REVIEW ON STEAM CONDENSATION HEAT TRANSFER COEFFICIENT IN VERTICAL MINI DIAMETER TUBE

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Abstract - *Design of tubular heat exchanger is commonly part of technology system. The most efficient tubular heat exchanger use latent heat of fluid as is phase change from gas to liquid. Thermal design of tubular heat exchanger based on condensation requires knowledge of phase change process in the tubes. This paper focused on review of condensation of steam inside the vertical tube. Condensation experiment are conducted on water steam as a heating medium flow inside tube and water as a base fluid from in shell side by varying saturation temperature and mass flow of water steam and cooling water. Different theoretical or experimental correlation are present for calculating condensation heat transfer coefficient. All this correlation attributes the different condensation heat transfer coefficient. Nusselt gives less condensation heat transfer coefficient than other correlation. Nusselt not studied wave of water steam on condensate surface. Flow of water steam in tubes side form waves on condensate surface and wave effect increase condensation heat transfer coefficient.*

Key Words: Condensation, vertical tube, water steam, heat exchanger, condensation heat transfer coefficient, wave effect.

1. INTRODUCTION:

Condensers are used in a range of chemical, petroleum, processing and power facilities for distillation, for refrigeration and for power generation. Most condensers used in the chemical process industries are water-cooled shell-and-tube exchangers and air-cooled tube or platen exchangers. Shell-and-tube condensers, which are used for condensing process vapors, are classified according to orientation (horizontal and vertical) and according to the placement of the condensing vapor (shell-side and tube-side). This project deals with vertical shell-and-tube condensers with tube-side condensation. Calculations of the overall heat transfer coefficient necessary for the design of the condenser heat transfer area are well described in the literature, but for limited operating conditions only. The Nusselt's condensation model, which is often recommended for calculating the condensing side heat transfer coefficient, is derived for conditions which need not be satisfied in real operation. The Grober's method is commonly used for calculating the shell-side heat transfer coefficient. Many industrial systems use vertical tube condensers and industrial practice has indicated that, often, much higher condensation coefficients of heat transfer are obtained when vapors are condensed inside tubes rather than outside. Condensation heat transfer plays an important role in many engineering applications, electric power generation, refrigeration and air-conditioning, process industries. Many different physical phenomena are involved in the condensation process, their related importance depending on the circumstances and application.

1.1 Types of condensation:

Condensation occurs when the vapour temperature is reduced below its saturation temperature T_{sat}. This is usually done by bringing the vapor into contact with a solid surface whose temperature T_s is below the saturation temperature T_{sat} of the vapor. Two distinct forms of condensation are observed: film condensation and drop condensation.

1.1.1 In film condensation: In film condensation, the condensate wets the surface and forms a liquid film on the surface that slides down under the influence of gravity. The liquid film thickness increases in the flow direction as more vapor condenses on the film. This is how condensation normally occurs.

1.1.2 In drop condensation: In drop condensation, the condensed vapor forms droplets on the surface instead of a film, and the surface is covered by countless droplets of varying diameters.

In film condensation, the surface is blanketed by a liquid film of increasing thickness, and this "liquid wall" between solid surface and the vapor serves as a resistance to heat transfer. The heat of vaporization h_{fg} released as the vapor condenses must pass through this resistance before it can reach the solid surface and be transferred to the medium on the other side. In drop condensation, however, the droplets slide down when they reach a certain size, clearing the surface and uncover it to vapor.

There is no liquid film in this case to resist heat transfer. As a result, heat transfer rates that are more than 10 times larger than those associated with film condensation can be achieved with drop condensation. Therefore, drop-wise condensation is the preferred mode of condensation in heat transfer applications, and people have long tried to achieve sustained drop-wise condensation by using various vapor additives and surface coatings. These attempts have not been very successful, however, since

the drop-wise condensation achieved did not last long and converted to film condensation after some time.

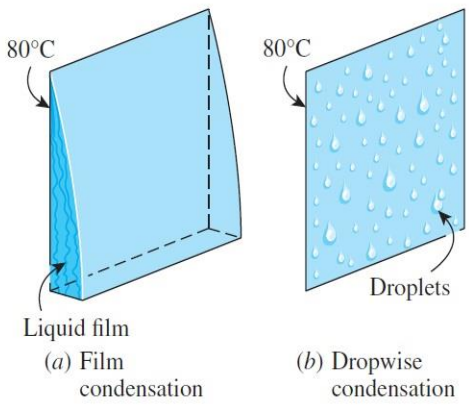

Fig 1.1: Types of condensation (a) film condensation and (b) drop condensation

Therefore, it is common practice to be conservative and assume film condensation in the design of heat transfer equipment. For better heat transfer, it is desirable to use short surfaces because of the lower thermal resistance.

