◎ 研究論文

An Experimental Investigation on Condensation Heat Transfer Inside Vertical Tubes

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수직관내 응축열전달에 관한 실험적 연구

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Key words : Condensation Heat Transfer, Non - annular Model, Refrigerant, Pressure Drop, Double Pipe Heat Exchanger

Abstract

냉동 · 공조 및 각종 화학공업에 널리 사용되는 열교환기인 응축기의 고성능화 및 합리적인 설계를 위해서는 냉매의 정확한 응축열전달률 예측과 그 메카니즘 규명이 필수 요건이다. 본 연구에서는 내경 9.7mm, 외경 12.7mm, 길이 1200mm의 수직 이중관 응축기의 압력강하 및 응축열전달특성을 실험적으로 밝혔다. 실험으로부터 Lockhart - Martinelli의 상관 관계식을 이용한 수직 응축관내 압력강하 특성을 종래의 실험식들과 비교 · 검토하고 새로운 압력강하식을 제안하였다. 그리고 종래의 해석방법과는 달리 비환상류 모델을 가정한 해석결과로부터 전 유동양식에 걸쳐 적용할 수 있는 새로운 응축열전달 예측식을 제안하였다.

Nomenclature

A	: Flow area	[m²]
c_p	: Specific heat at constant pressure	[kJ/(kgK)]
D	: Inside tube diameter	[m]
g	: Acceleration due to gravity	[m/s²]
G	: Mass velocity	[kg/(m²s)]
h	: Heat transfer coefficient	[W/(m²K)]
h*	: Dimensionless heat transfer coefficient	[-]
h_f	: Heat transfer coefficient in annular region of the slug unit	[W/(m²K)]
h_{tp}	: Heat transfer coefficient in the slug unit	[W/(m²K)]
λ	: Thermal conductivity	[W/(mK)]

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λ_w : Thermal conductivity of the wall [W/(mK)]
L_v : Length of vapor bubble in the slug unit [m]
L_l : Length of liquid slug in the slug unit [m]
Nu : Nusselt number [-]
P : Pressure [kPa]
Pr : Prandtl number [-]
q : Heat flux [W/m^2]
Re : Reynold number
T : Temperature [K, ℃]
T^+ : Dimensionless temperature
U_v : Mean velocity of vapor bubble in the slug unit [m/s]
U_l : Mean velocity of liquid slug in the slug unit [m/s]
W : Flow rate [kg/h]
X : Quality
y_i : Distance from wall to the interface [m]
y^+ : Dimension distance from wall
z : Coordinate, distance from vapor inlet [m]

Greek Symbols

α : Void fraction [-]
β : Ratio between length of vapor bubble and length of slug unit [-]
δ_w : Thickness of the tube [m]
ν : Kinematic viscosity [m^2/s]
η : Fujii parameter[$=(\rho_v\rho_l)^{1/2}$] [kg/m^3]
ρ : Density [kg/m^3]
σ : Surface tension [Pa]
τ_o : Wall shear stress [Pa]
τ_o^* : Dimensionless wall shear stress [-]
μ : Viscosity [kg/(ms)]
Φ_v : Friction pressure drop multiplier [-]
X_{tt} : Lockhart - Martinelli parameter [-]

Subscripts

c : Coolant
f : Film
F : Friction
i : Interface, inlet
l,f : Liquid, film
m : Mean, momentum
o : Outlet, inside tube wall
p : Pressure
s : Saturation

