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CALCULATING THE HEAT TRANSFER COEFFICIENT OF UNSTEADY HOMOGENEOUS OIL-GAS FLOW

DR. TAWFEEQ WASMI M. SALIH
Dept. of Materials Engineering, University of Al-Mustansiriayah-Iraq

ABSTRACT

It is important to compute the heat transfer coefficient in a phase-change flow and enhance the empirical correlations to approximate the performance in many applications. In this work, the heat transfer coefficient of homogeneous two-phase flow with unsteady-state in vertical tube has been studied. The model that used to predict the flow pattern and the heat transfer coefficient for various flow velocities is determined theoretically. Some empirical relations are enrolled in order to find the fluid properties. The results are compared with that obtained experimentally and show responsible values. It is observed that the variation of heat transfer coefficient have been more oscillated and complicated as the flow velocity is increasing. This behavior could not be observed if the flow assumed as single flow, especially when the fluid seems to be gas. However, the theoretical value of void fraction has an average percentage difference ranges between (10-22) % to that obtained experimentally. While, the theoretical value of heat transfer coefficient has an average percentage difference ranges between (10-15) % to that obtained experimentally.

KEYWORDS: Heat transfer, Two-phase flow, Unsteady flow, Homogenous flow, Oil extraction

Nomenclature

A= Cross-sectional area (m^2)
C= Circumference (m)
e= Internal energy (J/kg)
h= Enthalpy (J/kg)
P= Pressure (Pa)
q= Heat transfer (W)
u= flow velocity (m/s)
w= Work done (W)
X = Dryness fraction
θ= Inclination angle (deg)
v = Kinematic viscocity in (cSt)
ρ = Density of fluid (kg/m^3)
τ_w= Shear stress at the wall of the tube (Pa)

1. INTRODUCTION

The objective of the study is to recognize the fluid properties in oil and gas extraction that deals with the two-phase flow analysis where many states (solid, liquid or gas) could be involved during the treatment, the reactions, and the transportation, hence enhance the design of pipes, valves and pumps. (Brill J. P. and Beggs H. D., 1991). Whether the two-phase flow exists in the form of different components or occurs as a result of phase change caused by evaporation or condensation of a single fluid, the void fraction is an important parameter in the analysis of pressure drop, heat transfer, and mass transfer. The prediction of the void fraction in vertical two-phase flow with reliable accuracy is essential and requires suitable methods to estimate void fraction correctly and accurately (Sarah M. Monahan and et al, 2010). The flow pattern of vertical flow may be continuous liquid with a dispersion of bubbles within the liquid so it is denoted as bubble flow. The slug flow is another pattern where the bubbles being larger and approach the circumference of the tube. There is also a possibility of churn flow where the bubbles might be destroyed to give fluctuating pattern. The annular flow is more common where the liquid being close to the tube wall as a thin layer and the gas flows in the center. When the liquid flow rate is increased, the concentration of drops in the gas core increases forming large lumps (wisps) of liquid or what called wispy annular flow (Wallis G. B., 1969).

2. BACKGROUND

As a literature review, it is important here to refer to some studies corresponding to the recent work where they gave a good background of the subject with initial indications.

Afshin J. Ghajar and Jae-yong Kim (2006) have measured the flow parameters and heat transfer coefficients of air-water upward flow in pipe of inclined angles of (2°, 5°, and 7°). The measuring was done in a steel pipe of 27.9 mm diameter where the ratio of (length/diameter) was 100. The heat flux conditions are ranging from 3000 W/m^2 to 10,600 W/m^2. The Reynolds numbers depending on superficial velocities were between 820 and 26,000 and between 560 and 48,000 for water and air respectively. A general two-phase heat transfer formula is presented depending on the experimental results obtained for different flow patterns with deviation range about (± 20%).

Abdulkadir M. and et al. (2010) have studied a combination of air-silicone oil flow in a vertical tube within a riser of 6 m length. The results that obtained experimentally is corresponded to superficial air velocities ranged from 0.047 to 2.836 m/s, and a liquid superficial velocity at 0.047 m/s. The void fraction was measured radially using a wire mesh sensor (WMS). The data recorded every 60 seconds within a range of flow conditions. However, the average void fraction was observed to vary between 0.1 and 0.9 and it is observed that the void fraction was strongly affected by the superficial gas velocity.

