Polymer Process Engineering
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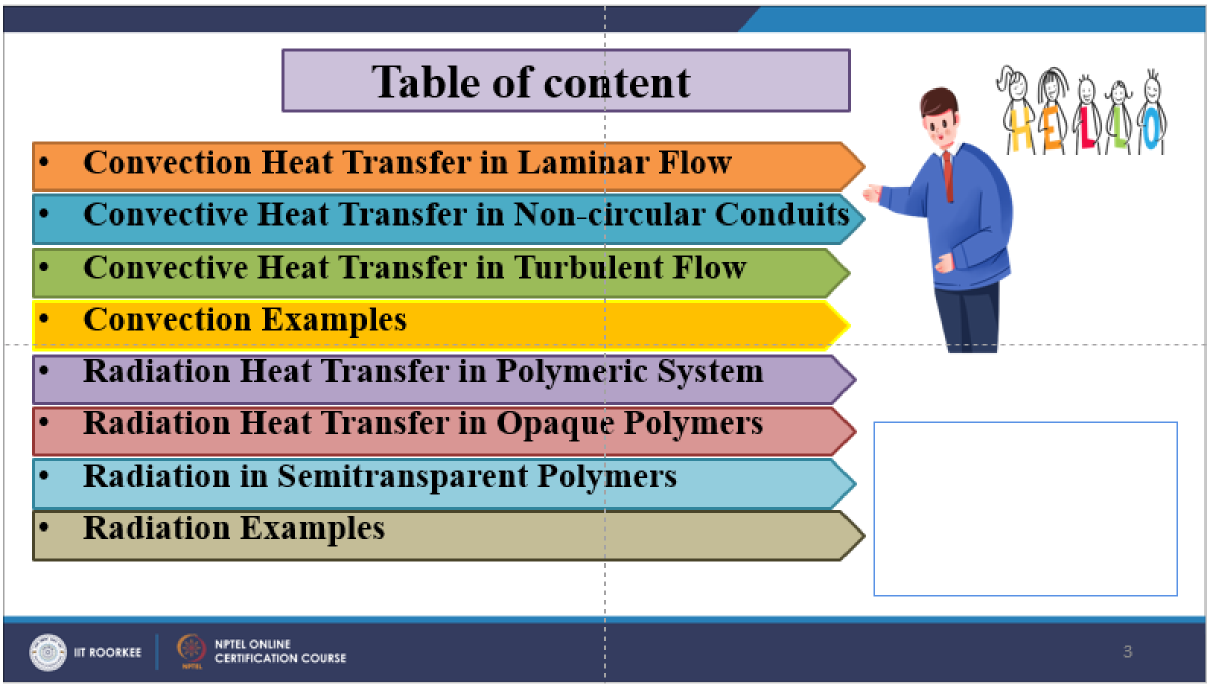
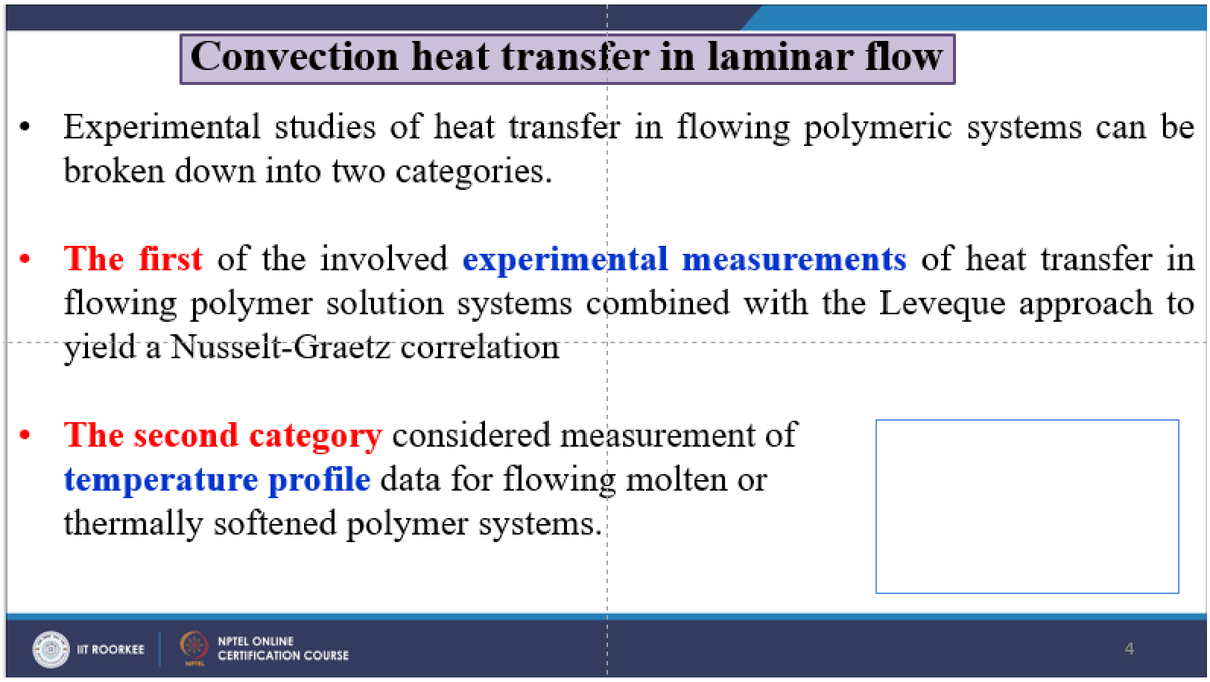
Lecture – 20
Heat Transfer Phenomenon in polymer systems: Convection and Radiation

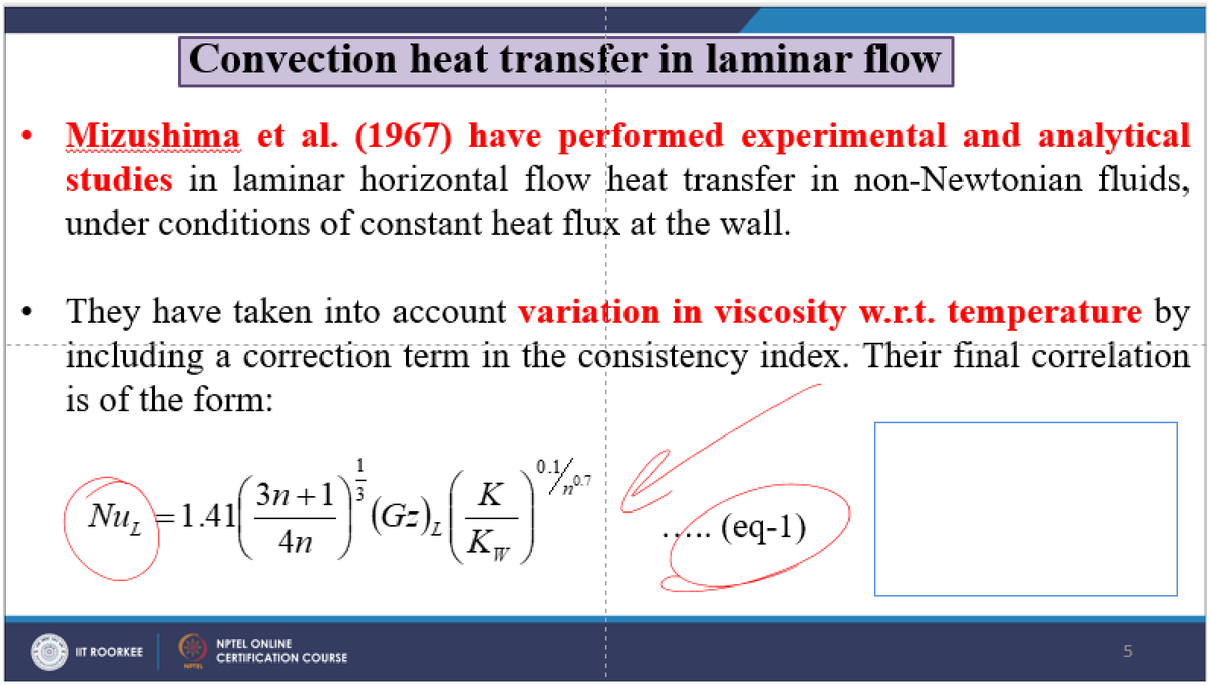
Hello friends, welcome to the convection and radiation part 2. Under the edges of heat transfer phenomena in polymer process engineering. Now, let us have a brief outlook on what we discussed in the previous lecture. We discussed the conduction in polymeric systems with a couple of examples. Then, we discussed about convection in the polymeric system, convection in the laminar flow polymeric system, and convection in the laminar flow of the molten polymers. Then we discussed about the heat transfer data of the molten polymers and soft polymers, and then heat transfer in the turbulent flow of polymers.



In this particular aspect, we are going to discuss in detail the convection heat transfer in the laminar flow, and convection heat transfer in non-circular conduits. We are going to discuss the convective heat transfer in the turbulent flow with certain examples. Then apart from this, we are going to discuss the heat transfer in the polymeric system, convection heat transfer in the opaque polymers, and radiation in the semi-transparent polymers, and we will discuss some of the examples pertaining to radiation. Now, let us talk about the convective heat transfer in laminar flow.



So, various experimental studies of heat transfer in the flowing polymeric system, can be broken down into various categories, especially we are going to discuss the two categories. The first category of the involved experimental measurement of heat transfer in a flowing polymer solution, this system combined with the Leibnitz approach to yield an asset grayish correlation. The second category is considered the measurement of temperature profile data for flowing molten or thermally softened polymer systems. Now, several authors have performed experimental and analytical studies in laminar horizontal flow heat transfer in non-Newtonian fluids under the condition of constant heat flux at the wall. They have considered the variation in the viscosity with respect to temperature by including a correlation term in the consistency index.



