MANUAL
HEAT TRANSFER LABORATORY

CHEMICAL ENGINEERING

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LIST OF EXPERIMENTS

EXPERIMENT NO. 1

SHELL AND TUBE HEAT EXCHANGER

AIM : To determine the overall and individual heat transfer co-efficient under thermal steady state conditions using shell and tube heat exchanger and also to find clean coefficient using correlations.

APPARATUS: Shell and tube heat exchanger setup, thermometers, stop watch, bucket and weighing balance.

THEORY

Shell and tube heat exchangers are used in process industries, where large surface area in a small volume is required, flow rates are high, flow is continuous, hot and cold fluids do not come in contact with each other, to handle wide range of liquids and need to be cleaned frequently. Essentially, it consists of a bundle of tubes enclosed in a cylindrical shell. The ends of the tubes are fitted into tube sheets, which separate the shell-side and tube-side fluids. Baffles are provided in the shell to direct the fluid flow and support the tubes. The assembly of baffles and tubes is held together by support rods and spacers. Many tubes in parallel are usually arranged in an equilateral triangular, square or rotated square pattern, where one fluid flows through these tubes and the other fluid flows outside the tubes in the shell side.

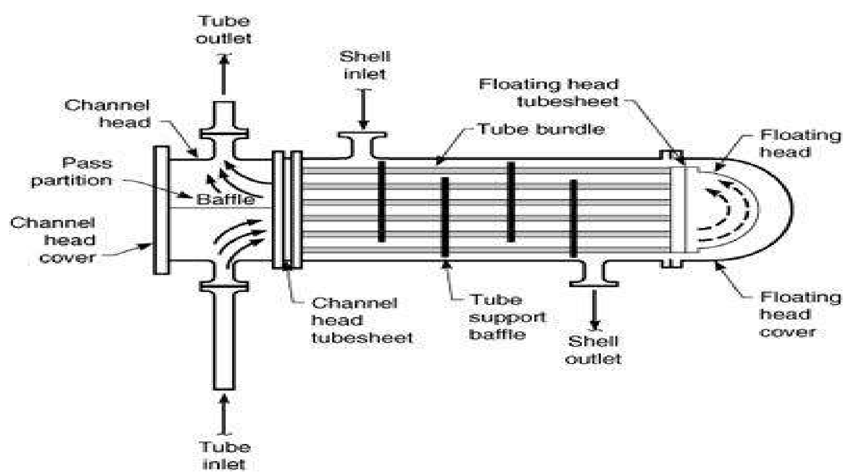


Figure1. Schematic sketch of Shell and Tube Heat Exchanger

The fluid in the tube is usually directed to flow back and forth in a number of passes through groups of tubes arranged in parallel, to increase the length of the flow path. Exchangers are built with from one to about sixteen tube passes. Several designs are available like fixed tube sheet, U-tube, internal or external floating head exchangers.

SPECIFICATIONS OF THE SHELL AND TUBE HEAT EXCHANGER

- Shell inside diameter, D_s = 150 *mm*
- Baffle spacing , B = 200 *mm*
- Number of passes, tube side –1 and Shell side – 1.
- Number of tubes, N = 37
- Length of each tube L= 600 mm
- Outer diameter of tube = d_o = 12.7 *mm*
- Inside diameter of tube = d_i = 9.3 mm
- Pitch is square and p = 23 *mm*
- Clearance , C' = [p – do] mm =10.3 *mm*
- Shell material: Stainless steel, its thermal conductivity, k = 54 *W/m K*.
- Tube material: Copper, its thermal conductivity, k = 386 *W/m K*.

PROCEDURE

1. Fill the tank with water to about ¾ th of its capacity. Adjust the temperature sensor suitably to get the desired temperature (about 70˚C).
2. Switch on the heater and allow the water bath to attain constant temperature. Start the pump to circulate hot water on shell side.
3. The equipment is allowed to attain thermal equilibrium (indicated by steady outlet temperature of hot water).
4. Open the cold water inlet valve and set a constant flow rate on the tube side.
5. Once the steady state is attained (indicated by constant cold water outlet temperature) measure the cold water flow rate using a bucket and a stop watch.
6. Record the inlet and outlet temperatures of hot and cold fluids (T_{in}, T_{out}, t_{in}, t_{out}).
7. Repeat the experiment for different cold water flow rates.

OBSERVATION TABLE

Sl.NO	HOT WATER		COLD WATER				
	T_{in} ˚C	T_{out} ˚C	t_{in} ˚C	t_{out} ˚C	Water collected kg	Time s	Mass flow rate kg/s [m_c]
1							
2							
3							
4							

<u>CALCULATIONS</u>

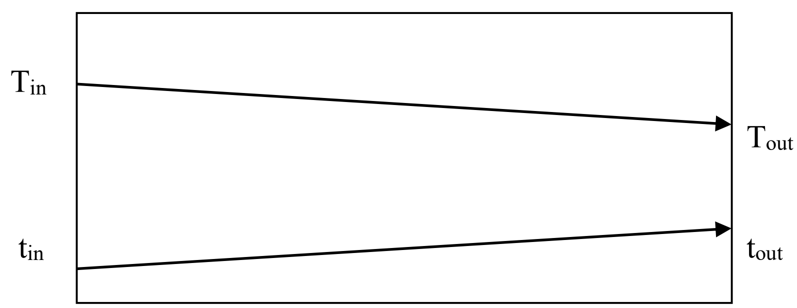
1. Heat load calculation,

$$Q = m_c * C_{pc} * (\otimes t)_c \qquad [W]$$

Where C_{pc} = specific heat of cold fluid at mean temperature, [J/Kg ˚C]

Mean temperature, $t_{c,\,mean} = (t_{out} + t_{in})/2$ and $(\otimes t)_c = t_{out} - t_{in}$.

2. LMTD calculation for co-current flow



Distance from cold fluid inlet, m

T*emperature – length profile*

$\otimes T_1 = T_{in} - t_{in}$

$\otimes T_2 = T_{out} - t_{out}$

$LMTD = (\otimes T_1 - \otimes T_2) / \ln \{\otimes T_1 / \otimes T_2\} \qquad [°C]$

3. Heat transfer area , $A_o = [\angle * do * L * N] \qquad [m^2]$

Where, do is outer diameter of the tube

L is length of the tube

N is the number of tubes

4. TABLE **<u>COLD WATER PROPERTIES AT MEAN TEMPERATURE</u>**

Sl. No.	$t_{c\ mean}$ °C	〉 kg/m^3	⌠ kg/ms	C_p kJ/kg °C	K W/m °C	N_{Pr}
1.						
2.						
3.						
4.						

<u>Calculation of overall heat transfer coefficient based on outside area</u>
$U_{oEXP} = Q / \{ A_o * LMTD \}$ [W / m² °C]

<u>CALCULATION OF THEORETICAL/CLEAN U_O</u>

REFER TABLE OF PROPERTIES OF COLD WATER AT MEAN TEMPERATURE

(From "data sheets provided"), Mean temperature of cold water

$$t_{c\ mean} = \{t_{in}+t_{out}\}/2$$

<u>PROPERTIES OF HOT WATER AT MEAN TEMPERATURE</u>

Mean temperature of hot water $T_{h\ mean} = \{T_{in}+T_{out}\}/2$

Sl. No.	$T_{h\ mean}$ °C	〉 kg/m^3	⌠ kg/ms	C_p kJ/kg °C	K W/m °C	N_{Pr}
1						
2						

3						
4						

TUBE SIDE CALCULATIONS

CALCULATION OF INSIDE FILM COEFFICIENT (COLD WATER SIDE)

5. Volumetric flow rate, F = mass flow rate / density at mean temperature

$$F = m_c / \rho \qquad [m^3 / s]$$

6. Flow area, $a_f = (\pi d_i^2) / 4$ $[m^2]$

Where d_i is the inner diameter of the tube

7. Velocity of fluid in the tube, $v = F / N * a_f$ $[m/s]$

8. Reynolds number, $N_{Re} = [di * v * \rho] / \mu$

where ρ = density at mean temperature

μ = viscosity at mean temperature

9. h_i calculation

Check for flow nature

If N_{Re} <2100 [flow is laminar]

Use $N_{Nu} = 1.86 [N_{Gz}]^{1/3}$

Where N_{Nu} = *Nusselt number* $= h_i d_i / k$

N_{Gz} = *Greatz number* $= (4 * m_c * C_{pc}) / (\pi * k * L)$

k is thermal conductivity of tube side fluid at mean temperature.

L is the total length= l *N.

If N_{Re} > 10,000 [flow is turbulent]

Use Dittus-Boelter equation

$$N_{Nu} = 0.023 * [N_{Re}]^{0.8} [N_{Pr}]^{0.4}$$

N_{Pr}= Prandtl number

& therefore $h_i = [N_{Nu} * k] / d_i$

If 2100 < N_{Re} < 10,000 [flow is in transition region]

(Note: The calculation method is based on graphs of above two equations after modifications on a common plot of the Colburn j_H factor versus N_{Re}.)

Use j_H Vs N_{Re} chart.

First find L/d_o: L/d_o=600/12.7= 47.25. Read j_H to the nearest L/d_o curve corresponding to the N_{Re}, then using j_H value compute h_i.

