HEAT TRANSFER LAB REPORTS

Experiments:

1) HEAT TRANSFER IN NATURAL CONVECTION
2) STUDIES OF HEAT TRANSFER IN VERTICAL CONDENSER
3) DETERMINATION OF CRITICAL HEAT FLUX
4) DROPWISE AND FILMWISE CONDENSATION
5) STUDIES ON HEAT TRANSFER IN PARALLEL FLOW/COUNTER FLOW HEAT EXCHANGERS
6) STUDIES ON RADIATIVE HEAT TRANSFER IN STEFAN BOLTZMANN APPARATUS

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HEAT TRANSFER IN NATURAL CONVECTION

AIM

- Study of convection heat transfer in natural convection.
- To find out the heat transfer coefficient of a vertical cylinder in natural convection.

THEORY

Natural convection is a type of flow of fluid, in which the fluid motion is not generated by any external source. The driving force for natural convection is gravitational force. The natural convection occurs due to the change in density of fluid caused due to change in temperature of fluid.

The average heat transfer coefficient during the natural convection is calculated using the value of heat flux, area of heat transfer and average temperature difference.

$$h = \frac{Q}{A\left(T_s - T_a\right)}$$

EXPERIMENTAL SET-UP

- A brass tube fitted in a rectangular duct open at the top and bottom.
- A heating element in the vertical tube
- Digital temperature indicator to measure the temperature at the different points
- The heat input to the heater is measured by digital ammeter and digital voltmeter and can be varied.

OBSERVATIONS

Specifications

- Length of the tube = 0.5 m
- Diameter of the tube = 0.038 m

Observation Table

Sl. No.	V (V)	I (A)	Time (min)	T_1 (°C)	T_2 (°C)	T_3 (°C)	T_4 (°C)	T_5 (°C)	T_6 (°C)	T_7 (°C)	T_8 (°C)
1.	40	0.22	30	36.4	37.0	37.4	37.6	37.8	36.7	36.4	30.6
2.			35	36.2	36.8	37.3	37.5	37.7	36.7	36.3	30.6
3.			40	36.2	36.8	37.3	37.5	37.7	36.6	36.3	30.6
4.	50	0.27	30	38.6	39.5	40.2	40.8	40.7	39.0	38.2	30.9
5.			35	39.1	40.0	40.7	41.2	40.9	39.5	38.6	31.0
6.			40	39.1	40.1	40.8	41.2	40.9	39.5	38.6	31.0

CALCULATIONS

Sl. No.	Q (W)	Tavg(°C)	h(W/m² °C)	Havg (W/m² °C)	Average h (W/m² °C)
1	8.8	22.879	22.88	23.17	24.354
2		23.292	23.29		
3		23.345	23.34		
4	13.5	26.382	26.38	25.52	
5		25.126	25.13		
6		25.046	25.05		

Heat Transfer Area = $\pi D_c L_c$ = 0.0597 m^2

(i) Calculating for 40V and 0.22A at steady state

Heat transfer Rate (Q) = I × V = 8.8 W

Avg. Temperature of Cylinder (T_s) = $\sum_{i=1}^{7} T_i / 7$ = 37.042°c

Air Temperature, T_a = 30.6 °c

Heat Transfer Co-efficient (h) = $Q/A(T_s - T_a)$ = 22.88 w/m²°c

For readings at 40V, 0.22A, h_{avg} = 23.17 W/m²°c

(ii) Calculating for 50V and 0.27A at steady state

Heat transfer Rate (Q) = I × V = 13.5 W

Avg. temperature of cylinder (T_s) = $\sum_{i=1}^{7} T_i / 7$ = 39.571°C

Air Temperature, T_a = 31°c

Heat Transfer Co-efficient (h) = $Q/A(T_s - T_a)$ = 26.38 W/m²°c

For readings at 50V, 0.27A, h_{avg} = 25.52 w/m²°c

∴ Average heat transfer co-efficient (h) = 24.345 w/m²°c

DISCUSSION

Heat transfer by convection is proportional to the surface area A of the object. So, the value of h will change depending on how we kept the plate. We are interested in temperature variation along the length of a metal cylinder, so we will take the characteristic length *L* to be the length of the cylinder.Thermocouples are placed along the rod as white bands, whereas there are other thermocouples inside the setup which measure the surrounding temperature. Because of the opening at the bottom and top, hot air moves to the top because of density difference and buoyancy force. The opening at the top ensures that there’s no recirculation of hot air, letting the natural convection properties sustain. While performing the experiment we need to ensure that there’s no intervention of external air in the laboratory such as Fan and Air conditioners because it hampers the condition of natural convection. Ta which we consider in this experiment is more or less constant along the length of the rod because it is measured in the enclosure far enough from the rod so that the thermal boundary layer is crossed and bulk temperature is being measured. The temperature readings measured, follow the pattern of first increasing and then decreasing, which means if we locally measure the heat transfer coefficient, then it is going to decrease and increase. This suggests that our readings are crossing from the laminar region to the turbulent region as resistance first increases and then the turbulent region comes into effect. This can be tested by finding the Reynolds number. The steady-state assumption is made when three consecutive readings are closely equal. Precautions to be taken while performing the experiment

- Ensure that all ON/OFF switches given on the panel are at OFF position.
- Ensure that the variance knob is at ZERO position, given on the panel.
- Switch on the panel with the help of Mains ON/OFF switch given on the panel.

RESULTS

- The calculated value for average heat transfer coefficient is 24.345 $W/m^2 \, {}^0C$

STUDIES ON HEAT TRANSFER IN VERTICAL CONDENSER

AIM

- To find out the overall heat transfer coefficient
- Steam side coefficient, Water side coefficient
- Plotting and Analysing the Wilson plot

THEORY

Special heat transfer devices used to liquefy vapour by removing their latent heat are called condensers. The latent heat is removed by absorbing it in a cooler liquid, called the coolant, since the temperature of the coolant obviously is increased in a condenser, the unit also acts as a heater but functionally it is the condensing action that is important and the name reflects this fact.

Different type of condensers:

- Air-cooled condenser: Natural Convection, Forced Convection
- Water-cooled condenser: Double Tube, Shell and Coil Condenser, Shell and Tube Condenser
- Evaporative Condenser

To obtain higher heat transfer coefficient larger velocities and shorter tubes should be used. The multi pass principle used in heat exchangers may also be used for the coolant in a condenser.

APPARATUS

The vertical condenser is mounted on a square tube frame. On one side of the condenser is connected by a monoblock pump with a bypass valve. An acrylic rotameter is provided to note the flow rate of water allowed to the condenser. A steam boiler with accessories is provided with a condensate measuring tank to collect the condensate. Panel board is mounted on the front with a temperature indicator and selector switch, D.P. switches for heater and for the pumps.

