**Indian institute of technology madras**
**NPTEL**
**National Programme on technology enhanced learning**
**Video lectures on**
**Convictive heat transfer**

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**Lecture 26**
**Hydro dynamicallyand thermallyfully**
**Developed internal laminar flows**

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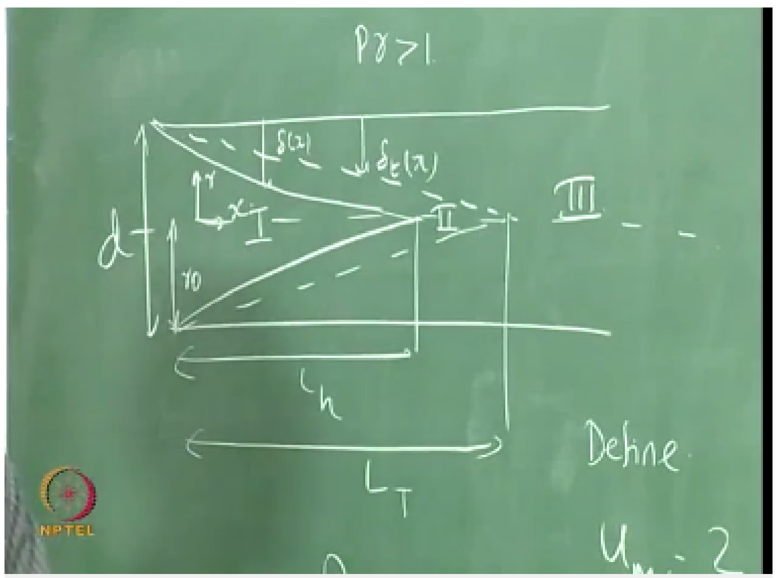


So good morning all of you so as yesterday we were looking at internal flows we started with the basic introduction about the fundamentals of internal laminar internal flows so I was talking about the growth of the laminar boundary layer in the hydrodynamic boundary layer from the different walls of the duct and at some point downstream from the entrance the boundary layers merge and that is that is what is called the the distance where they emerge from.

The entrance of the duct is called the entry length okay the hydrodynamic entry length and beyond that you will find the entire region is dominated by viscous effects okay and also the velocity profiles if you draw draw somewhere downstream of the region - we will find that they take the shape of a parabolic velocity profile so how do we know that this is parabolic we will derive the from the basic equations we will show that it is a parabolic profile and apart from. That if you also have an Associated heat transfer either in the form of uniform temperature or a uniform heat flux applied onto the walls of the duct so you can say that this is constant Q wall.

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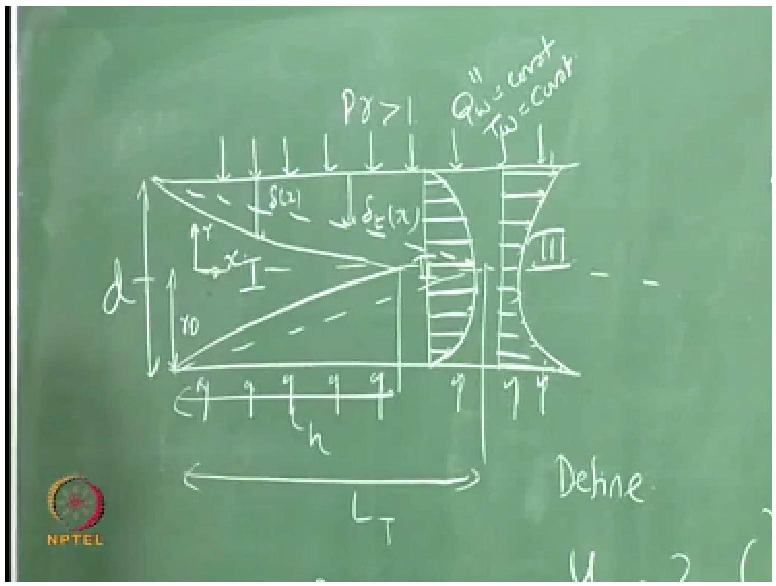


Prime or T wall is constant in this case you will you will find a temperature profile so where the temperature is higher at the wall and is minimum at the center of the duct now the criteria for defining these three regions one two three that I have indicated here so in region one the velocity profile keeps changing along with the axial position okay, so therefore this is a region where we cannot make much approximations in the nervier-stokes once it merges here so once the entrance length is reached as the two boundary layers merged if you plot the profile on from there onwards you will find that it is a parabolic profile and it is invariant of the location downstream so there are therefore four regions 2 1 3 you can put a criteria that D u/ DX = 0.

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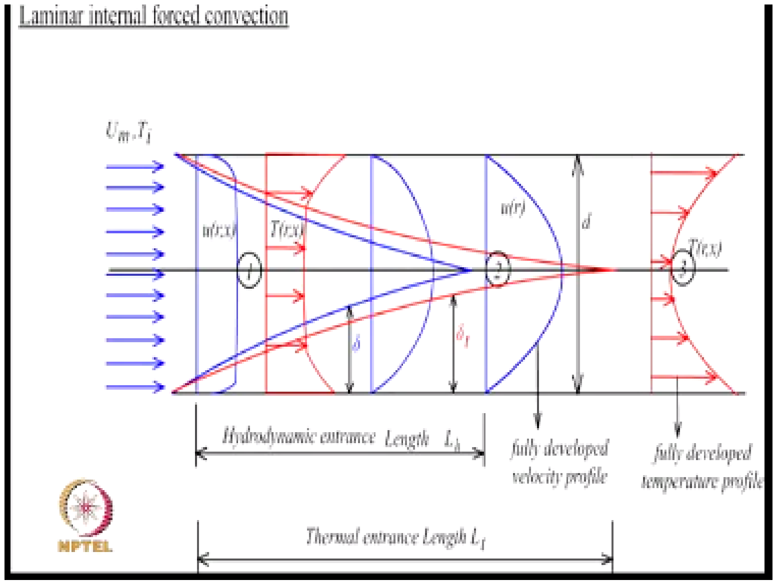


You that means the profiles are invariant of position downstream and this region is called as hydro dynamically fully developed region okay and when it comes to the temperature profiles however as you can see that when you apply a uniform heat flux or a wall temperature you keep on supplying energy and therefore the wall temperature keeps changing keeps increasing along with the location simultaneously the fluid which is receiving heat from the wall also.

Keeps increasing in its temperature profile therefore you cannot put a similar criteria saying that DT / DX = 0 because DT /DX will keep on changing because of continuous heat addition therefore we have to look at a non-dimensional temperature θ which when defined in this particular fashion t - T w/ T- T w so here unlike the external flows where you use the free stream temperature as the reference we have to use a new reference temperature which we call.

As a mean temperature or bulk temperature of the mixing cup temperature and we define the mean temperature we derived this yesterday based on that if you replace a non uniform temperature with a constant or a uniform temperature which we call as T mean so the enthalpy of this and this should be the same okay because this is a flow process so therefore we talk in terms of enthalpy rather than internal energy so for.