1.2 Flow Regimes:

The flow of liquid film exhibits different regimes, depending on the value of the Reynolds number. It is observed that the outer surface of the liquid film remains smooth and wave-free for about Re ≤ 30 and thus the flow is clearly laminar.

As the Reynolds number increases waves appear on the free surface of the condensate flow. The condensate flow is called wavy-laminar in the range of 30 ≤ Re ≤ 1800 and turbulent for Re >1800

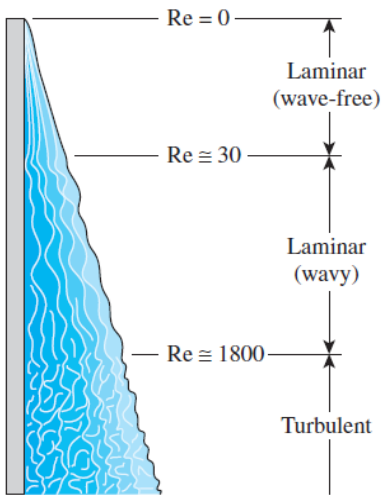

Fig. 1.2: Flow regimes during film condensation in a vertical tube.

2. METHOD FOR PREDICTING CHTC:

Many Scientist and Engineers have done the research on condensation of steam flow through vertical tube. The effect of steam flow rate in vertical tube gives wide range of condensation heat transfer coefficient. General experimental equation for different condition by various investigators has been discussed in following sub-section.

2.1 Thermal Resistance Method:

The overall heat transfer coefficient of cylindrical wall includes inverted sum of Thermal resistance of solid tube wall R_T and two unknown surface Thermal resistances on internal R_V and external surface R_w of tube. The external Thermal resistance R_W of tube on cooling water site can be estimated by average heat transfer coefficient hw .The Nusselt number Nu_W for cooling water flow along the tube is calculated according to Grober. Subsequently the condensation heat transfer coefficient hv on inner surface of tube can be obtained.

$$\frac{1}{u} = \frac{1}{\pi . D_i . h_v} + \frac{\ln\left(\frac{D_e}{D_i}\right)}{2 . \pi . k_T} + \frac{1}{\pi . D_e . h_w}$$

$$\frac{1}{u} = R_v + R_T + R_w$$

$$Nu_w = 1.86 \left(Re_w . Pr_w . \frac{D_e}{l}\right)^{0.33}$$

$$h_w = \frac{Nu_w . h_w}{D_e}$$

2.2 Nusselt Theory:

Nusselt [1] in 1916 is published the first article about laminar film condensation, where Nusselt analytically expressed condensation heat transfer coefficient dependent on mass of steam condensate. Condensation heat transfer coefficient is equal to ratio of thermal conductivity and condensate film thickness.

The analytical relation for the heat transfer coefficient in film condensation on a vertical plate described under the following simplifying assumptions:

1. Both the plate and the vapor are maintained at constant temperatures of T_s and T_{sat}, respectively, and the temperature across the liquid film varies linearly.

2. Heat transfer across the liquid film is by pure conduction (no convection currents in the liquid film).

3. The velocity of the vapor is low (or zero) so that it exerts no drag on the condensate (no viscous shear on the liquid–vapor interface).

4. The flow of the steam condensate is laminar and the properties of the liquid are constant.

5. The acceleration of the condensate layer is negligible.

The average heat transfer coefficient for laminar film condensation over a vertical flat plate of height H is determined to be

$$h_v = 0.9428\left[\frac{g \times \rho_c \times h_{fg} \times k_c^3}{\vartheta_c \times (T_{sat} - T_s) \times L}\right]^{0.25}$$

Reynold number (Re) :

$$Re = \frac{4g}{3\vartheta_c^2}\left(\frac{4k_c}{3h_v}\right)^3$$

2.3 Bromley and Rohsenow:

The condensate in an actual condensation process is cooled further to some average temperature between T_{sat} and T_s, releasing more heat in the process. Therefore, the actual heat transfer will be larger. Rohsenow [3] showed in 1956 that the cooling of the liquid below the saturation temperature can be accounted for by replacing h_{fg} by the modified latent heat of vaporization h^*_{fg}, defined as

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$

where c_{pl} is the specific heat of the liquid at the average film temperature.

