Chapter 6

Nucleate Pool Boiling Heat Transfer of Refrigerants Using Coated Surfaces

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Additional information is available at the end of the chapter

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Abstract

This work presents the experimental study of nucleated pool boiling heat transfer of R-134a and R-410A on a horizontal coated heating surface. The heating surface dimensions are 25.4 mm outer diameter and 116 mm effective length. The coated surfaces were fabricated by flame spraying technique. The copper powder was used as a coating material applied to the outer surface of copper tube. The experiments were performed for heat flux range of 5–50 kWm^{-2} at saturation temperature of 10°C. The heat transfer coefficients of both refrigerants demonstrated the same trends with applied heat flux increase and their magnitudes increases with increasing the value of applied heat flux. The present study also includes the effects of heat flux and coating parameter on boiling characteristics. The boiling heat transfer coefficient is enhanced by 1.9 times that of plain surface. An empirical correlation was also developed to predict the heat transfer coefficient with a mean error of 13%.

Keywords: pool boiling, heat transfer enhancement, coated surface, refrigerants, flame spraying

1. Introduction

Pool boiling characteristic has immense heat transfer applications because of the ability to remove enormous quantities of heat from heating surface with maintaining the lower temperature difference. This gives a reduced size of heat exchanger by enhancing the performance of equipments used in many industries such as refrigeration and air-conditioning industries, thermal power plants, process industries, and many other allied industries. In refrigeration and air-conditioning field, flooded evaporator is widely used as heat exchanger surface. The energy Industry demands for more enhancing surface and economic incentives as well, have spurred the development of methods to increase the heat transfer coefficients. The various types of surface modification are used to improve the performance of equipments. Among the various available surface coating methods, metallic coating is one of the appropriate coating material for use in pool boiling of refrigerants. This is considered for the basis on the overall consideration of enhanced performance, durability, the ease of manufacturing and application. Hence, a flame spraying technique was used to prepare the coated surfaces. The copper powder was used as a coating material applied over the surface of copper tube.

The impacts of Refrigeration and air conditioning system on environment are majorly supposed to release ozone-depleting refrigerants. The increase in emission of halogenated refrigerant used in refrigeration systems increases the concentration of greenhouse gases in the environment. As a result, adverse climatic changes being observed recently and contribute significantly to global warming. A reduction in GHG emissions can only be achieved by using alternative refrigerants. Thus to protect the earth surface from direct infrared radiation and to find solutions to socioeconomic favor for the mankind, further study in this area is indispensable.

In order to fill the gap caused by the phase out of CFCs, researches have been carried out extensively to find alternative refrigerants whose ozone depletion potential (ODP) is zero. Nowadays, design engineers have put remarkable efforts in designing efficient and compact systems thus reducing ODP. Therefore, R-134a can be better alternative refrigerant for CFCs because of its good thermodynamic properties as well as eco-friendly features. The refrigerant mixture R-410A is also a long-term alternative refrigerant with zero ODP for time being in developing countries. Due to the low temperature efficiency and lower discharge temperatures, favorable physical and transport properties, R410A are widely preferred in refrigeration and air conditioning applications. In present study, the refrigerants R-134a and R-410A were used as alternative refrigerants. A very few research works on pool boiling of refrigerants on coated surfaces are reported in the literatures.

Nowadays, many researches have focused on enhanced boiling heat transfer surface which fulfill the requirements of advance developments in energy generating equipments. Thome [1]; Webb [2]; and Bergles [3] were discussed the different enhancement techniques for fabricating the heating surfaces. Active, passive and compound were considered as enhancement techniques. A passive technique does not require any external power source and also its fabrication process of heating surface is easy and economical. The metallic coated surface is prepared by the passive techniques for enhancing the boiling heat transfer. The boiling of new refrigerants on metallic coated surfaces has been studied by very few researchers [4–18]. However, the experimental work on metallic coated surfaces and their impact on boiling heat transfer are still indispensable. The boiling heat transfer of refrigerants on these surfaces generates more active nucleation sites and it depends on the properties of refrigerants and surface geometry. The main objective of the present study is to conduct the pool boiling experiments of refrigerants R-134a and R-410A in metallic coated surfaces. This study provides the data to the refrigeration industry for the design of efficient heating surface. In addition, parametric study on the boiling characteristics is analyzed which determines the probability of flooding the reentrant cavities and the amount of superheat required for bubble growth.

2. Experimental facility and procedure

Figure 1 represents the experimental setup for the boiling of refrigerants on coated surfaces. This setup consists of boiling vessel, power supply arrangement, condensing loop and test section. The sealed cylindrical boiling vessel was fabricated with a 490 mm long stainless steel pipe of 150 mm internal diameter. It is closed at both ends with flange of same material. The bottom cover of boiling vessel contains fitting to mount a pipe with a valve to charge or drain out refrigerant from the vessel as and when required. This bottom cover has also fitting for preheater to control the saturation pressure of the system. Two inspection windows were welded at diametrically opposite side position of the boiling vessel body for visual observation of bubble dynamics on and near the tube surface. The boiling vessel was well insulated with glass wool to ensure adiabatic condition. A powder flame spraying method is applied to fabricate heating surface where copper powder used as coating material. The details of the test section along with the cartridge heater shown in **Figure 2**. The test sections were heated by cartridge heaters. Each cartridge heater having 16.5 mm diameter and actual heated length of 116 mm, and was inserted into the copper tube. K-type chromel-alumel thermocouples were embedded in the circumferential position of the tube to measure the wall temperatures. Four holes at top, two sides and bottom positions were made circumferentially in the wall thickness of test tube. Four thermocouple probes were embedded to

