Gear and Gear Unit Design: Theory and Practice

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Lecture – 14
Gear Unit Design – 1st. Stage Pinion & Gear Design-II (Probable Dynamic Load and Wear Load Capacity and Finalizing 1st. Stage Pinion & Gear Set)

We are continuing with design of general purpose industrial helical gear reduction unit this is part one. In this lecture I shall cover gear unit design 1st stage pinion and gear design, probable dynamic load and wear load capacity, and then I will finalize the 1st stage pinion and gears. In more details the covering portion will be design of 1st stage pinion and gear, estimation of wear load capacity, estimation of probable dynamic load at tooth contact, comparison of wear load capacity and probable dynamic load.

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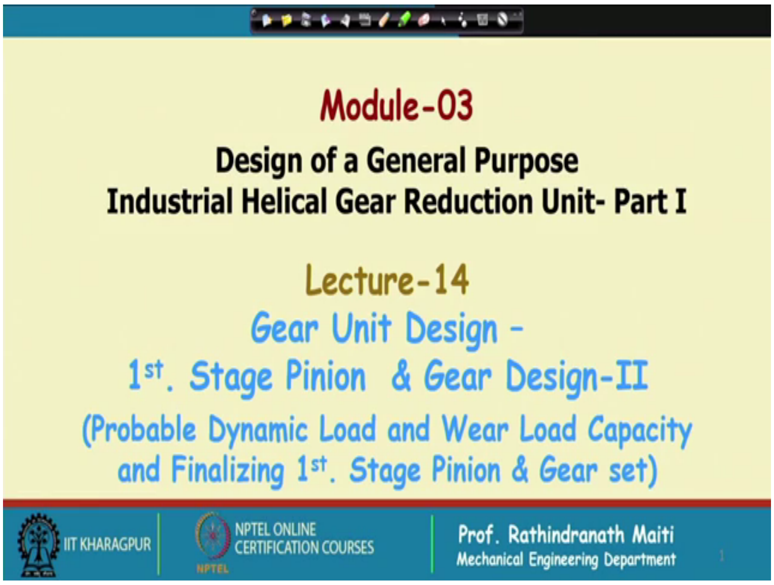

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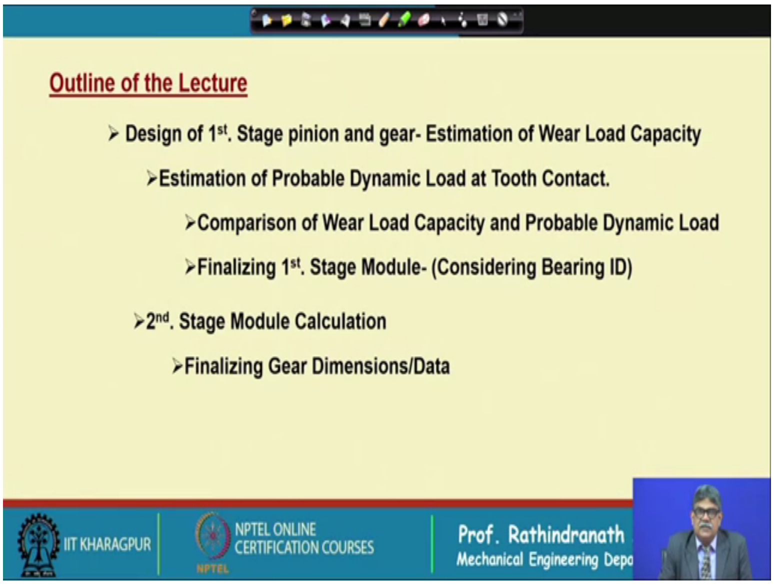

Finalizing 1st stage module considering bearing internal diameter ID 2nd stage module calculation and then, finalizing gear dimensions or data.

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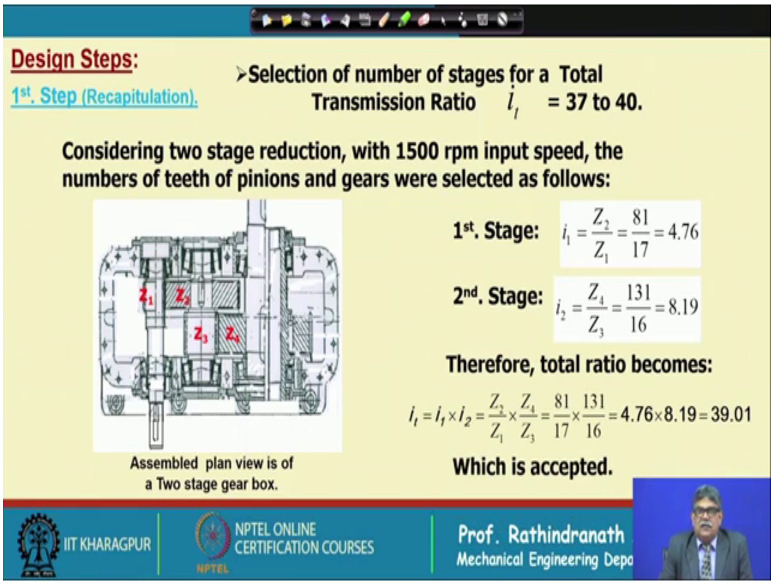

