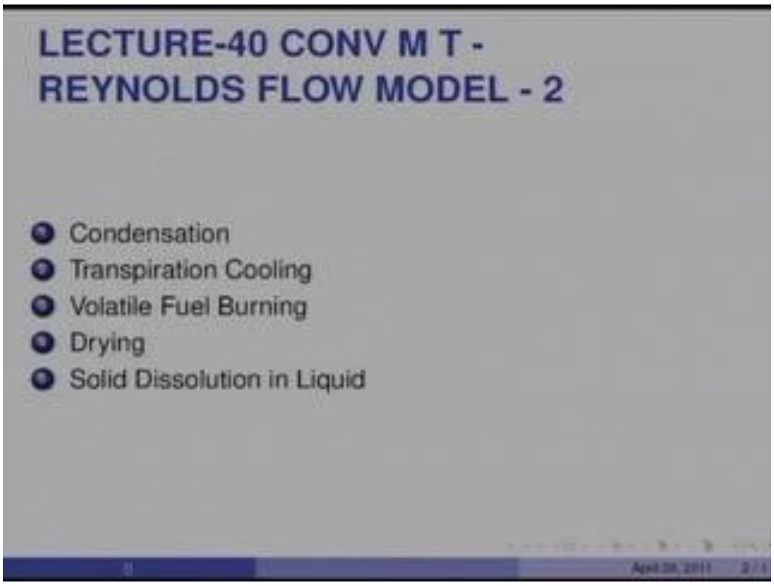
Convective Heat and Mass Transfer

Prof. A.W. Date
Department of Mechanical Engineering
Indian Institute of Technology, Bombay

Lecture No. # 40

CONV M T Reynolds Flow Model- 2

(Refer Slide Time: 00:41)



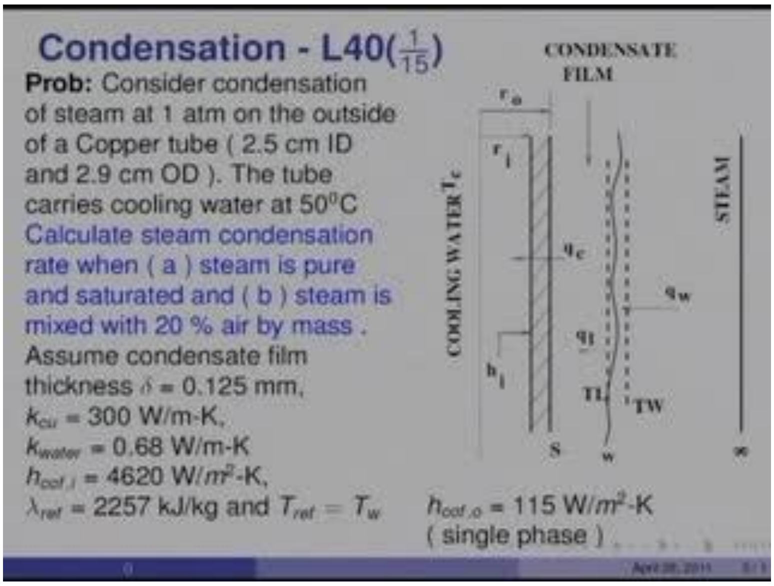
LECTURE-40 CONV M T - REYNOLDS FLOW MODEL - 2

- Condensation
- Transpiration Cooling
- Volatile Fuel Burning
- Drying
- Solid Dissolution in Liquid

In today’s lecture we shall take consider further problems in which, we shall apply Reynolds flow model for predicting mass transfer rates; and the new problems are condensation, transpiration cooling, volatile fuel burning, drying, and dissolution of solid in a liquid.

These are the vast range of problems in which Reynolds flow model can be applied, and I have selected a few, so as to help you understand how easily one can estimate the mass transfer rates, and arrive at very crucial engineering decisions in such problems. So, let us take the first problem of condensation.

(Refer Slide Time: 01:20)



Condensation - L40($\frac{1}{15}$)

Prob: Consider condensation of steam at 1 atm on the outside of a Copper tube (2.5 cm ID and 2.9 cm OD). The tube carries cooling water at 50°C Calculate steam condensation rate when (a) steam is pure and saturated and (b) steam is mixed with 20 % air by mass . Assume condensate film thickness $\delta = 0.125$ mm, $k_{cu} = 300$ W/m-K, $k_{water} = 0.68$ W/m-K, $h_{cof,i} = 4620$ W/m^2-K, $\lambda_{ref} = 2257$ kJ/kg and $T_{ref} = T_w$

$h_{cof,o} = 115$ W/m^2-K (single phase)

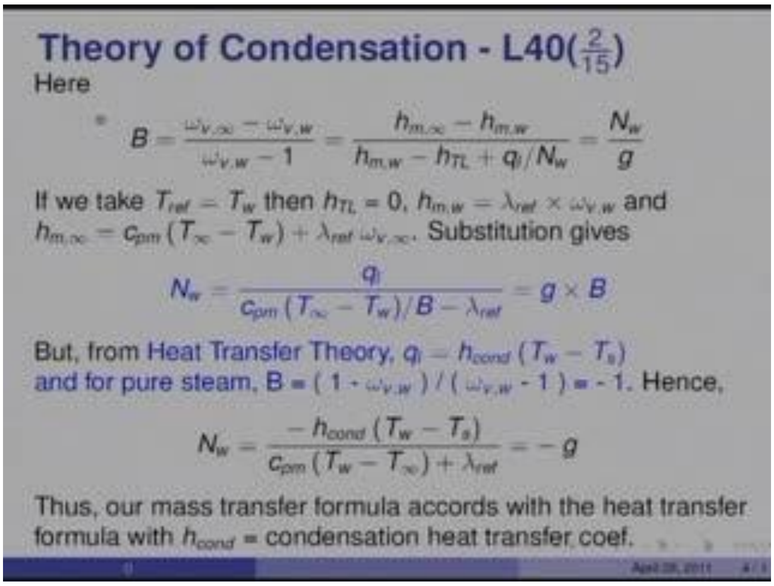
So, the problem statement is as follows; consider condensation of steam at one atmosphere on the outside of a copper tube, so here is a copper tube, this is the symmetry line of the tube of radius r I, and the wall thickness is r o minus r i through which cooling water is flowing and the steam is on the outside of the tube; the tube is made of copper, its diameter is 2.5 centimeter i d, and 2.9 centimeter o d, the tube carries cooling water at fifty degree centigrade.

Calculate the steam condensation rate on this tube when steam is pure and saturated, and b steam is mixed with twenty percent air by mass; when steam is mixed with air as you know the condensation rate reduces, and we was to estimate how much is the reduction. Further data given are assumed that condensate film thickness on the tube is .125 millimeters; the conductivity of copper is 300 watts per meter kelvin that of water is 0.68 watts per meter kelvin.

The heat transfer coefficient on the tube side h cof i is 4620 watts per meter square kelvin, we are going to assume t ref equal to t w where lambda ref the latent heat is 2257 kilo joules per kilo gram. Now, you are also given another information that if there was single phase flow of air in this case, then the h coefficient without mass transfer would be 115 watts per meter square kelvin.

So, this is the value we shall use to estimate g star; now, remember you can obviously solve this problem from heat transfer theory, and you are always used heat transfer theory for solving problems of this kind, but we are going to treat this as a mass transfer problem; so, my first task would be to show equivalence between Reynolds flow model, and the heat transfer theory of condensation and that is what i intended to do on the next slide.

(Refer Slide Time: 03:45)



Theory of Condensation - L40($\frac{2}{15}$)

Here

$$B = \frac{\omega_{v,\infty} - \omega_{v,w}}{\omega_{v,w} - 1} = \frac{h_{m,\infty} - h_{m,w}}{h_{m,w} - h_{TL} + q_l/N_w} = \frac{N_w}{g}$$

If we take $T_{ref} = T_w$ then $h_{TL} = 0$, $h_{m,w} = \lambda_{ref} \times \omega_{v,w}$ and $h_{m,\infty} = c_{pm}\,(T_\infty - T_w) + \lambda_{ref}\,\omega_{v,\infty}$. Substitution gives

$$N_w = \frac{q_l}{c_{pm}\,(T_\infty - T_w)/B - \lambda_{ref}} = g \times B$$

But, from Heat Transfer Theory, $q_l = h_{cond}\,(T_w - T_s)$ and for pure steam, B = (1 - $\omega_{v,w}$) / ($\omega_{v,w}$ - 1) = - 1. Hence,

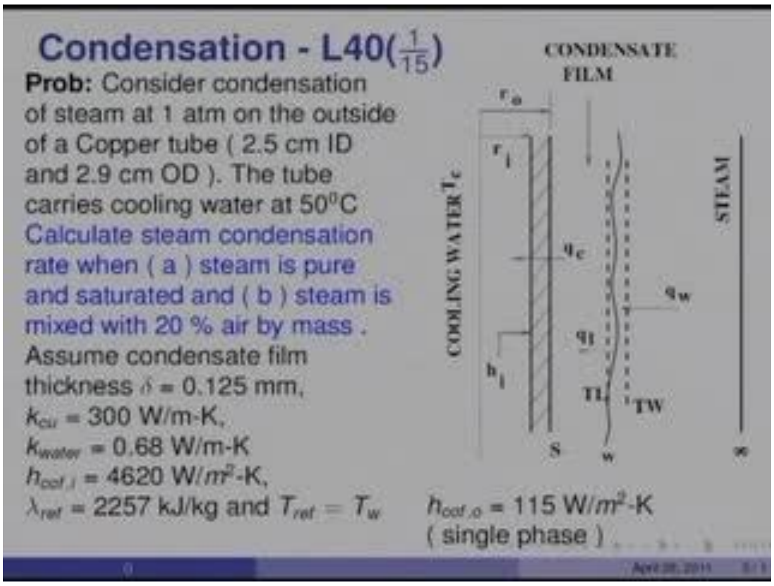
$$N_w = \frac{-\,h_{cond}\,(T_w - T_s)}{c_{pm}\,(T_w - T_\infty) + \lambda_{ref}} = -\,g$$

Thus, our mass transfer formula accords with the heat transfer formula with h_{cond} = condensation heat transfer coef.

So, in our case b would be omega v infinity minus omega v w divided by omega v w minus 1 and that would equal h m infinity minus h m w h m w minus h t l plus q l by n w and that would equal n w by g. Now, as I said we are going to take t ref equal to t w, then h t l will be of course 0, and h m w will be lambda ref into omega v w, there will be no sensible heat contribution to h m w, because t ref is equal to t w, but h m infinity would equate to c p m that is the mixture specific heat multiplied by t infinity minus t w plus lambda ref into omega v infinity; so, if we substitute for h m infinity and h m w in this expression and also make use of the definition of b and n w equal to g b.

Then substitution would show that n w can be written as q l, that is the conduction heat transferred inside the transferred substance that is in the liquid film, divided by c p m into t infinity minus t w divided by b minus lambda ref equal to g into b.

