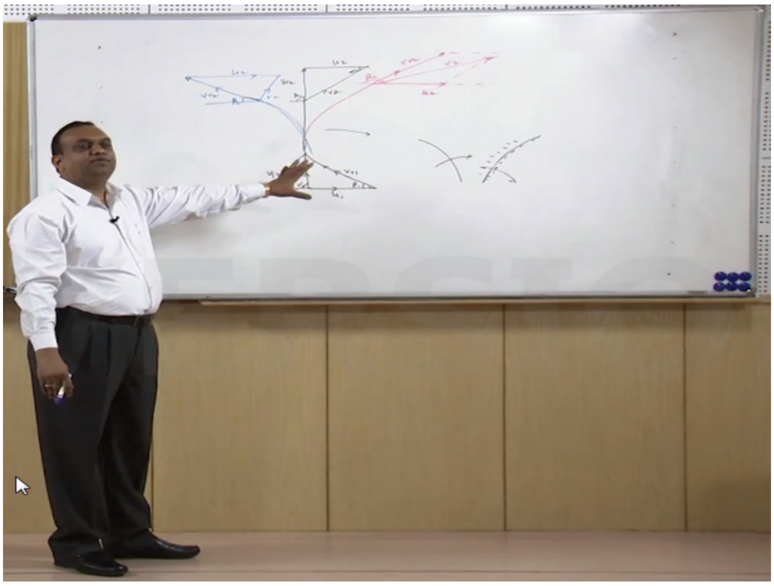
Steam and Gas Power Systems
Prof. Ravi Kumar
Department of Mechanical and Industrial engineering
Indian Institute of Technology – Roorkee

Module No # 08
Lecture No # 36
Centrifugal Compressors Characteristics

Hello I welcome you all in this course on steam and gas power systems today we will discuss centrifugal compressors characteristics the working of centrifugal compressor.

(Refer Slide Time: 00:37)



We have already discussed in the previous lecture and in the previous lecture we considered the radial blades of the centrifugal compressor right and we have drawn the velocity diagram for enlight and the blades are moving in this direction. so this is U1 this is V1 = VF1 and this is VR1 right and in the radial blades the air will be leaving blade from the radial direction this is VR2 and peripheral velocity at outlet and this is going to be the absolute velocity.

Ok and this is beta 1 this is beta 2 and this is alpha 2 and this is alpha 1 now in many of the applications the radial blades are not required the radial integral blades are not required the veins we can have backward curve veins and the forward curve veins. So if the wings are curved backward like this in that case the air will be leaving the vein in this direction opposite direction.

Right and then this beta 2 will be less than ninety degree and after this again U2 and then again V2 and this is VR2 here V2 is reduced right another arrangement can be done in the forward vein when there is a forward vein in that case air will be leaving the vein in this direction VR2 right and this is U2 this U2 and addition of this will give V2.

This is V2 ok now here the value of beta 2 is greater than 90 degree now the issue is which type of veins we should go for so this is the direction of U2 and ready velocity + U two will give the absolute velocity now we have three type of veins now the issue is for the particular application which type of vein we should go for now if we look at this type of geometry this is one type of geometry extreme geometry and this is another type of extreme geometry.

