AN EXPERIMENTAL INVESTIGATION ON EFFECT OF COATINGS ON THE SURFACES ON DROP WISE CONDENSATION

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ABSTRACT

The current report describes the experimentation that is conducted to examine condensation procedures and their types, calculations of HTC, and heat flux. Here, two different kinds of condensation are inspected.

- Drop wise condensation.
- Film wise condensation.

The organic STA films were coating on surface to promote the dropwise condensation. The experimental data show that the STA quoted surface show a better performance of hydrophobicity and a larger apparent contact angle. An increase of the droplet departure diameter and a lower departure frequency, for a larger contact angle hysteresis on the microblog surface during the drop wise condensation process also results in a decrease in the heat transfer performance of microgrooves nanostructure surface with STA.

KEYWORDS: Heat transfer coefficient, condensation, thermal analysis, surface coatings, condenser pressure, flow rate etc.,

Chapter-1

INTRODUCTION

INTRODUCTION

Introduction as it was stated earlier, the key target of this trial is to see the impact of fluctuating water flow rate on HTC in condensation. It is similarly required to inspect condensation and its dissimilar forms together with film-wise and dropwise condensation meanwhile trial is ended with both.

Conferring to the 2nd law of thermodynamics, energy is shifted from hot element to cold one. Similarly, once hot vapor is in interaction with colder matter, its energy is shifted. Throughout energy transmission procedure, molecules of vapor get closer to each other, which contributes to creation of molecular bands, henceforth, condensation alters its state from gas to liquid.

The condensation might be elucidated via the entropy. Thus, getting rid of energy contributes to dramatical reduction in the entropy level of the scheme, due to more stable state of fluid in contrast to the formed gas.

As it is noted above, there are 2 distinct types of condensation:

- Dropwise (low wettability surface)
- Film-wise (high wettability surface)

The difference is relied on the surface tension. Once there are more cohesive forces amid molecules of the same element than adhesive ones, gasses condense and accumulate in the form of drops on the surface – dropwise condensation occurs.

In contrast, in opposite case scenario, the liquid formed on the surface is looking like a film in film-wise condensation. Here, film causes a substantial surge in confrontation to heat transmission, while in dropwise is a bit lower.

In various fields of engineering activity, especially in petroleum industry, the film-wise condensation is much preferable for heat transfer procedures that are realized via heaters and condensers. Even though HTC is higher in dropwise condensation, building of apparatus for film-wise condensation is much cheaper. Otherwise, for construction of equipment based on drop-wise condensation, expensive metals, like gold and silver are required. Subsequently, in terms of efficiency and economy, it is totally unfeasible to use dropwise condensation method.

1.1 Mechanism of heat transfer in drop wise Condensation:

Heat transfer plays an important role in many engineering applications, notably electric power generation, process industries, refrigeration, and air-conditioning. Many different physical phenomena are involved in the condensation process, their relative importance depending on the circumstances and application. For examples, see Air Conditioning, Condensers, Desalination, and Refrigeration.)When a liquid and its vapor are in contact, molecules pass from liquid to vapor and from vapor to liquid. Condensation occurs when the number of molecules entering the liquid phase exceeds that of leaving molecules. Under these circumstances, the temperature of the vapor in the immediate vicinity (a few mean free paths) of the vapor-liquid interface is higher than that of the liquid. The *interface temperature drop* increases with increasing condensation rate and with decreasing pressure but in most circumstances (an exception being the case of liquid metals), this is very small and equilibrium conditions at the interface can be assumed. A brief summary of interface matter transfer during condensation is given by Niknejad and Rose (1981).

The most common and best understood case of condensation heat transfer is that of film *condensation* of a pure quiescent vapor on a solid surface. The problem of calculating the heat transfer rate for a plane vertical surface and for a horizontal cylinder with uniform surface temperatures, and where the condensate flow is laminar and governed only by gravity and viscous forces, has been solved by Nusselt (1916). Nusselt's main assumptions are that heat transfer across the condensate film is by pure conduction, the effect of vapor drag in supporting the falling condensate film is negligible, and the properties of the condensate may be taken to be uniform across the film, i.e., are essentially independent of temperature.

1.2 The well-known Nusselt equations are:

For the vertical plane surface, the mean value of Nusselt number is

(1)

$$Nu = 0.943\left\{\frac{\rho\Delta\rho g h_{lg} L^3}{\eta\lambda\Delta T}\right\}^{1/4},$$

and for the horizontal tube

(2)

$$Nu = 0.728\left\{\frac{\rho\Delta\rho g h_{lg} d^3}{\eta\lambda\Delta T}\right\}^{1/4},$$

where ρ is the liquid density, g is acceleration due to gravity, h_{lg} is the latent heat of evaporation, η is the liquid viscosity, λ is the liquid thermal conductivity, Δρ is the density difference

between vapor and condensate, ΔT is the vapor-to-surface temperature difference; L is the plate height; and d, the tube diameter. For the case of the tube, the additional assumption that film thickness is small compared with the tube radius is needed. Since the theory predicts that the radial film thickness tends to infinity at the bottom of the tube, this assumption is evidently invalid for the lower part of the tube. The fact that the heat transfer rate is inaccurate where the film becomes thick is relatively unimportant because it is small and makes a minor contribution to the total heat transfer rate for the tube. Equations (1) and (2) have been well verified experimentally for condensation of pure (only one molecular constituent) vapor. In order to obtain the constant in Eq. (2), numerical integration is required twice. Nusselt's slightly inaccurate value ($0.8024\ (2/3)^{1/4} = 0.725$) is due to his use of planimetry.

As discussed by Rose (1988), more recent theoretical studies in which the effects of inertia and convection in the condensate film and vapor drag on the condensate surface are included have shown that these effects are unimportant. More recently, it has been shown by Memory and Rose (1991) that the effect of variable wall temperature, which occurs in practice during condensation on a horizontal tube, also has a negligible effect on the *mean* heat transfer rate. Thus, Eqs. (1) and (2) can be used with confidence for condensation of pure "stationary" vapors when the condensate flow is laminar.

In the case of significant vapor flow rate over the condensing surface, the effect of drag on the condensate becomes significant. In some circumstances, the effect of vapor drag overwhelms that of gravity. Since 1960, *vapor shear stress effects* have been studied extensively. Some of the more important contributions are described by Rose (1988).

The relative importance of vapor shear stress and gravity on the motion of the condensate film is measured by the dimensionless parameter $F=\eta h_{lg} gx/\lambda \Delta T u^2_\infty$, where x is the relevant linear dimension (plate height or tube diameter) and u_∞ is the free-stream vapor velocity. For downward vapor flow over a horizontal tube, an approximate analysis gives

(3)

$$Nu_d \tilde{R}e_d^{-1/2} = \frac{0.9 + 0.728F^{1/2}}{(1 + 3.44F^{1/2} + F)^{1/4}}.$$

Nu_d is the mean Nusselt number for the condensate film and $\tilde{R}e_d$ is a Reynolds number using the vapor approach velocity and condensate properties. Equation (3)—which indicates that for F > 10, gravity dominates while for F < 0.1, vapor shear stress is controlling—agrees quite well with experimental data from several investigations using various condensing fluids. (See Tubes, Condensation on Outside in Crossflow.)

When the vapor contains more than one molecular species, the problem is complicated by diffusion of species in the vapor. For example, for a two-constituents vapor where only one constituent condenses (e.g., steam-air), the mixture is rich in the *noncondensing gas* near the interface where vapor molecules are removed. The tendency is for noncondensing gas molecules to diffuse away from the interface so that in the steady state, the rate at which gas molecules arrive at the surface with the condensing vapor is equal to their diffusion rate away from the surface. Even in the absence of forced convection of the vapor-gas mixture, the density difference, which results from the composition difference between that of the bulk vapor and that of the vapor adjacent to the interface, leads to natural convection. The process by which the steady state is reached is therefore one of diffusion in the presence of convection. The fact that the vapor-gas mixture adjacent to the condensate surface is rich in non-condensing gas causes the temperature at the interface to be lower than in the bulk. Assuming equilibrium at the interface, the temperature is equal to the saturation temperature corresponding to the partial pressure of the vapor, which may be significantly lower than in the more remote vapor. Composition (or partial pressure) and temperature boundary layer are set up in the vapor adjacent to the interface. This gives an effective heat transfer resistance since the temperature drop across the condensate film, and hence the heat transfer rate, is significantly reduced. Detailed boundary layer solutions of this problem for free and forced convection, notably by Koh, Sparrow, Fuji et al., have been given. Earlier works are discussed, and approximate equations are given by Rose (1969) for the free convection problem, and Rose (1980) for the forced convection case. When two or more constituents of the vapor condense together, the situation is similar to that described above since the more volatile constituent is more concentrated at the condensate surface. Extensive treatments of these problems for the case of the plane vertical condensing surface and laminar flow of vapor and condensate have been given by Fujii (1991). (See Condensation of Multicomponent Vapors.)

Approximate methods, based on the *analogy* between heat and diffusive mass transfer in the vapor, are widely used for multi constituent problems. The essence of the method is that the
differential equations expressing conservation of energy and molecular species can be arranged in identical form by appropriate nondimensionalization. Known results (theoretical or experimental) for
heat transfer problems are used to infer results for the corresponding mass transfer diffusion problems. The method has wide utility but is approximate since the boundary condition on the normal
vapor velocity at the surface is not the same for the heat and mass transfer problems. For the heat transfer problem, the normal velocity at the (solid) heat-exchanger wall is zero. In the case of condensation, where molecules pass through the vapor-liquid interface, the normal velocity is not zero. The validity of the analogy depends on the smallness of the "suction parameter" $(-v_o/u)Re^{1/2}$, where v_o is the normal outward (i.e., negative for condensation) vapor velocity at the condensate surface, u is the free-stream velocity parallel to the surface and Re is the free-stream vapor Reynolds number. The results are strictly correct only in the limiting case of zero condensation rate. A widely used approximate *stagnant film model* extends the range of validity of the analogy. The heat-mass transfer analogy and the stagnant film model are discussed by Lee and Rose (1983), Butterworth (1983) and Webb (1990).

