Fluid Mechanics

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Lecture No.# 17

Welcome to this lecture number 17 on the NP-TEL course on fluid mechanics for undergraduate chemical engineering students. In lecture number16 we started discussing the integral energy balance.

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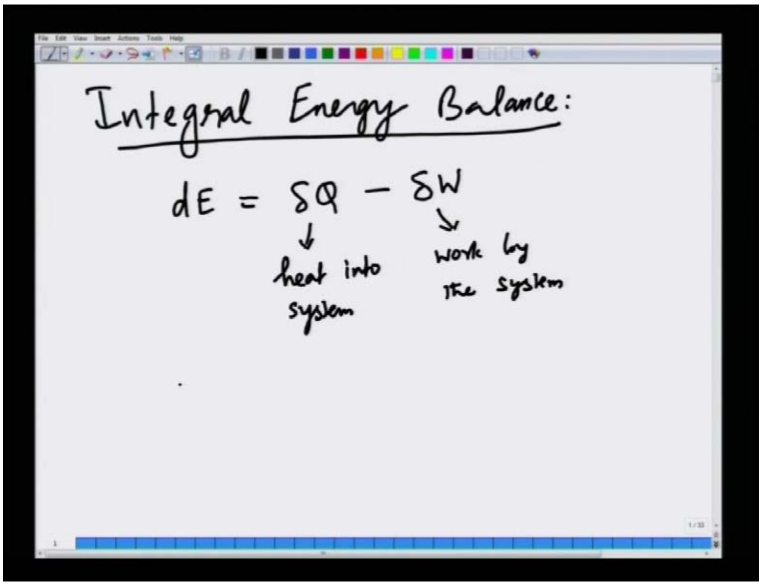

So, let me briefly recapitulate what we did before proceeding further. In order to derive the integral energy balance the underline fundamental principle is the first law of thermodynamics, which says that the change in total energy of a system is equal to the amount of heat transferred to the system, minus the amount of work done by the system. So, this is heat into the system, this is work by the system. So, this is the sign convention that we use, that the work done by the system on the surroundings is positive and the work done by the surroundings on the system is negative. That is the sign convention that is normally used in engineering thermodynamics. So, we also pointed out that in engineering applications you will always have continuously flowing systems.

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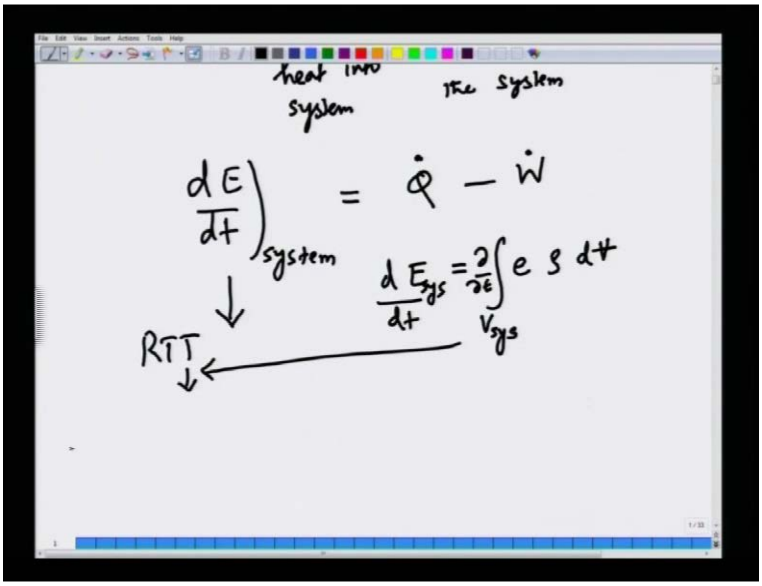

So, it is good to rewrite the first law of thermodynamics in a rate form, which says that the rate at which the energy of a system changes is equal to the rate of the heat transferred to the system, minus rate at which work is done by the system on the surroundings. So, Q dot means it is a rate at which heat is transferred to the system and W dot is the rate at which work is done by the system on the surroundings. Now, in order to convert this to a control volume kind of an approach, control volume kind of a formulation this gives, we have to use the Reynolds transport theorem as usual. Which will give and before we do that we realize that the energy of the system is equal to the volume of the system of the specific energy, times which is energy per unit mass, times density which is mass per unit volume integrated over the volume of the system. So, we want d d t of this, so which is d d t of this.

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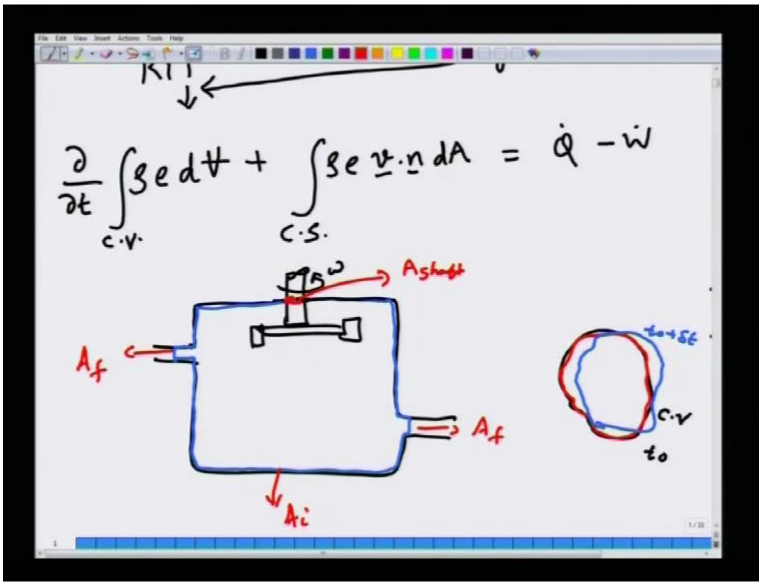

In order to do this we have to use the Reynolds transport theorem which will tell us that this is nothing but the rate of change of energy of the control volume. Remember that when we use the Reynolds transport theorem, we always imagine that you have a C V, at sometime t0. The system and C V coincide, the system and the C V are exactly the same at time naught. But C V is fixed in space, its independent of time but the system which I am marking in red and blue at a later time, blue is at a later time, t 0 plus delta t the system would have moved out of the C V by virtue of flow.

And the rate of change that we are talking about of the system is that, at instant at that particular instant of t 0what is the rate of change of energy of the system, and that is given by Reynolds transport theorem in terms of the variables pertaining to the control volume as follows. Rho e av plus integral over the control surface Rho e V dot n d S or dA. If you have to use dA let us keep using d a, is equal to Q dot minus W dot. In order to understand various contributions to the work interactions between the control volume and the surroundings,

we imagined in the last lecture drawing a general kind of an engineering equipment. Which is typical in many fluid mechanics applications. Where you had a shaft and the shaft has a rotor which could be an impeller, which is rotating at some angular velocity.

And the system sorry the control volume is the blue line and we want to write the balance about this control volume and you want to understand what are the work interactions. Now there are three types of areas as we said in the last lecture, the internal areas Ai, where there is the velocity of the fluid is 0then these areas through which flow is happening, A f at the inlet and outlet and this area over which the control surface is cutting across the rotating shafts that can be procured out of the C V to the surroundings.