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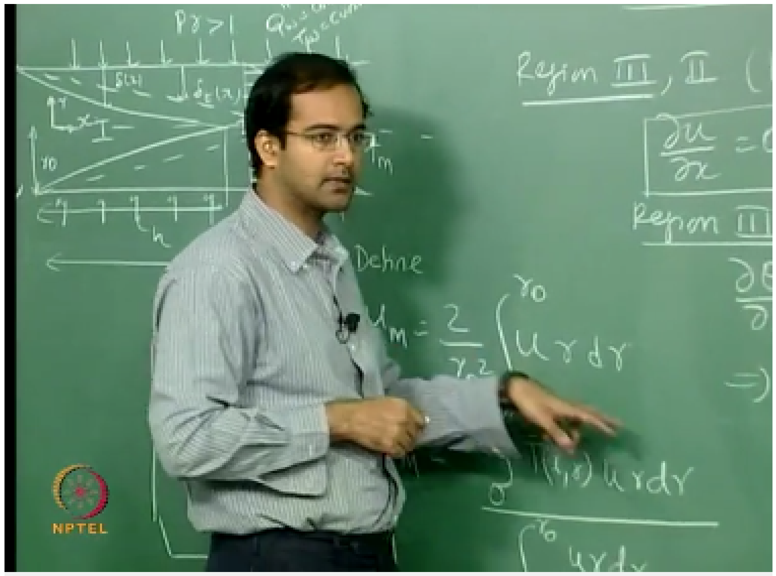


That condition you get an expression for the mean temperature or bulk temperature which Nothing but mass weighted average temperature you see this this is the temperature and this is the mass so you are giving a mass weighted average at different locations radically and the you are summing or integrating them together ok so this is how you define a mean temperature and therefore conditioned on theta okay now the condition on theta will be now your wall temperature will be only a function of X and so is your mean temperature because you are averaging out along the radial direction so therefore if you look at the numerator so this t -.

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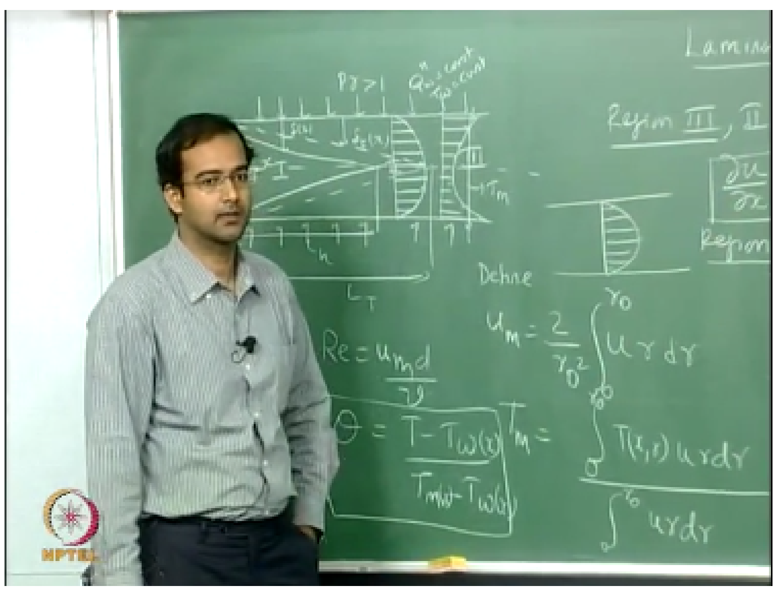


T wall will have a certain variation along X which is exactly similar to the variation of T my T mean - t1 so therefore your θ as a whole does not have any variation with respect to X Okay so therefore theta will be only a function of R right so therefore we can put a condition. That if it has to be thermally fully developed then Dθ / DX should be 0 so that means theta should be only a function of R okay, so therefore if you plot the non-dimensional temperature profile it will look very similar to your velocity profile so where it will go 0 at the wall and you get some kind of a profile like this and this will remain invariant along the axial location ok.

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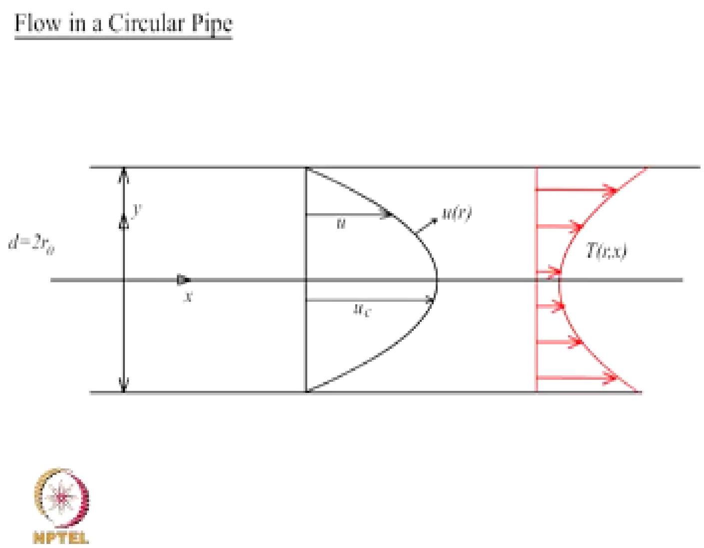


So we also showed that if d θ DX has to be zero consequently the heat transfer Coefficient in region 3 okay where it is hydro dynamically and thermally fully developed has to be a constant irrespective of what thermal boundary condition that you use whether it is a constant wall temperature is constant heat flux okay so this is a very important condition that you get in internal flows which is unlike the external flows where your heat transfer coefficient is a local.

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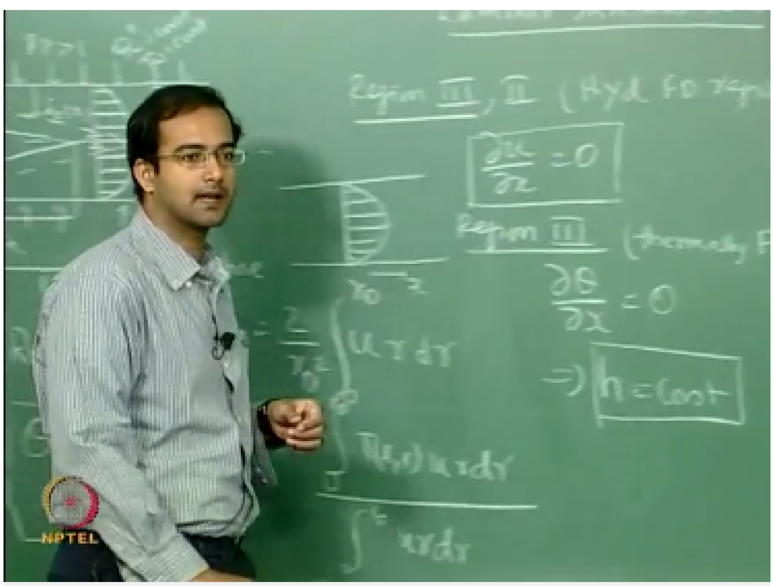

Heat transfer coefficient and it keeps very continuously okay here once you get a fully developed profile both hydrodynamically and thermally the value of heat transfer coefficient has to be a constant so we will have to see now what this constant value is and it depends on the boundary condition okay so the next couple of lectures we will focus on only the region 3 which is the simplest region to start with where you can make approximations with respect to.

The velocity as well as the temperature profiles and we can derive the expression for it is just a Value constant value for the two different boundary conditions one for the constant wall flux the other for constant wall temperature and now other thing is when you define Reynolds Number in internal flows you have to be careful that you should use again another characteristic Velocity called the mean velocity or the bulk velocity so once again what it means if you.

Replace your velocity profile with the uniform variation all along the cross section okay the it should satisfy conservation of mass at that particular location so the mass flow across

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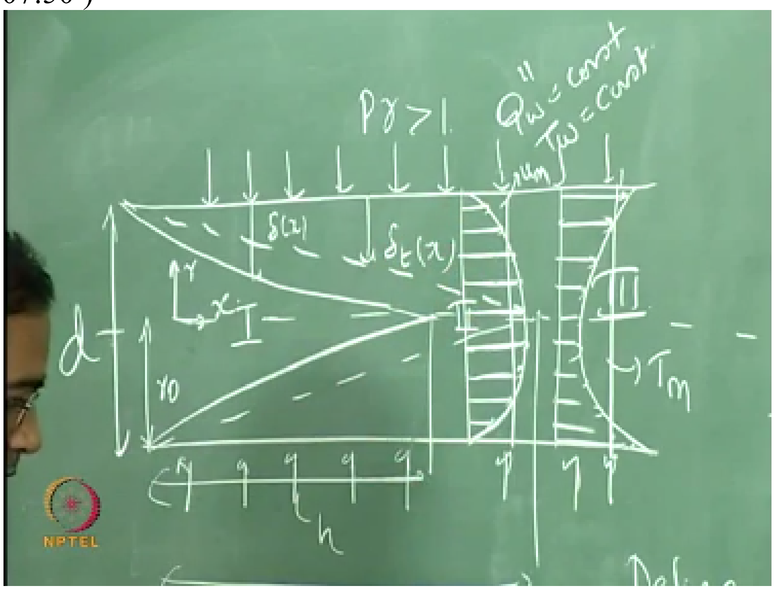


This particular cross section in this profile should be equal to the uniform profile so if you do that you will get a definition for the mean velocity which is something like this which we saw yesterday and based on that you define your Reynolds number okay now for a circular cross section of course you know this is the diameter of the cross section but you can also extend this definition to non circular cross sections right so if you have ducked with the say rectangular or triangular or square cross-section or some other cross-section so you have to define your Reynolds number based on what is called the.