10. h_{io} calculation: $h_{io}= h_i d_i/d_o$ $[W/m^{2\,o}C]$

SUMMARY OF TUBE SIDE CALCULATIONS

Sl. No.	m_c kg/s	N_{Re}	N_{Pr}	N_{Nu}	h_i $W/m^2\ °C$
1					
2					
3					
4					

SHELL SIDE CALCULATIONS

CALCULATION OF OUTSIDE FILM COEFFICIENT (HOT WATER SIDE)

11. Shell side mass flow rate, $m_h = Q / C_{ph} * (\otimes T)_h$

where C_{ph} is the specific heat of the hot fluid at mean temperature

$$(\otimes T)_h = T_{in} - T_{out}$$

12. Shell side area, $a_h = [Ds * C' * B] / p,$

Shell side fluid mass velocity, $G_h= m_h/a_h$ $[kg/s\ m^2]$

Where D_s is inner diameter of the shell

C' is the clearance between the tubes

B is the baffle spacing

p is the pitch of the tubes

13. Equivalent diameter, $d_e = 4 * [\{ p^2 - \angle / 4 * d_o^2 \} / \angle * d_o]$ **[*m*]**

where p is the square pitch [center to center distance between tubes]

d_o = outside diameter of tubes

14. Calculation of h_o

Donohue equation [McCabe and Smith page no. 388] to find h_o

$$[h_o * d_e] / k = 0.2 * [N_{Re}]^{0.6} * [N_{pr}]^{0.3}$$

Where $N_{Re} = d_e * G_e / \int$

$N_{pr} = C_p * \int / k$

$G_e = \square [G_b * G_c]$

$G_b = m_h / S_b$ and $G_c = m_h / S_c$

And $S_b = f_b * [\{ \angle * D_s^2 \} / 4] - N_b * [\{ \angle * d_0^2 \} / 4]$

Where f_b = fraction of the cross- sectional area of shell occupied by baffle window [Commonly 0.1955]

D_s = inside diameter of shell

N_b = Number of tubes in baffle window = $f_b * N$

d_o = outside diameter of tubes

and $S_c = P * D_s * [1 - \{ d_o / p \}]$

Note: $S_b = 0.00254 m^2$

$S_c = 0.01343\ m^2$

$N_b = 7.234$

SUMMARY OF SHELL SIDE CALCULATIONS

Sl. No.	m_{h}, *kg/s*	N_{Re}	N_{Pr}	h_o *W /m² °C*
1				
2				
3				
4				

15. Calculation of U_o [theoretical]

$$1 / U_o = 1 / h_{io} + 1 / h_o + R_w + R_d$$

Where, R_w is metal wall resistance = $d_o * \ln [d_o / d_i] / 2 * k_w$ $[(m^2 \ °C) / W]$

k_w = thermal conductivity of tube material

R_d is dirt resistance **$[(m^2 °C) /W]$** (Neglect the dirt resistance)

RESULTS TABLE

Sl. No.	**h_i** ***W/m² °C***	**h_o** ***W /m² °C***	**$U_{O,EXPT}$** ***W /m² °C***	**$U_{O,THEO}$** ***W /m² °C***
1				
2				
3				
4				

EXPERIMENT NO. 2

DOUBLE PIPE HEAT EXCHANGER

AIM: To determine the overall heat transfer co-efficient under thermal steady state conditions using double pipe heat exchanger and also to find clean co-efficient.

APPARATUS: Double pipe heat exchanger setup, thermometers, stop clock, bucket, weighing balance.

THEORY

H.T. from a warmer to colder fluid, usually through a solid wall separating the fluids is common in chemical engineering practice. The heat transferred may be latent heat or it may be sensible heat. A device used for exchanging energy in the form of heat between two fluids is known as heat exchanger. This heat exchange can be either direct or indirect method. In direct method heat exchange takes place without mixing. Two fluids are separated by a metallic or non metallic surface through which H.T. takes place generally in the direction perpendicular to flow direction. Heat exchangers may also be fired or unfired type. Heat is transferred from a flame to the fluid by radiation in fired type. Heat exchange is by convection in the fluids and by conduction across solid surface separating two fluids in unfired type. DPHE is used particularly when the flow rates are low, high pressure drops and when the temperature range is relatively high.

In this type one fluid flows inside a pipe, while a second fluid flows either co or counter currently in the annulus between a larger pipe and the outer side of the inner pipe carrying the first fluid. DPHE is assembled using standard metal pipes, return heads and connecting tees. The two lengths of inner pipe are connected by a return bend which is usually exposed and does not provide effective heat transfer surface. When arranged in two legs, the unit is a hairpin. DPHE permits true counter current which is more advantageous when very close temperature approaches are required. The number of sections and series parallel arrangement can be varied to meet the required conditions. DPHE are of greatest use where the total required heat surface is small, 100 to 200ft^2 or less. Disadvantage of it is small amount of heat transfer area contained in a single hairpin. Large number of hairpins requires considerable space, and there will be more leakage points.

Figure 2. Schematic sketch of Double Pipe Heat Exchanger

SPECIFICATIONS OF THE DOUBLE PIPE HEAT EXCHANGER

- Outer diameter of inner pipe = 12.7 *mm*
- Inner diameter of inner pipe = 9.3 *mm*
- Outer diameter of outer pipe = 25 *mm*
- Inner diameter of outer pipe = 21 *mm*
- Straight length of pipe = 1200 *mm*
- Number of hairpins=1.5
- Inner tube and outer pipe material is stainless steel, thermal conductivity of is 54 *W/m℃*

PROCEDURE

1. Fill the tank with water to about ¾ th of its capacity. Adjust the temperature sensor suitably to get the desired temperature (about 70˚C).
2. Switch on the heater and allow the water bath temperature to attain the set temperature.
3. Start the pump to circulate hot water in the annulus side and allow the equipment to attain thermal equilibrium (indicated by steady outlet temperature of hot water).
4. Open the cold water inlet valve and set to a constant flow rate. The system is allowed to attain steady state.
5. Measure the cold water flow rate using a bucket and a stop clock. Record the inlet and outlet temperatures of hot and cold fluids.(T_{in}, T_{out}, t_{in}, t_{out})
6. Repeat the experiment for different cold water flow rates.

OBSERVATION TABLE

Sl. NO.	**HOT WATER**		**COLD WATER**				
	T_{in}, °C	**T_{out} °C**	**t_{in} °C**	**t_{out} °C**	**Water collected *kg***	**Time s**	**Mass flow rate, m_c *kg/s***
1							
2							
3							
4							

CALCULATIONS

1. Heat load calculation

$$Q = m_c * C_{pc} * (\otimes t)_c \qquad [W]$$

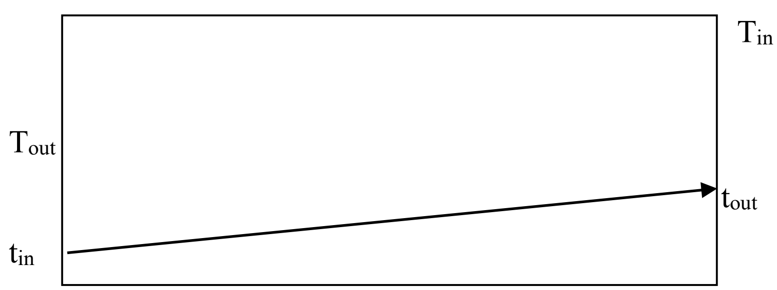
Where m_c= mass flow rate of cold water in kg/s

C_{pc} = specific heat of cold fluid at its mean temperature [J/kg °C]

$$t_{c\ mean} = \{ t_{in} + t_{out} \} / 2$$

$$(\otimes t)_c = t_{out} - t_{in}$$

2. LMTD calculation for counter current flow



Counter current flow pattern

$$\otimes T_1 = T_{out} - t_{in}$$

$$\otimes T_2 = T_{in} - t_{out}$$

$$LMTD = (\otimes T_1 - \otimes T_2) / \ln \{\otimes T_1 / \otimes T_2\} \qquad [°C]$$

3. Heat transfer area, $A_o = [\ \angle\ \ d_o\ L\ \ N\ ]$ $\qquad$ $[m^2]$

Where d_o is outer diameter of the inner tube

L is straight length of the tube.

N is the number of tubes

4. Calculation of overall heat transfer coefficient based on outside area of inner tube

$$U_{oEXP} = Q\ /\ \{ A_o * LMTD \} \qquad [\ W/m^2\ °C\ ]$$

CALCULATION OF U_{theo}

TABLE OF PROPERTIES of COLD WATER at $t_{c\ mean}$

Sl. No.	$t_{c\ mean}$ °C	〉 kg/m³	⎧ kg/m s	c_p J/kg °C	K W/m °C	N_{Pr}
1						
2						
3						
4						

PIPE {COLD WATER} SIDE CALCULATIONS

CALCULATION OF INSIDE FILM COEFFICIENT, h_i

5. Volumetric flow rate, F= mass flow rate / density at mean temperature

$$F = m_c\ /\ 〉 \qquad [m^3\ /\ s]$$

6. Flow area, $\quad a_p = (□ * d_i^2)\ /\ 4 \qquad [m^2]$

7. Velocity in the tube, $\quad v = F\ /\ a_p \qquad [m/s]$

8. Reynolds number, $\quad N_{Re} = [\ d_i * v * 〉]\ /\ ⎧$

Where, 〉 = density at mean temperature

⎧ = viscosity at mean temperature

9. h_i calculation

Check for flow nature

If N_{Re} <2100 [flow is laminar]

Use $N_{Nu} = 1.86\ [N_{Gz}]^{1/3}$

Where N_{Nu} = Nusselt number= $h\ d_i / k$

N_{Gz} = Greatz number = $(4 * m_c * c_{pc})\ /\ (\pi * k * L)$

k = thermal conductivity of cold water at mean temperature, **W/mK**

If N_{Re} > 6000 [flow is turbulent]

Use Dittus Boelter equation, $N_{Nu} = h\ d_i / k = 0.023 * [N_{Re}]^{0.8}\ [N_{Pr}]^{0.4}$

Where N_{Pr} = Prandtl number

$= c_p \mu / k$ (Can also be obtained from properties table)

n =0.4 since the fluid is heated

$$h_i = [\ N_{Nu} * k\ ] / di \qquad [\ W/m^2\ °C]$$

If 2100 < N_{Re} <6000 [flow is in transition region]

(Note: No simple equation applies here. The calculation method is based on graphs of Sieder-Tate equation, $N_{Nu} = h\ d_i / k = 0.023 * [N_{Re}]^{0.8}\ [N_{Pr}]^{0.4}\ [\ \mu / \mu_w]^{0.14}$

after modifications on a common plot of the Colburn j_H factor versus N_{Re}.)

Use j_H Vs N_{Re} chart. First find L/d_o: $L/d_o = 1200*3/12.7 = 283.5$

Read j_H to the nearest L/d_o curve corresponding to the N_{Re}, then using j_H value compute h_i.

10. Calculation of h_{io}(Inside film coefficient based on outer surface of inner tube)

$$h_{io} = h_i\ [\ d_i / d_o\ ]$$

SUMMARY OF PIPE SIDE CALCULATIONS

Sl. No.	m_c kg/s	N_{Re}	N_{Pr}	N_{Nu}	h_I $W/m^2\,°C$	h_{io} $W/m^2\,°C$
1.						
2.						
3.						
4.						