Consider the following temperature profiles,

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1/\Delta T_2\right)}$$

Schematic and Experimental Setup:

Experimental Set-Up

SPECIFICATIONS

1. Shell Length :1000 mm
   Shell Diameter: 150 mm
   Material: M.S.
   Insulation: Asbestos Cloth

2. Tube Length :1020 mm (Approx.)
   Tube Diameter: 12 mm
   Material: M.S.
   No. of Tubes: 14

3. Boiler Height: 450 mm
   Boiler Diameter: 350 mm
   Material: M.S.
   Heaters: 2 Kw /2 Nos

4. Pump H.P.: 0.5 HP
   Monoblock Make: Lubi/
   Any Reputed

5. Sump Tank Capacity: 36 Litres Approx.
   Material: SS
   Size: 300x300x400 mm
   Insulation: Asbestos Cloth

6. Digital Temperature Indicator:
   No. of Digits: 4
   Range: 400 ºC
   Resolution: 1 ºC

7. Thermocouple Types: Cr-Al
   No. of Thermocouples: 4
   Material: M.S.

8. Condensate Tank: 36 Litres
   Material: SS
   Size: 300X300X400 mm

OBSERVATIONS

Steam Pressure, kg/cm^2	**Water temperature, (ºC)**		**Steam temp., ºC (T_3)**	**Condensate temp., ºC (T_4)**	**Cooling water flow rate, cm^3/s**	**Condensate flow rate, cm^3/s**
	Inlet (T_1)	**Outlet (T_2)**				
1.5	35	65.5	106.5	39.0	30	130
1.5	35	63.5	106.5	40.0	40	140
1.5	35	58.5	106.5	50.0	50	150

CALCULATIONS

$\rho_w = 997\ kg/m^3 \qquad \rho_c = 0.579\ kg/m^3$

Latent heat of vapourisation, $L = 2260\ kJ/kg$
Thermal heat conductivity, $k = 0.604\ W/mK$
Viscosity, $\mu = 0.5465\ cp$

① Water Quantity
$M_w = Q_w \times \rho_w\ (kg/s)$, $Q_w = 30 \times 10^{-6}\ m^3/s$
$\Rightarrow M_w = 0.02991\ kg/s$

② Heat Transfer to water
$q_w = M_w \times C_{p_w} \times (T_2 - T_1)\ (J/s)$, $C_{p_w} = 4187\ J/kg\ K$

$\therefore q_w = 3819.6\ J/s$

③ Heat given out by stream
$q_c = Q_c \times \rho_c \times L \Rightarrow q_c = 130 \times (10^{-6}) \times 0.579 \times 2260 \times 10^3$
$= 170\ J/s$

④ $LMTD = \dfrac{T_2 - T_1}{\ln\left(\dfrac{T_3 - T_1}{T_3 - T_2}\right)} = \dfrac{65.5 - 35}{\ln\left(\dfrac{106.5 - 35}{106.5 - 65.5}\right)} = 54.84°C$

⑤ Heat Transfer Surface Area

$A = \pi d_o L N\ m^2$, $d_o = 12 \times 10^{-3} m$, $L = 1m$, $N = 14$

$\therefore A = 0.527787\ m^2$

⑥ Experimental overall heat transfer co-eff

$U_{exp} = \dfrac{q_w}{A \times LMTD} = 131.96\ W/m^2K$

⑦ Cooling water velocity through tubes

$v = Q_w / \left(\frac{\pi}{4} d_i^2 N\right) = 0.0189\ m/s$

⑧ Water side Heat Transfer Coefficient

$Re = \dfrac{\rho v d}{\mu} = 415 \qquad Pr = \dfrac{\mu C_p}{k} = 3.57$

$h_i = 1.86\ (Re \times Pr \times d_i/L)^{1/3} \times (k/d_i) = 259\ W/m^2K$

⑨ Shell side Heat Transfer Co-efficient

$h_o = 0.943\ \left(\dfrac{g k^3 L \rho_w^2}{\mu L (T_3 - T_w)}\right)^{0.25}$

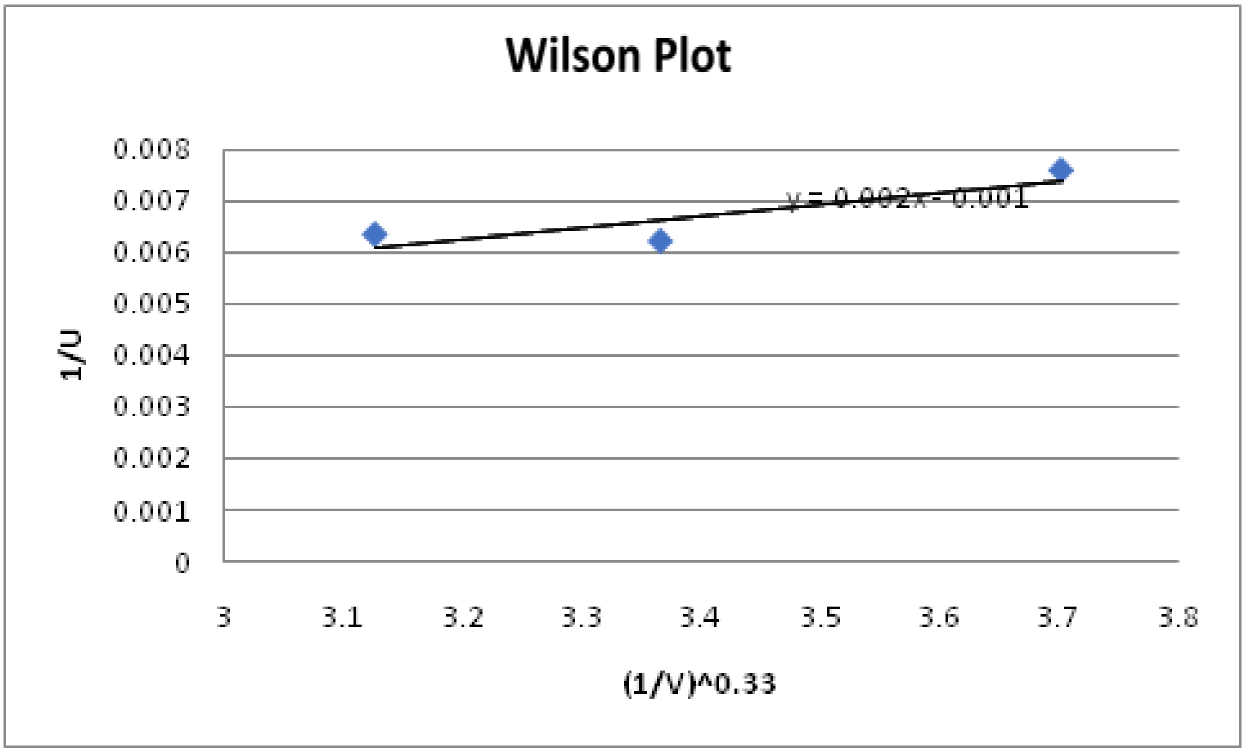
$\therefore h_o = 4152\ W/m^2K$

GRAPH

Note: For all the three set values, the flow falls in laminar region hence Dittus-Boelter equation cannot be used.

$Nu = 1.86(Re \times Pr \times di/L)^{0.33}$ => $Nu \; \alpha \; Re^{0.33}$ => $h \; \alpha \; Re^{0.33}$ => $h \; \alpha \; u^{0.33}$

$\frac{1}{U} = \frac{1}{au^{0.8}} + c$ gives the equation for **Wilson plot**



Slope = 1/a = 0.002

DISCUSSION

To get the film heat transfer coefficient, utilise the Wilson plot. The heat transfer coefficient of a horizontal condenser is higher due to drop formation, whereas the heat transfer coefficient of a vertical condenser is lower due to film formation. Because fluid must be forced up against gravity in a vertical condenser, the pumping cost is higher than in a horizontal condenser. We used a vertical condenser in our experiment because they take up less storage space and have less fouling, so we can ignore the dirt problem. Vertical condensers are mostly utilised in the food processing industry.