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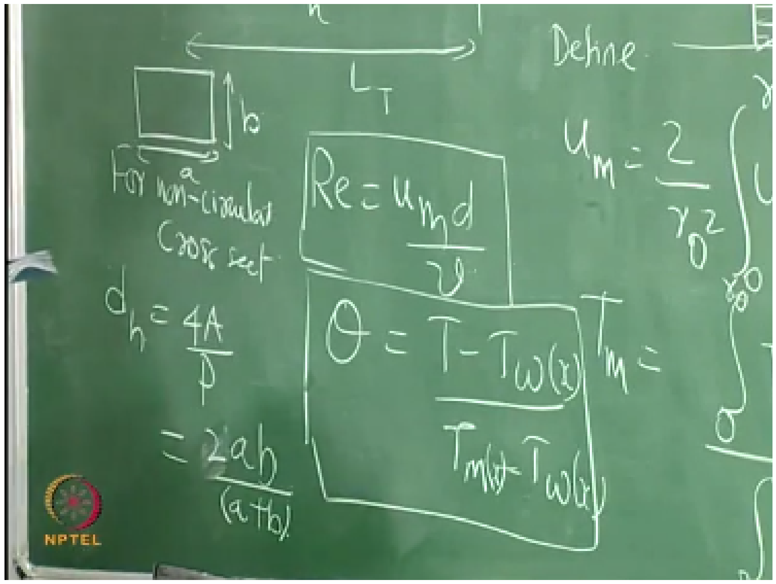


Hydraulic diameter so you use hydraulic diameter which is defined as four times area by three Meter okay so if you have a rectangular cross-section with the dimensions let me just so if I if I Must tell you that this is your a and this is your b okay so accordingly this will be so your area now will be what this will be four times ab divided by two times a plus B right so this will be two a b by a plus b okay so this is how you have to replace your diameter with an equivalent hydraulic diameter and define your Reynolds number based on the hydraulic diameter so this is for the case of non circular cross-sections okay so now with the sufficient understanding also we have classified the different regimes whether it is a laminar or turbulent depending on the value of Reynolds number below a certain critical Reynolds number critical Reynolds number.

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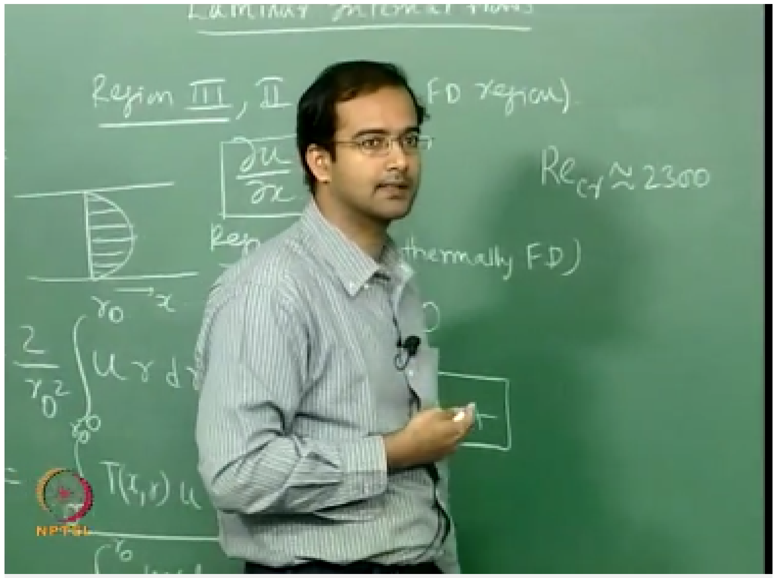


Generally is about 2300 so below this value it is generally classified as a laminar flows and above this generally considered as transition or turbulent although this value may vary in cases where you maintain a very uniform streamlined profile and your turbulent intensities are very low at the inlet and your wall is extremely smooth you can delay the transition to turbulence to as high as 8,000 or sometimes even 10,000 okay experimentally it has been that you can delay.

The transition okay but this is a generic thumb rule okay so today what we will do is we will take up first the region three okay and in Region three we will get the solution to the velocity and the temperature profile and now just before that I just want to write down the conservation equations for momentum and energy in Region three okay so we will make certain approximations when it comes to Region three so what are the approximations so why we are Looking at Region threes it is one of the simplest to start with because you can make several approximations so that your nervier-stokes equations becomes much simpler so you can get any.

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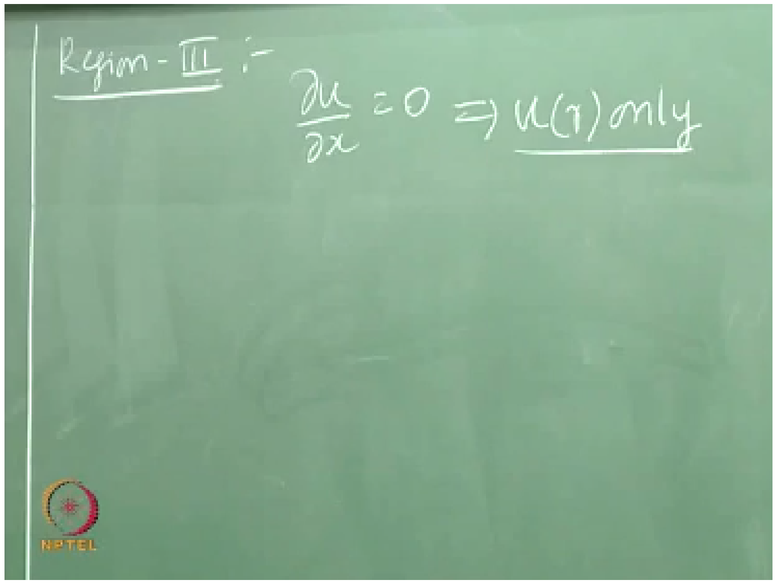


Direct exact solution to that so directly you found D u / DX is equal to 0 therefore your velocity is a function of and you are now assuming although this is a three dimensional flow you are assuming that right now the profile is two dimensional that means you do not have any dependence of velocity on the theta direction so strictly speaking you have to write down the nervier-stokes equation in the cylindrical coordinate system because this is a duct with a.

Circular cross section okay so but you are neglecting the variation of velocity and temperature with the third direction and also you are neglecting the variation with respect to the X direction so it is a function of only R okay and once you reach the fully developed condition therefore in Region three all your inertial terms get knocked off because d u / DX =zero okay and V velocity is zero correct there is no V velocity there is only axial velocity there is no V radial.

Velocity there is no azimuthal velocity so the inertial terms all disappear okay so there is no so you have your X momentum our momentum θ momentum so our momentum and θ momentum are not there right now only X momentum will be there and in X momentum on the left hand side you have D u /D X =zero V R 0 V θ =0 so there are entire inertial terms disappear so only you have your diffusion terms and your pressure gradient okay so therefore for this case you.

