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Performance of heat transfer mechanism in nucleate pool boiling -a relative approach of contribution to various heat transfer components

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HIGHLIGHTS

- At small flux of heat values the natural convection plays governing role.
- In intermediate heat flux range all the three components are important.
- In high heat flux region it has been found that almost all the heat is transferred due to latent heat transport.
- A single heat transfer component cannot explain the heat transfer mechanism in the entire nucleate boiling region.

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ABSTRACT

Nucleate pool steaming is an effective mode of transfer of heat that helps to reduce the use of fossil fuels and thus reduce pollution. Transfer of heat in nucleate pool steaming is examined to occur through the combination of natural convection, enhanced latent heat and convection transport. At the intermediate heat flux range, all three components play a principal character. In the elevated flux of heat area as the heat flux increases the enhanced convection contribution decreases while latent heat transport contribution has been found to increase considerably. In this study, we attempt to develop a heat transfer relationship for the coefficient of transfer of heat and suggested based on the relative benefactions of three components to the boiling flux of heat. The current work stressed the absolute motion of warmth guaranteeing from the summation of each of

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the three segments of regular convection, upgraded convection and idle warmth transport for water and methanol with round mathematical formed. The hypotheses of air pocket development in nucleate pool bubbling hypothesize for a significant bit of warmth move to the air pocket happens by conduction through a fluid microlayer framed on the warmed surface has been thought of. The maximum deviation of error between the present analytical and experimental one for water, methanol, ethanol and benzene is 6.89%, 5.24%, 5.64% and 6.21% respectively. The highest divergence in the middle of the forecasted data and different (analytical and investigational) outcomes is found to be ±2.54. Results from many other investigators have also been compared for the better visualization of heat transfer correlations to the present one. The highest divergence in the middle of the forecasted data and investigational one for Nusselt number is observed to be ±3.27 along with the present analytical ones.

Nomenclature

q_w	Wall flux of heat, W/m2
q_{nc}	Natural convection flux of heat, W/m2
q_{fc}	Enhanced convection flux of heat, W/m2
q_{th}	Latent heat transport, W/m2
h	Coefficient of transfer heat, W/m2-K
A'_{nc}	Fraction of natural convection area (non-dimensional)
A'_{fc}	Fraction of enhanced convection area (non-dimensional)
ΔT	Wall superheat, K
Nu	Nusselt number
Gr	Grashoff number
Pr	Prandtl number
Re	Reynolds number
V_b	Bubble rise velocity, m/s
D_d	Bubble departure diameter, m
ρ	Density, kg/m3
g	Acceleration due to gravity, m/s2
σ	Surface tension, N/m
N	Nucleation site density, m-2
ϑ	Kinematic viscosity, m2/s
f	Frequency of bubble emission, s-1
h_{fg}	Latent heat of vaporization, J/kg
k	Thermal conductivity, W/m-k
β	Thermal expansion coefficient, k-1
C	Specific heat, J/kg-k
T	Temperature, K
J_a	Jacob number
α	Thermal diffusivity, m2/s
R	Heating Surface
H	Vertical stature over the base
R	Radius of the sphere of which bubble is a segment
Subscript	
l	Liquid
v	Vapor

1. Introduction

Bubbling is the most recognizable warmth move measure in view of its component of eliminating a huge measure of warmth at a moderately low warm main thrust. A lot of work has been done here to comprehend the bubbling marvels and a few connections have been produced for associating the exploratory information. In early models, it was conjectured that the high motion of warmth densities feasible in nucleate pool steaming are because of solid fomentation of fluid close to the warming surface by the fume bubbles produced during the cycle. In a large portion of these models, nonetheless, the air pocket has been considered as an inactive component that advances the warmth move by string activity yet never elaborate straightforwardly in the warmth move measure. It implies that in three models, the dormant warmth transport was nearly dismissed. In any case, Bankoff [1] appeared in his 1D warmth pass on calculation whichever assigned that inactive warmth passes on can add to the significant bit of the warmth transition. Rallis and Jawurek [2] announced that the inert warmth pass on is critical at all spots. This varies from the earlier held perspective that the thrilling activity of bubbles considered for most of the flow of heat. Further, it has been recognized that a single mechanism cannot explain the heat transfer phenomena in the entire nucleate pool boiling region. A comprehensive work of pool steaming reveals that it may be because of the reality that the heat transfer apparatus varies greatly in different regimes of nucleate steaming. However, in the last five decades, numerous investigations have been worked out in the form of co-relations to predict the heat transfer due to boiling in pure liquids, a binary mixture to that of the pure liquid components and many more.