Now , in the problem it is given that transmission ratio should be 37 to 40, and we have already selected the stage ratios. We have selected 2 stage gearbox and we have considered that, in 1st stage teeth number of pinion is 17 and gear is 81.

In 2nd stage it is that is the teeth number of pinion is 16 and depth number of gear is 131. Although I have discussed, but here I would like to tell, usually it is not preferred that transmission ratio should be more than 6 in 1 stage, but to make this gearbox within 2 stage we have taken in 2nd stage 8.19, this is allowable . 1st stage it should not be more than 6 in any case, we could have go for 3 stage gearbox but, it would be expensive and also design will be cumbersome, not it is not required rather this is one.

2nd is that we have not taken the exact ratio, it is fraction that is to avoid tooth hunting which is after the repeated cycle; that means, if say 1st stage is 4; that means, every after 4 cycles there will be same pair in contact and that increases dynamics because, there will be some sort of pitch error and surface error in gears, to avoid that we have taken in fractions and ultimately our total transmission ratio has come 39.01. Which is acceptable because, of the reason that usually in prime mover although we wanted at 1500 rpm, but there will be plus minus 5 percent variations or maybe little less and as well, in output also there will be some variations. So, this variation is acceptable in general purpose.

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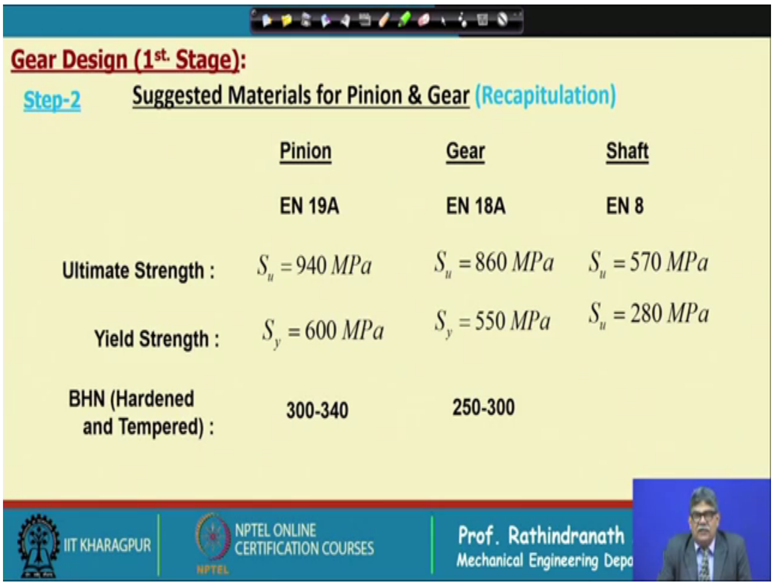

now, the material already discussed in earlier lecture, that for pinion we have taken EN 19A that is European norms 19 A, and gear EN 18A and shaft EN 8. Now, 19A or there are also there is c d those are very close to EN 19, basic material is EN 19 only small variation. So, even if it is EN 19 or EN 18 the properties will be same, whether it is a or b etcetera. Now, for the material pinion EN 19 the ultimate strain this 940 MPa and yield strength is 600 MPa, and renal hardness which can be raised to 300 to 340 before hauving. And it will produce quite aqudate gears. Similarly, for EN 18 it is ultimate strength is ache 860 MPa, and yield strength is 550 MPa, and hardness 250 to 300 which could be raised to 300 to 340, but it is not required, usually the gla the gear may have less hardness than the pinion that we will discuss later and shaft (Refer Time: 6:25) it is not hardened, if necessary it can be hardened.

