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COLLEGE OF ENGINEERING & TECHNOLOGY

UNIT: III PHASE CHAGNE HEAT TRANSFER AND HEAT EXCHANGERS

1. What is burnout point in boiling neat transfer? Why is it called so? (May /June 2013)

In the Nucleate boiling region, a point at which heat flow is maximum is known as burnout point. Once we cross this point, large temperature difference is required to get the same heat flux and most material may burn at this temperature. Most of the boiling heat transfer heaters are operated below the burnout heat flux to avoid that disastrous effect.

2. Define NTU and LMTD of a heat exchanger. (May/June 2013 & May/June 2016)

LMTD (Logarithmic Mean Temperature Difference)

The temperature difference between the hot and cold fluids in the heat exchanger varies from point in addition various modes of heat transfer are involved. Therefore based on concept of appropriate mean temperature difference, also called logarithmic mean temperature difference, the total heat transfer rate in the heat exchanger is expressed as

$Q = U\,A\,(\Delta T)_m$

Where U – Overall heat transfer coefficient W/m^2K

A – Area m^2

$(\Delta T)_m$ – Logarithmic mean temperature difference.

NTU (No. of Transfer Units)

It is used to calculate the rate of heat transfer in heat exchangers, when there is insufficient information to calculate the Log-Mean Temperature Difference (LMTD). In heat exchanger analysis, if the fluid inlet and outlet temperatures are specified or can be determined, the LMTD method can be used; but when these temperatures are not available The NTU or The Effectiveness method is used.

3. What are the different regimes involved in pool boiling? (May/June 2014)

The different boiling regimes observed in pool boiling are

1. Interface evaporation
2. Nucleate boiling
3. Film boiling.

4. Write down the relation for overall heat transfer coefficient in heat exchanger with fouling factor. (May/June 2014)

Overall heat transfer coefficient in heat exchanger

$$\frac{1}{Uo} = \frac{1}{ho} + R_{fo} + \frac{ro}{k}\ln\frac{ro}{ri} + \frac{ro}{ri} R_{fi} + \frac{ro}{ri}\frac{1}{hi}$$

Where R_{fi} and R_{fo} are the fouling factors at inner and outer surfaces.

[HMT Data Book, P.No.157]

5. How heat exchangers are classified? (May/June 2015)

The heat exchangers are classified as follows

1. Direct contact heat exchangers
2. Indirect contact heat exchangers
3. Surface heat exchangers
4. Parallel flow heat exchangers
5. Counter flow heat exchangers
6. Cross flow heat exchangers
7. Shell and tube heat exchangers
8. Compact heat exchangers.

6. What are the limitations of LMTD method? Discuss the advantage of NTU over the LMTD method. (May/June 2015 & Nov/Dec 2012 & Nov/Dec 2013)

The LMTD method cannot be used for the determination of heat transfer rate and outlet temperature of the hot and cold fluids for prescribed fluid mass flow rates and inlet temperatures when the type and size of heat exchanger are specified.

Effectiveness NTU is superior for the above case because LMTD requires tedious iterations for the same.

7. Differentiate between pool and forced convection boiling. (Nov/Dec 2012 & Nov/Dec 2013 & Nov/Dec 2015) (NOV/DEC 2016)

Boiling is called pool boiling in the absence of bulk fluid flow, and flow boiling (or forced convection boiling) in the presence of it.

In pool boiling, the fluid is stationary, and any motion of the fluid is due to natural convection currents and the motion of the bubbles due to the influence of buoyancy. Example: Boiling of water in a pan on top of a stove.

8. What is pool boiling? Give an example for it. (Nov/Dec 2014)

If heat is added to a liquid from a submerged solid surface, the boiling process referred to as pool boiling. In this case the liquid above the hot surface is essentially stagnant and its motion near the surface is due to free convection and mixing induced by bubble growth and detachment.

Example: Boiling of water in a pan on top of a stove.

9. What do you understand by fouling and effectiveness? (Nov/Dec 2014 & Nov/Dec 2015)

The surfaces of heat exchangers do not remain clean after it has been in use for some time. The surfaces become fouled with scaling or deposits. The effect of these deposits affecting the value of overall heat transfer coefficient. This effect is taken care of by introducing an additional thermal resistance called the fouling resistance or fouling factor.

10. Define effectiveness. (May/June 2016)

The heat exchanger effectiveness is defined as the ratio of actual heat transfer to the maximum possible heat transfer.

$$\text{Effectiveness } \varepsilon = \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}}$$

11. What is meant by sub-cooled and saturated boiling? (Nov/Dec 2015)

The sub-cooled boiling or saturated boiling, depending on the bulk liquid temperature.

Sub-cooled boiling:

There is sharp increase in temperature near to the surface but through most of the liquid, temperature remains close to saturation temperature. ($T_\alpha < T_{sat}$)

Saturated boiling:

When the temperature of the liquid equals to the saturation temperature. ($T_\alpha = T_{sat}$)

12. What is a compact heat exchanger? Give applications. (May/June 2016)

Special purpose heat exchangers called compact heat exchangers. They are generally employed when convective heat transfer coefficient associated with one of the fluids is much smaller than that associated with the other fluid.

In variety of applications including,

- Compressed Gas / Water coolers
- Condensers and evaporators for chemical and technical processes of all kinds.
- Oil and water coolers for power machines
- Refrigeration and air-conditioning units

13. What are the assumptions made in Nusselt theory of condensation? (May/June 2016)

1. The plate is maintained at a uniform temperature which is less than the saturation temperature of vapour. ($T_w < T_{sat}$)
2. Fluid properties are constant.
3. The shear stress at the liquid vapour interface is negligible.
4. The heat transfer across the condensate layer is by pure conduction and the temperature distribution is linear.

14. How fouling affect the rate of heat transfer? (May/June 2016)

"Fouling" is any kind of deposit of extraneous material that appears upon the heat transfer surface during the life time of the heat exchanger.

This fouling will cause an additional resistance to heat transfer is introduced and the operational capability of the heat exchanger is correspondingly reduced. In many cases, the deposit is heavy enough to significantly interfere with fluid flow and increase the pressure drop required to maintain the flow rate through the exchanger.

1. **Discuss briefly the pool boiling regimes of water at atmospheric pressure (May/June 2013,May/June 2014,Nov/Dec 2013)**

Boiling is classified as pool boiling or flow boiling, depending on the presence of bulk fluid motion. Boiling is called pool boiling in the absence of bulk fluid flow and flow boiling in the presence of bulk fluid motion.