The effect of sub-cooling condensate on wall surface is published later by Bromley and non-linear temperature distribution in film condensate is studied by Rohsenow. Bromley and Rohsenow investigated the effect of condensate subcooling on the heat transfer coefficient. They noted that the effect of nonlinear temperature profiles is negligible for small Jakob numbers and thin films, i.e., low-reduced pressures, but that subcooling is an important consideration at higher reduced pressures. Their analysis showed that subcooling the condensate would result in larger heat transfer coefficients compared to the value predicted by Nusselt film theory.

They proposed a modified phase-change enthalpy including the Jakob number that accounted for the degree of subcooling. Both of these observations suggest that increasing reduced pressure leads to conditions where the trends predicted by the Nusselt film model are not valid, and the effect of a parameter such as ΔT could be opposite of the typically expected dependence. Nusselt's modified equation is

$$h_v = 0.9428\left[\frac{g \times \rho_c \times h^*_{fg} \times k_c^3}{\vartheta_c \times (T_{sat} - T_s) \times L}\right]^{0.25}$$

JAKOB NUMBER (Ja) : It is ratio of sensible to latent energy absorbed during liquid-vapor phase change. Mathematically representation of jakob number is

$$Ja = \frac{c_p(T_s - T_{sat})}{h_{fg}}$$

2.4 Kapitsa and Mc-Adams:

At Reynolds numbers greater than about 30, it is observed that waves form at the liquid–vapor interface although the flow in liquid film remains laminar. The flow in this case is said to be wavy laminar. The waves at the liquid–vapor interface tend to increase heat transfer. But the waves also complicate the analysis and make it very difficult to obtain analytical solutions. Therefore, we have to rely on experimental studies. The increase in heat transfer due to the wave effect is, on average, about 20%, but it can exceed 50%. The exact amount of enhancement depends on the Reynolds number.

The wave's effect is studied by kapitsa [4] (1948) and later McAdams [5] (1954) went even further and suggested accounting for the increase in heat transfer in the wavy region. It is suggested using Nusselt equation for the wavy region also, with the understanding that this is a conservative approach that provides a safety margin in thermal design.

$$h_{v,wave} = 1.2\, h_v$$

The wave's effect increases condensation heat transfer coefficient about 20% as published Whitham equation is

$$h_v = 1.137\left[\frac{g \times \rho_c \times h_{fg} \times k_c^3}{\vartheta_c \times (T_{sat} - T_s) \times L}\right]^{0.25}$$

2.5 Kutateladze:

Kutateladze (1963) recommended the following relation for the average heat transfer coefficient in wavy laminar condensate flow for $\rho_v \ll \rho_l$ and $30 < Re < 1800$.

$$h_v = \frac{Re\, k_c}{1.08\, Re^{1.22} - 5.2}\left(\frac{g}{\vartheta_c}\right)^{0.33}$$

A simpler alternative to the relation above proposed by Kutateladze [6] (1963) is

$$h_{c,wavy} = 0.8\, Re^{0.11}\, h_{c,smooth}$$

which relates the condensation heat transfer coefficient in wavy laminar flow to that in wave-free laminar flow.

2.6 Hobler:

The next theoretically determined equation which includes the wave's effect is published by Hobler [7]. This

equation is chosen for comparison because the equation is often applied in engineering tasks. Hobler equation is valid for many kind of fluids with pressure 0.07 < pv [MPa]< 17 and specific heat flux 1.0 < qv [kW/m2]< 1000.

$$h_v = 0.00252 \times \left(\frac{\rho_v \times l_{23}}{\rho_c - \rho_v} \cdot \frac{\rho_c}{\sigma_c}\right)^{0.33} \times \frac{k_c^{0.8} q_v^{0.7}}{\mu_c^{0.5} \times c_c^{0.167} \times T_v^{0.37}} \times p_v^{\frac{10}{T_v - 273.15}}$$