Researchers have done a detailed study on the boiling of pure liquids. Vinayak and Balakrishnan [3] worked on the heat transfer in nucleate pool boiling of multi-component mixtures. The coefficient of transfer of heat was established to be a purpose of the contrast in the middle of the equilibrium liquid and vapor attentiveness of the thin element(s) and the least coefficient of transfer of heat takes place at the greatest of this worth. An experimental investigation was carried out by Fazel et al. [4], to show the pool boiling heat transfer characteristics of the pure and binary mixture along with its correlations. In this investigation, it was found that the heat transfer coefficients in boiling solutions were less on regular basis than that of pure component liquids having similar physical properties. Although it is necessary to calculate the heat transfer coefficient of a pure liquid to predict the transfer rate of the binary mixture, several studies have been carried out to forecast the transfer of heat coefficient of pure liquids in sub-atmospheric pressures. Expected atmospheric pressures do not appear similar to the traits of pool boiling in subatomic pressure. Raben et al. [5], Cole and Shulman [6], Van Stralen [7] and Stralen et al. [8] studied the nucleate pool boiling of water at sub-atomic pressures. Surface temperature gradients and oscillations are the results of low-frequency bubble departure, which causes boiling water at sub-atmospheric pressure. Gangto et al. [9] identified that large scale enrichment like porous mesh and porous foam enrich the heat transfer coefficient as well as critical heat flux. Enrichment in the censorious flux of heat varies during the coefficient of transfer of heat increment, which causes large structure enrichment like micro fins and microchannels. Stephan and Korner [10] developed an equation which showed the significance of a factor in the reduction of heat transfer in a binary mixture. Furthermore, researchers recommended the formulas and values of the empirical constant later making some changes for specific binary arrangements. Researchers like Jungnickel et al. [11] modified the Stephan and Korner [10] equation by adding the heat flux multiplier. They predicted the heat transfer coefficient for refrigerant mixtures by experimenting on a copper plate. Calus and Rice [12] developed an empirical system that was based on the theories proposed by Scriven [13] and Van Stralen [14] on bubble growth. They presented the data for acetone-water and isopropanol-water binary mixture and the individual components on boiling under free convection. Their correlation was unable to apply the accurate pure component correlations or the data available at that time because their methods yield the pool boiling coefficients directly. Their model couldn't predict the suppression, which was noticed in their experiment. Calus and Leonidopoulos [15] proposed an equation entirely based on literature. Their equation could not present the composition of transfer of heat precisely. This supplied a mean error which was lower than the Calus and Rice [12] and Stephan and Korner [10] equations. Schlunder [16] introduced a correlation of the pure component in which the parameter was the difference between the saturation temperatures. His correlation presented the dependency of the heat transfer coefficient on heat flux density and pressure. Thome [17], considered that the immersion temperature ascends at the interface of fluid fume of an air pocket. He built up another boundary to lessen the temperature distinction, which was the scope of bubbling (it is the differentiation in the purpose of dew and the temperatures of breaking point at a given condition). Later on, Thome and Shakir [18] determined another factor that was proposed by Shlunder [19] for example, the mass exchange adjustment factor. They accepted to frame their connection which was that close to the warming surface the air pocket point temperature isn't steady. Wenzel et al. [20] followed a technique, which was very like that of Shlunder [19], however, they needed to utilize the mass exchange condition to decide the real estimation of interface focus at the air pocket limit. The interface temperature was then controlled by utilizing the interface fixation. Fujita and Tsutsui [21] changed the connection proposed by Thome and Shakir [22] by including a term subordinate warmth motion and supplanting the mass exchange term. Fujita et al. [23] later modified the Fujita and Tsutsui [21] condition by modifying the term subject to the warmth with an ideal divider superheat subordinate term.

In recent years, several investigators have given their contribution towards the heat transfer mechanism for boiling purposes by utilizing different techniques. The change of phase of transfer of heat is unavoidable in most of the applications in industries that are receivable to its very elevated coefficient of transfer of heat [24]. However, the rates of transfer of heat correlated accompanied by these occurrences can be enhanced through most of the methods [25,26]. The enhancement of execution of the transfer of heat is dependent on Surface medication when it is difficult to modify the fluid and the other operating conditions [27,28]. Moreover, an unresisting plane alternation technique has been enticed by researchers around the globe, as a matter of interest. Modification on various contiguous laminas (that is hierarchical micro-nano, micro, and nano) have capitulated engrossing outcomes [29,30].