(Refer Slide Time: 01:20)



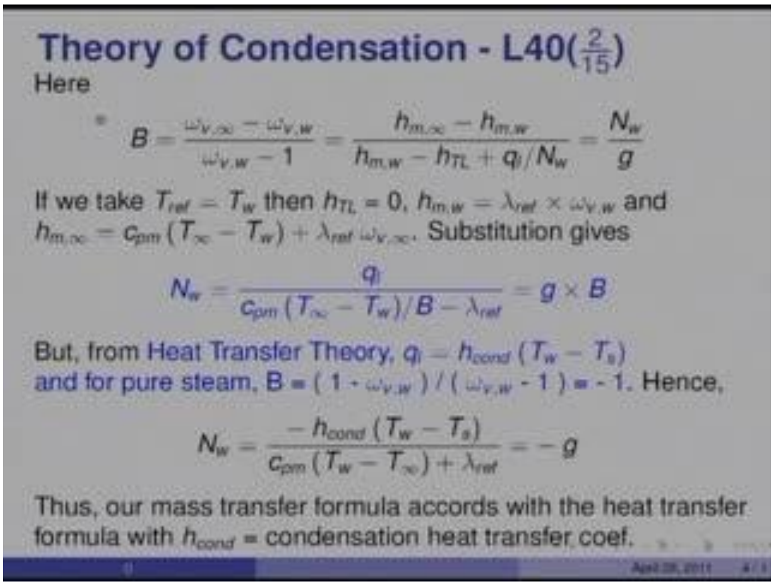
Condensation - L40($\frac{1}{15}$)

Prob: Consider condensation of steam at 1 atm on the outside of a Copper tube (2.5 cm ID and 2.9 cm OD). The tube carries cooling water at 50°C Calculate steam condensation rate when (a) steam is pure and saturated and (b) steam is mixed with 20 % air by mass . Assume condensate film thickness $\delta = 0.125$ mm, $k_{cu} = 300$ W/m-K, $k_{water} = 0.68$ W/m-K, $h_{cof,i} = 4620$ W/m^2-K, $\lambda_{ref} = 2257$ kJ/kg and $T_{ref} = T_w$

$h_{cof,o} = 115$ W/m^2-K (single phase)

So this is our mass transfer or Reynolds flow model formula for calculating condensation rate; but from heat transfer theory we know that q l is actually written as h condensation into t w minus t s, where t s is the outside surface temperature of the tube.

(Refer Slide Time: 03:45)



Theory of Condensation - L40($\frac{2}{15}$)

Here

$$B = \frac{\omega_{v,\infty} - \omega_{v,w}}{\omega_{v,w} - 1} = \frac{h_{m,\infty} - h_{m,w}}{h_{m,w} - h_{TL} + q_l/N_w} = \frac{N_w}{g}$$

If we take $T_{ref} = T_w$ then $h_{TL} = 0$, $h_{m,w} = \lambda_{ref} \times \omega_{v,w}$ and $h_{m,\infty} = c_{pm}\,(T_\infty - T_w) + \lambda_{ref}\,\omega_{v,\infty}$. Substitution gives

$$N_w = \frac{q_l}{c_{pm}\,(T_\infty - T_w)/B - \lambda_{ref}} = g \times B$$

But, from Heat Transfer Theory, $q_l = h_{cond}\,(T_w - T_s)$ and for pure steam, B = (1 - $\omega_{v,w}$) / ($\omega_{v,w}$ - 1) = - 1. Hence,

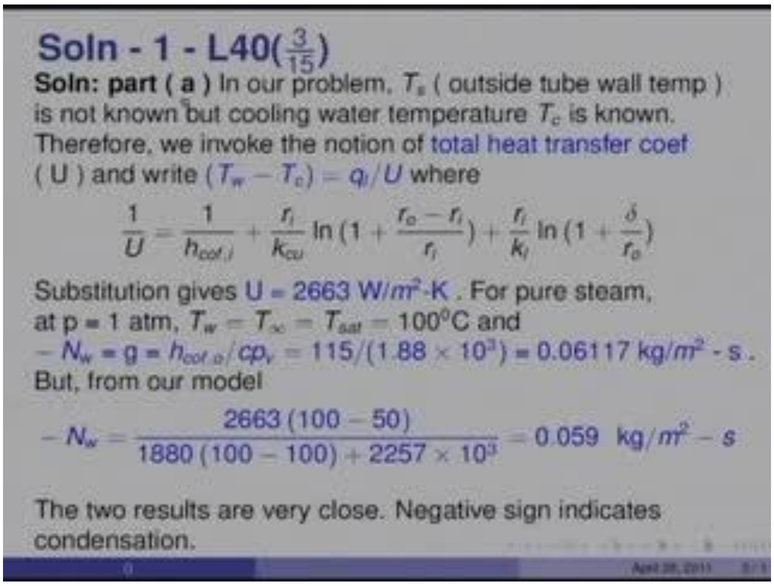
$$N_w = \frac{-\,h_{cond}\,(T_w - T_s)}{c_{pm}\,(T_w - T_\infty) + \lambda_{ref}} = -\,g$$

Thus, our mass transfer formula accords with the heat transfer formula with h_{cond} = condensation heat transfer coef.

And for pure steam omega v infinity will be equal to 1, so b will be equal to 1 minus omega v w divided by omega v w minus 1, and therefore b will be equal to minus 1; and hence our formula would simply read as n w equal to minus h condensation t w minus t s c p m into t w minus t infinity because b is minus 1 plus lambda ref.

Now, this is precisely the formula you have used in heat transfer theory to calculate condensation rate when steam is pure; so, thus our mass transfer formula accords with the heat transfer formula with h condensation equal to condensation heat transfer coefficient, and therefore there is a complete equivalence between the two.

(Refer Slide Time: 06:20)



Soln - 1 - L40($\frac{3}{15}$)

Soln: part (a) In our problem, T_s (outside tube wall temp) is not known but cooling water temperature T_c is known. Therefore, we invoke the notion of total heat transfer coef (U) and write $(T_w - T_c) = q_l / U$ where

$$\frac{1}{U} = \frac{1}{h_{cof,i}} + \frac{r_i}{k_{cu}} \ln\left(1 + \frac{r_o - r_i}{r_i}\right) + \frac{r_i}{k_l} \ln\left(1 + \frac{\delta}{r_o}\right)$$

Substitution gives U = 2663 W/m^2-K . For pure steam, at p = 1 atm, $T_w = T_\infty = T_{sat} = 100^\circ$C and $- N_w = g = h_{cof,o} / cp_v = 115/(1.88 \times 10^3) = 0.06117$ kg/m^2 - s . But, from our model

$$- N_w = \frac{2663\,(100 - 50)}{1880\,(100 - 100) + 2257 \times 10^3} = 0.059 \ \text{kg}/m^2 - s$$

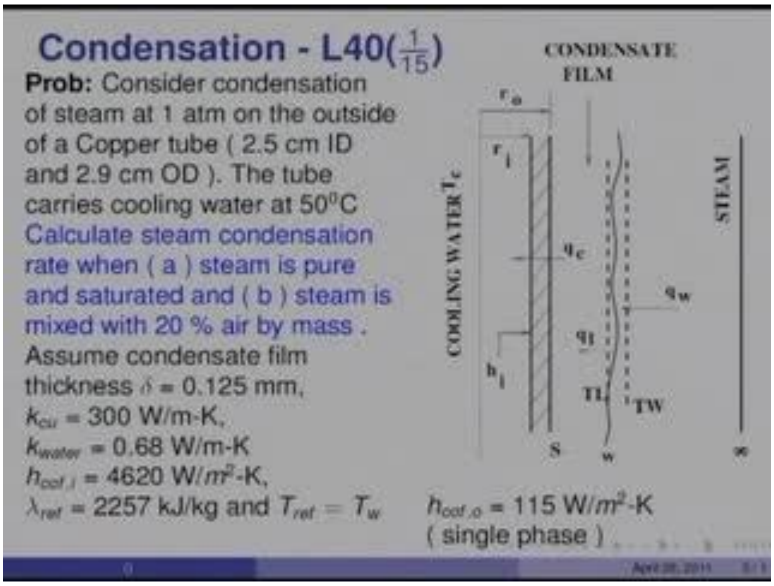
The two results are very close. Negative sign indicates condensation.

So, let us turn to our part a of the problem when steam is pure. So, in our problem t s is the outside tube wall temperature is not known, but the cooling water temperature t c is known, and therefore we can invoke the notion of the total heat transfer coefficient and write the condensation heat transfer Q l equal to divided by universal heat transfer coefficient u into t w minus I mean total heat transfer coefficient u into t w minus t c, where simply what I have done is 1 over u is the total resistance is equal to the resistance due to inside heat transfer coefficient plus resistance due to thickness of the copper wall and its conductivity.

This is the resistance due to the thickness of the liquid film having conductivity k liquid that is the water film; so, there are three resistances adding up to 1 over u, and substitution if I substitute these values from the data given here h cof i is 4620, k copper is 300, k water .68.

Then it works out that the total heat transfer coefficient u will be 2663 watts per meter kelvin. Now, for pure steam at p equal to 1 atmosphere, t w t infinity will equal t saturation, and that would equal hundred degree c.