The foregoing refers exclusively to laminar flow conditions. For tall condensing surfaces, or under conditions of high vapor shear stress, transition to turbulent flow in the condensate film may occur. This brings to the problem unresolved difficulties associated with the general problem of Turbulence. For moderate or low vapor velocities, the "effective height" of the surface for a horizontal tube (≈d) is small and laminar flow of the condensate is expected. Moreover, since turbulent mixing enhances heat transfer across the condensate film, the Nusselt solution [Eq. (2)] is conservative and is widely used in design calculations. Various models for turbulent film condensation from an essentially stationary vapor exist in the literature and predict somewhat different results. For gravity-dominated flow, transition to turbulence has been found to occur at film Reynolds numbers, $4\Gamma/\eta$, rather lower than 2 000, where Γ is the condensate flow rate per width of surface. In the presence of high-vapor shear stress, the problem is more complicated. Turbulent film condensation on a vertical surface under free-convection conditions can, in principle, be analyzed by an approach similar to that used for single-phase pipe flow. However, this problem is relatively unimportant since in practice, condensing surfaces are usually not sufficiently tall for turbulence to occur under purely free-convection conditions. Significant shear stress, due to vapor flow along the condensate surface, promotes the onset of turbulence. The analysis is then more complicated, particularly when the vapor flow is not in the same direction of gravity.

Turbulence is more often encountered for condensation inside a tube. In this case, the problem is generally complicated by the presence of significant vapor shear stress on the condensate film since even when all of the vapor is condensed in the tube, the shear stress for a portion of the tube towards the inlet end is generally significant owing to the high vapor velocity resulting from the small tube cross-section. Condensation inside tubes is beset with all the problems and uncertainties of Two-Phase Flow. The only case which can be analyzed wholly satisfactorily is that of downward vapor flow in a vertical tube with a laminar condensate film on the wall (stratified flow). In this case, the problem is the same as that for external flow, except that account must be taken of the progressive reduction of vapor flow rate, and hence shear stress, due to condensation. Approaches used in other cases are outlined by Butterworth (1983).

In many practical applications, condensation occurs in bundles or banks of horizontal tubes (shell-side condensation). In these cases, there is the additional complication of *inundation* (condensate from higher or upstream tubes falling or impinging on lower or downstream tubes). This leads to thicker condensate films on the inundated tubes. At the same time, the condensate film on inundated tubes is disturbed and the heat transfer coefficient may be enhanced. Nusselt's (1916) approach for a vertical, in-line column of horizontal tubes assumes that condensate drains to lower tubes in the form of a continuous laminar film. This leads to a simple expression for the average heat transfer coefficient for a column of N tubes:

(4)

$$\overline{\alpha}_N = \alpha_1 N^{-1/4},$$

where α_1, for the uppermost tube, is given by Eq. (2). In view of the more probable mode of *drainage* or inundation with condensate film disturbance due to splashing from droplets, columns or unstable broken films of liquid, it is not surprising that Eq. (4) has been found to be overconservative. Many experimental studies of condensation on tube banks have been made. The data are widely scattered owing primarily to the effects of noncondensing gases, turbulence and vapor velocity. Various correlations have been proposed and approximate methods used in practiced are discussed by Butterworth (1983). (See Tube Banks, Condensation Heat Transfer in.)

Numerous techniques for *enhancement of film condensation heat transfer* have been proposed. shell-side condensation, are low (fin height small in relation to tube diameter) integral-fin tubes. In this case, it is found that for horizontal tubes,the enhancement of the heat transfer coefficient can significantly exceed the increase in surface area due to the presence of the fins. The reasons for this are: 1) the vertical or near-vertical fin flanks have small heights so that the heat transfer coefficients are large [see Eqs. (1) and (2)] surface tension effects give rise to an additional mechanism for draining condensate from parts of the surface. The latter arises from the pressure gradient set up in the presence of a condensate surface of varying curvature, e.g., for the two-dimensional case:

(5)

$$\frac{dp}{ds} = \sigma \frac{d(r^{-1})}{ds},$$

where P is pressure, s is distance measured along the surface, σ is surface tension and r is the local radius of curvature of the condensate film. At the same time, surface tension has a detrimental effect on the heat transfer coefficient due to capillary retention of condensate between the fins and the consequent "blanketing" of heat transfer surface on the lower part of the surface. The extent of condensate retention can be calculated from:

(6)

$$\phi = \cos^{-1}\{(4\sigma \cos\beta / \rho g b d_o) - 1\}$$

as formulated by Honda et al. (1983). In Eq. (6), φ is the angle measured from the top of the tube to the position where the inter fin tube space is filled with retained condensate; β is the angle between the fin flank and radial plane; b is the distance between adjacent fins measured at the fin tip; and d_o is the tube diameter over the fins.

In an early theoretical solution of the problem of condensation on low-finned tubes by Beatty and Katz (1948), the vertical fin-flanks and horizontal inter fin tube spaces were treated based on the Nusselt theory and effects of surface tension were ignored. This simple approach proved quite successful in practice for relatively low-surface tension fluids. This is partly because condensate retention in this case is small, and partly because the beneficial and detrimental effects of surface tension tend, to some extent, to nullify each other. More detailed and complicated models have been proposed, notably by Honda and Nozu (1987). These require numerical solution and are less readily applied than the simple analytical result of Beatty and Katz. The various models are discussed by Marto (1988). A recent semi-empirical approach, which includes surface tension effects, has been given by Rose (1994). The result, in the form of an equation for the "enhancement ratio", is in good agreement with experimental data from seven investigations using four condensing fluids and 41 tube/fin geometries.

The foregoing relates to the case when the condensate wets the condensing surface and forms a continuous film. When the surface is not wetted, a quite different mode, namely Dropwise Condensation, may occur. In this case, minute droplets form at nucleation sites on the surface and growth takes place by condensation and coalescence with neighbors until drops reach a size at which they are removed from the surface by gravity or vapor shear stress. Moving drops sweep up stationary drops in their path, making available new area for condensation. The maximum-to-minimum drop size is around 10^6. To date, dropwise condensation has only been obtained with a few high-surface tension fluids (notably water). A nonwetting agent or "promoter" is required to promote dropwise condensation on metal surfaces. Heat transfer coefficients for dropwise condensation are much higher than those for film condensation under the same conditions. For steam at atmospheric pressure, the factor is around 20. Not surprisingly, this has stimulated research work over the past 60 years with the aim of finding an effective promoter. Although good promoters (e.g., dioctadecyl disulfide) are available, which form stable monomolecular layers on copper or copper-containing surfaces, and give dropwise condensation for hundreds or thousands of hours under clean laboratory conditions, effective promoters for use under industrial conditions have yet to be found. Recent surveys of dropwise condensation heat transfer have been given by Tanasawa (1991) and Rose (1994).

In some applications, such as desalination and geothermal power plant, use is made of direct contact condensation.

1.3 NUSSELT THEORY

A German engineer, born on November 25, 1882, at Nürnberg, Nusselt studied machinery at the Technical Universities of Berlin-Charlottenburg and München where he graduated in 1904 and conducted advanced studies in mathematics and physics. He became an assistant to O. Knoblauch at the laboratory for technical physics in München and completed his doctoral thesis on the conductivity of insulating materials in 1907, using the *"Nusselt Sphere"* for his experiments. From 1907 to 1909 he worked as an assistant of Mollier in Dresden, qualifying himself for a professorship with a work on heat and momentum transfer in tubes.

In 1915, Nusselt published his pioneer paper: "The Basic Laws of Heat Transfer" in which he first proposed the dimensionless groups now known as the principal parameters in the similarity theory of heat transfer. Other famous works were concerned with the film condensation of steam on vertical surfaces, the combustion of pulverized coal and the analogy between heat and mass transfer in evaporation. Among the primarily mathematical works of Nusselt, the well-known solutions for laminar heat transfer in the entrance region of tubes, for heat exchange in crossflow and the basic theory of regenerators should be mentioned.

Nusselt was professor at the Technical Universities of Karlsruhe from 1920 to 1925 and at München from 1925 until his retirement in 1952. He was awarded the Gauss-medal and the Grashof commemorative medal. Nusselt died in München on September 1,1957.

Assumptions Made in Nusselt's Analysis of Film Condensation

Nusselt's analysis of film condensation makes the following assumptions:

1.The film of the liquid formed flows under the action of gravity.
2.The condensate flow is laminar, and the fluid properties are constant.
3.The liquid film is in good thermal contact with the cooling surface and therefore, the temperature at the inside of the film is taken equal to the surface temperature T_s. Further, the temperature at the liquid-vapor interface is equal to the saturation temperature T_{sat} at the prevailing pressure.
4.Viscous shear and gravitational forces are assumed to act on the fluid; thus normal viscous force and inertia forces are neglected.

1.4 Mc Adams equation:

Experimental results have shown that the Nusselt's equation is convective , i.e. it yields results which are approximately 20% less than the actual results. According to Mc Adams

$$\mathbf{h = 1.13[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25} \quad W/m^2\,{}^oC}$$

1.5 Bubble Formation

The heat transfer rate in nucleate boiling is greatly influenced by the nature and condition of the heating surface and surface tension at the solid – liquid interface. The surface tension signifies wetting capability of the surface with the liquid and that influences the angle of contact between the bubble and the solid surface. If the surface is contaminated, it's wetting characteristics are effected with eventually the size and shape of the vapour bubbles.

The shape of the bubbles is shown in below fig.

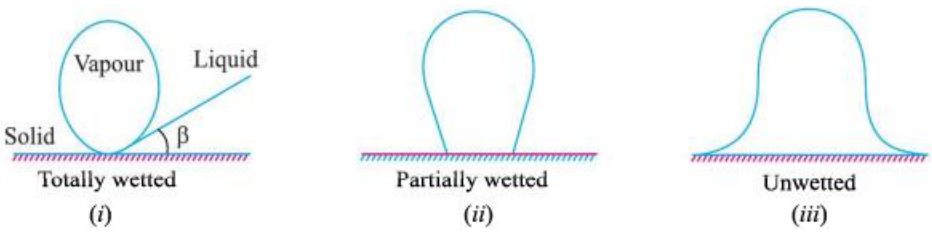


Fig 1: Typical shapes of steam bubbles

Bubble growth and collapse:

From the experiment it has been observed that the bubbles are not always in thermodynamic equilibrium with surrounding liquid. The vapour inside the bubble is not necessarily at the same temperature as the liquid

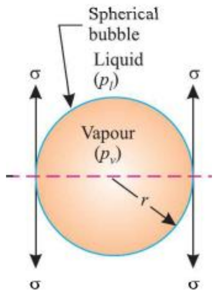


Fig 2: Force balance on a spherical vapour bubble

Critical Diameter of Bubble :

The maximum diameter of the bubbles formed on the heating surface depends on the following parameters:

σ_{lv} = Tension between liquid and vapour
σ_{ls} = Tension between liquid and solid surface
σ_{vs} = Tension between vapour and solid surface
β = Angle formed by the bubble as shown in Fig. 9.5
d_c = Maximum or critical diameter of bubble.
$g(\rho_l-\rho_v)$ = Buoyance force

Thus $d_c = f\left[\beta, \sigma_{lv}\ g\ (\rho_l - \rho_v), \dfrac{\sigma_{lv}}{\sigma_{ls}}\right]$

By the use of the dimensional analysis technique, we get

$$d_c = C.\beta \left(\frac{\sigma_{lv}}{\sigma_{ls}}\right) \sqrt{\frac{\sigma_{lv}}{g\ (\rho_l - \rho_v)}} \quad ...(9.6)$$

where C is constant which is generally calculated by experimental results.