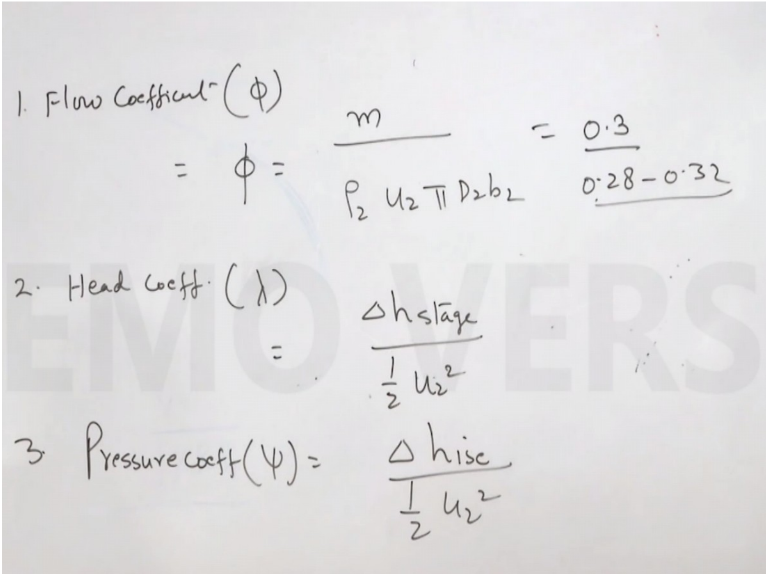
Now in backward veins and the forward veins if you look at the forward veins the absolute velocity which is giving the compressor is very high of baller is very high here in this case the absolute velocity which is leaving the impeller is low and it means in forward curve vein the other parameters are same RPM efficiency is same pressure raise will be more because it is it has important more kinetic energy because change in peripheral velocity is going to be the same.

Right and change in the relative velocity they vary but the outlet absolute velocity is very high so pressure raise in for the other parameters are same the pressure raise is quite high in case of forward curved vein but in forward curve vein the emperor is moving on this direction this side will have high pressure this side will have low pressure slip will take place further this is a convex surface.

So definitely the separation will be more losses will be more so forward type of vein is not very efficient type of arrangement efficiency of backward curve vein is more in comparison to the forward curve vein but if want to have high pressure raise then we should go for forward curve vein right. So these are the velocity diagrams for forward and backward and radial flow veins for (()) (05:38) in the previous lecture also i explained (()) (05:44) is provided just to avoid exceeding the sonic velocity.

Inside the compressor because compressor is rotating with very high speed rotation is being forty thousand fifty thousand or thing so that number M is going to be VR1 over gamma RT one .So we should check from this formula whether insides the impeller velocity of air is not exceeding the maximum one that their other dimensional parameters to judge the performance of the compressor these dimensional parameters are number one flow coefficient

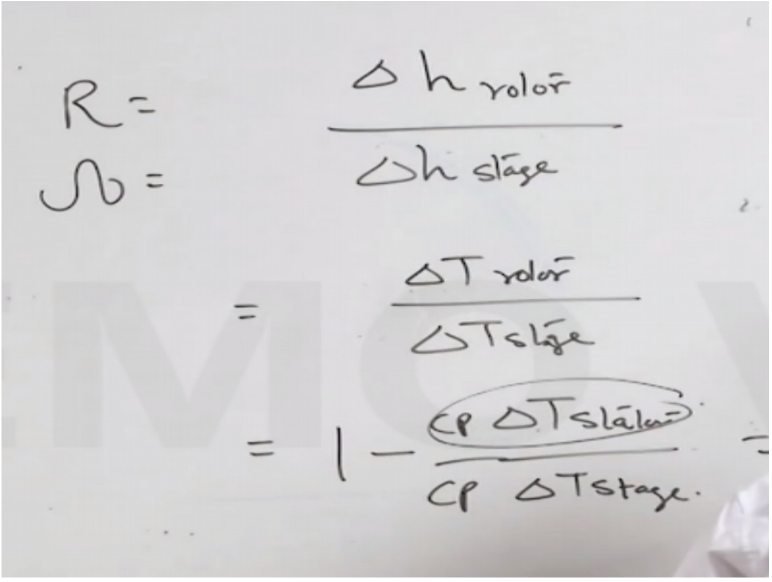
(Refer Slide Time: 06:36)



Now flow coefficient denoted by five is the actual mass flow rate and mass flow rate corresponding to the imparalell tip velocity the mass flow rate corresponding to imparellel tip velocity will be this is the cos section area the velocity and density this is volume representing the volume and this is multiplied by the density it will be the mass florate so this is the value of flow coefficient for a centrifugal compressor the optimum value.

Normally the optimum value of coefficient is .3 KT value between 0.28, - 0.32 second one is head coefficient the head coefficient is denoted by lambda and it is delta H in a stage divided by kinetic energy corresponding to the blade tip velocity that is half U2 square. Now these parameters are used for generating the performance of a centrifugal compressor or comparing the performance of two different centrifugal compressors.