(Refer Slide Time: 01:20)



Condensation - L40($\frac{1}{15}$)

Prob: Consider condensation of steam at 1 atm on the outside of a Copper tube (2.5 cm ID and 2.9 cm OD). The tube carries cooling water at 50°C Calculate steam condensation rate when (a) steam is pure and saturated and (b) steam is mixed with 20 % air by mass . Assume condensate film thickness $\delta = 0.125$ mm,

$k_{cu} = 300$ W/m-K,

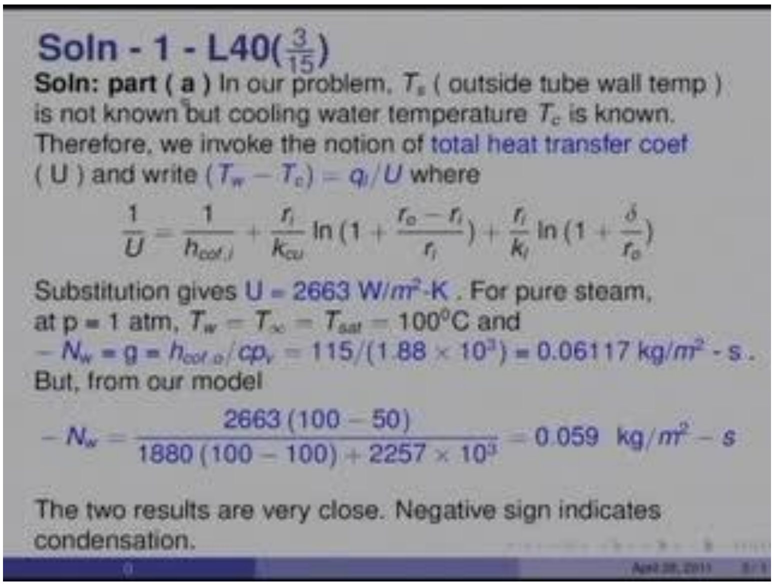
$k_{water} = 0.68$ W/m-K

$h_{cof,i} = 4620$ W/m^2-K,

$\lambda_{ref} = 2257$ kJ/kg and $T_{ref} = T_w$

$h_{cof,o} = 115$ W/m^2-K (single phase)

(Refer Slide Time: 06:20)



Soln - 1 - L40($\frac{3}{15}$)

Soln: part (a) In our problem, T_s (outside tube wall temp) is not known but cooling water temperature T_c is known. Therefore, we invoke the notion of total heat transfer coef (U) and write $(T_w - T_c) = q_I / U$ where

$$\frac{1}{U} = \frac{1}{h_{cof,i}} + \frac{r_i}{k_{cu}} \ln\left(1 + \frac{r_o - r_i}{r_i}\right) + \frac{r_i}{k_l} \ln\left(1 + \frac{\delta}{r_o}\right)$$

Substitution gives U = 2663 W/m^2-K . For pure steam, at p = 1 atm, $T_w = T_\infty = T_{sat} = 100^\circ$C and $-N_w = g = h_{cof,o}/cp_v = 115/(1.88 \times 10^3) = 0.06117$ kg/m^2 - s . But, from our model

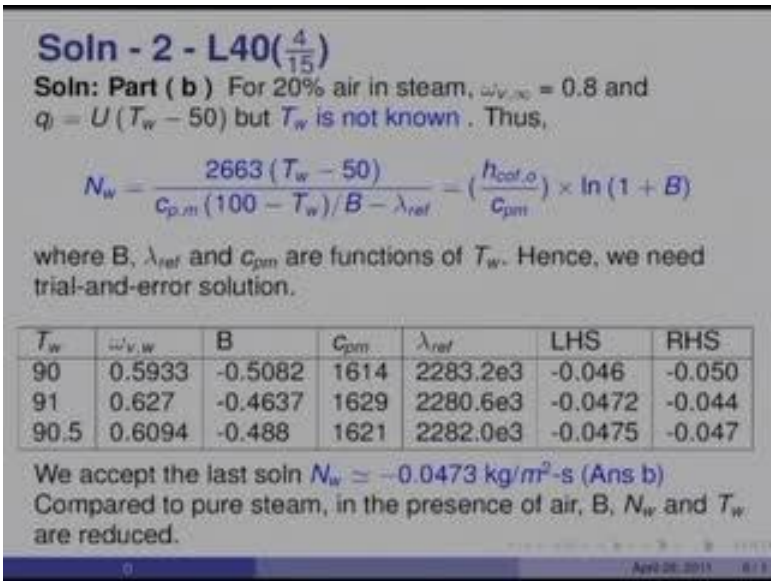
$$-N_w = \frac{2663\,(100 - 50)}{1880\,(100 - 100) + 2257 \times 10^3} = 0.059 \ \text{kg}/m^2 - s$$

The two results are very close. Negative sign indicates condensation.

And therefore, our formula is minus n w equal to g equal to h cof by c p of steam, and h cof for natural convection was given as a 115 watts per meter square kelvin, and therefore we readily calculate g is equal to 115 divided by 1.88 into 10 is to 3, equal to .06117 kg per meter square also, so this is a straightforward application of our formula to estimate n w which is negative n w equal to that; and from our model writing q l equal to u into t infinity minus t c that is 2663 into 100 minus 50 into 1880 which is the specific heat of the mix of the vapor into t w minus t infinity both are 100 degrees, and therefore, that term makes no contribution divided by the lambda ref which is 2257 into 10 raise to 3, and the result is .059 k g per meter square second.

So, the two results are very very close .0617 and .059; now, of course, the difference arises mainly because we assume h cof divided by c p v, and assuming h cof given is of the right order magnitude that the two results are extremely close, and that verifies our Reynolds flow model for estimating condensation rate.

(Refer Slide Time: 09:40)



Soln - 2 - L40($\frac{4}{15}$)

Soln: Part (b) For 20% air in steam, $\omega_{v,\infty} = 0.8$ and $q_l = U\,(T_w - 50)$ but T_w is not known . Thus,

$$N_w = \frac{2663\,(T_w - 50)}{c_{p,m}\,(100 - T_w)/B - \lambda_{ref}} = \left(\frac{h_{cof,o}}{c_{pm}}\right) \times \ln\,(1 + B)$$

where B, λ_{ref} and c_{pm} are functions of T_w. Hence, we need trial-and-error solution.

T_w	$\omega_{v,w}$	B	c_{pm}	λ_{ref}	LHS	RHS
90	0.5933	-0.5082	1614	2283.2e3	-0.046	-0.050
91	0.627	-0.4637	1629	2280.6e3	-0.0472	-0.044
90.5	0.6094	-0.488	1621	2282.0e3	-0.0475	-0.047

We accept the last soln $N_w \simeq -0.0473$ kg/m^2-s (Ans b)
Compared to pure steam, in the presence of air, B, N_w and T_w are reduced.

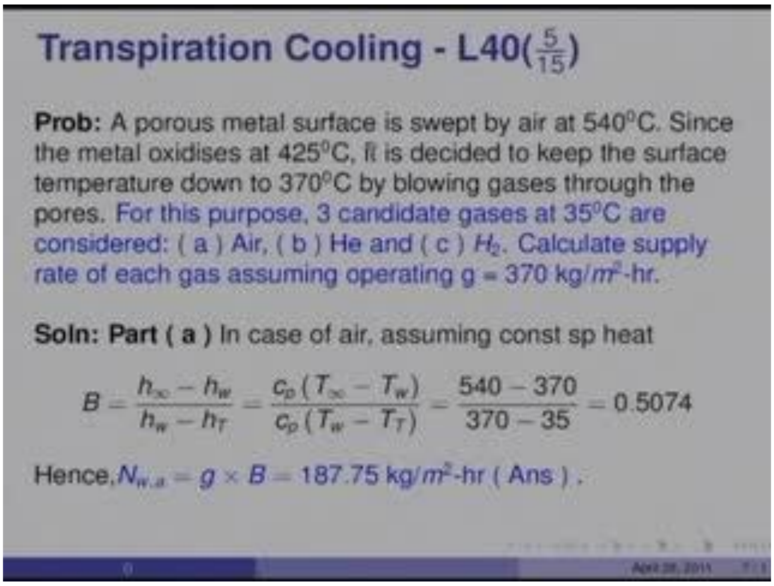
Negative sign, of course, in both these results minus n w equal to that, and minus n w equal to minus 059 indicates condensation. We now turn to the part b of the problem where there is twenty percent air in the steam, and therefore omega v infinity will be .8, and q l as before will be u into t w minus 50, but t w is not known, and therefore, our formula for n w will be 2663 t w minus 50 divided by c p m into 100 minus t w divided by b minus lambda ref and that would equal h cof o by c p m l n 1 plus b.

That is the formula for where b lambda ref and c p m all functions of t w, but we do not know t w, and therefore we must do iterations; so, the thing is you assume t w, and therefore, evaluate omega v w from saturation condition there and that gives us the value of b. Knowing omega v w we calculate mean specific heat as 1614, and also evaluate lambda ref equal to 2283.2 into 10 raise to 3, so the left hand side of this relationship gives us minus .046, whereas the right hand side on here gives us minus .05 when t w was 90, so obviously there is a difference in the between the left and right hand side, so we take next guess of 91 degree centigrade.

And we find that the left hand side is now minus .0472, and the right hand side is minus 044, so obviously the result must be in between the two; and we now take 90.5, and we find that the result is minus .0475 at the left hand side and we accept that result as nearly correct, and take n w equal to minus .0473 k g per meter square second; now, you will see that the our earlier result was minus .059, whereas now the result is minus .0473, and therefore the condensation rate has reduced because of the 20 percent air in steam; and this is a well-known problem in condensers, because the condenser operates at a very low pressure, there is always a chance of air being ingressed, and therefore, the rate of condensation reduces and that is why air ejectors are used in condensers.

This is just a problem of that variety which checks out what we normally do in order to prevent falling of condensation rate, we always remove air from the steam which enters the condenser. I will now take up the problem of transpiration cooling, which is always used to protect surfaces which are exposed to very high temperature gases. So, consider a problem as given here.

(Refer Slide Time: 12:34)



Transpiration Cooling - L40($\frac{5}{15}$)

Prob: A porous metal surface is swept by air at 540^{0}C. Since the metal oxidises at 425^{0}C, it is decided to keep the surface temperature down to 370^{0}C by blowing gases through the pores. For this purpose, 3 candidate gases at 35^{0}C are considered: (a) Air, (b) He and (c) H_2. Calculate supply rate of each gas assuming operating g = 370 kg/m^2-hr.

Soln: Part (a) In case of air, assuming const sp heat

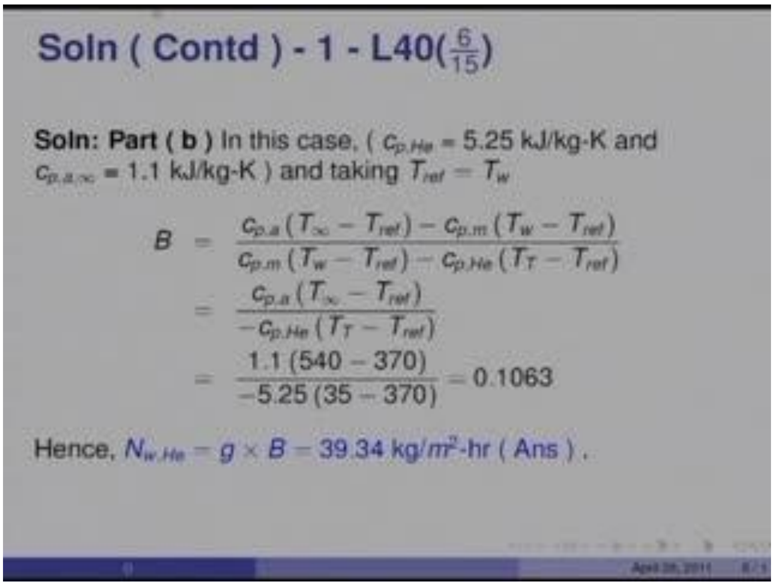
$$B = \frac{h_\infty - h_w}{h_w - h_T} = \frac{c_p\,(T_\infty - T_w)}{c_p\,(T_w - T_T)} = \frac{540 - 370}{370 - 35} = 0.5074$$

Hence, $N_{w,a} = g \times B = 187.75$ kg/m^2-hr (Ans) .