Thermal enlargement on the boiling plane occurs due to averting transfer of heat to the volume of liquid and generally underneath a very elevated flux of heat, the vapor blanket envelops the entire plane. Usually, an instant hike in temperature (managed through the flux of heat) is disagreeable and could be disasters also. The critical heat flux (CHF) or burnout heat flux is being defined as the greatest flux of heat, overhead that the heating structure should not be employed. The CHF is a known vital parameter in the boiling process [31]. Efforts are made in detaining the CHF and therefore securing the welfare of structures [32–34]. Under various conditions numbers of associations and replicas are suggested for fluid and planes [35–38]. However, the execution of structures accompanied by altered planes overtime is rarely examined or described. Laser texturing is one of these techniques which are being used for heat transfer applications by enabling indefinite mixtures for boiling planes accompanied by wanted wettability, plane ruggedness, plane outline and lamina of attributes. Kumar U.G. et al. [39], have given an assessment report on the part of laser textured planes on the boiling heat transfer, in which the present position of laser texturing methods and their utilization in boiling applications of transfer of heat such as melting and sintering, have been shown. The review report gives a summary of the investigations prime to this automation and their part in controlling the execution of boiling structures. It has also given a conclusion with future directions for recognizing the gaps in research and enhancing the processes of transfer of heat through laser-textured surfaces. Applications in different industries that are evaporators, boilers tubes, cooling of the elevated flux of heat dissipation electronic structures, air conditioning, some chemical process and cooling of nuclear reactors have been encountered in boiling and two-phase flow processes [31,40–43].

Bubbling of the unmoving fluid inside the vessel involves pool bubbling and is knowledgeable about the past applications in businesses. For the upgrade of the pace of move of warmth all through this activity, numerous endeavors have been done on before examinations to redress the components of the warming plane and its usage as a mix of different unadulterated liquids [45–49], while different analysts have indicated the aftereffect of enhancements to the working liquid [50–52]. The origination of new-designed warm liquids joined by efficient warm properties was right off the bat presented by Choi [53]. This examination was predominantly founded on the utilization of strong nano-scale material and subsequently known as “nanofluid”. These are the designing arrangement interference of nanoparticles as base fluid, having the point of strengthening of warm conductivity of those liquids. During the most recent decade, pool bubbling has been explored broadly by utilizing unadulterated fluids and nanofluids, yet data on the demonstrating of pool bubbling of nanofluids is very limited. The trade-in live stages, their associates, and the warming plane builds the level of intricacy to bubbling of unadulterated fluids because of the presence of ultrafine particles. It has been as of late demonstrated that the pool bubbling of nanofluids, for example, focus, molecule size, bubbles dynamic and warming surface structure is being influenced by a portion of these variables and henceforth, the hypothetical investigation of such apparatuses required further experimentations to accomplish an ideal guaging reproduction [54–56]. A mathematical examination has been completed on the new propagation of nanofluid 2-stage section inside the two-dimensional rectangular bubbling chamber [57]. The Eulerian–Eulerian approach has been utilized for the expectation of the bubbling bend and the trade between 2 stages. During the bubbling cycle plane correction was introduced through-plane toughness and wettability to lay it into thought in this multiplication. New conclusion affiliations concerning the nucleation zones thickness and air pocket flight breadth throughout steaming of silica nano-liquid were put to extended the steaming reproduction. In the study flux of heat partitioning (HFP) model was used to enhance the vital outcomes [58,59].

Although all the above researches are pointed out on an individual component basis, in this study, we attempt to develop a heat transfer relationship based on relative benefactions of natural convection, enhanced convection and latent heat transport. The quantitative evaluation of these components has been made based on the available information on bubble dynamics to arrive at an appropriate heat transfer relationship.

The current work stressed the absolute motion of warmth guaranteeing from the summation of each of the three segments of regular convection, upgraded convection and idle warmth transport for water and methanol with round mathematical formed. The hypotheses of air pocket development in nucleate pool bubbling hypothesize for a significant bit of warmth move to the air pocket happens by conduction through a fluid microlayer framed on the warmed surface has been thought of. It is hence a significant actuality that the base contact territory of the air pocket, through which warmth is moved, is precisely represented. For this reason, another air pocket development model as appeared in Fig. 1 is thought of. In this model, the state of a developing air pocket is spoken to by a round portion of which both the span of the base in contact with the warming surface, R_1, and the vertical stature over the base, H.

The expository connection has been created for the warmth move coefficient, which has been additionally contrasted with the test esteems got from Saini [44,60]. The biggest difference in the gauge information and Saini [44,60] (investigative and trial) was ±2.54. Results from Rohsenow [64], Kutateladze [71] and Wenzel [20] have additionally been analyzed for the better creative mind of warmth move relationships to the current one.