ANNULUS {HOT WATER} SIDE CALCULATIONS

PROPERTIES OF HOT WATER AT $T_{h\ mean}$ [$T_{h\ mean}$ = {T_{in}+T_{out}}/2]

Sl. No.	$T_{h\ mean}$ °C	〉 kg/m³	⌠ kg/m s	C_p kJ/kg°C	k W/m K	N_{Pr}
1.						
2.						
3.						
4.						

CALCULATION OF HOT WATER (OUTSIDE) FILM COEFFICIENT, h_O :

11. Shell side mass flow rate

Assumption: Under steady state condition, heat lost by hot fluid is equal to the heat gained by cold fluid.

Shell side mass flow rate, $m_h = Q / c_{ph} * (⊗T)_h$ [Kg /s]

where c_{ph} is the specific heat of the hot fluid at mean temperature

$$T_{h\ mean} = \{T_{in}+T_{out}\}/2 \text{ and } (⊗T)_h = T_{in} - T_{out}$$

12. Flow area, $A_a = \angle (D_i^2 - d_o^2) / 4$ [m^2]

where D_i = inner diameter of outer pipe and d_o = outer diameter of inner pipe.

13. Equivalent diameter, $D_e = [D_i^2 - d_o^2] / d_o$ [m]

14. Mass velocity $\boldsymbol{G_a = m_h / A_a}$ $\boldsymbol{[kg / s\ \ m^2]}$

15. Reynold's number, $\boldsymbol{N_{Re} = [\ D_e\ *\ G_a\ ] /}$ ⌠

where c_p **,**⌠, k are properties of hot fluid at mean temperature.

16. Computation of h_o:

Check for flow nature

If $N_{Re} < 2100$ [flow is laminar]

Use $\boldsymbol{N_{Nu} = 1.86\ [N_{Gz}]^{1/3}}$

Where N_{Nu} = Nusselt number= $\boldsymbol{h_o\ D_e / k}$

N_{Gz} = Greatz number = $\boldsymbol{(\ 4\ *\ m_c * C_{pc}) \ / \ (}$□ $\boldsymbol{* \ k\ *\ L)}$

k =Thermal conductivity of annulus side fluid at mean temperature.

If $N_{Re} > 6000$ [flow is turbulent]

Use $\boldsymbol{N_{Nu} = \ 0.023\ \ [N_{Re}]^{0.8}\ [N_{Pr}]^{\ n}}$

Where N_{Pr} = Prandtl number

=$\boldsymbol{C_p}$ ⌠ $\boldsymbol{/k}$ (Can also be obtained from properties table)

n =0.33 since the fluid is cooled & $\boldsymbol{h_o = \ [\ N_{Nu}\ \ * \ k\ ] / D_e}$ $\boldsymbol{[\ W/m^2\ {}^{\circ}C]}$

If 2100 < N_{Re} < 6000 [flow is in transition region]

(Note: No simple equation applies here. The calculation method is based on graphs of above two equations after modifications on a common plot of the Colburn j_H factor versus $N_{Re.}$)

Use j_H Vs N_{Re} chart..

First find $\boldsymbol{L/ D_e}$**:** $\boldsymbol{L/D_e=1.2*3/.022= 163.64}$

Read j_H to the nearest L/ d_o

First find $\boldsymbol{L/d_o}$**:** $\boldsymbol{L/d_o=1200*3/12.7= 283.5}$

Read j_H to the nearest L/ D_e curve corresponding to the N_{Re}, then using j_H value compute h_0.

SUMMARY OF ANNULUS CALCULATIONS

Sl. No	m_h kg/s	N_{Re}	N_{Pr}	N_{Nu}	h_o W/m^2 °C
1.					
2.					
3.					
4.					

14. Calculation of U_o [clean]

$1 / U_o = (1 / h_{io}) + (1 / h_o) + R_w$

where R_w is the metal wall resistance $= d_o * \ln [d_o / d_i] / 2 k_w$ [(m^2 °C) / W]

k_w = thermal conductivity of tube material

RESULT TABLE

Sl. No.	**h_i W/m^2 °C**	**h_O W/m^2 °C**	**$U_{O, expt}$ W/m^2 °C**	**$U_{O, Theo}$ W/m^2 °C**
1				
2				
3				
4				

EXPERIMENT NO. 3

VERTICAL CONDENSER

AIM: To determine the overall heat transfer co-efficient under thermal steady state conditions using vertical condenser and also the film coefficients.

APPARATUS: Vertical condenser setup, thermometers, stops watch, bucket, weighing balance.

THEORY

Condensers are heat transfer devices, used to liquefy vapour by removing their latent heat. Two types based on contact of vapor and coolant are shell and tube condensers (condensing vapour and coolant are separated by a tubular heat transfer surface) and contact condensers(both the vapor and liquid are usually water and are mixed physically and leave the condenser as a single stream). The condensation of vapors on the tube surfaces cooler than the condensation temperature of the vapor is commonly met in evaporation, distillation, and drying. The condensing vapor may consist of a single substance, a mixture of condensable and non condensable substances, or a mixture of two or more condensable vapors. Friction losses in a condenser are normally small, so that condensation is essentially a constant pressure. The condensing temperature of a single pure substance depends only on the pressure, and therefore the process of condensation of a pure substance is isothermal. Also the condensate is a pure liquid. Mixed vapors, condensing at constant pressure, condense over a temperature range and yield a condensate of variable composition until the entire vapor stream is condensed, when the composition of the condensate equals that of the original uncondensed vapor. Dropwise and filmtype are the two types based on condensation of vapor on a cold surface. In filmtype, which is more common, the liquid condensate forms a film, or a continuous layer of liquid that flows over the surface of the tube under the action of gravity. It is the layer of liquid interposed between the vapor and the wall of the tube that provides the resistance to heat flow. Heat transfer coefficient is comparatively less than that from dropwise type.

SPECIFICATIONS OF THE VERTICAL CONDENSER

Inside diameter of inner pipe = 19 mm

- Outside diameter of inner pipe = 22 mm
- Inside diameter of outer pipe = 60 mm
- Length of the condenser = 1500 m m
- Inner diameter of condensate tank = 147 mm

Inner and outer pipe is made of stainless steel; thermal conductivity is 54 w/m k.

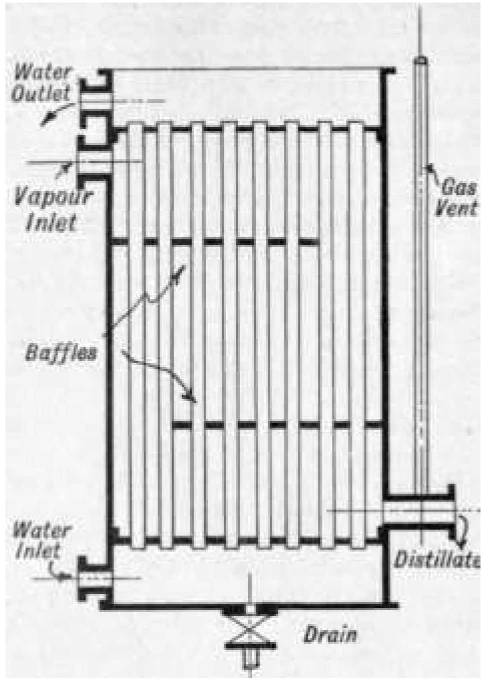


Figure3. Schematic sketch of Vertical Condenser

PROCEDURE

1. Check the boiler for the required pressure of steam (2.5 kg/cm^2).
2. Open the cold water inlet valve and set a constant flow rate in the pipe. Water in the condensate receiver is removed.
3. Initially when steam is allowed to flow through the annulus of the condenser, open the steam vent valve in the condenser is kept completely so that air present is replaced. The vent valve is slightly kept open throughout the experiment.
4. Adjust the steam inlet valve to about 0.5 kg/cm^2 (gauge) pressure and maintain constant throughout the experiment.
5. Allow the system to attain thermal equilibrium (or steady state indicated by constant cold water outlet temperature).
6. Note down the mass flow rate of condensate by calculating the time taken for 1cm rise in condensate level in the condensate receiver.
7. Measure the cold water flow rate using a bucket and a stopwatch. Record the inlet and outlet temperatures of cold water (t_{in}, t_{out}) and the steam pressure.
8. Repeat the experiment for different cold water flow rates.

Properties of steam:

Properties of steam are evaluated at absolute pressure using steam tables.

Absolute pressure = Gauge pressure + Atmospheric pressure.

Atmospheric pressure=1.03 kg/cm^2

1kg/cm^2=0.98 bar

Note: since steam is produced using pure water, steam temperature = condensate temperature, Condensation (steam saturation or vaporization) temperature is referred against absolute pressure.

λ_s= Latent heat of condensation at steam condensation temperature, J/kg

OBSERVATION TABLE

Steam pressure = 0.5 kg/cm^2 (gauge)

Sl. No.	**Rise in condensate level, m**	**Time taken, s**	**COLD WATER**				
			t_{in} °C	**t_{out} °C**	**Water collected kg**	**Time s**	**Mass flow rate kg/s [m_c]**
1.							
2.							
3.							
4.							

CALCULATIONS

1. **Heat load calculation**

Assumption: Heat exchanged between the fluids is given by

Heat lost by steam condensation, Q_h is equal to Heat gained by cold water, Q_c

$$Q_h = m_h * \lambda_s \qquad [W]$$

m_h= Mass flow rate of condensate = $\rho_c * l_c * a_c \qquad [kg/s]$

= density of condensate (water) at saturation temperature * height of rise in the condensate receiver per unit time * c/s area of condensate receiver

Verification : Whether the heat exchange between the cold and hot fluids is same or not.

$$Q_c = m_c * C_{pc} * (\otimes t)_c \qquad [\,W\,]$$

Where c_{pc} = specific heat of cold fluid at mean temperature, ***J/kg°C***

$$t_{c\ mean} = \{ t_{in} + t_{out} \} / 2$$

$$(\otimes t)_c = t_{out} - t_{in}$$

Check whether $Q_H = Q_c$ or not LMTD calculation for condenser

$$\otimes T_1 = T_{sat} - t_{in}$$

$$\otimes T_2 = T_{sat} - t_{out}$$

$$LMTD = (\otimes T_1 - \otimes T_2) / \ln \{\otimes T_1 / \otimes T_2\} \qquad [°C]$$

(For co-current température profile)

3. Heat transfer area, $A_i = [\, \angle * d_i * L \,]$ $[m^2]$

Where d_i is inner diameter of the inner tube

L is length of the tube.