The third one is pressure coefficient that is isotropic enthalpy change divided by tip velocity so if there is an posses the head coefficient and peruse coefficient are going to be the same.
(Refer Slide Time: 08:57)



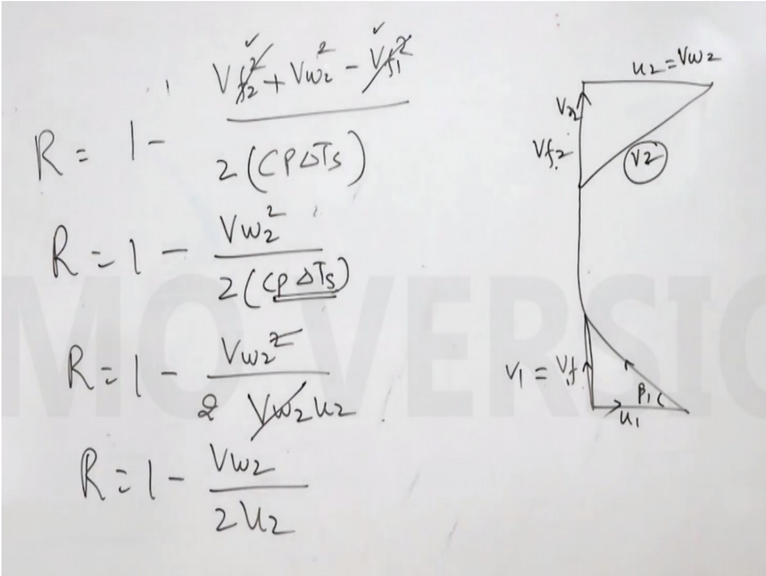
Now third one which is very important is reaction or degree of reaction is denoted by R in some of the author denoted by omega also right and this reaction is delta H in rotor divided by delta H in stage or change is enthalpy is always be delta T in case of gas turbines sorry in the central fuel compressors.

So it is delta T rotor divided by delta T stage ok and it is about = 1 - delta T state or veins rotor means which is the part of the compressor which is being rotation motion and stated means the part of the compressor. Which is static right and divided by delta T at stage.

Now delta T is stated or this is let us take in an CP delta T stage so CP delta T is stated if you remember that in vein less part and the vein diffuser the enthalpy change is due to change in kinetic energy so it is going to be = 1 - V2 square - V2 square divided by CP delta T into the stage right. Now again this is equal to this R = 1 – VF2 square.

V2 square is VF2 square + VW2 square under any case will component and flow component square will be V2 square - VF1 square divided by CP delta TS.

(Refer Slide Time: 11:30)