2. Components of heat transfer in nucleate boiling

We examined the transfer of heat from the heating surface to the liquid occurs as a result of the combination of natural convection, enhanced convection from the influence of domain of nucleating sites and the latent heat convey carried away by the bubbles which

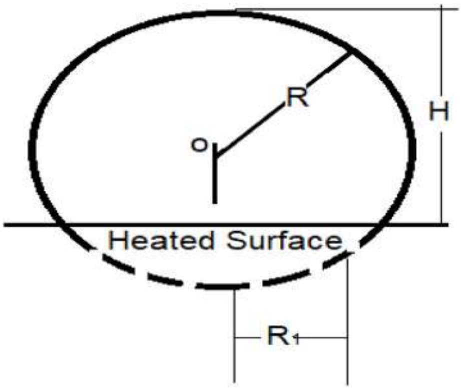

Fig. 1. Proposed bubble growth model.

are proportional to bubble emission frequency. Thus, total heat flux from the wall surface to liquid is given by:

$$q_w = q_{nc} + q_{fc} + q_{lh} \tag{1}$$

where q_w, is the total heat flux associated with the combined effect of all three heating components, q_{nc}, is the heat flux produced due to natural convection effect, q_{fc}, is the heat flux produced due to forced convection effect and q_{lh}, is the heat flux produced due to latent heat effect.

The heating surface area has been considered to consist of natural convection area (influence domain of nucleating sites), and these area fractions are related as follows:

$$A_{nc} + A_{fc} = A_{total} \tag{2}$$

$$A_{nc}^{'} + A_{fc}^{'} = 1 \tag{3}$$

Han and Griffith [62] showed that the velocity field all over a sphere dragged from a flat plane expands out to about one sphere diameter as a radius from the center. A criterion was developed for bubble initiation from a gas filled cavity on a surface in contact with a superheated layer of liquid. It was found that the temperature of bubble initiation on a given surface is a function of the temperature conditions in the liquid surrounding the cavity as well as the surface properties themselves. It was also found that the delay time between bubbles is a function of the bulk liquid temperature and the wall superheat and was not constant for a given surface. By consideration of the transient conduction into a layer of liquid on the surface, a thermal layer thickness was obtained. With that thickness and a critical wall superheat relation for the cavity, a bubble growth rate was obtained. Bubble departure was considered and was found that the Jakob and Fritz relation works as long as the true (non-equilibrium) bubble contact angle is used. At one gravity the primary effect of bubble growth velocity on bubble departure size was found to be due to contact angle changes.

Based on this, the area influenced by the departing bubble has been taken to be the circular area having the diameter equal to twice the diameter of the bubble at departure.

$$A_{fc}^{'} = \pi D_d^2 N \tag{4}$$

$$A_{nc}^{'} = 1 - A_{fc}^{'} \tag{5}$$

At very low heat flux, i.e. below the boiling incipience, the natural convection plays a vital part in transferring the heat from surface to liquid. Since no nucleation occurs at this low heat flux, enhanced convection and latent heat transport were zero, and the total area of the surface is available for natural convection. As the heat flux increases the generation of bubbles starts from the active nucleation sites, their growth and departure begin to play an important role in transferring the heat. An increase in heat flux causes a decrease in natural convection contribution because the area available for natural convection decreases while enhanced convection and latent heat transport components tend to increase. At the intermediate heat flux range, all three components are important. With further increase in heat flux the nucleation site density and bubble emission frequency experience substantial increase and at some heat flux value the natural convection component becomes negligible small since enhanced convection occurs everywhere on the heating surface and no surface available for natural convection. In the high heat flux region, the nucleation site density and bubble emission frequency is so high that vertical as well as lateral coalescence of bubbles takes place. In the high heat flux region as the heat flux increases, the enhanced convection contribution starts decreases while latent heat transport has been found to increases considerably. Near the censorious flux of heat range almost all the heat is transferred because of latent heat convey only.

The contribution of natural convection in Eqn. (1) has been estimated from:

$$q_{nc} = h_{nc} A_{nc}^{'} \Delta T \tag{6}$$

where, $A_{nc}^{'}$ is the fraction of heated plane undergoing natural convection.