4. Calculation of overall heat transfer coefficient based on inside area of inner tube

$$U_i = Q_h / \{ A_i * LMTD \} \qquad [\, W/ m^2\ °C \,]$$

<u>TUBE SIDE CALCULATIONS</u>

<u>CALCULATION OF INSIDE FILM COEFFICIENT, h_i (COLD WATER SIDE)</u>

5. Volumetric flow rate, F = mass flow rate / density at mean temperature

$$= m_c / \rangle \qquad (m^3 / s)$$

6. Flow area, a_p $= (\square * d_i^2) / 4$ $[m^2]$

where d_i is the inner diameter of the inner pipe

7. Velocity in the pipe , v $= F / a_p$ $[m/s]$

8. Reynolds number, N_{Re} $= [\, d_i * v * \rangle \,] /$ ⌠

where ρ = density at mean temperature

μ = viscosity at mean temperature

9. Inside HT coefficient, h_i calculation

Check for flow nature

If N_{Re} <2100 [flow is laminar] and also for Greatz number greater than 20

Use $N_{Nu} = h_i d_i/k = 2.0 [N_{Gz}]^{1/3}$

Where N_{Gz}= *Greatz number* $= (4 * m_c * c_{pc}) / (\pi * k * L)$

k is thermal conductivity of pipe side fluid at mean temperature.

If $N_{Re} > 6000$ [flow is turbulent]

Use Dittus-Boelter equation

$$N_{Nu} = h_i d_i/k = 0.023 * [N_{Re}]^{0.8} [N_{Pr}]^n$$

Where N_{Pr} = Prandtl number

$= C_p \mu/k$ (Can also be obtained from properties table)

n = 0.4 since the fluid is heated

And $h_i = [N_{Nu} * k] / d_i$ $[W/m^2\ ^\circ C]$

If $2100 < N_{Re} < 6000$ [flow is in transition region]

Note: The calculation method is based on graphs of Sieder-Tate equation ,

$$N_{Nu} = h\ d_i/k = 0.023 * [N_{Re}]^{0.8} [N_{Pr}]^{0.4} [\mu/\mu_w]^{0.14}$$

Use j_H Vs N_{Re} chart.

First find L/d_i: $L/d_i = 1.5/19 = 78.95$

Read j_H to the nearest L/d_i curve corresponding to the N_{Re}, then using j_H value compute h_i.

TABLE OF PROPERTIES OF COLD WATER AT MEAN TEMPERATURE $T_{C\ MEAN}$

Sl. No.	$t_{c\ mean}$ °C	〉 kg/m^3	⌠ kg/m s	c_p J/kg °C	k W/m °C	N_{Pr}
1						
2						
3						
4						

SUMMARY OF PIPE SIDE CALCULATIONS

Sl .NO.	m_c kg/s	N_{Re}	N_{Pr}	N_{Nu}	h_I $W/m^2\ {}^oC$
1					
2					
3					
4					

ANNULUS SIDE CALCULATIONS

Calculation of outside film coefficient , h_o [annulus or steam side]

10. Wall temperature $T_w . = T_s + t_{c,\ mean}$

Where T_s = Saturation temperature of steam at absolute pressure

$t_{c,\ mean}$ = Mean temperature of cold fluid

13. Film temperature $T_{f\ =}\ T_s - \ (3/4)\ (T_s - T_w)$

14. $\otimes T = \otimes T_{local} = T_s - T_w$

TABLE OF CONDENSED STEAM AT FILM TEMPERATURE

Sl. NO.	T_w °C	T_f °C	ρ_f kg/m^3	μ_f kg/m s	k_f W/m K	ΔT °C	h_o W/m^2 °C
1							
2							
3							
4							

15. $\boldsymbol{h_o = 0.943\, [\{ K_f^3\, \rho_f^2\, g\, \lambda_s\} / \{\mu_f\, L\, \Delta T\}]^{1/4}}$

where, the properties are of condensate are evaluated at T_f

16. Calculation of U_i [clean co-efficient]

$$\boldsymbol{1 / U_i = (1 / h_i) + (d_i / d_o\ h_o) + R_w}$$

Where R_w is metal wall resistance

$$\boldsymbol{R_w = (d_o - d_i)\ d_i / k_w * \ln(d_o / d_i)} \qquad [\ (m^2\ °C) / W\ ]$$

k_w = thermal conductivity of pipe material

17. Wilson plot ; A graph of (I/U_i) versus ($1/G^{0.8}$) gives straight relationship with the assumptions that the terms within a and b are almost constant.

Reference equation: Siedar Tate equation

$$\boldsymbol{I/Ui = (1/a)\ 1/G^{0.8} + b}$$

$$= [1/\{ (0.023 d_i/k_{cold\ fluid,\ mean}) \quad (d_i//\lceil_{cold\ fluid,\ mean})^{0.8} \quad (N_{pr}))^{0.33} \}] \ * \ (1/G^{0.8}) \ +$$

$$[\{(d_o - d_i) * d_i / k_w \ \ln(d_o / d_i) \} + \{d_i / (h_o d_o)\}]$$

$$a = \{ (0.023 d_i/k_{cold\ fluid,\ mean}) \quad (d_i//\lceil_{cold\ fluid,\ mean})^{0.8} \quad (N_{pr}))^{0.33} \}]$$

$$b = [\{(d_o - d_i) * d_i / k_w \ \ln(d_o / d_i) \} + \{d_i / (h_o d_o)\}]$$

Slope=b and intercept=1/a.

TABLE FOR WILSON PLOT

Sl. No.	$1/U_{i\ EXPT}$	$1/G^{0.8}$
1		
2		
3		
4		

EXPERIMENT NO. 4

EMISSIVITY

AIM: To determine the radiation heat transfer coefficient on a square or cylindrical metal block.

APPARATUS: Experimental setup, thermometer, stopwatch etc.

THEORY

Radiation is the energy streaming through space at the speed of light may originate in many ways. All substances at temperatures above absolute zero emit radiation that is independent of external agencies like in electron bombardment, electric discharge, or radiation of definite wavelengths. Radiation that is result of temperature only is called thermal radiation.

The fraction of radiation falling on a body that is reflected is called the reflectivity. The fraction of radiation falling on a body that is absorbed is called the absorptivity. The fraction of radiation falling on a body that is transmitted is called the transmissivity. The sum of these fractions must be unity. Radiation as such is not heat, and when transformed to heat on absorption, it is no longer radiation. However, reflected or transmitted radiation falling on other absorptive bodies will be converted to heat after several reflections. The maximum possible absorptivity is unity, attained only if the body absorbs all radiation incidents upon it and reflects or transmits none. A body that absorbs all incident radiation is called a black body.

Emissivity usually increases with temperature. Emissivities of polished metals are low, in the range of 0.03 to 0.08. Those of most oxidized metals range from 0.6 to 0.85; those of nonmetals such as refractories, papers, and building materials, from 0.65 to 0.95; and those of paints, other than aluminium paint, from 0.8 to 0.96. No actual substance is a blackbody, although some materials such as carbon black approach blackness.

Stefan–Boltzmann law states that the total emissive power of blackbody is propotional to the fourth power of the absolute temperature.

Kirchhoff's law states that, at temperature equilibrium, the ratio of the total radiating power of any substance to its absorptivity depends only upon the temperature of the substance.

SPECIFICATIONS OF THE CUBE/CYLINDER

- Length of the cube /cylinder = 5 cm/7.5 cm
- Diameter of cylinder= 5cm
- Mass of cube /cylinder = 0.975 kg/1.2 kg
- Block material = Mild steel (thermal conductivity is 72 W/m ºk)
- Specific heat capacity = 465 J/ kg K

PROCEDURE

1. Heat the given block of sample to about 250ºC.
2. Remove the block from heater and allow to cool in air and simultaneously start the stop watch
3. Record the time for every 10ºC drop in temperature till it is reduced to 70ºC.
4. Note the room temperature.
5. Plot a graph of temperature in Degree K versus time on an ordinary graph sheet. Draw tangents at several points (about eight) to the curve drawn. The values of the temperatures on the Y-axis where the tangent intercepts and the point at which the tangent is drawn are recorded. The time at which the tangent is drawn is also noted.

OBSERVATION TABLE

Table 1:

Sl. No.	**T** K	**Time** s
1		
2		
3		

TABLE 2: From Graph

Sl. No.	**$T_{tangent}$** K	**$T_{intercept}$** K	**Time** s	**Slope** $dT/d\theta$ (K /s)
1				
2				

CALCULATIONS

1. From the plot of temperature versus time

   Slope = $dT/d\theta$ at θ_1

   $$Q_{\theta 1} = m\ c_p\ (dT/d\theta \text{ at } \theta_1)$$

   Where $Q_{\theta 1}$ = Heat conducted through the cube, **[W]**

2. $$N_{Gr} = l^3 \beta\ g\ \Delta T / \nu^2$$

Where N_{Gr} = Grashoff number

l = length of the cube, m

T_s= Temperature at which tangent is drawn, **[K]**

$T_{film} = (T_s + T_{air})/2$, ***[K]***

β = Coefficient of thermal expansion = $1/T_{film}$, in k

g = 9.81 m^2/s

ν = Kinematic viscosity, in m^2/s at T_{film}

$\Delta T = T_s - T_{air}$, ***[K]***

PROPERTIES OF AIR AT FILM TEMPERATURE

Sl. No.	**T_f K**	**ν_f m^2/s**	**N_{Pr}**	**k_f W/m K**
1				
2				
3				

4. Evaluation of ($N_{Gr} * N_{Pr}$)

a) For $(10^4) < (N_{Gr} * N_{Pr}) < (10^9)$

$$N_{Nu} = 0.59\ (N_{Gr} * N_{Pr})^{1/4}$$

b) For $(10^9) < (N_{Gr} * N_{Pr}) < (10^{12})$

$$N_{Nu} = 0.13\ (N_{Gr} * N_{Pr})^{1/3}$$

N_{Pr} = Prandtl number

$= C_p \mu / k$ (Can also be obtained from properties table)