A porous metal surface is swept by air at 540 degree centigrade that is very high temperature. Now, since metal oxidizes at 425 degree centigrade. It is decided to keep the surface temperature down to 370, so our t w we want is at 370 by blowing gases through the pores. For this purpose, three candidate gases available at 35 degree centigrade are considered, one is air itself that is you inject air in air, you inject helium in air, and thirdly hydrogen in air. Calculate the supply rate of each gas assuming operating g of 370 kg per meter square hour; and we assume that the in all three cases of injection g mol as remains constant.

In case of air assuming constant specific heat we assume that between 370 and 540, there is not much differences specific heat of air, then the problem becomes not that it cannot be handled with variable specific heat, but for simplicity c p is equal to t infinity minus t w c p t w minus t t, so that simply becomes 540 minus 370, 370 minus 35 which is t t, and that is equal to .5074, that is the b.

(Refer Slide Time: 14:19)



Soln (Contd) - 1 - L40($\frac{6}{15}$)

Soln: Part (b) In this case, ($c_{p,He} = 5.25$ kJ/kg-K and $c_{p,a,\infty} = 1.1$ kJ/kg-K) and taking $T_{ref} = T_w$

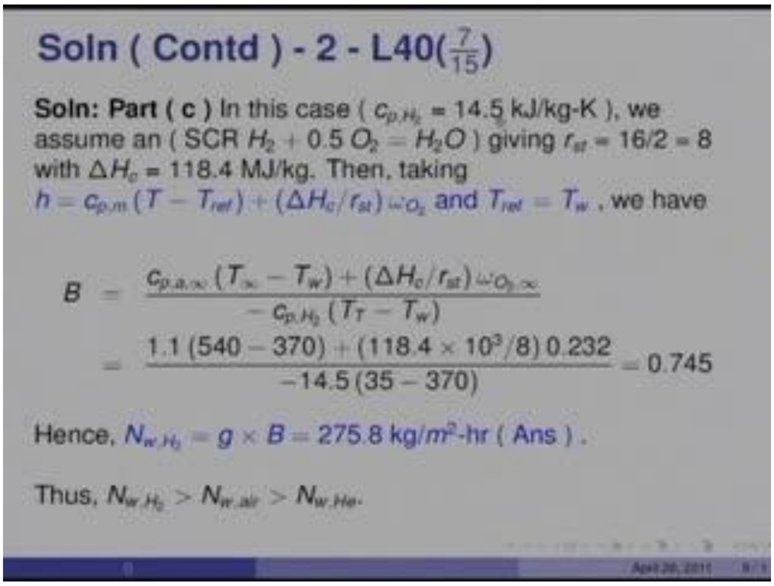
$$\begin{aligned} B &= \frac{c_{p,a}\,(T_\infty - T_{ref}) - c_{p,m}\,(T_w - T_{ref})}{c_{p,m}\,(T_w - T_{ref}) - c_{p,He}\,(T_T - T_{ref})} \\ &= \frac{c_{p,a}\,(T_\infty - T_{ref})}{-c_{p,He}\,(T_T - T_{ref})} \\ &= \frac{1.1\,(540 - 370)}{-5.25\,(35 - 370)} = 0.1063 \end{aligned}$$

Hence, $N_{w,He} = g \times B = 39.34$ kg/m^2-hr (Ans) .

And therefore, n w of air would be g times b equal to 187.75 kg per meter square per hour, so that is the answer for the part one of the problem. Now, we consider helium and hydrogen - part b; so, in this case now specific heat of helium is 5.25 kilo joules per kg kelvin and c p a infinity is 1.1 kilo joules kg per kelvin.

So therefore, if I take t ref equal to t w, then you will see I do not have to calculate mixture specific heat, because it will simply mean t w minus t ref gets canceled here as t w minus t ref gets canceled here, this is h infinity, and this is h t, and that would simply be c p a into t infinity minus t ref divided minus c p helium into t t minus t ref, so I will get 1.1 into 540 minus 370 divided by minus 5.25 into 35 minus 370 b is .0163. So, b for helium is smaller than that for air, because of high specific heat and you will see n w helium is g times b will be about 40 kg per meter square hour, so much reduced quantity of helium is required compare to air.

(Refer Slide Time: 15:35)



Soln (Contd) - 2 - L40($\frac{7}{15}$)

Soln: Part (c) In this case ($c_{p,H_2} = 14.5$ kJ/kg-K), we assume an (SCR $H_2 + 0.5\,O_2 = H_2O$) giving $r_{st} = 16/2 = 8$ with $\Delta H_c = 118.4$ MJ/kg. Then, taking $h = c_{p,m}(T - T_{ref}) + (\Delta H_c / r_{st})\,\omega_{O_2}$ and $T_{ref} = T_w$, we have

$$B = \frac{c_{p,a,\infty}(T_\infty - T_w) + (\Delta H_c / r_{st})\,\omega_{O_2,\infty}}{-\,c_{p,H_2}(T_T - T_w)} = \frac{1.1\,(540 - 370) + (118.4 \times 10^3/8)\,0.232}{-14.5\,(35 - 370)} = 0.745$$

Hence, $N_{w,H_2} = g \times B = 275.8$ kg/m^2-hr (Ans) .

Thus, $N_{w,H_2} > N_{w,air} > N_{w,He}$.

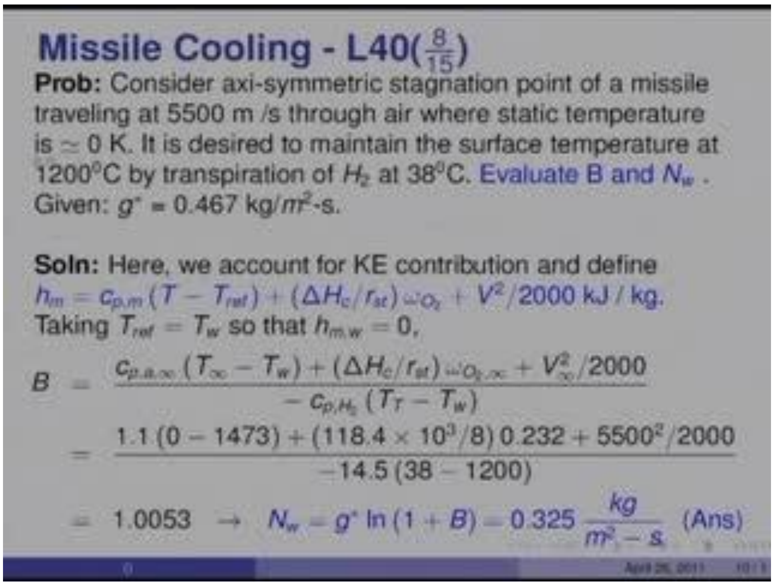
Now, let us take the case of hydrogen. In this case hydrogen specific heat is 14.5, and hydrogen is going to burn in such a hot environment; so, we assume a simple chemical reaction hydrogen plus half o 2 equal to h 2 o giving r s t is equal to 16 by 2 equal to 8, that is the stoichiometric ratio; for this case is 16 by 2 equal to 8, and the latent heat of hydrogen is 118 mega joules per kilogram, and therefore taking h equal to c p m into t minus t ref plus del h c by r s t omega o 2.

As we know we have associated the latent heat with oxygen, then therefore we divide this by r s t; and if we take t ref equal to t w we have b equal to h infinity h w of course will be 0, because omega o 2 cannot survive at the surface as well as t is equal to t ref, and we will have minus c p h two into t t minus t w which will be the h t, this is 1.1 c p a infinity is 1.1 into 540 minus 370 plus 100 and 18.4 into 10 raise to 3 divided by 8 into .232 divided by minus 40.5 into 35 minus 70.

And now, the b increases to .745, and therefore n w of hydrogen will be g into b 275.8; remember, we are using hydrogen which is going to burn to keep the surface cool at 370 degree centigrade. And you will see now that, in the three application if we find that the amount of hydrogen required will be greater than that for air and that for helium, this sort of simple calculations enable us to select the right kind of gas to inject through the porous surface depending on our requirement.

I will now take up the next problem and that is of missile cooling. Now, missiles as you know travel at very high altitudes at very very high speeds of the order of 5 to 7000 meter meters per second very similar to the kind of velocities that are encountered when the reentry vehicle enters the upper atmosphere at around 10000 meters per second.

(Refer Slide Time: 18:23)



Missile Cooling - L40($\frac{8}{15}$)

Prob: Consider axi-symmetric stagnation point of a missile traveling at 5500 m /s through air where static temperature is $\simeq 0$ K. It is desired to maintain the surface temperature at 1200°C by transpiration of H_2 at 38°C. Evaluate B and N_w . Given: $g^* = 0.467$ kg/m^2-s.

Soln: Here, we account for KE contribution and define

$h_m = c_{p,m}(T - T_{ref}) + (\Delta H_c/r_{st})\,\omega_{O_2} + V^2/2000$ kJ / kg.

Taking $T_{ref} = T_w$ so that $h_{m,w} = 0$,

$$B = \frac{c_{p,a,\infty}(T_\infty - T_w) + (\Delta H_c/r_{st})\,\omega_{O_2,\infty} + V_\infty^2/2000}{-\,c_{p,H_2}(T_T - T_w)}$$

$$= \frac{1.1\,(0 - 1473) + (118.4 \times 10^3/8)\,0.232 + 5500^2/2000}{-14.5\,(38 - 1200)}$$

$$= 1.0053 \;\rightarrow\; N_w = g^* \ln(1 + B) = 0.325\,\frac{kg}{m^2 - s} \quad \text{(Ans)}$$

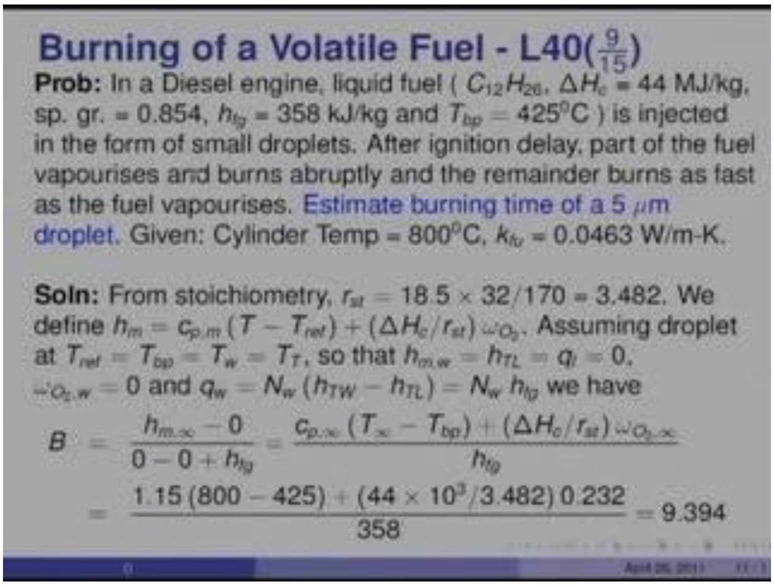
So, these are very very high velocity projectiles which have to be kept cool, because of the viscous heating that takes place near the surface. So, the problem reads as follows, consider axis symmetric stagnation point of a missile travelling at 5500 meters per second through air where static temperature is almost say 0 k. It is desired to maintain the surface temperature at about 1200 degree centigrade by transpiration cooling of hydrogen at 38 degree centigrade. Evaluate B and N w, given g star in this case equal to .467 kg per meter square second.