A downward pipe of 4.5 cm square section has used to investigate the heat transfer coefficient experimentally by Rafel H. Hameed (2013) with two superficial velocities (0.411 m/s and 1.1506 m/s) for the air-water flow. The tube was supported by eight thermocouples in-line. The recorded temperatures are used to calculate the heat transfer coefficient as well as Nusselt number for three inclination angle 5^o, 10^o, and 15^o. The heat transfer coefficient increased by increasing of water flow rate, air flow rate, and tilt angle. Many flow patterns were observed include: stratified, intermittent, and annular flow.

3. Theoretical Analysis

Navier-Stokes equations, that describe the motion of fluid, consist of a set of differential equations include many parameters like: velocity, pressure, temperature, and density of a moving fluid (Brill J. P. and Beggs H. D., 1991). It is assumed two-phase flow of one-dimensional, unsteady, homogenous, vertical-upward and no-boiling occurred. It should mention that the analysis refers to the liquid phase by subscript (l), the gas phase by subscript (g) and the mixed case by subscript (m).

The continuity equation is:

$$\frac{\partial \rho_m}{\partial t} + \frac{\partial (\rho_m\, u)}{\partial z} = 0 \tag{1}$$

The momentum equation is:

$$\rho_m \left(\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial z}\right) = -\frac{\partial p}{\partial z} - \rho_m\, g\cos\theta - \frac{C}{A}\tau_w \tag{2}$$

The energy equation is:

$$\frac{\partial}{\partial t}\left[\rho_m\left(e + \frac{u^2}{2}\right)\right] + \frac{\partial}{\partial z}\left[\rho_m\, u\left(h + \frac{u^2}{2}\right)\right] = \frac{1}{A}\left(\frac{\partial q}{\partial z} - \frac{\partial w}{\partial z}\right) - \rho_m\, u\, g\cos\theta \tag{3}$$

The internal energy is considered as:

$$e = h - \frac{p}{\rho_m} \tag{4}$$

By substitution, equation (3) becomes:

$$\frac{\partial}{\partial t}\left[\rho_m\left(h - \frac{p}{\rho_m} + \frac{u^2}{2}\right)\right] + \left(h + \frac{u^2}{2}\right)\frac{\partial\, \rho_m\, u}{\partial z} + \rho_m\, u\left(\frac{\partial h}{\partial z} + u\frac{\partial u}{\partial z}\right) = \frac{1}{A}\left(\frac{\partial q}{\partial z} - \frac{\partial w}{\partial z}\right) - \rho_m\, u\, g\cos\theta \tag{5}$$

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Hence:

$$\left(h-\frac{u^2}{2}\right)\left[\frac{\partial\,\rho_m}{\partial t}+\frac{\partial\,\rho_m\,u}{\partial z}\right]+\ \rho_m\ \left(\frac{\partial h}{\partial t}+\ u\ \frac{u^2}{2}\right)+\ \ \rho_m\ u\left(\frac{\partial u}{\partial t}-u\frac{\partial u}{\partial z}+g\cos\theta\ \right)=\ \frac{\partial p}{\partial t}+\frac{1}{A}\left(\frac{\partial q}{\partial z}-\ \frac{\partial w}{\partial z}\right) \quad (6)$$

Substitute continuity equation and momentum equation into last equation yields:

$$\frac{\partial h}{\partial t}+\ u\frac{\partial h}{\partial z}=\frac{1}{\rho_m}\left(\frac{\partial p}{\partial t}-\ u\ \frac{\partial p}{\partial z}\right)+u\ \frac{C}{A}\ \frac{\tau_w}{\rho_m}+\frac{1}{\rho_m A}\left(\frac{\partial q}{\partial z}-\ \frac{\partial w}{\partial z}\right) \quad (7)$$

Since flow pressure changes and viscous forces are small for oil extraction applications (Wallis G. B., 1969), so the terms (τ_w and dP) can be neglected in equation (7). If it assumed no work, the equation is expressed easily to be (Khalid M. A., 2013):

$$\frac{\partial h}{\partial t}+\ u\frac{\partial h}{\partial z}=\frac{1}{\rho_m A}\left(\frac{\partial q}{\partial z}\right) \quad (8)$$