1. Introduction

Condensation heat transfer coefficient varies depending on the geometry of the surface, the flow rate of condensate, physical properties, and thickness of the liquid film. A great amount of researches, since the first work by Nusselt[1] in 1916, have been conducted by numerous workers to elucidate the mechanism of filmwise condensation heat transfer, and many empirical or semi - theoretical predictions have been reported. Until today, however, there is no available general predictions concerned with condensation inside tubes. Borishanskiy et al.[2], Bell et al.[3], and Oh[4], showed that there are no agreement in the reported correlations. They concluded that the discrepancy is attributed to the different flow regimes that existed during the development of the various heat transfer correlations and the determination of the flow regime is one of the most important steps in proper prediction of tube side condensing heat transfer coefficients. The condensation proceeds along the condenser, the liquid flow rate increases and vapor flow rate correspondingly decreases, and a gradual transition may occur to a flow pattern which is non - annular flow regime. Some investigators analyzed this by regarding liquid film as a turbulent boundary layer, but none has considered the characteristics of non - annular flow. In this study, the condensing heat transfer across the liquid film is analyzed by the introduction of shear stress on the liquid - vapor interface and the hydrodynamic behavior of non - annular flow regime. The purpose of this investigation is to give a new model for predicting condensation heat transfer including non - annular flow, and thereby increase on understanding of the effects of non - annular flow upon the condensing flow. Based on this model, a new correlation for the condensing heat transfer is developed.

2. Experimental Apparatus and Method

A schematic diagram of the experimental apparatus is shown in Fig. 1. The test fluid side of the apparatus consists of a closed - loop refrigerant flow circuit driven by a refrigerant pump. Cooling water was continuously circulated in the coolant loop. The equipment in the R - 11 circuit consisted of a boiler, superheater, test section, condensate cooler, condensate flow measuring station, and reservoir. The boiler was a tank with immersion heaters at the bottom of the tank. The super heater was a tube heater with nichrome wire spirally wrapped around the vapor line. Down stream of the test section, an auxiliary condenser was provided to ensure fully condensed refrigerant at the pump inlet.

As shown in Fig. 2, the test section itself is an annular shaped double pipe heat exchanger, the vapor condensed inside the inner tube and cooling water flowed in the annulus between the inner and outer tubes. The total length of the test section was 1200mm. It was divided into four 300mm long section of 9.7mm I.D. Each section has a separate cooling water circuit. 4 thermocouples are placed in the middle of the

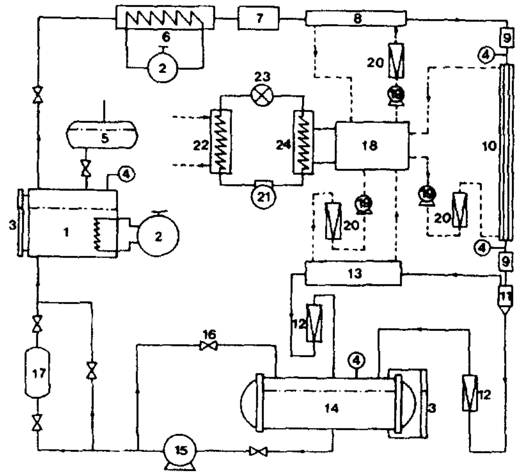

Fig. 1 Schematic diagram of experimental apparatus

1. Boiler	13. Auxiliary condenser
2. Voltage transformer	14. Reservoir
3. Liquid level indicator	15. Liquid pump
4. Pressure gage	16. By - pass valve
5. Liquid supply tank	17. Strainer
6. Superheater	18. Coolant tank
7. Calming section	19. Coolant pump
8. Precondenser	20. Coolant flow meter
9. Sight glass	21. Compressor
10. Test section	22. Condenser
11. Vapor liquid separator	23. Expansion valve
12. Refrigerant flow meter	24. Evaporator
— : Refrigerant line	----: Coolant line

300mm sections. All temperatures were measured with 0.32mm in diameters nylon - sheathed copper and constantan wire. The coolant thermocouples were placed at the midpoint of the annular gap between the inner and outer tubes with the leads projecting radically outward. All temperatures were recorded on a self - balancing data logger. Total condensate flow rate was measured volumetrically. The system consisted of a quick - shut off valve located downstream of a 1200mm long section of tubing. The tube was equipped with a sight glass that allowed visual observation and timing of the liquid - level rise when the valve was closed. The temperature of condensate at the