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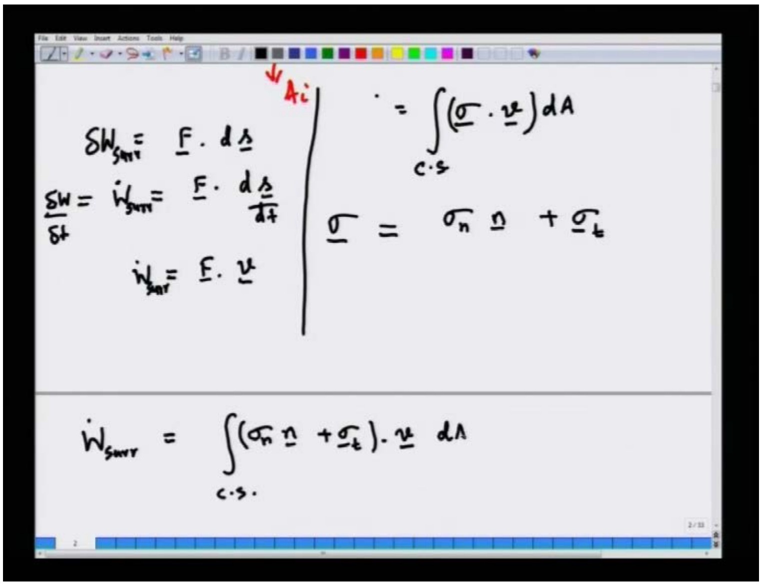

So, there are three types of areas, the work interactions as we said last lecture is simply force dotted with displacement. The rate at which work is done by this. So, first we discuss the rate at which work is done by the C V on the system and then we took a negative sign. So, rate at which work is done is delta W by delta t so ds by d t is the velocity. So, rate at which work is done is F dot v. Now, all we know in fluid mechanics is the stress vector. So, W dot is integral of the stress. Stress is force per unit area as we have been mentioning, over the entire control surface area. Now, the entire control surface area has three contribute types of areas one is Ai A f and A shaft.

So, we have to split this integral into three areas. Now, before we do that we also said that the total stress vector can be written as the normal component plus the tangential component .So, this is work done by the C V on the system. So, let me just put it as WC

V sorry work by the surroundings on the C V. So, let us just write this as delta W surroundings. When we want the work done by the C V on the surroundings we will take the negative signs. So, let me just do this right now. So, rate at which work is done by the surroundings on the C V is nothing but integral sigma n n over all the control surfaces plus sigma t dot v dA integrated over all the area.

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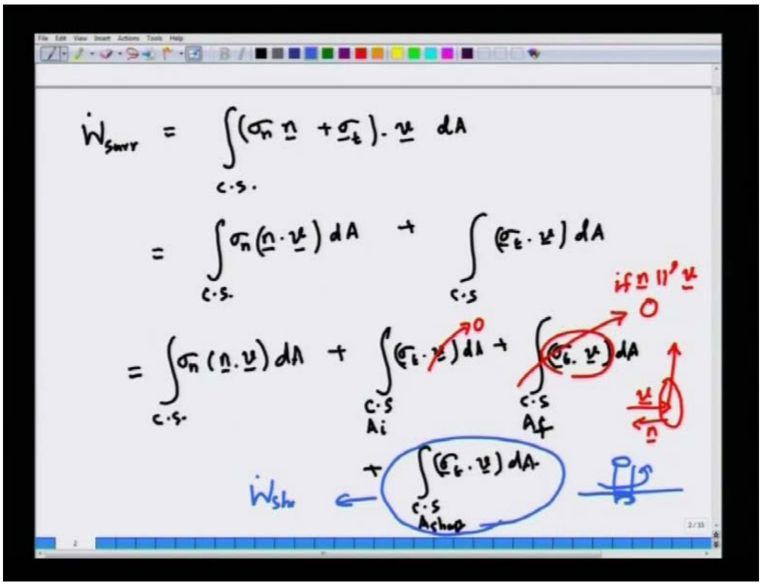

So, let us split the normal and tangential component separately, integral over C S sigma n n dot v, dA plus integral over C S sigma t dot v integrated over dA. Now, sigma n is now, let us now split this further. Sigma n n dot v dA plus. Now, let us split this surface area into, control surface area into control surface over A internal areas dA plus control surface over flow areas dA, plus control surface over shaft areas dA. Now, if you look at the internal areas, the velocity at solid surface is of the control surface is 0, so this contribution is 0. If we look at the control surfaces over the flow areas, this can be, this is let say the flow.

This is the direction of velocity, if you choose the control surface perpendicular to the unit normal to the control surface perpendicular to the flow direction, then the tangential stresses will act in the direction perpendicular to the velocity. So, these two vectors are orthogonal. So, this can be set to 0. If n is perpendicular to v. If the unit sorry, if the unit If v is perpendicular to the control surface, that is if n is parallel to v that means that the velocity vector and the shear vector are orthogonal to each other so that is 0. Now, this contribution is not 0 because whenever there is a rotating element and the C S is cutting across it there is a stress and then there is a velocity in the direction of stress. So, this is the shaft work which is denoted as W dot shaft, rate at which shaft work is done.

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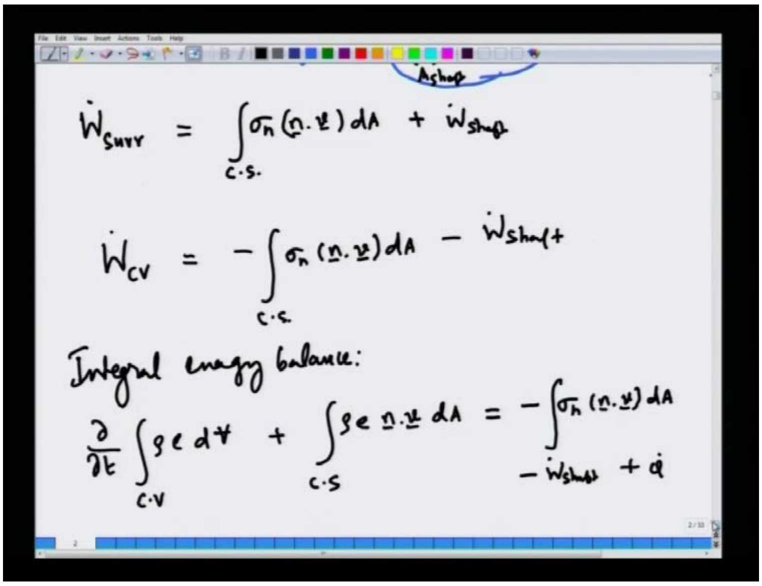

So, rate at which shaft work is done by the surroundings on C V. So, we are still looking at the rate at which work is done by the surroundings on the C V. So, this is integral over C S sigma n n dot v, dA plus W shaft. Now, rate at which, but the quantity that we are interested in is rate at which work is done by C V on the surroundings. It is minus integral C S sigma n n dot v, dA plusor minus because it is negative of the previous expression W shaft.

Therefore, the integral energy balance becomes d d t of integral rho e av integral over control volume, plus integral over control surface rho e n dot v dA is equal to minus integral sigma n, n dot v dA minus W shaft, plus Q dot. So, this is the simplification that we get after we use the fact that the shears tresses the rate at which the work is done by the shear stresses are 0 on the internal areas, and as well as the fact that the shear stresses cannot do work at the inlet and exit as long as you choose the control surface, such that it is perpendicular to the inlet velocity vector. So, that the velocity vector and shear stresses vector are perpendicular to each other.

