Machinery Fault Diagnosis and Signal Processing
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Lecture - 09
Rotordynamics

Welcome to this lecture on Rotordynamics.

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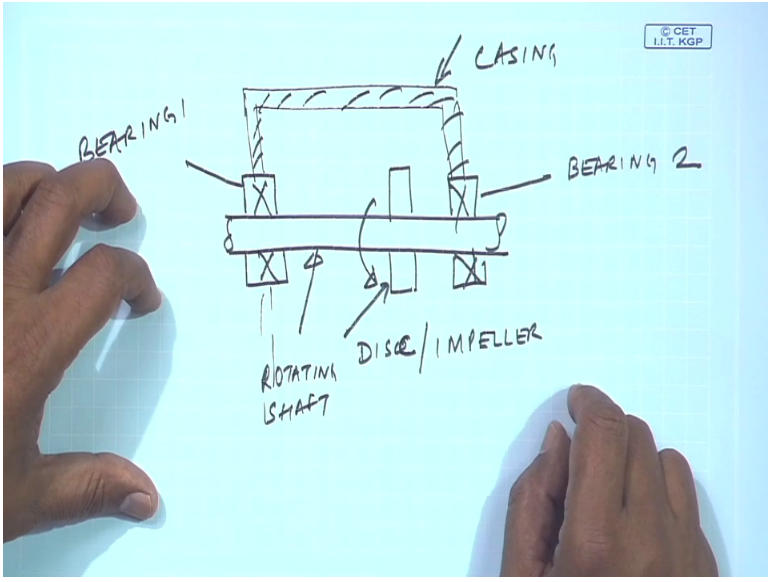

As you know every machine actually essentially comprises of a shaft which is supported on bearings and invariably it rotates and this there could be a housing etcetera and so on, if we just draw 1 half and this shaft could be carrying a disc. So, these are my bearing 2 bearing 1 this is my disc, which could be an impeller and this is the rotating shaft and this is the machine casing; I am sorry there is another 1 and this side which I am not drawing.

Now in rotordynamics we see and if you think of any machine beat electric motor, beat pump, beat gearbox, beat IC engine, essentially each 1 of them consists of a shaft and the shafts will not be there in area they are supported on bearings and in the shaft rotate they could be carrying and disc or an impeller; think of a large turbine, think of a compressor think of an aircraft engine. So, all these machines or rotating machines have a shaft which is supported on bearings.

So, rotordynamics studies the dynamics of such rotating shafts which are subjected to different conditions.

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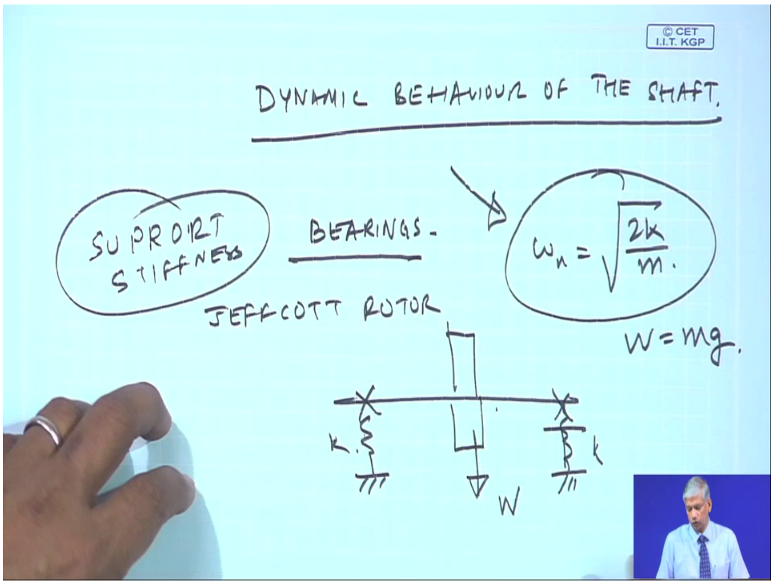

Now, what are the conditions here, it is essentially how does the dynamic behavior of this of the shaft, how does it affect the machines loosely same vibration level, let us say machine stability etcetera. Now rotordynamics we will see these bearings play a very important role and there are many simple models to study this and 1 is the simple JEFFCOTT rotor, which is nothing rotor shaft supported on bearings carrying a disc.

So, if you think of this shaft which has a load w acting and then this is supported on some stiffness with these bearings give some stiffness k, you will see the natural frequency of the system omega n is nothing but in this case 2 k by m, where W is equal to mg. So, you see this system has a natural frequency and which is influence obviously, by the mass carried by this shaft and this support stiffness.

So, support stiffness is very important and they control what is known as the natural frequency of the system, now we know from our basic theory on vibrations that any external forcing frequency omega, if it is equal to omega n we have the condition of resonance right.

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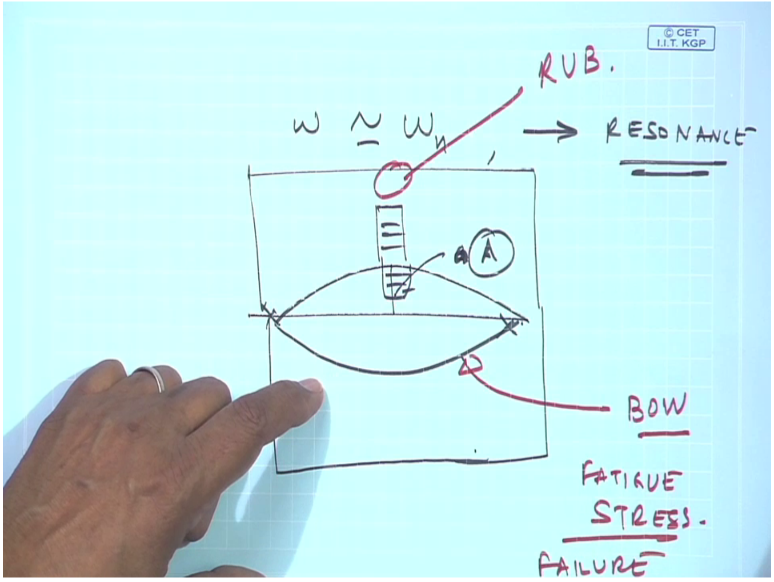

So, at resonance what happens the amplitude of the motion would increase, this amplitude could come to a level that at resonance may be this is and this amplitudes is may be undesirable or some amplitude a which is undesirable and now there are many reasons why this amplitude has to be unnerved and control.