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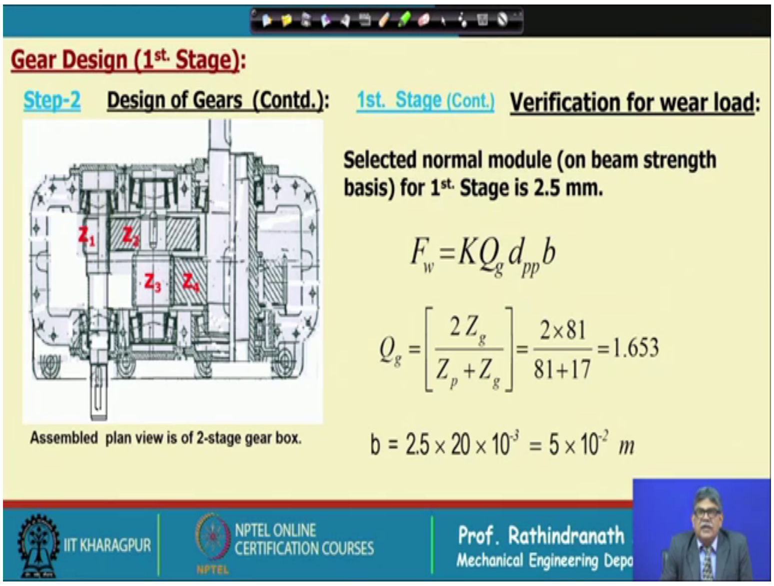

Now, in 1st stage design selected normal module we have already taken 2.5mm here, I would like to mention that allowable strength we have taken 2.5 times less than the yield strength. Yield strength divided by 2.5 in both cases has been taken as the allowable strength of the material, and Cv Fw etcetera has been taken and we have already selected the module 2.5 mm is satisfactorily. Now, we shall calculate the allowable strength in wears load or contact load. Which is expressed by Fw is expressed by K which is a factor depends on material accuracy hardness etcetera, Qg that a ratio teeth number of the gears and pinion it is related to the mo teeth number of gears and pinion, and dpp that is pitch diameter of pinion 1st subscript for pitch 2nds sub subscript for pinion, and then b is the active width of the gear this is we have considered for spur gear state tooth spur gab.

Now, this relations are Qg is given by twice teeth number of gears divided by summation of pinion and gear teeth number, which comes to 1.653 and b is the active width of the gear. We have considered the width factor size is equal to 20. So, 2.5 into 20 eh it comes 5 into 10 to the power minus 2 meter that is 50 mm.

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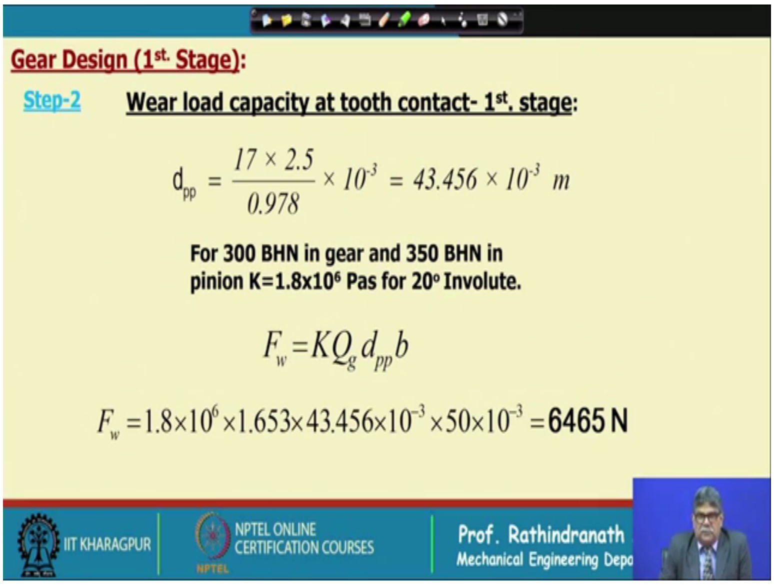

Now, dpp that is pitch diameter of pinion is 17 into 2.5 divided by cos beta we have considered helical gear. So, we have taken 0.978 into 10 to the power 3, which becomes 43.456 into 10 to the power minus 3 meter . That is, it is 43.456 mm. For 300 BHN and in gear and 350 BHN in pinion, K it can be taken as 1.8 into 10 to the power 6 pascals for 20-degree involute. This is from available data in machine design books or design data book.

Now, for this calculations therefore, it comes 1.8 into 10 to the power 6 into 1.653 into 43.456 into 10 to the power minus 3, which becomes into 50 into 10 to the power minus 3 that is b width. So, this becomes 6465 N so; that means, at the tooth contact safely we can apply 6465 N load .