N_{Nu} = Nusselt number = h_o l/ k

h_o =Air side convective heat transfer coefficient , ***[W/m² K]***

5. Nusselt number = $\boldsymbol{h_o\, l/ k}$

$\boldsymbol{h_o = N_{Nu}\, k / l}$

Where k = Thermal conductivity of air at T_{film} , ***[W/mK]***

6. $\boldsymbol{Q_c = h_o A_o \Delta T}$

Where Q_c = Heat lost by convection, ***[W]***

A_o = total surface area of cube, 6 l^2 , $\boldsymbol{m^2}$

$\Delta T = T_s - T_{air}$

7. $\boldsymbol{Q_r = Q_{\theta 1} - Q_c}$

Where Q_r = Heat lost by radiation, ***[W]***

8. Emissivity(□) is obtained from Q_r = ʃ□A ($T_s^4 - T_{air}^4$)

Where ʃ = Steffan- Boltmann constant = 5.671 * 10^{-8} ***[W/m² K⁴]***

□= Q_r / ʃA_o ($T_s^4 - T_{air}^4$)

9. Radiation heat transfer co-efficient h_{rad} is obtained from

$\boldsymbol{Q_r = h_{rad} A_o \ (T_s - T_{air})}$ ***[W]***

CALCULATION TABLE

Sl. No.	$T_{intercept}$ K	$Q_{\theta 1}$ W	T_f K	β 1/ K	ΔT K	N_{Gr}	N_{Pr}	$N_{Gr} * N_{Pr}$	N_{Nu}
1.									
2.									
3.									

RESULT TABLE

Sl. No.	h_o W/m² K	Q_c W	Q_r W	□	h_{rad} W/m² K
1.					
2.					
3.					

EXPERIMENT NO. 5

HELICAL COIL HEAT EXCHANGER

AIM: To determine the film coefficients and the overall heat transfer coefficient under thermal steady state conditions for various cold fluid flow rates using helical coil heat exchanger.

APPARATUS: Experimental set up of helical coil heat exchanger, thermometers, measuring jar, stop watch etc.

THEORY

A simplest and cheapest form of heat transfer surface for installation inside a vessel is a helical coil. The pitch and diameter of coil can be made to suit the application and area required. The diameter of the pipe used for the coil is typically equal to $d_v/30$, where d_v is the vessel diameter. The coil pitch is usually around twice the pipe diameter. Small coils can be self supporting, but for large coils some form of supporting structure will be necessary. Single or multiple turn coils are used.

A simple jacketed kettle is very commonly used in the chemical industry as a reaction vessel. In many cases, such as in sulphonation reactions, heat has to be removed or added to the mixture in order to either control the rate of the reaction or to bring it to completion. The addition or removal of heat is conveniently arranged by passing steam or water to the jacket fitted to the outside of the vessel or through helical coils fitted inside. In either case, some form of agitator is used to obtain even distribution in the vessel. This may be of the anchor type for very thick mixes or a propeller or turbine if the contents are not too viscous.

SPECIFICATIONS OF THE HELICAL COIL HEAT EXCHANGER

- Inner diameter of tube = 9.7 mm
- Outer diameter of tube = 12.7 mm
- Number of coils = 05
- Vessel material = Stainless steel (k=54 W/m °c)
- Tube material = Copper(k =386 W/m °c)
- Diameter of coil = 200 mm
- Speed of agitator = 23.7 rps
- Diameter of the vessel = 300 mm
- Diameter of the paddle = 75 mm

PROCEDURE

1. Fill the water bath so that the helical coil is immersed completely in water.
2. Switch on the heater so as to attain a temperature of about 80°C.
3. Allow cold water to flow through the helical coil at minimum flow rate. System is allowed to attain thermal equilibrium (or steady state indicated by constant cold water outlet temperature).
4. Measure the volumetric flow rate of water through helical coil by collecting known volume of water and record the time taken for it.
5. Record the inlet and outlet temperatures of cooing fluid and the water bath temperature.
6. Repeat the experiment for various cold water flow rates maintaining water bath temperature constant.
7. Calculate the film heat transfer coefficients and overall co-efficient.

OBSERVATION TABLE

Sl. No	T_{bath} °C	$T_{cold, in}$ °C	$T_{cold, out}$ °C	Volume of water collected cc	Time s	Volumetric flow rate cc/s
1.						
2.						
3.						

CALCULATIONS

1. Heat load calculation, $Q = m_c * c_{pc} * (\otimes t)_c$ [W]

Where m_c = mass flow rate of water through coil, kg/s

$= F * \rho_{ave}$

F= volumetric flow rate, m^3/s

ρ_{ave} = density at average cooling water temperature

C_{pc} = specific heat of cold fluid at mean temperature

$t_{c\ mean} = \{ t_{in} + t_{out} \} / 2$

$(\otimes t)_c = t_{out} - t_{in}$

2. LMTD calculation

$\otimes T_1 = T_h - t_{in}$

$\otimes T_2 = T_h - t_{out}$

$$LMTD = (\otimes T_1 - \otimes T_2) / \ln \{\otimes T_1 / \otimes T_2\} \quad [°C]$$

3. Heat transfer area, $A_o = [\angle * d_o * L]$ [m^2]

Where d_o is outer diameter of the tube

L is straight length of the pipe = $\angle n\, d_{coil}$

N is the number of turns

3. Calculation of $U_{o,\ expt}$

$$U_{o,\ expt} = Q / A_o * LMTD$$

OVERALL HEAT TRANSFER COEFFICIENT

Mean temperature of cold water $t_{c\ mean} = \{ t_{in} + t_{out} \} / 2$

Sl No.	$t_{c\ mean}$ °C	C_p kJ/kg °C	m_c kg/s	Q W	LMTD	U_o W/m^2 °C
1.						
2.						
3.						

4. Flow area, $a_p = \pi * d_i^2 / 4$

5. Velocity of water through tube= **F/$\mathbf{a_p}$**

6. Reynolds number, $N_{Re} = [\, d_i * v * \rho \,] / \mu$

where ρ = density at mean temperature

μ = viscosity at mean temperature

7. Inside film co-efficient , h_i calculation

Check for flow nature

If N_{Re}<2100 [flow is laminar] and also for Greatz number greater than 20

Use $$N_{Nu} = h_i d_i/k = 2.0 \, [\, N_{Gz} \,]^{1/3}$$

Where N_{Gz}= Greatz number = $(4 * m_c * c_{pc}) \; / \, (\pi * k * L)$

k is thermal conductivity of pipe side fluid at mean temperature.

If N_{Re} > 6000 [flow is turbulent]

Use Dittus-Boelter equation

$$N_{Nu} = h_i d_i/k = 0.023 * [N_{Re}]^{0.8} [N_{Pr}]^n$$

Where N_{Pr} = Prandtl number

= $C_p \, \mu / k$ (Can also be obtained from properties table)

n = 0.4 since the fluid is heated

And $h_i = [\, N_{Nu} * k \,] / d_i$ [$W/m^2\ °C$]

If 2100 < N_{Re} < 6000 [flow is in transition region]

(Note: No simple equation applies here. The calculation method is based on graphs of Sieder-Tate equation, $N_{Nu} = h \, d_i / k = 0.023 * [N_{Re}]^{0.8} [N_{Pr}]^{0.4} \, [\, \mu / \mu_w]^{0.14}$ after modifications on a common plot of the Colburn j_H factor versus N_{Re}.)

Use j_H Vs N_{Re} chart.

First find L/d_i: $L/d_i = 1.5/19 = 78.95$

Read j_H to the nearest L/d_i curve corresponding to the N_{Re}, then using j_H value compute h_i.

8. h_i based on outside area of tube $h_{io}= h_i * d_i/ d_o$
9. h_{io} when liquid flows through coil, h_{ic}

$$h_{ic}= h_{io}\{1+3.5(d_i/ d_c)$$

where dc is diameter of coil

INSIDE FILM CO-EFFICIENT [h_{ic}] TABLE

Sl. No.	$t_{c,mean}$ ˚C	〉 kg/m³	⌠ kg/m s	N_{Re}	k W/m°K	N_{Pr}	h_i W/m²°C	h_{io} W/m²°C	h_{ic} W/m² °C
1.									
2.									
3.									

10. Hot water side co-efficient, h_o

$$h_o= 0.87\ k/ d_v \quad (N_{Re})^{0.66} \quad (N_{Pr})^{0.33}$$

Reynolds number, $N_{Re} = \{d_p^2\ N\ 〉/ ⌠\}$

where 〉 = density at bath temperature, kg/m³

⌠ = viscosity at bath temperature, Kg/m-s

d_p= diameter of paddle, m

N= Number of revolutions per second

OUTSIDE FILM CO-EFFICIENT TABLE (ho)

(Hot water properties at T_{bath})

Sl. No.	T_{bath} °C	〉 kg/m³	⌠ kg/ms	C_p J/kg °C	k W/mK	N_{Pr}	h_{io} W/m²°C
1.							
2.							
3.							

11. Calculation of U_i [clean co-efficient]

$$1 / U_{o,clean} = (1 / h_{ic}) + (1/ h_{o}) + R_w$$

Where R_w is tube metal wall resistance

$$R_w = d_o * \ln [d_o / d_i] / 2 k_w \qquad [(m^2 \ ^\circ C) /W]$$

k_w = thermal conductivity of tube material

RESULTS TABLE

Sl. No.	**$U_{o, expt,}$ W/m²°C**	**h_{ic} W/m²°C**	**h_o W/m²°C**	**$U_{o, clean,}$ W/m²°C**
1.				
2.				
3.				

EXPERIMENT NO: 6

TRANSIENT HEAT CONDUCTION
(CONSTANT TEMPERATURE)

AIM: To study unsteady state heat conduction when one of the ends of a rod is exposed to hot atmosphere.

To obtain the temperature profile for conduction through a mild steel rod by a constant temperature source and to compare experimental temperature profile with that of theoretically predicted one.

APPARATUS: Experimental setup, stopwatch, thermometers.