The value of $C = 0.0148$ for water bubbles.

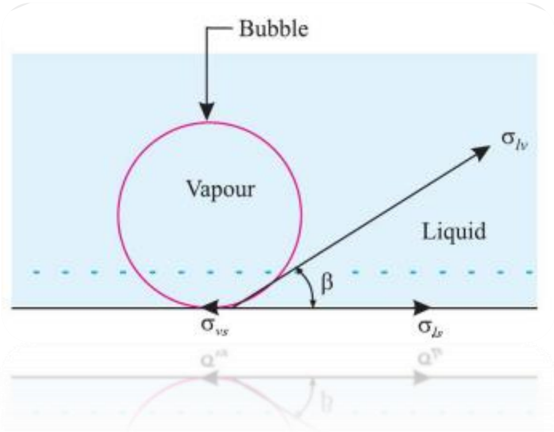


Fig 3:Critical diameter of bubble

Chapter-2
EQUIPMENT DESCRIPTION

Instruments description
2.1.Rota flow meter:

Fig 1: Rota flow meter

A rotameter is a gauge for measuring fluid flow using a graduated glass tube with an enclosed free float. Here we used the rotameter in order to regulate the cooling water flow into both the tubes in the condenser.

The operation of a rotameter is based on the variable area principle. That is, the flow of a liquid raises the float inside a tapered tube, increasing the area through which the liquid can pass. The larger the flow, the higher the float will be raised.

Basically, Rotameter is three types:

- Glass tube rotameter.
- Metal tube rotameter.
- Heavy duty rotameter.

Here we used a glass tube rotameter regulate the flow of the cooling water circulating through the condenser tubes.

2.2. Flow Regulating valve:

Fig 2: Flow control valve

The purpose of flow control valve is to maintain desired portion of the steam and water flow to the condenser and specimen tubes. Steam is supplied at a specific pressure to the upstream side of the control valve through which it passes to a heat exchanger, also operating at a specific pressure. Steam passes through the control valve and into the steam space of the equipment where it comes in contact with the heat transfer surfaces.

2.3. Temperature sensors:

An RTD (resistance temperature detector) is a sensor, whose resistance changes with change in temperature. The resistance increases as the temperature of the sensor increases. An RTD is a passive device.

Fig 3: Thermocouples

Here, we used a total 6 sensors for temperature measurements of inlet and outlet water , condenser chamber and specimens surface temperatures.

RTD does not produce an output on its own. External electronic devices are used to measure the resistance of the sensor by passing a small electric current through the sensor to generate a voltage.

2.4. Steam Generator:

Fig 4: Steam generator

we used a domestic pressure cooker to generate the steam, instead of a steam boiler. And the pressure relief valve is replaced by a diaphragm pressure gauge. The capacity of the cooker is 10 litres.

2.5 Pressure gauge:

Fig 5: Diaphragm pressure gauge

A pressure gauge is a method of measuring steam intensity in a steam powered machine to ensure there are no leaks or pressure changes that would affect the performance of the system.

Here, we used a diaphragm type pressure gauge. A diaphragm is elastic and becomes displaced when pressure is applied. The diaphragm, which is placed between two flanges, is used to determine the difference between the applied pressure and reference pressure.

We used 2 pressure gauges one is to measure steam pressure in the steam boiler and the other one is to check the steam pressure in the condenser chamber.

2.8 Specimens used:

S.No.	Material
1	Stainless steel
2	Stainless steel with wax coating
3	Tin
4	Aluminum-Tin Alloy

Table 1 : Specimens used

Fig 6: Temperature Indicator

2.7 Temperatue indicator:

The temperature Indicators are used for processing signals from temperature sensors or thermocouples. The temperature results will be shown on the led display. The specifications of the indicator is given below.

Specifications:

Brand	Hpg Systems
Size	48 x 96 mm
Model Name/Number	G-1008M
Display Type	Digital
Usage/Application	Industrial
Size/Dimension	48x96mm
Input Type	RTD
Material	Plastic
Color	black

Table 2: Specifications of indicator

2.8 Test rig set-up:

Fig 7: Test rig set-up

An experimental facility was developed to investigate boiling and condensation heat transfer coefficient of different materials multiport brass mini channel. Fig. 7 shows the schematic view of the experimental apparatus in the present study. The experimental system consists of a natural circulation loop. The loop itself comprises an oil free magnetic gear pump, a coriolis mass flow meter, electrical preheater, a multiport test section and a condenser.

Chapter-3
Experimental Observations & Calculations

Experimental observations:

Specimens used:

Copper tube - for film wise condensation

Stainless steel tube - for drop wise condensation.

Instruments specification:

Condenser length - 297 mm

Condenser diameter - 115 mm

Operating pressure - 1.5 bar

Tube specimen specification:

Length - 140 mm

Diameter - 19 mm

Observations: (no coatings on surface**)**

S.no	Mass Flow rate, m (LPH)	T_1 (^{0}C)	T_2 (^{0}C)	T_3 (^{0}C)	T_4 (^{0}C)	T_5 (^{0}C)	T_6 (^{0}C)
1	50 LPH	110.9	71.4	59.5	39.4	43.4	44.2
2		113.1	79.5	60.3	33.1	37.2	39.1
3		113.7	83.1	64.3	31.7	35.7	37.9

Table 3: Observations for no coated surface(Copper & SS Tube)

T_1 - steam (or) condenser temperature, (^{0}C)

T_2 - stainless steel tube surface temperature, (^{0}C)

T_3 - copper tube surface temperature, (^{0}C)

T_4 - cooling water inlet temperature, (^{0}C)

T_5 - stainless steel tube outlet water temperature, (^{0}C)

T_6 - copper tube outlet water temperature, (^{0}C)

Calculations:

1. **First observation calculation**

Area of the condenser , A = πDL

Where,

D = diameter of the tube

L = length of the tube

$A = \pi * 0.19 * 0.140$

$\mathbf{A = 8.36*10^{-3}\ m^2}$

- Mass flow rate , $M_w = 0.00028 * m$ kg/s

$M_w = 0.00028*50$

$\mathbf{M_w = 0.014\ kg/s}$

- At pressure 1.5 bar

$T_{sat} = 111.4\ ^0C$

$T_s = 59.5\ ^0C$

$H_{fg} = 2226.2$ kj/kg

$T_{mean} = ((111.4 + 59.5)) / 2 = 85.45^0C$

At 80^0C,

$\rho = 974\ kg/m^3$

$\mu = 3.47*10^{-3}$ Nm/s

K = 0.6687 w/m-k

For Film wise condensation:

- Heat carried away by water , $Q = M_w*C_{pw} * (T_6-T4)$ Watts

$= 0.014*4187*(44.2-39.4)$

Q = 281.37 Watts

- Theoretical heat transfer co-efficient

$h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*974^2*9.81*(0.6687)^3)/(4*3.47*10^{-3}*(111.9-59.5)*0.140)]^{.25}$

$\mathbf{h = 468.33\ w/m^2\ ^0C}$

- LMTD = $\theta_i - \theta_0 / \ln(\theta_i/\theta_0)$

  $\theta_i = T_1 - T_4 = 110.9 - 39.4 = 71.5\ ^oC$

  $\theta_0 = T_1 - T_6 = 110.9 - 44.2 = 70.7\ ^oC$

  LMTD = 71.5-70.7/ln(71.5/70.7)

  LMTD = 71.09

- Experimental heat transfer co-efficient , h = Q/a*LMTD

  h = 281.37/0.00836*71.09

  h = 473.43 w/m^2 ^{o}C

For drop wise condensation:

- Heat carried away by water , Q = M_w*C_{pw} * (T_5-T4) Watts

  = 0.014*4187*(43.4-39.4)

  Q = 234.47 Watts

- T_{mean} = ((111.4 + 71.4)) / 2 = 91.4^{0}C

  At 90^{0}C,

  ρ = 967.5 kg/m^3

  μ = 3.1178*10^{-3} Nm/s

  K = 0.6745 w/m-k

- **Theoretical heat transfer co-efficient**

  $h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

  $h = 0.943 * [(2226.2*967.5^2*9.81*(0.6745)^3)/(4*3.117*10^{-3}*(111.9-71.4)*0.140)]^{.25}$

  h = 514.67 w/m^2 ^{0}C

- LMTD = $\theta_i - \theta_0 / \ln(\theta_i/\theta_0)$

  $\theta_i = T_1 - T_4 = 110.9 - 39.4 = 71.5\ ^oC$

  $\theta_0 = T_1 - T5 = 110.9 - 43.4 = 67.5\ ^oC$

  LMTD = 71.5-67.5/ln(71.5/67.5)

  LMTD = 69.48

- **Experimental heat transfer co-efficient**

$h = Q/a*LMTD$

$h = 234.47/0.0.00836*69.48$

$h = 403.66$ w/m^2 ^{o}C

2. **Second observation calculation**

At pressure 1.5 bar

$T_{sat} = 111.4\ ^0C$

$T_s\ = 60.3\ ^0C$

$H_{fg}\ = 2226.2$ kj/kg

$T_{mean} = ((113.1 + 60.3))\ /\ 2 = 86.7^0C$

At 80^0C,

$\rho = 974$ kg/m^3

$\mu = 3.47*10^{-3}$ Nm/s

$K = 0.6687$ w/m-k

For Film wise condensation:

- Heat carried away by water , $Q = M_w*C_{pw} * (T_6\text{-}T4)$ Watts

$= 0.014*4187*(39.1-33.1)$

$Q = 351.7$ Watts

- Theoretical heat transfer co-efficient

$h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*974^2*9.81*(0.6687)^3)/(4*3.47*10^{-3}*(113.1-60.3)*0.140)]^{.25}$

$h = 467.45$ w/m^2 ^{0}C

- $LMTD = \theta_i\text{-}\theta_0/\ln(\theta_i/\theta_0)$

$\theta_i = T_1\text{-}T_4 = 113.1\text{-}33.1 = 80\ ^oC$

$\theta_0\ = T_1\text{-}T_6 = 113.1\text{-}39.1 = 74\ ^oC$

$LMTD = 80\text{-}73.6/\ln(80/73.6)$

$LMTD = 76.96$

- Experimental heat transfer co-efficient , h = Q/a*LMTD

h = 351.7/0.00836*76.96

h = 452.54 w/m^2 ^{o}C

For drop wise condensation:

- Heat carried away by water , Q = M_w*C_{pw} * (T_5-T4) Watts

= 0.014*4187*(37.2-33.1)

Q = 240.33 Watts

- T_{mean} = ((113.1 + 79.5)) / 2 = 96.3^{0}C

At 100^{0}C,

ρ = 961 kg/m^3

μ = 2.76*10^{-3} Nm/s

K = 0.6804 w/m-k

- **Theoretical heat transfer co-efficient**

$h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*961^2*9.81*(0.6804)^3)/(4*2.76*10^{-3}*(113.1-79.5)*0.140)]^{.25}$

h = 557.69 w/m^2 ^{0}C

- LMTD = θ_i-θ_0/ln(θ_i/θ_0)

θ_i = T_1-T_4 = 113.1-33.1 = 80 ^{o}C

θ_0 = T_1-T5 = 113.1-37.2 = 75.9 ^{o}C

LMTD = 80-75.9/ln(80/75.9)

LMTD = 77.92

- **Experimental heat transfer co-efficient ,** h = Q/a*LMTD

h = 240.33/0.1073*77.92

h = 368.89 w/m^2 ^{o}C

3. Third observation calculation

At pressure 1.5 bar

T_{sat} = 113.7 ^{0}C

T_s = 64.3 ^{0}C

H_{fg} = 2226.2 kj/kg

T_{mean} = ((113.7 + 64.3)) / 2 = 89^{0}C

At 90^{0}C,

ρ = 967.5 kg/m^3

μ = 3.1178*10^{-3} Nm/s

K = 0.6745 w/m-k

For Film wise condensation:

- Heat carried away by water , Q = M_w*C_{pw} * (T_6-T4) Watts

= 0.014*4187*(38.9-31.7)

Q = 422.04 Watts

- Theoretical heat transfer co-efficient

$h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*967.5^2*9.81*(0.6745)^3)/(4*3.118*10^{-3}*(113.7-64.3)*0.140)]^{.25}$

h = 489.7 w/m^2 ^{0}C

- LMTD = θ_i-θ_0/ln(θ_i/θ_0)

θ_i = T_1-T_4 = 113.7-31.7 = 82 ^{o}C

θ_0 = T_1-T_6 = 113.7-38.9 = 74.8 ^{o}C

LMTD = 82-74.8/ln(82/74.8)

LMTD = 78.34

- Experimental heat transfer co-efficient , h = Q/a*LMTD

h = 422.04/0.00836*78.34

h = 644.37 w/m^2 ^{o}C

For drop wise condensation:

- Heat carried away by water , Q = M_w*C_{pw} * (T_5-T4) Watts
  = 0.014*4187*(35.7-31.7)
  Q = 234.472 Watts

- T_{mean} = ((113.7 + 83.1)) / 2 = 98.4^0C

  At 100^0C,

  ρ = 961 kg/m^3

  μ = 2.76*10^{-3} Nm/s

  K = 0.6804 w/m-k

- **Theoretical heat transfer co-efficient**

  $h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

  $h = 0.943 * [(2226.2*961^2*9.81*(0.6804)^3)/(4*2.76*10^{-3}*(113.7-83.1)*0.140)]^{.25}$

  h = 570.88 w/m^2 ^{0}C

- LMTD = θ_i-θ_0/ln(θ_i/θ_0)

  θ_i = T_1-T_4 = 113.7-31.7 = 82 ºC

  θ_0 = T_1-T5 = 113.7-35.7 =78 ºC

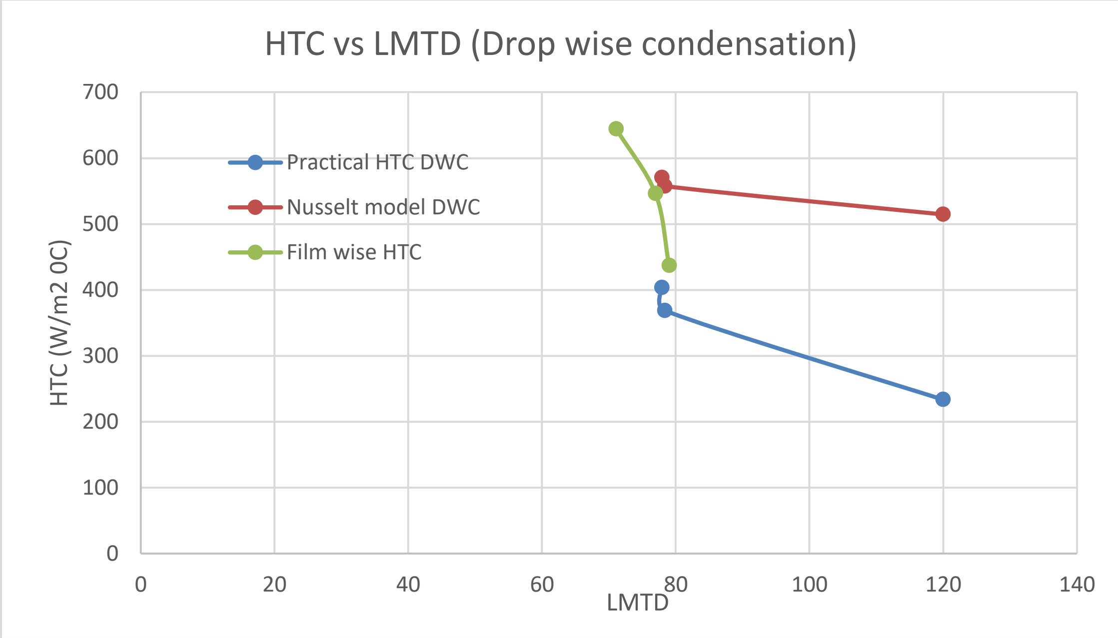
  LMTD = 82-78/ln(82/78)

  LMTD = 119.97

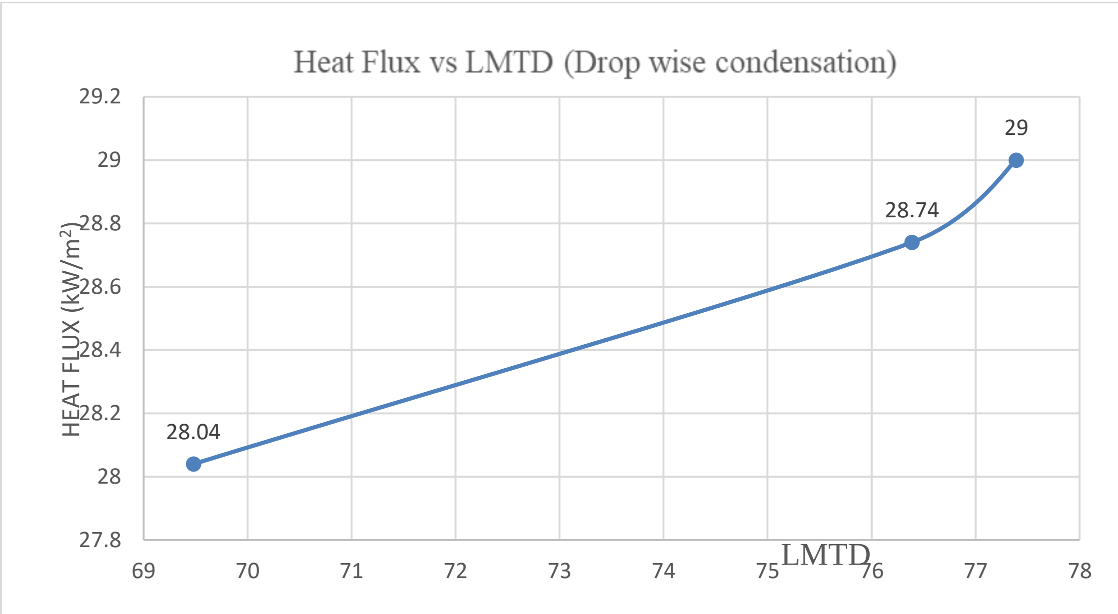
- **Experimental heat transfer co-efficient** , h = Q/a*LMTD

  h = 234.47/0.00836*119.97

  h = 233.77 w/m^2 ºC



Graph 1:HTC vs LMTD (no coated surfaces) DWC



Graph 2:HEAT FLUX vs LMTD (no coated surfaces) DWC

Observations: (wax coated surface)

S.no	Mass Flow rate, m (LPH)	T $_1$(^{0}C)	T$_2$ (^{0}C)	T$_3$ (^{0}C)	T$_4$ (^{0}C)	T$_5$ (^{0}C)	T$_6$ (^{0}C)
1	50 LPH	102.5	69.4	67.2	34.5	40.5	40.7
2		122.1	92.9	87.9	37.3	41.3	46.2
3		118.4	93.3	86.2	36.8	41.6	45.3

Table 4: Observations for wax coated surfaces(Copper & SS Tube)

T_1 - steam (or) condenser temperature, (^{0}C)

T_2 - stainless steel tube surface temperature, (^{0}C)

T_3 - copper tube surface temperature, (^{0}C)

T_4 - cooling water inlet temperature, (^{0}C)

T_5 - stainless steel tube outlet water temperature, (^{0}C)

T_6 - copper tube outlet water temperature, (^{0}C)

Calculations:

1. First observation calculation

Area of the condenser , $A = \pi DL$

Where,

D = diameter of the condenser

L = length of the condenser

$A = \pi * 0.0.19 * 0.0.140$

$A = 0.00836\ m^2$

- Mass flow rate , $M_w = 0.00028 * m$ kg/s

  $M_w = 0.00028*50$

  $M_w = 0.014$ kg/s

- At pressure 1.5 bar

  $T_{sat} = 102.5\ ^0C$

  $T_s = 67.2\ ^0C$

  $H_{fg} = 2226.2$ kj/kg

  $T_{mean} = ((102.5 + 67.2)) / 2 = 84.85^0C$

  At 80^0C,

  $\rho = 974\ kg/m^3$

  $\mu = 3.47*10^{-3}$ Nm/s

  K = 0.6687 w/m-k

For Film wise condensation:

- Heat carried away by water , $Q = M_w*C_{pw} * (T_6\text{-}T4)$ Watts

  $= 0.014*4187*(40.7-34.5)$

  Q = 363.43Watts

- Theoretical heat transfer co-efficient

$h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*974^2*9.81*(0.6687)^3)/(4*3.47*10^{-3}*(102.5-67.2)*0.140)]^{.25}$

$h = 516.95\ w/m^2\ {}^0C$

- LMTD = $\theta_i-\theta_0/\ln(\theta_i/\theta_0)$

$\theta_i = T_1-T_4 = 102.5-34.5 = 64\ {}^oC$

$\theta_0 = T_1-T_6 = 102.5-40.7= 61.8\ {}^oC$

LMTD = 64-61.8/ln(64/61.8)