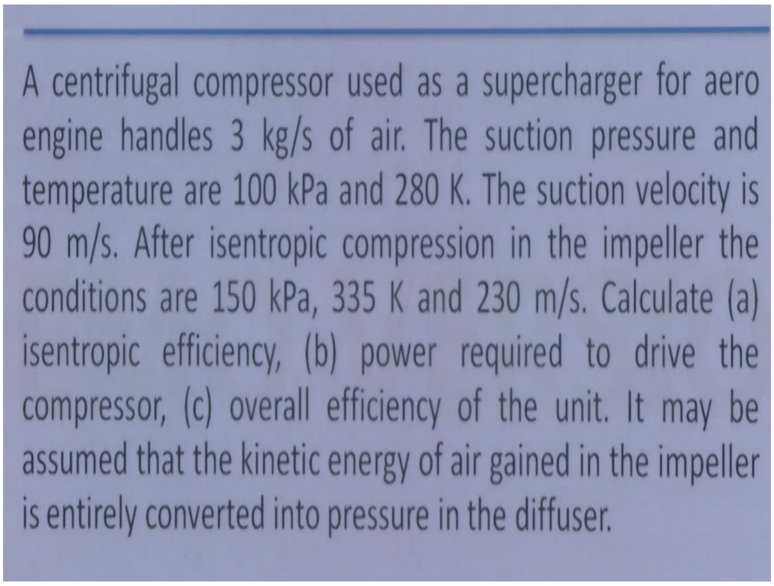


Why i have done this in a so suppose there will be a compressor for gas turbine this is VF and this is U1 and beta1 VF V1 ok so VF1 is V1 at the outlet this is VR2, U2, V2 right and this V2 = VF2 square or VR2 = VF2, VF2 square + VW two square now velocity of flow it is constant in this type of arrangement so this can be cancelled out and R = 1 – VW2 square divided by CP delta TS sorry.

We have to take kinetic energy so this has to be divided by 2 actually this R is 1 - V 2 square - V1 square divided by 2 CP delta TS so this two are missing in earlier expressions so we can make a correction. So this is going to be 1 - 2 CP delta TS right because this R is1 - V2 square - V1 square divided by CP delta TS divided by 2 these are energy.

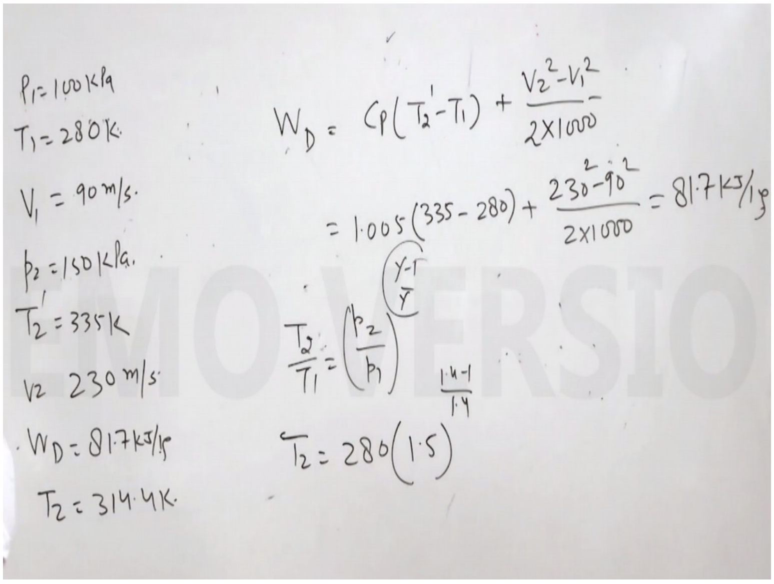
So this two was missing in earlier expression. So i have added it here and now this work can further be replaced as 1 - V W2 square divided by 2 VW2 U2 now this VW two will be cancelled with this 1 so R = 1- VW2 divided by 2 U2 and only in the case when VF1 = VF2 now after this we will solve some of the numerical on phase on centrifugal compressor the first one is a centrifugal compressor.

(Refer Slide Time: 14:21)



Used as a super charger for aero engine handles three kg per second of air so the centrifugal compressor is ending here at the rate of 3 kg per second the suction pressure and temperature are 100 kilo Pascal and 28 Kelvin. So suction pressure and temperature.

(Refer Slide Time: 14:43)



So P1 = 100 kilo Pascal and T1 = 280 K the suction velocity is 90 meters per second. So this suction velocity is 90 meters per second after isotopic compression the impeller and in the empeller the compressions are 150 kilo Pascal so P2 is 150 kilo Pascal and temperature is 335 Kelvin and velocity is 230 meters per second calculate isotropic efficiency right.

So work done is in the centrifugal compressor is CP T2 dash - T1 + V2 square - V1 square by 2 and because this is always in kilo joules and we should initially we should divide it by 1000 to concert the expression in kilo joules so work done is 1.005 this T2 is T2 dash 335 - T1 280

+ V2 is to 30 this is V1 is to 30 square - 90 square divided by 2 into 1000 and this gives 81.7 kilo joules per kg.