The coefficient of transfer of heat h_{nc} has been evaluated using the corresponding correlation for natural convection heat transfer [60]:

$$Nu_{nc} = 0.31(GrPr)^{1/3} \tag{7}$$

Enhanced convection contribution has been estimated from:

$$q_{fc} = h_{fc} A_{fc}^{'} \Delta T \tag{8}$$

The coefficient of transfer of heat h_{fc} can be determined from the correlation for Nusselt number [61]:

$$Nu_{fc} = C(Re)^{1/3}(Pr)^{1/3} \tag{9}$$

where, C = 1.32 for water.

= 0.332 for organic liquids.

Bubble Reynolds number $\left(Re = \frac{V_b L}{\vartheta}\right)$ has been evaluated using the expression for bubble rise velocity [62]:

$$V_b = \frac{gD_d\Delta\rho}{2(\rho_l+\rho_v)} + \frac{2\sigma}{D_d(\rho_l+\rho_v)}^{0.5} \tag{10}$$

and the characteristic length (L) has been taken to be the influence diameter of departing bubble $(2D_d)$. Latent heat transport is given by:

$$q_{th} = \frac{\pi}{6}D_d^2 fD_d \rho_v h_{fg} N \tag{11}$$

where, N is the nucleation site density.

For water [63]:

$$N = \left(q_w/117.1\right)^{1.5} \tag{12}$$

For organic liquids [64]:

$$N = \left(q_w/81.72\right)^{2} \tag{13}$$

By rewriting the expressions for $q_{nc'}$, q_{fc} and q_{lh} components Eqn. (1) becomes:

$$q_w = 0.31K\left[\frac{g\beta\Delta TP_r}{\vartheta^2}\right]^{1/3} A'_{nc}\Delta T + CK(Re)^{1/2}(P_r)^{1/3}A'_{fc}\Delta T/_{2D_d} + \frac{\pi}{6}D_d^2 fD_d\rho_v h_{fg}N \tag{14}$$

Consequently, the expression for the heat transfer coefficient can be written as:

$$\mathrm{h} = 0.31K\left[\frac{g\beta\Delta TP_r}{\vartheta^2}\right]^{1/3} A'_{nc} + CK(Re)^{1/2}(P_r)^{1/3}A'_{fc}\Big/ 2D_d + \frac{\pi}{6}D_d^2 fD_d\rho_v h_{fg}N/_{\Delta T} \tag{15}$$

As the thermo-physical properties for water and organic liquids differ considerably appropriate expressions are used for evaluating the bubble departure diameter (D_d) and the product of bubble emission frequency and departure diameter (fD_d).

Bubble departure diameter for water [65]:

$$D_d = 1.6\times10^{-4}\left[\frac{\sigma}{g\Delta\rho}\right]^{0.5}\times\left[\frac{\rho_l C_l T_{Sat}}{\rho_v h_{fg}}\right]^{1.25} \tag{16}$$

Product of bubble emission frequency and departure diameter for water: [66], [67], [68]

$$fD_d = \frac{X^{1/3}(6.6\,Y)^{1/3}}{X_2Z^2 + X_1^{1/3}\frac{(6.6\,Y)^{2/3}}{\pi\,(J_a)^2}}(J_a \le 16) \tag{17}$$

$$fD_d = \frac{X^{1/3}\left[1.35\,(J_a)^{4/3}1+\frac{2.67Y}{{J_a}^4}\right\}\left\{1+\frac{2.67Y}{{J_a}^4}\right\}\left\{1+\frac{2.67Y}{{J_a}^4}\right\}^{0.5}\Big)^{2/3}\Big]}{X_2Z^2 + X_1^{1/3}\left[0.578\,J_a^{\frac{2}{3}}1+\frac{2.67Y}{J_a}\right\}\left\{1+\frac{2.67Y}{J_a}\right\}\left\{1+\frac{2.67Y}{J_a}\right\}^{0.6}\Big)^{4/3}\Big]}\left(16 \le J_a \le 100\right) \tag{18}$$

$$fD_d = \frac{9.18\,X^{1/3}J_a}{X_2Z^2 + 3.4\,X_1^{1/3}{J_a}^{1/2}}(J_a > 100) \tag{19}$$

where,

$$X = \frac{\alpha_l^2}{g}, \cdot X_1 = \frac{\alpha_l}{g^2}$$

$$X_2 = \frac{0.865}{\alpha_l}, \cdot Y = \frac{C_l\Delta T\sigma}{\alpha_l\,q_w}, \cdot Z = \frac{K_l\Delta T}{q_w}$$

For organic liquids: [69], [70]

$$D_d = \left[\frac{6\sigma K_l\Delta T}{g\Delta\rho\,q_w}\right]^{1/3} \tag{20}$$

$$fD_d = 0.59\left[\frac{\sigma g\Delta\rho}{\rho_l^2}\right]^{1/4} \tag{21}$$

It may be noted that both heat flux (q_w) and wall superheat (ΔT) values are required for Eqn. (14) and Eqn. (15) with the above-given expressions for bubble dynamic parameters. Since evaluation can only be based on one of these parameters, preferably heat flux (q_W) an iterative process has been employed to arrive at suitable values of superheat.