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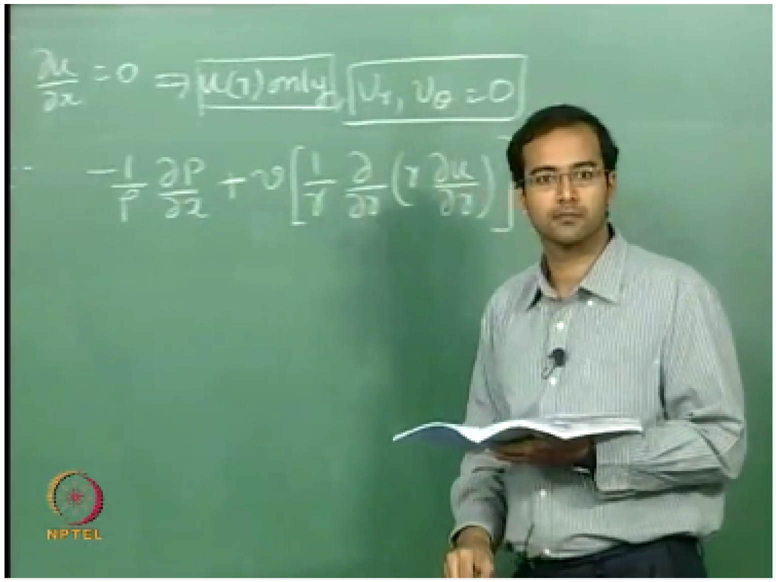


Are I will say V R and V theta these are 0 and also variation so this anyway I have said that U is a function of R 1 okay so therefore the momentum will be only X momentum so you have -1 / ρ DP / DX this is your pressure gradient term on the right hand side plus you have your diffusion term once again in the diffusion terms the θ variation since you are d u/ DX =0 so your $d^2$ u / DX $^2$ will be 0 also the variation with the direction is also not there therefore your d square u.

By D theta square is also 0 okay so the only term what is the only term then which has to be included with respect to R okay the diffusion term due to the gradient with respect to R right so that will be in the cylindrical system mu into 1 by r d by d r into r bu-by right so this is all in cylindrical system this is not cartesian in cartesian you do not have 1 by r so if you write the laplasian operator in cylindrical coordinate system you will get 1 by rd by dr into R okay and.

The shear stress in the radial Direction is d u by mu into du by D R okay so that mu divided by ρ is your kinematic viscosity here so this should be equal to 0 so this is a big approximation okay so that is why it is much simpler to look at region 3 because you are not including any of the inertial terms okay only pressure gradient term balances your your viscous forces okay so this is the very significant terms which which are important in internal plus an external flows.

For flat plate you can say that DP / DX = zero because your free stream velocity is invariant of X but here you have only these two terms and they have to balance each other so you cannot say simply that DP by DX is zero in fact the pressure gradient is driving the flow here okay so that is why internal flows are pressure driven flows okay so therefore they incur a considerable pressure drop to drive flow internal whereas external flows even if you use a fan or a blower for.

Convection so you do not have any pressure drop or very negligible whereas an internal flow internal flows pressure drop is a very important parameter because to drive the flow you need a pressure drop okay so let us write down the energy equation so the complete energy equation.

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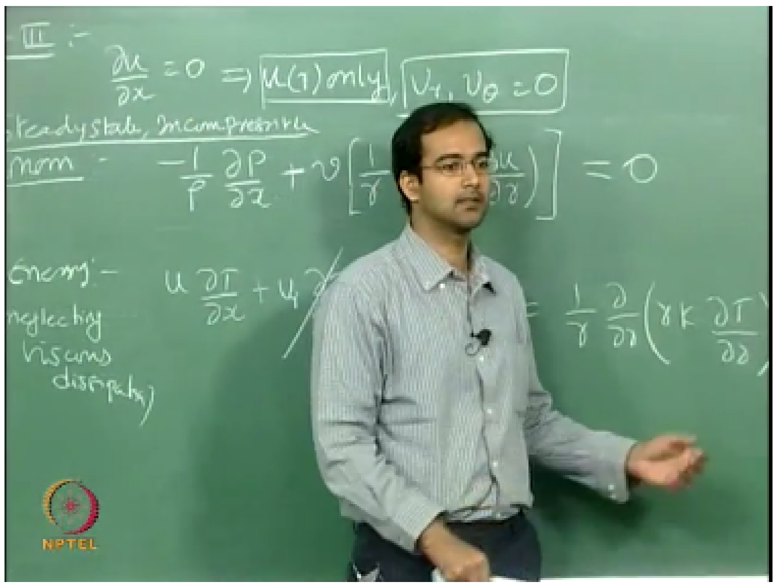

DT / DX plus V DT by D are in the cylindrical coordinate system plus okay so I call this as VR V θ / r DT / D θ this is your inertial term so I am writing everything under steady state income steady state incompressible these are the other approximations incompressible flow so this should be equal to the diffusion terms that is 1 / r d / d r into r into k dt / d r this is your diffusion term along the radial coordinate so you have 1 by rd by dr r into QR QR is K into DT.

By DL okay plus you have d by DX of K DT by DX plus one by R Square D by D theta into K into DT by DT so we are also neglecting what viscous dissipation we have included everything all the terms except the viscous dissipation term now once again for this case you can directly say since you do not have V R and V theta so therefore the convective terms involving V R and V theta are zero right and we have also said that variation of both velocity and temperature with.

Respect to the circumferential direction is zero correct so therefore you have only one convective term involving the axial velocity which is balancing your diffusion in the radial direction as well as diffusion in the axial direction this is your axial conduction this is your radial conduction but once again we will make an approximation that compared to your radial diffusion a radial conduction your axial conduction is very minor because if you look at the profiles of temperature the gradient is dominant along the r direction therefore the diffusion will be much pronounced in the radial direction so compared to the radial conduction we will also.

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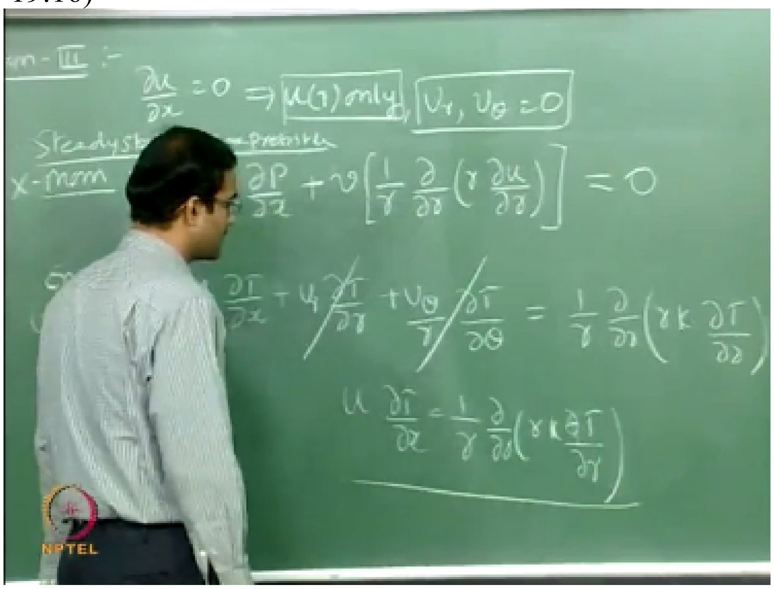


Neglect the axial conduction term so finally you are left left with u DT by DX is equal to 1 by r so 1 by rd by dr r k dt by we are so this is your energy equation in region 3 okay so now since we have written down the equations we will go ahead and start solving solving them so I will erase this okay so let us call this as equation number one and let us start with the momentum solution first see also when you write down the Y momentum equation it comes out that your.