2.7 Jack H. Goodykoontz and Robert G. Dorsch:

Local heat-transfer data were obtained for steam condensing in vertical downward flow inside a tube. A 5/8-inch-inside-diameter and 8-foot-long stainless-steel water-cooled tube was used as the test condenser. The coolant flowed counter currently in the surrounding annulus. Complete condensation occurred in the test section. The downstream vapor-liquid interface was maintained inside the tube in all runs by throttling the condensate at the exit. Axial variations of the local condensing heat transfer coefficient are presented. High condensing heat transfer coefficient occurred at the vapor inlet decreasing with distance down the tube at the downstream end of the condensing section. The local condensing heat transfer coefficients were strongly dependent on the local vapor flow rate. The average condensing heat transfer coefficients for the entire condensing region showed an approximate linear relation with the total mass velocity of the test fluid. Axial temperature distributions for the condenser tube wall, test fluid, and coolant are also presented. The measured axial temperature profiles of the vapor agreed closely with the local saturation temperature profiles obtained from measured static pressures when the inlet vapor was near saturation conditions. However, temperature measurements made with the inlet vapor in a superheated state showed that the core of the vapor could remain superheated the entire length of the two phase region, although condensation was occurring at the wall [8].

2.8 E. M. Sparrow and W. J. Minkowyez:

The developing work of Nusselt has been modified recently to include both the effect of interfacial shear stress on condensate flow and the characteristic of vapour velocity reducing along the length of the tube. All these investigations were covered with condensation of pure saturated vapours. However, in practical operations of tube condensers, small amount of non-condensable gas may exist in working vapours due to the leakage of the system or dissolution of working vapours. By the investigations regarding condensation in unconfined spaces, such as on flat plates or outside horizontal tubes, it has been well established that the existence of non-condensable gas in vapours can greatly reduce condensation heat transfer and deteriorate the performance of condensers. Thus, predicting the effects of non-condensable gas on annular filmwise condensation of vapours in a vertical tube seems to be of important technical and theoretical interest. It was found that the effects of non-condensable gas on condensation is more significant in ducts than in an unconfined space [9].

2.9 Chamra, Webb and Yang:

Chamra and Webb [14] described the results of condensation experiments on R-22 in microfinned 14.66mm diameter tubes, and Yang and Webb [13] reported condensation heat transfer coefficients of R-12 in 2.64mm plain and microfinned tubes. Both Chamra and Webb and Yang and Webb noted that the heat transfer coefficient increases with heat flux. Their experiments covered mass fluxes from 150–327 kgm^{-2} s^{-1} and 400–1000 kg m^{-2} s^{-1}, respectively. Both papers reported that the heat transfer coefficient is proportional to heat flux to the power of 0.22 and 0.20, respectively.

2.10 Dobson and Chato:

Dobson and Chato conducted an extensive set of experiments using different fluids, saturation temperatures (reduced pressures) and tube diameters,. Three fluids and two saturation temperatures, i.e., 35 and 45 0C, were investigated, corresponding to the following reduced pressures: R134a (Pr = 0.21- 0.32), R22 (Pr = 0.27- 0.39), and R32-R125 mixture (Pr = 0.43- 0.55). The experiments were conducted for small temperature differences (1, 2, 3, and 4 0C). In the classified flow regime, they noted an inverse relationship between heat transfer coefficient and temperature difference at low mass fluxes, approximately 75 kg m-2 s-1 . For larger mass fluxes, >300 kg m-2 s-1, they noted the direct relationship between heat transfer coefficient with temperature difference, i.e., increasing temperature difference resulted in greater heat transfer coefficients. They attributed this trend to increased heat transfer occurring due to forced convection in the liquid pool at the base of the tube.

2.11 Agarwal and Hrnjak:

Agarwal and Hrnjak observed an increase in the heat transfer coefficient with an increase in heat flux in the condensation region for heat fluxes between 5 and 10kW m-2 and a decrease in heat transfer coefficient for heat fluxes between 15 and 20Kw/m^2. They showed that the heat transfer coefficient in the desuperheating stage increased with heat flux.

3. INFLUENCING PARAMETER FOR CHTC:

Nusselt's film condensation theory presumes an even increase in thickness of the film due to further condensation. However experiments, show that even in that is clearly laminar, wave can develop at the film surface. Obviously this means that the disturbances in velocity that are always present in the steam are not damped under certain condition, and so wave form. They

lead to improvement in the heat transfer of 10 to 25 % compared to the prediction from Nusselt's theory.

If the effect of wave is ignore and calculation are done according to Nusselt's film condensation theory, then the heat transfer coefficient is too small, meaning that condenser would be too big. Unfortunately at the moment there is no reliable theory for calculating of wave formation on the heat transfer. In practical application, the heat transfer coefficient h_v according to Nusselt's film condensation theory is multiply by the correction factor i.e. wave factor or wave effect.