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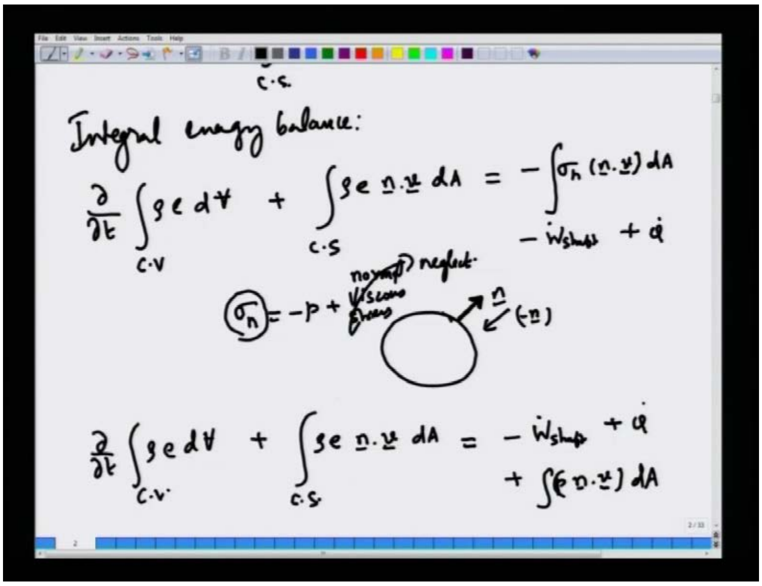

So, finally we get, but you also know that sigma n is minus p. So, if you take, if you remember our very first discussion on hydrostatics. If n is pointing out wards then pressure is pointing in the direction of minus n. So, this is the normal component of the stress but pressure is directing and acting in the direction of minus n. So, sigma n is minus p, so but there are also we pointed out in the last lecturer that there are also normal viscous stresses. Which are usually negligible in liquids like, fluids like ariel water.

So, we get the following expression d d t of integral rho ea v, C V plus integral rho e, n dot V, dA, C S is equal to minus W dot shaft plus Q, plus integral p, n dot v, d A. Now, we can pull this, both these expressions haven dot V times A.

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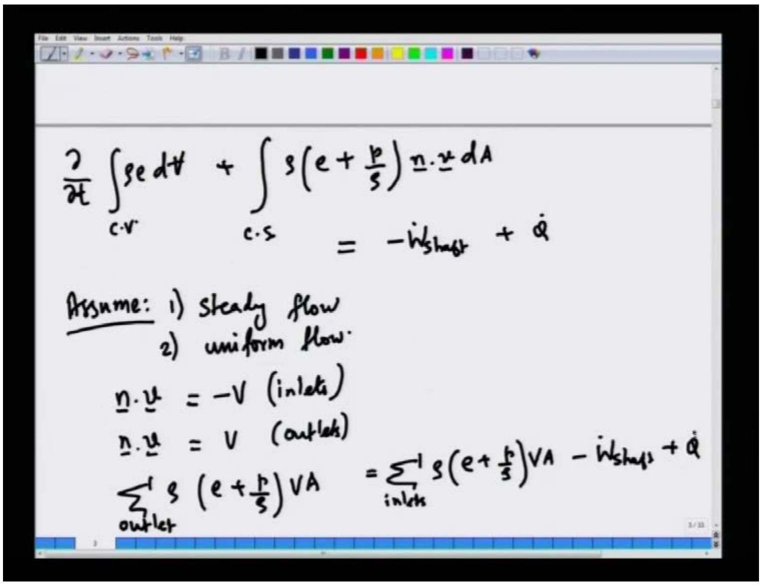

So, we can pull this to the other side to write rho e av, C V integrated over C V plus integral over C S, rho times e plus p by rho n dot V dA is minus W shaft plus Q dot. So, this is our final simplified energy balance. We will further simplify it by assuming. Now, we will assume certain things, that it is uniform flow, first you will assume steady flow, uniform flow where the properties are uniform across the inlets and outlets of the control surface.

So, that the integrals can be simplified, we will of course relax this assumption little later. And so we will derive try to simplify this energy equation for the special case of uniform and steady flows, wherein you will get summation and when we do this we should also realize that n dot v is minus V at inlets and n do tv is plus V at outlets. So, the integral over the series will become summation over outlet, rho e plus p by rho times V dA is summation over inlets rho e plus p by rho, not V dA just V a minus, W shaft plus Q dot.

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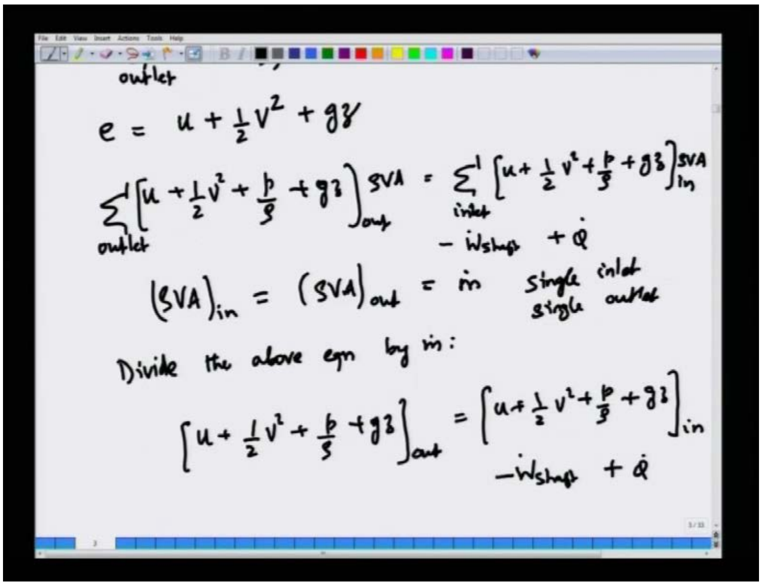

So, you remember that the specific energy, specific total energy per unit mass is the internal energy plus per unit mass plus the plus kinetic energy per unit mass plus kinetic energy per I am sorry the kinetic energy per unit mass plus he potential energy per unit mass. So, e is u plus half V square plus g z. That is something that we should keep at back of our mind. So, we will write summation over outlets u plus half V square plus p by rho plus g z, times rho at outlet at rho V A at outlet is summation over inlet u plus half V square plus p by rho plus g z at inlets, rho V A minus W dot shaft plus Q.

Now, rho V A for a steady flow at the inlet should be equal to rho V A at outlets. This is mass conservation, if there is only one inlet and one out let then you know that, the rho V A at inlet is rho V A outlet minus m dot. S, we can divide the entire which is the mass flow rate. Divide the entire equation by above equation by the mass flow rate to give u plus half V square plus p by rho plus g z at outlets. So, let just specialize to the case of single inlet and single outlet. Out is u plus half V square plus p by rho plus g z in minus W dot shaft plus Q dot.