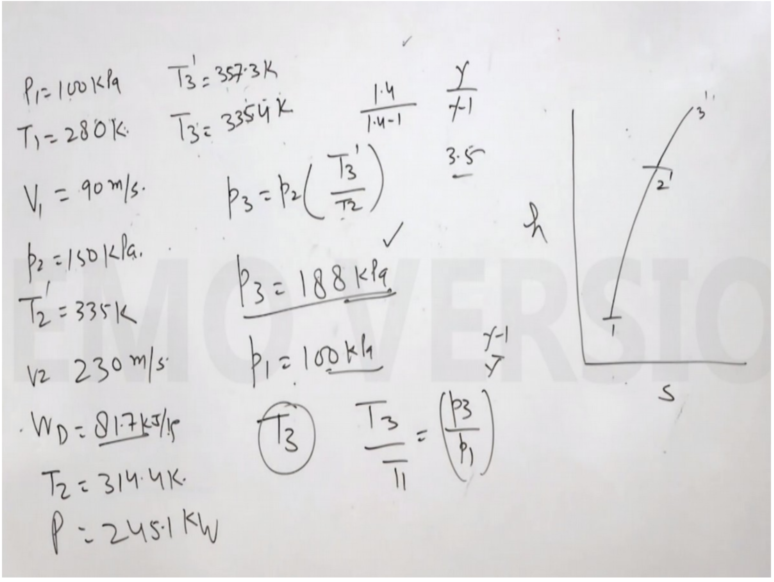
So work is 81.7 kilo joules per kg and we have to find isotropic efficiency now T2 is T2 by T1 = T2 by T1 raise to power gamma - 1 over gamma right gamma is 1.4. So we can take this as 0.286 or so let us take 1.4 here. So T2 is T1 is 280 T2 is 150 and T1 is 100.

So it is 1.5 raise to power 1.4 -1 divided by 1.4 ok and this gives the value of T2 as 314.4 Kelvin now we have the value of T2 we have the value of T2 dash now isotropic work this is the actual; work we have calculated now the isotropic work because in rotator questions kinetic energy cannot be neglected as i told you earlier so work done.

Isotropic is going to be CP T2 - T1 + V2 square – V1 square by 2 into 1000 = 1.005 and T2 is we have calculated 314.4 - T1 280 + again V2 square that is that expression is same 90 square by 2 into 100 and then we get isotropic work as 56.97 kilo joules per kg and we have to calculate the isotropic efficiency = 56.97 divided by 81.7 multiplied by 100.

We will give 69.7% ok so that is the difference between a isotropic efficiency of a gas turbine what we calculate in gas turbine right there we do not consider this kinetic energy but in actually when in a compressor when there is a change in the absolute velocity so this kinetic energy has to be considered while calculating the isotropic efficiency the total work total power is work done is 81.7 the total power is going to be this multiplied by mass flow rate that is three.

(Refer Slide Time: 19:50)



So power is 3 into 81.7. So power is going to be 245.1 kilo watt now overall, efficiency of the unit after the impeller if the air goes to the diffuser right when air goes to the diffuser then CP T3 dash - T2 dash that is from impeller sorry the impeller tip to the diffuser it is between impeller tip to the diffuser and this is = C2 square - C 1 square divided by 2 into 1000.

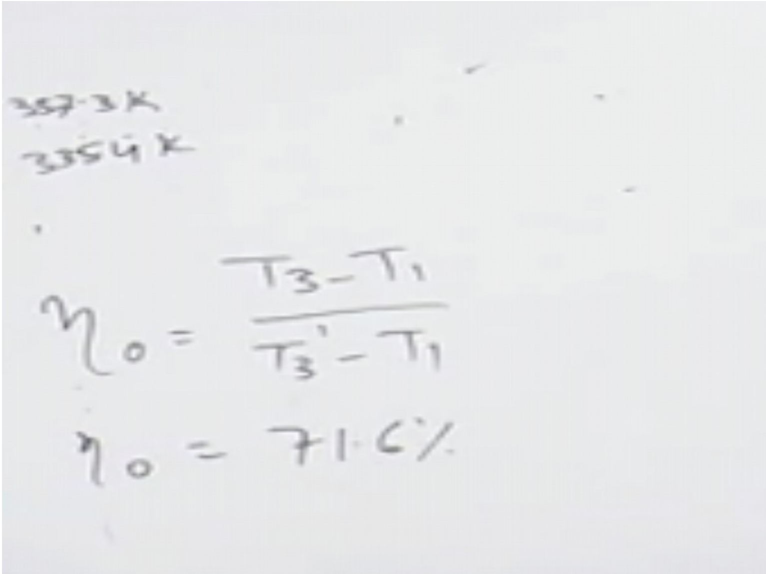
Right and that is = 230 square - 90 square divided into 2 into 1000 now we have taken part earlier we have calculating work consumed we included this also but now we are exclusively dealing with this and this will give us the temperature T3 dash because the rest of the things are known to us so temperature T3 dash is going to be = 357.3 Kelvin. So if we look at entropy enthalpy diagram.