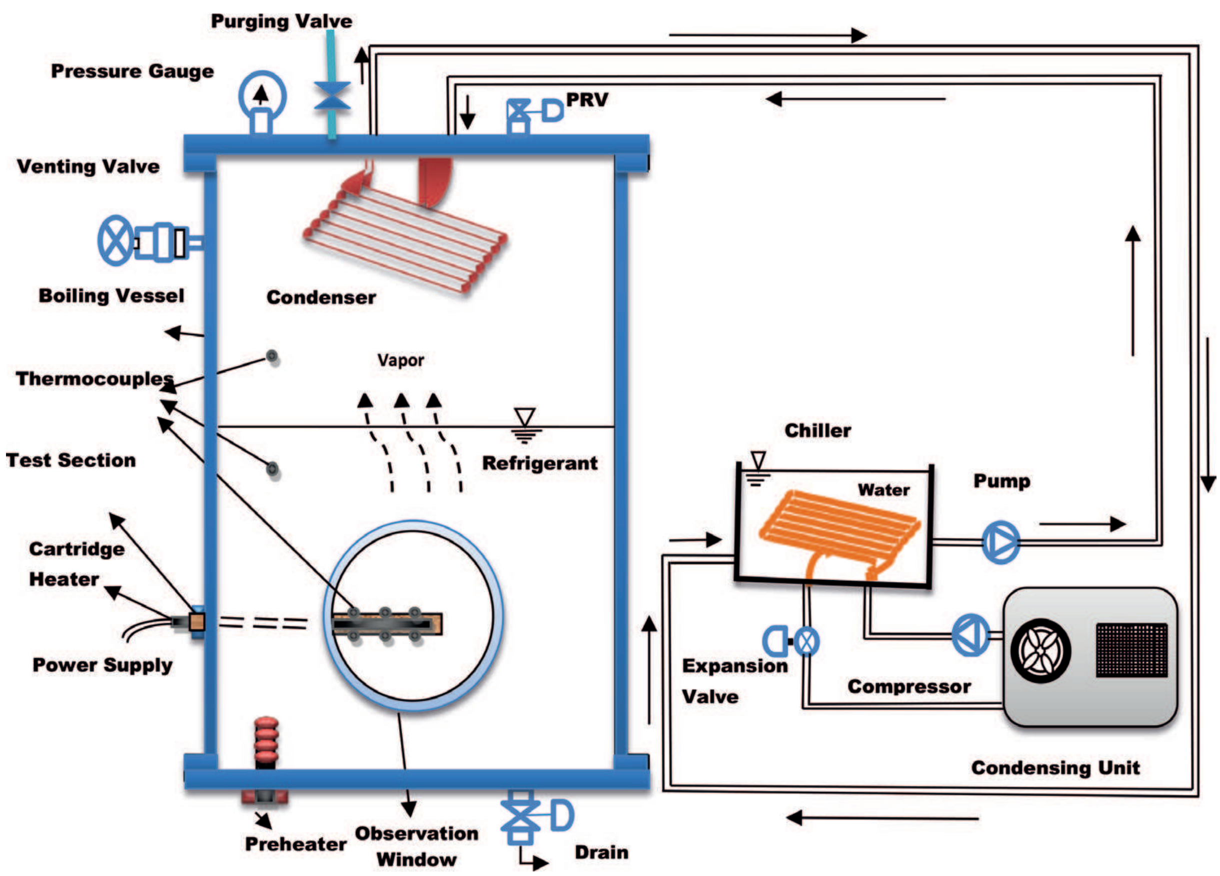

Figure 1. Schematic diagram of experimental set up.

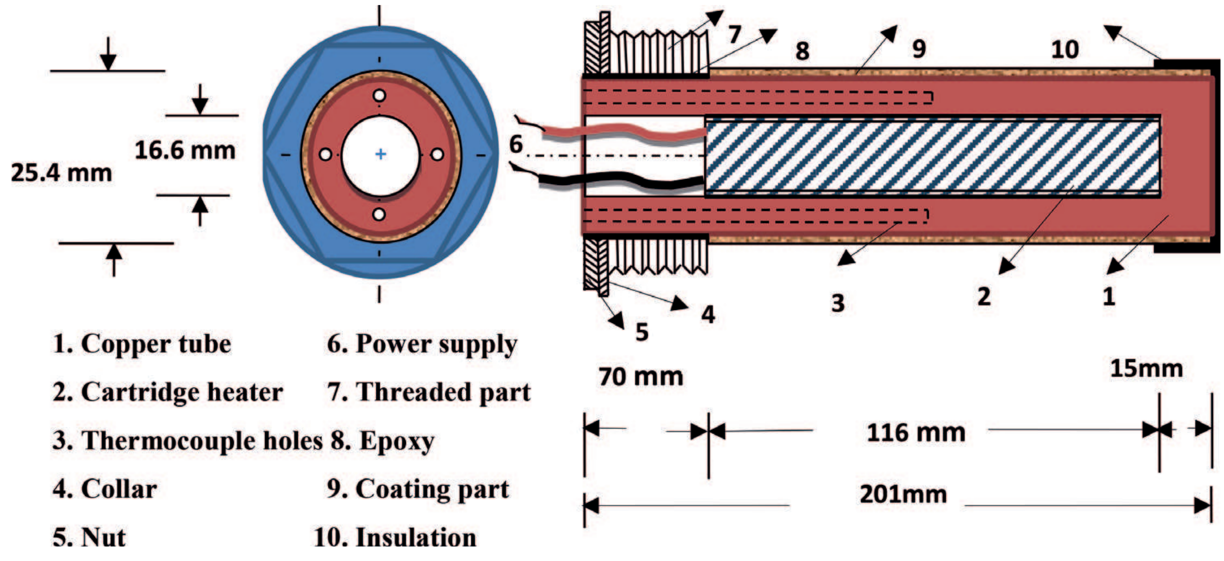

Figure 2. Test section along with the heating arrangement.

measure the temperature at each hole and the average of these temperatures indicates the wall temperatures. Two thermocouple probes were also inserted through the top cover of the boiling vessel at suitable positions to measure the liquid pool temperature on opposite sides of heating tube.