So, because imagine if this was a long shaft and it was carrying a set of blades and this came across the casing. So, what could happen once this touches the casing it will give a phenomenon of rub, if this flexes is bow fluxes too much it could induced fatigue stress on the shaft and there could be failure. So, this will be giving you an idea the importance of understanding the dynamics of a rotating shaft which is supported on bearings. And then all the designs associated with ensuring that, I have a stable design of the rotating system is what will studying in rotordynamics.

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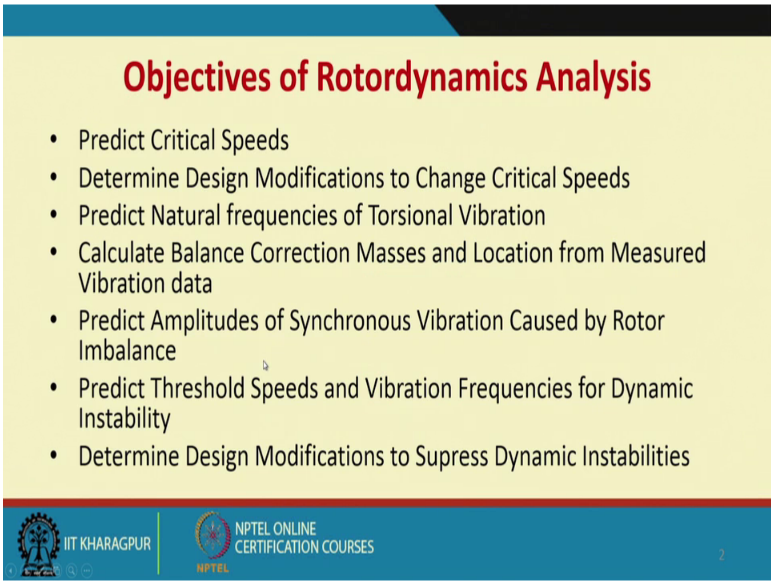

So, what are the objectives of rotordynamics, 1 is the predicting the critical speed because the shafts could be having the natural frequency because of rotation.

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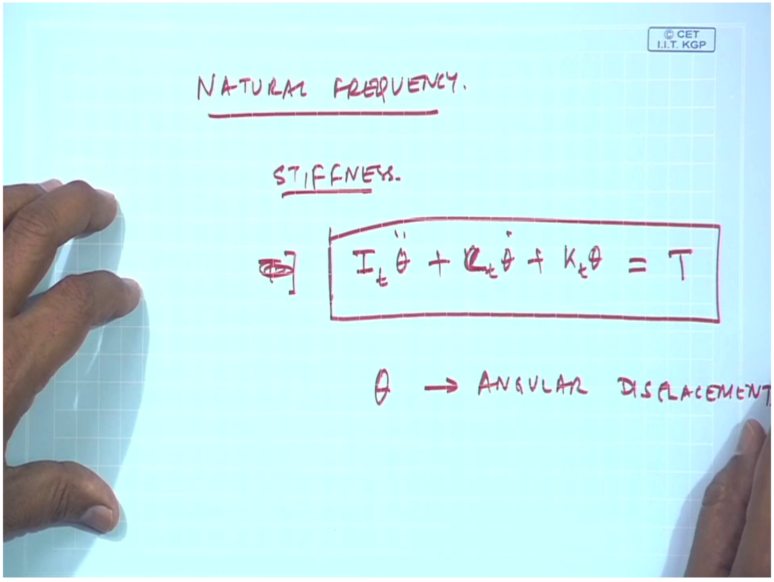

So, 1 should always avoid operating the shaft at the critical speeds, we can do certain design modifications to change critical speeds; so playing around with the stiffness predicting the natural frequencies of torsional vibration. So, far we had studied about the linear vibration, but in the torsion vibration systems I have what is known as the inertia term plus the damping term and the stiffness term, which is equal to the external torque in a rotating system.

So, this theta is the angular displacement and the subsequent derivatives of the angular velocity and then angular accelerations. So, even in a rotating shaft I would not like to have high motions or on stable motions or large motions and so on and then many things happen in these shafts.

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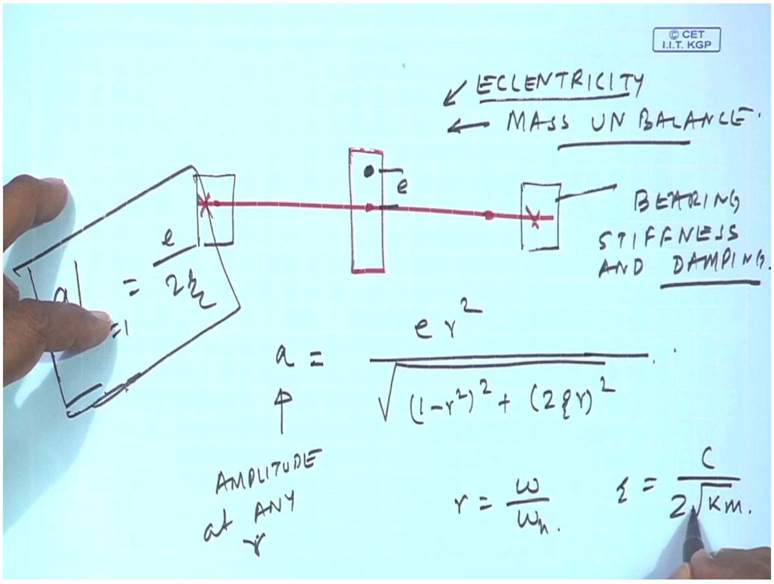

We have assumed that the shaft to be concentrated between the 2 bearings; that means, the geometry center of all the systems are in the same line, imagine if there was an or an unbalanced or an eccentricity in this soft in terms of a geometric eccentricity e.