Boiling takes different forms, depending on the value of the excess temperature $\Delta T_{excess.}$ Four different boiling regimes are observed: natural convection boiling, nucleate boiling, transition boiling, and film boiling. These regimes are illustrated on the boiling curve in fig, which is a plot of boiling heat flux versus the excess temperature.

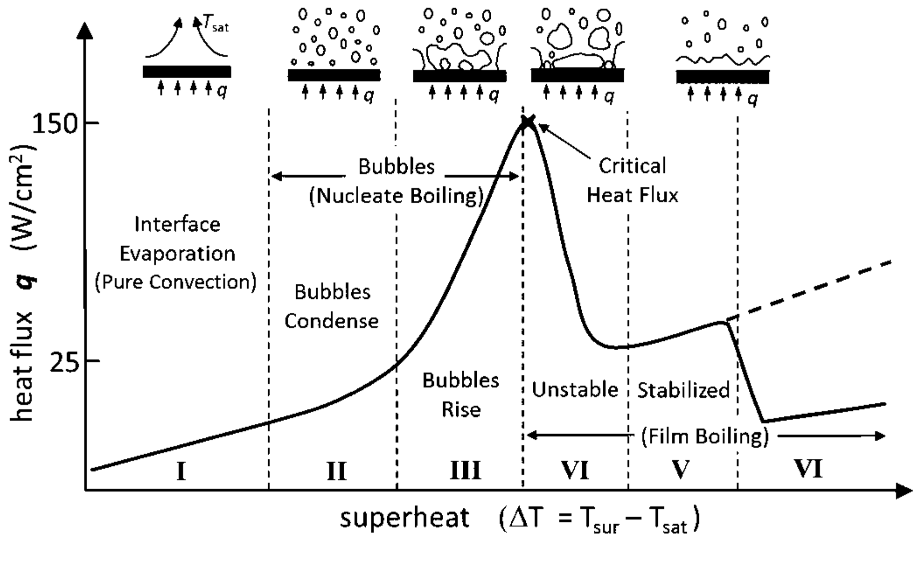


Fig: Typical boiling curve for water at 1 atmospheric pressure

NATURAL CONVECTION BOILING (to point A on the Boiling curve)

We know from thermodynamics that a pure substance at a specified pressure starts boiling when it reaches the saturation temperature at that pressure. But in practice we do not see any bubbles forming on the heating surface until the liquid is heated a few degrees above the saturation temperature (about 2 to 6^{o} C for water). Therefore, the liquid is slightly superheated in this case and evaporates when it rises to the free surface. The fluid motion in this mode of boiling is governed by natural convection currents, and heat transfer from the heating surface to the fluid is by natural convection. For the conditions of fig, natural convection boiling ends at excess temperature of about 5^{o} C.

NUCLEATE BOILING (between points A and C)

The first bubbles start forming at point A of the boiling curve at various preferential sites on the heating surface. Point A is referred to as the onset of nucleate boiling (ONB). The bubbles form at an increasing rate at an increasing number of nucleation sites as we move along the boiling curve toward point C. From fig nucleate boiling exists in the range from about 5^{o} C to about 30^{o}C.

The nucleate boiling regime can be separated into two distinct regions. In regions A-B (5^{o}C $\leq \Delta T_{excess} \leq$ 10 ^{o} C), isolated bubbles are formed at various preferential nucleation sites on the heated surface. But these bubbles are dissipated in the liquid shortly after they separate from the surface. The space vacated by the rising bubbles is filled by the liquid in the vicinity of the heater surface, and the process is repeated. The stirring and agitation caused by the entrainment of the liquid to the heater surface is primarily responsible for the increased heat transfer coefficient and heat flux in this region of nucleate boiling.

In region B-C (10^{o}C $\leq \Delta T_{excess} \leq$ 30 ^{o} C), the heater temperature is further increased, and bubbles form at such great rates at such a large number of nucleation sites that they form numerous continuous columns of vapour in the liquid. These bubbles move all the way up to the free surface, where they break up and release their vapor content. The large heat fluxes obtainable in this region.

At large values of ΔT_{excess}, the rate of evaporation at the heater surface reaches such high values that a large fraction of the heater surface is covered by bubbles, making it difficult for the liquid to reach the heater surface and wet it. Consequently, the heat flux increases at a lower rate with increasing ΔT_{excess}, and reaches a maximum at point C. the heat flux at this point is called the critical heat flux.

TRANSITION BOILING (between points C and D)

As the heater temperature and thus the ΔT_{excess}, is increased past point C, the heat flux decreases, as shown in fig. this is because a large fraction of the heater surface is covered by a vapour film, which acts as an insulation due to the low thermal conductivity of the vapour relative to that of the liquid. In the transition boiling regime, both nucleate and film boiling partially occur. Nucleate boiling at point C is completely replaced by film boiling at point D. for water, transition boiling occurs over the excess temperature range from about 30°C to about 120°C.

FILM BOILING (beyond point D)

In this region the heater surface is completely covered by a continuous stable vapour film. Point D, where the heat flux reaches a minimum, is called the Leidenforst point. The liquid droplets on a very hot surface jump around and slowly boil away. The presence of a vapour film between the heater surface and the liquid is responcible for the low heat transfer rates in the film boiling region. The heat transfer rate increases with increasing excess temperature as a result of heat transfer from the heated surface to the liquid through the vapour film by radiation, which becomes significant at high temperatures.

2. Water is to be boiled at atmospheric pressure in a polished copper pan by means of an electric heater. The diameter of the pan is 0.38 m and is kept at 115° C. calculate the following 1. Power required boiling the water 2. Rate of evaporation 3. Critical heat flux. (Nov/Dec 2012, Nov/Dec 2015)

Given:

Diameter, d = 0.38 m;

Surface temperature, T_w = 115° C.

To find:

1. Power required, (p)
2. Rate of evaporation, ($\dot{m}$)
3. Critical heat flux, (Q/A)

Solution:

Step 1:

Need to find the nucleate pool boiling or film pool boiling process.

ΔT = Excess Temperature = $T_w - T_{sat}$ = Answer, which is less than 50° C then it is Nucleate pool boiling or greater than 50° C then it is film pool boiling.

$$\boxed{\Delta T = T_w - T_{sat}}$$

We know that saturation temperature of water is 100° C. i.e. $T_{sat} = 100^\circ$ C

$\Delta T = 115 - 100 = 15^\circ$ C so this is nucleate pool boiling process.