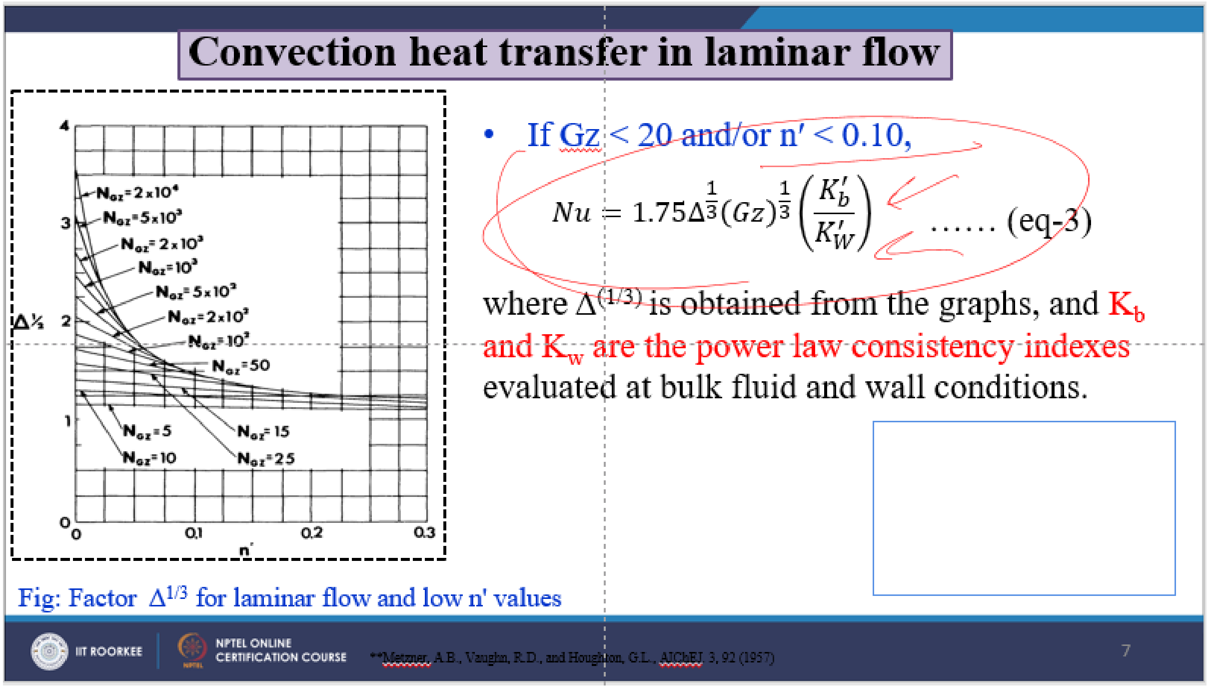
$$Nu_L = 1.41\left(\frac{3n+1}{4n}\right)^{\frac{1}{3}}(Gz)_L\left(\frac{K}{K_W}\right)^{0.1/n^{0.7}}$$

So, their final correlation which is represented in this form, you can see that Nusselt number in the laminar flow, this is equation number 1.

$$Nu = 1.75\left(\frac{3n' + 1}{4n'}\right)^{\frac{1}{3}}(Gz)^{\frac{1}{3}}\left(\frac{K_b'}{K_W'}\right)^{0.14}$$

Now, from this particular equation, some of the co-workers modified the first category by doing their experiments and their findings. They have summarized that if the grade's number is greater than 10 and the eta is greater than 0.1, then the equation can be modified to this one. This is represented as equation number 2, where k b and k w are the power law consistency indexes evaluated at bulk fluid and wall conditions and this is well represented in this particular figure. Now, if the grade's number is less than 20 and the eta dash is less than 0.

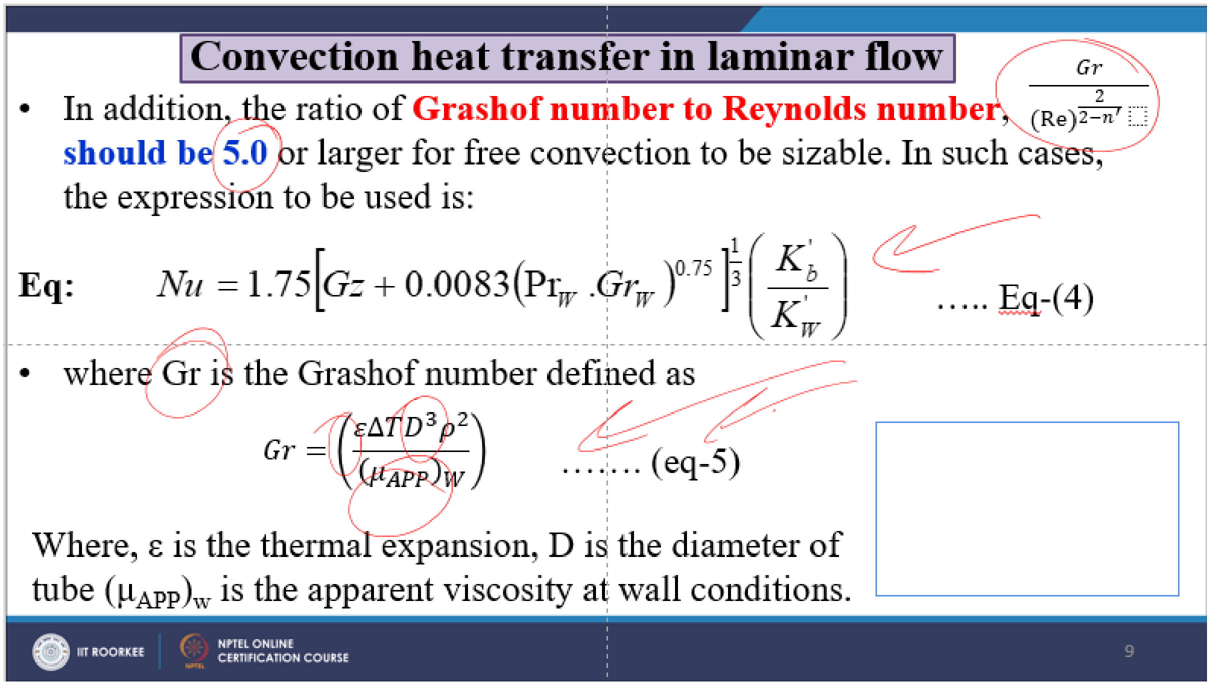
$$Nu = 1.75\Delta^{\frac{1}{3}}(Gz)^{\frac{1}{3}}\left(\frac{K_b'}{K_W'}\right)$$



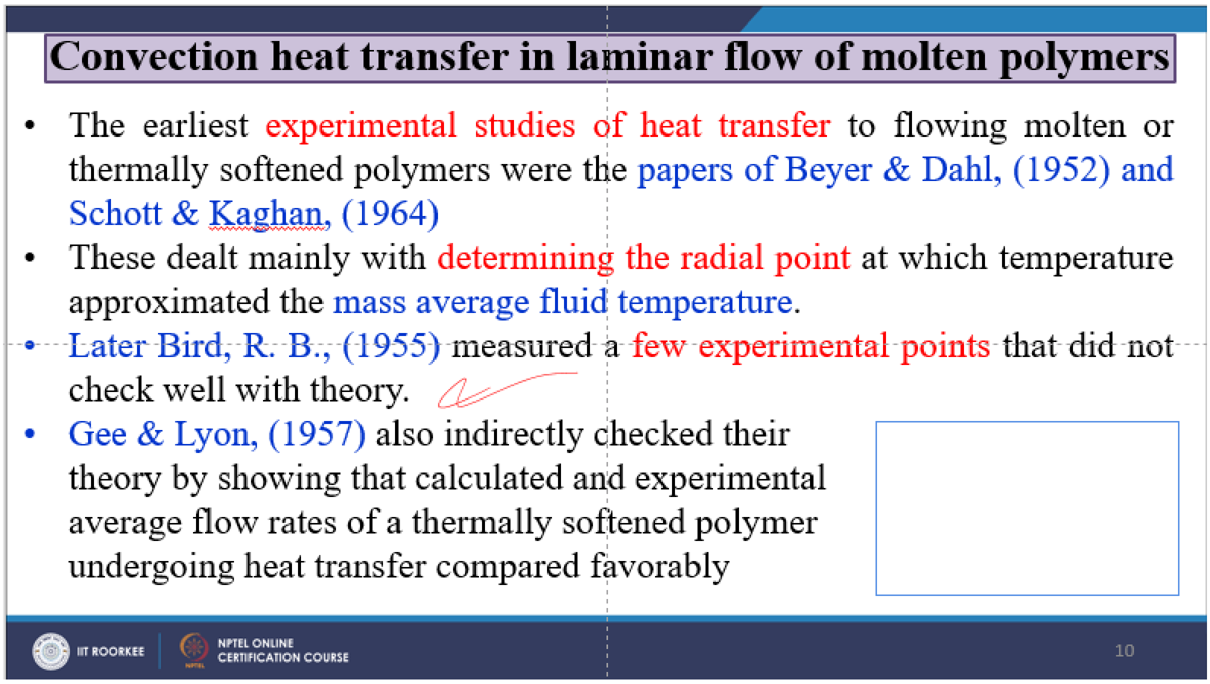
1, then the equation can be modified like this, we have represented in equation number 3. Delta to the power one upon 3 is obtained from the graphs, and k b and k w are the power law consistency index and evaluated at a bulk fluid and wall conditions. Now, other experimental studies for the polymer solution involve the work of the thermal entry reason and a constant wall flux. Some of the workers have used a numerical solution of the energy equation to test the experimental data over the grade's number which ranges from 80 to 1600. Then, Joshi and Bergles in 1974, used a constant wall flux and correlated the power law fluids, the effect of fluid property variation with the temperature.

$$Nu = 1.75\left[Gz + 0.0083\left(\Pr_W . Gr_W\right)^{0.75}\right]^{\frac{1}{3}}\left(\frac{K_b'}{K_W'}\right)$$

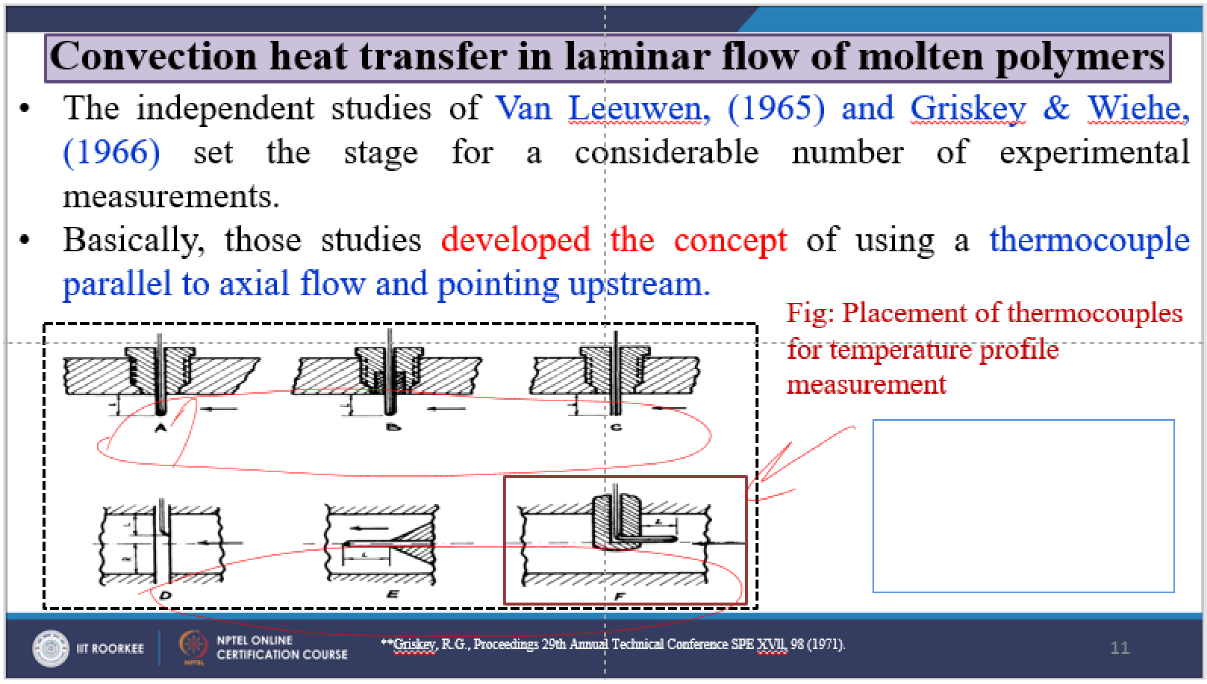
$$Gr = \left(\frac{\varepsilon \Delta T D^3 \rho^2}{(\mu_{APP})_W}\right)$$



