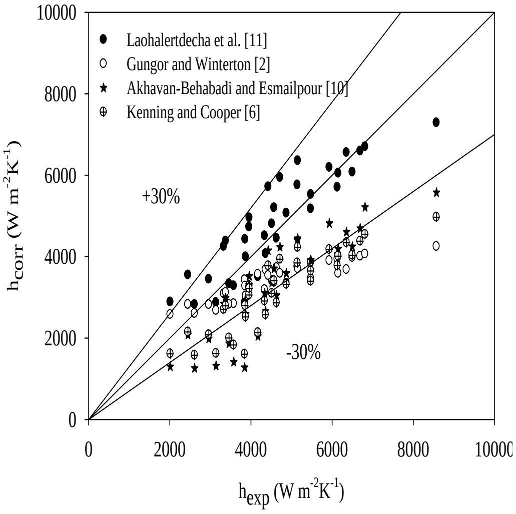
a)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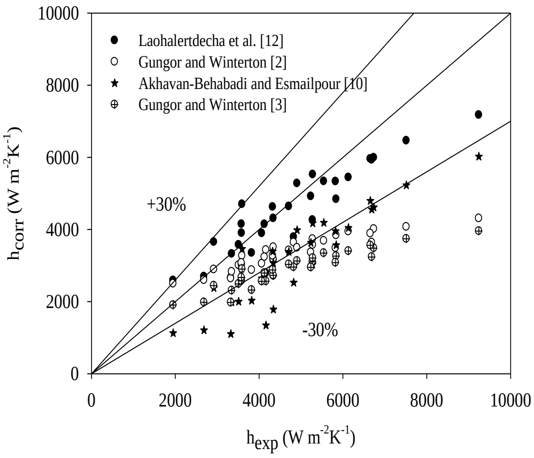
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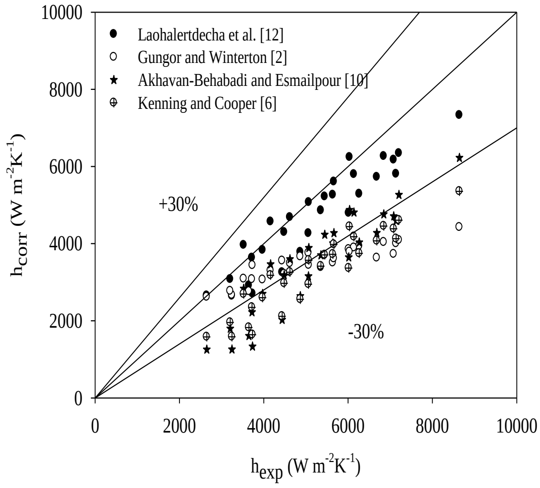
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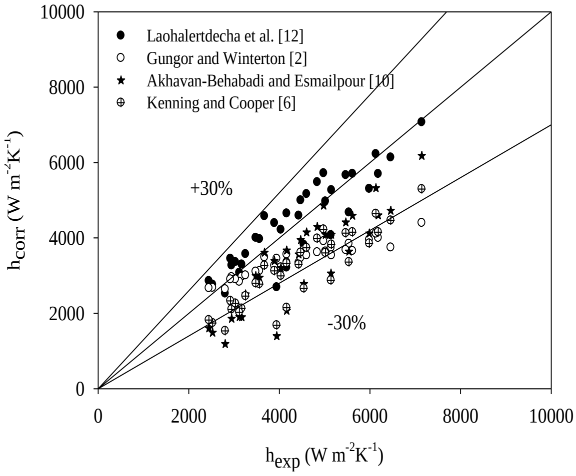
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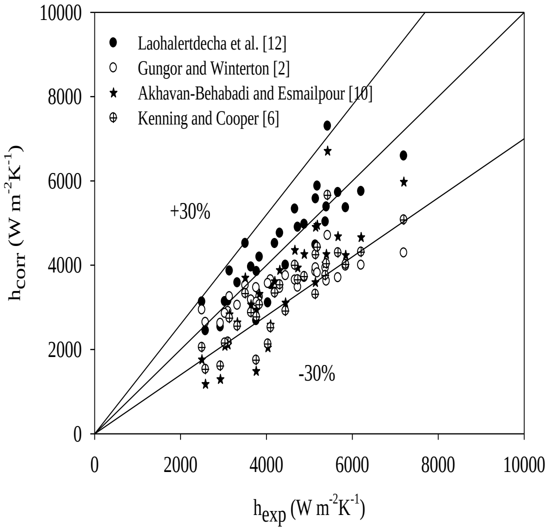
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Kenning and Cooper [6]
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h_corr (W m^-2K^-1)
h_exp (W m^-2K^-1)

f)