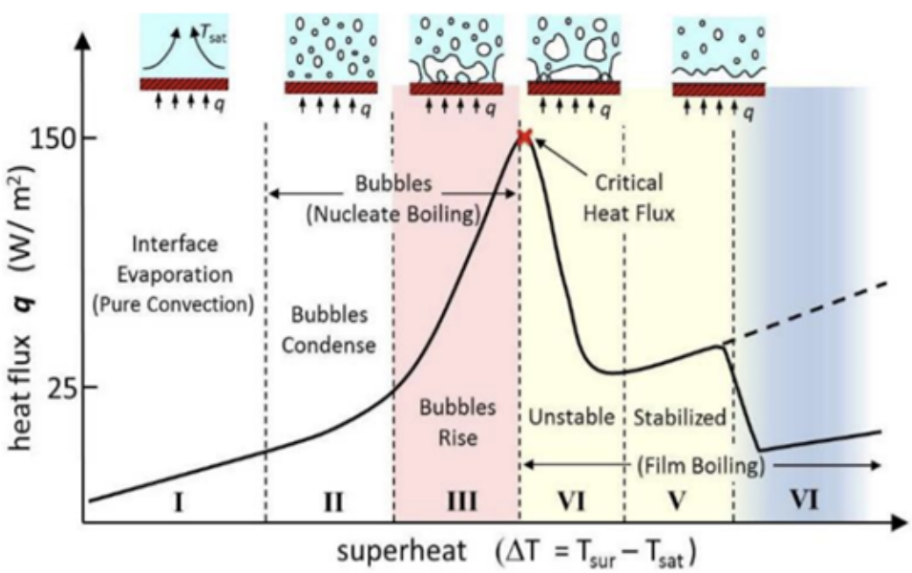
DETERMINATION OF CRITICAL HEAT FLUX

AIM

- Study of critical heat flux of a submerged nichrome wire.

THEORY

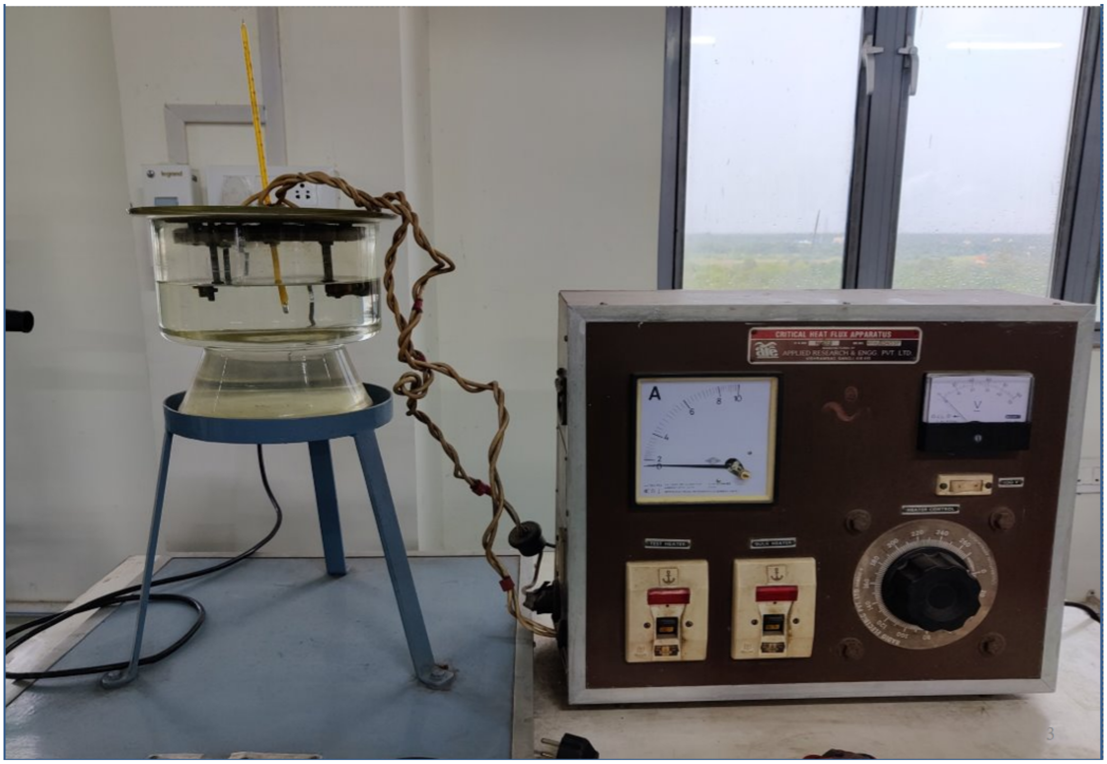
When heat is added to a liquid from a submerged solid surface which is at a temperature higher than the saturation temperature of the liquid, it is for a part of the liquid to change phase. This change of phase is called boiling. Boiling is of various types, the types depending on the temperature difference between the surface and liquid. The different types are indicated below in the figure in which a typical experimental boiling curve obtained in a saturated pool of liquid is drawn.



The heat supplied to the surface is plotted against (Tw-Ts), the difference between the temperature of the surface and the saturation temperature of the liquid. It is seen that the boiling curve can be divided into three regions; **1. Natural convection region, 2. Nuclear boiling region, 3. Film boiling region.**

As the temperature difference increases, the nichrome wire reaches the melting point and ultimately gets snapped. Heat flux value corresponding to this melting point of nichrome wire is approximately the same as the critical heat flux.

EXPERIMENTAL SET-UP



Specifications:

Length of test heater (R2) = 100 mm

Diameter of heater wire (d) = dwire

Surface area A =Π × d × LR2

OBSERVATIONS

Wire	Bulk water temperature (^{0}C)	Ammeter reading, I (Amp)	Voltmeter reading, V (Volts)
36 R [dwire = 0.005″]	50	6.6	30
	60	6.0	28
	70	5.8	28
	80	5.6	28
38 R [dwire = 0.004″]	50	5.8	44
	60	4.8	36
	70	4.4	32
	80	4.0	30
40 R [dwire = 0.0031″]	50	4.2	46
	60	3.8	42
	70	3.4	40
	80	3.0	34

CALCULATIONS

For wire - 1,

$T_b \rightarrow$ Bulk Temperature, $d_w = 0.127\,mm$
Length of test heater, $L_{R_2} = 100\,mm$

① $T_b = 50°c$, $V = 30V$, $I = 6.6A \Rightarrow Q = VI = 198\,W$
$A = \pi d L_{R_2} = \pi \times 0.127 \times 10^{-3} \times 100 \times 10^{-3} = 3.99 \times 10^{-5}\,m^2$
$q_{crit} = Q/A = 4.96\ MW/m^2$

② $T_b = 60°c$, $V = 28V$, $I = 6A \Rightarrow Q = VI = 168\,W$
$A = \pi d L_{R_2} = \pi \times 0.127 \times 10^{-3} \times 100 \times 10^{-3} = 3.99 \times 10^{-5}\,m^2$
$q_{crit} = Q/A = 4.21\ MW/m^2$

③ $T_b = 70°C$, $V = 28V$, $I = 5.8\ A \Rightarrow Q = VI = 162.4\ W$
$A = \pi d L_{R_2} = \pi \times 0.127 \times 10^{-3} \times 100 \times 10^{-3} = 3.99 \times 10^{-5}\ m^2$
$q_{crit} = Q/A = 4.07\ MW/m^2$

④ $T_b = 80°C$, $V = 28V$, $I = 5.6\ A \Rightarrow Q = VI = 156.8\ W$
$A = \pi d L_{R_2} = \pi \times 0.127 \times 10^{-3} \times 100 \times 10^{-3} = 3.99 \times 10^{-5}\ m^2$
$q_{crit} = Q/A = 3.93\ MW/m^2$