So, what is going to happen to these displacements at certain speeds omega is again a factor which we need to study. In fact, if this eccentricity e is e the displacement is given by a is equal e r square by 1 minus r square, a is the amplitude of vibration at any r where r is nothing but omega by omega n. So, at r is equal to 1 you will see in this amplitude is again controlled by a at r is equal to 1 is nothing but e by 2 zeta. So, this is an important conclusion from vibrations that at resonance all the amplitudes are controlled by playing around with the damping.

So, we can introduce damping in the system to reduce this amplitude and in rotordynamics people design on systems, where in that they can have some external damping played around at the bearing stiffness and we will come to this later on. So, another reason was I or once I told 1 was eccentricity, where there is a difference in the geometric centers and other is the mass unbalance.

We will talk about mass unbalance in the later chapters, but in the eccentricity case we can find out the amplitude at any point r given by this expression, where zeta is equal to c by root over 2 km you already know that ok and r is equal to omega n.

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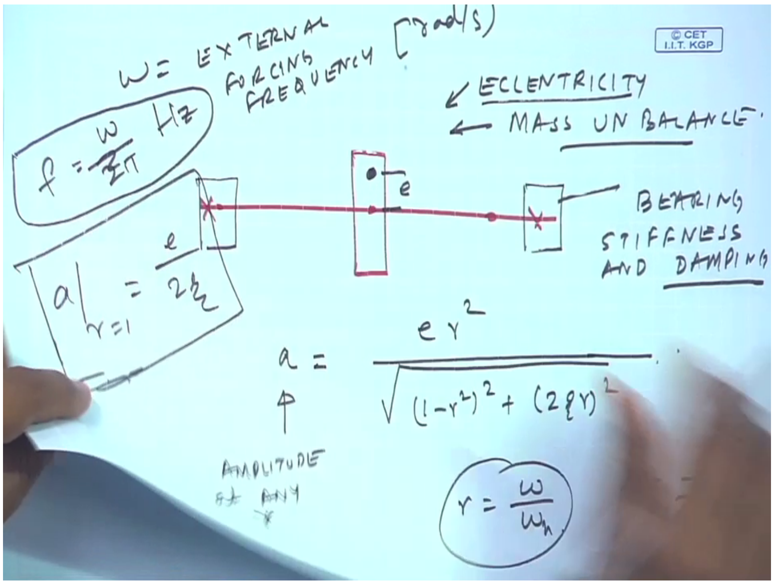

Omega n is the natural frequency system and omega is the external forcing frequency of causes this is circular frequency. So, the unit is radians per second and not hertz, where f is equal to omega by 2 pi and this given in hertz.

Now what happens in rotordynamics? We also would like to find out the amplitudes of synchronous vibration caused by rotary imbalance and predict these thresholds speeds and vibrations frequencies for dynamic instability.

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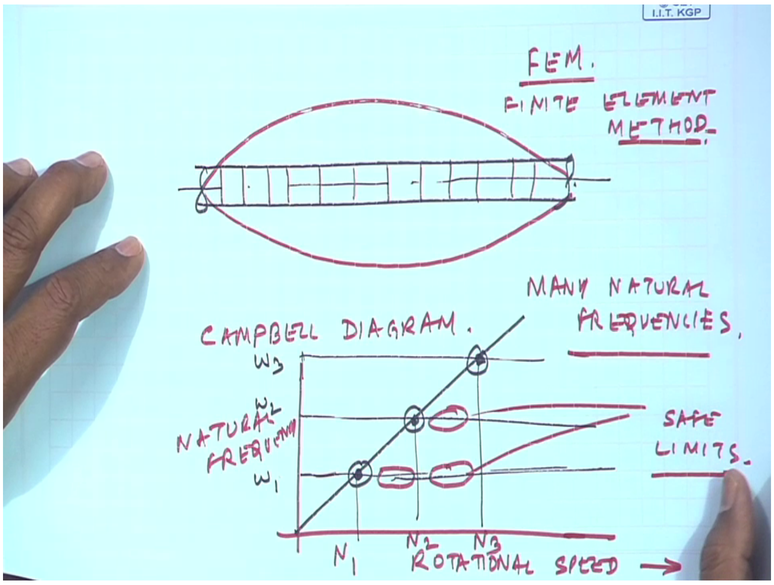

So, I will just explained you with 1 degree or freedom system, but you will see this shafts are strictly not rigid; if they were rigid they would not have any deflections. So, they are elastic so that is why they deform while rotation they will bow out.

Now, question is because I could divide this shaft into many degrees of freedom. So, I can have strictly speaking theoretically speaking in finite degrees of freedom system, but now within the limited range there will be many natural frequencies of this shaft. So, there is a very important diagram which is used in rotordynamics which is known as camp bell diagram, which plots the rotational speed to the different natural frequency of the system.