Up to date researchers tries to increases the condensation heat transfer coefficient by increasing velocity of steam i.e. increasing Reynolds number and by increasing the pressure of the steam . both this effect tries to decreases condensate film thickness along the flow direction by reducing condensate film thickness along the flow direction resistance to heat transfer is decreases and improve the heat transfer.

Past literature shows condensation of steam inside the tube or tube bundle having tube inner diameter is about 15 to 40 mm and length is about 1000 to 1500 mm. up to now all literature for L/di ratio is less than 200. No such literature is present for L/di is more than 200. This ratio is increases by decreasing the inner diameter of tube as diameter of tube decreases the condensation heat transfer coefficient is increases.

NOMENCLATURE:

Latin symbols

CHTC	Condensation heat transfer coefficient
c	specific thermal capacity [J kg^{-1} K^{-1}]
d	characteristic length in Nusselt number [m]
D	diameter of tube [m]
g	gravity acceleration [m s^{-2}]
h	specific enthalpy [J kg^{-1}]
U	overall heat transfer coefficient [Wm_1 K_1]
h_{fg}	latent heat of phase change [J kg^{-1}]
L	total length of tube [m]
m	mass flow rate [kg s^{-1}]
n	total count of tubes [pcs]
p	static pressure [Pa]
q	specific heat flux [W m^{-2}]
Q	total heat flux [W]
R	Thermal resistance [m KW^{-1}]
t	temperature in Celsius scale [^{0}C]
T	temperature in Kelvin scale [K]
ΔT	logarithmic mean temperature difference [K]
V	volume flow rate [m^3 s^{-1}]
x	variable on x-axis [–]
y	variable on y-axis [–]

Greek symbols

h	heat transfer coefficient [W m^{-2} K^{-1}]
k	thermal conductivity [W m^{-1} K^{-1}]
μ	dynamic viscosity [Pa s]
ν	kinematic viscosity [m^2 s^{-1}]
π	mathematical constant [–]
ρ	bulk density [kg m^{-3}]
σ	surface tension [N m^{-1}]

Subscripts

c	condensate
e	external
i	internal
min	minimal value
T	wall of tube
v	vapour
w	water
in	inlet
out	outlet

Dimensionless numbers

Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
Ja	Jakob number

4. CONCLUSIONS

This review paper has consider the different experimental correlation compare with the thermal resistance method and Nusselt's theory which shows different correlation gives different condensation heat transfer coefficient and heat transfer inside the vertical tube or tube bundle. Such as Nusselt theory, Bromley and Rohsenow theory, Kapitsa and Mc-Adams theory, Kutateladze theory, Hobler theory, Jack H. Goodykoontz and Robert G. Dorsch theory, etc. parameter studied in this literature is condensation heat transfer coefficient mass flow rate of steam and pressure of steam at inlet of condenser.

- Condensation heat transfer coefficient from Nusselt's is less than other experimental correlation because Nusselt assume the velocity of steam flow through the tube is very small of negligible or for stationary stream , in other literature flow velocity of steam is consider. As the vapour velocity increased, the shear stress on the condensate film became stronger and made the film thinner. Moreover, the disturbance effect also became stronger. Finally the thermal resistance decreased and the heat transfer coefficient increased. It could be seen that, the heat transfer coefficient increased with the increase in vapour velocity.
- The Nusselt equation does not take to account wave effect on condensate surface. These waves on condensate surface are caused by flow of water steam in

tube. The Nusselt film theory, which is applicable only to laminar smooth falling-film condensation, incorrectly predicts the trend for the conditions investigated.

- Condensation heat transfer coefficient increase by increasing the vapour pressure. The surface tension of the condensate film is mainly determined by the temperature, and decreases with the increase in the temperature. the temperature of the condensate film increases with increasing the vapour pressure. Thus, the surface tension of the condensate film decreased with the increase in vapour pressure. Under the high vapour pressure, the condensate film became thinner under the effect of gravity, and then the heat transfer was promoted.

Therefore, in future study condensation heat transfer coefficient is enhancing by using vertical tube of small diameter and studying the effect of mass flow rate, pressure and temperature of steam and mass flow rate of water on shell side. Hence, condensation of steam inside the tube needs further improvement.

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