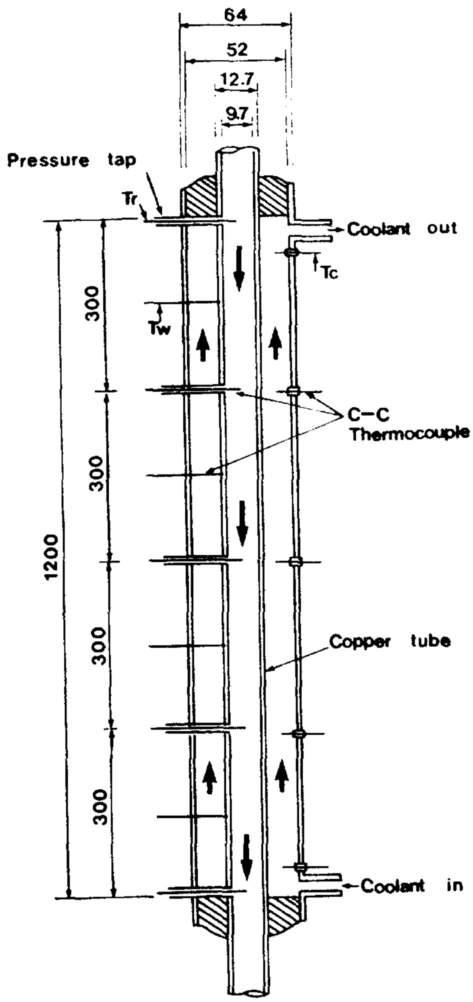

Fig. 2 Sectional view of test section

flow - measuring station was recorded so that the flow rate could be evaluated in mass unit. Coolant flow rate was measured with flow cell type flow meter. Prior to the acquisition of the experimental data, noncondensable gases were removed from the test loop and U - tube manometers attached at the test section. All data were taken after the facility reached a steady state condition where the vapor inlet and coolant exist temperatures had not changed for a period of at least 30 minutes. System pressure did not change when the quick shut off valve of the condensate flow rate measuring station was closed, therefore it was assured that the flow remained constant throughout the flow measuring operation. The heat flux of the test section was obtained from the coolant flow rate and the temperature change. The condensing wall temperature was determined from the outside tube wall temperature and the heat flux.

3. Experimental Results

3.1 Data analysis and heat balance

The experimental data obtained in the test are tabulated in Fig. 3. The experimental measurements were used to calculate local heat fluxes, local condensing heat transfer coefficients, mean condensing heat transfer coefficients, and overall pressure drops. The local condensing heat transfer coefficient, h, was calculated from the following relation

$$h = q/(T_s - T_o) \tag{1}$$

where heat flux, q, was taken to be the product to the coolant flow rate and the coolant temperature rise, T_s, is the saturated vapor temperature corresponding to the measured vapor pressure. The inner wall temperature of inner tube, T_o, heat flux, q, are calculated from the following relations :

$$q = W_c \cdot c_{pc} \cdot \Delta T_c/(\pi \cdot z \cdot D) \tag{2}$$

$$T_o = T_w + D \cdot q \cdot \ln(1 + \delta_w/D)/(2\lambda_w) \tag{3}$$

where T_w, z, D, δ_w and λ_w are outside tube wall temperature of inner tube, tube length, diameter, thickness of the tube and heat conductivity of the tube, respectively. The temperature of the condensate was assumed to be calculated by the following equation from the reference.[5]