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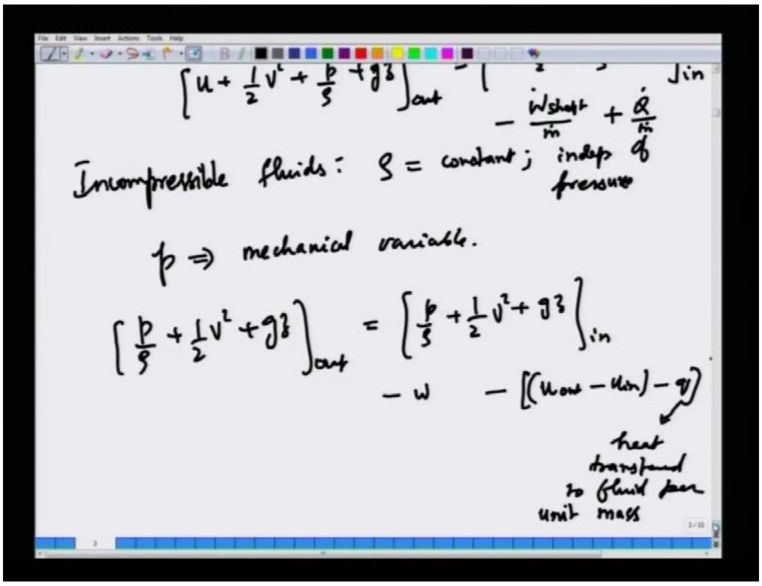

Now, this is the most general energy balance, this is the special form for a energy balance for a steady flow uniform flow also with single inlets and outlets. If we restrict ourselves to incompressible fluids. Incompressible fluids are fluids, where as we are said sometime back a rho is constant that is it is independent of pressure. The pressure changes that are associated with the fluid flow does not change the density of the fluid flow. Then the fluid is said to be incompressible, in such a case the pressure p is purely a mechanical variable. It is not a thermodynamic variable. So, people normally write, take u where u the internally is a thermodynamic variable, so take u to the other side to write this as, p by rho half V square plus g z at outlet is p by rho plus half V square plus g z at inlet, minus W dot shaft minus u out.

So, we are trying to divide the entire equation by W dot. So, you will get W dot shaft by m dot Q dot by m dot. Now W dot shaft by m dot is nothing but small w. It is the work done by the C V per the unit mass of the fluid, there is no rate because we are dividing one rate with another. This is the rate work done by the C V on the surroundings. This is the rate of mass flow, the two rates will cancel of to give the rate at which the work is done by the C V on the surroundings per unit mass of the fluid. And you will get minus u out minus u in. Minus u sorry you have already taken the minus sign out, where small Q is the rate at which heat is transferred to the fluid per unit mass. There is no rate sorry there is no rate the heat transferred to fluid per unit mass. There is no rate involved with here.

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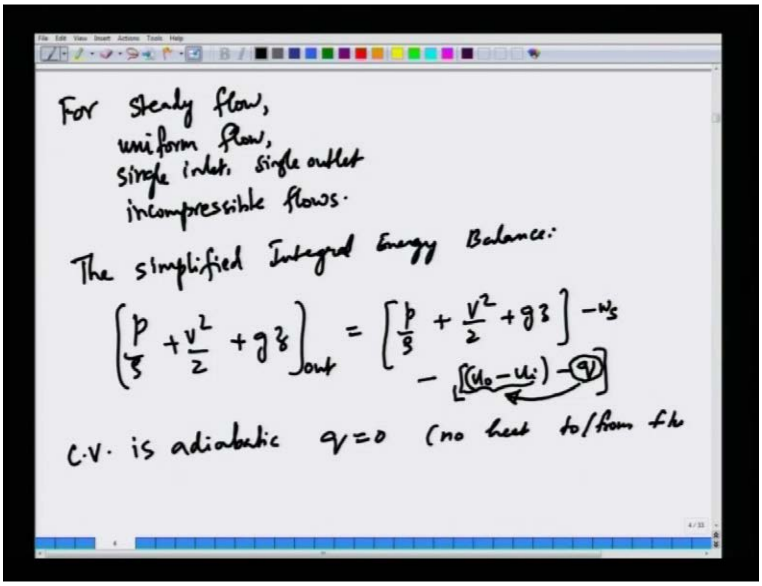

So, finally we have a simplified equation for steady flow, uniform flow that is the flow property such as velocity, density, internal energy there are in variant with the cross section at inlets and outlets. Steady flow, uniform flow, single inlet and single outlet and finally, we also have incompressible flows. The simplified energy balance, integral energy balance becomes simply. So, let us start with p by rho that is conventional V square by 2plus g z. So, I can start with p by rho here also, p by rho plus V square by 2 plus g z minus W s. The rate at which shaft work done by sorry the amount of shaft work done by the fluid work by the C V per unit mass of the fluid, minus u out minus u in minus q.

So, what we are saying essentially is that, for an incompressible fluid the heat transferred can affect only the internal energy for example, the heat that we are putting in cannot go into increasing any of these. That is why these two terms are club together. But suppose for the moment let us say that the C V is adiabatic. That is when the C V is adiabatic q is 0, no heat is transfer to or from the fluid.

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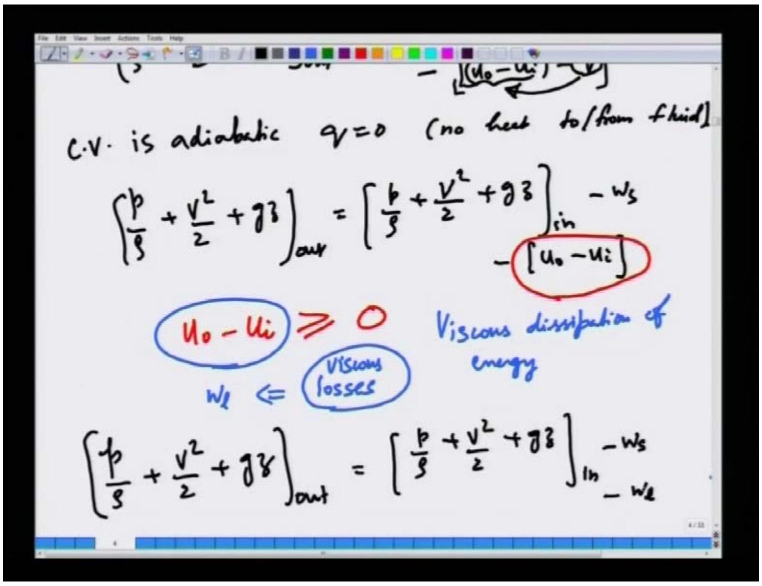

So, you get p by rho plus half V square plus g z at outlet is p by rho plus half V square plus g z at inlet, minus W s minus u out minus u in. Now, there is no heat transfer. Now, the question that we can ask is, will this term be 0 or non 0. There is no heat transfer whenever there is a fluid flow. So, it turns out that whenever there is a fluid flow happening within pipes or you know in equipment like pumps or compressors, there is always an irreversible transfer or conversion of macroscopic mechanical energy to internal energy.

So, this is fact that whenever there is a fluid flow the viscous action of the flow of the fluid, the viscous nature of the fluid always dissipates energy from macroscopic motions to microscopic degrees of freedom. Which is essentially the internal energy of the fluid. So, u 0 minus ui is always greater than or equal to 0. The equality happens when you consider a fluid with 0 viscosity. Of course there is no real fluid with 0 viscosity, if you consider a hypothetical or ideal fluid that has 0 viscosity, then you will have no conversion of mechanical energy to internal energy. This is called the viscous dissipation of energy.

So, this term will always be greater than or equal or to 0. Now in the context of fluid flows. This is the energy that is lost so these are termed as losses, viscous losses because the macroscopic energy that you have supplied by way of kinetic energy or gravitation potential energy or the pressure has been lost to internal energy. These are termed simply as viscous losses. So, in engineering fluid mechanics we will simply for incompressible flows, with all the assumptions of steady uniform flow one inlet and one outlet and incompressible fluids. The energy equation is normally written as plus g z at inlet minus W s, this is the shaft work done per unit mass of the fluid by the C V on the surroundings minus W loss.

This is the amount of energy lost to internal energy by the viscous dissipation action. So, this is a new, we have to tell what the losses are in order to solve a problem. But we certainly know that there are losses. How these losses are computed for simple systems, we will come to a little latter when we do differential balance of momentum. But right now we know for sure that losses are there. Whenever there is a fluid flow there will be loss of energy from macroscopic form, be it pressure head or kinetic energy ahead or the viscous loss sorry or the gravitational potential energy in any form to the integral degrees of freedom which is the internal energy. But if you are looking everything from a macroscopic point of view, that is energy that is lost from the macroscopic side, so we just term it as viscous losses. So, normally then therefore, we write the internal energy balance like this. Sometimes in engineering fluid mechanics.