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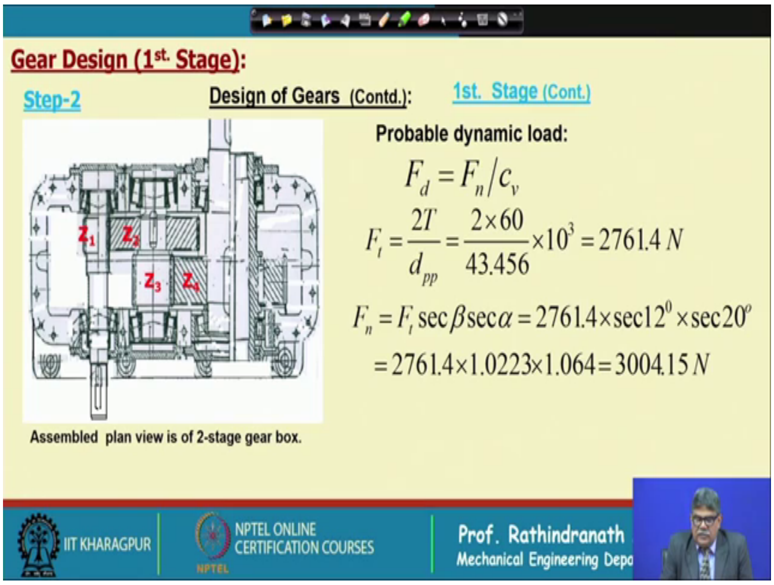

Now, we shall consider the probable dynamic load, one thing I would like to mention that in earlier equation; that means, for wear load it is for helical gear cos cube cos beta is the denominator, but here we have not considered, that is 0.978 we have not considered for that wear load will be slightly more.

Now, probable dynamic load is Fd is equal to we are considering in the direction of normal directions because, what the wear load strength we have calculated it is in the direction of normal . So, we considered a the normal load that divided by the velocity factor this velocity factor already we have calculated earlier. Now, the Ft which can be given is 2T by dpp that is (Refer Time: 11:40) of pini pinion. So, 2 into 60 now here, as we have considered the a factor of safety of 2 multi that we have multiplied with the torque in calculation of gear teeth module. So, here also we consider the same load 2 into 60 divided by 43.456 into 10 to the power 3 which becomes 2761.4 N.

And in normal directions this load will be 3004.15N. This formula I have shown earlier here, I would like to mention that helix angle we have considered the 12 degree, that is nominal. It may need to refine, why we we shall finalize the dimensions and pressure angle is 20 degree? So, that gives us 3004.15N in that normal directions.

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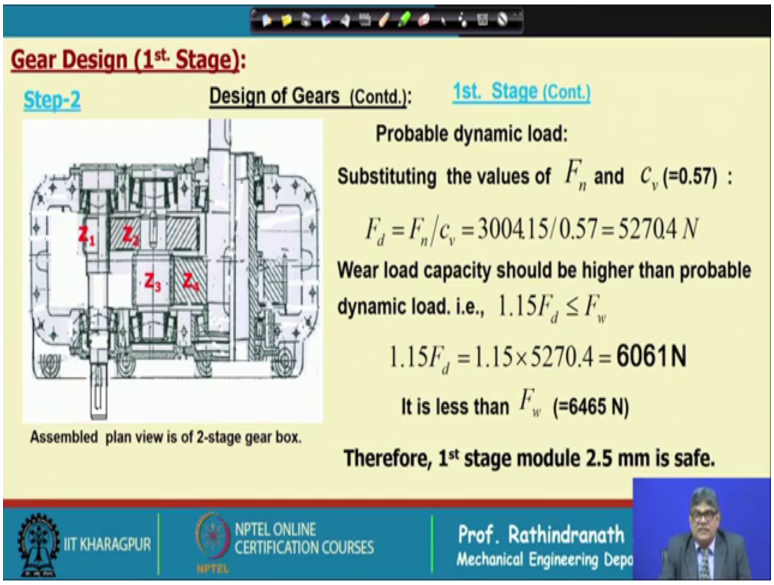

Now, already for 2.5 we have calculated the velocity factor is 0.57. Therefore, we get the probable dynamic load in the normal directions 5270.4 N. Now, this is has in earlier lecture, we have discussed that 1.15 times of the dynamic load should be less than the wear load capacity. What does it mean? Actually I would say that due to the dynamics load may increase up to calculated Fd into 1.15. This is although momentarily, but that may cause damage to the surface even if there is no breakage, no damage at the root due to the bending; to take care of that we have taken that, we consider that whatever the surface capacity we should dynamic loads should not increase that now, in this case this 1.15Fd becomes 6061N and we have wear load capacity 6465N. So, this we should call consider the design is satisfactory.

Now, here is the question if it is not satisfied, if the dynamic lord load part is more than wear load capacity then, what would do? At 1st instant we can increase the hardness, once the hardness is increased then K factor increases and therefore, the capacity load carrying capacity wear load capacity also increases. So, for example, we have taken 350 BHN or for pinion, and 300 BHN for gear why it is like that? Actually, if we consider the momentary radius of curvature, where the maximum contact load is coming the radius of curvature of gear tooth will be more than the pinion tooth, if the ratio is high in the order of 3, 4.