LMTD = 62.89

- Experimental heat transfer co-efficient , h = Q/a*LMTD

h = 363.43/0.00836*62.89

$h = 691.2\ w/m^2\ {}^oC$

For drop wise condensation:

- Heat carried away by water , $Q = M_w*C_{pw} * (T_5-T4)$ Watts

= 0.014*4187*(40.5-36.5)

Q = 234.47 Watts

- $T_{mean} = ((102.5 + 69.4)) / 2 = 85.95^0C$

At 90^0C,

$\rho = 967.5\ kg/m^3$

$\mu = 3.1178*10^{-3}$ Nm/s

K = 0.6745 w/m-k

- **Theoretical heat transfer co-efficient**

$h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*967.5^2*9.81*(0.6745)^3)/(4*3.117*10^{-3}*(102.5-69.4)*0.140)]^{.25}$

$h = 541.307\ w/m^2\ {}^0C$

- LMTD = $\theta_i - \theta_0 / \ln(\theta_i/\theta_0)$

  $\theta_i = T_1 - T_4 = 102.5 - 36.5 = 66\ ^oC$

  $\theta_0 = T_1 - T5 = 102.5 - 40.5 = 62\ ^oC$

  LMTD = 66-62/ln(66-62)

  LMTD = 63.979

- **Experimental heat transfer co-efficient** , h = Q/a*LMTD

  h = 234.47/0.00776*63.979

  h = 438.37 w/m^2 ^{o}C

2. **Second observation calculation**

   At pressure 1.5 bar

   $T_{sat} = 122.1\ ^0C$

   $T_s = 87.9\ ^0C$

   $H_{fg} = 2226.2$ kj/kg

   $T_{mean} = ((122.1 + 87.9)) / 2 = 105^0C$

   At 110^0C,

   $\rho = 953$ kg/m^3

   $\mu = 2.52*10^{-3}$ Nm/s

   K = 0.6827 w/m-k

For Film wise condensation:

- Heat carried away by water , Q = $M_w*C_{pw} * (T_6 - T4)$ Watts

  = 0.014*4187*(46.2-37.3)

  Q = 521.7 Watts

- Theoretical heat transfer co-efficient

  $h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

  $h = 0.943 * [(2226.2*953^2*9.81*(0.6827)^3)/(4*2.52*10^{-3}*(122.1-87.9)*0.140)]^{.25}$

  h = 567.06 w/m^2 ^{0}C

- LMTD = $\theta_i - \theta_0 / \ln(\theta_i/\theta_0)$

  $\theta_i = T_1 - T_4 = 122.1-37.3 = 84.8\ ^oC$

  $\theta_0 = T_1 - T_6 = 122.1-46.2 = 75.9\ ^oC$

  LMTD = 84.8-75.9/ln(84.8/75.9)

  LMTD = 80.27

- Experimental heat transfer co-efficient , h = Q/a*LMTD

  h = 521.7/0.00836*80.27

  h = 777.4 w/m^2 ^{o}C

For drop wise condensation:

- Heat carried away by water , Q = M_w*C_{pw} * (T_5-T_4) Watts

  = 0.014*4187*(41.3-37.3)

  Q = 234.472 Watts

- T_{mean} = ((122.1 + 92.9)) / 2 = 107.5^{0}C

  At 110^{0}C,

  ρ = 953 kg/m^3

  μ = 2.52*10^{-3} Nm/s

  K = 0.6827 w/m-k

- **Theoretical heat transfer co-efficient**

  $h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

  $h = 0.943 * [(2226.2*953^2*9.81*(0.6827)^3)/(4*2.52*10^{-3}*(122.1-92.9)*0.140)]^{.25}$

  h = 589.92 w/m^2 ^{0}C

- LMTD = $\theta_i - \theta_0 / \ln(\theta_i/\theta_0)$

  $\theta_i = T_1 - T_4 = 122.1-37.3 = 84.8\ ^oC$

  $\theta_0 = T_1 - T5 = 122.1-41.3 = 80.8\ ^oC$

  LMTD = 84.8-80.8/ln(84.8/80.8)

  LMTD = 82.78

- **Experimental heat transfer co-efficient** , h = Q/a*LMTD

h = 234.472/0.00836*82.78

h = 338.8 w/m^2 ^{o}C

3. **Third observation calculation**

At pressure 1.5 bar

T_{sat} = 118.4 ^{0}C

T_s = 86.2 ^{0}C

H_{fg} = 2226.2 kj/kg

T_{mean} = ((118.4 + 86.2)) / 2 = 102.3^{0}C

At 100^{0}C,

ρ = 961 kg/m^3

μ = 2.76*10^{-3} Nm/s

K = 0.6804 w/m-k

For Film wise condensation:

- Heat carried away by water , Q = M_w*C_{pw} * (T_6-T4) Watts

= 0.014*4187*(45.3-36.8)

Q = 498.25 Watts

- Theoretical heat transfer co-efficient

h = 0.943*[h_{fg}ρ^{2}k^{3}g/4μ(T_{sat} - T_s)L]$^{0.25}$

h = 0.943 * [(2226.2*961^2*9.81*(0.6804)^3)/(4*2.76*10^{-3}*(118.4-86.2)*0.140)]$^{.25}$

h = 563.66 w/m^2 ^{0}C

- LMTD = $ϴ_i$-$ϴ_0$/ln($ϴ_i$/$ϴ_0$)

$ϴ_i$ = T_1-T_4 = 118.4-36.8 = 81.6 ^{o}C

$ϴ_0$ = T_1-T_6 = 118.4-45.3= 73.1 ^{o}C

LMTD = 81.6-73.1/ln(81.6-73.1)

LMTD = 77.27

- Experimental heat transfer co-efficient , h = Q/a*LMTD

$$h = 498.25/0.1073*77.27$$

$$\mathbf{h = 771.3\ w/m^2\ {}^oC}$$

For drop wise condensation:

- Heat carried away by water , $Q = M_w*C_{pw} * (T_5-T_4)$ Watts
  $= 0.014*4187*(41.6-36.8)$
  Q = 281.36 Watts

- $T_{mean} = ((118.4 + 93.3)) / 2 = 105.85^0C$

  At 110^0C,

  $\rho = 953\ kg/m^3$

  $\mu = 2.52*10^{-3}$ Nm/s

  K = 0.6827 w/m-k

- **Theoretical heat transfer co-efficient**

  $h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

  $h = 0.943 * [(2226.2*953^2*9.81*(0.6827)^3)/(4*2.52*10^{-3}*(118.4-93.3)*0.140)]^{.25}$

  h = 612.66 w/m² ⁰C

- $LMTD = \theta_i - \theta_0/\ln(\theta_i/\theta_0)$

  $\theta_i = T_1-T_4 = 118.4-36.8 = 81.6\ {}^oC$

  $\theta_0 = T_1-T5 = 118.4-41.6 = 76.8\ {}^oC$

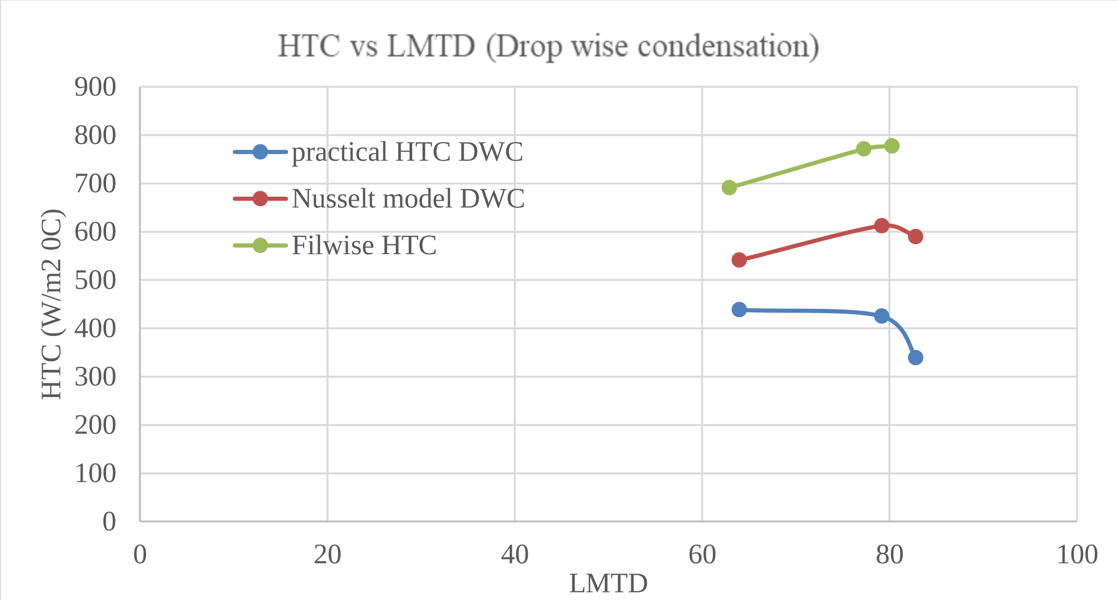
  LMTD = 81.6-76.8/ln(81.6/76.8)

  LMTD = 79.17

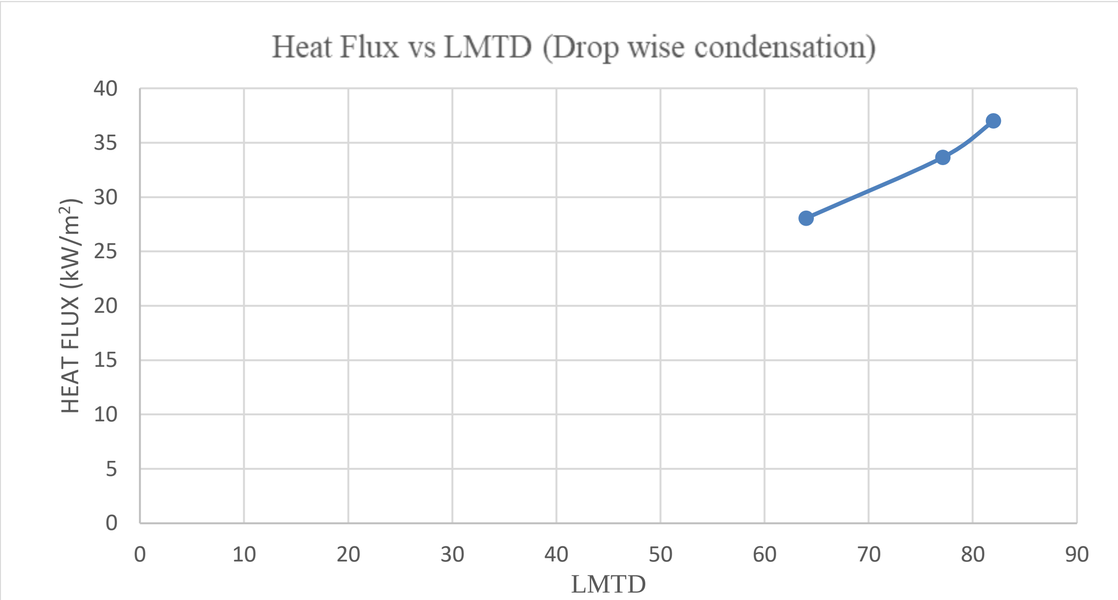
- **Experimental heat transfer co-efficient** , h = Q/a*LMTD

$$h = 281.36/0.00776*79.17$$

$$\mathbf{h = 425.1\ w/m^2\ {}^oC}$$



Graph 3: HTC vs LMTD (wax coated surfaces) DWC



Graph 4:Heat Flux vs LMTD (wax coated surfaces) DWC

Observations: (Aluminum-Tin alloy tube**)**

S.no	Mass Flow rate, m (LPH)	T_1 (^{0}C)	T_2 (^{0}C)	T_3 (^{0}C)	T_4 (^{0}C)	T_5 (^{0}C)	T_6 (^{0}C)
1	50 LPH	110.5	89.2	78	37.5	39.2	41.5
2		118.2	97.8	81.2	38.6	40	44.2
3		121.6	109.2	95.8	37.2	40.5	45