The refrigerant vapor produced in the boiling vessel get condensed in an internal water cooled condenser and returned to the vessel. The internal condenser was mounted vertically below the top cover of the vessel to ensure return of the condensate by gravity to liquid pool. An external chiller with an accurate temperature controller was used to condense to maintain the pool temperature to 10°C. The heat transfer experiments were conducted for boiling of three refrigerants R-134a, and R-410A on plain and coated tubes at saturation temperature of 10°C. The following procedure was adopted for pool boiling of given refrigerants on coated surfaces:

- Before each test, the boiling vessel and fabricated test surfaces were cleaned with acetone, chlorinol and water.
- A pressure and vacuum gauges were used to check the leakage inside the boiling setup maintaining a pressure of 2.0 MPa and a vacuum of 60 cm of Hg for 24 hours. When the system is ensured from leakage then refrigerant was filled into boiling vessel as vapor form to the level of 35 mm above the boiling surface.
- To remove the non-condensable gases and air from the boiling vessel through purging valve, the refrigerant was heated at 50 kWm^{-2} for 1 hour. When the complete removal of air from the boiling vessel was ensured, the chiller was started 3 hours prior to beginning of the experiments.
- The cartridge heater was used to supply the power to test surface in the range of 5–50 kWm^{-2} for both increasing and decreasing heat flux levels. This was done for the purpose of avoiding a boiling hysteresis. The saturation temperature of the refrigerant was maintained at 10°C within 0.2°C temperature fluctuation.

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Test tubes	Coating thickness, t_c (μm)	Porosity, ε (%)	Mean pore diameter, d_{mp} (μm)
C-1	42	11.03	2.58
C-2	95	13.1	2.52
C-3	151	8.5	1.75
C-4	271	13.8	2.39
C-5	395	10.4	2.45
C-6	423	11.9	2.51

Table 1. Coating parameters of tested tubes used in present study.

- Data were acquired under steady state within the variation of the wall temperature of 0.1°C in 5 minutes. For each power input, the condenser mass flow rate of liquid was adjusted to maintain the constant pressure. An 8 channel data acquisition module (ADAM-4019) was used to collect the experimental data. The above procedure was repeated for each test surface.

2.1. Fabrication of test surface

In current study, the thermal spraying coating surface was prepared at Metallizing Equipment Co. Pvt. Ltd. (MEC) Jodhpur, India. The oxy- acetylene flame was used to melt copper powder by a spraying gun. For atomization and acceleration of the particles onto heating surface, high pressure air is passed over the molten material to solidify and forming a coating. For supplying the oxygen and acetylene gases, two stage precision regulators were used in this spray system. The spray distance for the coating surfaces is 18 cm. This spray conditions depend on the pressures and flow rates of oxygen and acetylene gases. Oxygen as the oxidizing gas at a pressure of 0.25 MPa and a flow rate of 1.27 m^3/hour with acetylene gas pressure of 0.11 MPa and a flow rate of 1.56 m^3/hour were maintained. The specifications of prepared coated surfaces were as given in **Table 1**. Scanning electron microscope (SEM) images of test surfaces were analyzed using 'Image J' offered by Research Services Branch (RSB) of the National Institute of Mental Health (NIMH). The SEM images of one coated tube of 42 μm thick and plain tube is shown in **Figure 3**. The image analysis procedure is well described by Dewangan

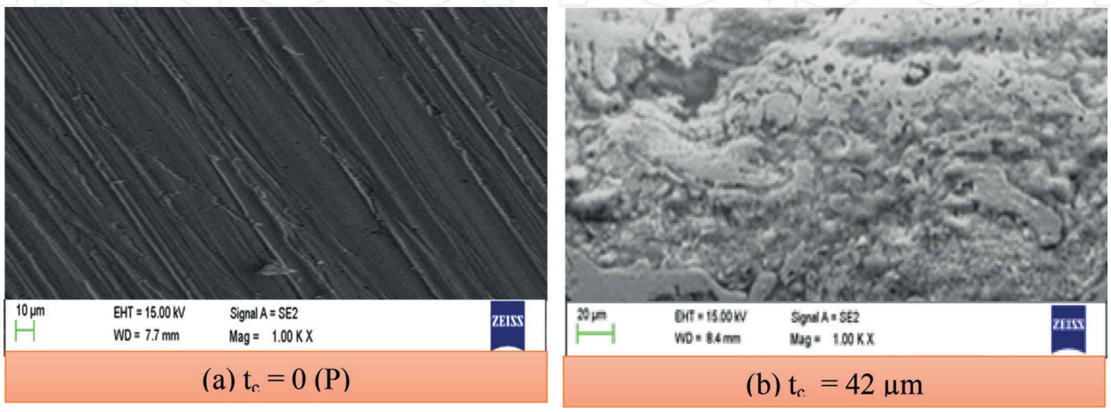

Figure 3. SEM images of plain and coated surfaces.

et al. [18]. This analyze is done to obtain coating parameters of the coated surfaces. As a result, the coating parameters are obtained and as shown in **Table 1**.

3. Results and discussion

In order to establish the integrity of the experimental set-up and verify the temperature measurement in the present test arrangement, preliminary tests have been conducted with R-134a, and R-410A. The heat transfer results of the pool boiling of refrigerants over a horizontal tube are compared with conventional correlations of Stephan and Abdelsalam [19]; Cooper [20]; Cornwell and Houston [21]; Gorenflo [22]; and Jung et al. [23] correlations. The comparisons of the experimental results have been depicted in **Figures 4** and **5**. These figures show the discrepancy between the current experimental data and the predicted results incurred with the same operating condition for different refrigerants. This shows that, the experimental results on pool boiling using the test apparatus and measurement system are reliable. The straight line in each plot signifies that, no discrepancy between the experimental and correlated data.

Dashed lines are appended to show the deviation of calculated values from the experimental data. The experimental result shows ±15% errors for R-134a with mean deviation (MD) ranging from −8.91% to 15.21% and a mean absolute deviation (MAD) of 7.91 to 15.21% for R-134a and ± 19% errors for R-410A with a MD of −19.19 to 18.98% and MAD of 5.85 to 18.98% for R-410A. Among them, the deviations are smallest by using Cornwell-Houston correlation for R-134a. Cooper correlation also shows the minimum deviation the smallest deviation between the experimental and the calculated values for R-134a and R-410A. The variation of imposed

