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STUDY OF THE PERFORMANCE OF PLATE HEAT EXCHANGER(PHE) (LOW PRESSURE AND LOW REYNOLDS NUMBER)

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SUMMARY - In this paper, condensation inside a vertical tube is analysed theoretically, and the main factors of condensation in a plate heat exchanger (PHE) are studied experimentally. The experimental results show that under low pressure and low Re conditions, heat transfer by PHE is satisfactory.

1. INTRODUCTION

Because of its well-known advantages, heat transfer by PHE of liquid/liquid phase has already been studied widely. In view of heat transfer and flow resistance, a process with a phase change is superior to that of a continuous phase (Huang and Wei, 1988). Since 1970's using PHE as a condenser has been studied and developed (Clark, 1974). *So* far, literature search shows that only pressures above 1 bar gauge have been investigated, and there is no information on the heat transfer performance of PHE as a condenser under lower pressures. Hence, low pressure and low Re conditions of heat transfer performance of plate condenser were chosen for this theoretical and experimental study.

2. THEORETICAL ANALYSIS

It is difficult to find a good theoretical model for steam condensation in the channel between two plates (Kumar, **1979).** Here, the vital problem is the complexity of the phenomenon taking place because both the condensation between plates and inside a tube represent condensation in restricted channel, and the flow in the channel influences the liquid film. The forces involved are gravity and shear stress at the wall and at the steam-liquid interface. The theoretical analysis **of** steam condensation in a vertical tube is considered here for an experimental study of the condensation between plates.

For analysis, the model of steam condensation inside a vertical tube is simplified as shown in Figure 1 and equation (1). Equation (1) is obtained in terms of force relationship.

$$\tau_w = \tau_i + \rho_f g y_f \quad (1)$$

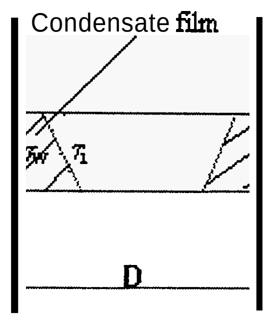


Figure 1: Steam condensation inside a vertical tube

Laminar and turbulent analyses have been carried out. For a turbulent film, assuming a three-layer flow model of Von Karman (1934), Huang and Wei (1988) used the following dimensionless equations (2a, 2b, 2c).

$$u^+ = y^+ \qquad y^+ \le 5 \quad (2a)$$

$$u^+ = -3.05 + 5 \ln y^+ \qquad 5 < y^+ \le 30 \quad (2b)$$

$$u^+ = 5.5 + 2.5 \ln y^+ \qquad y^+ > 30 \quad (2c)$$

For a laminar film, the following equation (3) was used (Huang and Wei, 1988).

$$u^+ = y^+ - \frac{1}{2}\left(\frac{y_i^*}{y_i^+}\right)^3 \left(y^+\right)^2 \quad (3)$$

Using momentum-heat analogy and dimensionless analysis, the relationship between temperature and thickness of liquid film can be found. The following are the equations (4a, 4b, 4c) for turbulent flow based on von Karman three-layer flow model (Huang and Wei, 1988).,

$$T^+ = Pr\, y^+ \qquad y^+ \le 5 \qquad (4a)$$

$$T^+ = Pr\, y^+ + 5\ln(y^+/5 - 1) \qquad 5 < y^+ \le 30 \qquad (4b)$$

$$T^+ = Pr\, y^+ + 2.5\ln(y^+/2.5) \qquad y^+ > 30 \qquad (4c)$$

and the laminar flow is given by equation (5).

$$T^+ = Pr\, y^+ \qquad (5)$$

Heat transfer coefficient, α_i , is defined by the condensate temperature difference, $\Delta t = T_w - T_i$, and is given by equation (6).

$$\alpha_i = \frac{q_w}{T_w - T_i} \qquad (6)$$

Local Nusselt Number and average Nusselt Number are expressed by the following equations (7 and 8) respectively.