Table 5 : Observations for Aluminum – Tin Tube

T_1 - steam (or) condenser temperature, (^{0}C)

T_2 - stainless steel tube surface temperature, (^{0}C)

T_3 - copper tube surface temperature, (^{0}C)

T_4 - cooling water inlet temperature, (^{0}C)

T_5 - stainless steel tube outlet water temperature, (^{0}C)

T_6 - copper tube outlet water temperature, (^{0}C)

1. First observation calculation

For Film wise condensation:

- Heat carried away by water , $Q = M_w*C_{pw} * (T_6\text{-}T4)$ Watts

$= 0.014*4187*(41.5-37.5)$

Q = 234.47 Watts

- Theoretical heat transfer co-efficient

$h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*961^2*9.81*(0.6804)^3)/(4*2.76*10^{-3}*(110.5-78)*0.140)]^{.25}$

h = 562.35 w/m^2 ^{0}C

- LMTD = θ_i-θ_0/ln(θ_i/θ_0)

$\theta_i = T_1\text{-}T_4 = 110.5-37.5 = 73\ ^oC$

$\theta_0 = T_1\text{-}T_6 = 110.5-41.5= 69\ ^oC$

LMTD = 73-69/ln(73/69)

LMTD = 70.98

- Experimental heat transfer co-efficient , h = Q/a*LMTD

h = 234.47/0.00836*70.98

h = 395.13w/m^2 ^{o}C

For drop wise condensation:

- Heat carried away by water , $Q = M_w*C_{pw} * (T_5-T_4)$ Watts
  $= 0.014*4187*(39.2-37.5)$
  Q = 99.65 Watts

- $T_{mean} = ((110.5 + 89.2)) / 2 = 99.85^0C$

  At 100^0C,

  $\rho = 961\ kg/m^3$

  $\mu = 2.76*10^{-3}\ Nm/s$

  $K = 0.6804\ w/m\text{-}k$

- **Theoretical heat transfer co-efficient**

  $h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

  $h = 0.943 * [(2226.2*961^2*9.81*(0.6804)^3)/(4*2.76*10^{-3}*(110.5-89.2)*0.140)]^{.25}$

  h = 625 w/m² ⁰C

- LMTD = $\theta_i-\theta_0/\ln(\theta_i/\theta_0)$

  $\theta_i = T_1-T_4 = 110.5-37.5 = 73\ ^oC$

  $\theta_0 = T_1-T5 = 110.5-39.2 = 71.3\ ^oC$

  LMTD = 73-71.3/ln(73/71.3)

  LMTD = 72.14

- **Experimental heat transfer co-efficient , h** = Q/a*LMTD

  h = 99.65/0.00776*72.14

  h = 165.2 w/m² ºC

2. Second observation calculation

For Film wise condensation:

- Heat carried away by water , $Q = M_w*C_{pw} * (T_6-T4)$ Watts
  $= 0.014*4187*(44.2-38.6)$
  Q = 328.26 Watts

- Theoretical heat transfer co-efficient

$h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*961^2*9.81*(0.6804)^3)/(4*2.76*10^{-3}*(118.2-81.2)*0.140)]^{.25}$

h = 544.42 w/m² ⁰C

- LMTD = $\theta_i-\theta_0/\ln(\theta_i/\theta_0)$

$\theta_i = T_1-T_4 = 118.2-38.6 = 79.6\ ^oC$

$\theta_0 = T_1-T_6 = 118.2-44.2= 74\ ^oC$

LMTD = 79.6-74/ln(79.6/74)

LMTD = 76.76

- Experimental heat transfer co-efficient , h = Q/a*LMTD

h = 328.26/0.00836*76.76

h = 511.49w/m² ᵒC

For drop wise condensation:

- Heat carried away by water , $Q = M_w*C_{pw} * (T_5-T_4)$ Watts

= 0.014*4187*(41-38.6)

Q = 140.68 Watts

- $T_{mean} = ((118.2 + 97.8)) / 2 = 108^0C$

At 110^0C,

$\rho = 953\ kg/m^3$

$\mu = 2.52*10^{-3}$ Nm/s

K = 0.6827 w/m-k

- **Theoretical heat transfer co-efficient**

$h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*953^2*9.81*(0.6827)^3)/(4*2.52*10^{-3}*(118.2-97.8)*0.140)]^{.25}$

h = 645.26 w/m² ⁰C

- LMTD = $\theta_i - \theta_0 / \ln(\theta_i/\theta_0)$

  $\theta_i = T_1 - T_4 = 118.2 - 38.6 = 79.6\ ^oC$

  $\theta_0 = T_1 - T5 = 118.2 - 41 = 77.2\ ^oC$

  LMTD = 79.6-77.2/ln(79.6/77.2)

  LMTD = 78.39

- **Experimental heat transfer co-efficient ,** h = Q/a*LMTD

  h = 140.68/0.1073*78.39

  h = 214.66 w/m^2 ^{o}C

3. Third observation calculation

For Film wise condensation:

- Heat carried away by water , $Q = M_w * C_{pw} * (T_6 - T_4)$ Watts

  = 0.014*4187*(45-37.5)

  Q = 439.635 Watts

- $T_{mean} = ((118.2 + 97.8)) / 2 = 108^0C$

  At 110^0C,

  $\rho = 953\ kg/m^3$

  $\mu = 2.52*10^{-3}$ Nm/s

  K = 0.6827 w/m-k

- Theoretical heat transfer co-efficient

  $h = 0.943*[h_{fg}\rho^2 k^3 g/4\mu(T_{sat} - T_s)L]^{0.25}$

  $h = 0.943 * [(2226.2*953^2*9.81*(0.6827)^3)/(4*2.29*10^{-3}*(121.6-95.8)*0.140)]^{.25}$

  h = 623.2 w/m^2 ^{0}C

- LMTD = $\theta_i - \theta_0 / \ln(\theta_i/\theta_0)$

  $\theta_i = T_1 - T_4 = 121.6 - 37.5 = 84.1\ ^oC$

  $\theta_0 = T_1 - T_6 = 121.6 - 45 = 76.6\ ^oC$

  LMTD = 84.1-76.6/ln(84.1/76.6)

  LMTD = 80.29

- Experimental heat transfer co-efficient , h = Q/a*LMTD

h = 439.63/0.00836*80.29

h = 654.95w/m^2 ^{o}C

For drop wise condensation:

- Heat carried away by water , Q = M_w*C_{pw} * (T_5-T_4) Watts
  = 0.014*4187*(40.5-37.2)
  Q = 193.43 Watts

- T_{mean} = ((121.6 + 109.2)) / 2 = 115.4^{0}C

  At 120^{0}C,

  ρ = 945 kg/m^3

  μ = 2.29*10^{-3} Nm/s

  K = 0.6850 w/m-k

- **Theoretical heat transfer co-efficient**

  $h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

  $h = 0.943 * [(2226.2*945^2*9.81*(0.6850)^3)/(4*2.29*10^{-3}*(121.6-109.2)*0.140)]^{.25}$

  h = 747.2 w/m^2 ^{0}C

- LMTD = θ_i-θ_0/ln(θ_i/θ_0)

  θ_i = T_1-T_4 = 121.6-37.2 = 84.4 ^{o}C

  θ_0 = T_1-T5 = 121.6-40.5 = 81.1 ^{o}C

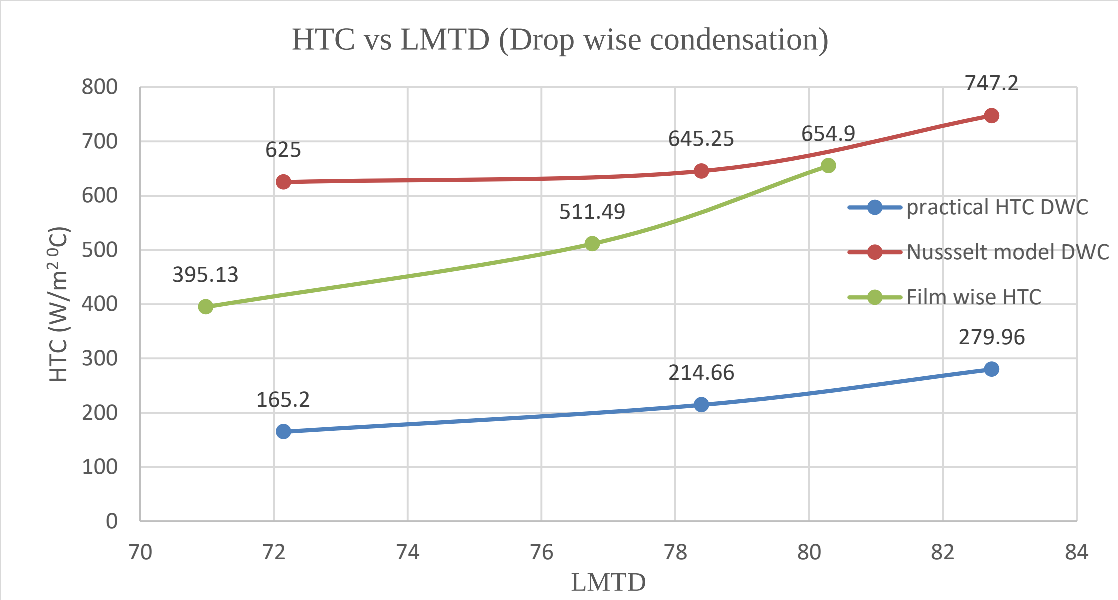
  LMTD = 84.4-81.1/ln(84.4/81.1)