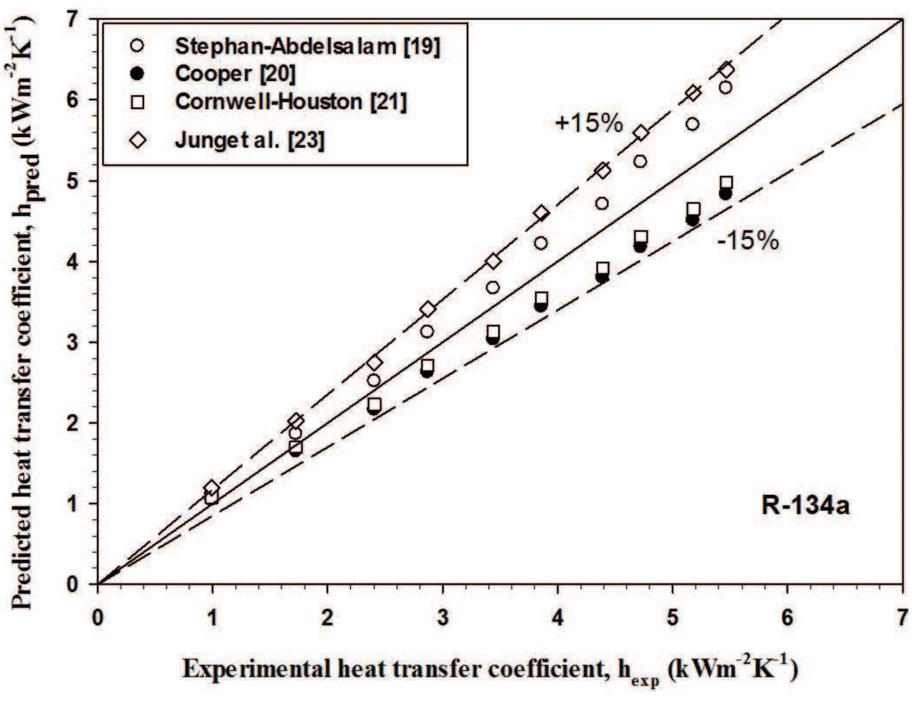

Figure 4. Comparison of experimental data of R-134a with correlations.

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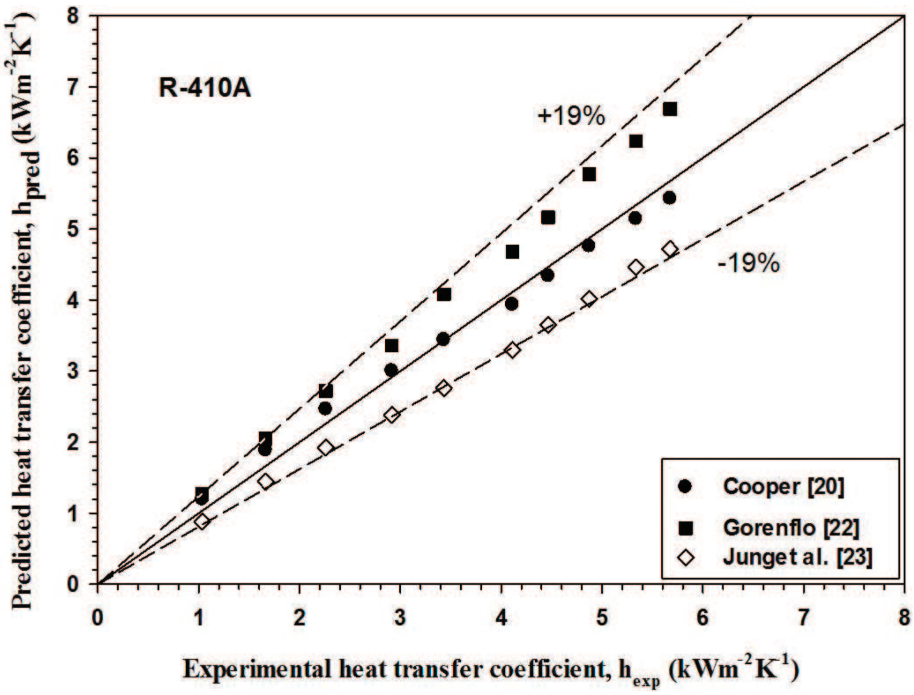

Figure 5. Comparison of experimental data of R-410A with correlations.

heat flux with wall superheat for the boiling of all three refrigerants on a plain heating tube surface at saturation temperature of 10°C. From this observation salient features can be inferred:

- At a given saturation temperature, the wall superheat increases with increase in imposed heat flux and the variation between two by power law, $q\ \alpha\ \Delta T\ n$.
- Heat flux in nucleate boiling of R-134a and R-410A are proportional to the wall superheat raised to a power of 3.86, and 4.22 respectively.

The main reason can also be seen from the physical properties of the tested refrigerants as in **Table 2**.

Figure 6 depicts the variation of heat transfer coefficients of a plain surface for saturated boiling of all three refrigerants at saturation temperature of 10°C with heat flux as a parameter. From this figure, the heat transfer coefficient as a function of superheat for the surface tested and the variation between two can be represented by a power law, $h \propto \Delta T^m$ where m varies between 2.87 and 3.21.

These features can be explained as follows: Wall superheat increases with increase in imposed heat flux on heating tube and therefore variation of the local heat transfer coefficient is found at the top, two sides of the middle and bottom of heating plain tube. This, in turn, increases the wall superheat and thereby value of minimum radius of nucleation sites at which bubble can originate decreases. This can be substantiated by following Eq. (1):

$$r_{cr} = \frac{2\sigma}{\left(\frac{dp}{dT}\right)_{sat} \Delta T_w} \quad (1)$$

Properties	R-134a	R-410A
T_{sat} (°C)	10	10
p (MPa)	0.415	1.09
ρ_l (kgm^{-3})	1261	1128.4
ρ_v (kgm^{-3})	20.23	41.917
k_l (Wm^{-1} K^{-1})	0.088	0.0974
k_v (Wm^{-1} K^{-1})	0.012	0.0132
$\nu_l \times 10^{-4}$ (m^2 s^{-1})	0.0019	0.0013
$\nu_v \times 10^{-4}$ (m^2 s^{-1})	0.0055	0.0030
σ (Nm^{-1})	0.0101	0.0075
M (gmol^{-1})	102.03	72.6
C_{pl} (kJkg^{-1} K^{-1})	1.37	1.58
h_{lv} (kJkg^{-1})	190.74	208.5

Table 2. Properties of tested refrigerants at 10°C [24].