So, here the value of g star is again given; now, because of the high velocity here we must account for the kinetic energy contribution, and define h m equal to c p m t minus t ref plus delta h c divided by r s t omega o 2, which is as usual, plus v squared by 2000 as the kinetic energy contribution to enthalpy in kilo joules per k g, therefore, v squared by two into 1000 that is just what in kilo joules per kg will be.

So, taking t ref equal to t w so that h m w again is 0, then the first x, the numerator will be h infinity divided by minus h t will be c p h 2 into t t minus t w, and you get ah c p a infinity equal to 1.1 minus 0 minus 1473, because t infinity is almost taken as 0. The wall temperature is 1200 degree centigrade or 1473 kelvin plus 118.4 into 10 raise to 3, which is the delta h c of hydrogen divided by 8 as before into .232, which will the omega o 2, which is that which is the mass fraction of oxygen in air plus 5500 square divided by 2000 divided by minus 14.5 into 38 minus 1200.

So, this gives us the large v of value 1, and n w will be g star into l n 1 plus b will give you g star is .467, and therefore .325 kg per meters square second. Missiles of this type have to carry a certain amount of gas with them, so that in order to keep the surface cool, the particularly the stagnation point of the missile has to be kept very very cool, because it is it is travelling at a very very high velocity; under such conditions a hif, you know the time of flight knowing the mass transfer rate as we have calculated here, we can calculate so knowing the time of flight that is seconds and knowing the surface area over which you are going to inject the gas. We can calculate the amount of hydrogen that the missile must carry with it; so, problems of this kind give you a first estimate of how much hydrogen to carry in a missile without solving any differential equation.

(Refer Slide Time: 21:30)



Burning of a Volatile Fuel - L40($\frac{9}{15}$)

Prob: In a Diesel engine, liquid fuel ($C_{12}H_{26}$, $\Delta H_c = 44$ MJ/kg, sp. gr. = 0.854, $h_{fg} = 358$ kJ/kg and $T_{bp} = 425^0$C) is injected in the form of small droplets. After ignition delay, part of the fuel vapourises and burns abruptly and the remainder burns as fast as the fuel vapourises. Estimate burning time of a 5 μm droplet. Given: Cylinder Temp = 800^0C, $k_{fu} = 0.0463$ W/m-K.

Soln: From stoichiometry, $r_{st} = 18.5 \times 32/170 = 3.482$. We define $h_m = c_{p,m}(T - T_{ref}) + (\Delta H_c/r_{st})\,\omega_{O_2}$. Assuming droplet at $T_{ref} = T_{bp} = T_w = T_T$, so that $h_{m,w} = h_{TL} = q_l = 0$, $\omega_{O_2,w} = 0$ and $q_w = N_w(h_{TW} - h_{TL}) = N_w\, h_{fg}$ we have

$$B = \frac{h_{m,\infty} - 0}{0 - 0 + h_{fg}} = \frac{c_{p,\infty}(T_\infty - T_{bp}) + (\Delta H_c/r_{st})\,\omega_{O_2,\infty}}{h_{fg}}$$

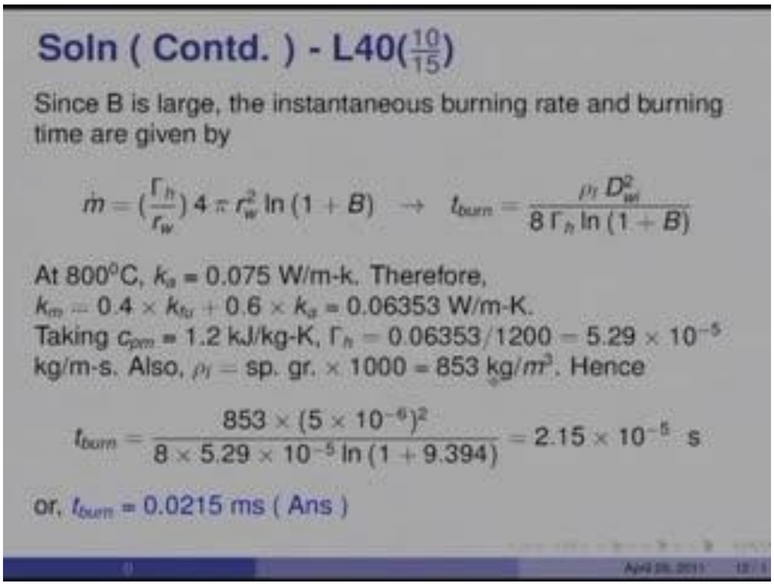
$$= \frac{1.15\,(800 - 425) + (44 \times 10^3/3.482)\,0.232}{358} = 9.394$$

Let us now look at another problem, and that is of burning of a volatile fuel. So, in a diesel engine liquid fuel c twelve h 26 with heat of combustion 44 mega joules per k g and specific gravity 0.854, and latent heat 358, and boiling point 425 degree centigrade is injected in the form of small droplets.After ignition delay, part of the fuel vaporizes and burns abruptly and the remainder of the fuel burns as fast as the fuel vaporizes. Estimate the burning time of a 5 micron droplet.

You are given that temperature in the cylinder, the gas temperature of the cylinder is about 800 degree centigrade, and the conductivity of the fuel is .0463 watts per meter kelvin. Now, from stoichiometry for c 12 h 26 fuel the oxygen to fuel ratio will be 18.5, which is m plus n that is 12 plus 26 by 4 will be 18.5 multiplied by 32, which is the molecular weight of oxygen divided by 170, which is the molecular weight of the fuel, and therefore the oxygen to fuel ratio will be 3.482.

We define h m equal to c p m into t minus t ref plus del h c by r s t into omega o 2. And assume the droplet to be at it is boiling point, so all temperatures t ref t b p t w and t t are all at the boiling point of 425 degree centigrade. So, that h m w h t l and q l, there will be no conduction heat transfer inside the liquid fuel; and therefore we will simply have b equal to h m infinity minus 0, which is h m w h m w again 0 minus h t l, which is 0, and this will be simply h f g, because q l by n w would be…, as you can see q w will be n w h t w minus h t l will be simply n w into h f g, and therefore our formula will simply be c p infinity into t infinity minus t b p plus del h c by r s t omega o 2 infinity by h f g and taking c p infinity equal to 1.15 800 minus 425, which is the boiling point 44 into 10 raise to 3, which is the heat of combustion into divided by 3.482, which is r s t into .232 divided by 348, which is the value of h f g you have been given there, and therefore 9.394 very very high b in case of liquid volatile burning. Now, let us go on to calculate the burning time of a 5 micron droplet.

(Refer Slide Time: 24:24)



Soln (Contd.) - L40($\frac{10}{15}$)

Since B is large, the instantaneous burning rate and burning time are given by

$$\dot{m} = \left(\frac{\Gamma_h}{r_w}\right) 4 \pi r_w^2 \ln(1 + B) \rightarrow t_{burn} = \frac{\rho_l D_{wi}^2}{8 \Gamma_h \ln(1 + B)}$$

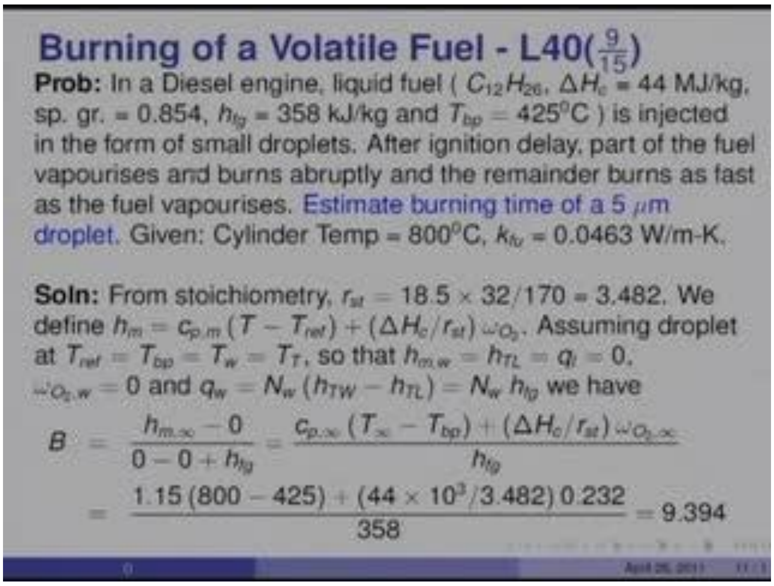
At 800°C, $k_a = 0.075$ W/m-k. Therefore, $k_m = 0.4 \times k_{fu} + 0.6 \times k_a = 0.06353$ W/m-K. Taking $c_{pm} = 1.2$ kJ/kg-K, $\Gamma_h = 0.06353/1200 = 5.29 \times 10^{-5}$ kg/m-s. Also, $\rho_l =$ sp. gr. $\times$ 1000 = 853 kg/m^3. Hence

$$t_{burn} = \frac{853 \times (5 \times 10^{-6})^2}{8 \times 5.29 \times 10^{-5} \ln(1 + 9.394)} = 2.15 \times 10^{-5} \text{ s}$$

or, $t_{burn} = 0.0215$ ms (Ans)

Since b is large, the instantaneous burning rate and the burning time would be given by, m dot into gamma h by r w 4 pi r w square l n 1 plus b and that will give us t burn equal to rho l into d w i square 8 into gamma h into l n 1 plus b. Now, at 800 degree centigrade conductive heat of air is .075 such calculation the mean conductivity is evaluated by as .4 times fuel conductivity plus point six multiplied by air conductivity, and therefore the mean conductivity will be .06353. And mean specific heat we can take as 1.2 kilo joules per kg kelvin, and therefore gamma h which is k m by c p m would be .06353 divided by twelve hundred equal to 5.29 into 10 raise to minus 5 kilograms per meter second.