Therefore, we may not need that much hardness of the gear that is why 50 difference is usual. Now, if this would not be satisfactory in that case, probably we could go for pinion hardness 400 or 450 and gear had this 350 or 300. And with that the Key factor would and wear load capacity would also increase, but we should Keep in mind in any case dynamic load is not increasing rather, if we go for increasing the hardness, if the hardness is 400 or so, then that hardened to be done after the hop cutting. This means that if it is hardened after hop cutting, there will be distortion of the teeth there will be distortion it is profile.

Usually a little more material is kept and then the gear after hardening, the pinion and gears they are ground. Once it is ground then surface will be smooth as well as the core strength will increase. So, in that way we may find in grinding also pitch error will be less. So, dynamic load may decrease as well as, we are increasing the wear load carrying capacity and dynamic load is decreasing.

So, that is one method that we can have satisfactory design, but we should keep in mind that hardness process, grinding process they are expensive. So, looking into the purpose sometimes what is done that width of the pinion, active width that is b it is slightly increased. So, instead of here we have taken the factor is 20, in case of helical gear we can go slightly more easily we can go for 25, if it is accurately cut. So, if we go for 25 then also the wear load capacity will increase, but the dynamic load is not going to increase because, it is not dependent on the width.

So, this is another method whether for that it might be size will be slightly increase, but cost may not increase that much because, we are not doing for hardening further or grinding. Anyway, in this case the 1st is module 2.5 it is safe it is satisfactory but, we will face another problem. When we will select the bearing before going into that let us look into what are the major dimensions that to be decided once the module is calculated.

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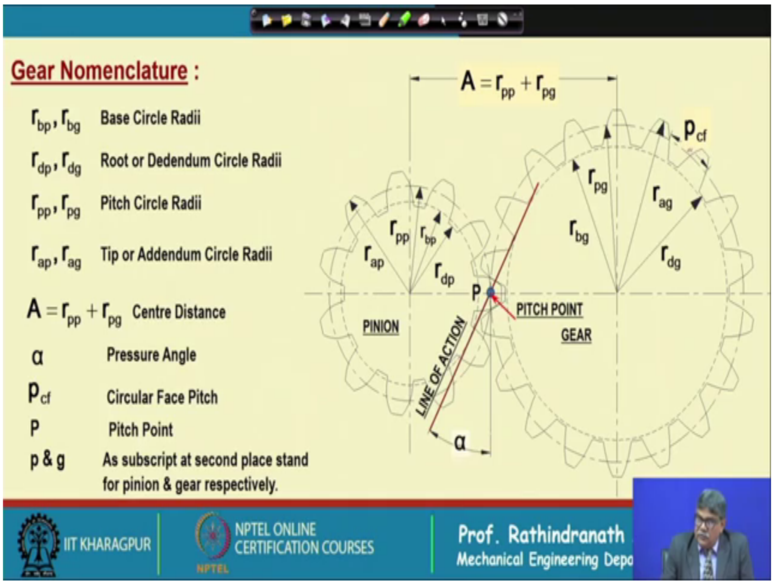

First of all, the base circle which can be calculated from the al already it is shown that will remain fixed. So, here I have shown in the gear and pinion in mace, and their the base circle you can see from this nomenclature, parametric identification that base circles and then route or dedendum circle radii. Here, it is interestingly you can see that base circle of the pinion is above the root circle whereas; in case of gear it is below the root circle. It is normal in opinion. If the teeth number is 17 or close to that even smaller then base circle will be inside, but that portion what are the portion below the base circle that is not in use.

Then, we need to calculate the I have shown, the 2nd one is the root circle and dedendum circle radii, and then pitch circle radii rpp rpg and then, rap rag is pitch or addendum circle radius and then finally, the center distance rpp plus rpg. This is without any corrections, if there is a correction still this center distance will be rpp rpg, but that pitch circle radius of pinion and pitch circle radius of gear is not the standard one, that is we need to calculate after the miss or gear tooth corrections both we should consider. In this case we are not going to calculate or considering gear tooth corrections therefore, we will consider this is the standard.

Then alpha is the pressure angle, and Pcf is the face pitch of the this is face pitch because, normal pitch will be different in case of helical gear normal pitch will be different ok. And p and g stands for the subscript in subscript stand for pinion and gears.

So, also it is shown that pitch point we had through which the load is acting. Then, we will Consider that we will finalize now, what should be the module.