The high heat flux condition enhances the number of nucleation sites and thereby the population of vapor bubbles form on heating surface. The bubbles grow and detach from the heating surface to travel in the pool of refrigerant, which increases the intensity of turbulence near the heating surface and increases heat removal rate. As a result, heat transfer coefficients are obtained to be

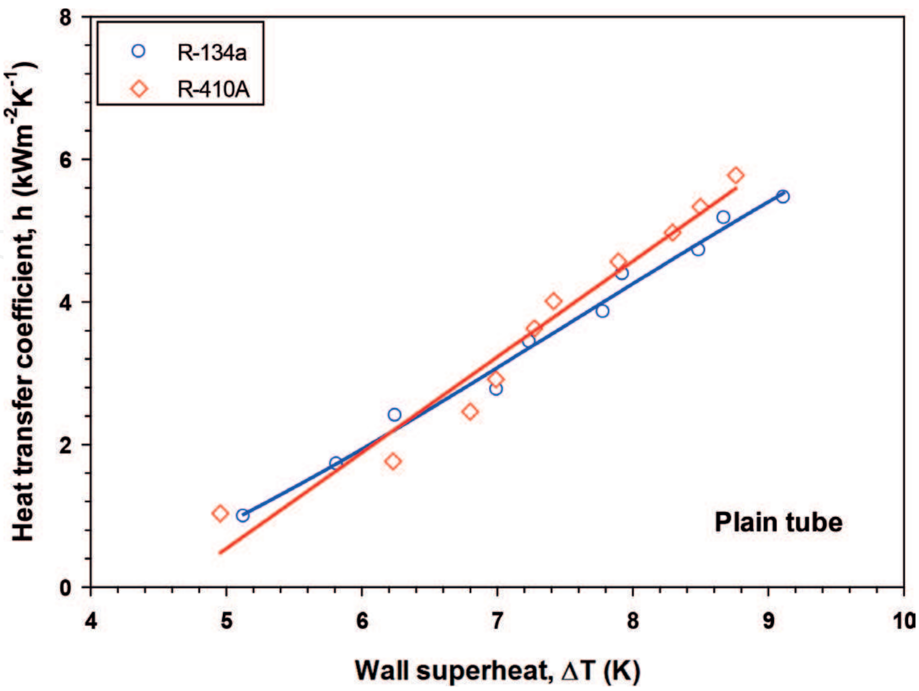

Figure 6. Variation of heat transfer coefficient with wall superheat.

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higher at high heat flux condition. The magnitude of heat transfer coefficient at given circumferential position on plain surface is found to differ from liquid to liquid. This is due to variation in thermo-physical properties of liquids under consideration. Similar features have also been observed during pool boiling of R-134a and R-410A refrigerants at saturation temperature of 10°C as shown in **Figure 7** which shows that the local heat transfer coefficients increases with increase in imposed heat flux and shown as $h \propto q^n$ where n values varies between 0.75 and 0.76.

Figures 8 and **9** contain the graphs drawn for heat transfer coefficients versus wall superheat as well as imposed heat flux for seven types of surface. The experimental data points lies in the nucleate pool boiling region as the wall superheat ΔT falls in the range of 2.2–9 K. The coating thickness of 151 μm needs high heat fluxes to achieve a particular wall superheat. Boiling on coated surfaces has differed due to the differences in their surface characteristics. The surface characteristics include the use of different material of substrate, coating material, fabricating method and coating thickness. But in the present study all these surface characteristics namely substrate material, coating material and fabricating method were ensured to be the same. The boiling heat transfer coefficients on the coated surfaces increased with increasing heat flux. The copper coating over heating surface enhances to the formation of porous matrix consisting of micro porous layers. This surface contains large amount of cavities of different sizes. Some of the cavities may have the size that meet the requirement of wall superheat for boiling incipience at low heat flux condition. This causes the coated surface to provide a higher heat transfer coefficient than the plain surface. The very tiny vapor bubbles originate on coated surface due to lower surface tension. Consequently, population of vapor bubble increases and their merging as vapor agglomerates also increases the resistance of bubble departure from coated surface. Therefore, heat removal rate and heat transfer coefficient reduces.

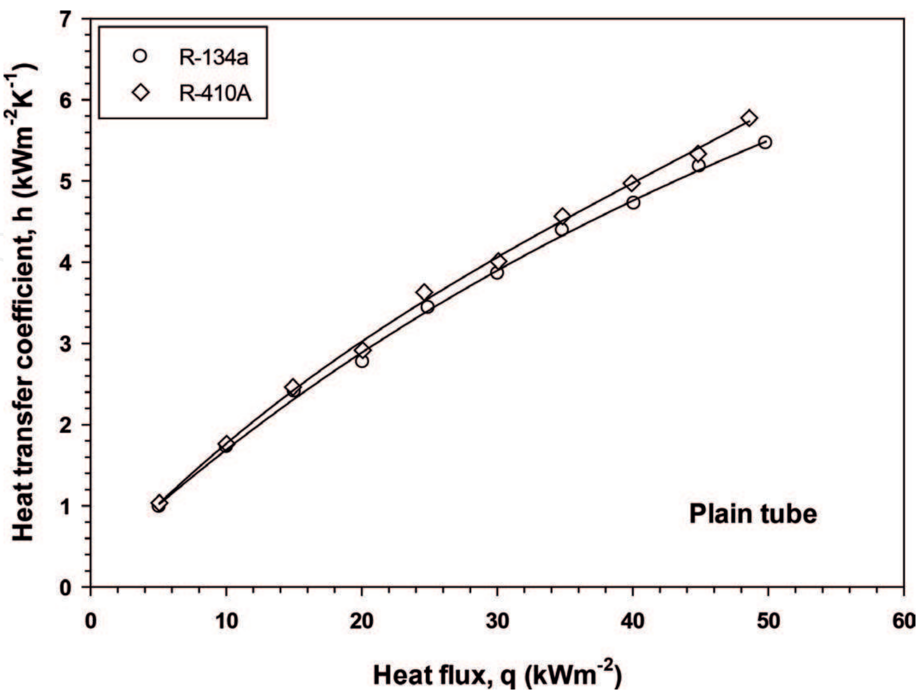

Figure 7. Variation of heat transfer coefficient with heat flux.