Further, based on Eqn. (15), the values of heat transfer can be evaluated by using the following Eqn. for the cases, Water and Methanol:

$$Nu = hD/K \tag{22}$$

3. Results and discussions

The components of natural convection, enhanced convection and latent heat transport and the total flux of heat resulting from the summation of these components have been plotted in Fig. 2 (a) for water and (b) methanol respectively. Fig. 2(a and b), represents that for little flux of heat values, the natural convection plays a dominant role. However, in the intermediate heat flux range, all three components are important. In high heat flux-region, almost all the heat is transferred due to latent heat transport. The above results have also been tabulated in Tables 1 and 2.

Fig. 3 (a) the heat flux values predicted by using the proposed Eqn. (14) have been compared with those values predicted by using some well-known correlations available in literature along with the experimental values. The greatest divergence in the middle of the forecasted data and Saini [44,60] is ±2.54 along with the present analytical ones. Results from Rohsenow [29], Kutateladze [71] and Wenzel [20] have also been compared for the better visualization of heat transfer correlations to the present one.

Fig. 3 (b) shows the comparison of heat transfer coefficients predicted by using the proposed Eqn. (15) with the corresponding experimental values [44,60]. During the calculation water was found the highest diverging component, as compared with the experimental ones. The greatest divergence of error in the middle of the forecasted and investigational one for water, methanol, ethanol, and benzene are 6.89%, 5.24%, 5.64% and 6.21%, respectively. The prediction errors were evaluated by using the following definition:

$$Error = \frac{h_{Pred.} - h_{Exp.}}{h_{Exp.}} \tag{23}$$

Further, by using Eqn. (21), the comparison of heat transfer rate in the form of Nusselt number for the proposed one with those of the values predicted by different available analytical values in literature along with the experimental values [44,60] has been shown in Fig. 4. The maximum deviation between the predicted data and Saini [44,60] has been found to be ±3.27 along with the present analytical ones. Notwithstanding above referred to writing [60], a short correspondence was given by Saini and Gupta, which could likewise have been utilized to locate similar outcomes [44], in which, the hypotheses of air pocket development in nucleate pool bubbling hypothesize for a significant bit of warmth move to the air pocket happens by conduction through a fluid microlayer framed on the warmed surface, was appeared. Results from Rohsenow [64], Kutateladze [71] and Wenzel [20] have also been compared for the better visualization of Nusselt number to the present one. In all of the above studies, it was found that the values of Nusselt number is increasing with the increasing values of heat transfer coefficients values. It was also found that the associated results for the present study were even better not only than that of the experimental one [44,60], but also to the given analytical studies in the literature. The maximum differences between the values of heat flux for present and experimental one [44,60], Rohsenow [64], Kutateladze [71] and Wenzel [20] were found to be (25.02–24.89 = 0.13), (25.02–24.12 = 0.9), (25.02–28.9 = −3.88) and (25.02–24.5 = 0.52), respectively. Henceforth, it could be concluded that the maximum rate point of increment for the comparative values of heat flux is 0.13 for