(Refer Slide Time: 21:30)



Burning of a Volatile Fuel - L40($\frac{9}{15}$)

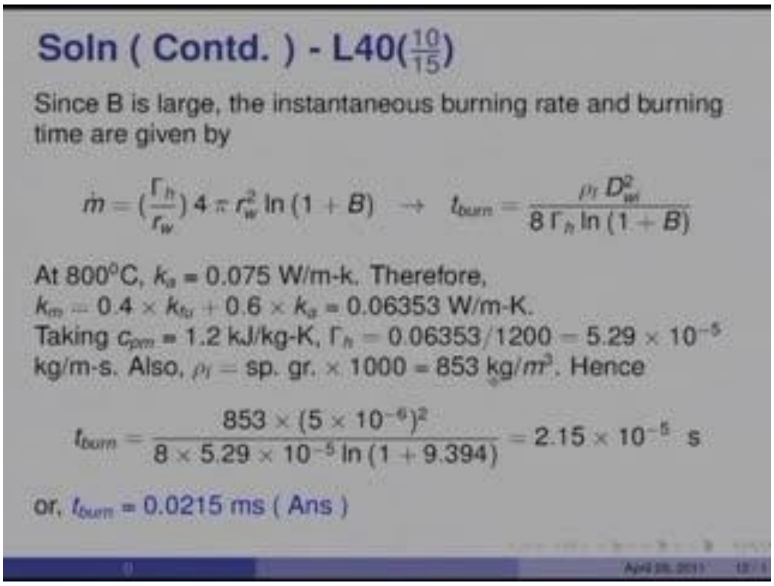
Prob: In a Diesel engine, liquid fuel ($C_{12}H_{26}$, $\Delta H_c = 44$ MJ/kg, sp. gr. = 0.854, $h_{fg} = 358$ kJ/kg and $T_{bp} = 425^\circ$C) is injected in the form of small droplets. After ignition delay, part of the fuel vapourises and burns abruptly and the remainder burns as fast as the fuel vapourises. Estimate burning time of a 5 μm droplet. Given: Cylinder Temp = 800°C, $k_{fu} = 0.0463$ W/m-K.

Soln: From stoichiometry, $r_{st} = 18.5 \times 32/170 = 3.482$. We define $h_m = c_{p,m}(T - T_{ref}) + (\Delta H_c / r_{st})\,\omega_{O_2}$. Assuming droplet at $T_{ref} = T_{bp} = T_w = T_T$, so that $h_{m,w} = h_{TL} = q_l = 0$, $\omega_{O_2,w} = 0$ and $q_w = N_w\,(h_{TW} - h_{TL}) = N_w\,h_{fg}$ we have

$$B = \frac{h_{m,\infty} - 0}{0 - 0 + h_{fg}} = \frac{c_{p,\infty}\,(T_\infty - T_{bp}) + (\Delta H_c / r_{st})\,\omega_{O_2,\infty}}{h_{fg}}$$

$$= \frac{1.15\,(800 - 425) + (44 \times 10^3/3.482)\,0.232}{358} = 9.394$$

(Refer Slide Time: 24:24)



Soln (Contd.) - L40($\frac{10}{15}$)

Since B is large, the instantaneous burning rate and burning time are given by

$$\dot{m} = \left(\frac{\Gamma_h}{r_w}\right) 4\,\pi\,r_w^2 \ln(1 + B) \rightarrow t_{burn} = \frac{\rho_l\,D_{wi}^2}{8\,\Gamma_h \ln(1 + B)}$$

At 800°C, $k_a = 0.075$ W/m-k. Therefore, $k_m = 0.4 \times k_{fu} + 0.6 \times k_a = 0.06353$ W/m-K. Taking $c_{pm} = 1.2$ kJ/kg-K, $\Gamma_h = 0.06353/1200 = 5.29 \times 10^{-5}$ kg/m-s. Also, ρ_l = sp. gr. × 1000 = 853 kg/m^3. Hence

$$t_{burn} = \frac{853 \times (5 \times 10^{-6})^2}{8 \times 5.29 \times 10^{-5} \ln(1 + 9.394)} = 2.15 \times 10^{-5} \text{ s}$$

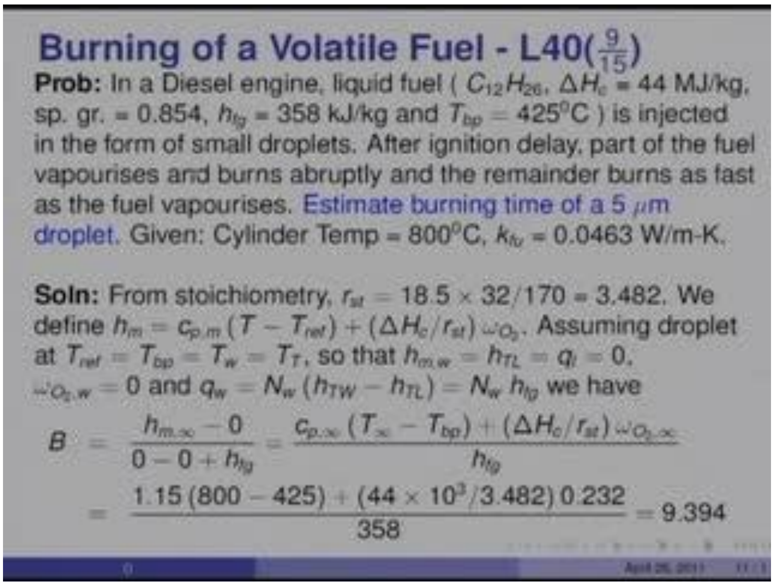
or, $t_{burn} = 0.0215$ ms (Ans)

Now, you are also given rho l, the liquid the specific gravity is .84, and therefore rho l will be 853 kg per meter cube, and you will recall that our…, and if we substitute that here t burn equal to 853 5 into 10 raise to minus 6 raise to 2 into 8 into 5.29 into 10 raise to minus 5 l n into 1 plus 9.394, which is the value of b, and that gives us 0.0215 milliseconds.

Now, of course, here I have estimated the burning time based on assuming that the environment inside the diesel engine is more less stagnant, that is at the top deck center where the burning takes place there is not much velocity; in reality there would be some velocity between the injected droplet and the gas which is turned as a result of the piston movement. And therefore, there would be some enhancement in the effective value of gamma h, because of convection pass the droplet; of course, the his role will be very very small, because the droplet diameter is very very small of 5 microns, and therefore the answer is reasonable. Such time is very very important, because this time affects the cutoff ratio of the diesel engine, and therefore its efficiency.

As you know we want very small cutoff ratio, and if all the fuel burns out in a smaller time then the cutoff ratio will be smaller, and therefore, it is of crucial importance to estimate what the burning time of a droplet would be.

(Refer Slide Time: 21:30)



Burning of a Volatile Fuel - L40($\frac{9}{15}$)

Prob: In a Diesel engine, liquid fuel ($C_{12}H_{26}$, ΔH_c = 44 MJ/kg, sp. gr. = 0.854, h_{fg} = 358 kJ/kg and T_{bp} = 425°C) is injected in the form of small droplets. After ignition delay, part of the fuel vapourises and burns abruptly and the remainder burns as fast as the fuel vapourises. Estimate burning time of a 5 μm droplet. Given: Cylinder Temp = 800°C, k_{fu} = 0.0463 W/m-K.

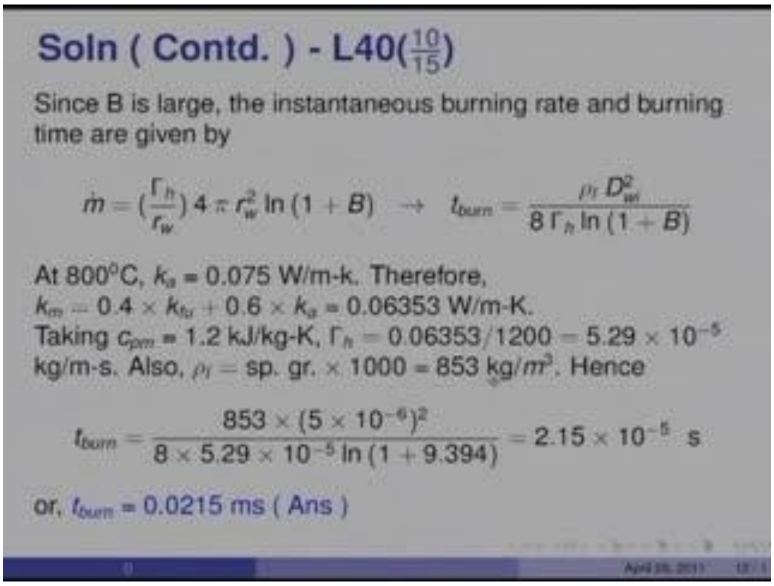
Soln: From stoichiometry, $r_{st} = 18.5 \times 32/170 = 3.482$. We define $h_m = c_{p,m}(T - T_{ref}) + (\Delta H_c/r_{st})\,\omega_{O_2}$. Assuming droplet at $T_{ref} = T_{bp} = T_w = T_T$, so that $h_{m,w} = h_{TL} = q_l = 0$. $\omega_{O_2,w} = 0$ and $q_w = N_w(h_{TW} - h_{TL}) = N_w\, h_{fg}$ we have

$$B = \frac{h_{m,\infty} - 0}{0 - 0 + h_{fg}} = \frac{c_{p,\infty}(T_\infty - T_{bp}) + (\Delta H_c/r_{st})\,\omega_{O_2,\infty}}{h_{fg}}$$

$$= \frac{1.15\,(800 - 425) + (44 \times 10^3/3.482)\,0.232}{358} = 9.394$$

Of course, in a diesel engine there will cloud of droplets and in our estimate of b; we have assumed that each droplet experiences omega o 2 in infinity equal to .32. In reality some droplets that is at the front of the cloud will experience omega o 2 infinity equal to .232, but those that follow will be facing a mixture of burnt products and air, and therefore, they will experience somewhat lower value of omega o 2 infinity, and therefore the b will be smaller, they will take a little longer then to burn.