DISCUSSION

Distilled and demineralized water is used so that there is no happening of electrolysis, which ensures current is only conducted through the wire and not through the solution by ions after ionization. This should be taken care of because our power readings are dependent on current readings, which may no longer be valid if there's unmeasured current circulation inside. Later increments of the current power supply should be minute and gradual when compared to the initial readings. When compared to the actual Boiling curve, our plot indicates that the region in which we are operating is under a transition boiling regime and hence the decreasing flux. This applies even when the x-axis doesn't include T_{sat} deviation because, with T_{sat} deviation, the curve shape still remains the same, whereas only a shift occurs. On an actual boiling curve, the Heat flux at which wire snaps is at the same level as the critical heat flux means the critical heat flux values are a very close approximation for the actual values. The material of wire is chosen after taking this phenomenon into consideration. Reaching higher temperatures inculcates the influence of Radiation properties. Due to the surface tension property of the wire, we observe some bubbles released from the water pool surrounding the wire dipped into it, this is because the surface tension properties of it help the bubbles to coalesce on their surface. Critical heat flux is an important parameter for heating operations and ensures that the equipment works in the operating, safe region. The nichrome wire should not be placed too loose or too tight as for a loose wire we would add extra length of wire in our calculations while for a tightly clamped wire mechanical stress would generate leading to error in the calculations.

RESULTS

Wire	Bulk water temperature (^{O}C)	Ammeter reading, I (Amp)	Voltmeter reading, V (Volts)	$q_{critical\ heat\ flux}$ (in MW/m^2)
36 R [dwire = 0.005″]	50	6.6	30	4.96
	60	6	28	4.21
	70	5.8	28	4.07
	80	5.6	28	3.93
38 R [dwire = 0.004″]	50	5.8	44	7.99
	60	4.8	36	5.42
	70	4.4	32	4.41
	80	4	30	3.76
40 R [dwire = 0.0031″]	50	4.2	46	7.81
	60	3.8	42	6.46
	70	3.4	40	5.50
	80	3	34	4.13

STUDIES ON HEAT TRANSFER IN PARALLEL FLOW/COUNTER FLOW HEAT EXCHANGER

AIM

- Temperature distributions in parallel flow and counter flow Heat Exchanger.
- Heat transfer rate in the two runs.
- Heat transfer coefficient in parallel and counter flow runs.
- To obtain the effectiveness of the given heat exchangers.

THEORY

Heat Exchangers are devices in which heat is transferred from one fluid to another. The necessity for doing this arises in a multitude of industrial applications. Common examples of best Heat Exchanger are the radiator of a car, the condenser at the back of a domestic refrigerator and the steam boiler of a thermal power plant.

Heat Exchanger are classified in three categories:

(1) Transfer type (2) Storage type (3) Direct contact type

A transfer type of Heat Exchanger is one in which both fluids pans simultaneously through the device and heat is transferred through separating walls. In practice most of the Heat Exchangers used are transferred types.

The transferred type Exchangers are further classified according to flow arrangements as

(1) **PARALLEL FLOW** in which fluids flow in the same direction.

(2) **COUNTER FLOW** in which they flow in opposite direction.

(3) **CROSS FLOW** in which they flow at right angles to each other.

EXPERIMENTAL SET-UP

- Simple example of a transferred type of Heat Exchanger can be the form of a tube in tube type arrangement.
- A fluid flows through the inner tube and another fluid flows through the annular space and heat transfer takes place through the walls of the inner tube.
- Heat Transfer resistance $R_T = \frac{1}{UA} = \frac{1}{A_i h_i} + \frac{\ln ln \frac{r_o}{r_i}}{2\pi KL} + \frac{1}{A_o h_o}$

APPARATUS

The apparatus consists of a tube in tube type concentric tube Heat Exchanger. The hot fluid is hot water which is obtained from an electric geyser and it flows through the inner tube. The hot water flows always in one direction and the flow rate which is controlled by means of a valve. The cold water can be admitted at one of the ends enabling the Heat Exchanger to run as a parallel flow apparatus or a counter flow apparatus.

The following procedure is to be followed,

1. **Heat transfer rate, q is calculated as**
   a. q_h = heat transfer rate from hot water = $m_h C_{ph}(T_{hi}-T_{ho})$
   b. q_C = heat transfer rate to cold water = $m_c C_{pc}(T_{co}-T_{ci})$
   c. $q = (q_h + q_c)/2$ Watts or J/sec.
   (Assuming $C_{pc} = C_{ph} = 4184$ J/Kg.K)
2. **LMTD – Logarithmic mean temperature difference**:

   The LMTD can be calculated as: $LMTD = \Delta T_m = \frac{\Delta T_1 - \Delta T_2}{Ln\frac{\Delta T_1}{\Delta T_2}}$

3. **Overall heat transfer coefficient** can be calculated by using

   $q = UA\,\Delta T_m$, $\therefore U = \frac{q}{A\Delta T_m}$ (Watt/m^2 K)

- $\mathbf{U_{ri}}$ based on $A_i = \pi d_i L$
- $\mathbf{U_{ro}}$ based on $A_o = \pi d_o L$

4. The **effectiveness of the heat exchanger** can be calculated by using the expression:

$$Effectiveness = \frac{actual\ heat\ transfer}{maximum\ possible\ heat\ transfer} = \frac{m_c C_{pc}(T_{co} - T_{ci})}{m_h C_{ph}(T_{hi} - T_{ci})}$$ with $m_h C_{ph} < m_c C_{pc}$

5. The **overall heat transfer coefficient** value can be predicted by using the forced convection heat transfer correlations for flow through tube and annulus and can be compared with experimentally determined values.

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{r_i}{K} ln\left(\frac{r_o}{r_i}\right) + \left(\frac{r_i}{r_o}\right)\frac{1}{h_o}$$ where,

- hi = heat transfer coefficient for inner tube between hot water and inner tube inside surface.
- ho = heat transfer coefficient for annulus flow between outer surface of inner tube and cold water.
- $r_i = d_i/2$, $r_o = d_o/2$, K = 70 Watt/ m K. (thermal conductivity of tube material, for G.I pipe).

- $\mathbf{h_i}$ is calculated by using the correlation.

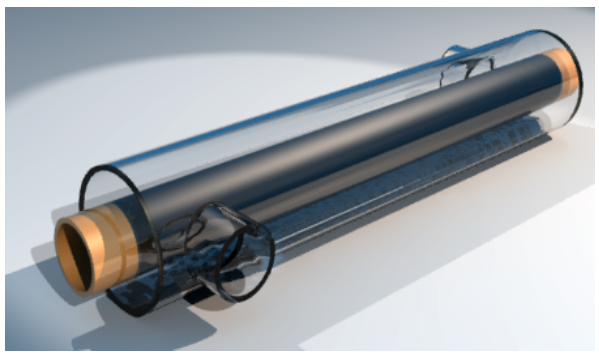
  $Nu(d_i) = 0.023\, Re_{(di)}^{0.8} (P_r)^{0.3}$, and $Nu(d_i) = h_i\, d_i/k$, (n = 0.3) when $T_w < T_f$