  LMTD = 82.73

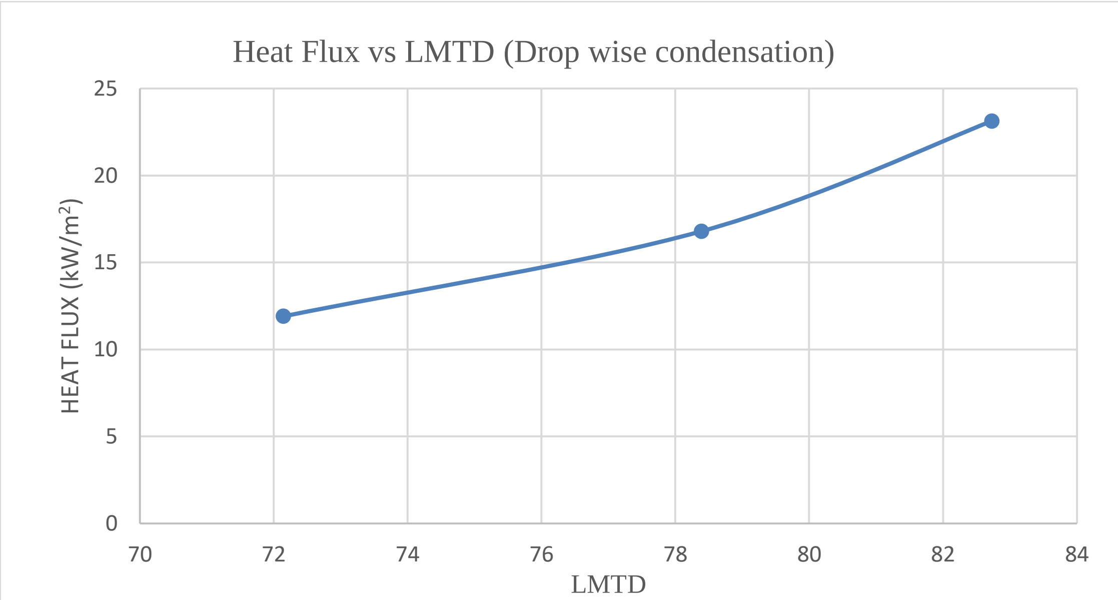
- **Experimental heat transfer co-efficient ,** h = Q/a*LMTD

h = 193.43/0.00776*82.73

h = 279.96 w/m^2 ^{o}C



Graph 5: HTC vs LMTD (Aluminum – Tin alloy) DWC



Graph 6 : Heat flux vs LMTD (Aluminum – Tin alloy) DWC

Observations: (Tin tube)

S.no	Mass Flow rate, m (LPH)	T_1 (^{0}C)	T_2 (^{0}C)	T_3 (^{0}C)	T_4 (^{0}C)	T_5 (^{0}C)	T_6 (^{0}C)
1	50 LPH	118.1	109.2	78.4	36.9	38.7	41.5
2		120.5	116.7	90.7	37.1	39.4	44.4
3		118.8	113.5	94.2	33.9	38.3	43.8

Table 6 : Observations for Tin Tube

T_1 - steam (or) condenser temperature, (^{0}C)

T_2 - stainless steel tube surface temperature, (^{0}C)

T_3 - copper tube surface temperature, (^{0}C)

T_4 - cooling water inlet temperature, (^{0}C)

T_5 - stainless steel tube outlet water temperature, (^{0}C)

T_6 - copper tube outlet water temperature, (^{0}C)

1. First observation calculation

For Film wise condensation:

- Heat carried away by water , $Q = M_w*C_{pw} * (T_6-T_4)$ Watts
  $= 0.014*4187*(41.5-38.7)$
  Q = 164.13 Watts
- $T_{mean} = ((118.1 + 78.4)) / 2 = 98.24^0C$

  At 100^0C,

  $\rho = 961\ kg/m^3$

  $\mu = 2.76*10^{-3}$ Nm/s

  K = 0.6804 w/m-k
- Theoretical heat transfer co-efficient

  $h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

  $h = 0.943 * [(2226.2*961^2*9.81*(0.6804)^3)/(4*2.76*10^{-3}*(118.1-78.4)*0.140)]^{.25}$

  h = 534.9 w/m² ⁰C
- LMTD = $\theta_i-\theta_0/\ln(\theta_i/\theta_0)$

  $\theta_i = T_1-T_4 = 118.1-36.9 = 81.2\ ^oC$

  $\theta_0 = T_1-T_6 = 118.1-41.5 = 76.6\ ^oC$

  LMTD = 81.2-76.6/ln(81.2/76.6)

  LMTD = 78.87
- Experimental heat transfer co-efficient , h = Q/a*LMTD

  h = 164.13/0.00836*78.87

  h =248.9w/m² ºC

For drop wise condensation:

- Heat carried away by water , Q = $M_w * C_{pw} * (T_5 - T_4)$ Watts
  = 0.014*4187*(38.7-36.9)
  Q = 105.51 Watts

- $T_{mean} = ((118.1 + 109.2)) / 2 = 113.65^0C$

  At 110^0C,

  $\rho = 953\ kg/m^3$

  $\mu = 2.52*10^{-3}$ Nm/s

  K = 0.6827 w/m-k

- **Theoretical heat transfer co-efficient**

$h = 0.943*[h_{fg}\rho^2 k^3 g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*953^2*9.81*(0.6827)^3)/(4*2.52*10^{-3}*(118.1-109.2)*0.140)]^{.25}$

h = 657.86 w/m² ⁰C

- LMTD = $\Theta_i - \Theta_0 / \ln(\Theta_i/\Theta_0)$

  $\Theta_i = T_1 - T_4 = 118.1-36.9 = 81.2\ ^oC$

  $\Theta_0 = T_1 - T5 = 118.1-38.7 = 79.4\ ^oC$

  LMTD = 81.2-79.4/ln(81.2/79.4)

  LMTD = 80.29

- **Experimental heat transfer co-efficient** , h = Q/a*LMTD

  h = 105.51/0.00836*80.29

  h = 157.19 w/m² ᵒC

2. Second observation calculation

For Film wise condensation:

- Heat carried away by water , Q = $M_w * C_{pw} * (T_6 - T_4)$ Watts
  = 0.014*4187*(44.4-37.1)
  Q = 427.9 Watts

- T_{mean} = ((120.5 + 90.7)) / 2 = 105.6^{0}C

At 110^{0}C,

ρ = 953 kg/m^3

μ = 2.52*10^{-3} Nm/s

K = 0.6827 w/m-k

- Theoretical heat transfer co-efficient

$h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*953^2*9.81*(0.6827)^3)/(4*2.29*10^{-3}*(120.5-90.7)*0.140)]^{.25}$

h = 601.14 w/m^2 ^{0}C

- LMTD = $\Theta_i-\Theta_0/\ln(\Theta_i/\Theta_0)$

$\Theta_i = T_1-T_4$ = 120.5-37.1 = 83.4 ^{o}C

$\Theta_0 = T_1-T_6$ = 120.5-44.4 = 76.1 ^{o}C

LMTD = 83.4-76.1/ln(83.4/76.1)

LMTD = 79.69

- Experimental heat transfer co-efficient , h = Q/a*LMTD

h = 427.9/0.00836*79.69

h = 642.26 w/m^2 ^{o}C

For drop wise condensation:

- Heat carried away by water , Q = $M_w*C_{pw} * (T_5-T_4)$ Watts

= 0.014*4187*(39.5-37.1)

Q = 140.68 Watts

- T_{mean} = ((120.5 + 116.7)) / 2 = 118.6^{0}C

At 120^{0}C,

ρ = 945 kg/m^3

μ = 2.29*10^{-3} Nm/s

K = 0.6850 w/m-k

- **Theoretical heat transfer co-efficient**

$h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*945^2*9.81*(0.6850)^3)/(4*2.29*10^{-3}*(120.5-116.7)*0.140)]^{.25}$

$h = 832.14\ w/m^2\ {}^0C$

- LMTD = $\theta_i - \theta_0 / \ln(\theta_i/\theta_0)$

$\theta_i = T_1 - T_4 = 120.5-37.1= 83.4\ {}^oC$

$\theta_0 = T_1 - T5 = 120.5-41.4 = 79.1\ {}^oC$

LMTD = 83.4-79.1/ln(83.4/79.1)

LMTD = 81.23

- **Experimental heat transfer co-efficient** , h = Q/a*LMTD

h = 140.68/0.00836*81.23

$h = 207.16\ w/m^2\ {}^oC$

3. Third observation calculation

For Film wise condensation:

- Heat carried away by water , $Q = M_w*C_{pw} * (T_6-T_4)$ Watts

= 0.014*4187*(43.8-33.9)

Q = 580.3 Watts

- $T_{mean} = ((118.8 + 94.2)) / 2 = 106.5^0C$

At 110^0C,

$\rho = 953\ kg/m^3$

$\mu = 2.52*10^{-3}$ Nm/s

K = 0.6827 w/m-k

- **Theoretical heat transfer co-efficient**

$h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

$h = 0.943 * [(2226.2*953^2*9.81*(0.6827)^3)/(4*2.29*10^{-3}*(118.8-94.2)*0.140)]^{.25}$

$h = 630.66\ w/m^2\ {}^0C$

- LMTD = $\theta_i - \theta_0 / \ln(\theta_i/\theta_0)$

  $\theta_i = T_1 - T_4 = 118.8 - 33.9 = 84.9\ ^oC$

  $\theta_0 = T_1 - T_6 = 118.8 - 43.8 = 75\ ^oC$

  LMTD = 84.9-75/ln(84.9/75)

  LMTD = 79.85

- **Experimental heat transfer co-efficient ,** h = Q/a*LMTD

  h = 580.3/0.00836*79.85

  h = 869.33 w/m^2 ^{o}C

For drop wise condensation:

- Heat carried away by water , Q = $M_w * C_{pw} * (T_5 - T_4)$ Watts

  = 0.014*4187*(38.3-33.9)