$$Nu_i = \frac{\alpha_i D}{\lambda_f} = \frac{Pr_f \rho_f}{u_f T^+}\sqrt{\frac{\tau_w}{\rho_f}} \qquad (7)$$

$$Nu = \frac{1}{L}\int Nu_i\, dl \qquad (8)$$

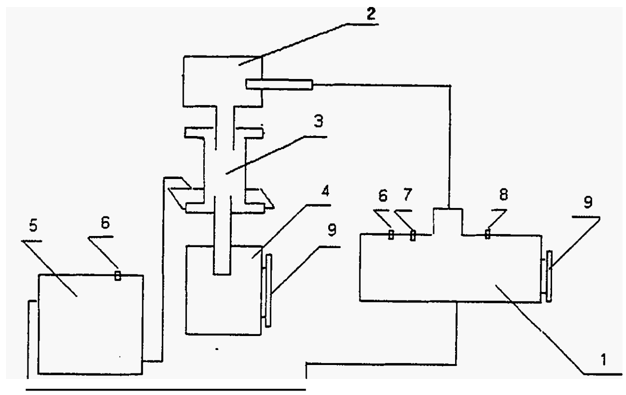
Equations (1) to (8) show that the thickness of film influences heat transfer directly. In order to find the other influential factors, an equation of the form as shown in equation (9) should be derived (to find a theoretical relationship).

$$Nu = f(Re_f, Pr, r/Cp\,\Delta t, Re_g) \qquad (9)$$

3. EXPERIMENT

Figure 2 shows the experimental layout. It contains a steam system and a cooling water system. The steam system comprises a boiler, a top stabilising pressure tank, a plate condenser, and a bottom stabilising pressure tank. The cooling water system comprises a pump and a cooling water tank. The plates have chevron shape corrugation. The heat transfer area is 0.028 m^2, the chevron angle is 90 degree., and the triangular corrugation is 4 mm deep. The material of the steam channel is copper. A copper plate and a steel plate form the cooling water channel. When the device was designed, a 0.5 mm corrugation was added between the edge of the plate and the first corrugation. The purpose was to increase turbulence and to drain the condensate so as to decrease the heat resistance and to enhance the heat transfer.

During the experiment the measured parameters were: top stabilising pressure P_1, temperature T_1, bottom stabilising pressure P_2, temperature T_2, condensate volume flux V_f, temperature T_f, mass flux of inlet cooling water Gw_1, mass flux of outlet cooling water Gw_2, inlet temperature of cooling water Tw_{11}, Tw_{21}, outlet temperature of cooling water Tw_{12}, Tw_{22}, and plate temperature sensor electrical potential values U_1.......U_{24}.



1. boiler
2. top stabilising pressure tank
3. plate condenser
4. bottom stabilising pressure tank
5. cooling water tank
6. pressure sensor
7. temperature sensor
8. thermostat
9. liquid level indicator

Figure 2: Experimental layout.

The range of steam pressure was 0.08 to 0.28 bar gauge, outlet Re Number of condensate was 60 to 150, and inlet Re Number of steam was 1700 to 2800.

4. RESULTS AND DISCUSSION

Four main factors are discussed.

4.1 Effect of Pressure

Figure 3 shows two cases: (1) a decrease in condensate temperature difference enhances the heat transfer, and (2) an increase in steam pressure reduces the inlet Re Number of steam.

4.2 Effect of Film Thickness

Figure 4 shows that as the ratio $r/(Cp\Delta t)$ increases, the heat transfer improves. Δt is the condensate temperature difference and r is the latent heat of condensation. $r/(Cp\Delta t)$ is inversely proportional to the thickness of the condensate film. When the pressure changes slightly, both Cp and r change negligibly. Therefore, $r/(Cp\Delta t)$ is dominated by Δt. In other words, the thinner the film, the better is the heat transfer. This accords with the theoretical studies.

4.3 Effect of Liquid Film Reynolds Number

When the steam pressure is constant, condensation heat transfer improves with increasing Ref and Re,.

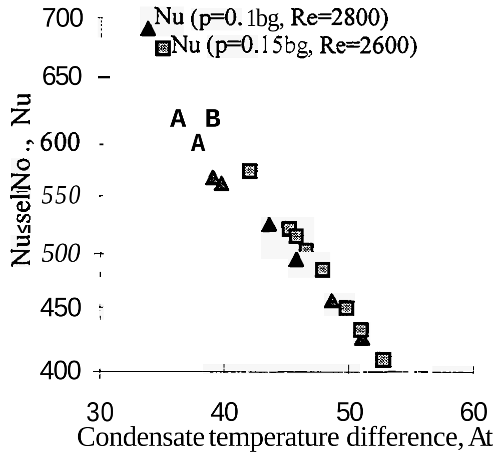


Figure 3: Nusselt No. versus condensate temperature difference.