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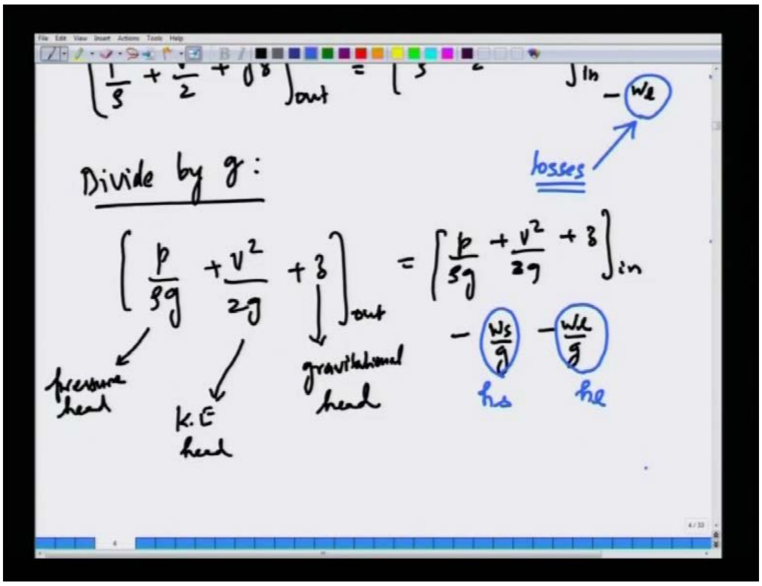

you divide this entire b y g, acceleration due to gravity to get p by rho g plus V square by 2 g plus z at outlet, is p by rho g plus V square by 2 g plus z at inlet minus W s by g minus W l by g. Now if you look at this equation, each term has a dimensions of length.

So, historically or traditionally people worried everything, people thought about various energies in terms of pressure heads. So, this the gravitational head. So, this is called the gravitational head because it has dimensions of length, a head called gravitational head. This is the kinetic energy head, this is the pressure head. Now, W s by g is denoted as h s, Wl by g is denoted as h l. There is no new information in this equation except that many times you see text books discuss this form of energy balance also. Where every quantity is represented in dimensions of length which essentially represents height of column of liquid in the gravitational potential, gravitational potential energy sense.

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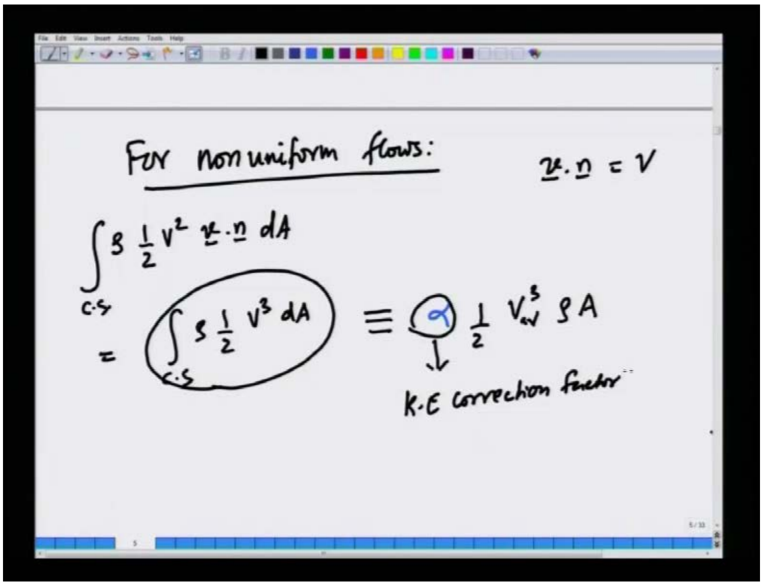

So, that is why all these terms are defined in term the form of heads. Now that is the simplest form of energy balance. As we told in the last lecture that if for the non uniform flows, that is where the velocities are a function of the cross section area. Then it is just as we introduced the momentum correction factor in the last lecture. For the flux terms in the energy balance you may have to carry out this integral. So, you have half, so if you recall you have a term of the type rho half V square, v dot n dA over the control surface.

So, this is essentially if you are thinking about outlet v dot n is positive. So, if you are considering about inlet v dot n will be negative V. But essentially you will have to evaluate an integral of the form half V cube dA, where V is the function of the cross sectional area, now you want to be able to write this in the form of a new correction factor. Alpha times half V average cubed times rho V A ism dot.

So, let us write this as sorry times rho A. So, we will just write rho A. We want to be able to write, instead of doing this integral. We want to write this integral as though if it is a uniform flow. Then the answer will be half V cubed times rho A. Now, we substitute V by V average because the flow is non uniform. But you also introduce a correction factor, this is called the kinetic energy correction factor. So, in order to find what this kinetic energy correction factor is, we have to simply…

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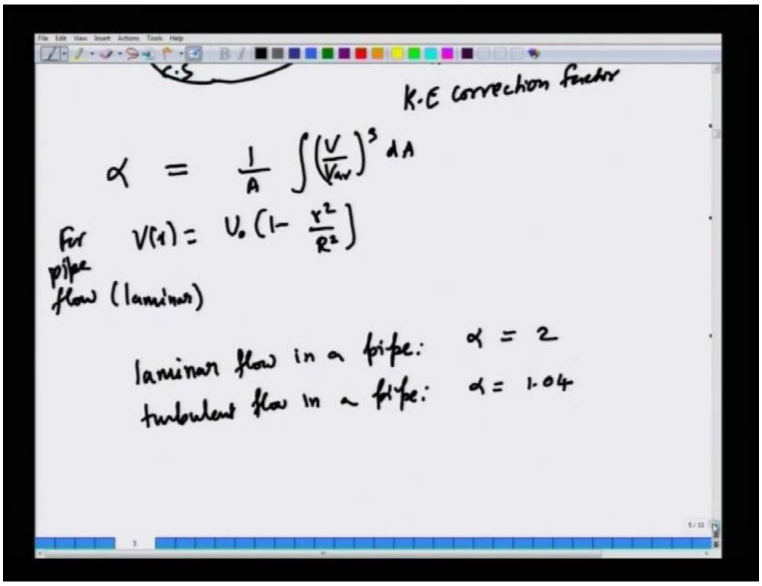

So alpha is equal to one over area u by V average whole cube dA. So, if we use V for a pipe flow, laminar pipe flow laminar. So, V of r is some maximum velocity times one minus r square by R square. So we can calculate what is V average and then we can calculate alpha. So, for laminar flow in a pipe, alpha becomes2.Whereas, for turbulent flow in a pipe alpha is simply 1.04. So, you could for non uniform flows use the correction f actor kinetic energy correction factor.