So this is the enthalpy increased from one to two and this is 2 to 3 enthalpy increase 3 dash this is 2 dash this is 3 dash ok now once we have the T3 dash now the P3 is P2 T3 dash divided by T2 this is we have not we are not considering any loss during this because it is not given in the numerical value. So we are consuming isotropic process.

So here that that case we can take 1.4 divided by 1.4 - 1 that is gamma 1 gamma - 1 PR 3.5 whatever value we want to take and this provides us the pressure P3 that is 188 kilo Pascal right we have the value of T3 and we have the value of T1 also right with the help of these two values we can calculate the value of T3 by T1 = P3 by P1 raise to power gamma - 1 over gamma.

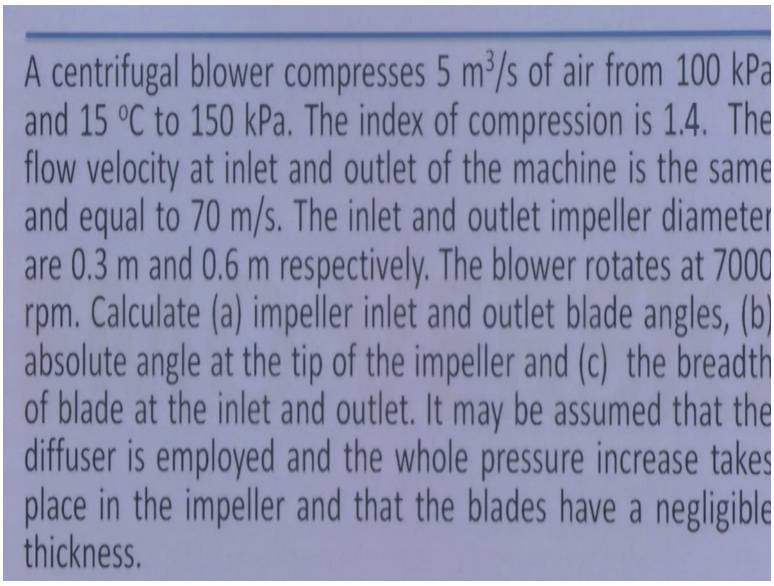
Now considering this we get the value of T3 as 335.4 Kelvin now e have the value of T3 dash we have the value of T3.

(Refer Slide Time: 23:21)



So overall efficiency the overall efficiency of the compressing machine overall efficiency is going to be T3 - T1 divided by T3 dash – T1 now we have all the value we have the value of T3 dash we have the value of T3 we have value of T1 also and this gives the overall efficiency as 71.6% right now.