- $\mathbf{h_o}$ is calculated by using the correlation

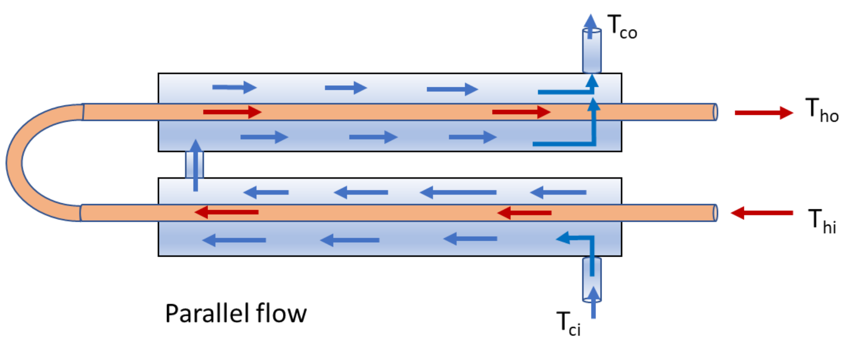
  $Nu_{(D_i - d_o)} = 0.023\, Re_{(D_i - d_o)}^{0.8} (P_r)^{0.4}$, (n = 0.4) when $T_w > T_f$

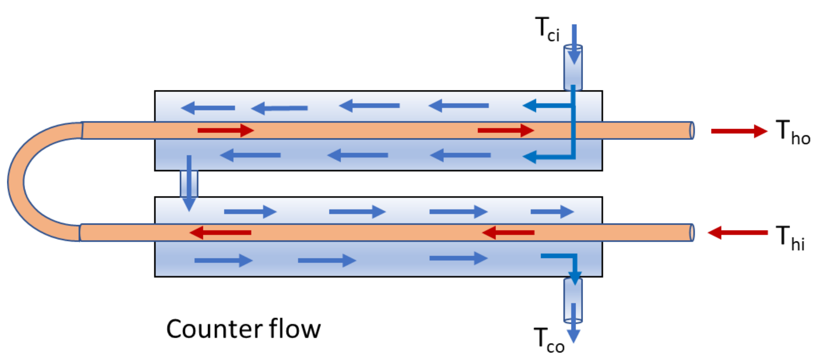
  (Above correlation applicable when $0.7 \leq Pr \leq 160$, $Re \geq 10000$, $d/L < 0.1$)

- In turbulent flow evaluate properties of water at average of bulk mean temperature $(T_{hi} + T_{ho})/2$.
  (For water k = 0.64 W/mK, $\mu = 0.5 \times 10^{-3}$ kg/m sec. at 50 ºC)



Structure of Double Pipe Heat Exchanger





OBSERVATIONS

Specifications:

Inner tube: Material: G.I. I.D. (di) = 8.10 mm O.D. (do) = 13.17 mm

Outer tube: Material: G.I. I.D. (Di) = 27.5 mm O.D. (Do) = 33.8 mm

Length of Heat Exchanger L = 2m

Thermal conductivity of tube material is 70 w/m-K.

PARALLEL FLOW

Sr. No.	Hot Water Side			Cold Water Side		
	Flow rate m_h, kg/hr	T_{hi}, ^0C	T_{ho}, ^0C	Flow rate m_c, kg/hr	T_{ci}, ^0C	T_{co}, ^0C
1.	61	58.7	47.3	156	32.0	35.9
2.	150	59.2	51.2	155	32.2	37.9
3.	192	59.3	52.7	152	32.2	38.3

COUNTER FLOW

Sl. No.	Hot Water Side			Cold Water Side		
	Flow rate m_h, kg/hr	T_{hi}, ^0C	T_{ho}, ^0C	Flow rate m_c, kg/hr	T_{ci}, ^0C	T_{co}, ^0C
1.	63	58.3	47.5	138	33.1	36.3
2.	146	59.2	51.5	137	32.9	38.0
3.	189	58.7	52.2	143	32.8	38.1

CALCULATIONS

Parallel Flow

$\dot{m}_h = 61\ kg/hr$

$T_{hi} = 58.7°c$

$T_{ho} = 47.3°c$

$\dot{m}_c = 156\ kg/hr$

$T_{ci} = 32°c$

$T_{co} = 35.9°c$

$$q_h = \dot{m}_h C_p (T_{hi} - T_{ho}) = \frac{61}{3600} \times 4184 \times (58.7 - 47.3)$$
$$= 808.21\ W$$

$$q_c = \dot{m}_c C_p (T_{co} - T_{ci}) = \frac{156}{3600} \times 4184 \times (35.9 - 32)$$
$$= 707.09\ W$$

$$q = (q_c + q_h)/2 = 757.65\ W$$

$$LMTD = \frac{(T_{hi} - T_{ci}) - (T_{ho} - T_{co})}{\ln\left(\frac{T_{hi} - T_{ci}}{T_{ho} - T_{co}}\right)} = 17.978°c$$

$$U_{ri} = q/(\pi d_i L)(LMTD) = 828.49\ W/m^2K$$

$$U_{ro} = q\ ((\pi d_o L)(LMTD) = 509.55\ W/m^2K$$

$$Re_h = \frac{\dot{m}_h d_i}{\mu \frac{\pi}{4} d_i^2} = 4800 \qquad Re_c = \frac{\dot{m}_c (D_i - d_o)}{\mu \times \frac{\pi}{4}(D_i^2 - d_o^2)} = 1385.67$$

$$Nu_h = 0.023\ Re_h^{0.8}\ Pr^{0.3} = 26.93 \qquad Nu_c = 0.023\ Re_c^{0.8}\ Pr^{0.4} = 17.97$$

$$h_i = \frac{Nu_h \times k_h}{d_i} = 2367.32\ W/m^2K \qquad h_o = \frac{Nu_c \times k_c}{(D_i - d_o)} = 211.43\ W/m^2K$$

$$U = \left[\frac{1}{h_i} + \frac{r_i}{K} \ln\left(\frac{r_o}{r_i}\right) + \left(\frac{r_i}{r_o}\right) \times \frac{1}{h_o}\right]^{-1}$$

$$\therefore\ U = 822.47\ W/m^2K$$

$$Effectiveness = \frac{\dot{m}_c\ C_{pc}\ (T_{co} - T_{ci})}{\dot{m}_h\ C_{ph}\ (T_{hi} - T_{ci})} = 0.374$$

Counter Flow

$\dot{m}_h = 63$ kg/hr $\quad\quad$ $\dot{m}_c = 138$ kg/hr

$T_{hi} = 58.3$ °c $\quad\quad$ $T_{ci} = 33.1$ °c

$T_{ho} = 47.5$ °c $\quad\quad$ $T_{co} = 36.3$ °c

$$q_h = \dot{m}_h C_p (T_{hi} - T_{ho}) = \frac{63}{3600} \times 4184 \times (58.3 - 47.5)$$
$$= 790.77 W$$

$$q_c = \dot{m}_c C_p (T_{co} - T_{ci}) = \frac{138}{3600} \times 4184 \times (36.3 - 33.1)$$
$$= 513.24 W$$

$$q = (q_c + q_h)/2 = 652.01 W$$

$$LMTD = \frac{(T_{hi} - T_{co}) - (T_{hi} - T_{co})}{\ln\left(\frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}}\right)} = 17.978°c$$

$$U_{ri} = q / (\pi d_i L)(LMTD) = 714.865 \ W/m^2K$$

$$U_{ro} = q((\pi d_o L)(LMTD) = 439.67 \ W/m^2K$$

$$Re_h = \frac{\dot{m}_h d_i}{\mu \frac{\pi}{4} d_i^2} = 4957.06 \quad\quad Re_c = \frac{\dot{m}_c (D_i - d_o)}{\mu \times \frac{\pi}{4}(D_i^2 - d_o^2)} = 1688.09$$

$$Nu_h = 0.023 \ Re_h^{0.8} \ Pr^{0.3} = 30.41 \quad\quad Nu_c = 0.023 \ Re_c^{0.8} \ Pr^{0.4} = 16.29$$

$$h_i = \frac{Nu_h \times k_h}{d_i} = 2429.21 \ W/m^2K \quad\quad h_o = \frac{Nu_c \times k_c}{(D_i - d_o)} = 735.63 \ W/m^2k$$

$$U = \left[\frac{1}{h_i} + \frac{r_i}{K} \ln\left(\frac{r_o}{r_i}\right) + \left(\frac{r_i}{r_o}\right) \times \frac{1}{h_o}\right]^{-1}$$

$$\therefore U = 783.79 \ W/m^2K$$

$$\text{Effectiveness} = \frac{\dot{m}_c C_{pc} (T_{co} - T_{ci})}{\dot{m}_h C_{ph} (T_{hi} - T_{ci})} = 0.278$$