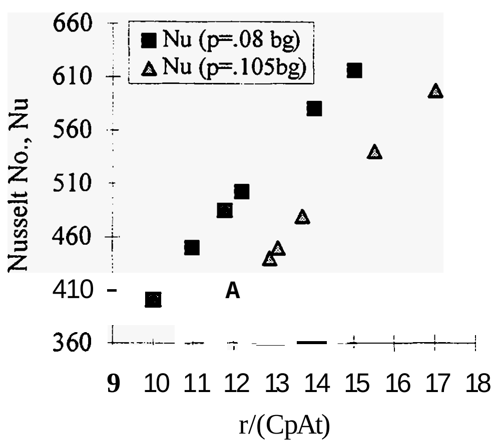


Figure 4: Nusselt No. versus r/(CpΔt).

4.4 Effect of Plate Shape

The heat transfer coefficient at the inlet end of the plate is larger than that at the outlet end. At the inlet end, the condensate film drops rapidly with gravity, and condensate drains immediately fiom the chevron corrugation where liquid deposits, so the film is very thin and heat transfer is high whereas at the outlet, there is much liquid deposited near the comers, so the film is relatively thick and heat transfer is not satisfactory.

Considering the flow of condensate film, steam pressure, condensation temperature difference, steam velocity, etc., a formula that describes the relationship of condensation heat transfer between plates is derived as shown in equation (10).

$$Nu = 0.05128\, Pr^{0.30}\, Re_f^{0.28}\, (r/Cp\Delta t)^{1.01}\, Re_g^{0.62} \qquad (10)$$

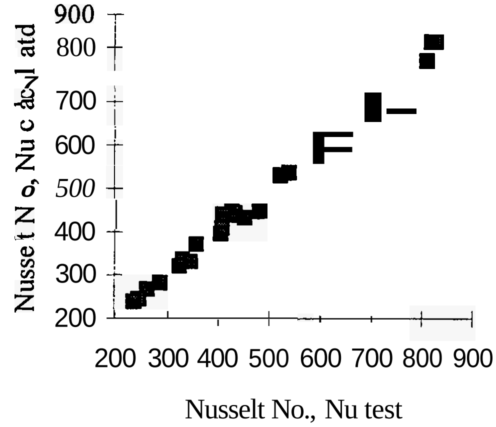


Figure 5: Calculated Nu versus test Nu.

Figure 5 shows that equation 10 fits the test data well. The maximum deviation is less than 10%.

5. CONCLUSIONS

In this paper condensation in a vertical tube is analysed theoretically. Through the theoretical models and dimensional analysis, an equation of the form as shown in equation **(9)** is expected.

$$Nu = f(Ref, Pr, r/Cp\Delta t, Re_g)$$

For steam pressure less than 0.3 bar gauge and steam temperature less than 110°C, the main influential factors have been investigated, and an experimental correlation equation is derived as shown in equation (10).

$$Nu = 0.05128\, Pr^{0.30}\, Re_f^{0.28}\, (r/Cp\Delta t)^{1.01}\, Re_g^{0.62}$$

Both theoretical and experimental studies show that the thickness of the condensate film is the most important factor. Therefore, how to reduce the condensate film to enhance heat transfer is to be studied further.

Heat transfer performance of plate condenser is also satisfactory under low pressure and low Re Number conditions. Not only is it available to recover waste heat, but satisfactory heat transfer can also be achieved.

6. ACKNOWLEDGEMENT

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7. NOMENCLATURE

Cp	heat capacity, J/kgK
D	equivalent diameter, m
g	gravitation acceleration
Nu	Nusselt Number
P	steam pressure, bar gauge
Pr	Prandtl Number
q	heat flux, W/m^2
r	latent heat of condensation, J/kg
Re	Reynolds Number
T	temperature, °C
U	electrical potential of temperature, mV
u	liquid film velocity, m/s
y	liquid film thickness, m
α	heat transfer coefficient, W/m^2°C
Δt	condensate temperature difference, °C
λ	thermal conductivity, W/mK
ρ	density, kg/m^3
τ	shear stress, N/m^2

Subscripts

f	liquid film
g	steam
i	liquid film and vapour interface
w	wall of inlet tube

Superscript

+	dimensionless parameter

8. REFERENCES

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