Certain investigators, they found some of the cases that free or natural convection, they occurred in the certain system and these are generally confined to the low grade's number that is around 20 reason. Now, in addition, the ratio of Grashof number to Reynolds number, this is like this, it should be 5 or larger for free convection to be sizable. In such cases, the expression is to be used like this, which is represented as equation number 4, where the Grashof number is defined as per this usual notation where this epsilon is a thermal expansion, D is the diameter of the tube, mu is the apparent viscosity at wall condition. The earliest experimental studies of heat transfer to the flowing molten or thermally softened polymers, papers of Byer and Dahl in 1952 and Scott and Kagan in 1964, these did mainly with the determining the radial point at which the temperature approximated the mass average fluid temperature. Bergles in 1955 measured a few experimental point that did not check well with the theory.



Thereafter, some of the co-workers they also indirectly checked their theory by showing the calculated and experimental average flow rates of thermally softened polymer undergoing heat transfer compared favorably.



So, one of the studies being carried out independently by Leeuwen and Griskey and Wiehe, set the stage for a considerable number of experimental measurements. So, basically those studies developed the concept of using thermocouple parallel to axial flow and pointing upstream. This is the placement of the thermocouple for the temperature profile measurement. Now, in this particular table, there is a comparison of various thermocouple arrangements, like thermocouple configuration here you see that the thermocouple configuration is given and the dynamic response and thermal errors and mechanical stability is given.

Convection heat transfer in laminar flow of molten polymers

Comparison of various thermocouple arrangements

Thermocouple Configuration	Dynamic Response	Thermal Errors	Mechanical Stability
A	Poor	Poor	Poor
B	Poor	Improved over A	Poor
C	Poor	Poor	Poor
D	Poor	Poor	Improved over A,B,C
E	Poor	Poor	Adequate
F	Good	Good	Poor

**Toor, H.L. Ind. Eng. Chem. 48, 922 (1956).

So, let us take one example that is for A, where the dynamic response is very poor and thermal and mechanical stability is also very poor. Now, if you take the example of D, where this is the placement, the dynamic response and thermal errors, they are poor, but mechanical stability improved over the different A, B and C. So, this is the classical representation of different positioning. When we are talking about the heat transfer in the laminar flow of the molten polymers, so, the point covered in this particular table, it should be recognized that the configuration F, this this configuration being parallel to flow and pointed upstream, this will produce the least flow disturbance. So, also this configuration F would minimize the shear heating at the thermocouple.

So, number of investigations being carried out and they made temperature profile measurement of flowing molten or thermally softened polymer system, they used the configuration F. Let us take an example, a number of these studies were directed for measuring temperature in the exit of a screw extruder or within the plunger or injection molding machines. Now, these placements, the results are large temperature fluctuations anywhere from 10 to 30 degree Celsius and these are the number of possible reasons for this occurrence. So, first both temperature and velocity profile would undergo rearrangement, therefore, influencing the reading. Next, although the configuration F which we discussed earlier, this was used to the thermocouple holder was quite sizable, this could have caused the flow disturbances and considerable viscous heating either of which would have influenced the reading.

Convection heat transfer in laminar flow of molten polymers

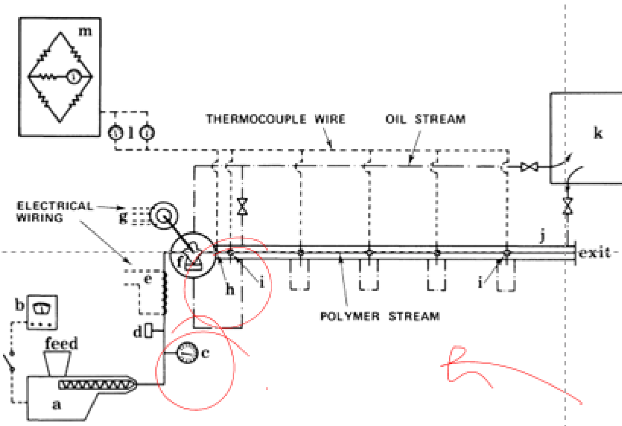


Fig: Device for measuring flowing polymer temperature profiles

a- Extruder
b- Control panel
c- Pressure gage
d- Safety rupture disc
e- Heating tape
f - Metering pump in oil jacket

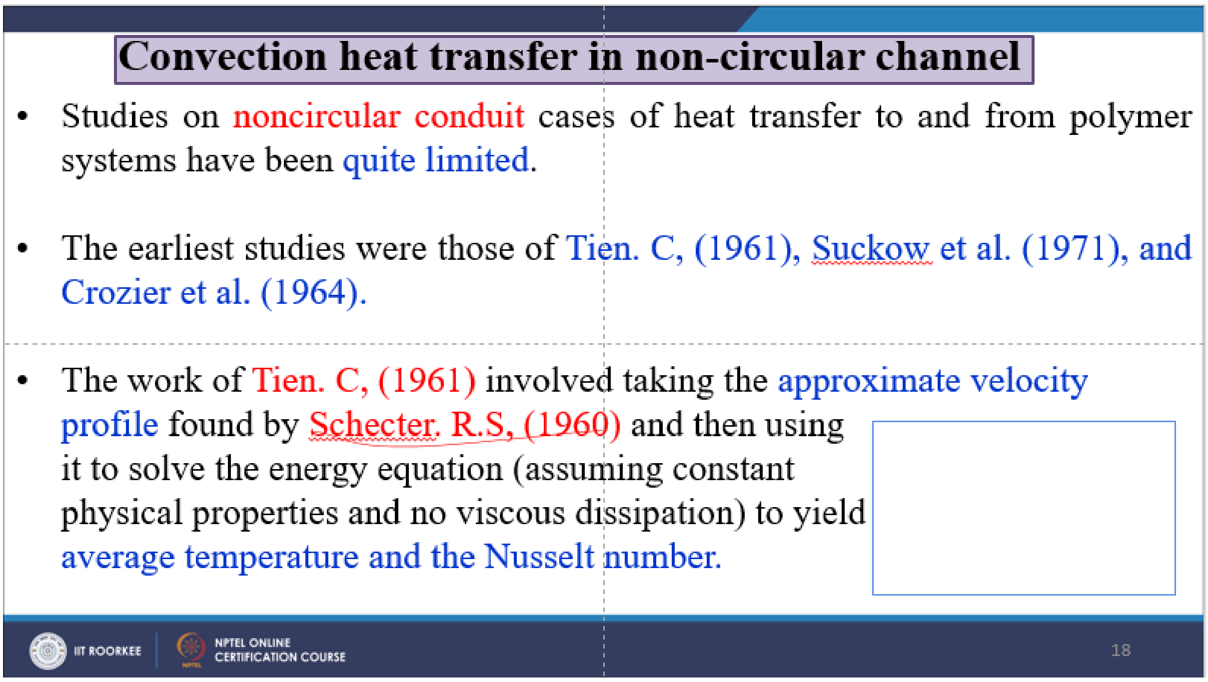
Now, here you see that this is the device for measuring the flowing polymer temperature profile. As usual, we are having the extruder, control panel, pressure gauge, safety rupture disk, heating tape and all these things and a metering pump in oil jacket. So, they perform experimentally to find out the things. Now, there are various device average inlet temperature thermocouples, multipro profile rings, heat exchanger sections, hot oil circulating units and thermocouple switches. So, this device is used to measure the temperature profile which is represented in this figure.

Convection heat transfer in laminar flow of molten polymers

- The device used to measure the temperature profiles is represented in figure previously.
- The thermocouples were iron-constantan, enclosed first in hypodermic tubing and then in a stainless steel sheath.
- Teflon® tubing was used as an insulation between the tubing and the sheath. This construction insured against heat-conduction errors.
- The outside diameter of each probe was 0.0625 in. This meant that the given thermocouple surface seen by the flowing polymer was only 0.4% of the total tube cross-sectional area.

The thermocouple they are they used, they are iron constantan, they enclosed first in the hypodermic tubing and then in a stainless-steel shell. Teflon tubing, they are used as an insulation between the tubing and the shell. The construction ensured against the heat conduction error. The outside diameter of each probe was 0.0625 inch and this means that the given thermocouple surface seen by the flowing polymer was only 0.

4 percent of the total tube cross sectional area. Now, studies on non-circular conduit cases for the convection heat transfer to and from the polymer system have been quite limited. This team, Sukov, they carried out the earlier studies and they involved taking in the approximate velocity profile which was found by the Schister and then using to solve the energy equation assuming constant physical properties and no viscous dissipation to yield the average temperature and the Nusselt number. Another study, Sukov and co-worker, they used to exact velocity distribution instead of approximate profile to solve the energy equation. Crossier and co-worker, they used the Leibniz technique to find appropriate equation for heat transfer.



Now, there are several equations developed by Crossier for the non-circular channels, the equation, various equations being developed by the Crossier in 1964. Now, the one equation that is H A d E over k is equal to 1 by 2 E square C p Z Z gamma B over gamma W to the power 0.14. This is equation number 6. And when the Grays number is less than 3, the Grays and for the Grays number greater than 20, then the equation becomes H A d E over k is equal to 1.

Convection heat transfer in non-circular channel

The equations developed by Crozier et al., (1964) were

Eq: $\frac{h_a D_e}{k} = \frac{1}{2}\,\frac{D_e^2 C_p G_z}{kL}\left(\frac{\gamma_B}{\gamma_W}\right)^{0.14}$ (6)

when the **Graetz number is less than 3.0**, and for Graetz number >20

Eq: $\frac{h_a D_e}{k} = 1.86\left(\frac{D_e^2 G_z C_p}{kL}\right)^{1/3}\left(\frac{2n+1}{3n}\right)^{1/3}\left(\frac{\gamma_B}{\gamma_W}\right)^{0.14}$

Where, h_a is the average heat transfer coefficient, D_e is the diameter, **k** is the thermal conductivity, γ_B is the fluid viscosity at bulk average temperature and γ_W is the viscosity at wall temperature

86 into d square Z Z C p over k L over 3 2 n plus 1 3 n to the power 1 by 3 gamma B over gamma W 0.14. Now, where this H A is the average heat transfer coefficient, d E is the diameter, k is the thermal conductivity, gamma B is the fluid viscosity at the bulk temperature and gamma W is the viscosity at wall temperature. Now, for the range of the H A, the H A is the average heat transfer coefficient. Now, 20 is greater than, I mean when the Graetz number is between 20 and 3, the work of Tien and Sukhoi et al, this can be used.