(Refer Slide Time: 24:24)



Soln (Contd.) - L40($\frac{10}{15}$)

Since B is large, the instantaneous burning rate and burning time are given by

$$\dot{m} = (\frac{\Gamma_h}{r_w})\, 4\, \pi\, r_w^2 \ln(1 + B) \quad \rightarrow \quad t_{burn} = \frac{\rho_l\, D_{wi}^2}{8\, \Gamma_h \ln(1 + B)}$$

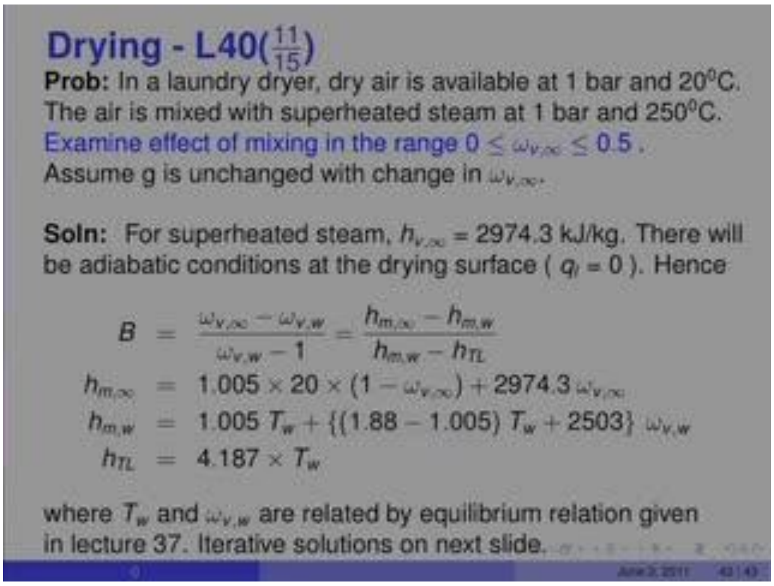
At 800°C, $k_a = 0.075$ W/m-k. Therefore,
$k_m = 0.4 \times k_{fu} + 0.6 \times k_a = 0.06353$ W/m-K.
Taking $c_{pm} = 1.2$ kJ/kg-K, $\Gamma_h = 0.06353/1200 = 5.29 \times 10^{-5}$ kg/m-s. Also, ρ_l = sp. gr. × 1000 = 853 kg/m^3. Hence

$$t_{burn} = \frac{853 \times (5 \times 10^{-6})^2}{8 \times 5.29 \times 10^{-5} \ln(1 + 9.394)} = 2.15 \times 10^{-5} \text{ s}$$

or, $t_{burn} = 0.0215$ ms (Ans)

But nonetheless the estimate that we have provided here 0.215 millisecond is a very valuable one in accessing how quickly the fuel shall burn inside a diesel engine.

(Refer Slide Time: 28:01)



Drying - L40($\frac{11}{15}$)

Prob: In a laundry dryer, dry air is available at 1 bar and 20°C. The air is mixed with superheated steam at 1 bar and 250°C. Examine effect of mixing in the range $0 \leq \omega_{v,\infty} \leq 0.5$. Assume g is unchanged with change in $\omega_{v,\infty}$.

Soln: For superheated steam, $h_{v,\infty} = 2974.3$ kJ/kg. There will be adiabatic conditions at the drying surface ($q_l = 0$). Hence

$$\begin{aligned} B &= \frac{\omega_{v,\infty} - \omega_{v,w}}{\omega_{v,w} - 1} = \frac{h_{m,\infty} - h_{m,w}}{h_{m,w} - h_{TL}} \\ h_{m,\infty} &= 1.005 \times 20 \times (1 - \omega_{v,\infty}) + 2974.3\, \omega_{v,\infty} \\ h_{m,w} &= 1.005\, T_w + \{(1.88 - 1.005)\, T_w + 2503\}\, \omega_{v,w} \\ h_{TL} &= 4.187 \times T_w \end{aligned}$$

where T_w and $\omega_{v,w}$ are related by equilibrium relation given in lecture 37. Iterative solutions on next slide.

We now take up another problem and that is of convective drying of cloth in a laundry dryer. So, the problem statement is as follows, in a laundry dryer dry air is available at 1 bar and 20 degree centigrade. Now, of course, this is a cold air, and therefore the engineer wants to try out mixing of this air with super-heated steam at 1 bar and 250 degree centigrade.

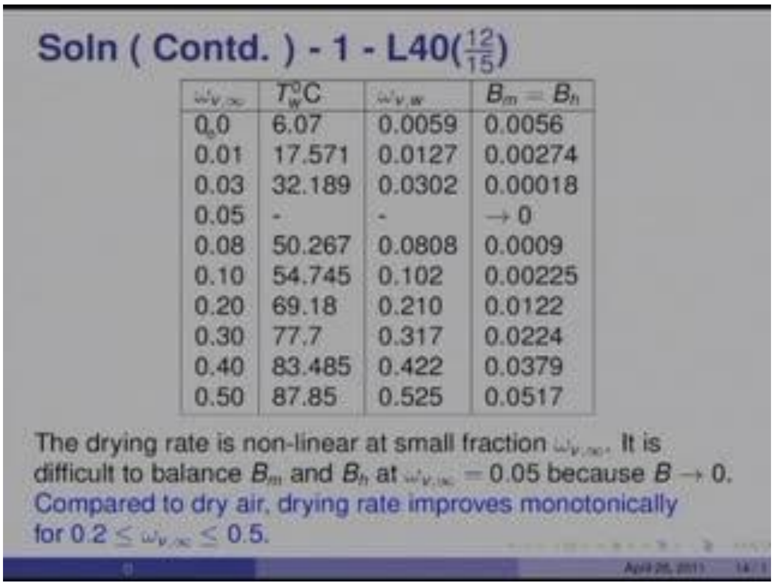
Now, use of steam in for super-heated steam for drying is quite routinely used in textile industry particularly in a device called the stenter, where the wet printed cloth is to be dried at a very fast rate; instead of using air which gives you rather, because air has to be heated somewhere, and that air has a lowest specific heat than specific heat of the of steam, and therefore what is preferred is super-heated steam drying rather than air drying.

So, we want to examine here the possibility of mixing 1 bar 250 degree centigrade super-heated steam with air which is available at 1 bar and 20 degree c. And let us see if the effective mixtures which will flow pass the drying cloth would actually dry it fast. So, examine the effect of mixing in the range of 0 omega v infinity to .5 assume g is unchanged with change in omega v infinity.

Now, for superheated steam at 1 bar and 250 degree centigrade from steam tables we have we can get h v infinity equal to 2974.3 kilo joules per k g. There will be adiabatic conditions at the drying surface, and therefore q l equal to 0, and hence our formula would be b omega v infinity minus omega v w omega v w minus 1 equal to omega h m infinity minus h m w h m w minus h t l, where omega h m infinity will be the enthalpy of air plus enthalpy of steam weighted by the mass fractions, 1 minus omega v infinity is the mass fraction of air, and omega v infinity would be the mass fraction of vapor or the steam.

At the surface, we do not know the value of t w, but we can take this as 1.005 t w 2503, because t ref is taken as 0, omega v w, and h t l would be 4.187 into t w, where t w and omega v w are related by equilibrium relation given in lecture thirty-seven, and therefore we must iterate as usual to assume t w, therefore calculate v w, therefore calculate h m w, and see whether the left hand side agrees with the right hand side.

(Refer Slide Time: 31:20)



Soln (Contd.) - 1 - L40($\frac{12}{15}$)

$\omega_{v,\infty}$	T_w^0C	$\omega_{v,w}$	$B_m = B_h$
0.0	6.07	0.0059	0.0056
0.01	17.571	0.0127	0.00274
0.03	32.189	0.0302	0.00018
0.05	-	-	$\rightarrow 0$
0.08	50.267	0.0808	0.0009
0.10	54.745	0.102	0.00225
0.20	69.18	0.210	0.0122
0.30	77.7	0.317	0.0224
0.40	83.485	0.422	0.0379
0.50	87.85	0.525	0.0517

The drying rate is non-linear at small fraction $\omega_{v,\infty}$. It is difficult to balance B_m and B_h at $\omega_{v,\infty} = 0.05$ because $B \rightarrow 0$.

Compared to dry air, drying rate improves monotonically for $0.2 \leq \omega_{v,\infty} \leq 0.5$.

April 26, 2011

So, we need iterative solution, and on the next slide I will show you the iterative solutions for a range of values of omega v infinity varying from 0 to .5; so, if omega v infinity is 0, that is only pure air is supplied at 20 degree centigrade, then t w actually turns out to be 6 degrees only, and b is .0056, therefore we expect very very low rates of drying.

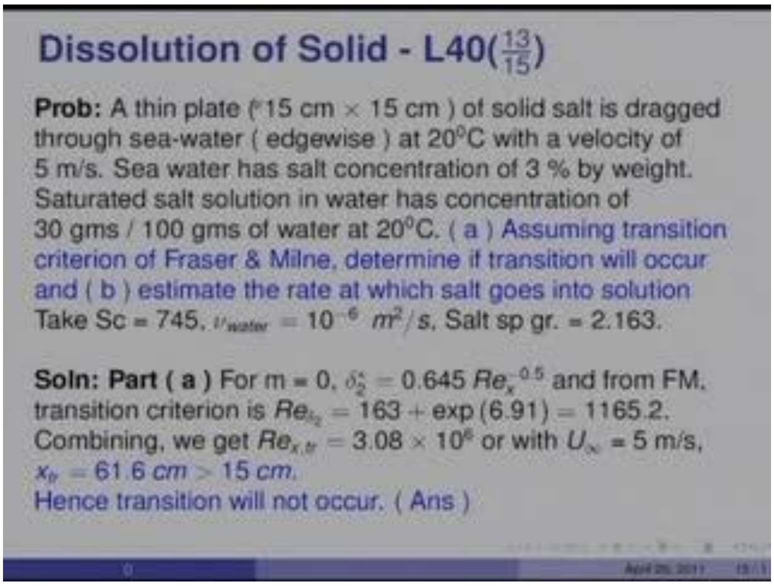
At .01 that is 1 percent steam addition when would have thought that the drying rate will actually increase, but you will see that the drying rate has actually decreased to .00274 b is decreased. If I make it .03, that is three percent, then it still further decreases to .00018, but the temperature increases of the surface to 32 degree centigrade; now, at .04 and .05, you one finds that the b values are extremely small, 10 raise to minus 5, 10 raise to minus 6 of that order.

And therefore, it is extremely difficult to balance b m equal to b h, and find t w exactly, so these are skipped now, and I go to eight percent air, and I find that it is .009, so it is better than three percent air, but still less than that with pure air. If I go to 10 percent air however I get .00225 again the drying rate will increase, but still lower than the value corresponding to pure air.

But at 20 percent I get .0122 that means, b has now doubled, and 20 percent steam mixing would achieve results in the desirable direction at 30 percent, of course, still further improvement in b. At 40 percent still further improvement. And at 50 percent .0517 to still further; so, drying rate is a non-linear at small mass fractions omega v infinity when they are small, it first decreases with increase in omega almost goes to 0, then starts increasing again and beyond .1 it exceeds the value of pure air case very interesting result.

So, here is a comment that it is difficult to balance b m equal to b h exactly when omega v infinity is around .05, because b itself tends to 0; so, compared to dry air, drying rate improves monotonically beyond omega v infinity .02, and of course, if you went for a pure steam you will certainly get much higher drying rate compared to pure air which is omega v infinity equal to 0.

(Refer Slide Time: 34:26)



Dissolution of Solid - L40($\frac{13}{15}$)

Prob: A thin plate (15 cm × 15 cm) of solid salt is dragged through sea-water (edgewise) at 20^{0}C with a velocity of 5 m/s. Sea water has salt concentration of 3 % by weight. Saturated salt solution in water has concentration of 30 gms / 100 gms of water at 20^{0}C. (a) Assuming transition criterion of Fraser & Milne, determine if transition will occur and (b) estimate the rate at which salt goes into solution Take Sc = 745, $\nu_{water} = 10^{-6}\ m^2/s$, Salt sp gr. = 2.163.