So, if shaft like this will have many natural frequencies, this is the 1 horizontal lines. So, I can have the rotational speed such that rotational speed, this is 1 cross over this is another cross over. So, these are regions where I should not operate my machines. So, maybe this is 1 N1 rpm N2 rpm N3 rpm and this is maybe the natural frequency omega 1 omega 2 omegas 3.

So, this region is only the safe operating region, if pressure going in this speed so there are regions of safe limits where in resonance can be avoided. So, you know people spend a lot of trying, a lot of doing analysis no to find out when the designer system. How do I find out the natural frequency of such long rotor shafts? we will talk about turbines we talk about compressors because, as an operator you are not allowed to run at N1 because N1 speed corresponds to the natural frequency ok.

So, you should move around and shift your N1, so that it does not interfere with the or co coincident with the natural frequency of the system. So, such kinds of diagrams Campbell diagrams are developed. In fact, how can you 1 do that people use the standard techniques of finite element methods to analyze the system, because we obviously cannot make such analysis by actually physically constructing a shaft and going through this.

So, as a machinery operator or a designer we have to specify, what is the safe limit for such machines which I have multiple operating speeds. So, that 1 avoids critical speeds and this conditions you will see when a particularly you know the speeds are very high.

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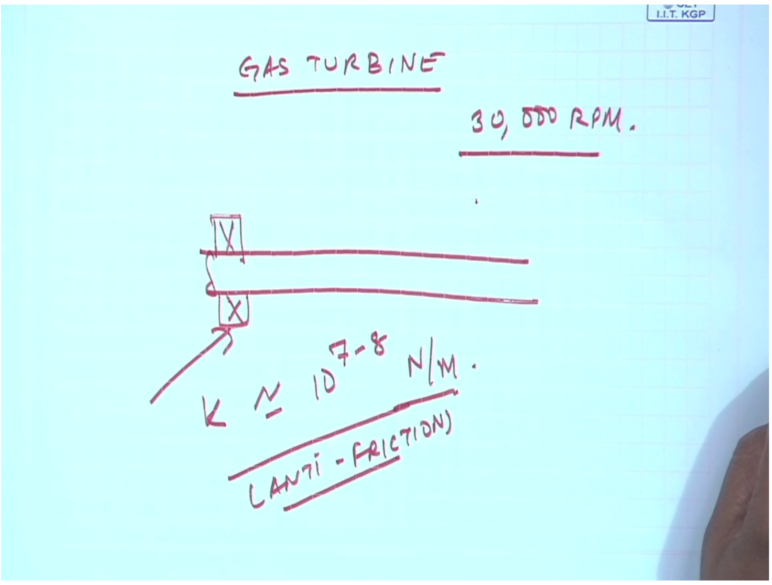

When you are operating a gas turbine we cannot really start and because gas turbines you know typically 30000 rpm. So, by the time you reach 0 to 30000 rpm when yours cranking up or start up did not start up we will cross many natural frequencies. So, you are not allowed to operate it at any of the natural frequencies otherwise residence would occur and fatigue set in.

So, you 1 is quickly asked to pass through these resonant conditions very quickly and all of you must have witnessed it particularly sitting in an aircraft, with the pilot has just started the engine and it is going up revving up to full speed you will see they will go for regions from we will see suddenly a large vibrations and then it will disappear and. So, on and. So, when you feel a large vibration that means we are just passing through the first critical speed second critical speed and so on.

So obviously, many times particularly in plant machineries there is always a limit that this is the safe operating speed and that is therefore any rotating systems, so but then 1 of the other objectives of the rotor dynamic analysis it to suppress such dynamic instabilities.

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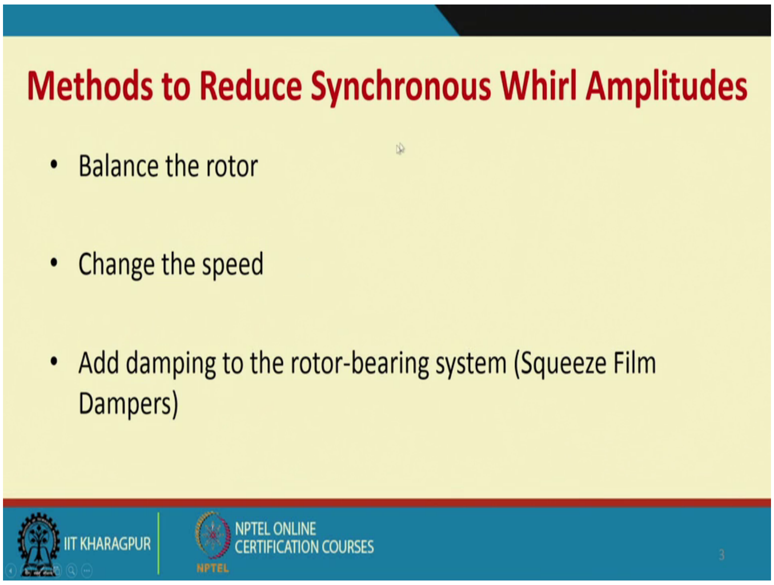

Well, to method the reduce this synchronous whirl amplitudes; obviously, if I have the balance the rotor, that means the geometric center the mass center they are all concentric and change the speed avoid the critical speed and of course add damping to the rotor bearing system in the case of the squeeze film dampers.

Particularly if you talk about bearings you know we come to that later on. So, your some of these bearings are anti friction bearings and the journal bearings. So, there is a certain stiffness the anti friction bearing stiffness is very high out of 10 to the power 7 to 8 Newton per meter, we high and this is being a very rigid in the case of anti friction bearings, we do not have much play in it. But if you talk of other bearing like the journal bearing or the fluid film bearing, we can have provision to add certain squeeze film dampers we will come to that later on.