Convection heat transfer in non-circular channel

- **For the range 20 > Gz > 3**, the work of **Tien. C, (1961) or Suckow et al. (1971)** can be used.
- In the latter case, point temperature is given as a function of reduced length (for a power law fluid of n = 1/2):

$$\theta = 1.222(1-\lambda^3)e^{-2.787\Psi} + (0.226)(1-14\lambda^3+13\lambda^6)e^{-31.934\Psi} \quad \text{...... (eq-8)}$$

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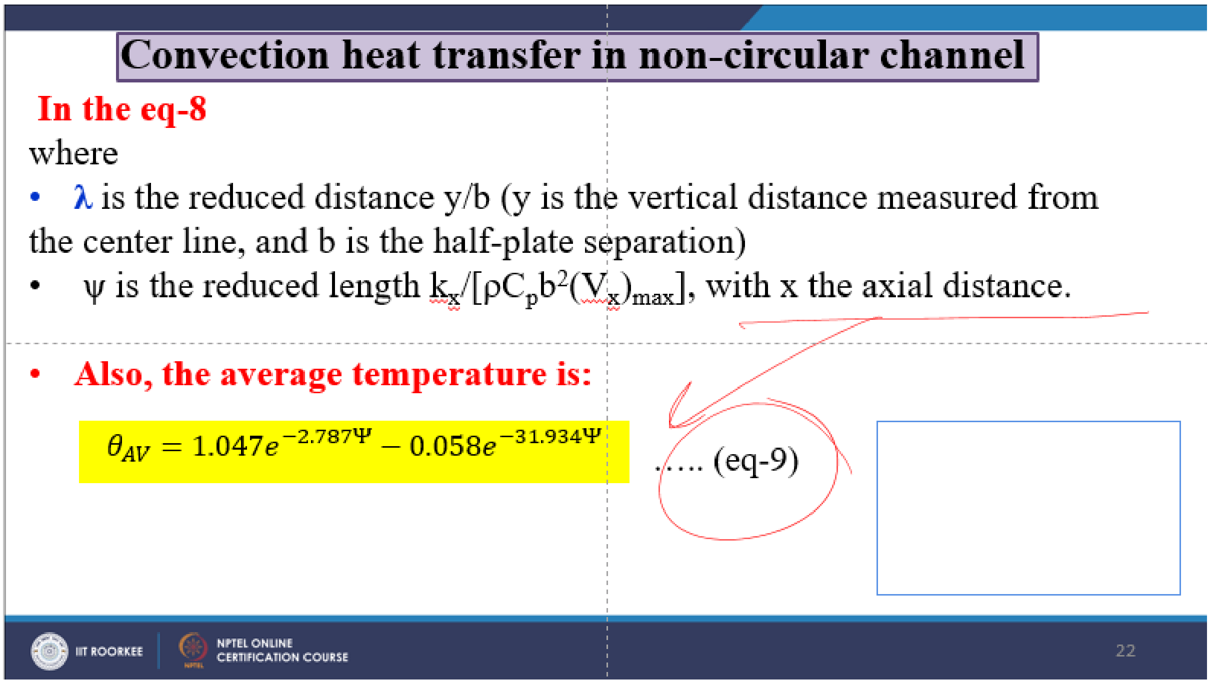
In the later case, the point temperature is given as a function of reduced length for the power law fluid H A is equal to 1 by 2, where this this particular equation can be utilized and this can be represented mathematically like this. Now, if we use the equation 8, where lambda is the reduced distance that Y B, Y is the vertical distance measured from the center line and B is the half plate separation and psi is the reduced length. So, k x over rho C p B square v x maximum with the x is the axial distance. So, the average number can be represented like equation number 9.



$$\theta_{AV} = 1.047e^{-2.787\Psi} - 0.058e^{-31.934\Psi}$$

The other work, some of the the researchers they performed considered the paddle plates or slits and all 3 studies which are listed here, they involve the theoretical solution of the energy equation that included the viscous dissipation, but neither temperature dependent properties nor compressibility cooling.

Convection heat transfer in non-circular channel

- According to **Kozicki et al., (1966)** work, a modified Reynolds number is given by :

Eq:

$$Re'' = \frac{4 R_h V \rho}{N_{eff}} (a+b)$$

10

where

R_H is the hydraulic radius for the conduit (cross-sectional area available for flow-wetted perimeter),

N_{eff} an effective viscosity, $\tau_{wall}/(2V/R_H)$;

V the average velocity,

ρ the density and a and b are the empirical constants.

Another co-worker of Lin and Hirst, they reported the theoretical solution for the the heat transfer with flow in an annulus with viscous dissipation and a moving inner cylinder. Now, they appear there appear to be no other heat transfer data available for flowing polymeric system in non-circular cross section. However, the situation can be rendered by using the geometrical parameter technique which was developed by Kozycki in 1966. As per the Kozycki in 1966, the work the modified Reynolds number is given by 4 r h v rho over n and this is equation number 10, where r h is the hydraulic radius for the conduit that is a cross sectional area available for flow weighted parameters and effective is the effective viscosity which is represented by this and v is the average velocity and rho is the density and a and b are the empirical constants. Now, in other segment, other co-workers, they reposted the Reynolds number value by using this equation that d equivalence v rho over an apparent and this is represented as equation number 11.

Convection heat transfer in non-circular channel

- Other author **Lin & Hsu, (1980)** reposted the Re value as:

$$Re = \frac{D_{eq}.V.\rho}{\eta_{APP}} \quad \text{..... (eq-11)}$$

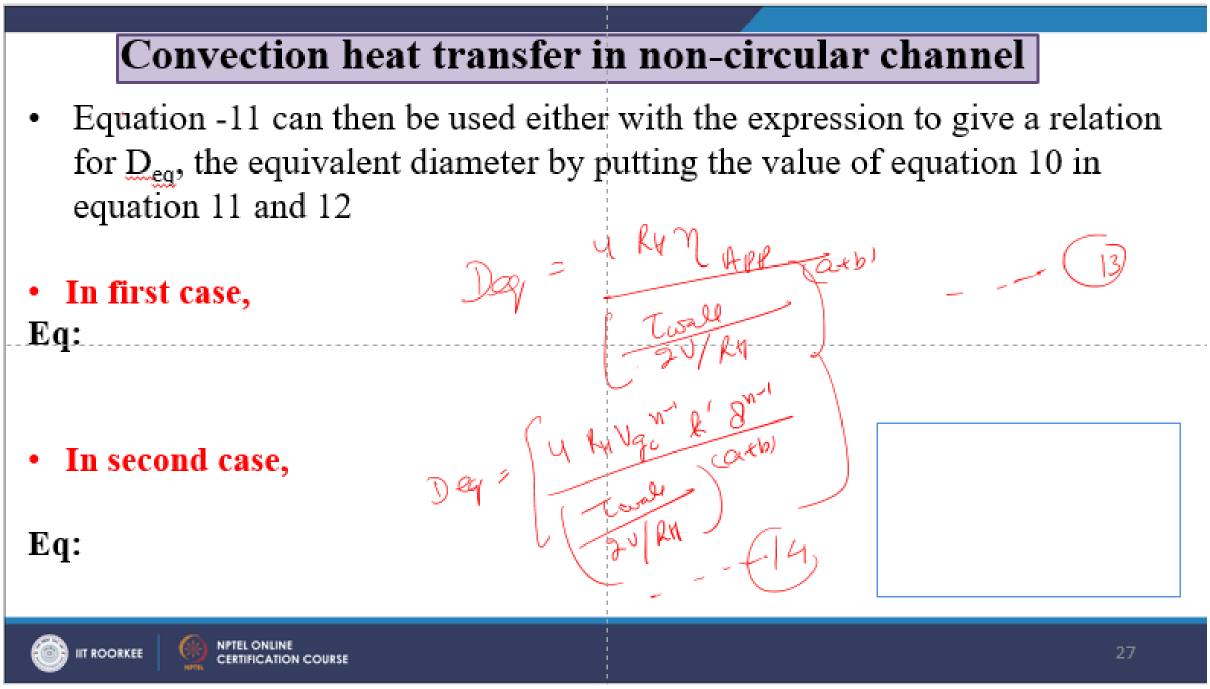
- **Dinh & Armstrong in 1982**, rewrite the eq 11 as

$$Re' = \frac{D_{eq}^{n'}.V^{2-n'}.\rho}{g_C.k'.8^{n-1}} \quad \text{....... (eq-12)}$$

Where, D_{eq} = equivalent diameter, V the average velocity, ρ the density, **n' , k' and n** are power law parameters.

$$\mathrm{Re} = \frac{D_{eq}.V.\rho}{\eta_{APP}}$$

$$\mathrm{Re}' = \frac{D_{eq}^{n\prime}.V^{2-n\prime}.\rho}{g_C.k'.8^{n-1}}$$



Another worker, they rewrite this particular equation or rearrange this equation in this particular form where d equivalent is represented as equivalent diameter, v is the average velocity, rho is the density, eta k and they are the power law parameters. Now, this equation 11 can be used either with the expression to give a relation of equivalent diameter and by putting the value of equation 10 in the equation 11 and 12. So, this can be represented as equivalent diameter 4 r h eta tau vol 2 v over n. So, this is the equivalent diameter 4 r h to the power a plus b. This is equation number 13 or equivalent is equal to 4 r h v to the power a plus b.

Slurry equation

$$\frac{hD}{k_{slurry}} = 0.027\left(\frac{D.V.\rho}{\mu_S}\right)^{0.8}\left(\frac{C_P.\mu_S}{k_{slurry}}\right)^{\frac{1}{3}}\left(\frac{\mu_L}{\mu_{LW}}\right)^{0.14}$$

Convection heat transfer in turbulent flow

- In general, **situations in which polymer turbulent flows** are encountered are **limited**.
- A slurry equation is:

$$\frac{hD}{k_{slurry}} = 0.027\left(\frac{D.V.\rho}{\mu_S}\right)^{0.8}\left(\frac{C_P.\mu_S}{k_{slurry}}\right)^{\frac{1}{3}}\left(\frac{\mu_L}{\mu_{LW}}\right)^{0.14} \quad \text{.....(eq-15)}$$

where $\mathbf{k_{slurry}}$ is the thermal conductivity of the slurry, $\boldsymbol{\mu_s}$ is the slurry viscosity, $\boldsymbol{\mu_L}$ **and** $\boldsymbol{\mu_{LW}}$ are viscosities of the suspending liquid in the bulk and at the wall.