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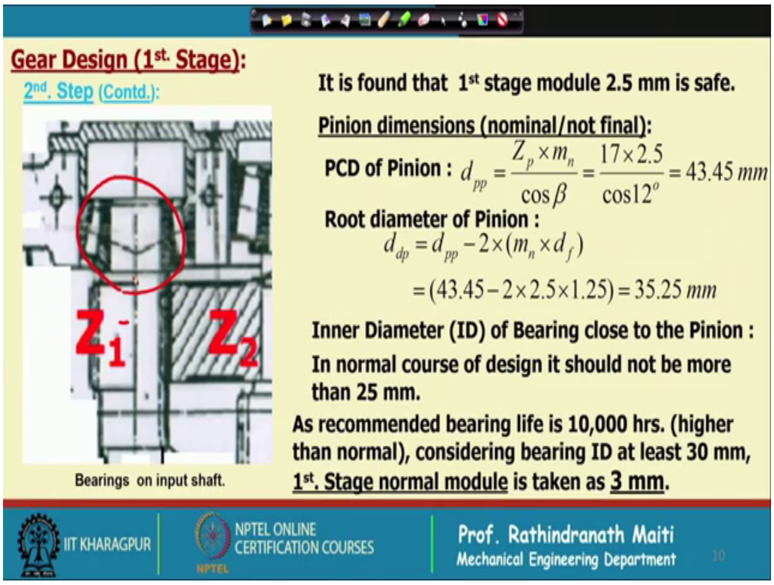

In 1st stage we have calculated the module is 2.5 mm and also considering the wear load and dynamic load, we have found that it is safe. But if we calculate now, that what is the pitch circle diameter? This is coming 43.45 which already we have calculated earlier, and if we consider the root diameter of pinion it is coming 35.25 mms. Root diameter is the pitch circle diameter minus twice into dedendum. Dedendum again is equal to normal module into dedendum factor. Dedendum factor in standard gears is 1.25. So, and module we have considered 2.5. So, the diameter room diameter opinion comes as 35.25 mm.

Now, if we consider the bearing, this is the bearing this is tapered roller bearing it is shown this is cut section. Now, this is the pinion root ok. So, this bearing has to rest on the sholder of the shaft therefore, shaft should be something like that there should be stepped down. Now, usually this chased down step down over the diameter not less than 7 mm, 10 mm is good, 10 mm or more is good. So, this means that this means that if we consider the bearing ID so, we should not it should be around 25 mm 30 mm will be difficult so, 25 mm. Now, we have considered the bearing life as recommended it should be about 10000 hours.

Here, it can be mentioned of course, that de depends on the experience or you have one should have some idea. 10000 hours is usually in higher side and an experienced designers he at this moment we may consider that bearing ID 25 will not be more enough, it should be at least 30 and if it is 30 mm to make it so, we need to increase the root diameter of the pinion also.

This means, there is no way we have to go for higher module, although on the strain basis design on wear load design it is 2.5 is enough, but we are going for 3 module considering the internal diameter of the bearing at the 1st stage also I would like to mention here that, as this pinion is very close to this bearing this bearing will have more load than the other side bearing therefore, 25 keeping the diameter 25 may be a risk that is why 3 module is.

So, 1st stage module 3 mm now, we should calculate the 2nd stage module also.

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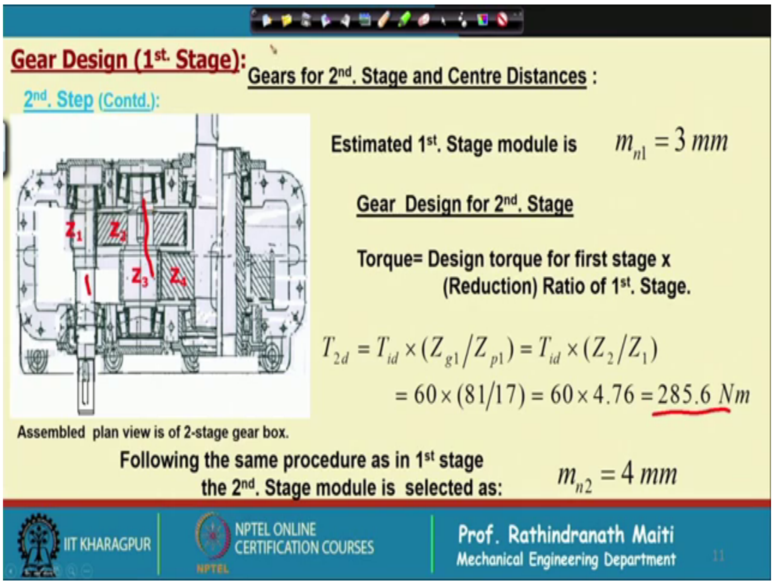

Now, for the 2nd stage 1st stage module is 3 mm, for 2nd stage the torque will be design torque for 1st stage multiplied the ratio of 1st stage. In this case this is we have considered the torque in the shaft. Which is output shaft in this case t twice d; that means, we are no this is designed of for the gear to design for the 2nd stage. So, 1st stage torque need now, we are considering the torque of this shaft intermediate shaft. This shaft this torque is nothing but this T2d means 2nd shaft and this is Tid is the input torque here and that multiplied by this ratio gear ratio 1st stage, which gives 60 into 80 one pie by 17 that becomes 285.6 Nm; that means, the torque here is 285.6 Nm. So, we have to design the 2nd stage gears that is Z 3 into Z 4 these 2 sets on the basis of the torque on the pinion to 285.6 Nm.