$$T_l = T_o + 0.3(T_s - T_o) \tag{4}$$

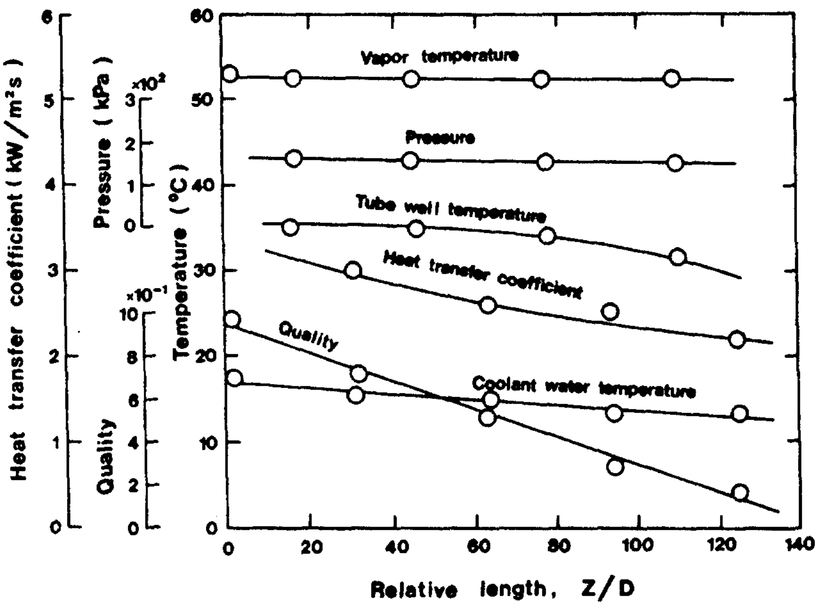

Fig. 3 Distribution of temperature, quality, pressure and heat transfer coefficient through test section in a single run, $G_t = 130 kg/m^2s$

where T_s is saturation vapor temperature. The refrigerant properties were obtained from the Dupont[6] and ASHRAE Data Book.[7]

The pressure drops along the test section were obtained from the pressure drop between the inlet tap and any of the pressure taps along the test section. The assumption was made the change in the temperature of the condensate and the vapor was the same as the change in the vapor saturation temperature. The range of the parameters are shown in following Table 1.

The apparatus was well insulated to minimize any error owing to heat loss from the test section. The heat balance check was based on a comparison of the heat gain by the cooling water flowing in the annular with the heat lost by the refrigerant vapor flowing inner tube, since these could be taken with considerable accuracy. Fig. 4 shows the comparison of the

Table 1. Range of the parameters.

Parameters	Range
Absolute pressure at entrance, p_i(kPa)	40 – 182
Vapor quality at entrance, X_i	1
Vapor quality at exit, X_o	0.1 – 0
Total refrigerant mass velocity, G_t(kg/m²s)	70 – 160
Coolant flow rate, W_c(kg/h)	100 – 450
Coolant inlet temperature, T_{ci}(℃)	10 – 15
Vapor velocity at entrance, u_{vi}(m/s)	10 – 35

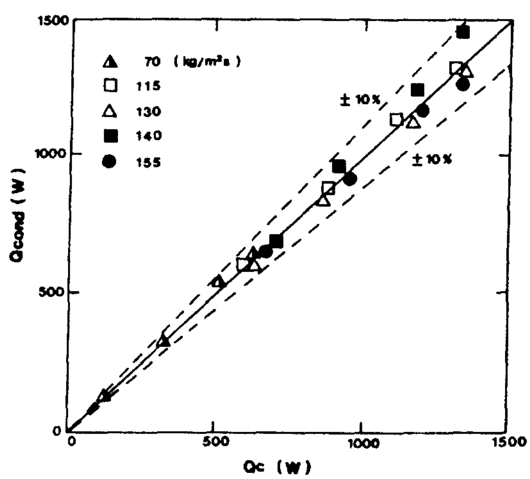

Fig. 4 Heat balance of the test section

heat gain by the coolant in the annulus, Q_c, with the heat loss of the refrigerant flowing in the inner tube, Qcond. As shown in the Fig. 4, in most runs, the heat balance error was less than 10 percent.