Fig. 3. Comparison of the convective heat transfer coefficients for (a) horizontal smooth tube, (b) horizontal corrugated tube 1, (c) horizontal corrugated tube 2, (d) horizontal corrugated tube 3, (e) horizontal corrugated tube 4, (f) horizontal corrugated tube 5

A lot of figures might be created from the findings of the analyses, as a result of space limitation, only typical results are shown. An extensive range of data on the explanations above can be seen from the authors' former works in the literature. It should be also noted that in-tube condensation version of this study can be seen from authors' published works [14, 15].

IV. Conclusion

Following specific results can be expressed from the calculations of this study in order designers to select the most compatible practical correlations.

a- The most predictive correlations for the smooth horizontal tube are Gungor and Winterton [2] and Wongsagnam et al. [11] with the average proportional error rates of 17.06% and 20.88%, respectively.

b- The most predictive correlations for the corrugated horizontal tubes are Laohalertdecha et al. [12] and Gungor and Winterton [2] with the maximum average proportional error rates of 13.46% and 29.46%, respectively.

c- Gungor and Winterton [2]'correlation are found to be capable of prediction on the flow boiling data belonging to the horizontally oriented smooth and corrugated tubes.

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IV. References

[1] Z. Liu, 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 Mass Transfer 34 (1991) 2759-2766.

[2] K.E. Gungor, R.H.S. Winterton, A general correlation for flow boiling in tubes and annuli, International Journal of Heat Mass Transfer 29 (1986) 351-358.

[3] K.E. Gungor, R.H.S. Winterton, Simplified general correlation for saturated flow boiling and comparison with data, Chemical Engineering Research and Design 65 (1987) 148-156.

[4] M.M. Shah, Chart correlation for saturated boiling heat transfer: equations and further study, ASHRAE Transactions 88 (1982) 185-196.

[5] S.G. Kandlikar, A general correlation for saturated two phase flow boiling horizontal and vertical tubes, Transactions of ASME 112 (1990) 219-228.

[6] D.B.R. Kenning, M.G. Cooper, Saturated flow boiling of water in vertical tubes, International Journal of Heat Mass Transfer 32 (1989) 445-458.

[7] P.A. Kew, K. Cornwell, Correlations for prediction of boiling heat transfer in small-diameter channels, Applied Thermal Engineering 17 (1997) 705-715.

[8] G.M. Lazarek, S.H. Black, Evaporative heat transfer pressure drop and critical heat flux in a small vertical tube with R-113, International Journal of Heat Mass Transfer 25 (1982) 945-960
.

[9] L. Sun, K. Mishima, An evaluation of prediction methods for saturated flow boiling heat transfer in mini-channels, International Journal of Heat Mass Transfer 52 (2009) 5323-5329.

[10] M.A. Akhavan-Behabadi, M. Esmailpour, Experimental study of evaporation heat transfer of R-134a inside a corrugated tube with different tube inclinations, International Communications in Heat and Mass Transfer 55 (2014) 8-14.

[11] J. Wongsa-ngam, T. Nualboonrueng, S. Wongwises, Performance of smooth and micro-fin tubes in high mass flux region of R-134a during evaporation, Heat and Mass Transfer 40 (2004) 424-435.

[12] S. Laohalertdecha, A.S. Dalkilic, S. Wongwises, Correlations for evaporation heat transfer coefficient and two-phase friction factor for R-134a flowing through horizontal corrugated tubes, International Communications in Heat and Mass Transfer 38 (2011) 1406-1413.

[13] K. Aroonrat, S. Wongwises, Evaporation heat transfer and friction characteristics of R-134a flowing downward in a vertical corrugated tube, Experimental Thermal and Fluid Science 35 (2011) 20-28.

[14] A.S. Dalkilic, S. Laohalertdecha, S. Wongwises, Experimental investigation of heat transfer coefficient of R134a during condensation in vertical downward flow at high mass flux in a smooth tube, International Communications in Heat and Mass Transfer 36 (2009) 1036-1043.

[15] A.S. Dalkilic, M. Aktas, O. Acikgoz, S. Wongwises, A review of recent empirical correlations for the calculation of determination of r134a's convective heat transfer coefficient in vertical condensers, International Communications in Heat and Mass Transfer 69 (2015) 41-50.