So, in the same procedure if it is calculated we the calculation is done in the same maintaining the same procedure, and it was found probably 3.5 mm would be quite satisfactory for the 2nd stage, but if we look into the size of the gear box if it is 3 mm, and if we give this is 3.5 there will be there is a possibility that here, the gear will foul. So, we have taken for the 2nd stage 4 mm. So, we have finalized the 2nd stage module is 4 mm.

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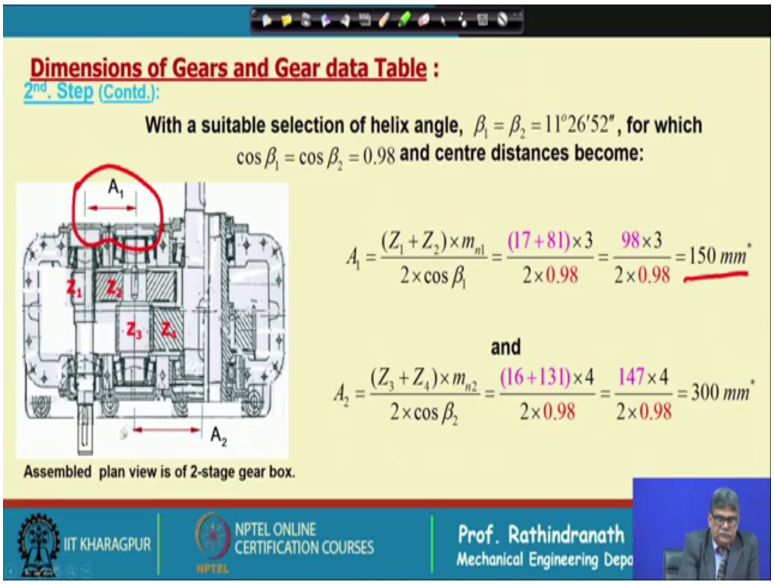

Now, helix angle, Helix angle we have considered it is 12 now, if we give this is 12 then, the factor at the denominator that cos beta which is coming into the pitch circle diameter as well as, in the center distance. We will find that number and that that value will be a fraction, which is not desired. It is to be mentioned for standard gearbox.

Industrial gearbox usually the center distance kept multiple of 10 mm; however, multiple of 5 mm is oso also acceptable, but other odd values are not normally accepted. Now, we have intentionally chosen the teeth number in such a way, with this value a slight modification in helix angle will get the center distance very square value. Now, what we have considered that cos beta 1 and beta 2; that means, helix angle in 1st stage and 2nd stage both are equal and 11-degree 26 minute 52 seconds for which cos beta 1 and cos beta 2 becomes 0.98, but here I would like to mention, it is not necessary that we have to give the helix angle same for both stage, it may be different.

Now, according to that this center distance the 1st stage center distance A1 this is calculated Z1 plus Z2 into module 1st stage divided by 2 I t into cos of 1st sta stage helix angle. So, this becomes 98 into 3 divided by 2 into 0.98, which 150 mm a very good value. Similarly, if we calculate for 2nd stage that becomes 300 mm the calculation is shown. So, that is also a good value this is the 2nd stage.

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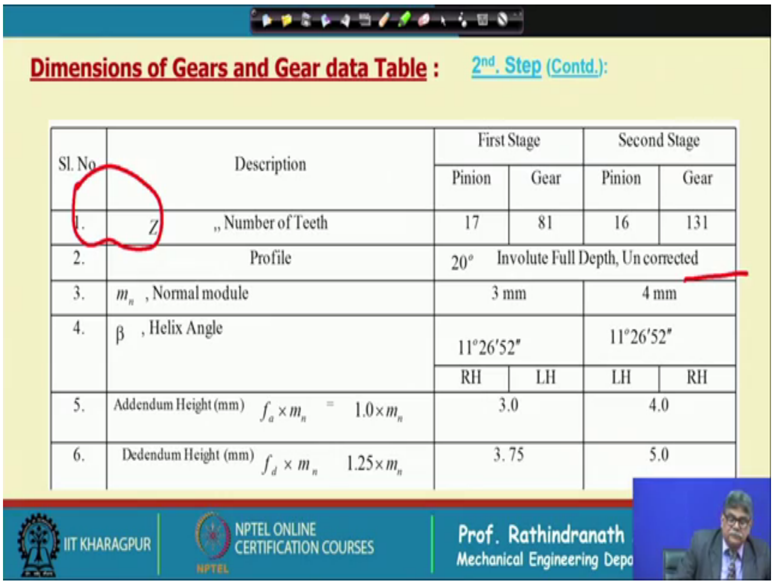