3.2 Pressure drop

During recent years a large number of empirical correlations for prediction of pressure drops in two phase flow have been postulated on the basis of experimental results, but even today there is still no satisfactory general correlation. One of this reason is that empirical or semi - theoretical relationships for pressure drop are only limited applicability because of the very complicated phenomena of two phase flow. In general, determination of the frictional pressure drop is important for the flow designing. However even today the pressure drop due to the friction of condensing two phase flow in a vertical tube can not be measured directly, and its calculation is still quite imprecise. Lockhart and Martinelli[8] established the relation between, friction pressure drop multiplier, ΦV and Lockhart - Martinelli parameter X_{tt} from the pressure drop measurements of the adiabatic two phase, two component flow in tubes. In a general way, the frictional pressure drop in two phase flow is referred to that of a single phase flow at the same total mass flow rate. A large number of works and calculation models have been reported on the friction pressure drop and friction pressure drop multiplier in two phase flow. But even today the most widely used method calculating the two phase pressure drop is the Lockhart and Martinelli relationship. Lockhart and Martinelli introduced the following parameter, pressure drop multiplier :

$$\Phi_v = [(dp/dz)_F/(dp/dz)_v]^{1/2} \tag{5}$$

where $(dp/dz)_F$ is the two phase friction pressure drop. and $(dp/dz)_v$ is the pressure drop assumed vapor phase flows alone. Equation (5) indicates how many times larger the friction pressure drop in a two phase flow than in a single phase flow. Lockhart and Martinelli further defined a new parameter X_{tt} such that :

$$X_{tt} = [(dp/dz)_l/(dp/dz)_v]^{1/2} = \left[\frac{\mu_l}{\mu_v}\right]^{0.1}\left[\frac{1-X}{X}\right]^{0.9}\left[\frac{\rho_v}{\rho_l}\right]^{0.5} \tag{6}$$

where $(dp/dz)_l$, $(dp/dz)_v$ are the friction pressure drops computed by assuming each phase flowing alone in the tube. For the investigation of the condensing two phase pressure drop the well known empirical correlation of Lockhart - Martinelli often correlated by the following equation :

$$\Phi_v = 1 + aX_{tt}^{\,b} \tag{7}$$

Fujii et al.[9] from their experiment during condensation R - 11 inside vertical tubes proposed by the equation :

$$\Phi_v = 1 + aX_{tt}^{\,0.2} \tag{8}$$

where, $a = 1.24(G/\eta)^{0.7}$, $G/\eta \leq 1.5(m/s))$, $a = 1.65$, $G/\eta > 1.5(m/s)$, where η is defined as, $\eta = (\rho_v\rho_l)^{1/2}$, The main difficulty in solving Equation (7) is the estimation of constant a in second term of the right hand side.

Fig. 5 shows the comparison of Fujii parameter, G/η, and correlation for friction pressure drop, $(\Phi_v - 1)/X_{tt}^{\,0.2}$. As shown in the Fig. experimental data are best fitted when $a = 2.4$. If we fix the power of Lockhart - Martinelli parameter is 0.2, the Equation (8) can be written as :

$$\Phi_v = 1 + 2.4X_{tt}^{\,0.2} \tag{9}$$

Fig. 6 shows the comparison of the experimental data with the calculated results by the Equation (9). As shown in the Fig. 6, it is evi-

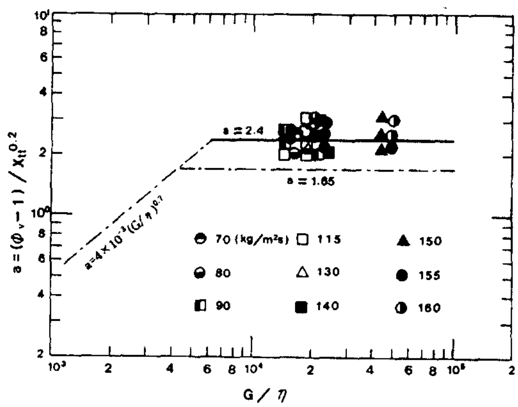

Fig. 5 Comparison of Fujii parameter, G/η, and correlation for friction pressure drop, $(\Phi_v - 1)/X_{tt}^{0.2}$)