Step 2:

Need to find the properties of water at 100° C.

(From HMT data book page No. 21)

Density, $\rho_l = 961$ kg/m^3

Kinematic viscosity, $\nu = 0.293 \times 10^{-6}$ m^2/s

Prandtl Number, $P_r = 1.740$

Specific heat, $C_{pl} = 4216$ J/kg K

Dynamic viscosity, $\mu_l = \rho_l \times \nu = 961 \times 0.293 \times 10^{-6} = 281.57 \times 10^{-6}$ Ns/m^2

Enthalpy of evaporation, $h_{fg} = 2256.9$ KJ/kg (from steam table)

Specific volume of vapour, $\nu_g = 1.673$ m^3/kg

Density of vapour, $\rho_v = (1/\nu_g) = 0.597$ kg/ m^3

Step 3:

Need to find the heat flux, power

Heat flux, $\frac{Q}{A} = \mu_l \times h_{fg} \left[\frac{g \times (\rho_l - \rho_v)}{\sigma}\right]^{0.5} \times \left[\frac{C_{pl} \times \Delta T}{C_{sf} \times h_{fg} P_r^n}\right]^3$ ....1 (from HMT data book page no. 142)

Where σ = surface tension for liquid vapour interface at 100° C.

$\sigma = 0.0588$ N/m (from HMT data book page no. 144)

For water – copper → C_{sf} = surface fluid constant = 0.013 and n=1 for water (from HMT data book page no.143)

Substitute the μ_l, h_{fg}, ρ_l, ρ_v, C_{pl}, ΔT, C_{sf}, n and P_r values in equation 1

$\frac{Q}{A}$ = 4.83 × 10^5 W/m^2

Heat transfer Q = 4.83 × 10^5 × A

Area A = $\left(\frac{\pi}{4}\right)d^2$ = 0.113 m^2

$\boxed{\text{Power = 54.7 kW}}$

Step 4:

Need to find Rate of evaporation, ($\dot{m}$)

Heat transferred Q = $\dot{m}$ × h_{fg}

Substitute Q and h_{fg}

$\boxed{\dot{m} = 0.024 \text{ kg/s}}$

Step 5:

Need to find the critical flux

For nucleate pool boiling, critical heat flux,

$$\frac{Q}{A} = 0.18 \times h_{fg} \times \rho_v \left[\frac{\sigma \times g \times (\rho_l - \rho_v)}{\rho_v^2}\right]^{0.25}$$

(from HMT data book page no. 142)

$\boxed{\text{Critical heat flux, } q = \frac{Q}{A} = 1.52 \times 10^6 \text{ W/m}^2}$

3. A wire of 1 mm diameter and 150mm length is submerged horizontally in water at 7 bar. The wire carries a current of 131.5 ampere with an applied voltage of 2.15 Volt. If the surface of the wire is maintained at 180° C, calculate the heat flux and the boiling heat transfer coefficient.(May/June 2014 Reg 2008)

Given:

Diameter, D = 1 mm =1 × 10^{-3} m;

Length, L = 150mm = 150 × 10^{-3} m;

Pressure, P = 7 bar

Voltage, V = 2.15 V

Current, I = 131.5 amps

T_w = 180° C

To find:

1. Heat flux, $\frac{Q}{A}$
2. Heat transfer coefficient, h

Solution:

Step 1:

Need to find heat flux

Q = V × I = 2.15 × 131.5 = 282.72W

A = πDL = π × 1 × 10^{-3} ×150 × 10^{-3} =471.23 × 10^{-6} m^2

Heat flux = $\frac{Q}{A}$ = 282.72/471.23 × 10^{-6} =599.950 × 10^3 W/m^2

$$\boxed{\frac{Q}{A} = 599.950 \times 10^3 \text{ W/m}^2}$$

Step 2:

Need to find the heat transfer co efficient h

At pressure P = 7 bar: ΔT = 180 – 100 =80° C

Heat transfer co efficient, h = 5.56 $(\Delta T)^3$

(From HMT data book page no: 143)

h= 2846720 W/m^2 K

Heat transfer coefficient other than atmospheric pressure

h_p = h P $^{0.4}$ =2846720 × $7^{0.4}$ = 6.19×10^6 W/m^2 K

$$\boxed{h_p = 6.19\times10^6 \text{ W/m}^2\text{ K}}$$

4. A vertical cooling fin approximating a flat plate 40 cm in height is exposed to saturated steam at atmospheric pressure. The fin is maintained at a temperature of 90° C. estimate the thickness of the film at the bottom of the fin, overall heat transfer coefficient and heat transfer rate after incorporating McAdam's correction, the rate of condensation of steam. (Nov/Dec 2015 Reg 2008)

Given:

Height (or) Length, L = 40 cm = 0.4m

Surface temperature, T_w = 90° C

To find:

1. The film thickness δ_x
2. Overall heat transfer coefficient h (McAdam's correction)
3. Heat transfer rate Q
4. Rate of condensation of steam $\dot{m}$

Solution:

Step 1:

We know that, saturation temperature of water is 100° C, i.e. T_{sat} = 100° C

h_{fg}=2256.9 KJ/kg (from steam table)

We know that

Film temperature, $T_f = \frac{T_{w} + T_{sat}}{2}$ = 95°C

Properties of saturated water at 95°C (from HMT data book page no: 21)

Density, ρ_l=967.5 kg/m³

Kinematic viscosity, $\nu=0.328 \times 10^{-6}$ m²/s

Specific heat, C_{pl}=4205.5 J/kg K

Thermal conductivity K =0.674 W/mk

Dynamic viscosity, $\mu_l = \rho_l \times \nu = 967.5 \times 0.328 \times 10^{-6} = 3.173 \times 10^{-4}$ Ns/m²

Step 2:

We need to find the film thickness

$$\delta_x = \left[\frac{4\, \mu\, K\, x\, (T_{sat} - T_w)}{g\, h_{fg}\, \rho_l^{2}}\right]^{0.25}$$ (from HMT data book page no: 148)

substitute all appropriate property value in above formula

$$\boxed{\delta_x = 1.13 \times 10^{-4} \text{ m}}$$

Step 3:

We need to find the heat transfer coefficient h

For vertical surface laminar flow (assume) or find by Re-Reynolds number

$R_e = \frac{4\, \dot{m}}{P \mu}$ here P = perimeter; R_e > 1800 then that flow is turbulent flow,

R_e < 1800 then that flow is laminar flow,

$$h = 0.943 \left[\frac{k^3 \times \rho^2 \times g\, h_{fg}}{\mu \times L \times (T_{sat} - T_w)}\right]^{0.25}$$ (from HMT data book page no: 148)

The factor 0.943 may be replaced by 1.13 for more accurate result as suggested by Mc Adams

$$h = 1.13 \left[\frac{k^3 \times \rho^2 \times g\, h_{fg}}{\mu \times L \times (T_{sat} - T_w)}\right]^{0.25}$$

Substitute all the properties in above formula

$\boxed{h = 1495.3\ W/m^2\ K}$

Step 4:

We need to find the heat transfer rate Q

Q=h A $(T_{sat} - T_w)$ = h L W $(T_{sat} - T_w)$

Q= 1495.3 × 0.4 × 1× 10 = 5981.26 W

$\boxed{Q = 5981.26\ W}$

Step 5:

We need to find the rate of condensation of steam ṁ

Q = ṁ h_{fg}

ṁ = Q/h_{fg}

$\boxed{\dot{m} = 0.00265\ kg/s}$

5. A condenser is to be designed to condense 600 kg/h of dry saturated steam at a pressure of 0.12 bar. A square array of 400 tubes, each of 8 mm diameters is to be used. The tube surface is maintained at 30° C. Calculate the heat transfer coefficient and the length of each tube. (April/May 2015) (NOV/DEC 2013)

Given:

ṁ = 600 kg/h = 0.166 kg/s

Pressure P = 0.12 bar

No. of tubes = 400

Diameter, D = 8 mm = 8 × 10^{-3} m

Surface temperature, T_w = 30° C.

To find:

1. Heat transfer coefficient h

2. Length

Solution:

Step 1:

We need find the properties of steam at 0.12 bar (from steam table)

T_{sat} =49.45 ^{o} C.

h_{fg} = 2384.3 $\times$ 10^3 J/kg

Film temperature, $T_f = \frac{T_{w+} T_{sat}}{2}$ = 39.72^{o}C =40 ^{o} C

Properties of saturated water at 40^{o}C (from HMT data book page no: 21)

Density, ρ_l=995 kg/m^3

Kinematic viscosity, ν=0.657 × 10^{-6} m^2/s

Thermal conductivity K =0.628 W/mk

Dynamic viscosity, $\mu_l = \rho_l \times \nu$ =995 × 0.657× 10^{-6} = 653.7× 10^{-6} Ns/m^2

With 400 tubes, a 20 × 20 tube of square array could be formed

N = $\sqrt{400}$ = 20

Step 2:

We need to find the heat transfer coefficient h

$$h= 0.728 \left[\frac{k^3 \times \rho^2 \times g\, h_{fg}}{\mu \times N\, D \times (T_{sat} - T_w)}\right]^{0.25}$$ (from HMT data book page no: 148)

$\boxed{h= 5304.75\ W/m^2\ K}$

Step 3:

Q=h A $(T_{sat} - T_w)$ = h D L $(T_{sat} - T_w)$ =1.05 × 10^6 L -----1

We know that

Q = ṁ h_{fg} = 0.3957× 10^6 W -----2

Equating (1) and (2) We get, $\boxed{L = 0.37\ m}$

6. **In a double pipe counter flow heat exchanger,10000 kg/hr of an oil having a specific heat of 2095 J/kg-k is cooled from 80^{o}c to 50^{o}c by 800kg/hr of water entering at 25^{o}c. Determine the heat exchanger area**

for an overall heat transfer co-efficient of 300 W/m²k.Take C_P for water as 4180 J/kg-k.

Given:

Hot fluid – oil (T_1-T_2)   Cold fluid - water (t_1-t_2)

The mass flow rate of oil (Hot fluid),m_h =10000 kg/hr

$$= \frac{10000\ kg}{3600\ s}$$

$$\boxed{m_h = 2.277 kg/s}$$

Specific heat of oil, C_{ph} = 2095 J/kg-k

Entry temperature of oil , $T_1 = 80^0C$

Exit temperature of oil , $T_2 = 50^0C$

Mass flow rate of water (Cold fluid),m_c =8000 kg/hr

$$= \frac{8000\ kg}{3600\ s}$$

$$\boxed{m_c = 2.22\ kg/s}$$

Entry temperature of water, $t_1 = 25^0C$

Overall heat transfer co-efficient, U = 300 W/m²k

Specific heat of water, C_{pc} = 4180 J/kg-k

To find:

Heat exchanger area, A

Solution:

Heat lost by oil (Hot fluid) = Heat gained by water (Cold fluid)

$$Q_h = Q_c$$

$$m_h\ C_{ph}\ (T_1\text{-}T_2) = m_c\ C_{pc}\ (t_1\text{-}t_2)$$

$$2.277 \times 2095\ (80\text{-}50) = 2.22 \times 4180 \times (t_2 - 25)$$

$$174.53 \times 10^3 = 9.27 \times 10^3\ t_2 - 231.99 \times 10^3$$

$$t_2 = 43.85^oc$$

$$\boxed{\text{Exit temperature of water, } t_2 = 43.85^oc}$$

Heat transfer,Q = $m_h\ C_{ph}\ (T_1\text{-}T_2)$ or $m_c\ C_{pc}\ (t_1\text{-}t_2)$

$$Q = 2.22 \times 4180 \times (43.85 - 25)$$

We know that, $\boxed{Q = 174.92 \times 10^3 \text{ W}}$

Heat transfer, $Q = UA\,(\Delta T)_m$ .....(1)

Where,

$(\Delta T)_m$ – Logarithmic Mean Temperature Difference. (LMTD)

For counter flow, $(\Delta T)_m = \dfrac{[(T_1 - t_2) - (T_2 - t_1)]}{ln\left[\dfrac{T_1 - t_2}{T_2 - t_1}\right]}$

$$= \frac{[(80 - 43.85) - (50 - 25)]}{ln\left[\frac{80 - 43.85}{50 - 25}\right]}$$

$$\boxed{(\Delta T)_m = 30.23^0 c}$$

Substitute $(\Delta T)_m$, U and Q value in eqn (1)

$$Q = UA\,(\Delta T)_m$$

$$174.92 \times 10^3 = 300 \times A \times 30.23$$

$$\boxed{\text{Heat exchanger area } A = 19.287 \text{ m}^2}$$

7.In a cross flow heat exchangers, both fluids an mixed, hot fluid with a specific heat of 2300 j/kg k ,enters at 380^0 and leaves at 300^0c. Cold fluids enter at 25^0c and leaves 210^0C. Calculate the required surface area of heat exchanger. Take overall heat transfer co-efficient is 750 w/m^{2}k. Mass flow rate of hot fluid is 1Kg/s.