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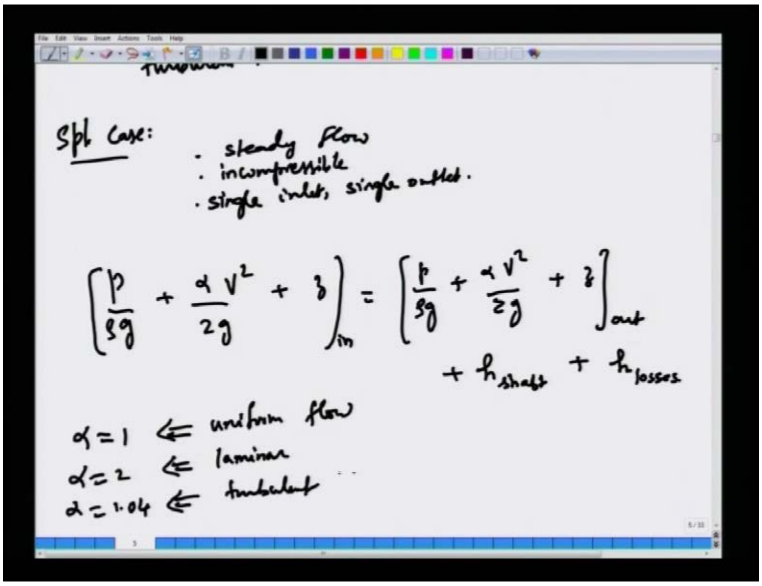

So, for the simplest case special case of steady flow, uniform flow, incompressible fluid and then single inlet and single outlet. You get p by rho g plus alpha V square by 2 g plus z at inlet, is p by rho g plus alpha V square by 2 g plus z at outlet plus the head due to shaft work plus h due to losses, viscous losses, the head due to losses. So, this is the most, this is the special case where you assume steady flow, uniform flow incompressible fluid and single inlet and single outlet. But sorry we are not considering uniform anymore because you are removing the uniform flow assumption by introducing the kinetic energy correction factor.

If the flow is uniform alpha is 1.If the flow is laminar alpha is 2 and alpha is 1.04 if the flow is turbulent. So, this is very close to one. That is because of the fact that the turbulent flow velocity profile is in fact uniform, very close to being uniform. Therefore it makes sense for us to approximate alpha to 1 for turbulent flows. So, that completes the basic discussion on energy balance. Now, I want to point out an important distinction between this energy equation, which we derived from the first of thermodynamics to another equation which many of us will be familiar with.

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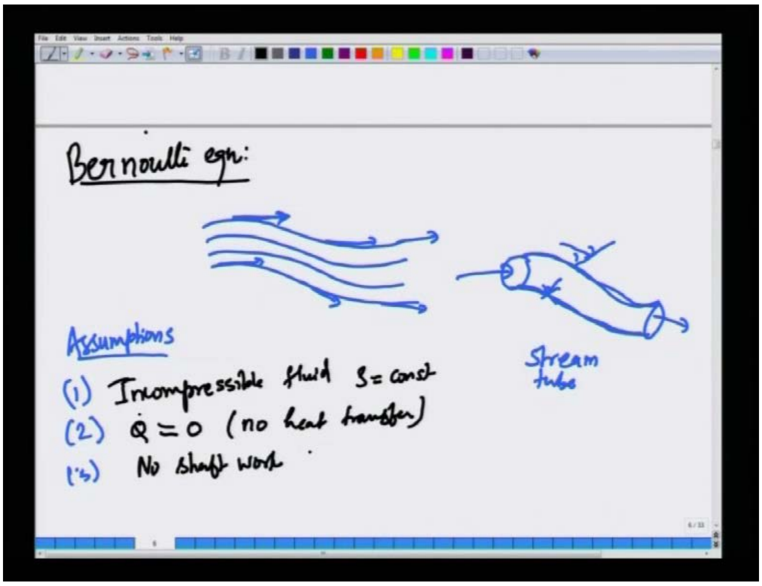

That is the Bernoulli equation, in if you look at the Bernoulli equation, if you are already familiar with the Bernoulli equation it may appear very similar to the energy equation. But let me try to derive the Bernoulli equation from the energy equation that we just derived. So, in general if we have a fluid, the fluid velocity. We define what are called stream lines. Stream lines are always parallel to the local velocity vector. So, in a fluid flow in a steady fluid flow, you can describe the flow pattern using a bunch of stream lines by plotting the stream lines. If you collect a set of stream lines you can form what is called a stream tube. It is a bunch of stream lines that are collected together it is like a tube. So, the stream tube, the surface of the stream tube itself are comprised of stream lines.

A basic definition of stream line is that fluid velocity is tangential to stream line and there is no flow normal to the stream line. So, therefore, the stream tube is almost like a tube of a complicated shape but it is a still a tube, in the sense that there is no flow, net flow out of the stream tube in the normal direction. Fluid can come in and flow but there is no a flow in this direction because by definition the stream lines are parallel to the flow velocity vector. But this not like a rigid conduit because this stream tube is surrounded by a fluid and in general the fluid can exert stresses, viscous shear stresses.

So, I want to apply the energy balance to this control volume, where you have a stream tube the stream tube is our control volume. Now we will make some major assumptions in order to derive the Bernoulli equation, that these will tell us the crucial distinctions between the Bernoulli equation that we are going to derive and the most general form of energy balance which is statement of first law of the thermodynamics. We will assume that fluid is incompressible, rho is constant. Second we will assume that there is no heat transfer from the outside to inside or vice versa, and there is no shaft work within this stream tube, no shaft work. We are just considering plain flow there is no reason for us to expect that there is a shaft work, present no shaft work.

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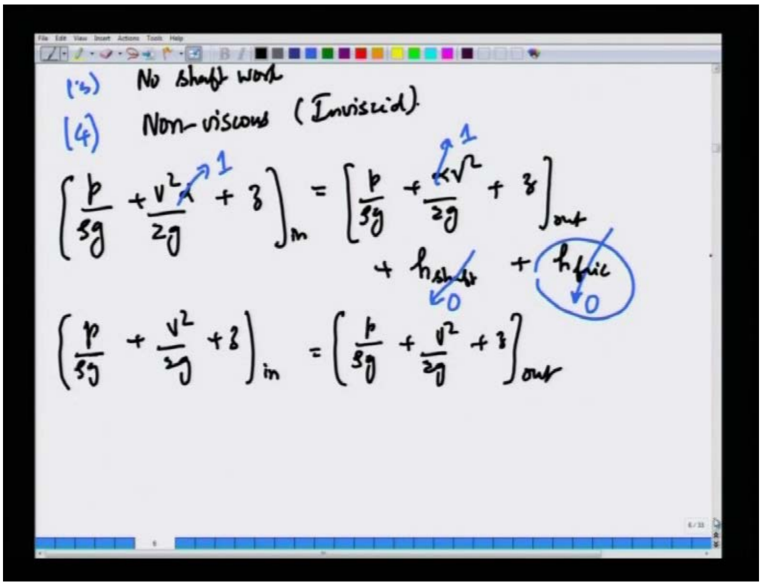

So, these are the three major assumptions. Once we do this and further assume a most important assumption that the fluid is non viscous or inviscid fluid is hypothetical fluid with 0 viscosity. Once you do this we can simplify the energy balance to the Bernoulli equation. So, by making this specific assumptions that flows is incompressible and there is no heat transfer, there is no shaft work, most importantly that the fluid is non viscous inviscid, there is no viscosity.

And we apply the energy equation to this specific control volume called a stream tube. A stream tube is essentially a bunch of stream lines that are present in a flow and we collect a bunch of stream lines and encircle them. And the stream tube itself the surface of the stream tube itself is comprised of the stream lines and so there is no flow in or out of stream tube to the sides. So, it can only come in and go out through the entry and exit.

There is no flow in through, there is no normal flow through the sides of the stream tube. By definition the fluid velocity is parallel to the surface, it cannot go perpendicular.