Now, this is equation number 14. In general, situation in which the polymer turbulent flow are encountered, they are very limited. Usually, a slurry equation in the turbulent flow is important and this is represented as equation number 15 in this aspect here. The important factor is the k slurry. This is the thermal conductivity of the slurry and mu is the slurry viscosity mu l and mu l w.

Convection heat transfer in turbulent flow

- The slurry viscosity μ_S can be given by:

Eq:

$$\mu_S = \frac{\mu_L}{\left[1-\left(\frac{X_\mu}{X_{\mu b}}\right)\right]^{1.8}} \quad ---(16)$$

- Another slurry relationship is given by **Thomas, D.G., (1960)**

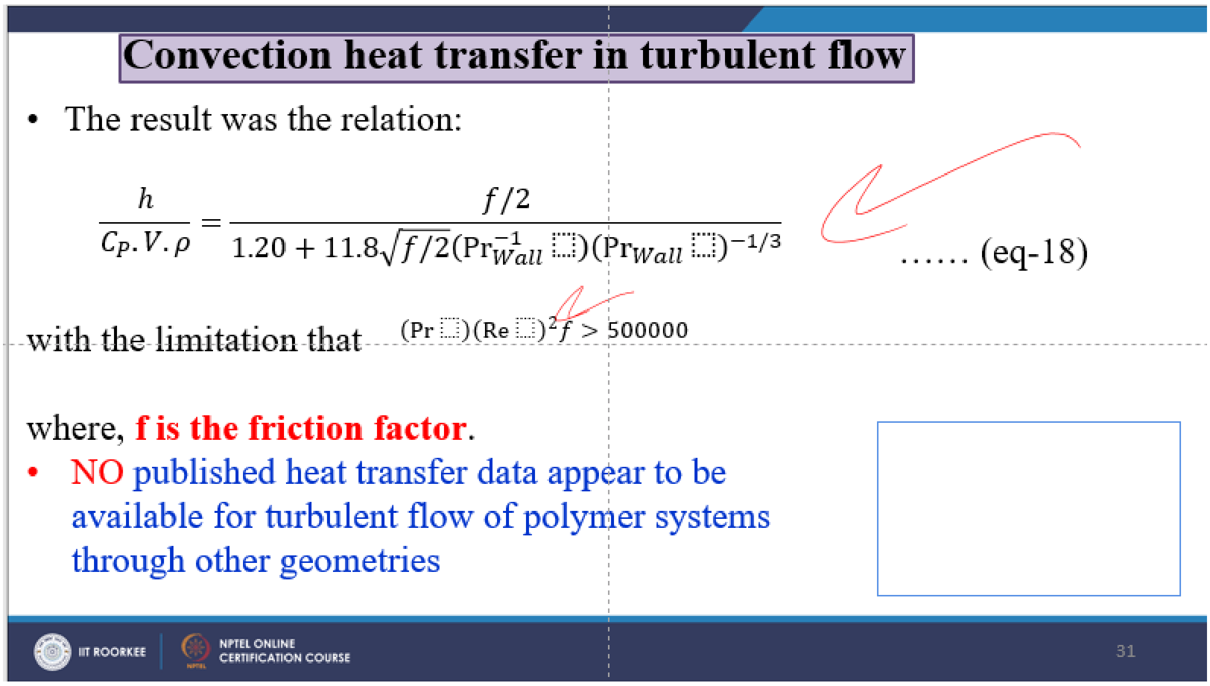
Eq:

$$\left(\frac{h}{C_p V}\right)\left(\frac{C_p \eta_{app}}{k_{slurry}}\right)^{2/3}\left(\frac{\eta_{apparent}}{\eta_{app}}\right)^{0.14} = 0.027\left(\frac{D V \rho}{\eta_{app}}\right)^{-0.2} \quad ---(17)$$

- The **general relation** recommended for **turbulent heat transfer to flowing non-Newtonians** is that developed by Metzner et al., (1958). This was done by analogy with momentum transport.

They are viscosities of the suspending liquid in the bulk and at the wall. So, for slurry viscosity mu s, this can be given as mu s is equal to mu l and mu l w over 1 minus X mu or X mu b to the power 1.8. This is our equation number 16. So, another relationship Thomas in 1960, they he gave another relationship which is like this H over C p v C p k slurry over 1 minus X mu b into 1 minus X mu b.

To the power 2 by 3 into wall over eta apparent to the power 0.14 and which is 0.027 d v rho eta apparent to the power 0.2.



Now, this is our equation number 17. Now, this is our equation number 17. Now, the general relation recommended for the turbulent heat transfer to flowing non-Newtonian is developed by Metzner in 1958 and this was done by the analogy with the momentum transfer. Now, this particular equation includes the constant heat flux, equality of E d thermal and momentum diffusivity at all radius, no dependence of fluid on time, no fluid elasticity, large Prandtl number, the assumption that heat flux is a function of tube radius. So, thus this result can be represented as equation number 18 with the limitation that Prandtl number into Reynolds number square f is greater than 5 lakhs, where f is the friction factor. And no published heat transfer data appear to be available for turbulent flow of polymeric system through other geometries.

$$\frac{h}{C_P.V.\rho} = \frac{f/2}{1.20 + 11.8\sqrt{f/2}(\mathrm{Pr}_{Wall}^{-1})(\mathrm{Pr}_{Wall})^{-1/3}}$$

Convection Examples

Question 1: A polymer solution (n of 0.5; K at 90°F of 51 lb_m s^{n-2} ft^{-1}; viscosity activation energy of 14,900 Btu/lb mole) is fed into a 1-in. i.d. stainless steel tube (10 ft long) at a mass flow rate of 750 lb_m/h and a temperature of 90°F. The velocity profile is fully developed before the solution enters the heated tube. Heat is supplied by steam condensing at 20 psia. Remaining fluid properties are: density = 58 lb_m/ft^3, specific heat = 0.6 Btu/ lbm. °F, thermal conductivity = 0.5 Btu/ft°F h. Calculate the exit temperature of the fluid.

The temperature of the condensing steam is 227.96°F. For purposes of calculation, we will take this as 228°F. The value of K_B = 11.182 lb_m 8^{2-n} ft^{-1}

K_w = 3.315 lb_m 8^{2-n} ft^{-1}

Now, let us take couple of examples in this particular first question a polymer solution of nita is 0.5 k is 90 degrees Fahrenheit and 51 pounds viscosity activation energy of 14900 B t u per pound mole is fed into a 1-inch internal diameter stainless steel tube of 10 feet long at mass flow rate of 750 pounds per hour and a temperature of 90 degree Fahrenheit. The velocity profile is fully developed before the solution enters the heated tube, heat is supplied by the steam condensing at 20 psia, remaining fluid provide properties are density which is obviously required which is given specific heat is given, thermal conductivity is also given. So, you need to calculate the exit temperature of the fluid, the temperature of the condensing steam is given as 227.96 Fahrenheit for the purpose of calculation.

$$h = \frac{0.5\ Btu/ft\,F\,h}{1/12\ ft}(1.75)(125)^{1/3}(90)^{1/3}\left[\frac{11.182}{3.315}\right]^{0.14} = 69.73\ \frac{Btu}{ft^2\cdot F\cdot h}$$

$$w\,C_p\,\Delta T_{fluid} = h_A\,(T_{wall} - T_{fluid})_{Av}$$

$$w\,c_p(T_2 - T_1) = h_A\left[T_{wall} - \left(\frac{T_2 - T_1}{2}\right)\right]$$

$$750\,\frac{lbm}{h}\left(\frac{0.6\ Btu}{lbm\,F}\right)(T_2 - 90°F) = \left(69.73\,\frac{\pi}{12}\times 10\right)ft^2\left[220 - \left(\frac{T_2 + 90}{2}\right)\right]$$

$$T_2 = 137°C$$

By using equation

$$Nu = 1.75\left(\frac{3n' + 1}{4n'}\right)^{\frac{1}{3}} (Gz)^{\frac{1}{3}} \left(\frac{K_b'}{K_W'}\right)^{0.14}$$

For given data

$$\left[\frac{3n+1}{4n}\right] = \left[\frac{2.5}{2.0}\right] = 1.25Gz = \frac{W.C_P}{k.L} = \frac{(750lb_m/h)(0.6Btu/lb_m{}^{\circ}F)}{(0.5Btu/ft^{\circ}F.h)(10ft)} = 90$$

The value of K_B = 11.182 $Ib_m\ 8^{2-n}\ ft^{-1}$

K_w = 3.315 $Ib_m\ 8^{2-n}\ ft^{-1}$

Using,

$$h = \frac{k}{D} 1.75\left(\frac{3n+1}{4n}\right)^{1/3} (Gz)^{1/3} \left(\frac{K_B}{K_W}\right)^{0.14}$$

putting all the values in the equation, we get

$$h = \frac{0.5Btu/ft^{\circ}F.h}{1/12ft}(1.75)(125)^{1/3}(90)^{1/3}\left[\frac{11.182}{3.315}\right]^{0.14} = 69.73\frac{Btu}{ft^2{}^{\circ}F.h}$$

Next by the enthalpy balance

$$W.C_P.\Delta T_{fluid} = h_A\left(T_{Wall} - T_{fluid}\right)_{Av}$$

$$W.C_{P.}(T_2 - T_1) = h_A\left[T_{Wall} - \left(\frac{T_2 - T_1}{2}\right)\right]$$

$$750\frac{lbm}{h}\left(\frac{0.6Btu}{lbm^{\circ}F}\right)(T_2 - 90^{\circ}F) = \left(69.73\frac{\pi}{12}\times 10\right)ft^2\left[228 - \left(\frac{T_2 + 90}{2}\right)\right] T_2 = 137^{\circ}C$$

Now, I will take is at 228 degrees Fahrenheit and the value of k B is given 11.182 pounds and k W is also given. So, putting all values in the equation, so we get H is equal to 0.5 B upon 1 by 12 into 1.75, 125 to the power 1 by 3 into 90 into 11.

182, 3.315, 0.14 which is equal to 69.73 B t u. And the value of k W is given by 0.5 into 10 to the power 1 by 3. Now, if we take the enthalpy balance which is W C p delta t, then it comes out to be this mathematical representation t wall minus t fluid which is average. So, W C p t 2 minus t 1 which is equal to H A t wall minus t 2 minus t 1 over 2.

So, this comes out to be 750 over into 0.6 into t 2 minus 90 Fahrenheit and this comes out to be 69.73 pi 12 into 10. So, this is the equation for the enthalpy balance. And this calculated value of t 2 is 137 degree Celsius and this is our answer. Now, let us take another example of flowing polyethylene melt which is maintained at 300 gram per minute is being heated in a rectangular duct which is having the dimension of 1 inch into 2 inches.