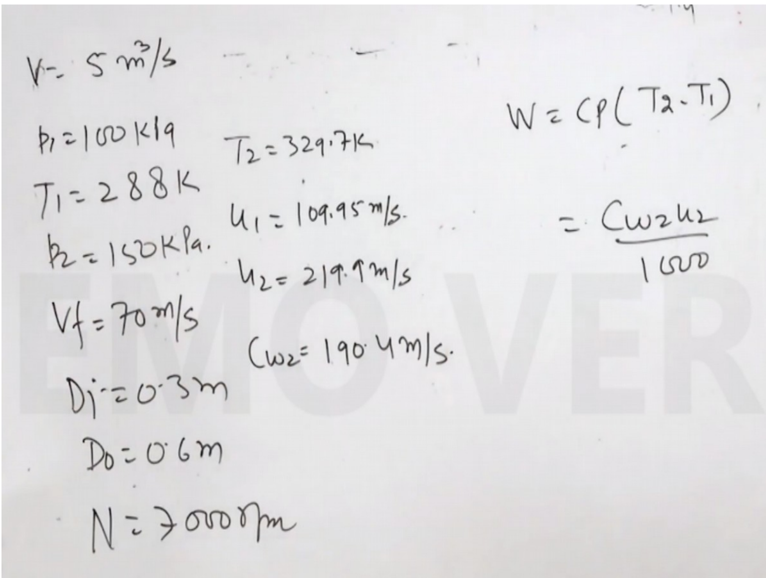
(Refer Slide Time: 24:00)



A centrifugal blower compresses 5 m^3/s of air from 100 kPa and 15 °C to 150 kPa. The index of compression is 1.4. The flow velocity at inlet and outlet of the machine is the same and equal to 70 m/s. The inlet and outlet impeller diameter are 0.3 m and 0.6 m respectively. The blower rotates at 7000 rpm. Calculate (a) impeller inlet and outlet blade angles, (b) absolute angle at the tip of the impeller and (c) the breadth of blade at the inlet and outlet. It may be assumed that the diffuser is employed and the whole pressure increase takes place in the impeller and that the blades have a negligible thickness.

We will take up another numerical this is about a centrifugal compressor centrifugal blower which compresses 5 metre cube per second of air from 100 kilopascal and 15 degree centigrade to 150 degree centigrade is the blower the working principle of blower is same as the working principle of centrifugal compressor only pressure ratio is different in blower the pressure ratio is very low incorporating into the compressor. So the compressor compresses 5 metre cube of air.

(Refer Slide Time: 24:25)



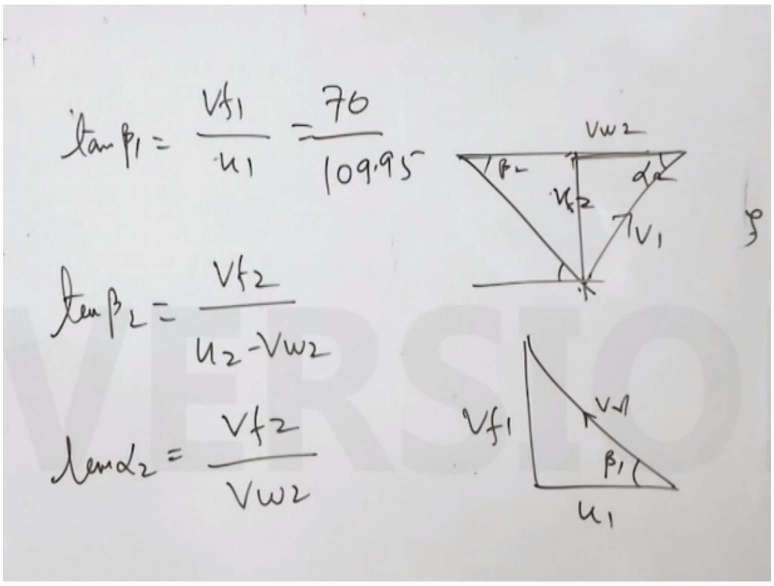
So V = 5 meter cube per second from 100 kilo Pascal and 15 degree centigrade to 150 kilo Pascal so P1 = 100 kilo Pascal T1 = 288, 15 degree centigrade 288 Kelvin T2 = 150 kilo Pascal the flow velocity of the inlet and outlet of the machine is the same and = 70 metres per second the inlet and outlet impeller diameter are .3 and .6 respectively.