If you assume all these things then the most general form of the energy balance which we wrote with single inlet and single outlet for incompressible fluids is the following. V square by 2plus g z, V square by 2 g plus p by rho g. So, let me write p by rho g first as usual. p by rho g plus V square by 2 g, alpha plus z at in, is p by rho g plus alpha V square by 2 g plus z out plus h shaft plus h friction. This is the most general equation. Now, we are going to simplify this, we are saying that there is no shaft work 0 and there is no viscous shear stresses at the sides because we are assuming the fluid to be in inviscid or non viscous. So, no viscous stresses, the work done by the viscous stresses on the C V,on the control surfaces of the C V. So, that is 0.There is no viscous loss because the fluid is inviscid. So, this is a very critical assumption and we will assume uniform flow alpha is said to 1, that is further an assumption. So, if we do all this you will get a simple equation. Now p by rho g plus V square by 2 g plus z at inlet is p by rho g plus V square by 2 g plus z at outlet.

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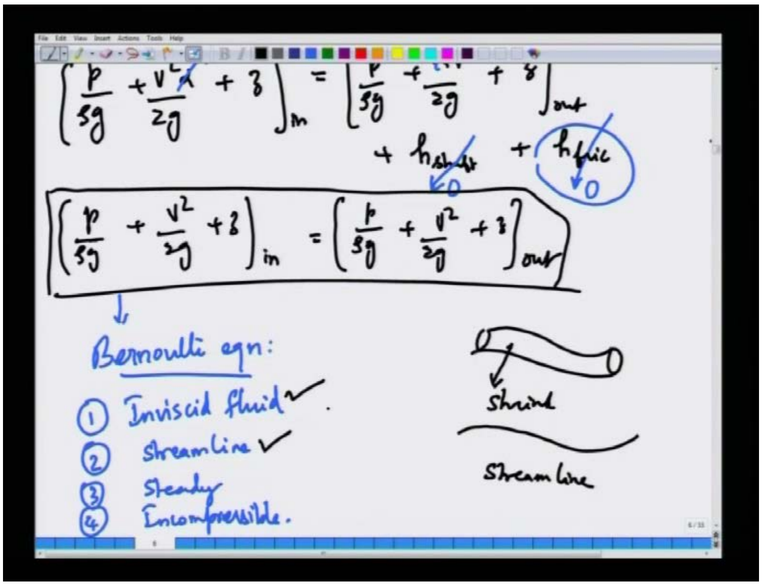

Now, this we can further simplify. The stream tube or further shrink the stream tube to an infinitesimal of infinitesimal thickness in the limit it becomes a stream line. So, this equation which is called the Bernoulli equation. The classical Bernoulli equation is applicable to an inviscid fluid, 0 viscosity to a stream line.

It is applicable only between the two points along a steam line, along the same stream line of course we have made further assumptions such as steady incompressible flow. Which are less restricted assumptions but the critical assumptions for the application for the application of Bernoulli equation is an inviscid fluid and a stream line. Whereas the energy balance that we wrote is generally applicable to any control volume. And as long as you can supply the problem on with the information about what are the viscous losses you can completely solve it. And it is applicable to any flow of viscous liquid into any equipment like a pump or a piping network and so on.

As long as we know what are the losses we can certainly solve that, most general energy balance equation internal energy balance equation. But this one is more restrictive the Bernoulli equation although it appears very close closely similar to the integral energy balance it is not, because it is applicable only to an inviscid fluid of course assuming steady and incompressible flow. And further it is applicable only along two points along a given g stream line. That is the very strict limitation on the application of the Bernoulli equation. Whereas the engineering energy balance equation sorry .

Whereas the integral energy balance equation is always valid for any complex flow process that you may encounter which may have pumps and compressors. So, you can take into account what is the shaft work contributions and so on very easily. But you need to know what are the losses involved in a given situations. That is the major input to the problem, before we can solve a problem. So, we can ask the question, when can the engineering Bernoulli equation, sorry when can the Bernoulli equation be valid?

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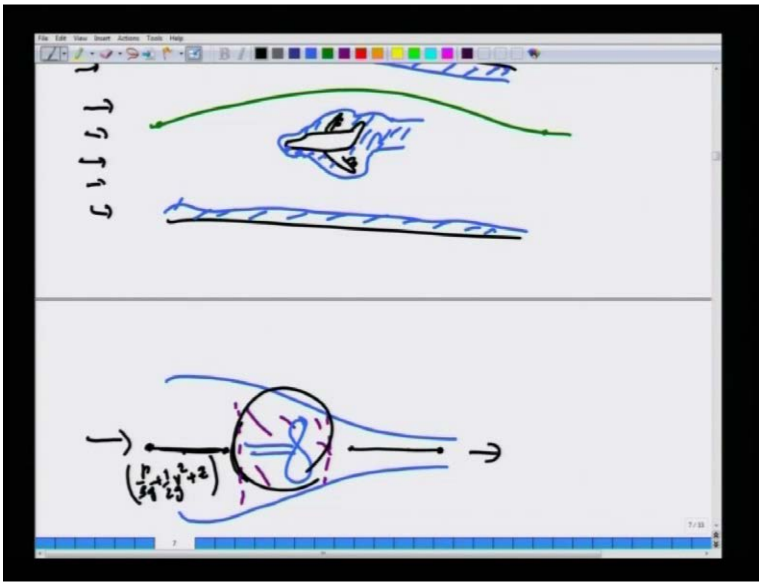

So, suppose you have a system like this, you have a flow past from a air plane model. You have a tiny air plane ,this is the model airplane and you have flow. Let us says the airplane is stationery when flow is coming like this, very close to the solid surfaces we see later that viscous effects will become important. This is called a boundary layer. So, here where ever I am patching with blue Bernouli equation is not valid. So, very close to the air plane surface as well there will be viscous effects that are very very important. So, the Bernoulli equation will be invalid in the blue patches but if you take a stream line where there is no blue patch then you can certainly apply the Bernouli equation between these two points, because you can assume that there is to a good approximation that viscous losses are negligible.

They are never 0but they are not that important. Likewise let us consider another context. You may have a conduit like this and you may have a fan or a blower and fluid is coming in like this and going out like this. So, close to these solid surfaces, close to this solid surfaces viscous losses will be important. So, you can apply Bernoulli equation in the upstream and downstream of this blower or a fan along a stream line. But you cannot take the stream line across this fan because viscous losses become important. So, the Bernoulli equation is valid. So, p by rho plus half V square p by rho g plus V square by 2 g plus z is a constant along this stream line. Tt is also a constant along this stream line in the downstream but the constants will be different because of the fact you have an equipment like a shaft or a fan which is trying to do work on the fluid and so on.

So, clearly the Bernoulli equation is not valid in this region. So, the Bernoulli equation is a very simple equation and it can be used in many contexts to get a first cut result But in certain contexts obviously, the Bernoulli equation is not valid and you should know when to apply the classical Bernoulli equation and when to apply the integral energy balance. The integral energy balance is certainly more general in that it accounts for viscous losses and it also accounts for shaft work contributions, either in or out of the C V But whereas the Bernoulli equation of course cannot taken to account all these competitions.

Now, this is a really what is I want to say about the energy balance, integral energy balance. Now, we have to apply this integral energy to some specific contexts. Now the contexts which I am going to discuss first is flow measurement.