DP / dy is approximately zero because all the other terms are negligible right so therefore P is invariant of Y it is only a function of X okay so this is the conclusion as far as your pressure is concerned it's a function of X okay so if you write the Y momentum all the inertial terms are zero all the diffusion terms are zero so that will leave with DP by dy is zero okay therefore the pressure will be only a function of X therefore you have pressure gradient only along the X .

Direction which balances the diffusion term diffusion of velocity in the radial direction so with that why is that required because now this is a constant term because this is if this has to balance this this is a function of R this is a function of X so they both have to be constants correct then only this can balance this so once this is a constant that means we can directly integrate it out so this is the simplification okay so therefore since P is a function of X only.

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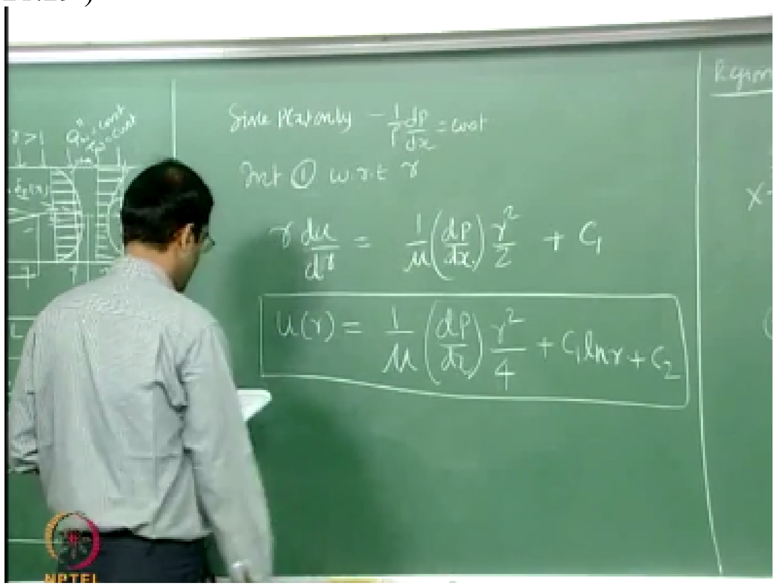

Okay so - 1 by P DP by DX should be a constant right since you know this is a function of X this is a function of R they can balance only if there they are constant therefore we can integrate one integrating one with respect to R so we get if you integrate once Rd u by D will be you have to now tell which one correct but DP by DX also has to be a constant because this is a function of P is a function of X and this is a derivative with respect to X this is the.

Derivative with respect to R and U is a function of R so these two can be equal only if they are equal to a constant that is the only possibility okay so this will be new by ρ cancels this will be 1 by 1 by **μ** DP by DX so therefore I can replace the partial derivative with an ordinary derivative differential because U is a function of only R and P is a function of X here so I can say this is R so there will be an R square by 2 right here R square by 2 plus a c1 ok so

Now if I integrate it again this will be 1 by **μ** so my R R cancels here so this will be DP by DX which is a constant and if I integrate it again this will become R square by 4 plus this is c1 by r if i integrate that that will be c1 1 of R plus c2 ok so this is my solution now I have to find the constant c1 and c2 how do I find the constant boundary conditions ok so what are the boundary conditions so you are R equal to R 0 so what is U at R equal to $R_0$ bar goes to $R_0$ and what is the other boundary condition in the case of external flows I have I use we can give a boundary condition where you are you going to very large y Y to ∞ then your U becomes.

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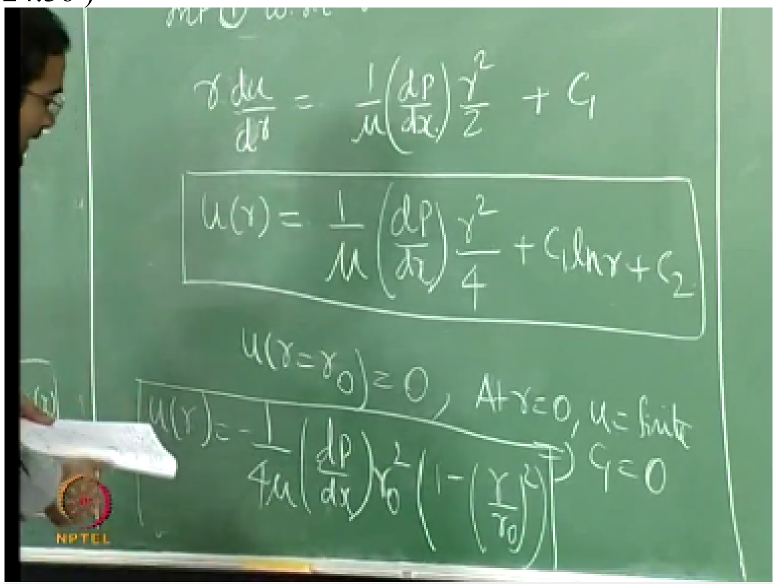


U ∞ whereas here you do not have anything like that exactly so at R=zero you should be finite now directly you can look at this equation if R goes to 0 this goes to ∞ so in order to make u finite the C 1 has to be 0 ok so directly this gives you C 1 is 0 so you can use U of our U at R equal to R 0 equal to 0 to calculate C 2 and if you do that your final velocity.

Profile comes out as okay so this is your velocity profile and you can directly see that for a given pressure gradient DP / DX for a fixed pressure gradient the variation with respect to R is parabolic okay so at R equal to R 0 this will be 1 - 1 0 and at R equal to 0 this will be 1 by 4 -1 by 4 **μ** DP / DX into $R_0^2$ ok so now we want to see what is the maximum velocity so because when we draw the profile we draw it like this and we know that we are plotting it with a maximum velocity at R equal to zero but let us show that the maximum velocity or centerline velocity is maximum velocity okay so to how do we show that how do we show the location of maximum velocity yeah .

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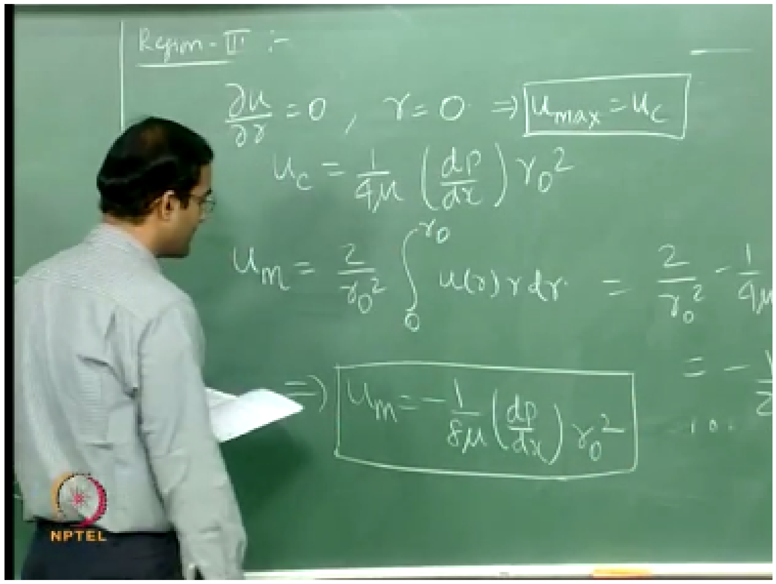


So you take the derivative D u with respect to D are equated to zero and you will find that our comes r out as zero so that means the center line is the inflection point okay so there could be a maxima or minima of course there has to be a maximum so this this is the region where so u max is equal to u centerline okay the centerline velocity is the maximum velocity and the value of the maximum velocity here is 1 / 4 **μ** DP / DX R 0 square this is the value of maximum.

Velocity so now we will go ahead and calculate the other the other is the mean velocity which is 2 by r 0 square integral 0 to r 0 u of r RDR ok so if you substitute the velocity profile and integrate it or not square - 1 by 4 mu DP by DX these are all constant terms which can be pulled out outside the integral 0 to r 0 1 - r by r 0 the whole square into r near okay so if you integrate it out you will get - 1 by 2 **μ** DP by DX into R.