Given:

Specific heat of hot fluid C_{ph} = 2300 J/Kg K

Entry temperature of hot fluid $T_1 = 380^0$ C

Exit temperature of hot fluid $T_2 = 380^0$ C

Entry temperature of Cold fluid $t_1 = 380^0$ C

Exit temperature of Cold fluid $t_2 = 380^0$ C

Overall heat transfer co-efficient, U = 750 w/m^{2}k

The mass flow rate of hot fluid ,m_h =1 kg/s

To find:

Heat exchanger area (A)

Solution:

This is Cross flow, both fluids unmixed type heat exchanger.

For cross flow heat exchanger,

$$Q = F\,UA\,(\Delta T)_{m\ (counter\ flow)} \quad .....(1)$$

[From HMT Data book page No. 152]

Where,

$(\Delta T)_m$ – Logarithmic Mean Temperature Difference for counter flow.

For counter flow, $(\Delta T)_m = \dfrac{[(T_1-t_2)-(T_2-t_1)]}{ln\left[\frac{T_1-t_2}{T_2-t_1}\right]}$

$$= \frac{[(380-210)-(300-25)]}{ln\left[\frac{380-210}{300-25}\right]}$$

$\boxed{(\Delta T)_m = 218.3^0c}$

Heat transfer, $Q = m_h\ C_{ph}\ (T_1\text{-}T_2)$

$Q = 1 \text{ X } 2300 \text{ X } (380\text{-}300)$

$\boxed{Q = 184 \text{ X } 10^3 \text{ W}}$

To find correction factor F, refer HMT data book page No 162

[Single pass cross flow heat exchanger – Both fluids unmixed]

From graph,

X_{axis} value $P = \left[\dfrac{t_2-t_1}{T_1-t_1}\right]$

$= \left[\dfrac{210-25}{380-25}\right]$ X axis Value is 0.52, Curve Value is 0.432,

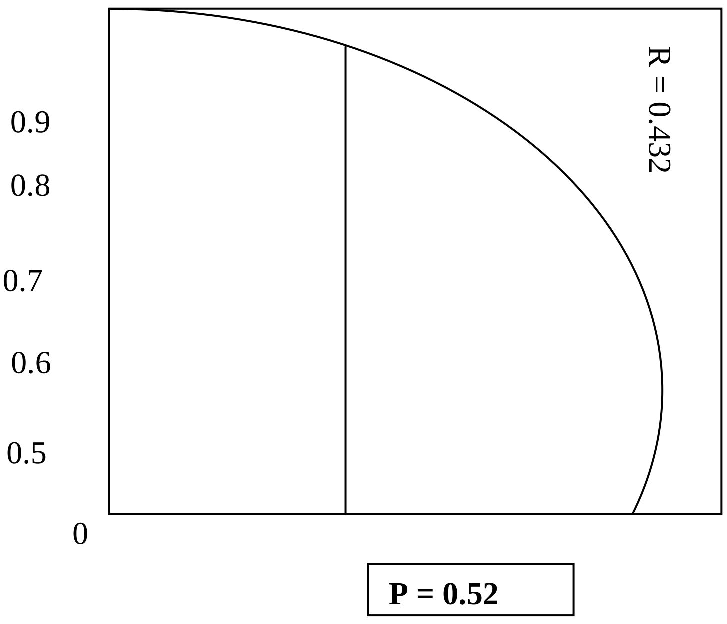
corresponding Y_{axis} Value is 0.97 i,e $\boxed{F = 0.97}$

$P = 0.52$

Curve Value $R = \left[\dfrac{T_1-T_2}{t_2-t_1}\right]$

$= \left[\dfrac{380-300}{210-25}\right]$

$R = 0.432$



Substitute, Q , F $(\Delta T)_{m}$, and U value in eqn (1)

$$Q = F\,UA\,(\Delta T)_m$$

$$184\times10^{3} = 0.97 \times 750 \times A \times 218.3$$

$$\boxed{\text{Surface Area } A = 1.15\,m^2}$$

8. Classify the heat exchangers, draw the temperature distribution in a condenser and evaporator.

There are several types heat exchangers which may be classified on the basis of

I. Nature of heat exchange process

II. Relative direction of fluid motion

III. Design and constructional features

IV. Physical state of fluids.

I.Nature of heat exchange process

On the basis of the nature of heat exchange processes, heat exchangers are classified as