Assume (φ) is the heat flux across the wall; then the heat transmitted per unit length is:

$$\frac{\partial q}{\partial z}=\pi\ d\ \varphi \quad (9)$$

The cross sectional area is;

$$A=\frac{\pi}{4}\ d^2 \quad (10)$$

Thus equation (8) will be:

$$\frac{\partial h}{\partial t}+\ u\frac{\partial h}{\partial z}=\frac{4\varphi}{\rho_m d} \quad (11)$$

The enthalpy and specific volume in term of mixed flow are:

$$h\ \ =h_l+\ \chi h_{lg} \quad (12)$$

$$\vartheta\ \ =\vartheta_l+\chi\,\vartheta_{lg}=1/\rho_m \quad (13)$$

Substitute equation (12) and equation (13) into equation (11) leads to:

$$\frac{\partial x}{\partial t}+\ u\frac{\partial x}{\partial z}=\left(\frac{\vartheta_l}{\vartheta_{lg}}+\ \chi\ \right)\Omega \quad (14)$$

Where:

$$\Omega=\frac{4\vartheta_{lg}\,\varphi}{d h_{lg}} \quad (15)$$

State equation (14) in total derivative, as following (Khalid M. A., 2013):

$$\frac{dx}{dt}=\left(\frac{\vartheta_l}{\vartheta_{lg}}+\ \chi\ \right)\Omega \quad (16)$$

Solving analytically by integrating factor method yields:

$$x=\frac{\vartheta_l}{\vartheta_{lg}}\left(\mathrm{e}^{\Omega t}-1\right) \quad (17)$$

The obtained wetness fraction (x) is very important to calculate the void fraction (α), where Hewitt and Roberts map could be used to find the flow pattern in a vertical flow (Wolverine Tube INC., 2007).

$$\alpha=\frac{1}{1+\frac{1-x}{x}\frac{\rho_g}{\rho_l}} \quad (18)$$

Now, in order to predict the two-phase heat transfer coefficient (h_{tp}), a general heat transfer formula has been developed by Ghajar A. J., and Tang C. C. (2009).

$$h_{tp}=F_p\ h_l\left[1+0.55\left(\frac{\chi}{1-\chi}\right)^{0.1}\left(\frac{1-F_p}{F_p}\right)^{0.4}\left(\frac{pr_g}{pr_l}\right)^{0.25}\left(\frac{u_l}{u_g}\right)^{0.25}(I^*)^{0.25}\right] \quad (19)$$

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The flow pattern factor (Fp) and inclination factor (I *) could be calculated by:

$$F_p = (1-\alpha) + \alpha \left[\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{\rho_g\,(ug-ul)^2}{g\,d\,(\rho_l-\rho_g)}}\right)\right]^2 \tag{20}$$

$$I^* = \frac{(\rho_l-\rho_g)gd^2}{\sigma}\,|\sin\theta| \tag{21}$$

The viscosity ratio (μ_g / μ_l) *is* estimated to be (0.015) and Prandtle ratio (pr_g / pr_l) is equal to (0.08) as mentioned by (Afshin J. Ghajar and Jae-yong Kim, 2006). The heat transfer coefficient for single phase(h_L) could be found from the known formula of Sieder and Tate (Afshin J. Ghajar and et al, 2004) below, while, Reynolds number and prandtle number are depending on liquid propetries and based preliminary on kinematic viscosity which obtained from experimental data of crude oils (Marisa F. Mendes and et.al., 2005).

$$h_l = 0.027\, Re_L{}^{4/5} Pr_L{}^{1/5}\left(\frac{K_L}{d}\right) \tag{22}$$

$$\log(\log(\nu+0.7)) = a + b\,(T_B)^C + d\,\log(T) \tag{23}$$

with a = 5.489, b = 0.148, c = 0.5 and d = -3.7.

4. RESULTS AND DISCUSSIONS

The flow properties and operation conditions for the proposed model are listed in table (1) with certain values related to the typical row oil (Kendoush A., Ghanim K. and B. Yaqob, 2006).