Soln: Part (a) For m = 0, $\delta_2^* = 0.645\ Re_x^{0.5}$ and from FM, transition criterion is $Re_{\delta_2} = 163 + \exp(6.91) = 1165.2$. Combining, we get $Re_{x,tr} = 3.08 \times 10^6$ or with $U_\infty = 5$ m/s, $x_{tr} = 61.6\ cm > 15\ cm$.
Hence transition will not occur. (Ans)

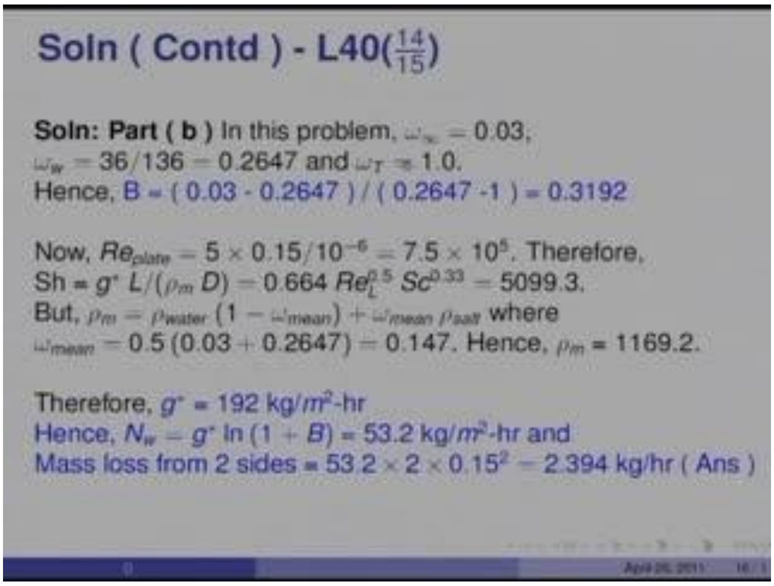
So, such industrial irrelevant problems also can be very effectively solved by Reynolds flow model. Finally, I take up the case of solid dissolving in liquid, and I am taking up the case of a thin plate, now there is no gas involved here, nor is there any chemical reaction, it is a dissolution of solid into a liquid; in fact it is a salt slab as you can see here in the problem statement, a thin plate 15 centimeters by 15 centimeters of solid salt is to be dragged through sea water edge wise at 20 degree centigrade with a velocity of 5 meters per second sea water has salt concentration of 3 percent by weight.

Saturated salt solution in water has concentration of 30 grams per 100 grams of water at 20 degree centigrade; assuming transition criterion of fraser and milne determine if transition will occur in this particular case and estimate the rate at which salt will go into the solution from the solid plate. Take for the salt solution Schmidt number is 745 which is very high the nu kinematic viscosity of water's sea water is 10 raise to minus 6 meters square per second, and the salt specific gravity is 2.163.

So, first part of the question we want to examine whether the when the sea water flows over the plate there will be transition or not now this is a case of m equal to 0, that is the pressure gradient is 0, and delta two star will be .645 Reynolds x to the minus .5. And from faser and milne, for m equal to 0 r e delta two will be 163 plus exponential of 6.91or 1165.2 will be the r e based on delta 2, and therefore if I substitute that here this result and this result.

Then we would get r e x transition equal to 3.08 into 10 raise to 6 or u infinity equal to 5 meters per second, which gives x transition at 61.6 centimeters per second, which is of course greater than our plate which is 15 centimeter. And therefore, over the plate transition will not occur we will simply have laminar boundary layer. Now, let us go on to the part b of the problem, where we wish to estimate the rate at which salt will go into the solution.

(Refer Slide Time: 36:57)



Soln (Contd) - L40($\frac{14}{15}$)

Soln: Part (b) In this problem, $\omega_\infty = 0.03$, $\omega_w = 36/136 = 0.2647$ and $\omega_T = 1.0$.
Hence, B = (0.03 - 0.2647) / (0.2647 -1) = 0.3192

Now, $Re_{plate} = 5 \times 0.15/10^{-6} = 7.5 \times 10^5$. Therefore,
Sh = $g^* L/(\rho_m D) = 0.664\, Re_L^{0.5}\, Sc^{0.33} = 5099.3$.
But, $\rho_m = \rho_{water}\,(1 - \omega_{mean}) + \omega_{mean}\,\rho_{salt}$ where
$\omega_{mean} = 0.5\,(0.03 + 0.2647) = 0.147$. Hence, $\rho_m = 1169.2$.

Therefore, $g^* = 192$ kg/m^2-hr
Hence, $N_w = g^* \ln(1 + B) = 53.2$ kg/m^2-hr and
Mass loss from 2 sides = $53.2 \times 2 \times 0.15^2 = 2.394$ kg/hr (Ans)

So in this problem omega infinity is .03, because that is the sea water concentration you have been given that; at the surface of the salt it will be 30 grams per hundred grams, and therefore 36 divided 100 grams of water, and therefore it will be 36 divided by 136 equal to .2647, omega t which is pure salt will be 1, and therefore b will be .03 minus 0.2647 divided by .2647 minus 1 equal to 0.3192.

And therefore, the plate Reynolds number of course here is 5 meters per second into the length of the plate, which is 0.15 divided by kinematic viscosity which is 10 raise to minus 6 is 7.5 into 10 raise to 5. And therefore, Sherwood number would be g star l divided by rho m into diffusivity will be equal to .664 r e l to the .5, Schmidt number equal to .33will be 5.99.3 that would be the Sherwood number.

Now, in order to calculate rho m, we take rho m equal to rho water plus 1 minus omega mean which is the mass fraction of water plus omega mean into rho of salt, where omega mean will be taken as .03 in the sea water, and .2647 at the surface, as we have shown here, and that is mean is .147, so therefore, mean density would be 1169.2, and therefore g star taking l equal to .15, rho m equal to 1169.2, and diffusivity value has been given to you, because Schmidt number is known, and nu is known.

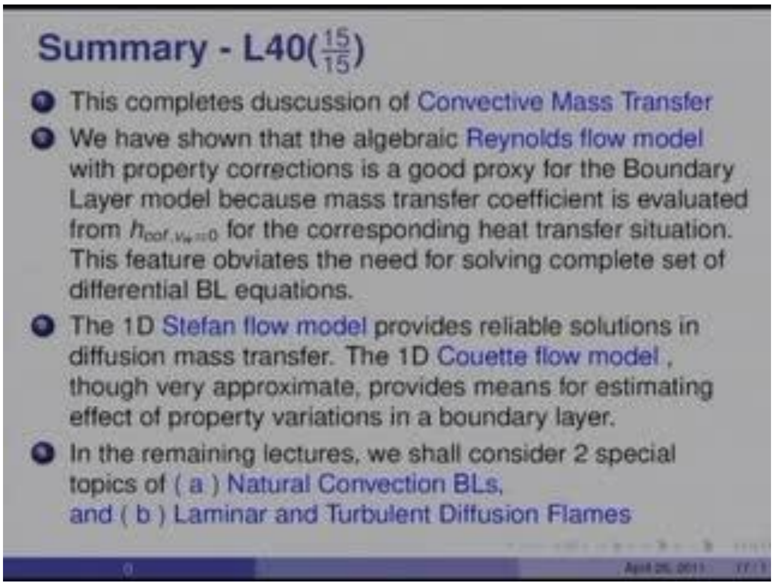
And therefore, we can get the value of diffusivity, therefore we get g star is estimated at 192 kilograms per meter square hour; and therefore, n w will be g star into l n plus into 1 plus b equal to 53.2 meter kilograms per meter square hour; and therefore, the mass loss from two sides of the plate will be 53.2 into two into .15 square, which is the area of the plate and the two sides.

So, 2.394 kg per hour would be the answer for the rate at which the salt will go into the solution. So, this is a departure from other problems of gaseous type, we here we have mass transfer into liquid from a solid, so with this I complete all our discussion on essentially force convection heat and mass transfer. In the first 10 lectures I dealt with laminar flow, the first 10 lectures I develop the main theory of laminar flows, external boundary layers then took up the case of internal laminar internal flows.

Simple as well as complex shaped (( )), but all under force convection situation then we spent 10 lectures that is lecture number twenty-one to lecture number thirty on turbulent flows, where we discussed the formal and predictive aspects of turbulent flows.

And develop methods for estimating calculating external boundary layers both starting from laminar through transition to turbulence over external surfaces; as well as we applied phenomenology as well as analogy methods to determine nusselt numbers for internal flows like pipe flows and included effects of roughness and so on so forth. So and then from lecture number thirty onwards we turn to convective mass transfer, where we develop the Stefan flow model, the Couette flow model, and the Reynolds flow model as proxies for the boundary layer flow model.

(Refer Slide Time: 41:17)



Summary - L40($\frac{15}{15}$)

- This completes duscussion of Convective Mass Transfer
- We have shown that the algebraic Reynolds flow model with property corrections is a good proxy for the Boundary Layer model because mass transfer coefficient is evaluated from $h_{cof, V_w=0}$ for the corresponding heat transfer situation. This feature obviates the need for solving complete set of differential BL equations.
- The 1D Stefan flow model provides reliable solutions in diffusion mass transfer. The 1D Couette flow model , though very approximate, provides means for estimating effect of property variations in a boundary layer.
- In the remaining lectures, we shall consider 2 special topics of (a) Natural Convection BLs, and (b) Laminar and Turbulent Diffusion Flames

And in the last few lectures I applied these models to practical problems of engineering relevance. So, our conclusion then is that, the that the algebraic Reynolds flow model with property corrections is a good proxy for the boundary layer flow model, because mass transfer coefficient is evaluated from h cof infinity at v w equal to 0 for the corresponding heat transfer situation, and we have number of co relations for h cof v w equal to 0, which we can readily use for the corresponding mass transfer situation.

And therefore, estimate g star and go on to calculate the mass transfer rate, it is this feature which obviates the need for solving complete set of boundary layer equation; the one d Stefan flow model provides a reliable solutions in diffusion mass transfer, the one d Couette flow model though very approximate provides mean for estimating effect of property variations in a boundary layer.

And we showed that the Couette flow model actually does predict separation of variable property mass transfer rates when property correction is applied as this found from the Reynolds flow model or invent from the boundary layer flow model. So, with this really I complete all issues of convective mass transfer at the interface as well as convective heat transfer, but all under forced convection situation.

And in the remaining two lectures I am going to take up two special topics, and these are described here at the bottom of the slide; so, we will consider two special topics - one is of natural convection boundary layers in which I will consider both heat transfer as well as mass transfer. And secondly where heat and mass transfer again play an important role is laminar diffusion flame, that is it will be heat and mass transfer with chemical reaction, but in a free jet situation, there is no wall, but we the what is of importance is to predict the length of the flame and the width of the flame. And these are two special topics which I think are relevant to convective heat transfer, and therefore, in the last two lectures I will be dealing with these two topics.