RESULTS

Parallel Flow

Sl. No.	q_h (W)	q_c (W)	Q (W)	LMTD (°C)	U_{ri} (W/m^2 K)	U_{ro} (W/m^2 K)	Effectiveness
1	808.21	707.10	757.65	41.395	359.93	221.37	0.374
2	1394.67	1026.82	1210.8	44.551	534.42	328.69	0.218
3	1472.77	1077.61	1275.2	46.248	542.21	333.48	0.178

Sl. No.	Re_i	Re_o	Nu_i	Nu_o	h_i (W/m^2 K)	h_o (W/m^2 K)	Ui (W/m^2 K)
1	5284.72	1886.82	31.17	7.39	2497.12	321.09	423.08
2	12995.22	1874.72	64.02	7.37	5129.18	320.41	462.49
3	16633.88	1838.44	77.99	7.32	6249.07	318.35	467.33

Counter Flow

Sl. No.	q_h (W)	q_c (W)	Q (W)	LMTD (°C)	U_{ri} (W/m^2 K)	U_{ro} (W/m^2 K)	Effectiveness
1	790.78	513.24	652.01	41.291	310.51	190.97	0.278
2	1306.57	812.04	1059.3	45.756	455.25	279.99	0.182
3	1427.79	880.85	1154.3	46.038	493.04	303.24	0.155

Sl. No.	Re_i	Re_o	Nu_i	Nu_o	h_i (W/m^2 K)	h_o (W/m^2 K)	Ui (W/m^2 K)
1	5457.99	1669.11	31.98	7.09	2562.40	308.36	410.97
2	12648.68	1657.01	62.65	7.08	5019.46	307.62	445.02
3	16373.98	1729.58	77.02	7.18	6170.84	312.00	458.39

The following have been used to derive these results,

- For water, $C_{ph} = C_{pc} = 4184$ J/kg K
- Calculating U_{ri} based on surface $Ai = \pi\, di\, L = 3.14*0.0081*2 = 0.050868\ m^2$
  and U_{ro} based on surface $Ao = \pi\, do\, L = 3.14*0.01317*2 = 0.082708\ m^2$
- $r_i = d_i/2 = 0.0041$ m, $ro = do/2 = 0.0066$ m
- ***Inner Tube:*** $D_e = d_i = 0.0081$ m => $A_f = \pi\, d_i^2/4 = 5.15\times10^{-5}\ m^2$

- ***Annular Space:*** $D_e = D_i - d_o = 0.01433$ m => $A_f = \pi (D_i^2 - d_o^2)/4 = 4.577\times10^{-4}$ m^2
- K = thermal conductivity of tube material = 70 Watt/m K (for G.I. pipe assumed)
- Nu (di) = 0.023 $(Re_{di})^{0.8} (Pr)^{0.3}$ = hi di/k (Re > 3000)
- Nu (Di-do) = 1.86 $(Re.Pr.De/L)^{0.33}$ = ho De/k (Re < 2300)
- Properties of water at average of bulk mean temperature (hot water side) 55 ºC
  μ = 0.504 × 10^-3 kg/m sec, k = 0.649 W/m K
  Pr = $(\mu C_{ph}/k)$ = 3.249
- Properties of water at average of bulk mean temperature (cold water side) 35 ºC
  μ = 0.719 × 10^-3 kg/m sec, k = 0.623 W/m K
  Pr = $(\mu C_{ph}/k)$ = 4.829
- Viscosity correction factor $(\mu / \mu_w)^{0.14}$ due to wall temperature is considered 1.

DISCUSSION

From the sample calculation, we find that the overall heat transfer coefficient is greater for parallel flow when compared with counterflow. Though this may appear to differ from the literature that for counterflow, heat transfer and heat transfer coefficient is higher, it is in accordance with the theory. We have considered different flow rates for the two cases hence it is not correct to compare U-values. Informed decisions must be taken while choosing one liquid to be placed in the inner tube and the other in the annular region. This will depend on several parameters like the corrosive or fouling nature of the fluid. For eg: a highly fouling liquid should be placed in the inner tube so that the shell is protected from damage. Countercurrent flows are preferred as less area is required to get the same heat transfer. Other commonly used heat exchangers are shell and tube, heat exchangers. In fact, the shell and tube heat exchanger is more compact and provides higher heat exchange due to the bundle of tubes. The double pipe is used when the heat exchange required is small.

Dropwise and Filmwise Condensation

AIM

- To determine the individual heat transfer coefficient in Dropwise and Filmwise condensation using a condensation apparatus.

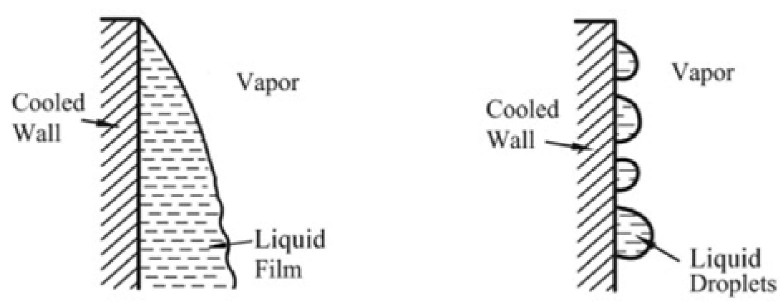
THEORY

Condensation is a natural phase change phenomenon by which matter changes from gaseous to liquid phase by the loss of heat energy. It has applications in power generation, process industry, fog harvesting, etc. During condensation, very high heat fluxes are possible.

Condensation usually occurs in 2 modes:

1. Filmwise condensation: A film spread on the surface
2. Dropwise condensation: Bead-like droplets on the surface

Wettable surfaces undergo filmwise condensation. Unless specially treated, most materials are wettable by water. Dropwise condensation is much more effective than filmwise condensation, and hence, much more desirable



(a) Filmwise condensation (b) Dropwise condensation

APPARATUS

1) Electrically heated boiler (steam generator) 2) Voltage regulator 3) Glass cylinder with two wire cooled copper condensers 4) One of the condensers is gold plated to make it hydrophobic 5) Rotameter 6) Pressure gauge 7) Digital temperature indicator to measure temperature of steam, cooling water inlet and outlet, and condenser midpoint.

Experimental Setup

The equipment consists of a metallic container in which steam generation takes place. A suitable electric heater is installed in the lower portion of the container which heats water and facilitates steam generation. An opening is provided in the cover for filling water.

The glass cylinder houses two water cooled copper condensers, one of which is chromium plated to promote drop wise condensation and the other is in its natural state to give film wise condensation. A pressure gauge is provided to measure the steam pressure. A Rotameter is provided to measure flow rate of cooling water through the condenser. A multi–channel digital temperature indicator is provided to measure temperature of steam, condenser surfaces, condenser cooling water inlet and outlet.