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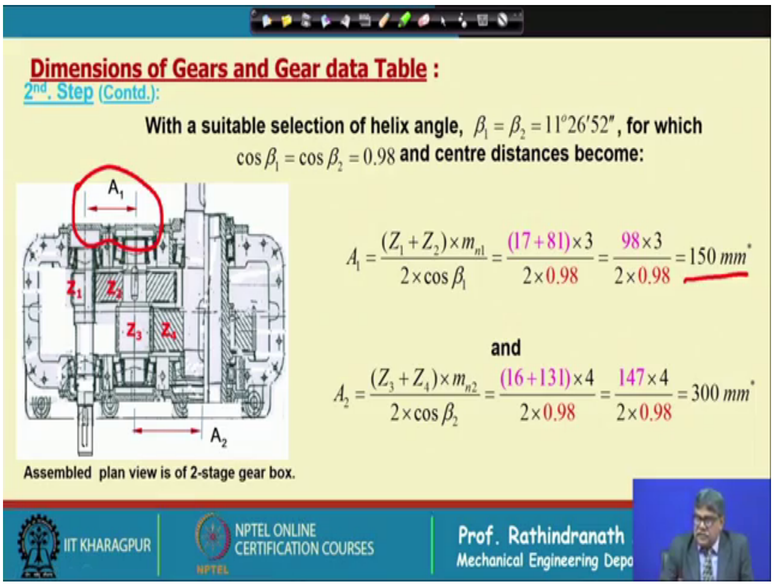

And then, we calculate the gear data or gear dimensions these are in the tabular tabular form. This with the drawing and also the below material this data is provided. This is for the manufacturers and as well for also for further design, what we find the 1st stage pinion is pinion teeth number is 17 and gear is 81, 2nd stage 16 131. Pressure angle 20 degree a stage module 3 mm, 2nd stage module 4 mm, and helix angle both are same for both stages which is 11-degree 26 minute 52 seconds.

And now, in 1st stage we have taken pinion is right hand gear is left hand, and 2nd stage pinion is left hand. Now, 2nd stage; that means, intermediate shaft both gear and pinion in the same direction of helix that is an important act that you we shall discuss a little later . In the next lectures and; obviously, this pinion and gear should be in the opposite direction. So, it is right hand and addendum height is 3 mm ah. In case of 1st stage 2nd stage 4 mm dedendum height is 3.75 and 2nd stage 5 and the pitch circle diameter 52.4 pinion and gear to 247.96 1st stage. 2nd stage 65.306 and 534.69 and so on, this addendum diameter materials face width.

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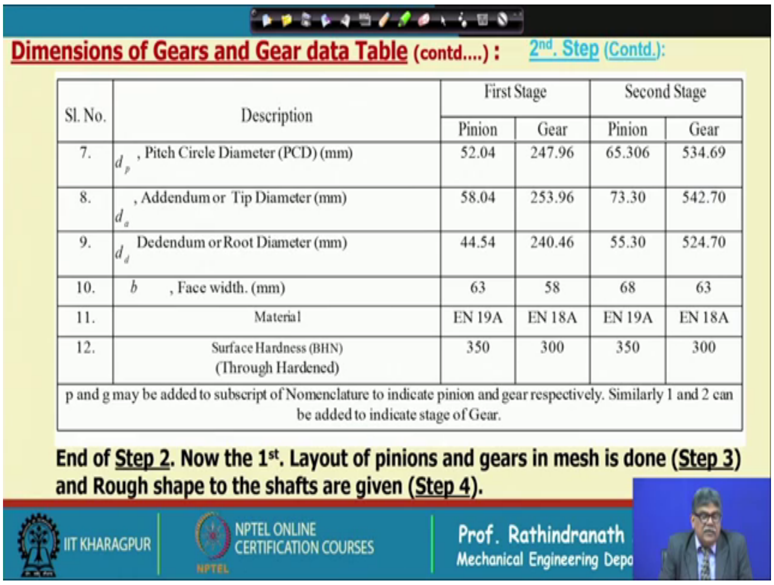

Now, another thing it is to be noted that width of the pinion is usually taken 5 mm more than the gear whereas, gear width is exactly taken as what we have calculated for the active contact length. Now, I shall discuss later why we take the 5 mm more in case of pinion later, and surface har hardness for pinion it is 350 and for gear it is 300 in both stages. So, this is end of step 2 and now, we need to make a layout of the gear and pinions. So, that we can find out the, what will be the shape of these shafts before going into the next step of calculations.

Thank you.