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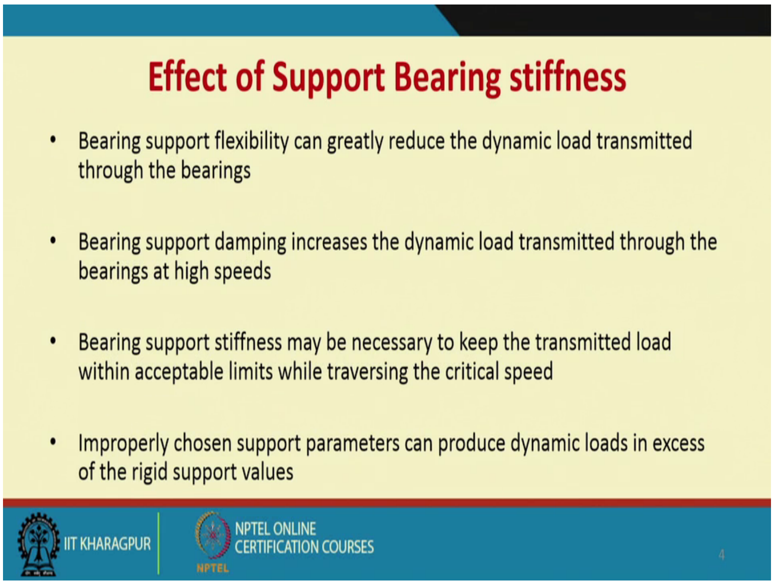

So, what does this bearings support stiffness do? So, bearing support flexibility can greatly reduce the dynamic load transmitted through the bearings increases, this damping increases the dynamic load transmitted through the bearings at high speeds, bearing supports keep the transmitted load within acceptable limits while traversing the critical speed because, I was telling you once you go from 0 to 3000 or 30000 rpm there will be many critical speeds 1 and will be traversing through, so 1 has to go through these methods.

So, 1 good idea of controlling these stability of rotating shaft at high speeds is to play around with the bearing stiffness.

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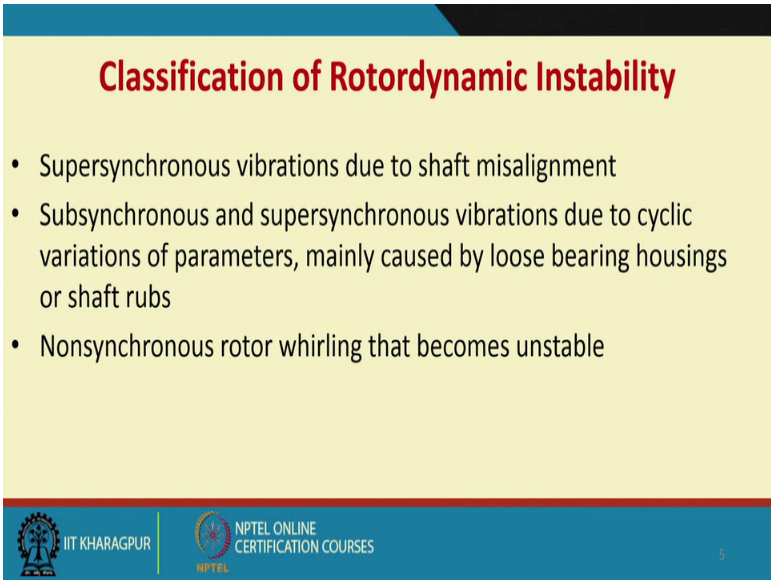

And of course, you know there will be some super synchronous vibrations sub synchronous vibrations because, of shaft misalignment those components and so on.

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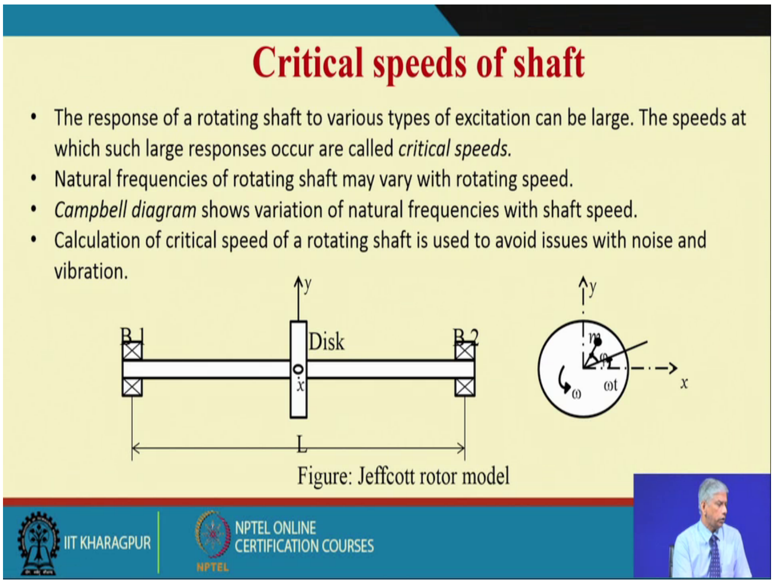

So, this is what we are talking about this critical speed, the response at such large when our critical speeds, natural frequency will vary with the rotating speed and like I told the Campbell diagram. So, the variation of natural frequencies it is shaft speeds.

So, we need to calculate the critical speed of rotating shaft, so as to avoid issues with noise and vibration and that is the analysis people do for large rotors, now if you think of a gas turbine ok.