OBSERVATIONS

SPECIFICATIONS

- <u>Copper tube condenser</u>

  Inner diameter, di = 11 mm

  Outer diameter, do = 12.7 mm

  Length of the tube, L = 250 mm

- <u>Gold plated condenser</u>

  Inner diameter, di = 11 mm

  Outer diameter, do = 12.7 mm

  Length of the tube, L = 250 mm

- <u>Measured temperatures (˚C):</u>

  T1 = Water inlet temperature for both tubes

  T2 = Water temperature in copper tube at centre distance

  T3 = Water temperature at outlet from copper tube

  T4 = Water temperature in gold plated tube at centre distance

  T5 = Water temperature at outlet from gold plated tube

T6 = Vapor temperature around copper tube

T7 = Vapor temperature around gold plated tube

Film wise condensation (Copper Condenser) (Plain)

S.no	Water flow rate, Mw lpm (≈ kg /min)	Steam Pressure,Ps Kg /cm2 (≈ bar)	Temperature(^{0}C)			
			T1	**T2**	**T3**	**T6**
1	20	1.0	39.4	43	45.9	100
2	30	1.0	40.1	42.4	45.4	96.5
3	40	1.0	39.6	42.6	44.5	93.8

Drop wise condensation: (Gold Plated Condenser):

S.no	Water flow rate, Mw lpm (≈ cm^3 /s)	Steam Pressure,Ps Kg /cm2 (≈ bar)	Temperature(^{0}C)			
			T1	**T4**	**T5**	**T7**
1	20	1.0	39.1	42.5	47.5	85
2	30	1.0	38.9	41.6	45.5	78.9
3	40	1.0	38.9	41.3	44.5	75

CALCULATIONS

a) Film wise condensation

Flow rate = 20 cm^3/s

Heat picked up water, $Q = m C_p (T_3 - T_1) = (0.02)(4187)(6.5)$
$= 544.31 W$

$$LMTD(T_m) = \frac{T_3 - T_2}{\ln\left(\frac{T_6 - T_2}{T_6 - T_3}\right)} = \frac{45.9 - 43}{\ln\left(\frac{100 - 43}{100 - 45.9}\right)} = 55.55^\circ C$$

Overall heat transfer coefficient,

$$U = \left(\frac{Q}{A T_m}\right) = \frac{544.31}{0.003175 \times 55.55} = 3086.16 \ W/m^2K$$

Experimental Surface heat transfer Co-efficient,

$$h = \left(\frac{Q}{A(T_s - T_a)}\right) = \frac{544.31}{0.003175 \times 57} = 3007.65 \ W/m^2K$$

b) Drop wise condensation

Flow rate = 20 cm^3/s

Heat picked up water, $Q = m C_p (T_5 - T_1) = (0.02)(4187)(8.4)$
$= 703.42 W$

$$LMTD(T_m) = \frac{T_5 - T_4}{\ln\left(\frac{T_7 - T_4}{T_7 - T_5}\right)} = \frac{47.5 - 42.5}{\ln\left(\frac{85 - 42.5}{85 - 47.5}\right)} = 39.95^\circ C$$

Overall heat transfer coefficient,

$$U = \left(\frac{Q}{A T_m}\right) = \frac{703.42}{0.003175 \times 39.95} = 5545.67 \ W/m^2K$$

Experimental Surface heat transfer Co-efficient,

$$h = \left(\frac{Q}{A(T_s - T_a)}\right) = \frac{703.42}{0.003175 \times 42.5} = 5212.93 \ W/m^2K$$

DISCUSSION

We find that the overall heat transfer coefficient and surface heat transfer coefficient is higher for dropwise condensation compared to filmwise condensation. This implies that dropwise condensation is much more effective than filmwise condensation.In filmwise condensation, the film of liquid provides a higher heat transfer resistance so less heat transfer occurs compared to dropwise condensation.In filmwise condensation we use copper tube whereas in dropwise condensation we use gold plated tube because gold plating makes it hydrophobic leading to the formation of droplets on the surface of the tube.

We must ensure that the steam generator remains submerged in water (at least 3/4th) otherwise the heating coil will be exposed to air and it might get burned out due to high voltages.Following precautions should be taken while performing the experiment,

- Do not start heater supply unless water is filled in the steam generator to nearly ¾ of its capacity. If water is insufficient in the steam generator the heater burns out.
- After switching on the boilers, temperature readings attaining steady-state should only be considered for the heat transfer coefficient calculation.
- Operate gently the selector switch of temperature indicator as well as control valves.

RESULTS

Film wise condensation (Copper Condenser) (Plain)

S.no	**Water flow rate (cm^3/s)**	**Q (W)**	**Tm (^0C)**	**U (W/m^2)**	**h (W/m^2K)**
1	20	544.31	55.55	3086.16	3007.65
2	30	665.73	52.58	3987.80	3875.76
3	40	820.65	50.24	5144.75	5048.29

Drop wise condensation: (Gold Plated Condenser):

S.no	**Water flow rate (cm^3/s)**	**Q (W)**	**Tm (^0C)**	**U (W/m^2)**	**h (W/m^2K)**
1	20	703.42	39.95	5545.67	5212.93
2	30	829.03	35.31	7394.84	5099.84
3	40	937.89	32.07	9211.05	8765.53

STUDIES ON RADIATIVE HEAT TRANSFER IN STEFAN-BOLTZMANN APPARATUS

AIM

- To determine the value of Stefan-Boltzmann constant for radiation heat transfer.

THEORY

Stefan-Boltzmann law states that the emissive power of a perfect black body is directly proportional to the fourth power of the absolute temperature

$$Q = \sigma \varepsilon A T^4$$

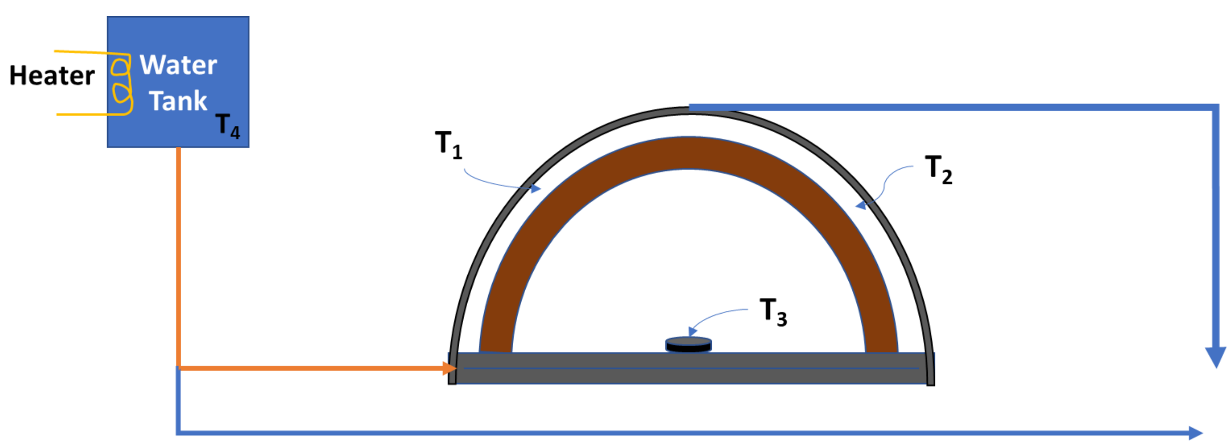
σ – Stefan Boltzmann constant (W/m^2K^4)

$E_a(W/m^2)$ = total emissive power = $\int_0^\infty E_{a\lambda} d\lambda = \sigma\varepsilon T^4$

whereas $E_{a\lambda}$ (W/m^2 µm) is the amount of radiant energy emitted by surface a per unit area per unit time per unit wavelength