THEORY

The term transient or unsteady state designates a phenomenon which is rime dependent. Conduction of heat refers to the transient conditions where the heat flow and the temperature distribution at any point ot the system vary continuously with time. The temperature and rate of heat conduction are then undoubtedly dependent both the time and space co-ordinates {t = f(x, y, z, |)}. Transient conduction occurs in heating and cooling of metal billets, cooling of IC engines cylinder, burning of bricks and vulcanization of rubber, and during starting and stopping of various heat exchange units in power installation. So the study of transient conduction situation is an important component of conduction studies.

Temperature variation during unsteady state may follow a periodic or a non-periodic pattern. In the first, the temperature change is in repeated cycles and the conditions get repeated after some fixed interval of time. Examples are temperature variation of a building during a full day period of 24-hours and heat processing of regenerators whose packings are alternately heated by fuel gases and cooled by air. In the second type, the temperature changes as some nonlinear function of time. Examples are heating of an ingot in a furnace and cooling of bars, blanks and metal billets in steel works.

Undoubtedly the time dependent effects occur in many industrial heating, cooling and drying processes. An increase or decrease in temperature at any instant continues until steady temperature distribution is attained. For example during quenching of steel, there occurs a gradual decrease in the temperature of hot steel rod and the quenching medium attains the same temperature.

Consider a solid cylinder. The body is initially at temperature equal to surrounding temperature, including the surface at x=0. The end temperature at x=0 is instantaneously changed and held at T_s for all times greater than | = 0. Thus the boundary conditions are T(x,0)=.T_i

$T(0, \tau)$ for $\tau > 0$

$T(\infty, \tau) = T_i$ for $\tau > 0$

The solution of the differential equation for transient heat conduction

$D^2T/dx^2 = (1/\alpha)\ dT/d\tau$

With these boundary conditions would give the following solution for temperature distribution at any time τ at a distance x from the surface.

$[T(x, \tau) - T_s] / [T_i - T_s] = \text{erf}\ [x/2\ \square(\langle\ \tau)]$

Where erf $[x/2\ \square(\langle\ \tau)]$ = Gaussian error function

SPECIFICATIONS OF THE SETUP

Material of rod = Mild steel

Diameter of rod = 25.4 mm

Specific capacity of rod material (C_p) = 465 J/kg K

Density of rod ()) = 7833 kg/m^3

Thermal conductivity of rod (k) = 72 W/m K

PROCEDURE

1. Set the steam pressure in the boiler about 0.5 kg/cm^2 is sufficient as per the requirement.
2. Make sure the thermometers are placed in all the seven provisions made.
3. Record the initial temperatures at different locations of the mild steel rod. And allow water to circulate at the cold end (which is not in direct contact with the heat source) of the rod.
4. Pass the steam at the other end and start recording the temperature at a time interval of 5 minutes. (Steam pressure is maintained at a constant value (say 0.2 kg/cm^2).

Note : Since observations are to be made during unsteady state, temperatures are noted till the steady state is attained. Steady state is indicated by constant temperature at each location with respect to time.

5. Draw the temperature profiles (temperature versus time plots) for three time intervals.
6. Compare the Experimental temperature profile with that of theoretically predicted one as a function of time.

OBSERVATION TABLE

Steam pressure = kg/cm^2

Sl. No.	**T_1, °C** **X_1=5 cm**	**T_2, °C** **X_2=10 cm**	**T_3 , °C** **X_3=15 cm**	**T_4 , °C** **X_4 =20 cm**	**T_5 , °C** **X_5 =25 cm**	**T_6 , °C** **X_6=30 cm**	**T_7, °C** **X_7=35 cm**
1.							
2.							
3.							
4.							
5.							

CALCULATIONS

1. $\boldsymbol{Q_o = -kA(T_i-T_s)/\square(\angle\langle /)}$

Where Q_o =Heat at source due to steam at T_s, W

T_i = Initial temperature at zero time (before steam is allowed to flow), °C

T_s = Steam saturation temperature, °C

Note: Properties of steam are evaluated at absolute pressure using steam tables.

Absolute pressure = Gauge pressure + Atmospheric pressure.

Atmospheric pressure=1.03 kg/cm^2

1 kg/cm^2=0.98 bar

Note: since steam is produced using pure water, steam temperature = condensate temperature

Condensation (steam saturation or vaporization) temperature is referred against absolute pressure.

2. $\boldsymbol{Q_1 = Q_o\, e^{-x\ /4\langle /}}$

Where Q_1= Heat conducted through rod through a distance x and after time | ,W

x = Distance from the hot end of the rod, m

⟨ = Thermal diffusivity of rod material, m^2/s

= **k/⟩ C_p**

k= Thermal conductivity of rod material, W/m K

ρ = Density of rod material, kg/m^3

C_p = Specific heat capacity of rod material, J/kg K

τ = Time, s

3. $z = x/2\sqrt{(\alpha\tau)}$

4. Obtain erf(z) value from table of erf(z) against z

   Note: Notation used in the error function table is x instead of z

5. Calculate $T_{(x,\tau)}$ using

   $erf(z) = T_{(x,\tau)} - T_s/(T_i - T_s)$

   $T_{(x,\tau)} = T_s + (T_i - T_s)\ erf(z)$

   Where

   $T_{(x,\tau)}$ = Temperature (Theoretically obtained) at distance x and after time, τ

RESULT TABLE

Q_o = W, τ = seconds

Sl. No.	**x_1 = cm**	Q1 **W**	**z**	**erf(z)**	**T_{The}, °C**	**Texpt, °C**
1.						
2.						
3.						
4.						
5.						

EXPERIMENT NO. 7

BARE TUBE CONDENSER

AIM : To determine air side Heat Transfer coefficient on bare tube condenser by natural convection under steady state conditions

APPARATUS: Experimental setup, stopwatch.

THEORY

On the basis of method adopted for fluid motion, convective HT can be classified as forced convection (FC) and free or natural convection (NC). In FC, fluid is forced to flow along the solid surface by means of external energy provided by pump, fan or a compressor. Examples are cooling of internal combustion engines, air conditioning installations, nuclear reactors, heat exchangers, condensers and other number of process equipments. In NC, the fluid motion is caused due to density difference produced by temperature gradients. A stagnant layer of fluid in contact with the hot body, receives energy by conduction. Therefore, internal energy of fluid particles increases. Due to temperature rise, the lighter fluid particles move upward to a region of low temperature where they mix with and transfer a fraction of their energy to the cold particles. Simultaneously, the cool heavier particles move downward to fill the space vacated by the warm fluid particles. Thus a circulation pattern sets up in the fluid. It is called as convection currents. The currents are set up due to gravity and are responsible for free convection.

In FC, the fluid velocity is much higher than that in NC. Therefore, the heat exchange rate is more. The intensity of mixing of fluid particles is generally less in NC. Therefore HT coefficients are very small. But many devices depend largely on NC for cooling purpose. Some examples are cooling of transmission lines, transformers, rectifiers, electrically heated wires such as heating elements of an electric furnace, electronic instruments. NC is also responsible for heat losses from pipes carrying steam or other hot fluids which are either not insulated or improperly insulated. Grashoff, Prandtl and Reynolds numbers are the important dimensionless groups in NC. Grashoff number is a ratio of product of buoyancy force and inertial force to the square of viscous force in NC boundary layer HT.

SPECIFICATIONS OF THE BARE TUBE CONDENSER

- Inside diameter of tube, d_i = 71 mm
- Outside diameter of tube, d_o = 75 mm

- Inner diameter of condensate receiver, d_c = 5.41 cm
- Length of the bare tube, L = 1

Material of the bare tube = stainless steel, its thermal conductivity (k) is 54 W/m K

PROCEDURE

1. Boiler is checked for the required pressure of steam (1.0 kg/cm^2 is sufficient).
2. Drain the water in the condensate receiver to bring down the water level and allow steam to flow through the condenser. Make sure the steam vent valve in the condenser is kept completely open so that air present is replaced. Keep the vent valve slightly open throughout the experiment.
3. Adjust the steam inlet valve is adjusted to about 0.015 kg/cm^2 (gauge) pressure and maintain constant throughout the run of the experiment.
4. Air surrounding the bare tube condenses the steam inside the tube. Allow the steam to attain thermal equilibrium (or steady state indicated by constant rate of rise in condensate level).
5. Find the mass flow rate of condensate by noting the time taken for 1cm rise in condensate level in the condensate receiver.
6. Repeat the experiment for different steam pressure (increment of 0.005 kg/cm^2).

OBSERVATION TABLE

Room temperature =_______°C

Sl. No.	p kg/cm^2	Time t_c, s	Rise in condensate, l_c (cm) Trial no. 1	Rise in condensate level, l_c (cm) Trial no. 2
1				
2				
3				
4				

CALCULATIONS

1. **Heat load calculation**

   Assumption: Heat exchanged between the fluids(steam and air) = Heat lost by steam by condensation, Q

$$Q = m * \lambda_s \qquad [W]$$

Where m= Mass flow rate of condensate = $\rho_c * l_c * A_c / t_c \qquad [kg/s]$

ρ_c = Density of condensate at saturation temperature, kg/m^3

Note: Density can be obtained as inverse of specific volume from steam table is another source.

l_c = Height of rise in the condensate receiver, ***m***

A_c = Cross area of condensate receiver, m^2

= $\pi d_c^2/4$

d_c = Inner diameter of the condensate receiver, ***m***

λ_s= Latent heat of condensation, ***J/kg***

Note: λ_s is taken corresponding to T_s from steam table.

2. **Heat transfer area ,**

$$A_o = [\pi * d_o * L] \qquad [m^2]$$

Where d_o is outer diameter of the bare tube

L is length of bare tube.

3. **Calculation of overall heat transfer coefficient**

$$U_o = Q / \{A_o * \Delta T\} \qquad [W/m^2\ ^\circ C]$$

Where $\Delta T = T_s - T_{room}$

T_s = Steam temperature corresponding to absolute pressure of steam, which must be taken from the steam tables

Absolute pressure = Gauge pressure + Atmospheric pressure.