  Q = 257.92 Watts

- $T_{mean} = ((118.8 + 113.5)) / 2 = 116.15^0C$

  At 120^0C,

  $\rho = 945\ kg/m^3$

  $\mu = 2.29*10^{-3}$ Nm/s

  K = 0.6850 w/m-k

- **Theoretical heat transfer co-efficient**

  $h = 0.943*[h_{fg}\rho^2k^3g/4\mu(T_{sat} - T_s)L]^{0.25}$

  $h = 0.943 * [(2226.2*945^2*9.81*(0.6850)^3)/(4*2.29*10^{-3}*(118.8-113.5)*0.140)]^{.25}$

  h = 924.12 w/m^2 ^{0}C

- LMTD = $\theta_i - \theta_0 / \ln(\theta_i/\theta_0)$

  $\theta_i = T_1 - T_4 = 118.8 - 33.9 = 84.9\ ^oC$

  $\theta_0 = T_1 - T5 = 118.8 - 38.3 = 80.5\ ^oC$

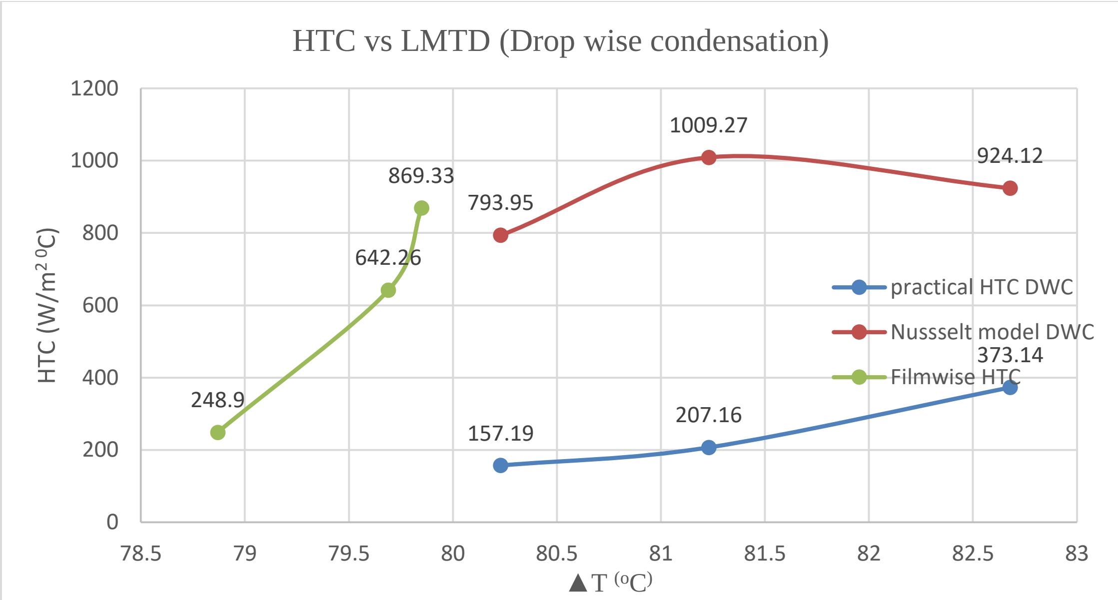
  LMTD = 84.9-80.5/ln(84.9/80.5)

  LMTD = 82.68

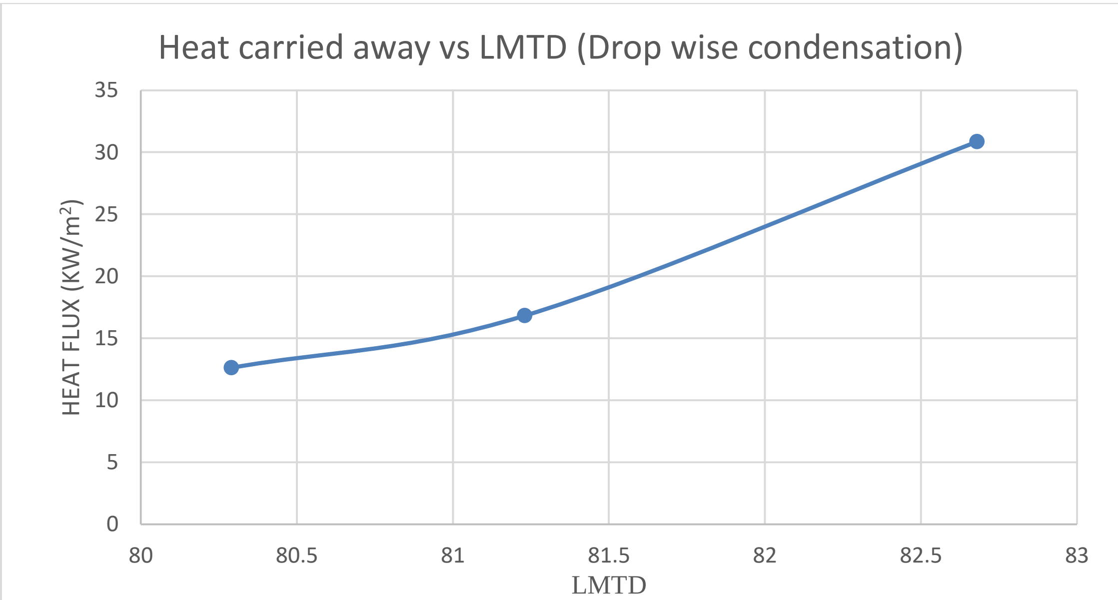
- **Experimental heat transfer co-efficient , h = Q/a*LMTD**

$$h = 257.92/0.00836*82.68$$

$$\mathbf{h = 373.14\ w/m^2\ {}^oC}$$



Graph 7: HTC vs LMTD (Tin tube) DWC

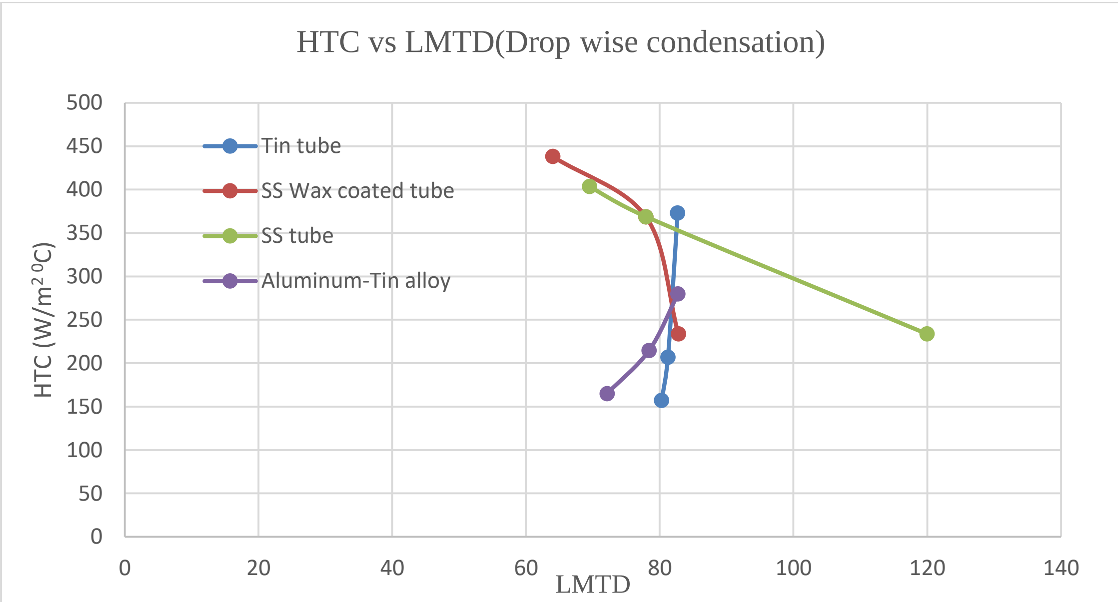


Graph 8 : Heat flux vs LMTD (Tin tube) DWC

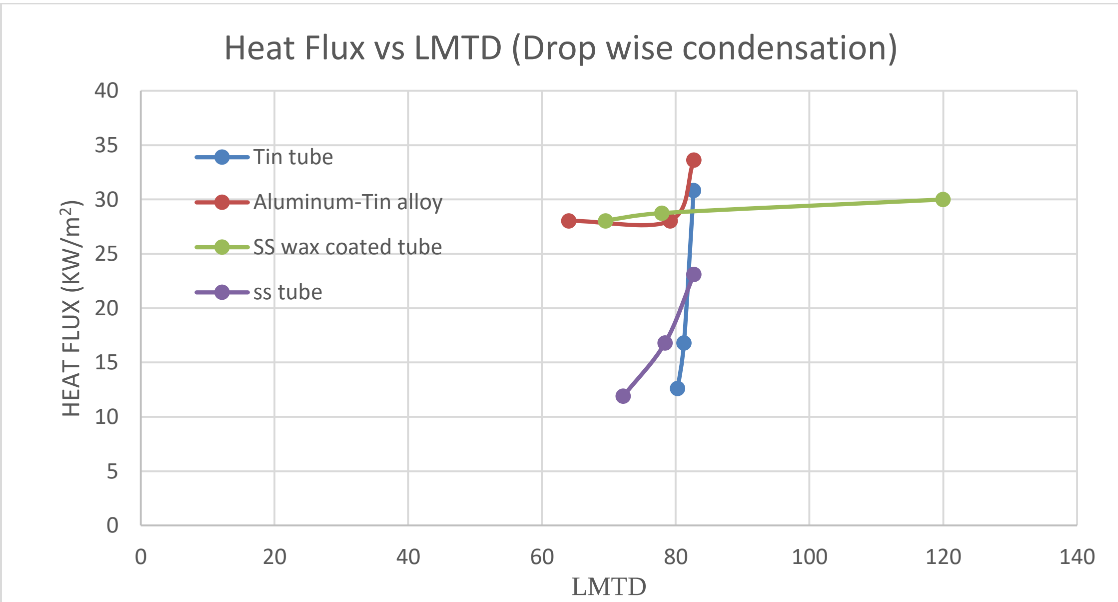
Conclusion

S.No.	Material	Flow rate (kg/s)	Average LMTD	Average Heat flux(kW/m2)	Average HTC,h(w/m2 oC)
1	Stainless steel with wax coating	0.014	74.75	29.91	400.75
2	Stainless steel	0.014	73.77	28.27	335.42
3	Aluminum-Tin Alloy	0.014	77.75	20.09	245.83
4	Tin	0.014	81.4	17.28	219.94

Table 7: Conclusion



Graph 9: Conclusion HTC vs LMTD



Graph 10 : Conclusion Heat flux vs LMTD

From the above experimental observations of condensation of steam, we can observe the heat transfer coefficient for different materials, i.e. for stainless steel, stainless steel with wax coating , aluminum – tin alloy , tin. Among the four materials stainless steel specimens are exhibiting better heat transfer coefficient than the tin & aluminum tubes. A specimen with wax coating is showing more heat transfer coefficient than the bare surfaces. So, from the above investigation we conclude that the metal with coatings exhibits more heat transfer coefficient and better condensation properties.

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