Direct contact heat exchangers or open heat exchangers

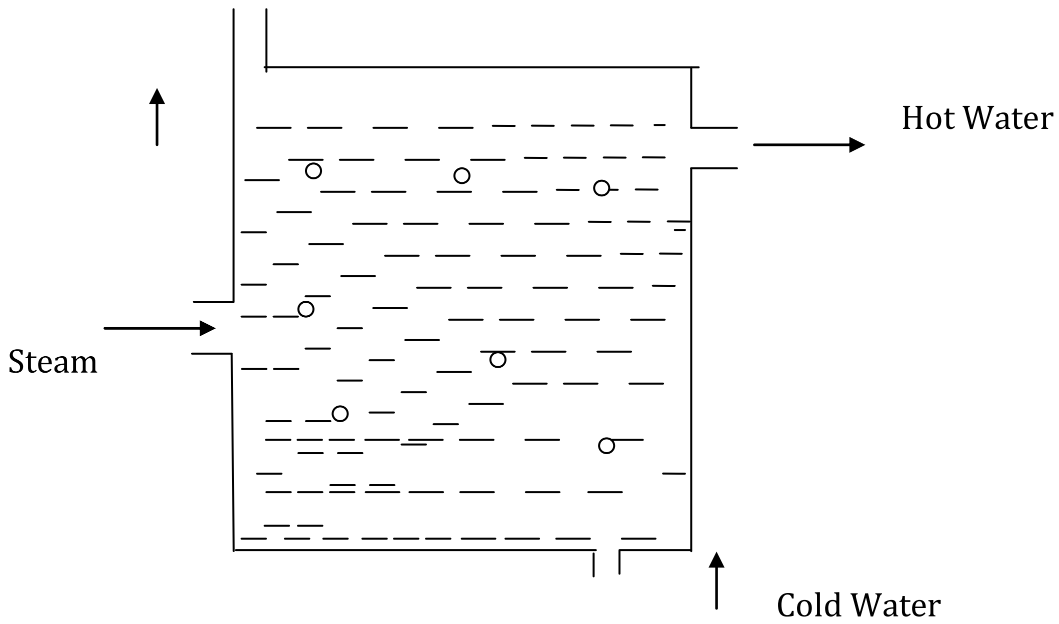
a) Indirect contact heat exchangers

a.Direct contact heat exchangers

The heat exchange takes place by direct mixing of hot and cold fluids. This heat transfer is usually accompanied by mass transfer.

Ex: cooling towers, direct contact feed heaters

Gas



b. Indirect contact heat exchangerscould be carried out by transmission through a wall which separates the two fluids

It may be classified as

i)Regenerators

ii)Recuperators

Regenerators

Hot and cold fluids flow alternately through the same space

Ex: IC engines, gas turbines

Recuperators

This is most common type of heat exchanger in which the hot and cold fluid do not come into direct contact with each other but are seperated by atube wall or a surface.

Ex: Automobile radiators, Air pre heaters,Economisers

Advantages

1.Easy construction

2. More economical

3. More surface area for heat transfer

Disadvantages

1.Less heat transfer co-efficient

2.Less generating capacity

II.Relative direction of fluid motion

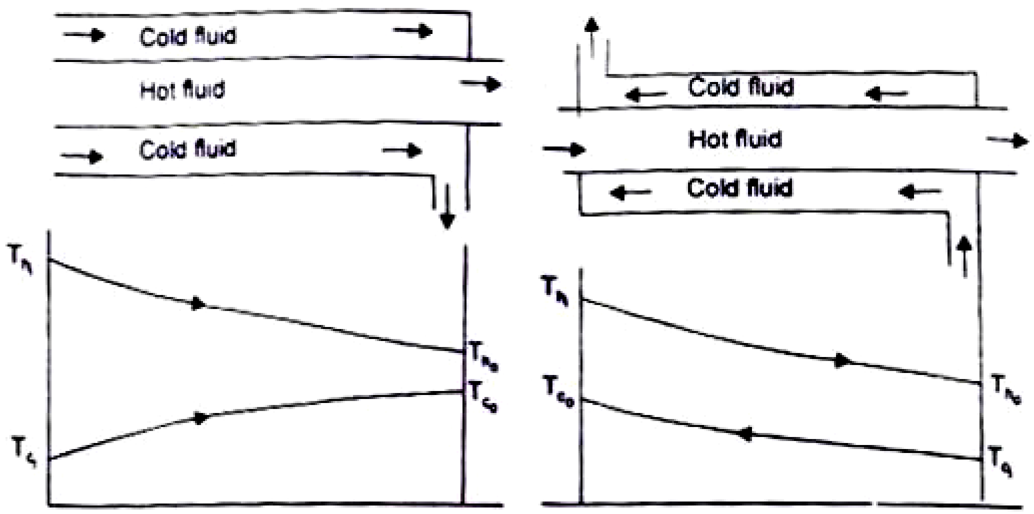
a.Parallel flow heat exchanger

b.Counter flow heat exchanger

c.Cross flow heat exchanger

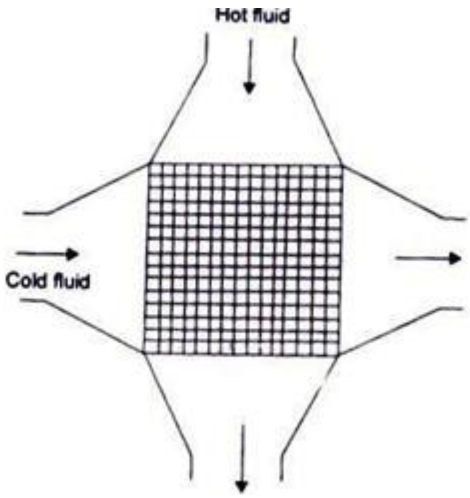
a)Parallel Flow – the hot and cold fluids flow in the same direction. Depicts such a heat exchanger where one fluid (say hot) flows through the pipe and the other fluid (cold) flows through the annulus.

(b) Counter Flow – the two fluids flow through the pipe but in opposite directions. A common type of such a heat exchanger. By comparing the temperature distribution of the two types of heat exchanger



We find that the temperature difference between the two fluids is more uniform in counter flow than in the parallel flow. Counter flow exchangers give the maximum heat transfer rate and are the most favoured devices for heating or cooling of fluids. When the two fluids flow through the heat exchanger only once, it is called one-shell-pass and one-tube-pass

(c) Cross-flow - A cross-flow heat exchanger has the two fluid streams flowing at right angles to each other. illustrates such an arrangement An automobile radiator is a good example of cross-flow exchanger. These exchangers are 'mixed' or 'unmixed' depending upon the mixing or not mixing of either fluid in the direction transverse to the direction of the flow stream and the analysis of this type of heat exchanger is extremely complex because of the variation in the temperature of the fluid in and normal to the direction of flow



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III.Design and constructional features

a.Concentric tubes

b.Shell and tube

c.Multible shell and tube passes

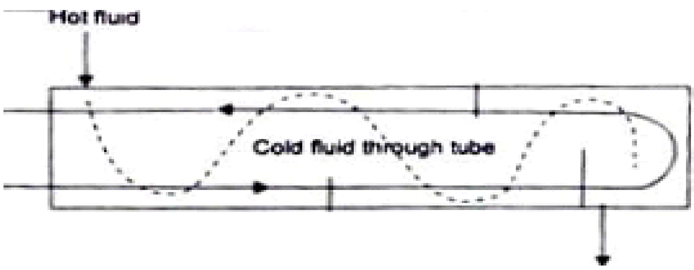
d.Compact heat exchangers

a Concentric tubes

Two concentric pipes ,each carrying one of the fluids are used as a heat exchanger.The direction of flow may be parallel or counter.

b. Shell and tube

One of the fluids move through a bundle of tubes enclosed by a shell.The other fluid is forced through the shell and it moves over the outside surface of the tubes.