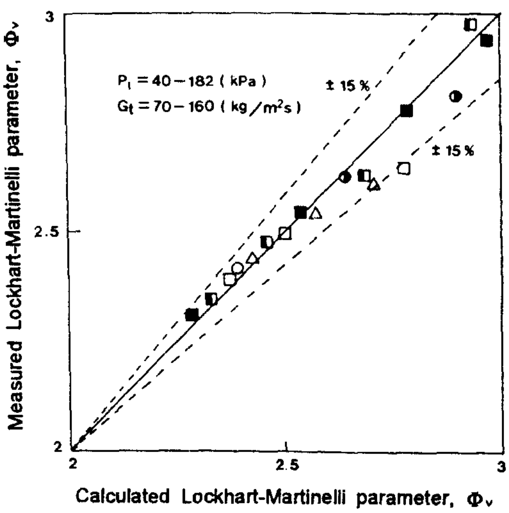

Fig. 6 Comparison of calculated and measured Lockhart - Martinelli parameter

dent that the great majority of the data are correlated by the Equation (9) within ± 15 percent.

3.3 Heat transfer coefficient

When condensation occurs inside a tube, the local condensing heat transfer coefficient can vary appreciably depending upon how the condensate is distributed within the tube. Generally, the flow passes through several different

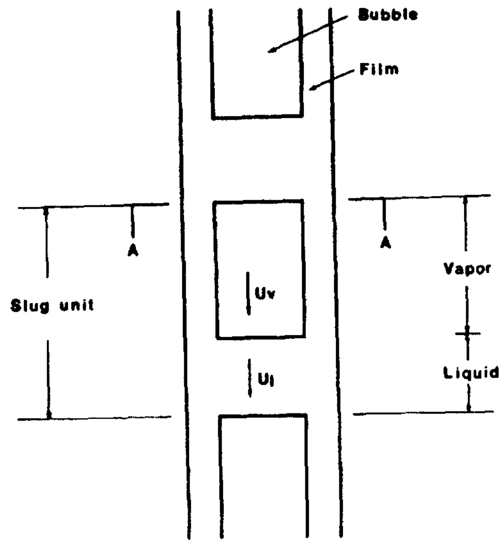

Fig. 7 Physical model for non - annular flow

patterns as the vapor proceeds from the tube inlet to the exit. For this purpose, however, the flow patterns during condensation of pure components inside vertical tubes are separated into two regimes : annular and non - annular. As the condensation proceeds along the condenser, the liquid flow rate increases and vapor flow rate correspondingly decreases. A gradual transition may occur to a flow pattern which is non - annular. The idealized non - annular flow model is presented as shown in Fig. 7. The model is based on cocurrent one - dimensional flow in steady state and equilibrium conditions.

If we assume this model in condensing problem inside a vertical tube, the heat transfer coefficient around the perimeter of cross section A - A, as shown in Fig. 7, can be calculated from the time average through the slug unit, which can be expressed as :

$$h_{tp} = \frac{\int_0^{\Delta t_1} h_f dt + \int_{\Delta t_1}^{\Delta t_1 + \Delta t_2} h_c dt}{\Delta t_1 + \Delta t_2} \tag{10}$$

where Δt_1 denotes the time interval that a vapor bubble with an annular liquid film passes across section A - A. It can be calculated from vapor bubble velocity, Uv, and the length of the vapor bubble, L_v, as :

$$\Delta t_1 = L_v/U_v \tag{11}$$

and Δt_2 is the time interval that the liquid slug passes through section A-A in Fig. 7, calculated as :

$$\Delta t_2 = L_l/U_l \tag{12}$$

where L_l is the length of the liquid slug, U_l is mean velocity of liquid slug in the slug unit. In Equation (10), hf is the annular-film condensation heat transfer coefficient and h_c is the convective heat transfer coefficient for single phase liquid flow. Assuming that h_f and h_c are constants with respect to time, Equation (10) can then be written as :

$$h_{tp} = h_f\beta + h_c(1-\beta) \tag{13}$$

or in terms of Nusselt number

$$Nu_{tp} = Nu_f\beta + Nu_c(1-\beta) \tag{14}$$

where β is ratio of volume void fraction, α_v, and void fraction, α, defined as :