Atmospheric pressure=1.03kg/cm^2

1kg/cm^2=0.98bar

Note: since steam is produced using pure water, steam temperature = condensate temperature

CALCULATIONS FOR AIRSIDE FILM COEFFICIENT, h_o

4. $N_{Gr} = L^3 \beta g \Delta T / \nu^2$

Note : N_{Gr} , Grashoff number refers to air flow conditions

Where L = length of the tube, m

β = Coefficient of thermal expansion = $1/T_{film}$, in K

$$T_{film} = (T_w + T_{room})/2$$

$$T_w = (T_s + T_{room})/2$$

$g = 9.81\ m^2/s$

ν = Kinematic viscosity of air at T_{film} obtained from air properties table, in m^2/s

5. **Prandtl number**

$$N_{Pr} = C_p \int / k$$

Note : All properties are obtained from air properties table.

N_{Pr} can also be directly obtained from air properties table.

6. **Evaluation of (N_{Gr} * N_{Pr})**

a) For $(10^4) < (N_{Gr} * N_{Pr}) < (10^9)$

$$N_{Nu} = 0.59\ (N_{Gr} * N_{Pr})^{1/4}$$

b) For $(10^9) < (N_{Gr} * N_{Pr}) < (10^{12})$

$$N_{Nu} = 0.13\ (N_{Gr} * N_{Pr})^{1/3}$$

N_{Nu} = Nusselt number = $h_o L/k$

7. **Air side HT coefficient**

$$h_o = N_{Nu}\, k / l$$

Where $k = k_{air}$ = Thermal conductivity of air at T_{film}, W/m K

TABLE OF PROPERTIES OF AIR AT FILM TEMPERATURE & h_o

Sl. NO.	T_{film} K	β /K	ν_f m^2/s	Cp J/kg °C	$\int$ kg/m s	k W/m K	ho $W/m^2\ K$
1							

2							
3							
4							

STEAM SIDE CALCULATIONS

Check for flow nature

If $N_{Re} < 1200$ [flow is laminar]

Note: Reynolds number refers to flow conditions of condensate as continuous film on inner surface of tube.

- Ripples or wave starts forming when $N_{Re} > 30$

For $N_{Re} > 1800$, becomes turbulent

In between this range, it is called as transition range.

Reynolds number, $N_{Re} = 4l' / ⌠_f$

where l' = Condensate loading, kg/ $m\ s = m_c / \pi\, d_i$

d_i = Inner diameter of bare tube, m

⌠ = viscosity of condensate film at film temperature, kg /m s

8. **Steam side HT coefficient, h_i**

Use $h_i = 0.943\ [\ \{k_f^3\ ⎠_f^2\ g\ ⌊\}/\{⌠_f\ L\ \otimes T\}\ ]^{1/4}$

Properties are evaluated at T_f

$$T_f = T_w - (3/4)\ \otimes T$$

Where T_w = Wall temperature

$$\otimes T = T_s - T_w$$

$$T_w = (T_s + T_{room})$$

k_f = Thermal conductivity of condensate film at T_f, W/m °C

ρ_f = Density of condensate film at T_f, kg/m^3

λ = Latent heat of condensation at T_s in J/kg

μ_f = Viscosity of condensate film at T_f, kg/m s

TABLE FOR STEAM SIDE HT COEFFICIENT

Sl. No.	N_{Re}	T_f °C	k_f W/m °C	ρ_f kg/m^3	λ J/kg	μ_f kg/m s	h_i W/m^2 K
1.							
2.							
3.							
4.							

RESULTS TABLE

Sl. No.	Q W	U_o W/m^2 K	h_i W/m^2 K	h_o W/m^2 K
1.				
2.				
3.				
4.				

EXPERIMENT NO. 8

FIN TUBE CONDENSER

AIM: To estimate airside film heat transfer coefficient by natural convection under steady state conditions and to find fin effectiveness and fin efficiency in a fin tube heat exchanger.

APPARATUS: Experimental setup, stopwatch etc.

THEORY

The use of fins (extended surfaces) on the outside of a heat exchanger pipe wall to give relatively high heat transfer coefficients is common. Examples are air cooled internal combustion engines of motor cycles, scooters, aero-engines, air cooled heat exchangers, refrigeration, cryogenic processes, electrical equipments, transformers etc.

In longitudinal type of fins, the fins are spaced around the tube wall and the direction of gas flow is parallel to the axis of the tube. In radial (transverse or circular) type of fins, the gas flows normal to the tubes. The performance of fins is evaluated in terms of effectiveness and efficiency. By using fin, the convective resistance decreases while the conductive resistance increases. A fin of low thermal conductivity does not serve the purpose of improvement in heat transfer. Similarly use of fin in system, where heat transfer coefficient is high may actually decrease the rate of heat transfer.

SPECIFICATIONS OF THE BARE TUBE CONDENSER

- Inside diameter of tube, d_i = 71 mm
- Outside diameter of tube, d_o = 75 mm
- Inner diameter of condensate tank, d_c = 5.41 cm
- Length of the bare tube, L = 1 m
- Number of fins,n = 12
- Width of fin =25.4 mm
- Thickness of fin = 2.5 mm

Material of the bare tube = Stainless steel

PROCEDURE

1. Drain the water in the condensate receiver.

2. Open the steam inlet valve and maintain pressure, say 0.01 kg/cm^2 (gauge) constant throughout this run.
3. Initially when steam is introduced, keep the vent valve completely open so that any condensate, if present is ejected out.
4. Keep the vent valve slightly open throughout. At steady state conditions, note the time taken for 1cm rise in the condensate level.
5. Repeat the above step to verify the first time of rise to find the mass flow rate of condensate.
6. Repeat the experiment for different pressure conditions. Conduct four runs.

OBSERVATION TABLE

Steam pressure = 0.01Kg / cm^2 (gauge)

Sl. No.	ρ kg/cm^2	Time s	Rise in condensate level (cm)
1.			
2.			
3.			
4.			

CALCULATIONS

8. **Heat load calculation**

$$Q = m_c * L_s \qquad [W]$$

Where L_s= Latent heat of condensation, J/kg

m_c = mass flow rate of condensate, kg/s

9. m_c = Density of condensate at T_{sat} * Cross sectional area of condensate receiver, A_c * Rise in condensate level/ Time

Where $A_c = \pi d_c^2/4$, d_c is inner diameter of the condensate receiver, m^2

Get density from steam table

10. Heat transfer surface area of bare tube, $A_b = [\,\angle * d_i * L\,]$ $[m^2]$

Where d_i is inner diameter of the bare tube

L is length of bare tube.

11. **Calculation of overall heat transfer coefficient based on inner area of bare tube**

$$U_i = Q \,/\, \{ A_i * \otimes T \} \qquad [\,W / m^2 \ ^\circ C\,]$$

Where $\otimes T = T_{sat} - T_{ambient}$

T_{sat} = Steam temperature corresponding to absolute pressure of steam, which must be taken from the steam tables

Get properties of steam from steam tables.

Absolute pressure = Gauge pressure + Atmospheric pressure.

CALCULATION OF AIRSIDE FILM COEFFICIENT, ho

4. $$N_{Gr} = L^3 \beta\, g\, \Delta T / \nu^2$$

Where N_{Gr} = Grashoff number

L = length of the tube, m

β = Coefficient of thermal expansion = 1/(T_{film}, in K)

g = 9.81 m^2/s

ν = Kinematic viscosity, in m^2/s at T_{film}

TABLE FOR PROPERTIES OF AIR AT FILM TEMPERATURE

Sl. No.	T_f °C	ν_f m^2/s	N_{Pr}	K_f W/m °C	β /K
1.					
2.					
3.					
4.					

5. Evaluation of (N_{Gr} * N_{Pr})

a) For (10^4) < (N_{Gr} * N_{Pr}) > (10^9)

$$N_{Nu} = 0.59\ (N_{Gr} * N_{Pr})^{1/4}$$

b) For (10^4) < (N_{Gr} * N_{Pr}) > (10^{12})

$$N_{Nu} = 0.13\ (N_{Gr} * N_{Pr})^{1/3}$$

N_{Pr} = Prandtl number

N_{Nu} = Nusselt number

6. $$h_{o,air} = N_{Nu} K_{air} / L$$

Where K_{air} = Thermal conductivity of air at T_{film} , [W/ m K]

9. **Steam side heat transfer coefficient, h_i**

Reynolds number, $N_{Re} = 4L/\mu_f$

where L = Condensate loading, = $m_c / \pi d_i$ [Kg/ ms]

μ = viscosity at film temperature

8. Check for flow nature

If $N_{Re} < 2100$ [flow is laminar]

Use $h_i = 1.13 [\{K_f^3 \rho^2 g \lambda\}/\{\mu_f L \Delta T\}]^{1/4}$

Properties to be evaluated at T_f

$$T_f = T_w - (3/4) \Delta T$$

Where T_w = Wall temperature

$$T = T_{sat} - T_w$$

$$T_w = (T_s + T_{amb})/2$$

CALCULATION TABLE FOR INSIDE HEAT TRANSFER COEFFICIENT

Sl. NO.	T_f °C	ν m^2/s	μ kg/m s	K W/m °C	N_{Pr}	N_{Re}	h_i W/m^2 °C
1.							
2.							
3.							
4.							

RESULTS TABLE

Sl. No	Q W	U_o W/m^2 °C	$h_{i,steam}$ W/m^2 °C	$h_{o, air}$ W/m^2 °C
1.				
2.				
3.				

EXPERIMENT NO. 9

PACKED BED HEAT EXCHANGER

AIM: To determine overall and individual heat transfer coefficients under thermal steady state conditions for different cold water flow rates in a packed bed heat exchanger,

APPARATUS: Experimental setup, stopwatch, bucket, thermometers, balance etc.

THEORY

The convective HT coefficient is a function of fluid flow, properties of fluids, surface conditions and thermal conditions. In convective HT the resistance to HT can be classified as resistance offered by i. hot fluid ii. cold fluid iii. Tube or pipe metal wall iv. scale formation on hot fluid side v. scale formation on cold fluid side. The first two resistances depend upon fluid velocity and viscous of the fluid. Fluid film thickness decreases with increase in fluid velocity. Viscous fluids induce thicker films. Metal wall resistance depends upon pipe or tube thickness and the thermal conductivity of the material. Its resistance is negligible if wall is thin and thermal conductivity is higher. Scale formation over the surface occurs due to dissolved solute in the fluid, build up of solid particles thickness over the surface increases the resistance.