Table (1): Operation conditions

Symbol	Definition	Value
C_p	Specific heat capacity	2130 J/kg.K
k	Thermal conductivity	0.138 W/m.K
ρ	Density	843.3 kg/m^3
φ	Heat flux	15 kW/m^2
σ	Surface tension	0.0278 N/m
μ	Dynamic viscosity	6.2 cP
T	Temperature	70 ^{o}C
T_B	Boiling temperature	210 ºC
d	Tube diameter	7 cm

The procedure of the calculation includes definition of flow properties at the entrance of the tube. The set of equations were solved for each time step until steady state is satisfied. The results are recorded each 60 seconds as a plan of work (Khalid M. A., 2013) in order to compare the results with that denoted by Abdulkadir M. and et. al. (2010). Various flow velocities are taken to calculate the void fraction values. It has been observed terminally in figure (1), that void fractions are approached a constant value of (0.92). The transition stages are quite rapid in the range of (0.2-0.75) void fraction. Figure (2) represents the heat transfer coefficient at range velocities between (0.05 m/s-0.20 m/s). The values of heat transfer coefficient for two-phase flow initially increased until maximum values, then decreased slowly. This behavior is due to the viscosity of the fluid, where initially is liquid, where viscosity of liquid decreases as temperature increases and that effects on the value of Reynolds number, hence leads to increase the heat transfer coefficient. The inverse of this behavior is happened when the fluid seems to be gas (x>0.5) because the viscosity of gas increases as temperature increases, and this behavior cannot be observed if the flow assumed as single. The variation between the assumption of single flow and two-phase flow can be appeared in figure (3), where heat transfer correlations were that obtained empirically are more reliable. Figure (4) represents comparison between the results obtained by recent theoretical work with that adapted from experimental results of Abdulkadir M. and et. al. (2010) for both void fraction and heat transfer coefficient. The slight difference between theoretical results and experimental results is due to the effect of the theoretical

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assumptions with many simplifications indeed, and the differences in oil properties like the density and the superficial velocity of the flow. However, the behavior of theoretical void fraction function is quite linear rather that of experimental void fraction which appeared as polynomial of second degree. This is mentioned by Somchai Wongwises and Paisan Naphon, (2000) and Nilanjana B., Andrey T. and Greg N., (2003), who suggested that a program like FLUENT could be used to get reasonable results. In the recent study, the theoretical values are approached to that recorded experimentally when the void fraction is closed to (0.9) and the difference is about (22%) when (α<0.5) and (10%) when (α>0.5). Same indications are observed by Nicklin D. J., Wilkes J. O. and Davidson J. F, (1962). In general, the theoretical value of heat transfer coefficient has an average percentage difference ranges between (10%-15%) to that obtained experimentally.