Convection Examples

Problem 2: A flowing polyethylene melt (300 g/min) is being heated in a rectangular duct (1 in. X 2 in.). Initial polymer temperature is 410°F. The exit temperature is 425°F. Determine the wall temperature needed to accomplish this for a 10-ft length of duct.

Given: The values of a and b are 0.2440 and 0.7276; W=500(g/min); C_P=0.68 (Btu/lbm°F); k=0.148 (Btu/ft.h.°F); τ_{wall}=2.9 x 10^{-2} lb_f/in^2; R_H= (1/36) ft; η_{app}=0.725 lb_f/in^2.s; V=5.27 x -10^{-4} ft/s

The Nusselt number for this case is $Nu = \frac{h.D_e}{k} = 10$

$$D_e = \frac{4.R_H.\eta_{App}}{\left(\frac{\tau_{Wall}}{2V/R_H}\right)} = \frac{4(1/36ft)\left(0.725lb_f/in^2.s\right)}{\left[\frac{2.9\times10^{-2}lb_f/in^2}{(2)(5.27\times10^{-4}ft/s)(36ft)}\right]^{0.9716}} = 0.1045ft$$

$$h = \frac{k}{D_e}.Nu = \frac{(0.148/Btu/h.ft.°F)}{0.1045ft}.10 = 14.18Btu/h.ft^2.°F$$

$$w.C_P(T_2 - T_1) = h_A\left[T_{Wall} - \left(\frac{T_1 + T_2}{2}\right)\right]$$

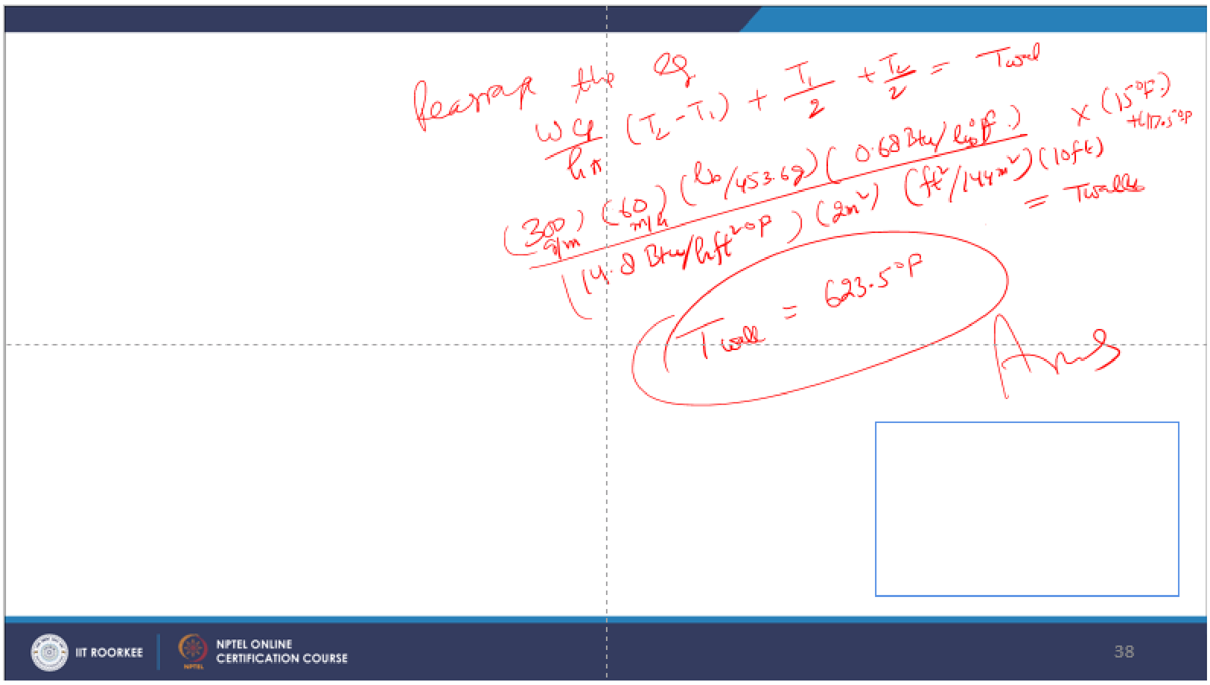
On rearranging above equations

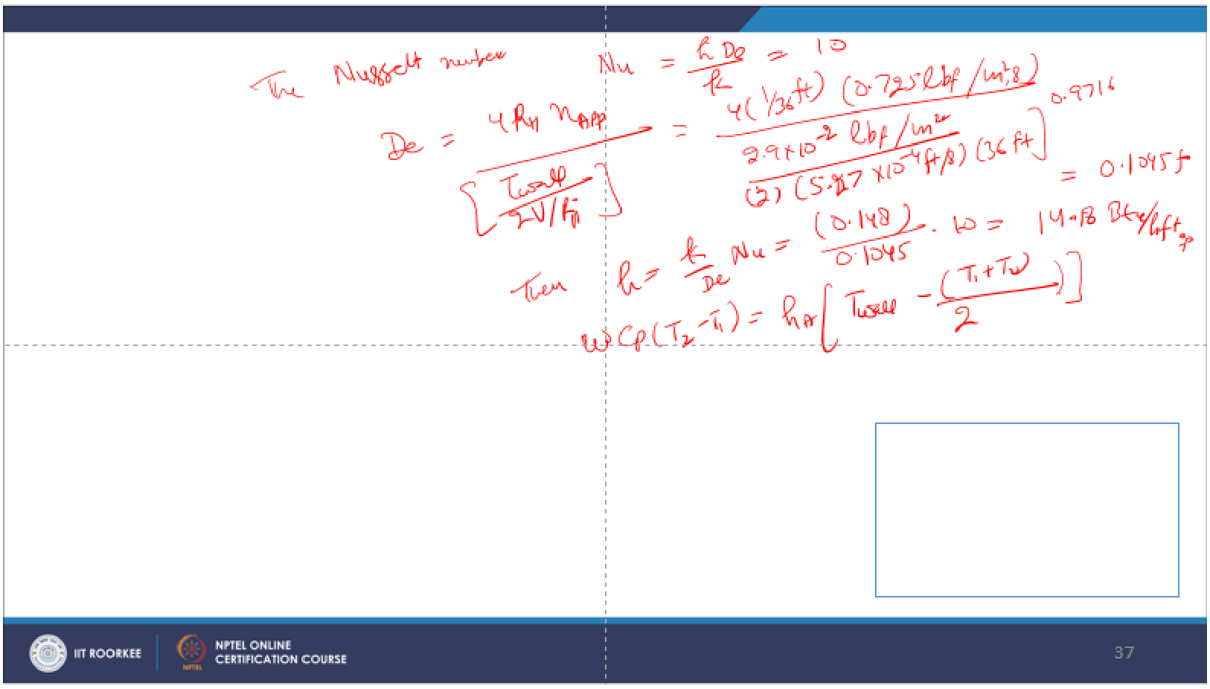
$$\frac{w.C_P}{h_A}(T_2 - T_1) + \frac{T_1}{2} + \frac{T_2}{2} = T_{Wall}$$

On substituting the values in the equations, we will get

$$\frac{(300g/min)(60\,\text{min}/h)(lb_m/453.6g)(0.68Btu/lb_m°F)}{(14.8Btu/h.ft^2.°F)(2m^2)(ft^2/144m^2)(10ft)}(15°F) + 417.5°F = T_{Wall}$$

Twall = 623.5oF (answer)





Now, initial polymer temperature is 410 Fahrenheit and the exit temperature is 425. Determine the wall temperature needed to accomplish this for 10 feet length of duct. It is given that the value of A and B which is required it is the value of A and B is given, the C p value is given, W value is given, k is given, tau value and R h and eta apparent and V is also given. So, let us try to solve this particular problem. Now, the Nusselt number for this case is given by this particular mathematical representation which is equal to 10.

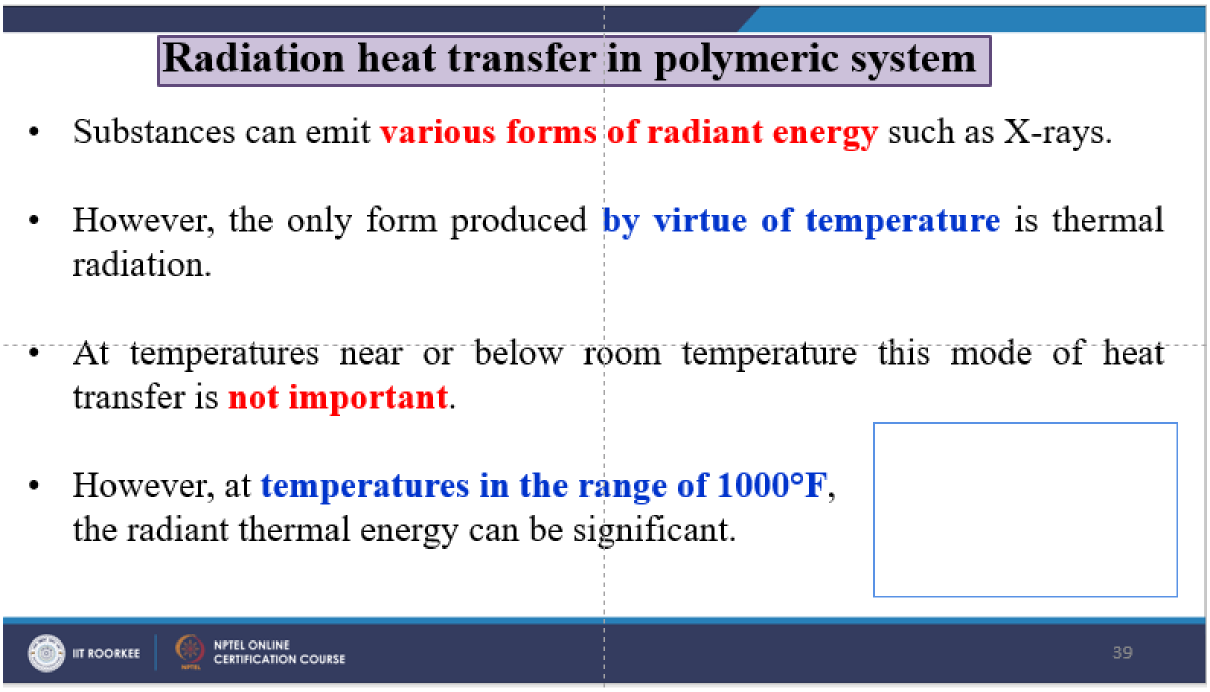
Now, this V is equal to 4 h over 10. Now, this is equal to 4 h over 10. Now, this is equal to 4 h over 2 V over R h which is equal to 4 into 1 by 36 0.725 over 2.9 into 10 to the power minus 2 over 10 to the power minus 2.