ε = (*energy radiated by a body*) / (*energy radiated by blackbody*), for blackbody Ɛ = 1

Experimental Setup



OBSERVATIONS

SPECIFICATIONS

- Diameter of the disc (d) = 20 mm
- Thickness of the disc = 1.5 mm
- Mass of the Disc (m) = 5 g
- Specific heat of the disc (C_p) = 380 J/Kg-K
- Inner-dia. of hemisphere surface = 200 mm

Table-1

Thermocouple Number	**Temperature (ºC)**	**Temperature (K)**
1- Sphere (left side temp)	t1 = 70.5	T1 = 343.6
2- Sphere (right side temp)	t2 = 70.4	T2 = 343.5
4- Hot water temp	t4 = 100	T4 = 373.1

Table-2

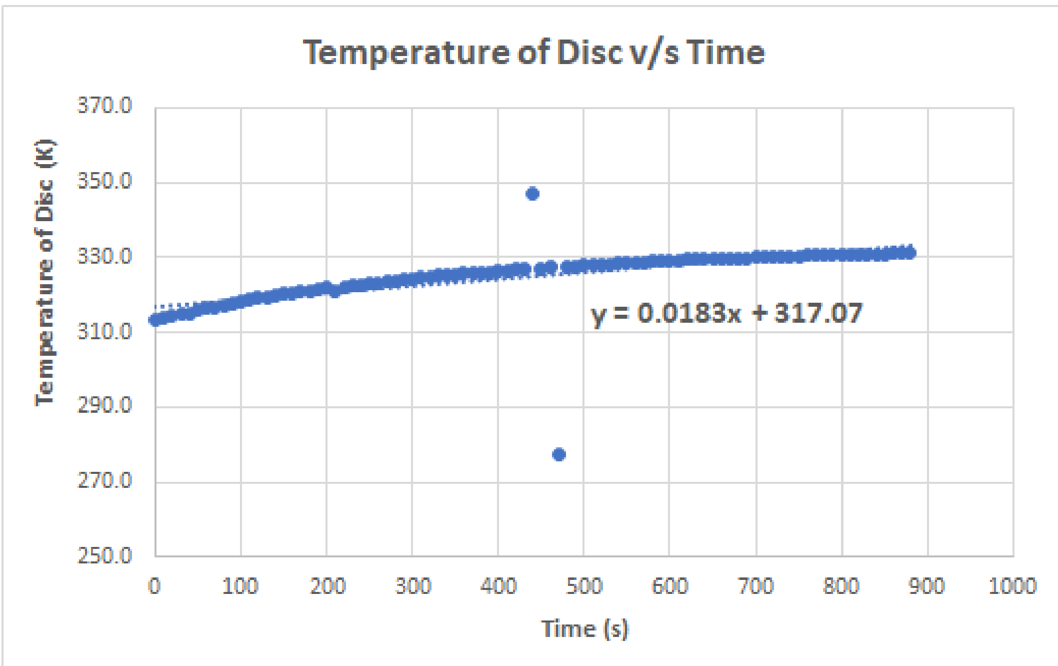
Time (s)	**Temperature of Disc (ºC)**	**Temperature of Disc (K)**	**Time (s)**	**Temperature of Disc (ºC)**	**Temperature of Disc (K)**
0	40	313.2	150	47	320.2
10	40.6	313.8	160	47.4	320.6
20	41.1	314.3	170	47.7	320.9
Time (s)	**Temperature of Disc (ºC)**	**Temperature of Disc (K)**	**Time (s)**	**Temperature of Disc (ºC)**	**Temperature of Disc (K)**
30	41.6	314.8	180	47.8	321.0
40	41.9	315.1	190	48.2	321.4
50	42.7	315.9	200	48.7	321.9

60	43.3	316.5	210	47.8	321.0
70	43.6	316.8	220	49	322.2
80	44.1	317.3	230	49.2	322.4
90	44.5	317.7	240	49.6	322.8
100	44.9	318.1	250	49.9	323.1
110	45.7	318.9	260	50.1	323.3
120	45.9	319.1	270	50.2	323.4
130	46.2	319.4	280	50.5	323.7
140	46.7	319.9	290	50.9	324.1
300	51.1	324.3	510	54.8	328.0
310	51.5	324.7	520	54.9	328.1
320	51.7	324.9	530	55	328.2
330	51.9	325.1	540	55.1	328.3
340	52.1	325.3	550	55.2	328.4
350	52.2	325.4	560	55.4	328.6
360	52.4	325.6	570	55.5	328.7
370	52.6	325.8	580	55.7	328.9
380	52.8	326.0	590	55.8	329.0
390	52.9	326.1	600	55.9	329.1

400	53	326.2	610	56	329.2
410	53.2	326.4	620	56.2	329.4
420	53.6	326.8	630	56.3	329.5
430	53.7	326.9	640	56.3	329.5
440	73.8	347.0	650	56.4	329.6
450	54	327.2	660	56.5	329.7
460	54.1	327.3	670	56.6	329.8
470	4.2	277.4	680	56.6	329.8
480	54.4	327.6	690	56.7	329.9
490	54.5	327.7	700	56.8	330.0
500	54.7	327.9	710	56.9	330.1
720	57	330.2	810	57.6	330.8
730	57.1	330.3	820	57.7	330.9
Time (s)	**Temperature of Disc (ºC)**	**Temperature of Disc (K)**	**Time (s)**	**Temperature of Disc (ºC)**	**Temperature of Disc (K)**
740	57.2	330.4	830	57.8	331.0
750	57.2	330.4	840	57.8	331.0
760	57.3	330.5	850	57.8	331.0
770	57.4	330.6	860	57.9	331.1

780	57.5	330.7	870	58	331.2
790	57.5	330.7	880	58	331.2
800	57.6	330.8			

GRAPH



CALCULATIONS

From the plot we get, slope of the line graph as 0.0183 K/s

Thus, $\frac{dT}{dt} = 0.0183$ K/s

Area of disc (A_D) = (CSA + Area of Top surface) = $4.084 \times 10^{-4} m^2$

Temp. of disc at t=0 (T_D) = 313.2 K

Avg. temperature of hemisphere $(T_{avg}) = \frac{1}{2}(T_1 + T_2) = 343.55$ K

By heat balance,

Rate of change of heat capacity of disk = Net energy radiated on the disk

$$m C_p \frac{dT}{dt} = \sigma A_D (T_{avg}^4 - T_D^4)$$

$$\sigma = \left(m C_p \frac{dT}{dt}\right) \times \frac{1}{A_D (T_{avg}^4 - T_D^4)} = \frac{5 \times 10^{-3} \times 380 \times 0.0183}{4.084 \times 10^{-4} (343.55^4 - 312.2^4)}$$

$$\therefore \sigma = 1.977 \times 10^{-8} \ W/m^2K^4$$

DISCUSSION

We know that the theoretical value of the Stefan-Boltzmann constant is 5.67 x 10-8 W/m^2K^4 whereas experimentally we obtain a lower value of 1.977 x 10^{-8} W/m^2K^4. This is because we have taken emissivity (ε) = 1 in our calculations but this is true only for black body radiation. In practice, the given disc will have an emissivity of less than 1. From the plot we observe that the temperature of the disc increases linearly with time i.e., dT/dt is a constant. Inlet hot water enters from the bottom and outlet hot water flows out from the top so that water fills the entire hemispherical annulus and this ensures uniform heat transfer. The apparatus is coated with Plaster of Paris as it is a good insulator and also it does not develop cracks.

RESULTS

- Stefan-Boltzmann constant for radiation heat transfer = 1.977 x 10^{-8} W/m^2K^4