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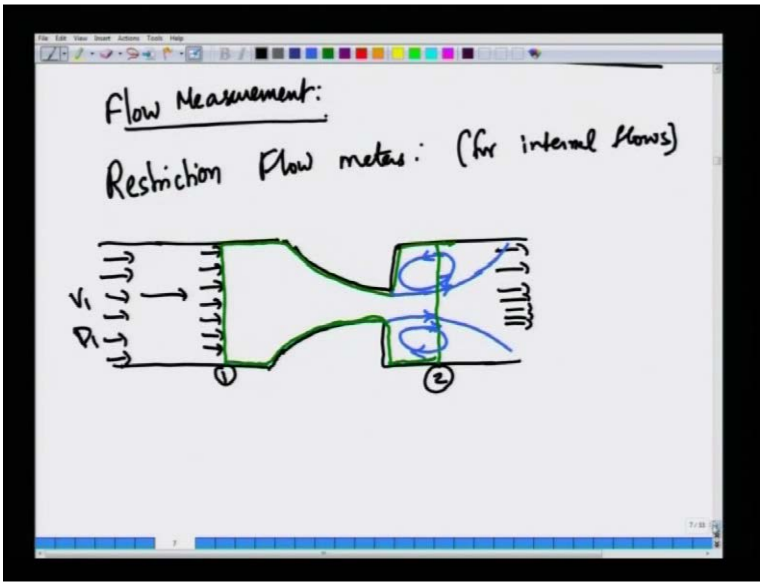

So, the question that we are going to ask is in an industrial setting, where you have complex networks of pipe pipelines are going here and there. How can you measure the flow inline? One obvious way to measure a flow is of course, if at the exit of the pipe you collect the water and you measure the volume of the water for a given amount of time. That gives you a sense of what is the volumetric flow rate that is happening in the process, in the network of pipes. But if you want to know you want to do something inline, that is without having to wait for the exit of the flow, somewhat in between the network itself.

If we want to know what is the flow rate which is very very important in many chemical engineering applications. How do you do that? So we want to do something within the pipeline network. So we have a set of devices called restriction flow meters for internal flows. That you have flow in conduits such as pipes or channels. And you want to know the flow that is happening at inside those pipes and channels. So these flow meters looks something like this. This is the general schematic there are of course specific details which we will come to a little later. The general schematic is as follows, this is the pipe line and the flow meter reduces the cross section of the flow and suddenly expands it. And of course the pipeline diameter is the same upstream and downstream.

So, this is the general schematic. Now, far away the fluid is flowing with some average velocity V 1 and the diameter of this tube is D1.Now we want to know what is the velocity and given that we know what is the diameter. In order to do this we have to draw a C V. The C V is the green line that I am going to draw, it ad joints the cross section of the flow meter very closely. This is my C V. So, fluid is coming in to the C V, at station one and it is leaving the C V at some other station. Now, that is the C V that we are going to work with. Now, if you look out the nature of the flow, the fluid will come like this and it will go like this and there will be re-circulating zones of fluid. These, zones do not contribute to any flow rate and we are going to, lets draw the C V like this we are going to draw the C V.

Where this is our C V ,let me draw the C V again. This is our C V. So fluid is coming in like this far away it is uniform and it is restricted deliberately by the flow meter. So, here the velocity will increase and because there is a sudden expansion there will be some re-circulating zone so f fluid, like this in the upstream, I mean in the downstream of the contraction and the fluid eventually leaves. So, the idea is to measure in order to find what is the velocity. We have to measure the pressure drop between the stations one and two and using that we want to infer the average velocity of stream of the flow meter.

So, that is the idea by measuring the pressure difference or pressure drop that happens between the upstream and downstream sections of the contraction of the flow meter. By measuring the pressure drop we want to be able to relate that to the velocity. So, what are the basic principles that we have? We have mass conservation, mass balance, integral mass balance and energy and energy balance.

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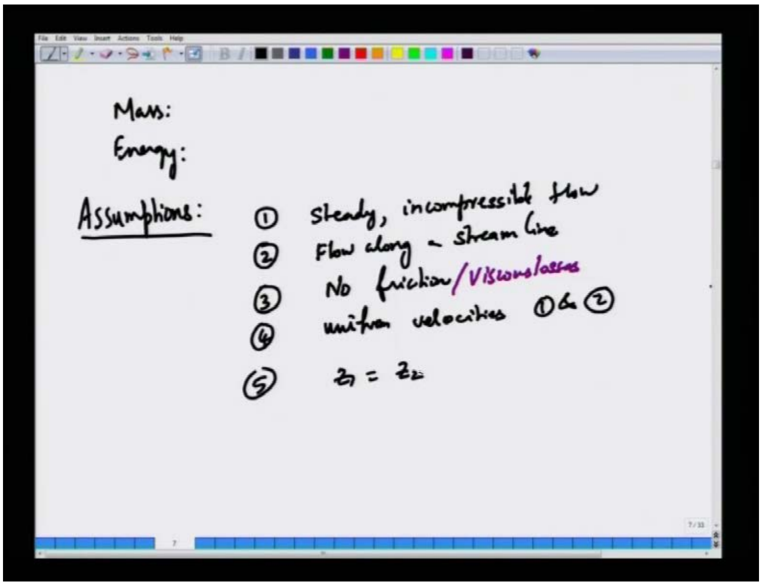

So, the mass balance, so let us first write, let us also make some assumptions because this is a fairly complex problem cannot be solved without making these assumptions. First is steady in compressible flow and we want to consider the flow along a stream line, from points one and two. We will assume no friction and uniform velocities at stations one and two. So, essentially we are considering a stream line and we want to write, we will use the Bernoulli equation and correct it later, because we are assuming no friction, no viscous losses, no frictions means no viscous losses.

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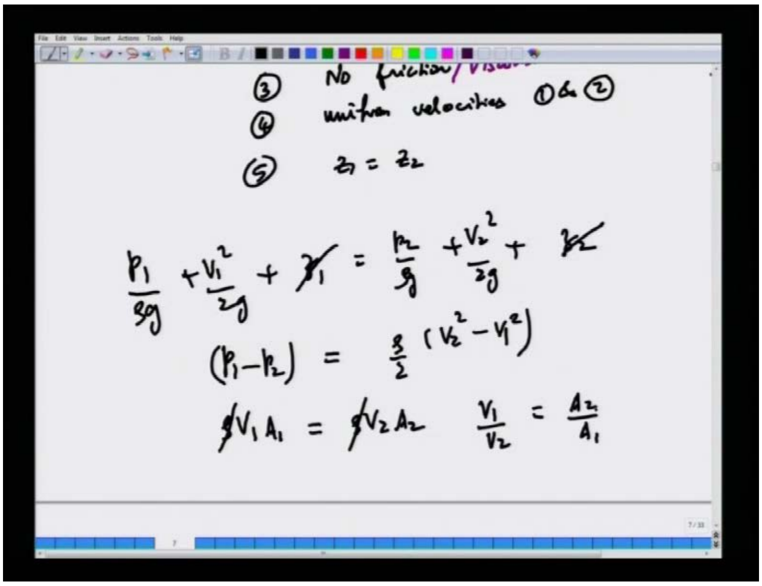

So, but it is not a good assumption, we will correct it later and we will also assume that the elevation is the same between points one and two. That is a good assumption. So, if you use the Bernoulli equation p 1 by rho V 1 square by 2 plus z 1,is p two by rho plus V 2 square by 2 plus g, z 2,z 1 is approximately equal to z 2. So, that is there so p 1minus p 2 becomes rho by 2, V 2 square minus V 1 square. Now if you use the continuity equation sorry the mass balance equation rho V 1 A1 is rho V2 A2. Since the fluid is incompressible density is constant. So, we can cancel it out. So, V 1 by V 2is simply A 2 by A 1. We will stop here and we will complete the discussions in next lecture. Soon this discussion, on application of energy balance to the case of flow measurement. These are called flow meters. So, we will stop here now and thank you for your attention.