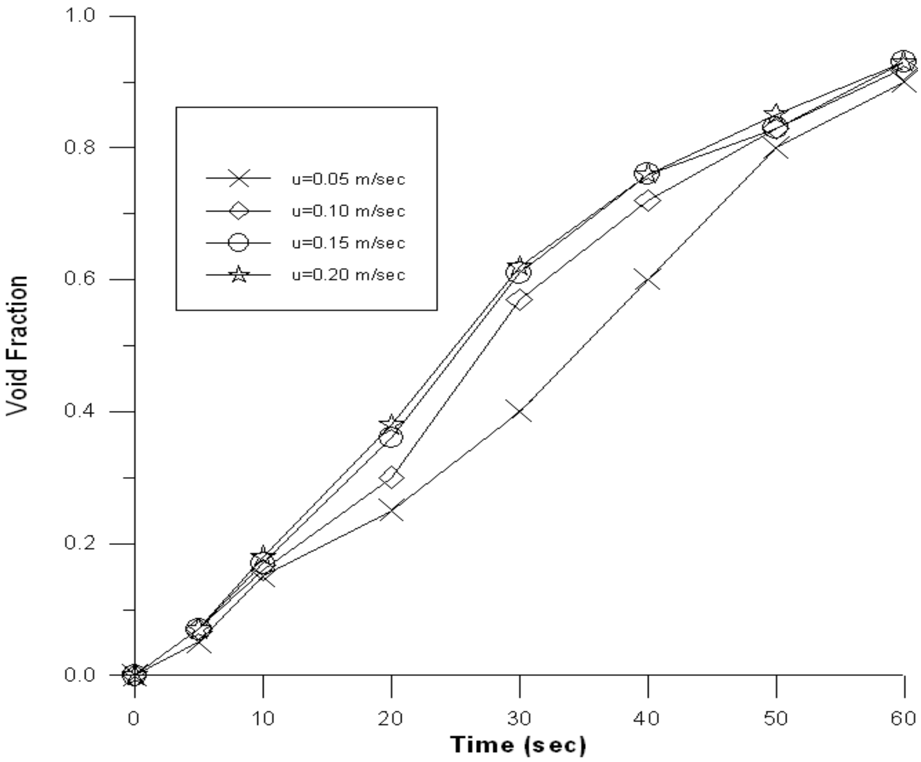

Fig. (1): Void fraction for various flow velocities

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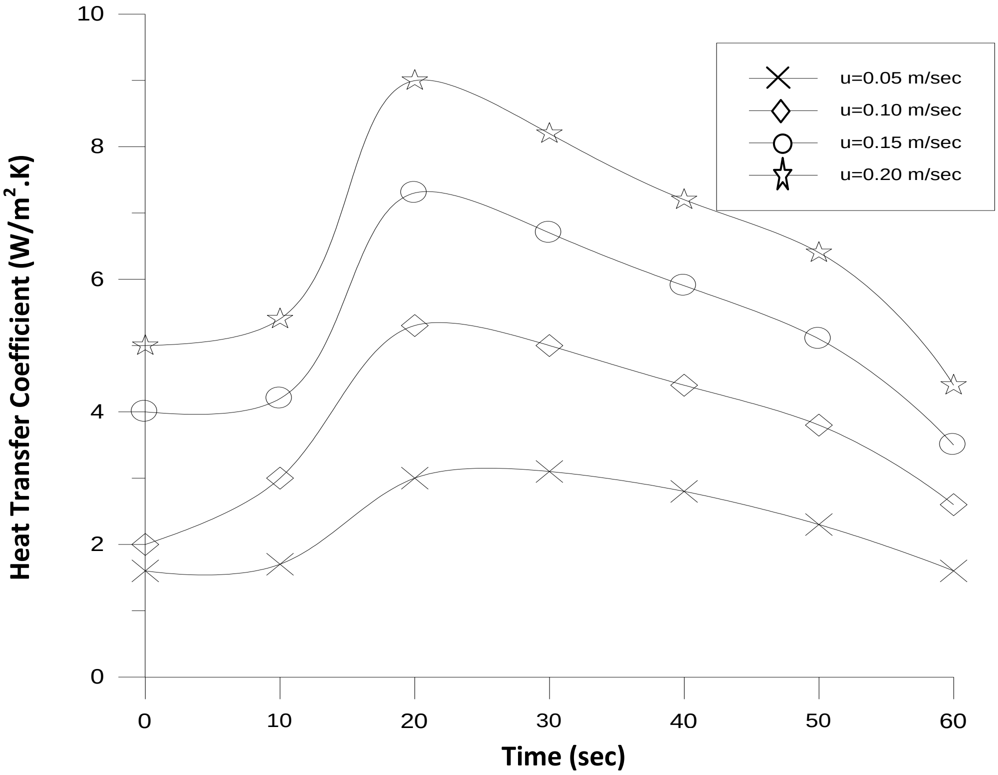