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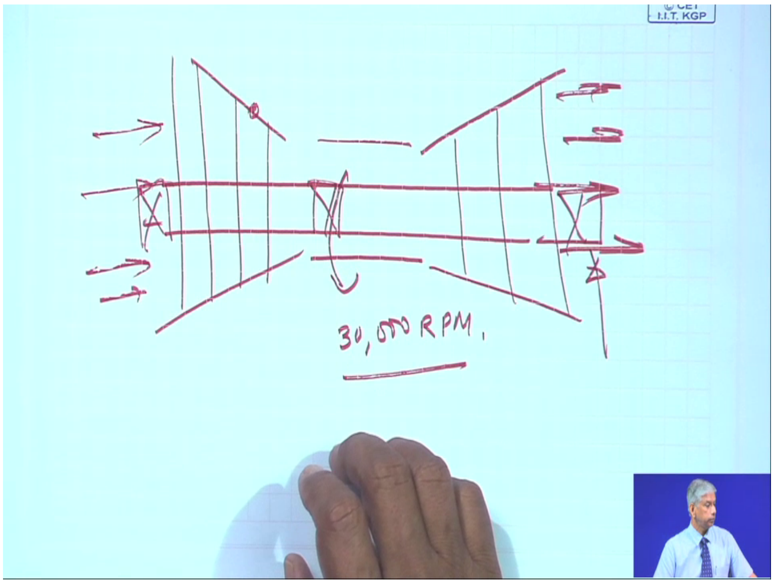

There will be sets of compressor blades you know multiple stages of compressors, when we compression chamber and then the turbine. So here going in and this shaft rotating at very high rpm 30000 rpm, so imagine if the exhaust coming out and then you get the thrust ok.

So, this all many critical speeds, so if 1 sort of critical speed now there will be a rub here. So, there will be instability and they are supported on bearings. So, 1 has to design this bearing stiffness as bearings. So, and the stiffness and damping so that the amplitudes are within limits, so one such study which we did with finite element method.

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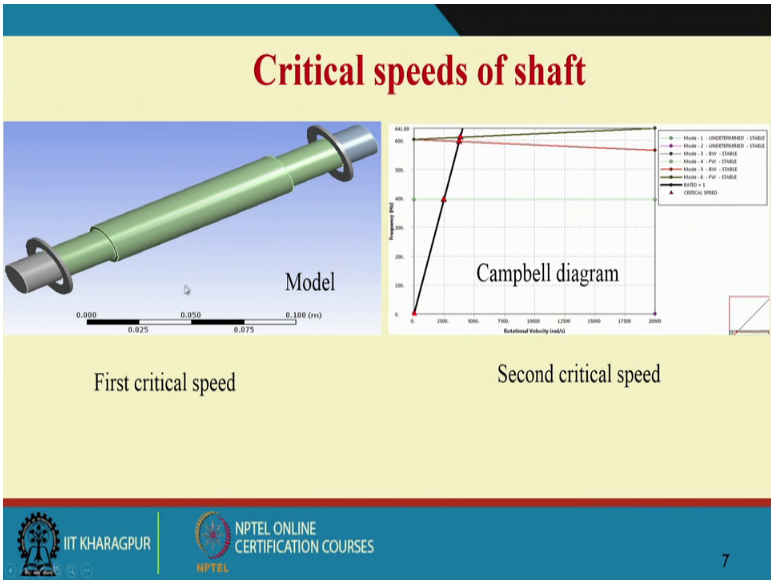

So, this is a shaft as you can see right, which has been modeled in finite element model the shaft supported on bearings. So, these are the bearings are represented just step did the step for the bearing and the bearing shaft or these shaft main shaft and another step in the bearings.

So, you will see this is the rotational velocity in radians per second which could be also in rps and these are the different natural frequencies. So, this is the first natural frequency comes down about 400 hertz. And then the second natural frequencies of this, gives you an operating idea from the Campbell diagram that this is the only the safe limits you have to avoid these critical speeds. So, different modes the critical speeds are indicated here and for different ratios where it is stable unstable 1 can study this ok.

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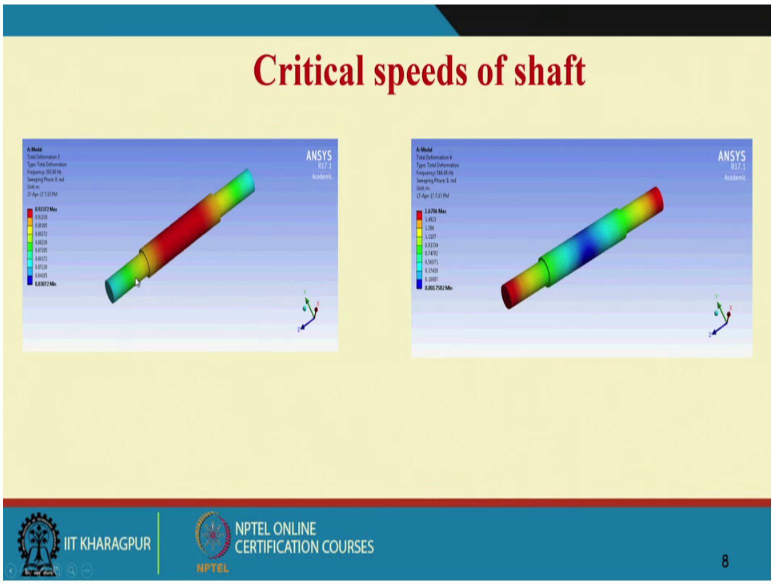

So, if you look at this mode shape of the critical speeds of the shaft, you look here that this red means maximum displacement; that means, this is going out like this now imagine this. So, the factor which was controlling these natural frequencies of course, you know if you the inertia of the shaft is not to be changed, it is this bearing stiffness which would influence then natural frequencies. I know we can do by the expression omega n is equal to root over 2 k by m, but to be more accurate you can do finite element model and then do it and this is the second critical speed.

So, here you see there is maximum deflection so in the shaft is flexible. So, it is going having a low motion like this. So, there is an actually a whirling will occur and then little bit better diagram of whirling in subsequent classes.