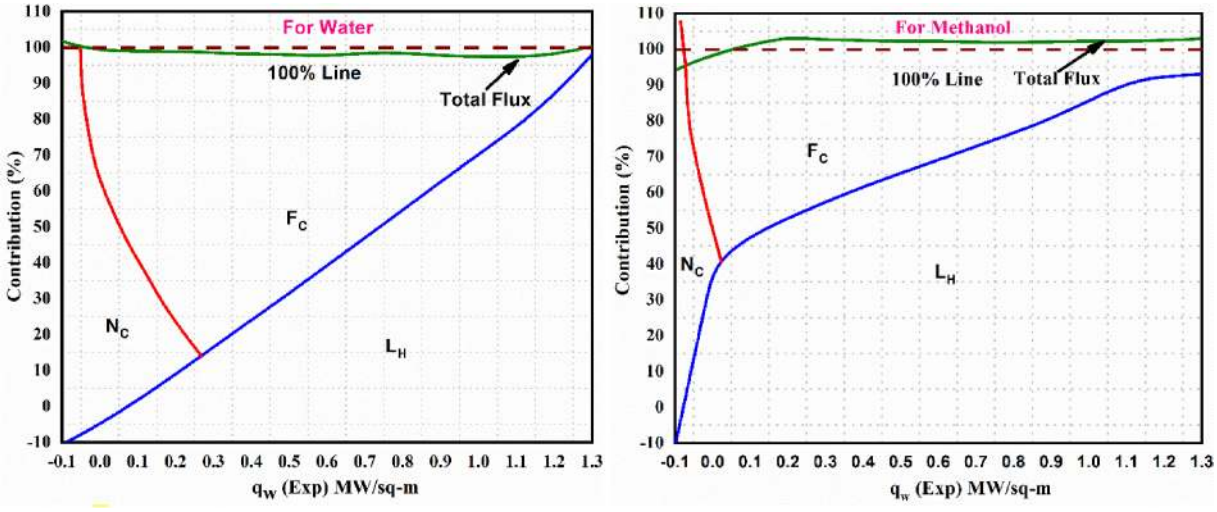

Fig. 2. Relative contributions of various heat flux components for (a) water; (b) methanol.

Table 1
Relative contributions of various heat flux components for water.

Water					
Heat Flux (qw) for NC (MW/sq-m)	**Contribution in %/NC**	**Heat Flux (qw) for FC (MW/sq-m)**	**Contribution in %/FC**	**Heat Flux (qw) for LH (MW/sq-m)**	**Contribution in %/LH**
−0.05179	99.52199	−0.09627	89.4571	−0.09627	−10.49713
−0.04899	89.23518	−0.0182	97.17782	0.02908	−3.30784
−0.03499	79.29254	0.12115	98.83365	0.19362	8.35564
−0.0154	67.28489	0.25521	96.83365	0.34977	20.32505
0.01788	58.03059	0.36096	98.14532	0.47543	29.57935
0.07107	45.71702	0.46983	98.14532	0.62597	41.24283
0.12115	37.11281	0.58989	97.45698	0.76843	52.21797
0.17434	26.86424	0.69565	98.48948	0.92177	64.22562
0.2689	21.8673	0.80731	98.48948	1.03033	72.44742
0.2867	17.8896	0.92737	97.45698	1.13639	80.66922
0.29991	16.6513	1.06672	97.45698	1.23966	91.95602
0.301265	16.3219	1.1321	96.9217	1.26871	95.821
0.316541	15.8741	1.39841	94.1235	1.28981	96.98761
0.335112	14.6721	1.158762	96.6542	1.299871	99.8875
0.3452	14.391	1.16998	97.45698	1.30093	100.78221
0.41731	13.82409	1.29813	99.2361	1.40341	102.14562

Table 2
Relative contributions of various heat flux components for methanol.

Methanol					
Heat Flux (qw) for NC (MW/sq-m)	**Contribution in %/NC**	**Heat Flux (qw) for FC (MW/sq-m)**	**Contribution in %/FC**	**Heat Flux (qw) for LH (MW/sq-m)**	**Contribution in %/LH**
−0.08507	107.74379	−0.09907	90.49713	−0.09907	14.41598
−0.07698	100.21033	−0.04619	85.54493	−0.04619	15.54493
−0.07138	94.70363	−0.0154	90.61185	−0.0154	30.61185
−0.06858	86.48184	0.00669	82.21033	0.00669	40.21033
−0.06019	76.88337	0.09596	97.74379	0.09596	47.74379
−0.04059	67.28489	0.23841	84.62715	0.23841	54.62715
−0.0154	55.62141	0.38056	80.78394	0.38056	60.78394
0.02348	40.55449	0.38056	84.53155	0.47823	64.53155
0.38056	38.6713	0.62597	80.03824	0.62597	70.03824
0.38056	35.01987	0.77932	85.85086	0.77932	75.85086
0.38056	31.5623	0.87978	89.63671	0.87978	79.63671
0.62597	28.98761	0.99705	85.44933	0.99705	85.44933
0.77932	25.88712	1.1224	91.64436	1.1224	91.64436
0.87978	19.4871	1.24526	92.63862	1.24526	92.63862
0.99705	15.18278	1.29813	92.98279	1.29813	92.98279
1.1224	13.986541	1.31341	89.98241	1.3761	100.291