$0^2$ by 4 so which can be simplified to UM is - 1 by 8 **μ** DP by DX into R 30 square so therefore you see um is a it is now a fixed value in the fully developed region because DP by DX is a constant so everything is constant so therefore um is a constant value in a fully developed region okay so so this is your expression for um we also found out expression for you see therefore you can also write you are you are you of are as this entire thing is nothing but you see okay and you can see from this expression that your centerline velocity is actually twice of the mean velocity right so so therefore you can express your you as.

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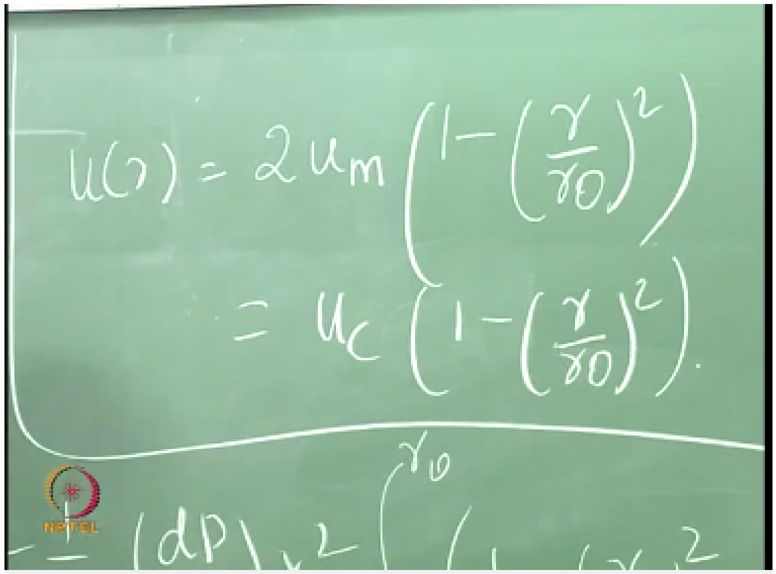


Twice of um into 1 - R by R not the whole square which is also u centerline's 1 - R by R not the whole square ok so so this is this is how you our general velocity profile is written as a variation in terms of R is it clear since you have already done this I am not spending too much of time you can just go and look into any fluid mechanics textbook and you will find the derivation so any any questions so far so you can clearly see that at center line you have R equal to R 0 you have your centerline and from there you have a parabolic velocity profile distribution okay so therefore we will find out other integral quantities the important integral.

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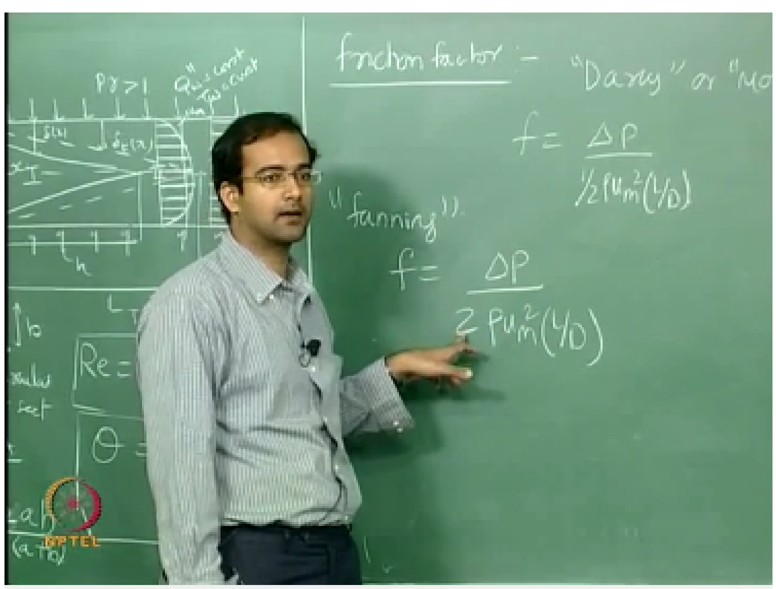


Quantity will define an internal flows is called the friction factor okay in external flows we define the wall shear stress and from there we calculated a non-dimensional skin friction coefficient CF okay the same way we can also define a skin friction coefficient okay but generally people refer to as we can we will show that we will show that the equivalent of a skin Friction coefficient is your Fanning's friction factor okay so there are two different frictions

Factors which are defined so one which is more popular is your Darcy or your Moody friction factor so named after these two people who have defined it and it is defined as F is equal to**Δ** P by half ρ um square L by D because here your L by D is also an important factor so as far as your pressure drop is concerned because DP by DX is nothing but **Δ** P by L so that has to be included as it is okay so this is how your Moody's friction factor is defined the other

Friction factor is also called Fanning's friction factor and he defines it so that is **Δ** p by twice ρ u M square L / D so you see the basic form is the same so only the factor here they use a factor of half and here finding uses a factor of two and you will see why Fanning defined it this way is that for the fully developed internal flows okay so the Fanning friction factor exactly comes out to be the same as skin friction coefficient okay so therefore you defined it in a convenient form

Where you can replace the skin friction coefficient which you which you are used to an external flows and that will become equal to your friction factor in the internal flows okay so this can also be written as - DP by DX into D by twice ρ u M square right because **Δ** P by L is nothing but - DP by DX this is a pressure gradient so since this is the pressure gradient keeps so your pressure changes this is decreasing therefore if you take a pressure gradient will

Be negative so therefore you have to put a negative sign here okay so this is the way that you write it and you can now substitute for DP by DX from this expression in terms of so you are let me call this as equation number 2 from Equation number 2 you can write DP by DX in terms of un correct and you can substitute that here so you will find that some of the terms um terms will cancel off so if you substitute that this will be 8 nu into um divided by R 0 square you

Have to ρ um square okay so this should come out as 8 mu / ρ um or not okay so the um-um cancels here and okay so this is a D here so D is twice or not right and now you can define an ass number okay so this is ρ um and this you can write as D 0 by two okay so this will become 16 divided by Reynolds number similarly if you had substituted here for the Moody's friction factor this expression will come out as 64 by Reynolds number okay so that is the

Difference the factor is the difference and now if you define your skin friction coefficient CF for internal flows as tau by half ρ um square okay and your tau is nothing but - nu D u by D R at R equal to R 0 divided by half P um square so you can calculate you have the velocity profile you can calculate the derivative with respect to R and you have to calculate derivative at R equal to R 0 and you substitute it so you will get an expression for CF

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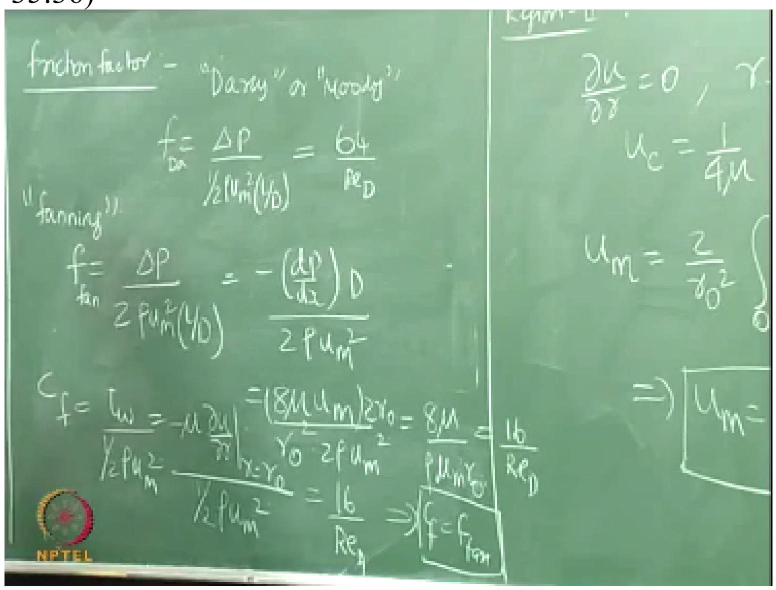