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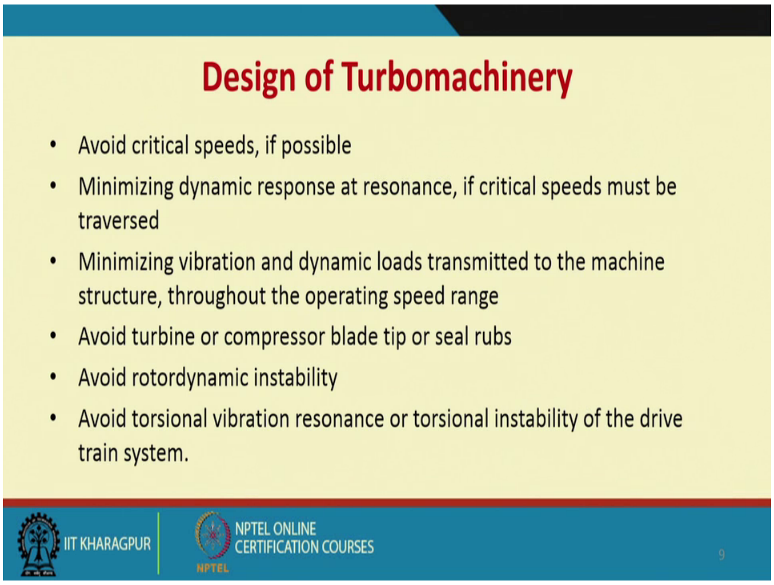

So, when you study rotordynamics we design a turbo machinery first and foremost is to avoid critical speeds, those of you who would have designed the shafts from simple strength of materials, we other than calculating the torsion strength of the shaft to withstand the external torque, we also must find out the critical speed of the shaft. So, that it never operates at the critical speed and then we are away from the vertical speed. while minimizing the dynamic response at resonance if critical speeds has to be traverse will when you go up to high speeds, there will be many critical speeds.

So, this has to be traversed and of course minimizing vibration and dynamic loads transmitted the machine structure throughout the entire operating speed range, avoid turbine or compressor blade tip or up seal rubs, because it is rubbing would occurs as I was telling you because of large motions.

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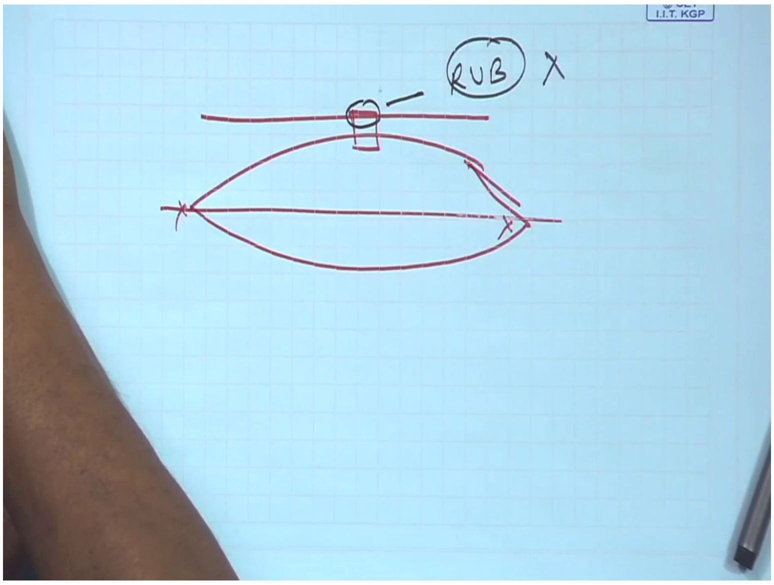

So, this is going to hit tip casing, so this rob would occur that has to be avoided and avoid torsion vibration resonance or torsion instability of the drive train system.

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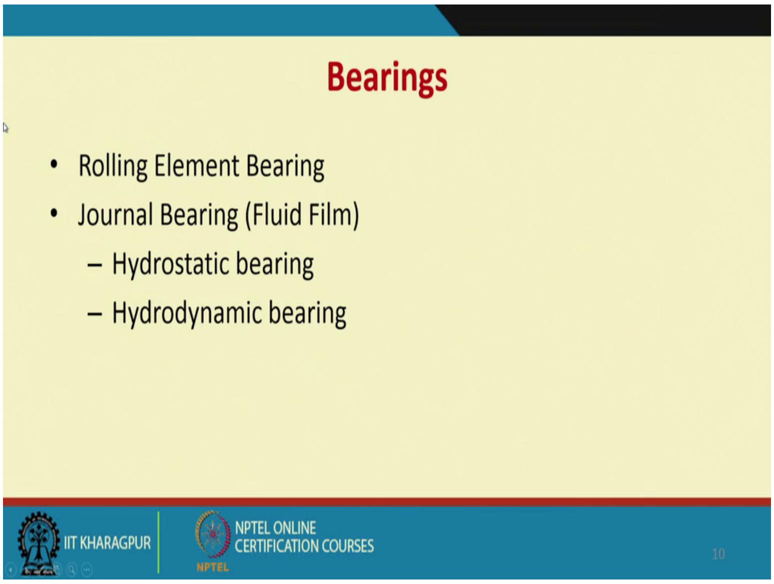

So, this bearing plays a very important role in the shaft. So, we have this rolling element bearing and the journal bearing or the fluid film bearing. So, in the rolling element bearing as I was telling you they have almost a constant high stiffness, but in the hydrostatic bearing or the hydrodynamic bearing, we can play around with the bearing stiffness depending on the viscosity of the oil depending on the eccentricity and so on.