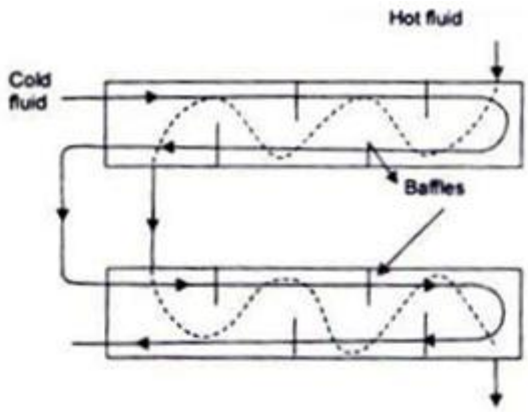


c. Multible shell and tube passes

If the fluid flowing through the tube makes one pass through half of the tube, reverses its direction of flow, and makes a second pass through the remaining half of the tube, it is called 'one-shell-pass, two-tube-pass' heat exchanger, Many other possible flow arrangements exist and are being used. depicts a 'two-shell-pass, four-tube-pass' exchanger.

d.Compact heat exchangers

There are many special purpose heat exchangers called compact heat exchangers.They are generally employed when convective heat transferco-efficient associated with one of the fluids is much smaller than that associated with the other fluid.



IV.Physical state of fluids

a.Condensers

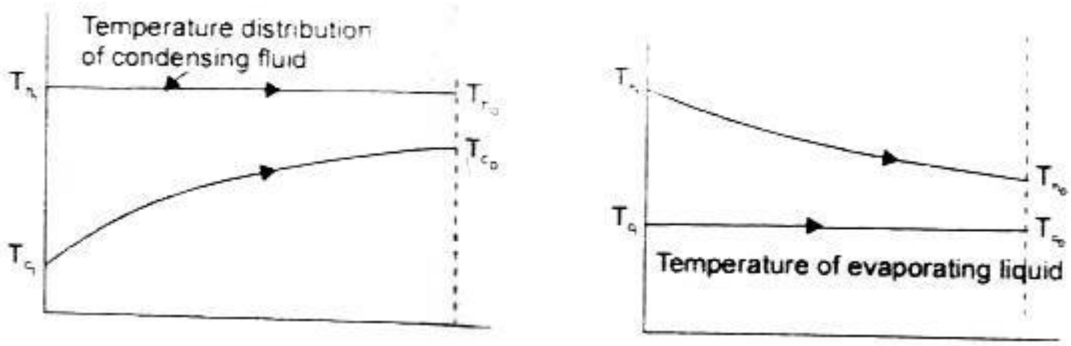
b.Evaporators

a)Condenser

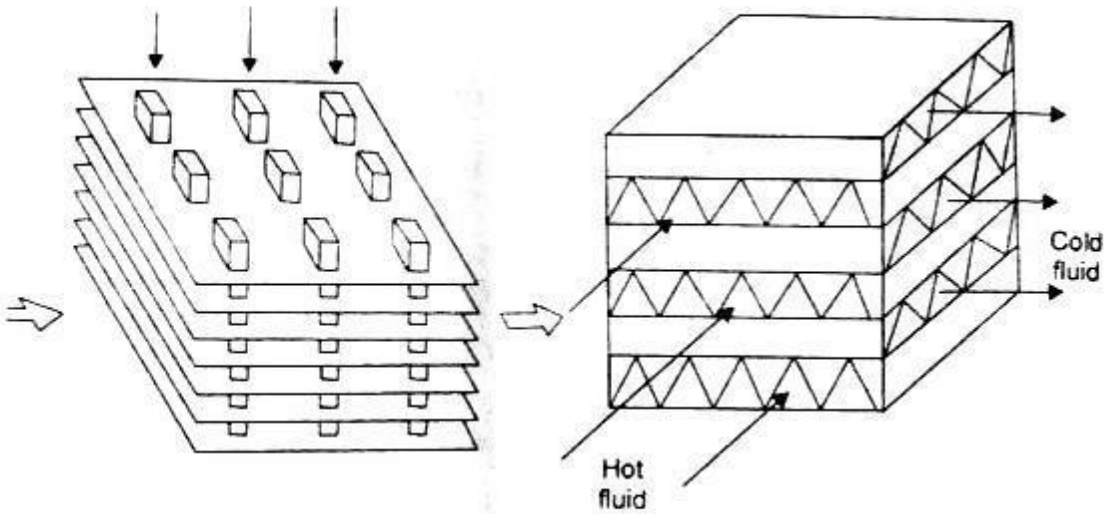
In a condenser, the condensing fluid temperature remains almost constant throughout the exchanger and temperature of the colder fluid gradually increases from the inlet to the exit.

b)Evaporator

Temperature of the hot fluid gradually decreases from the inlet to the outlet whereas the temperature of the colder fluid remains the same during the evaporation process. Since the temperature of one of the fluids can be treated as constant, it is immaterial whether the exchanger is parallel flow or counter flow.



(a) A condenser (b) An evaporator



9. Water at the rate of 4 kg/s is heated from 38°cto 55°c in a shell –and-tube heat exchanger .The water is flow inside tube of 2 cm diameter with an average velocity 35 cm/s. How water available at 95°c and at the rate of 2.0 kg/s is used as the heating medium on the shell side .If the length of tubes must not be more than 2m calculate the number of tube passes , the number of tubes per pass and the length of the tubes for one pass shell, assuming U_0 = 1500 w/m²k.