Now, this is equal to 4 h over 10. Now, this is equal to 2 into 5.27 into 10 to the power minus 4 feet per second into 36 feet and that whole to the power 0.9716, this comes out to be 0.1045 feet. Then h is equal to k over d e Nusselt number, this is coming out to be 0.

148 over 10 to the power 0.1045 into 10 which is comes out to be 14.18 B t u over h f t degree Fahrenheit. So, once again W C p t 2 minus t 1 is equal to h A t vol minus t 1 plus t 2 minus t 2 over 2. So, if we rearrange the equation, we get W C p over h A into t 2 minus t 1 plus t 1 over 2 plus t 2 over 2 is equal to t vol. If we substitute the values in the equation, we get 300 into 60 into 453.

6 grams into 0.68 over this is in gram per minute and minute per hour. So, this is equal to 0.68 into 10 then 14.8 B t u over h f t square which we have calculated then 2-meter square which is given f t square over 144-meter square into 10 f t multiplied by 15 degrees Fahrenheit plus 417.

5 degrees Fahrenheit. So, this is equal to 0.68 into 10 into 453.6 gram into which is equal to t vol. So, if if we solve this, we found it that temperature of the vol is 623.



5 degrees Fahrenheit. This is our answer. Now let us talk about the radiation heat transfer in the polymeric system. Now the substance can emit the various form of radiant energy such as x rays, gamma rays, all these things, the only form produced by virtue of the temperature is the thermal radiation. At temperature near or below room temperature, this mode of heat transfer is not at all important. So, at a temperature in the range of say 1000 degrees Fahrenheit, the radiant thermal energy can be significant. So, if the thermal radiation is conceived as a photon gas, it is possible from the thermodynamic to show that the energy density of the radiation is E b is equal to small theta t to the power 4.

Radiation heat transfer in polymeric system

- If thermal radiation is conceived of as a **"photon gas,"** it is possible from thermodynamics to show that the energy density of the radiation is

$$E_b = \sigma . T^4$$

where

J is the Stefan-Boltzmann constant (0.1714 x10^{-8} Btu/h.ft^2°F or 1.36 x10^{-12} cal/cm^{2}s°C),

T is absolute temperature,

E_b is the energy emitted from a blackbody (one in which absorptivity and emissivity are unity).

The J is the Stefan Boltzmann constant, t is the absolute temperature, E b is the energy emitted from a blackbody, one in which the absorptivity and emissivity are unity. So, the blackbody represents the idealization that only be closely approximated experimentally for any other body that is termed as a gray body, the emissivity epsilon which is represented as epsilon is equal to E over E b or the ratio of intensity emitted by a body relative to blackbody. So, for all bodies in the radiation equilibrium alpha is equal to epsilon where alpha is the fraction of incident radiation absorbed. A very important aspect of radiative heat transfer is the system geometry. This is accounted for by using the radiation shape factor and also called the view factor, angle factor or configuration factor and defined as say F 1 2 that is a fraction of energy leaving the surface 1 and reaches to surface 2 and F 2 1 that is a fraction of energy leaving the surface 2 and reaches the surface 1.

Radiation heat transfer in polymeric system

- The **blackbody** represents an idealization that can only be closely approximated experimentally. For any other body (termed a **gray body**), the **emissivity ε** is

$$\varepsilon = E/E_b$$

- or **the ratio of intensity emitted by the body relative to a blackbody**

$E_b = \sigma . T^4$ $\varepsilon = E/E_b$

Radiation heat transfer in polymeric system

- For all bodies in radiation equilibrium, $\boxed{\alpha' = \varepsilon}$ **the α' is that fraction of incident radiation absorbed.**

1 → 2

- A very important aspect of radiative heat transfer is the **system geometry**. This is accounted for by using **radiation shape factors**, also called **view factors, angle factors, or configuration factors** and defined as follows

F12= fraction of energy leaving surface 1 that reaches surface 2
F21= fraction of energy leaving surface 2 that reaches surface 1

So, ultimately it can be shown by this particular mathematical relation where phi 1 and phi 2 are the angle between the a 1 and a 2 respectively and the normal and r is the length of the ray joining a and a 2 that is the 2 bodies. Now the radiation shape factor this can be computed for the different situation and then used with the equation. So, it can be represented as q net is equal to a 1 F 1 2 E b 1 minus E b 2 which is equal to a 2 F 2 1 E b 1 minus E b 2. So, q net has become a 1 F 1 2 E b 1 1 minus E b 2 E b 1 which is equal to then a 1 F 1 2 E b 2. So, q net is equal to a 1 F 1 2 t 1 to the power 4 1 minus t 2 over t 1 to the power 4.

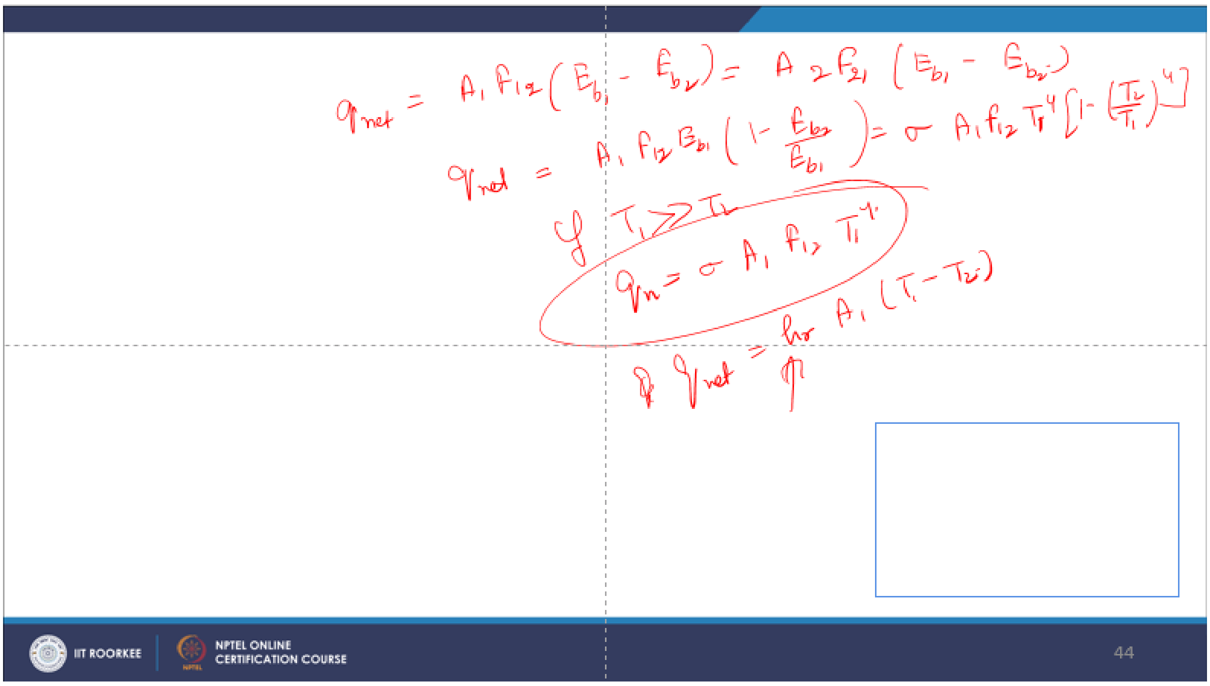
Radiation heat transfer in polymeric system

- Ultimately, it can be shown that

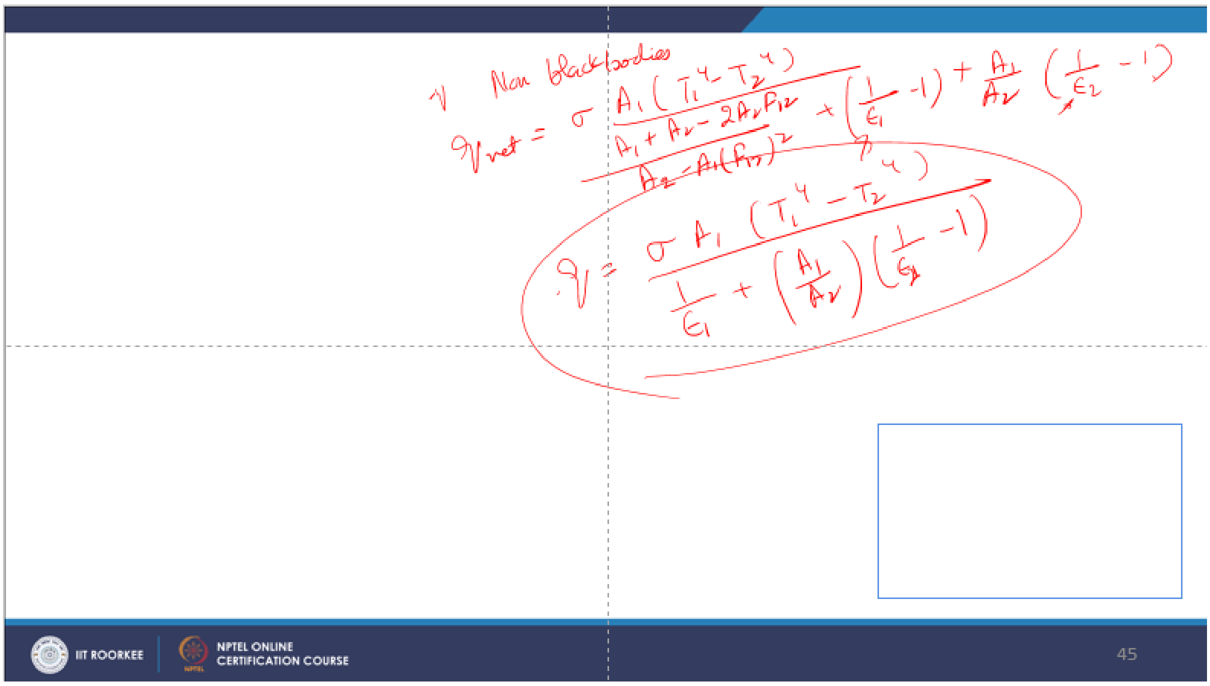
$$A_1.F_{12} = A_2.F_{21} = \int A_1 \int A_2 Cos\varphi_1.Cos\varphi_2 \frac{dA_1.dA_2}{\Pi.r^2}$$

where ϕ_1 and ϕ_2 are angles between A_1 and A_2, respectively, and the normal, and r is the length of the ray joining A_1 and A_2.