Which also comes out to be exactly identical to 16 / re D so that is why the way the Fanning friction factor was defined both Fanning friction factor and skin friction coefficient become identical okay so therefore for the fully developed case you have CF is equal to u RF let me call this as f Fanning here to differentiate it from f Darcy so this is a another important Result that you should remember okay now if you plot the friction factor I think all of you know

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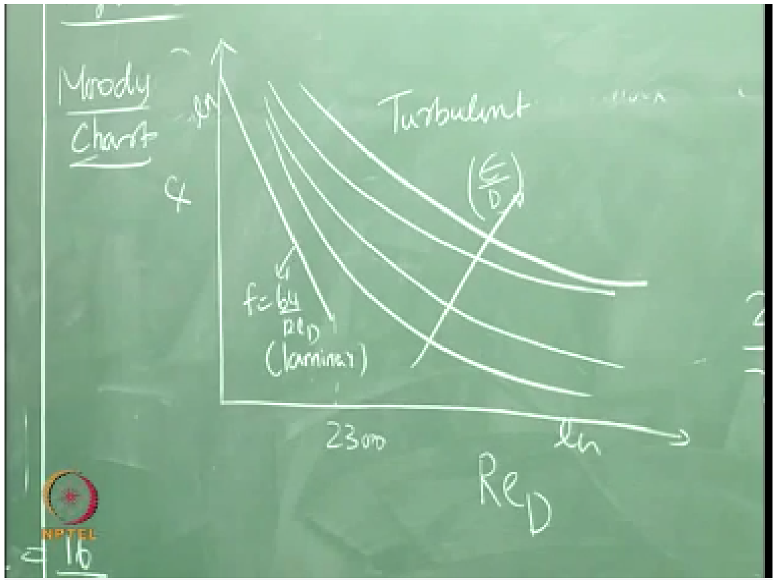


How it looks so this is called the Moody's chart right I hope all of you remember the Moody's chart in fluid mechanics that is the internal force that is a single most important thing that you Might have learnt okay if you brought half as a function of re D okay now remember if you plot on a log-log scale now this is this is a hyperbola right you have if say if Darcy F into re D is equal to 64 s a rectangular hyperbola so if you plot on a log-log scale this becomes linear okay

So on a log-log scale if you plot a log-log scale so this will look so with your increasing Reynolds number the friction factor comes down right so this is your relationship F is equal to 64 by re D and this is valid for certain Reynolds number till what critical Reynolds number you can say 2300 so once critical Reynolds number exceeds the flow becomes turbulent and the variation will not be as straightforward as this it becomes completely different and in the

Turbulent region you will find it is not a single line but you will have several multiple lines like this okay so what these multiple lines indicate is the friction factor is not only a function of Reynolds number but it's also a function of the roughness okay so you plot what is called non-dimensional roughness that is roughness height divided by the diameter of the tube so with increasing roughness your friction factor increases so this is completely turbulent whereas in

The laminar flow you find a very linearly decreasing profile if you plot it on a log-log scale so this is called the Darcy or Moody chart from which you can pick up for a corresponding value of Reynolds number what is the exact value of the friction factor okay I think all of you are familiar with this in the case of non circular ducts okay this is valid for a circular duct okay if you go to non circular ducts so your friction factor f becomes 64 by some factor fie I am sorry.

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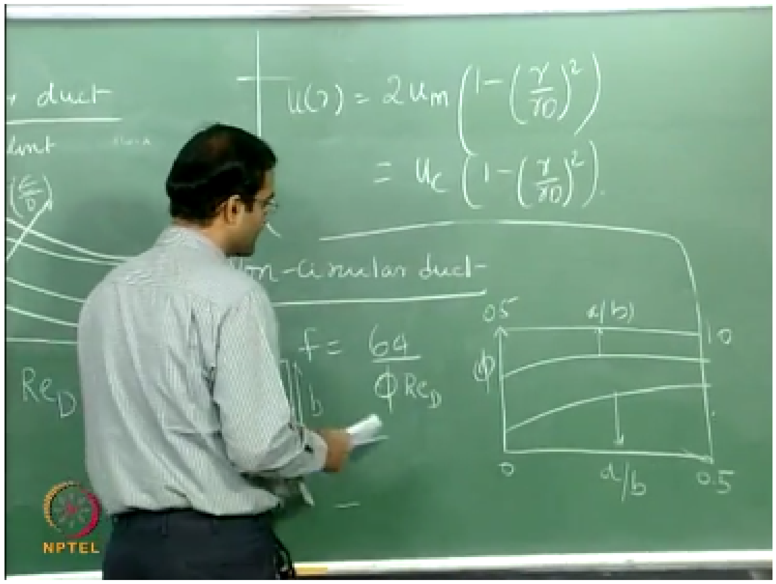


It is not 64/ re 64 by re is only for circular ducts because we have seen that we have substituted all this relation for circular ducts but generally for non circular ducts you have an additional Factor fee and this fee will be a function of the aspect ratio so if you have a non circular cross-section okay you have aspect ratio a over B okay so it is a function of this aspect ratio suppose you take a rectangular cross-section you can plot V as a function of a by B so say

Is a by B going from 0 to say 0.5 and on the top scale say from 0.5 to 1 so you have a variation something like this corresponding to this and you have a variation another variation corresponding to a by B of 0.5 to 1 so you have typically variation of V for a smaller range of a by b 0 to 0.5 you have one variation for a by be varying between 0.5 and 1 so the limiting value 1f becomes 1 basically okay so then it becomes equal to a circular cross a square cross-section ok

So then you have the value of V at somewhat approaching to 1 and you are a by B approaches 1 okay so therefore you have to know this factor and then you have to plug it for a non circular cross section with the different aspect ratio and that will give you the corresponding friction factor so anyway these are some additional details which you can find in text books so next what we what we are going to do is take the heat transfer problem so very.

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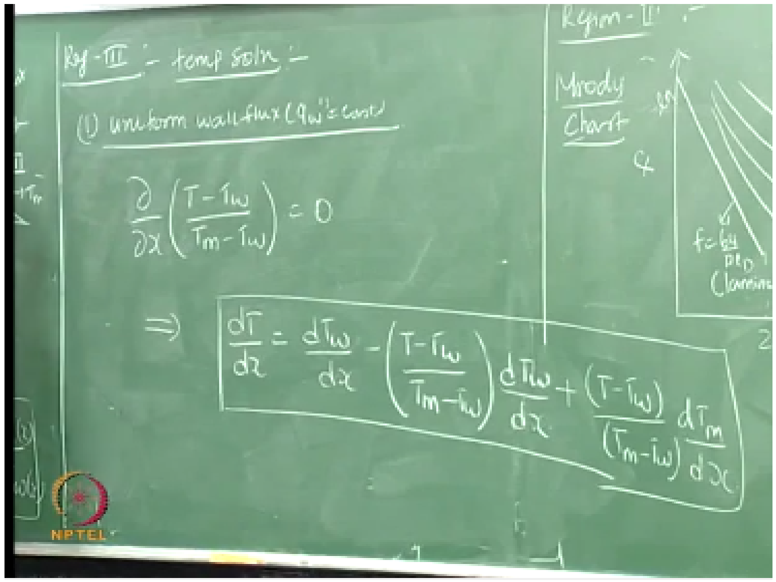


Quickly in another 5-10 minutes and see how we can find the temperature profiles in the thermally fully developed region okay now we know how the velocity profile looks so now in Region three temperature solution so once you get the velocity solution for velocity profile naturally the next part will be to get the solution for the temperature and we have already I already told you that the heat transfer coefficient is a constant in Region three so we have to know what is the value of this constant so therefore first we look at a boundary condition

Which will be the uniform wall flux that is Q w double prime this is the constant okay now we have already defined my D /DX of theta and theta is t - T wall by t - TM - T wall this is equal to zero for thermally fully-developed case so we will expand upon this term okay so if you integrate if you differentiate it by parts okay how so what do you get so this is VD u - u DV by v square right so if you differentiate and expand it you can just try I will.