Given:

M_c = 4 kg/s

T_{CI} =38^0 C

T_{CO} = 55^0C

U = 35m/s

T_{hi} = 95^oc

C_h = 2 kg/s

U_O = 1500 w/m^2k

To find:

1) Number of tubes per pass
2) Number of passes
3) Length of tube per pass

Solution:

The heat transfer rate for the cold fluid is

$$Q = m_c \, c_c \Delta T_C$$

$$= 4 \text{ X } 4186 \; (55-38)$$

$$Q = 284.65 \text{ KW}$$

The exit temperature of hot fluid can be calculated

$Q = m_h C_h \Delta T_h$

= 284.65 kw

$$\Delta T_h = \frac{284650}{4186 \; X \; 2}$$

$= 34^0$C

T_{ho} = 95-34 =61^oc

Counter flow heat exchanger

$$\Delta T_{ln} = \frac{(\Delta T1 - \Delta T2)}{ln \; (\Delta T_1 | \Delta T_2)}$$

$\Delta T_1 = T_{h,\,I} - T_{c,\,o}$

= 95-55 = 40^oc

$\Delta T_2 = T_{h,\,o} - T_{c,\,i}$

= 61-38 = 23^oc

$$\Delta T_{ln} = \frac{(40 - 23)}{ln \; (40|23)} = 30.72^o\text{c}$$

$A = \frac{Q}{U_{\Delta T ln}}$ = 284.65 X 1000 /((1500)X30.72)

= 6.177m^2

Using average velocity of water in the tubes and its flow rates

$m_c = \rho AU$

A = 4/[(1000)(0.35)]

$\boxed{A = 0.011429m^2}$

This area is can also be put as the number of tubes

$0.011429 = n\pi\frac{d^2}{4}$

$\boxed{n = 36.38}$

Taking n = 36, the total surface area of tubes for one shell pass exchanger in terms of L,

A = 6.177 =nπdL

L =6.177/[(36)π(0.02)]

$\boxed{L = 2.731m}$

Since this length is grater than the permitted length of 2m,

$P = \frac{t_o - t_i}{T_I - t_i}$

= 0.3

$R = \frac{T_I - T}{t_o - t_i}$

$\boxed{R = 2}$

Thus the total area required for one shall pass,2 tube pass exchanger is

$A' = Q / [UF\Delta T_{ln}]$

$\boxed{A' = 6.863m^2}$

Due to velocity requirement let the number of tubes pr pass still be 36

A' = 2nπdl

L = 6.863/[2 X 36 X π X0.02]

$\boxed{L = 1.517\ m}$

1. **A nickel wire carrying electric current of 1.5 mm diameter and 50 cm long, is submerged in a water bath which is open to atmospheric pressure.calculate the voltage at the burn out point, if at this point the wire carries a current of 200A.**

Given:

D = 1.5 mm = 1.5 *10^3 m

L =50 cm = 0.50 m

Current , I = 200 A.

To find:

Voltage (v)

Solution:

We know that, saturation temperature of water is 100^o C.

i.e., T_{sat} = 100^o C.

PROPERTIES OF WATER AT 100^0 c .

From HMT Data book page no 21

ρ_l = 961 Kg / m^3

ν = 0.293 * 10^{-6} m^2/s

P_r = 1.740

C_{pl} = 4216 J /Kg k

$\mu_l = \rho_l * \nu$

= 961 * 0.293 * 10^{-6}

= 281.57 * 10^{-6} Ns/ m^2

From steam table at 100^0 c .

h_{fg} = 2256.9 KJ/Kg

hfg = 2256.9 * 10^3 J/Kg

ν_g = 1.673 m^3/Kg

ρ_v = 1/ ν_g = 1/ 1.673

= 0.597 Kg / m^3

σ = surface tension for liquid – vapour interface

At 100^o C (From HMT databook page no 144)

σ = 0.0588 N/m

For Nucleate pool boiling critical heat flux (at burn out)

$Q/A = 0.18 * h_{fg} * \rho_v \; [((\sigma * g * (\rho_l - \rho_v)) / (\rho_v^{\;2}))]^{0.25}$

From HMT databook page no 142

Substitute hfg , ρ_l ,σ , ρ_v

$Q/A = \quad 0.18 * 2256.9 * 10^3 * 0.597\ 8\ [((0.0588 * 9.81 * (961 - 0.597)) / (0.597)^2)]$

$Q/A = \quad 1.52 * 10^6\ W/m^2.$

Heat transferred , Q = V * 1

$Q/A = (V*1) / A$

$1.52 * 10^6 = (V * 200) / (\pi dL)$

$1.52 * 10^6 = ((V * 200) / (\pi * 1.5 * 10^{-3} * 0.50))$

V = 17.9 Volts

2. An oil cooler of the form of tubular heat exchanger cools oil from a temperature of 90° C to 35° C by a large pool of stagnant water assumed at constant temperature of 28° C.The tube length is 32 m and diameter is 28 mm. The specific heat and specific gravity of the oil are 2.45 KJ / Kg K and 0.8 respectively. The velocity of the oil is 62 cm / s. Calculate the overall heat transfer co – efficient.

Given:

Hot fluid – oil	Cold fluid - water
(T1, T2)	(t1 , t2)

Entry temperature of oil T1 = 90° C

Exit temperature of oil T2 = 35° C

Entry and Exit temperature of water , t1 = t2 = 28° C

Tube length L = 32 m

Diameter D = 28 mm = 0.028 m

Specific heat of oil , C_{ph} = 2.45 KJ/Kg k = $2.45 * 10^3$ J/Kg k

Specific gravity of oil = 0.8

Velocity of oil, C =62 cm / s = 0.62 m/s.

To Find:

Overall heat transfer co- efficient U

Solution:

Specific gravity of oil = Density of oil / density of water

$= \rho_0 / \rho_w$

$0.8 = \rho_0 / 1000$

$\rho_0 = 800$ Kg / m^3.

Mass flow rate of oil, $m_h = \rho_0 * A * C$

$= 800 * ((\pi/4)*(D^2)*0.62$

$= 800 * ((\pi/4)*(0.028^2)*0.62$

$m_h = 0.305$ Kg / s.

Heat transfer , $Q = m_h * C_{ph} * (T1 – T2)$

$= 0.305 * 2.45 * 10^3 * (90 – 35)$

$Q = 41 * 10^3$ W.

We know that

Heat transfer , $Q = UA (\Delta T)_m$

From HMT databook page no 151

$(\Delta T)_m$ = logarithmic mean temperature difference (LMTD)

For parallel flow

$(\Delta T)_m = [((T1 – t1) – (T2- t2))] / \ln [((T1 – t1) /(T2 –t2))]$

$= [((90 – 28) – (35- 28))] / \ln [((90 – 28) /(35 –28))]$

$(\Delta T)_m = 25.2^o$ C.

Substitute $(\Delta T)_m$ value in Q Equation

$Q = UA (\Delta T)_m$

$41*10^3 = U * \pi *D* L *(\Delta T)_m$

$41*10^3 = U * \pi *0.028* 32 *25.2$

$U = 577.9$

Overall heat transfer co – efficient , U = 577.9 W / m^2 K