Various dimensionless numbers used in free convection

Many catalytic reactions are carried out in multitubular reactors that are similar to shell and tube exchanger. The solid catalyst particles are packed in the tube and the reactant enter and leave through headers at he ends of the reactor. For an exothermic reaction, the heat of the reaction is removed by circulating a coolant or boiling a fluid on the shell side. For endothermic reaction, the energy used for the reaction is transferred from hot fluid in the shell to the catalyst particles in the tube. The limiting heat transfer coefficient is usually of the tube side and the tube size and mass flow **rates are often choosen to ensure a nearly** constant reaction temperature or to prevent the maximum catalyst temperature from exceeding a safe value.

SPECIFICATIONS OF THE SET UP

- Outer diameter of packed bed, do = 6 cm
- Inner diameter of inner pipe = 5 cm
- Packing material = Ceramic, berl saddles
- Diameter of the packing material = 1.27 cm
- Inner diameter of jacket = 11.25 cm
- Length of bed = 1200 mm
- Material of construction= Stainless steel

PROCEDURE

1. Take sufficient water in the water bath.
2. Switch on the heater and let the water reach the desired temperature.
3. Switch on the pump and set the hot water to the desired flow rate by adjusting the bypass valve. The hot water flows in the annular space between the inner pipe and the jacket.
4. Allow the equipment to attain thermal equilibrium (indicated by steady outlet temperature of hot water).
5. Allow the cold water to flow through the inner pipe paced with ½’ berl saddles.
6. When the steady state (temperature of both hot and cold water are steady) is reached, measure the inlet and outlet temperature of cold and hot water.
7. Collect cold water at the outlet for a known period of time.
8. Repeat the experiment for different flow rates of cold water.

OBSERVATION TABLE

Sl. No.	HOT WATER		COLD WATER				
	T_{in}(˚C)	T_{out} (˚C)	t_{in}(˚ C)	t_{out} (˚C)	Weight of water Collected (kg)	Time S	Mass flow rate kg/s [m_c]
1							
2							
3							
4							

CALCULATION

1. Heat load calculation, $Q = m_c * C_{pc} * (\otimes t)_c$ $[W]$

Where C_{pc} = specific heat of cold fluid at mean temperature, J/Kg^oC

$$(\otimes t)_c = t_{out} - t_{in}$$

10. LMTD calculation for packed bed for co current flow

$$\otimes T_1 = T_{in} - t_{in}$$

$$\otimes T_2 = T_{out} - t_{out}$$

$$LMTD = (\otimes T_1 - \otimes T_2) / \ln\{\otimes T_1 / \otimes T_2\} \qquad [°C]$$

3. Heat transfer area, $A_o = [\angle \; d_o \, l]$ $[m^2]$

Where d_o is outer diameter of the packed pipe

L is length of the pipe

4. Calculation of overall heat transfer coefficient based on outside area

$$U_o = Q / \{A_o * LMTD\} \qquad [W/m^2 \; °C]$$

CALCULATION OF INSIDE HEAT TRANSFER COEFFICIENT, $h_{i,\,cold}$

4. Mean temperature of cold water, $t_{c\,mean} = \{t_{in} + t_{out}\} / 2$

5. Mass flow rate, m_c = Weight of water collected/Time of collection $[kg/s]$

6. Flow area, $a_p = (\Box * d_i^2) / 4$ $[m^2]$

where d_i is the inner diameter of the packed pipe.

7. Velocity in the tube, $v = m_c \,/\, \rangle * a_p$ $[m/s]$

8. Reynolds number, $N_{Re,p} = [d_p * v * \rangle] / \lceil$

where $\rangle$ = density at mean temperature

$\lceil$ = viscosity at mean temperature

9. $h_{i,p}$ calculation

$$N_{Nu} = 0.813 * [N_{Re,p}]^{0.9} [\exp\{-6d_p/di\}]$$

& $$h_{i,p} = [N_{Nu} * k] / di \qquad [W/m^2 \; °C]$$

10. Calculation of h_{io}, (inside H. T. coefficient based on outside area of tube)

$$h_{io,p} = h_{i,p} * [d_i / d_o]$$

SUMMARY OF TUBE SIDE CALCULATIONS

Sl. NO.	$t_{c\ mean}$ °C	〉 kg/m^3	© m^2/s	⌠ kg/m-s	C_p kJ/kg°C	K W/m K	N_{Re}	$h_{i,p}$ W/m^2°C	$h_{io,p}$ W/m^2 °C
1									
2									
3									
4									

ANNULUS CALCULATIONS

Calculation of outside film coefficient, h_o [annulus or hot water side]

11. Annulus side mass flow rate, $m_h = Q / C_{ph} * (\otimes T)_h$ [kg / s]

where C_{ph} is the specific heat of the hot fluid at mean temperature

Mean temperature of hot water $T_{h\ mean} = \{ T_{in} + T_{out} \} / 2$

$$(\otimes T)_h = T_{in} - T_{out}$$

12. Flow area, $A_a = \angle (D_j^2 - d_o^2) / 4$ [m^2]

where D_j = inner diameter of jacket, m

d_o = outer diameter of inner pipe., m

13. Equivalent diameter, $D_e = [D_j - d_o]$ [m]

14. Mass velocity, $G_a = m_h / A_a$ [$kg / s\ m^2$]

15. Reynold's number, $N_{Re} = [D_e * G_a] / ⌠$

where C_p , ⌠, k are properties of hot fluid at mean temperature.

16.Computation of h_o :

Check for flow nature

If N_{Re} <2100 [flow is laminar]

$$N_{Nu} = 1.86 [\{ 4 * m_c * C_{ph} \} / \square * k * L]^{1/3}$$

If N_{Re} > 10,000 [flow is turbulent]

$$N_{Nu} = 0.023 * [N_{Re}]^{0.8} [N_{Pr}]^{0.33}$$

$$\& \; h_o = [N_{Nu} * k] / D_e \qquad [W/m^2 \,°C]$$

If 2100 < N_{Re} < 10,000 [flow is in transition region]

(Note: No simple equation applies here. The calculation method is based on graphs of above two equations after modifications on a common plot of the Colburn j factor versus $N_{Re.}$)

Use j_H Vs N_{Re} chart from Perry's page no. [10–16] - vi edition using j_H value compute h_i .

SUMMARY OF ANNULUS SIDE CALCULATIONS

Sl. No.	$T_{h\ mean}$ °C	〉 kg/m³	© m²/s	⌠ kg/m s	C_p kJ/kg °C	K W/m K	N_{Re}	N_{pr}	h_o W/m² °C
1.									
2.									
3.									
4.									

17. Calculation of U_o [clean]

$$1 / U_o = (1 / h_{io}) + (1 / h_o) + R_w$$

where R_w is the metal wall resistance $= d_o * \ln [d_o / d_i] / 2 k_w \qquad [(m^2 °C) /W]$

k_w = thermal conductivity of tube material

RESULTS TABLE

Sl. No.	LMTD	U_O W/m² °C	$h_{ip,o}$ W/m² °C	h_o W/m² °C	R_d m² °C/W
1					
2					
3					
4					

EXPERIMENT NO. 10
TRANSIENT HEAT CONDUCTION
(CONSTANT HEAT FLUX)

AIM: To obtain the temperature profile for conduction through a rod heated by a constant heat flux source under transient conditions and also to compare the profile with theoretically predicted values.

APPARATUS: Experimental setup, stopwatch, thermometers etc.

THEORY

Conduction in homogeneous isotropic solids where the temperature distribution within the solid does change both with time and distance is called unsteady state condition.

The rate of change of heat content of the element will be equal to minus the rate of increase of heat flow from (x.y,z) to (x+dx, y+dy, z+dz). General solutions of unsteady state conduction are available for certain simple shapes such as infinite slab, infinitely long cylinder, and the sphere.

SPECIFICATIONS OF THE TRANSIENT HEAT CONDUCTION SETUP

Diameter of the rod = 25.4 mm

Rod material = Brass, thermal conductivity of brass is 110.7 W/m K

Density of brass= 8522 kg/m^3

Specific heat capacity of brass= 386 J/kg K

Distances from the hot end of the rod:

X_o = 0 cm	X_1 = 4.5 cm	X_2 = 9.5 cm	X_3 =14 cm
X_4 = 18.5 cm	X_5= 23cm	X_6=30 cm	

PROCEDURE

1. Place the thermometers along the rod.
2. Allow the cold water to flow through the cold end of the rod.
3. Record the temperatures.
4. Subject the rod to constant heat flux conditions by adjusting the voltage and current to fixed values.
5. Allow the flow of cold water through the other end of the rod to maintain the temperature constant.

6. Record the temperatures at various points at five minutes intervals till constant temperatures are attained.
7. Perform the calculations to obtain the temperature profile and compare with theoretically obtained values.

OBSERVATIONS

1. Heat flux, $\boldsymbol{Q = VI}$ $\quad$ ***[W]***

Where Voltage, V = ____Volts and Current, I = _____Ampere

2. Thermal diffusivity, $\boldsymbol{\alpha = k/(\rho\, C_p)}$ $\quad$ $\boldsymbol{[m^2/s]}$

Where K = Thermal conductivity of rod, W/m K

ρ = Density of rod, kg/m^3

3. Cross sectional area of rod, $\boldsymbol{A = \pi d^2/4}$ $\quad$ $\boldsymbol{[m^2]}$

Where d = diameter of rod, m

4. $\boldsymbol{Z = X/2\sqrt{\alpha\tau}}$

Here, τ = Time in seconds

Error function, erf(Z) to be obtained from table of Z versus error function

5. Theoretical temperature,

$$T_{the} = a + T_i$$

Where $\boldsymbol{a = ([\{2Q\sqrt{(\alpha\tau/\pi)}\ \exp\{-(x^2/4\alpha\tau)\}/kA] - [QX\{1-\mathrm{erf}(Z)\}]/kA)}$

OBSERVATION TABLE

Sl. No.	Time (min)	T_1 °C **X_o = 0 cm**	T_2 °C **X_1 = 4.5 cm**	T_3 °C **X_2 = 9.5 cm**	T_4 °C **X_3 = 14 cm**	T_5 °C **X_4= 18.5 cm**	T_6 °C **X_5= 23 cm**	T_7 °C **X_6= 30 cm**
1								
2								
3								

RESULTS TABLE

Sl. No.	X m	z	erf(z)	T_{the} °C	T_{expt} °C
1.					
2.					
3.					
4.					