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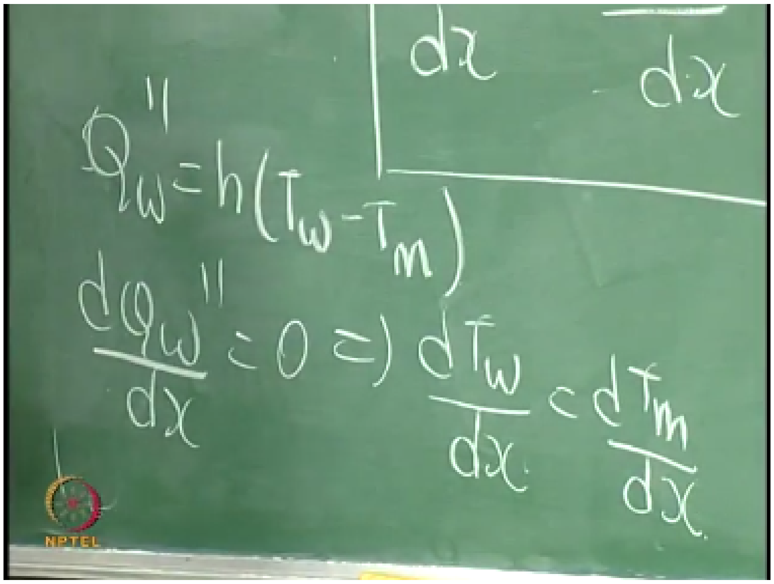

Give you some time you should be getting an expression like this so all of you please try expanding this differential here okay so I maybe you can complete it as a homework but you should be able to get very straightforward you should be able to get this now the important factors when you apply the uniform heat flux boundary condition from the Newton's law of cooling you can say that Q R double prime is H into **δ** T P wall -**δ** T mean so since your.

Wall flux is uniform therefore DQ /DX will be 0 right there is no variation with respect to X and therefore your this will give that DT w / DX should be equal to DT mean by DX very important conclusion as it because coming from the fact that you have a uniform heat flux so therefore if you look at the profiles if you substitute let us call this as now number one this is number two if you substitute this conclusion that is DT w/ DX should be equal to DT m / DX what.

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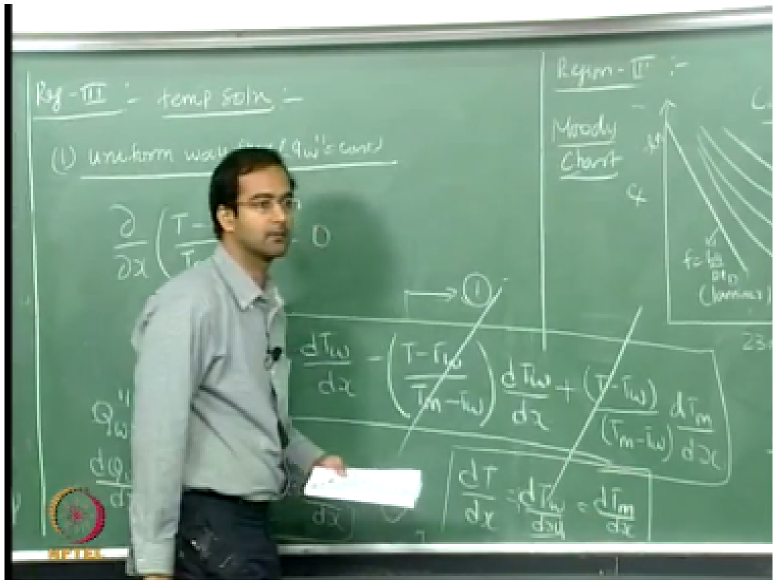


Happens these two terms will cancel out so therefore you directly get the fact also that DT /DX should be equal to DT w /DX which in fact is equal to DT m / DX okay so the slopes of all the three profile like we are any temperature at any location radial location your wall temperature and your mean temperature they are identical and it can be true only so how how should the profile look if you plot your temperature on the y axis and X if I want to show.

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Temperature at any radial location wall temperature and mean temperature and if the slopes have to be the same it has to be exactly parallel all the three have to be parallel and in fact we will see that this has to be just linear okay so you can say then this is your t wall okay this is maybe T at some radial location and say this is your tea me so this is a function of X this is a function of R and X this is a function of X but all of them in the region three will be linear okay.

And the slopes are identical okay so this is a very important conclusion which characterizes a fully developed flow with a constant heat flux boundary condition so therefore if you apply this fact to the energy equation okay so what was the energy equation that we had written we had written if u u / alpha DT / DX okay there is one more thing I forgot ρ CP on the left hand side of the convective terms in the when you wrote the energy equation there should be ρ CP.

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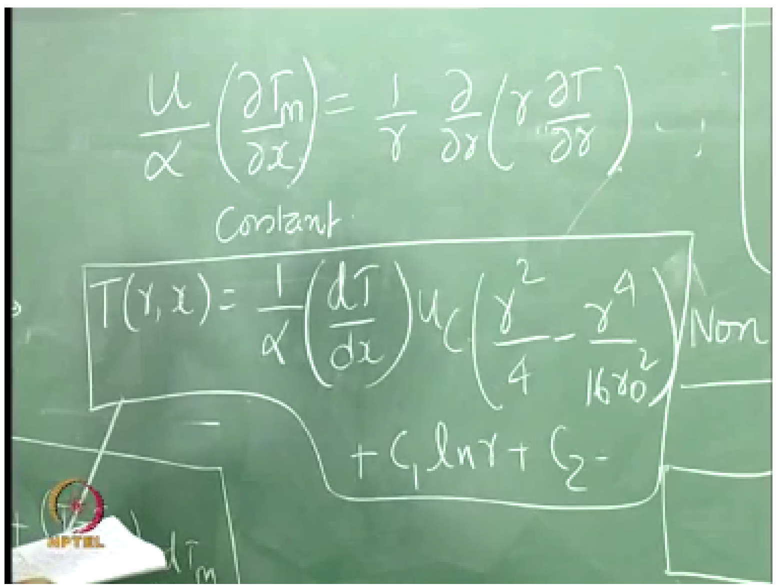


Okay on the right hand side you have K okay so I am taking P CP by K which is 1 by alpha DT by DX should be equal to 1 by r d / dr r dt / dr now the fact i am going to use here is my dt by DX is exactly identical to dt / DX so these are all just a constant ok so this can this can be through only if they are all constant because T is a function of both R and X if this slope has to be equal to this and this this has to be equal to X DT wall by DX the slope has to be a constant.

Value and therefore it is a linear variation okay so this is an important fact you can just replace this as DT mean by DX now this is a constant so therefore you can see that you can now integrate this equation with respect to R just like the way you integrated the velocity profile because now this term is a constant and therefore this does not become a partial this can be written as an ordinary differential equation okay so I will just give you the result of the integration and we.

Will stop there so if you integrate this equation you will get solution for T as a function of DT DT m / DX 1 /alpha DT / DX and you have to also substitute the parabolic velocity profile which you obtain from the fully developed hydrodynamic solution so then you integrate it with respect to R so you get so this is your temperature profile so we will stop here for today so. Tomorrow we will start continue from here apply the boundary conditions and calculate the constants and also similar to calculating integral quantities for velocity like skin friction coefficient and friction factor we will calculate the heat flux and the heat flux anyways the constant will calculate the heat transfer coefficient and we will know what what exactly the constant values.

**Hydrodynamically and thermally fully**

**Developed internal laminar flows**

**End of lecture 26**

**Next:Fullydeveloped laminar internal flow and heat transfer**

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