So DI = 0.3 metre DO = 0.6 meters the blow rotates 7000 RPM so N = 7000 RPM calculate impeller inlet and outlet blade angles so first of all we will calculate T2 temperature at the exit it is going to be T1, P2 by P1 raise to power gamma -1 - gamma and that is going to be equal to i will write the expression here T2 = 329.7 Kelvin because we have the value of T1 we can take from here P2 by P1 is 1.5.

Gamma is one gamma = 1.4 and from here we will get the value of T2 as this now U1 peripheral velocity at inlet U1 = pie D1 N by 60 we have the value of N we have the value of DI or D1 so U1 can be calculated as 109.95 meters per second since the diameter has driven from .3 metre to .6 metre so U2 will also be doubled as 2119.9 metres per second right now CP work is CP T2 - T1 is specific work = CW 2.

U2 by 1000 right now here we have the value of T2 we have the value of T1. we have the value of U2 also this will covers the value of CW2 and that is = 190.4 metres per second sorry yes 4 metres per second it is will component right so comparing simply T2 - T1 with the input to the blower we have calculated the CW2 now once we have CW2 with us now we want to calculate blade inlet angle blade inlet angle.

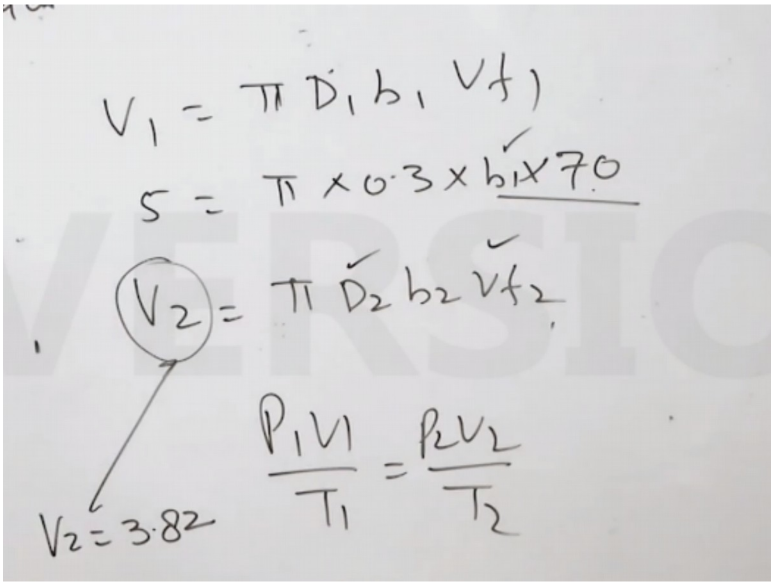
(Refer Slide Time: 27:39)



VF1 U1 VR1 and this is beta 1 so tan beta 1 = VF1 by U1 VF1 is 70 so it is 70 divided by 109.95 from here we will get the value of Beta1 is 32.4 degree right now similarly we will calculate now beta 2 is this is beta 2 right and this beta 2 is tan beta 2 is VF2 by U 2 - VW2 this is V1 and this is VW2 and this is VF2.

So VF2 divided by this part and we have all the values with us and from here we will get the value of beta 2 S 67.1 degree alpha 2 is this one similarly alpha 2 can also be calculated as tan alpha 2 = VF2 divided by VW 2 right and then alpha 2 = 20.2 degree now the third part is breadth of the blade at inlet and outlet. So we have to find the value of P and in order to find the value of P.

(Refer Slide Time: 29:36)



V1 = pie D1 B1 VF1 this is area velocity volume V1 N is 5 = pie is known to us D1 is 0.3 into B1 and VF is power 70 this will give the value of B1 now the B1 is 7.58 centimetre now similarly V2 for pie D2 V2 VF2 VF1 = VF2 but we do not have the value of V2 right we will use the relation P1 V1 T1 = P2 V2 T2 P1 and P2 and T1 and T2 are known to us V1 is with us.

We will calculate the value of V2 and this V2 is V2 = 3.82 metre cube per second and this is this value 3.8 we will put here D2 is with us F2 is with us with the 70 metres per second and that will give the value of B2 that is 2.19 centimetre that is all for today from the next class we will start with the axial flow compressors thank you very much.