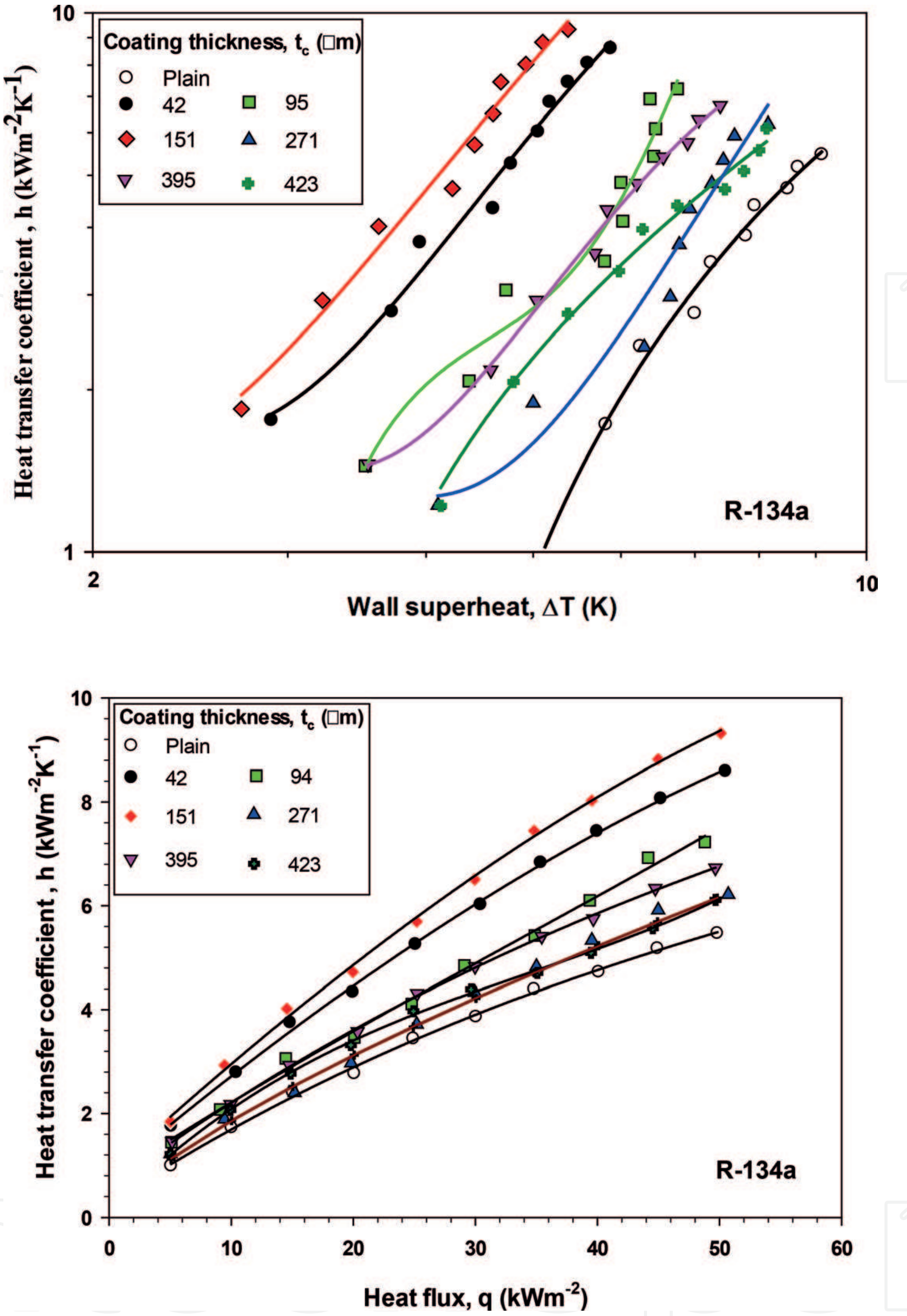

Figure 8. Variation of heat transfer coefficients for R-134a.

As a result, the recirculation of liquid in the inner portion of the coated surfaces increases and it enhances turbulence behavior of bubbles near the heating surface and heat removal rate. The effect of increase in coating thickness reveals two opposing characteristics [25] follows as: heat removal rate increases or decreases as a result of vapor bubble agglomerates and capillary action respectively. Coating thickness and applied heat flux are the main parameters to contribute of each effect. During initial stages of coating, the effect of capillary action is more pronounced than other coating surfaces for enhancing the boiling performance. Therefore,