Fig. (2): Heat transfer coefficient for various flow velocities

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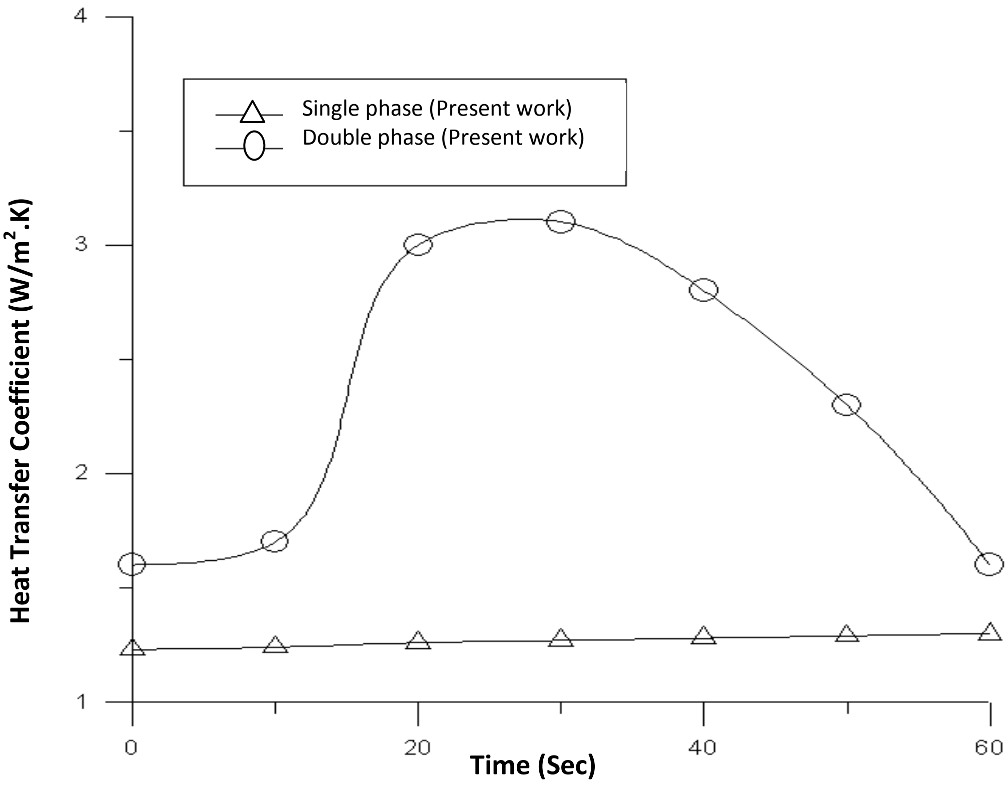
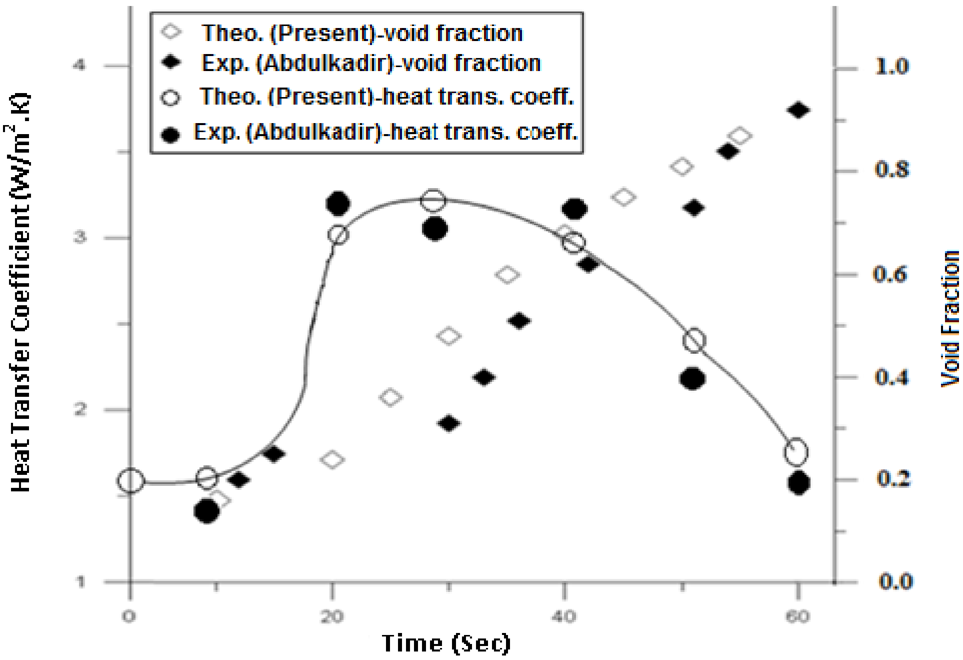


Fig. (3): Comparison for heat transfer coefficient at u=0.05 m/s



Fig, (4): Comparison between theoretical and experimental results at u=0.05 m/s

5. CONCLUSIONS

The effects of heat transfer coefficient due to flow pattern transition for oil-gas two-phase flow have investigated. Starting from the Navier-Stokes equations, the controlling of two-phase flow is described by the fluid properties, and flow characteristics: velocity, pressure and temperature. Even though that an analytical approach has used to solve the equations with a suitable program, but the results have been compared with experimental data and achieved an acceptable agreement. The results that obtained from the theoretical research have several advantages, where it showed the difference when the flow is assumed as a two-phase flow in place of single flow because of the essential effect of the viscosity which is related to the temperature and the corresponding phase change. The theoretical value of heat transfer coefficient has an average percentage difference ranges between (10-15) % to that obtained experimentally. Regarding to void fraction, theoretical and experimental values are closed each other at (α=0.9). The average percentage difference between theoretical and experimental values is (22%) for (α<0.5) and (10%) for (α>0.5).

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