$$\beta = \alpha_v/\alpha \tag{15}$$

where $\alpha = A_v/A$, $\alpha_v = V_v/V_T$, here A is the cross section area of tube, A_v is the cross section area occupied by vapor in the slug unit, V_T and V_v are total volume and volume of the vapor in the slug unit, respectively. The convective heat transfer coefficient, h_c, can be obtained from the well known Dittus-Boelter correlation for liquid-forced convection inside tubes, given by[10]

$$Nu_c = 0.023Re_l^{0.8}Pr_l^{0.3} \tag{16}$$

Equation (13) is based on two postulated heat transfer mechanisms. One is the microscopic heat transfer, associated with the bulk movement of the vapor and liquid, and its contribution to the total heat transfer coefficient is given by the first term of Equation (13). And the other is the microscopic heat transfer, associated with the liquid phase convection heat transfer, and its contribution to the total heat transfer coefficient is given by the last term of the equation. If T_i is the interface temperature and T_o is the temperature of the tube inside wall, by definition, the heat transfer coefficient is defined as follow :

$$h_l = q_o/(T_i - T_o) \tag{17}$$

The local heat transfer coefficient based on the interface can be expressed as follows :

$$h_l = \rho_l u^*/T_i^* = \frac{\rho_l c_p}{T_i^+}\left[\frac{\tau_o}{\rho_l}\right]^{1/2} \tag{18}$$

In equation (18) local heat transfer coefficient is a function of dimensionless interface temperature. This equation can be easily converted to :

$$h^* = Pr_l\left[\tau_o/\{\rho_l(g\nu_l)^{2/3}\}\right]/T_i^+ \tag{19}$$

where h* is the dimensionless heat transfer coefficient defined by :

$$h^* = h_i(\nu_l^2/g)^{1/3}/\lambda_l \tag{20}$$

In Equation (18), local heat transfer coefficient is a function of dimensionless interface temperature and wall shear stress.

Using the assumption of Carpenter et al.[11] in a liquid layer(the major thermal resistance occurs in a lamir sublayer of the condensate film), one may be written dimensionless distance from the wall to the interface $y_i^+ = c/Pr_l^n$, and $q/q_o = 1$ in the turbulent liquid film. In this case the dimensionless temperature drop across the liquid film, T_i^+ is :

$$T_i^+ = \int_0^{y_i^+} Pr_l dy^+ = Pr_l y_i^+ = cPr_l^{1-n} \tag{21}$$

where $y_i^+ = (y_i/\nu_l)(\tau_o/\rho_l)^{1/2}$, Substituting Equation

(21) into Equation (20)

$$h^* = cPr_l^n \tau_o^{*1/2} \quad (22)$$

where c, n, and τ_o^* are the empirical constant, power of Prandtl number, and dimensionless wall shear stress, respectively. Dimensionless wall shear stress is defined by the following equation :

$$\tau_o^* = \tau_o / \{\rho_l (g\nu_l)^{2/3}\} \quad (23)$$

The dimensionless heat transfer coefficient, h^*, is a function of the dimensionless wall shear stress, τ_o^*. The relation dimensionless heat transfer coefficient, h^*, and the dimensionless wall shear stress, τ_o^*, are shown in Fig. 8. The local heat transfer coefficient predicted by Carpenter and Colburn[11] were drawn as chain line and Soliman et al.[12] were drawn as two - dot chain line. As shown in the Fig., the data scattered with these predictions. A cause of the poor agreement with the data is due to the assumption of a two - layer model for interface friction force. The solid line reveals following equation, yields as :

$$h_{tp} = 0.99h_f + h_c(1 - 0.99) \quad (24)$$

Substituting Equation (24) into Equation(20), we get a new prediction as follows :

$$h^* = [0.99h_f + h_c(1 - 0.99)] + (\nu_l^2/g)^{1/3}/k_l \quad (25)$$

The solid line in the Fig. represents the prediction based on Equation (24). The equation satisfied experimental data as well as Oh's data.[13] Equation (25) can be applied to predict the condensation heat transfer in the entire condensation path including both annular and non - annular regimes.