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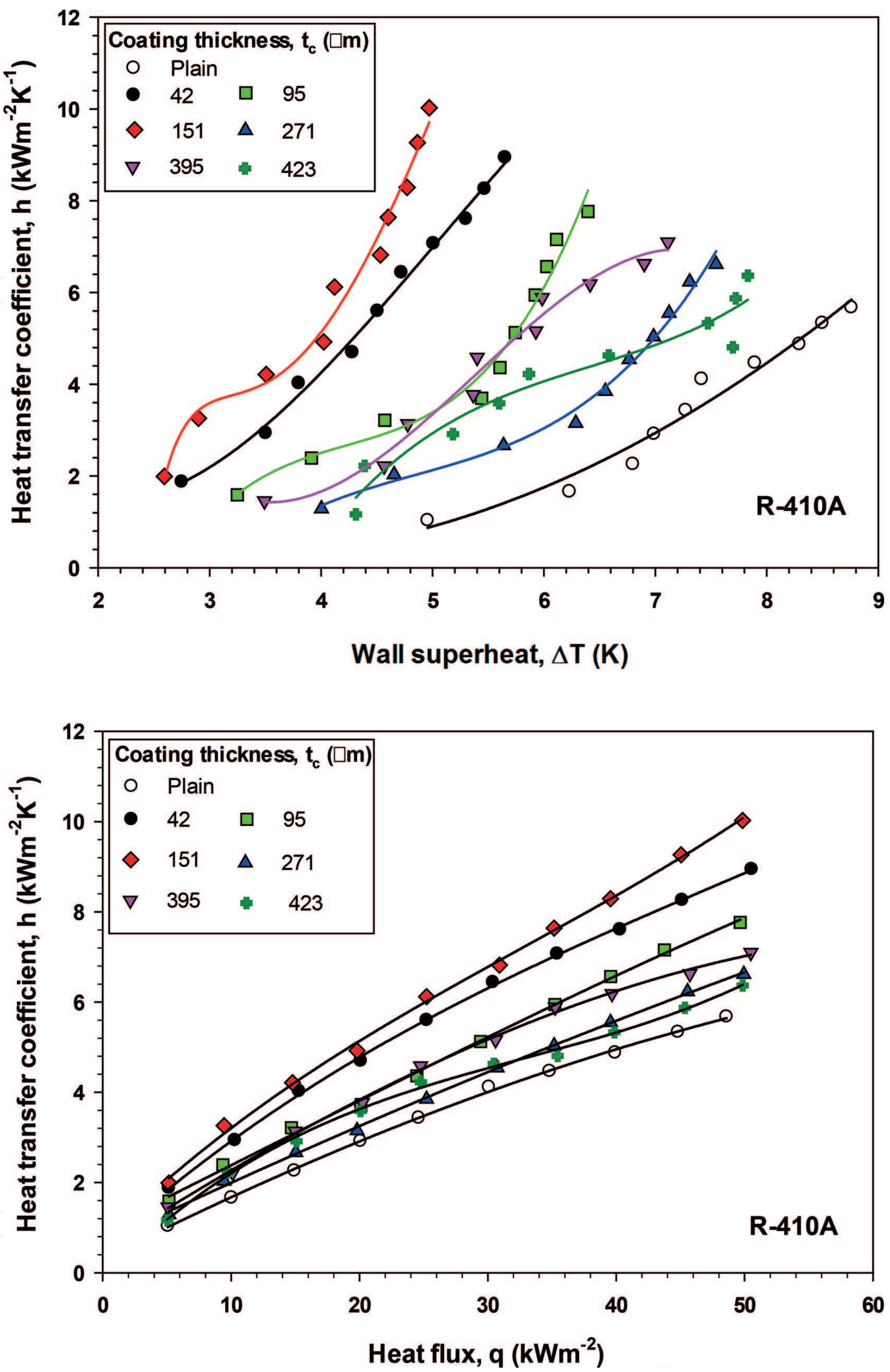

Figure 9. Variation of heat transfer coefficients for R-410A.

the boiling heat transfer coefficient is found to be more on a 42 μm thick coated surface than plain surface.

For an increase in coating thickness from 42 to 95 μm, the above characteristics are observed. However, the nucleation site density is the key parameter to observe the effect of heat removal rate from the heating surface as compared to capillary action. For a given applied heat flux, lower heat transfer coefficient is observed on a 95 μm thick coated tube than 42 μm thick tube due to the combined effect of above characteristics. Further increase in coating thickness from 95 to 151 μm, heat removal rate is more significant than other effects. The reason behind this phenomena is that the rise in recirculation intensity (liquid from bulk rushes to inner layer of the structure with greater intensity). Therefore boiling heat transfer coefficients on 151 μm is found to be greater than that on the 95 μm coated tube. Again increase in thickness of coating from 151 to 271 μm, the effect of heat removal rate is not to be significant than other effects. Thus for increase in coating thickness from 151 to 423 μm, heat removal rate and heat transfer coefficient decreases. For 395 μm coating surface, the intensity of recirculation of liquid play a dominating role than other effects which enhances the heat transfer rate. Due to this phenomenon, higher heat transfer coefficient is achieved on 395 μm coating surface than that on the 271 μm thick coated surface. As can be observed from **Figure 8**, at low heat flux value ≤30 kWm^{-2}, the effect of capillary action is more than that of nucleation site density. This caused the heat removal rate to be higher and therefore heat transfer coefficients on the 423 μm coated tube was found to be higher than that of on the 271 μm coated tube. However, at the heat flux condition ≥ 30 kWm^{-2}, the boiling heat transfer coefficient on 271 μm coating thickness surface is higher than that on 423 μm coating thickness surface. A rise in heat flux increases the number of active nucleation sites to form large number of vapor bubbles on heating surface. In fact, population at some stage becomes so large, and thereby the heat transfer rate and high heat transfer coefficient increases. However, when boiling of refrigerant R-134a occurs on a coated surface, the population of vapor bubble increases due to large heat flux, nucleation sites formed by coating layers.

4. Development of proposed correlation

The evolution of new refrigerants in the market makes the scientist to work for developing a nucleate pool boiling correlation based on a consistent database. Especially, many of the R-22 alternatives azeotropic refrigerant blends and to predict heat transfer coefficients of near azeotropic mixtures, a precise nucleate boiling correlation for pure refrigerants is required immediately. Therefore, a new correlation is developed based upon the present data of three refrigerants following Rudemiller and Lindsay approach [26]. Nucleate pool boiling heat transfer is affected by imposed heat flux, surface specifications, wall superheat, density of liquid and vapor, latent heat of evaporation, characteristic length, other thermophysical properties and their relationship may be summarized in Eq. (2) as described by Incropera [27].

$$h = h\left(q, \Delta T, h_{lv}, \sigma, k, \mu, C_p, \rho, t_c, d_{mp}, \varepsilon\right) \tag{2}$$

The non-dimensional groups can be formed by using the above properties. The major dimensionless groups according to Rudemiller and Lindsay are the Reynolds number *(Re)*, the Jakob number *(Ja)*, the constant heat flux number (N_{cf}) and the geometric scale factor (η). For gas flame coating heating surfaces, the conventional Jacob number (*Ja*) was written in terms of Nusselt number (*Nu*). This correlation considered the effects of system pressure in terms of reduced pressure ratio (p_r), a pressure function $F(p)$ [28] and the refrigerant vapor phase density on boiling heat transfer phenomena and pressure function, $F(p) = 1.8\, p_r^{\,0.17} + 4\, p_r^{\,1.2} + 10\, p_r^{\,10.}$.