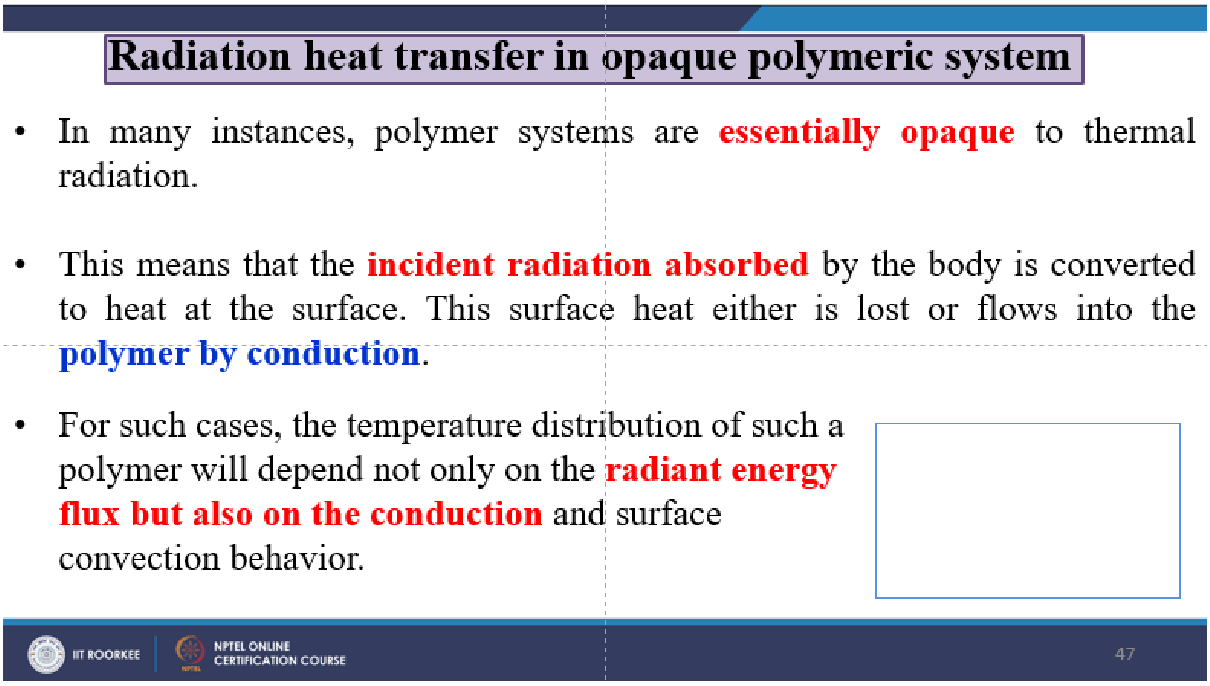
$$A_1.F_{12} = A_2.F_{21} = \int A_1 \int A_2 Cos\varphi_1.Cos\varphi_2 \frac{dA_1.dA_2}{\Pi.r^2}$$



Now if t 1 is greater than t 2 then q net is equal to small sigma a 1 F 1 2 t 1 to the power 4. Also it is possible to write the q net expression in the form of q net is equal to H r a 1 t 1 minus t 2 where H r is the radiative heat transfer coefficient. Now for non-black bodies connected non-black bodies connected by third surface which is not exchanging the heat then q net is equal to small sigma a 1 t 1 to the power 4 minus t 2 to the power 4 over a 1 plus a 2 minus 2 a 2 F 1 2 over a 2 minus a 1 F 1 2 square plus 1 over epsilon 1 minus 1 plus a 1 over a 2 into epsilon 2 minus 1. Where epsilon 1 and epsilon 2 are the emissivity of surface 1 and 2 respectively. For a special case of infinite parallel planes q is equal to small sigma a 1 t 1 to the power 4 minus t 2 to the power 4 over 1 epsilon 1 plus a 1 over a 2 F 1 2 over a 2 F 1 2.



So, this is equal to 1 over epsilon 2 minus 1. Now, let us talk about the radiation heat transfer in opaque polymeric system. In many instances, polymer systems are essentially opaque to thermal radiation. This means that incident radiation absorbed by the body is converted to heat at the surface. Now this surface heat either is lost or flows into the polymer by conduction. So, for such cases the temperature distribution of such a polymer will depend not only on the radiant energy plus but also on the conduction and surface conduction behaviour.



Now, some of the co-workers they have considered this case for slab constant initial temperature Ti. For such cases, the temperature distribution within the slab can be given by this particular equation where a is the half thickness of the slab, t is the time, x is the distance from the center line of the slab and q is the heat flux across the surface. Now this particular equation can be rewritten for considering the centre line temperature and the surface temperature expression. Now, if the ratio of this t s minus t i to t c minus t i is taken as i the uniformity index then i can be represented like this and with a certain modification can be represented like this. So, therefore, the uniformity index i can be represented as i is equal to phi alpha t over a square.

$$(T - T_i) = \frac{q_s . a}{k}\left[\left(\frac{\alpha t}{a^2}\right) + \left(\frac{3\alpha^2 - a^2}{6a^2}\right) - \frac{2}{\pi^2}\sum_{n=1}^{\infty}\frac{(-1)}{n^2}\, e\,\frac{-n^2\pi^2\alpha\theta}{a^2}\cos\left(\frac{n\pi\alpha}{a}\right)\right]$$

Radiation heat transfer in opaque polymeric system

- **McKelvey, J.M. (1965)** has considered this case for a slab at constant initial temperature T_i. For such a case, the temperature distribution within the slab is

$$(T - T_i) = \frac{q_s . a}{k}\left[\left(\frac{\alpha t}{a^2}\right) + \left(\frac{3\alpha^2 - a^2}{6a^2}\right) - \frac{2}{\pi^2}\sum_{n=1}^{\infty}\frac{(-1)}{n^2} e^{\frac{-n^2\pi^2\alpha\theta}{a^2}} \cos\left(\frac{n\pi\alpha}{a}\right)\right]$$

where

a is the half thickness of the slab,
t is time,
x is the distance from the centerline of the slab,
q_s is the heat flux across the surface.

Radiation heat transfer in opaque polymeric system

- If this equation is rewritten for T_c (centerline temperature) and T_s (the surface temperature), expressions for $(T_c - T_i)$ and $(T_s - T_i)$ are obtained.
- If the ratio of $(T_s - T_i)$ to $(T_c - T_i)$ is taken as I, a uniformity index, then

$$I = \frac{(T_S - T_i)}{(T_C - T_i)}$$

$$I = \frac{\frac{\alpha t}{a_2} + \frac{1}{3} - \frac{2}{\pi^2}\sum_{n=1}^{\infty}\frac{1}{n^2}\exp\frac{-n^2\pi^2\alpha\theta}{a^2}}{\frac{\alpha t}{a^2} - \frac{1}{6} - \frac{2}{\pi}\sum_{n=1}^{\infty}\frac{(-1)^n}{n^2}\exp \square \frac{-n^2\pi^2\alpha\theta}{a^2}}$$

$$I = \frac{(T_S - T_i)}{(T_C - T_i)} I = \frac{\frac{\alpha t}{a_2} + \frac{1}{3} - \frac{2}{\pi^2}\sum_{n=1}^{\infty}\frac{1}{n^2}\exp\frac{-n^2\pi^2\alpha\theta}{a^2}}{\frac{\alpha t}{a^2} - \frac{1}{6} - \frac{2}{\pi}\sum_{n=1}^{\infty}\frac{(-1)^n}{n^2}\exp \quad \frac{-n^2\pi^2\alpha\theta}{a^2}}$$

$$I = \varphi\left(\frac{\alpha t}{a^2}\right)$$

Radiation heat transfer in opaque polymeric system

- Therefore, uniformity index (I) can be written as

$$I = \varphi\left(\frac{\alpha t}{a^2}\right)$$

- The relation of I to $\alpha t/a^2$ can be used to set the slab dimensions or required time of heating for a given energy interchange situation

This relation of i to alpha t over a square can be used to set the slab dimension or required time of heating for a given energy interchange solution. Now, let us take another example of radiation. Now a 3 feet wide strip of an opaque polymer which is 0.02-inch-thick is being heated continuously between the banks of an infrared heater at 1500 Fahrenheit each heater bank has an area of 0.

Radiation Examples

Question: A 3-ft-wide strip of an opaque polymer (0.02 in. thick) is being heated continuously between banks of infrared heaters at 1500°F; each heater bank has an area of 0.08 ft. Assuming that heat is transferred entirely by radiation, calculate the net gain in enthalpy of the polymer if the rate remains 25 ft/min: Take polymer density as 80 lb/ft³. Given, σ=0.1714 x 10-8 Btu/h.ft2ºR, ε=0.90

$$q_{net} = \sigma . A_{heater} . \varepsilon_{heater} . T_{heater}^4$$

$$q_{net} = (0.1714 \times 10^{-8} Btu/h.ft^2 {}^\circ R)(2)(0.08 ft^2)(0.90)(1960^\circ R)^4 = 3650 Btu/h$$

$$q_{net} = \sigma A_{heater} \epsilon_{heater} T_{heater}^4$$

$$q_{net} = (0.1714 \times 10^{-8}\ Btu/h\,ft^2\,{}^\circ R^4)(2)(0.08\ ft^2)(0.90)(1960\ {}^\circ R)^4$$

$$= 3650\ Btu/h$$

08. If assuming that the heat is transferred entirely by radiation, calculate the net gain in enthalpy of the polymer if the rate remains 25 feet per minute and we can take the polymer density as 80 pounds per cubic feet and it is given that small theta is equal to 0.1714 into 10 to the power minus 8 and epsilon is given as the emissivity is given as 0.9. Now, this particular equation can be used with the same expression for the temperature with the additional stipulations that the radiant energy absorption by the heating element is negligible, and that the temperature of the heating element is much greater than the temperature of the polymer, then this can be written as like this.

Mass rate of polymer

$$= (25\ ft/m)(60\ min/h)(3\ ft)\left(\frac{0.02}{12}\ ft\right)(80\ lb/ft^3)$$

$$= 600\ lb_m/h$$

$$= \frac{3650\ Btu/h}{600\ lb/h} = 6.25\ Btu/lb_m$$

Mass rate of polymer

$$= (25ft/min)(60\,\text{min}/\,h)(3ft)\left(\frac{0.02}{12}ft\right)(80lb_m/ft^3) = 600lb_m/h$$

Enthalpy rise

$$= \frac{3650Btu/h}{600lb_m/h} = 6.25Btu/lb_m$$

Now, if you substitute the value, then this can be 0.1714 into 10 to the power minus 8 Btu per meter cube. Into 2 into 0.08 0.90, that is the emissivity 1960 and which is comes out to be 3650 Btu per hour.

Then the mass rate of the polymer is given as 0.714 into 10 to the power minus, which is given as 25 feet per minute into 60 minutes per hour 3 feet 0.0212 feet 80 pounds per cubic feet which is equal to 600 pounds per hour, and the enthalpy rise is equal to is 3650 Btu per hour over 600 pound per hour which is coming out to be 6.25 Btu. This is the required answer.

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So, dear friends, in this particular segment, we discussed the various aspects of conductive heat transfer, we discussed convective heat transfer among the polymeric system, we discussed radiative heat transfer, we discussed various correlations, and various theories, and all the aspects are being covered over here.

So, for your convenience, we have enlisted various references you can use as per your requirements. Thank you very much.