Fig. 9 shows the Equation (19) compared with previous studies. The general form of the condensing heat transfer coefficient given by Equation (19) is developed by the dimensionless temperature drop across the liquid film, Equation (21), and dimensionless wall shear stress, Equation (23), as was done by Traviss et al.[14]. The data of Altman et al.[15] correspond with the Traviss et al. data in the region of $\tau_o^* \geq 20$ and experimental data agree with the region of $10 \leq \tau_o^* \leq 40$ as like Bae et al.

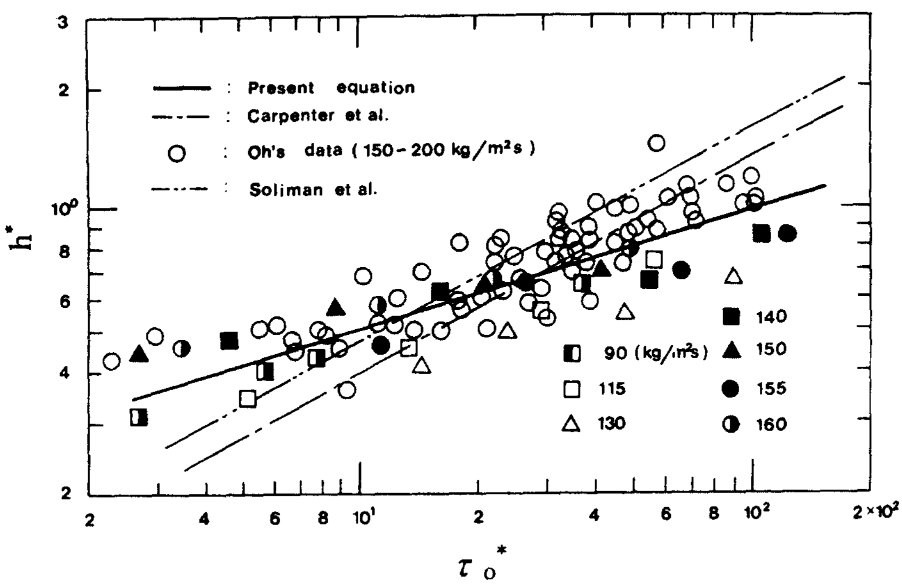

Fig. 8 Dimensionless heat transfer coefficient, h^*, vs. dimensionless shear stress, τ_o^*

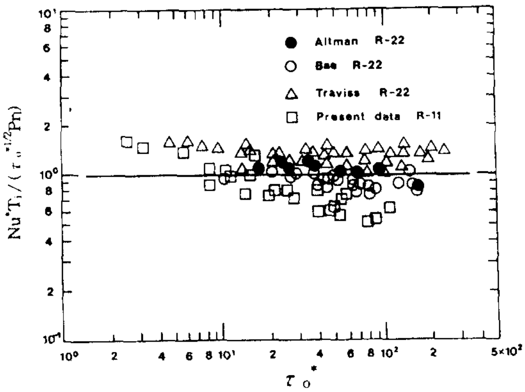

Fig. 9 Value of Traviss analysis, $Nu^*T_i/(\tau_o^{*1/2}Pr_l)$ vs. dimensionless shear stress, τ_o^*

4. Conclusions

The results of the investigation for condensing heat transfer in vertical downward flow within smooth double pipe heat exchanger may be summarized as follows :

1. The flow patterns are a major factor in determining the heat transfer mechanisms. Non - annular flow effects in condensing two phase flow is an important problem and must be analyzed accordingly.
2. In the estimation of friction pressure drop defined by Lockhart and Martinelli is proposed in the recommendable equation.
3. Equations developed on the basis of non - annular flow model can be used to predict the hydrodynamic behavior of non - annular flow, as well as annular which is necessary for heat transfer predictions.
4. By using the non - annular flow model proposed equation can be applied to predict the condensation heat transfer coefficient in the entire condensation path including both annular and non - annular regimes.

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