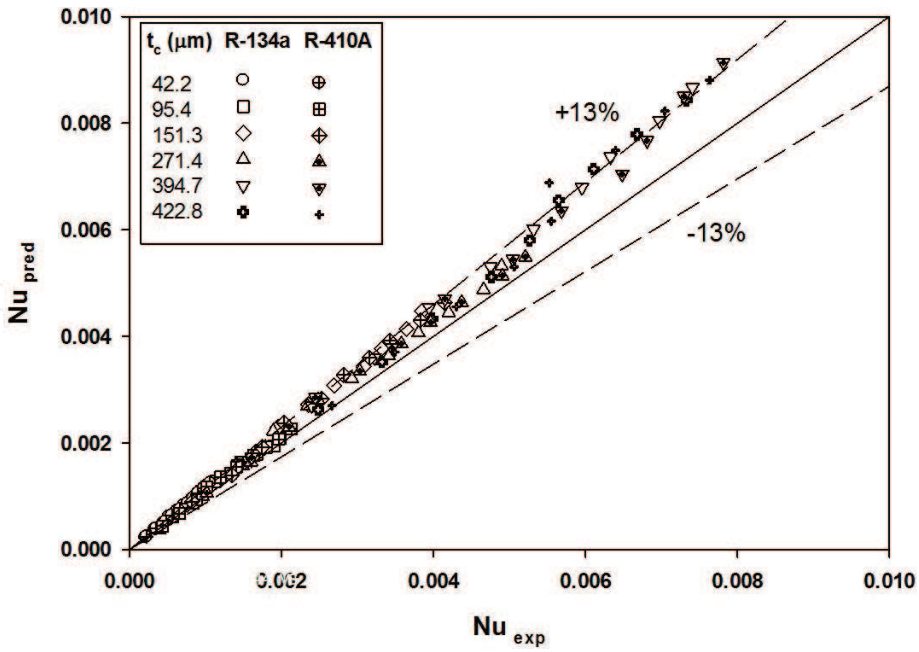

Figure 10. Deviation of present correlation against experimental data.

In this work, the values of constant and exponents depends upon the thermophysical properties of refrigerants R-134a, R-410A and measured experimental data and their values are:

$$C = 0.0132; m = 0.312; n = -0.043; a = 0.852; a_1 = 0.429; b_1 = 0.612$$

Hence, the relationship can be generated in the generalized form.

$$\begin{gathered} Nu = 0.013\,(Ra)^{0.31}\left(N_{cq}\right)^{-0.04}\left(\frac{1}{\eta_s}\right)^{0.85} F(p)\,(p_r)^{0.43}\left(\frac{\rho_l\,k_l}{\rho_v\,k_{eff}}\right)^{0.61} \\ 1.498 \times 10^{-5} \le Ra \le 3.938 \times 10^{-3} \\ 0.00167 \le N_{cq} \le 0.00267 \\ 16.356 \le \frac{t_c}{d_{mp}} \le 113.178 \end{gathered} \tag{3}$$

Figure 10 shows the comparison between the present experimental data with a correlation proposed in this study. The mean deviation was found to be within a maximum error of ±13 percent for all refrigerants tested.

5. Conclusions

In this study, nucleate pool boiling heat transfer coefficients (HTCs) of two refrigerants of R-134a, and R-410A were observed at saturation temperature of 10°C on heating surfaces. Data were collected for the heat flux range from 5 to 50 kW m^{-2} in the interval of 5 kWm^{-2}. As per the experimental results, following conclusions is made:

1. For refrigerant R-134a, the experimental data were predicted by Stephan and Abdelsalam [19], Cooper [20], Cornwell and Houston [21] and Jung et al. [23] correlations within an error band of±15%. For R-410A, the experimental data was predicted within an error band of ±19% by the above correlations. Gorenflo correlation was also predicted the HTCs of R-410A within an error band of±19%. This established the integrity of experimental set-up.

2. At a given saturation temperature and corresponding pressures for all three refrigerants, heat transfer coefficients increase with increase in heat flux. For the same heat flux range, the boiling heat transfer coefficients of R-134a and R-410A are 1.86 and 1.92 times higher than those of plain surfaces respectively.

3. Experimental data for pool boiling of R-134a, and R-410A on copper coated surfaces with different coating thicknesses of copper have been generated for various values of heat flux at saturation temperature of 10°C. Analysis has shown that the heat transfer coefficients increase with increase in imposed heat flux as in the form of $h \propto q\ n$ where n values depend upon the thickness of copper coating and boiling liquid. In fact, the value of n for boiling of refrigerants on coated surfaces is less than those of the plain surfaces.

4. The coated surface of 151 μm thick (C-3) shows the highest enhancement factor of 1.92 among all coated surfaces. In addition, the experimental data for all coated surfaces were correlated in terms of the major dimensionless groups of Nusselt number (Nu), the Rayleigh number (Ra), the geometric scale factor (tc/dp) and the constant heat flux number. The mean deviation was found to be within a maximum error of ±13 percent for all refrigerants tested.

Nomenclature

A	surface area of tube [m^2]
C_{pl}	specific heat of liquid [Jkg^{-1} K^{-1}]
D	tube diameter [mm]
$F(p)$	pressure function
d	diameter [μm]
Gr	grash of number, $Re^2\ P_r^{1/3}$ [−]
h	boiling heat transfer coefficient [Wm^{-2} K^{-1}]
h_{lv}	latent heat of vaporization
I	current [amp]
k	thermal conductivity [Wm^{-1} K^{-1}]
k_{eff}	effective thermal conductivity…, …$\varepsilon k_l + (1 - \varepsilon)\ k_c$ [Wm^{-1} K^{-1}]
M	molecular weight [gmol^{-1}]

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Nu	Nusselt number, ht/k_{eff} [−]
p	pressure [MPa]
p_r	reduced pressure, p_{sat}/p_{cr} [−]
Pr	Prandtl number [−]
Q	heat transfer rate [W]
Ncq	constant heat flux number, $\mu_l^2/\rho_l \sigma d_p$ [−]
q	heat flux [Wm^{-2}]
Ra	Rayleigh number, $GrPr$ [−]
Re	Reynolds number, $q\, d_p/\varepsilon\, \mu_l h_{lv}$ [−]
R_s	surface roughness [μm]
t	coating thickness [μm]
T	temperature [K]
ΔT	temperature difference [K]
V	voltage [volt]
Greek letters	
ρ	density [kgm^{-3}]
ν	kinematic viscosity [m^{2}s^{-1}]
σ	surface tension [Nm^{-1}]
ε	porosity [%]
Subscripts	
c	coating
cq	constant heat flux
cr	critical
exp	experimental
l	liquid
mp	mean pore
$pred$	predicted
s	scale
sat	saturation
v	vapor
w	wall

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