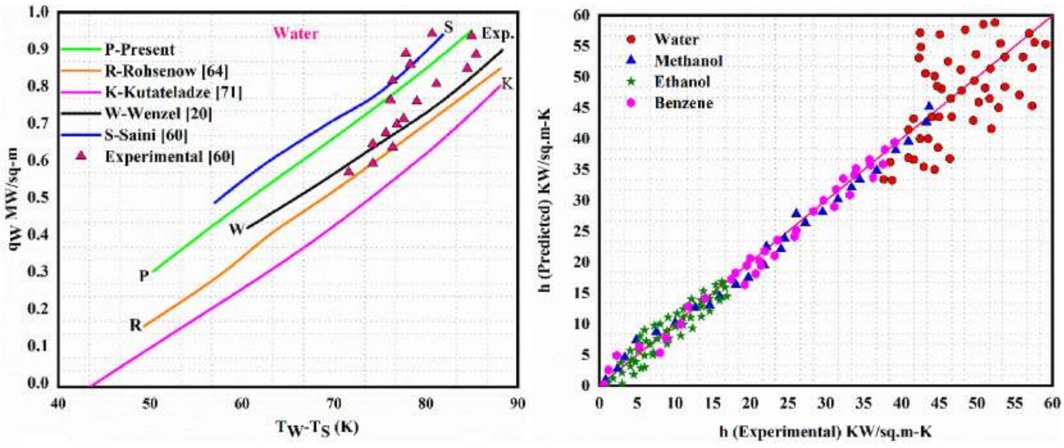

Fig. 3. Comparison of (a) heat transfer correlations; (b) experimental and predicted 'h' values.

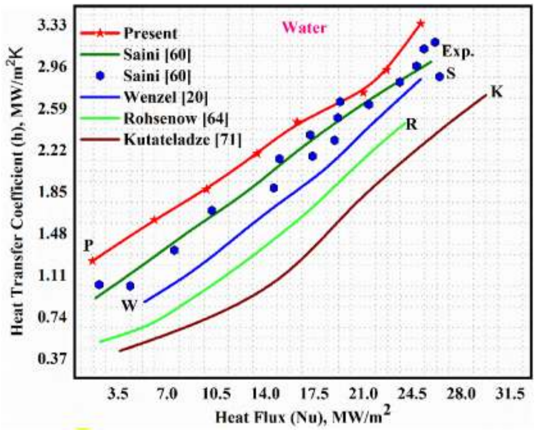

Fig. 4. Comparison of Nusselt number correlations.

experimental and for analytical one is 0.9.

4. Conclusions

As, the main purpose of the current study is to emphasize the development of a heat transfer relationship based on relative benefactions of natural convection, enhanced convection and latent heat transport. The current work stressed the absolute motion of warmth guaranteeing from the summation of each of the three segments of regular convection, upgraded convection and idle warmth transport for water and methanol with round mathematical formed. The hypotheses of air pocket development in nucleate pool bubbling hypothesize for a significant bit of warmth move to the air pocket happens by conduction through a fluid microlayer framed on the warmed surface has been thought of. The quantitative evaluation of these components has been made based on the available information on bubble dynamics to arrive at an appropriate heat transfer relationship. In the sight of above-mentioned objectives, results and discussion following conclusions have been made:

- It is clear that a single mechanism cannot explain the heat transfer phenomena in the entire nucleate boiling region because the heat transfer process varies greatly in various authorities of nucleate boiling.
- Consideration of relative contributions of natural convection, enhanced convection and latent heat transport components can represent the heat transfer phenomena in nucleate pool boiling more realistically.
- At small flux of heat values, the natural convection plays a governing role.
- In the intermediate heat flux range, all three components are important.
- In the high heat flux region, it has been found that almost all the heat is transferred due to latent heat transport.
- The maximum deviation of error between the proposed and experimental one for water, methanol, ethanol and benzene observed to be 6.89%, 5.24%, 5.64%, 6.21% respectively.
- The greatest divergence in the middle of forecasted data and investigational one for the heat transfer coefficient has been found to be ±2.54 along with the present analytical ones.
- The greatest divergence in the middle of forecasted data and investigational one for Nusselt number is found to be ± 3.27 along with the present analytical ones.
- The maximum rate point of increment for the comparative values of heat flux is 0.13 for experimental and for analytical one is 0.9.

Author Statement

Ashwini Kumar: Conceptualization, Software, Data curation, Writing – original draft preparation. Aruna Kumar Behura: Conceptualization, Validation, Data curation, Writing – original draft preparation. Dipen Kumar Rajak: Methodology, Writing – original draft preparation. Ravinder Kumar: Data curation, Writing – original draft preparation. Mohammad Hossein Ahmadi: Data curation, Supervision, Writing – review & editing. Mohsen Sharifpur: Supervision, Validation, Writing – review & editing. Olusola Bamisile: Conceptualization, Supervision, Writing – review & editing.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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