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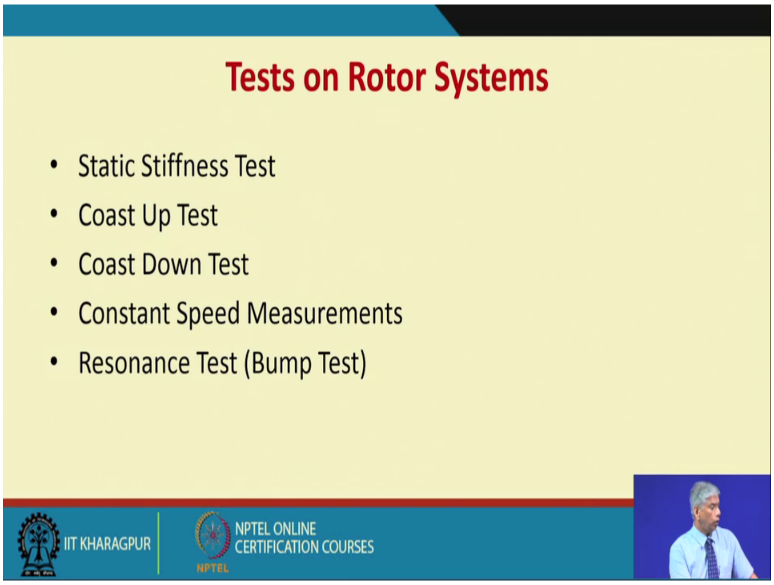

So, certain tests are done on the rotor systems, 1 to find out the natural frequencies 1 is the static test stiffness test.

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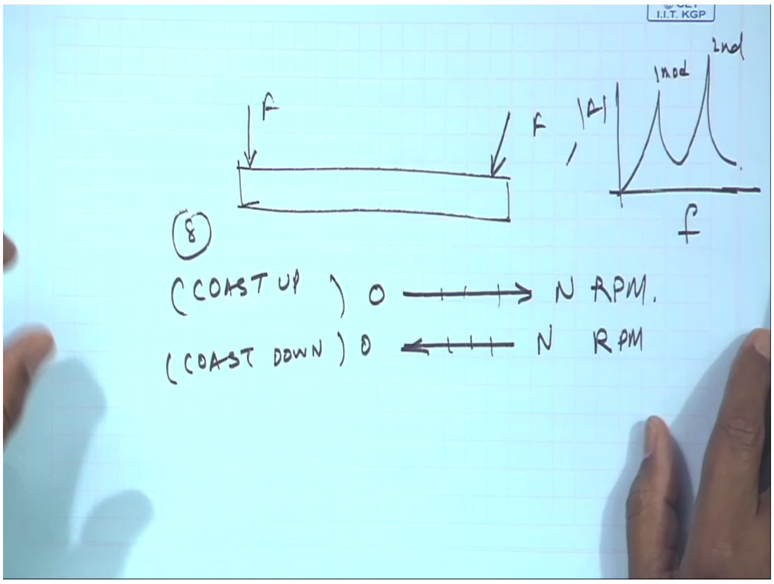

So, you will give certain force and measure the deflection, but another very important test which people do is coast up and coast down test. So, that you increase the speed from 0 to certain n rpm at a constant rate and then from n to 0, this is the coast up and this is the coast down and then the residents test is like a bomb test we give certain force and then from measured the natural response and plotted it is frequency and then. So, this is the first frequency first mode second mode and so on certain aptitudes, but whenever there is this coast up and coast down, we can see at certain rpm there will be high amounts of vibrations and these are the places where resonance will occur.

So, in summary in rotordynamics we understand the dynamics of a rotating shaft, which is inherently present in any machine and the effect of the bearing stiffness and how does the bearing stiffness play with the critical speeds of the shaft and no shaft is actually strictly speaking a rigid they are flexible. So, they will have many critical speeds, so our designer of rotordynamics would give the operative guidelines as to 1 of the safe operating limits of your machines and the reason these dynamic instability is occurred because of eccentricity because of unbalance. So, these are not looked in total we will have issue where and we could be unknowingly driving it at resonance and it will create other machinery problems.

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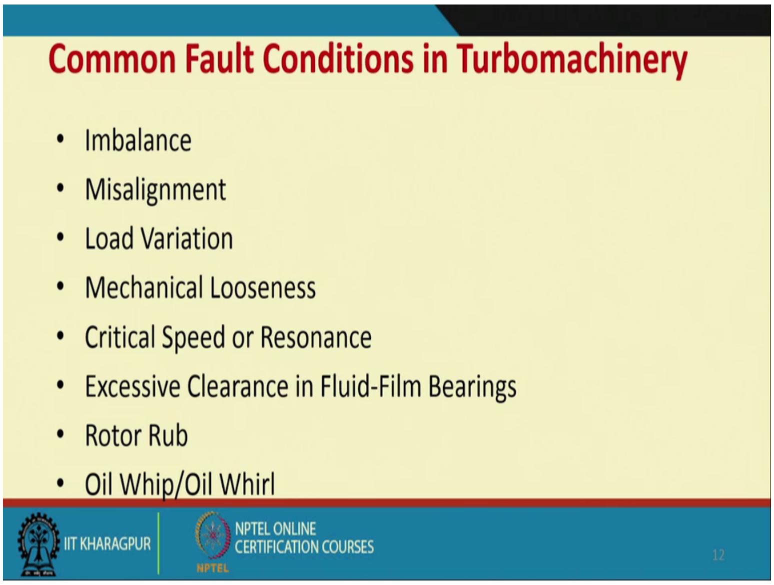

So, some of the common fault conditions in turbo machinery which we will be seeing later on is imbalance, misalignment load variation, mechanical looseness, critical speed or resonance excessive clearance and fluid film bearings rotor rub oil whirl and oil whipping.

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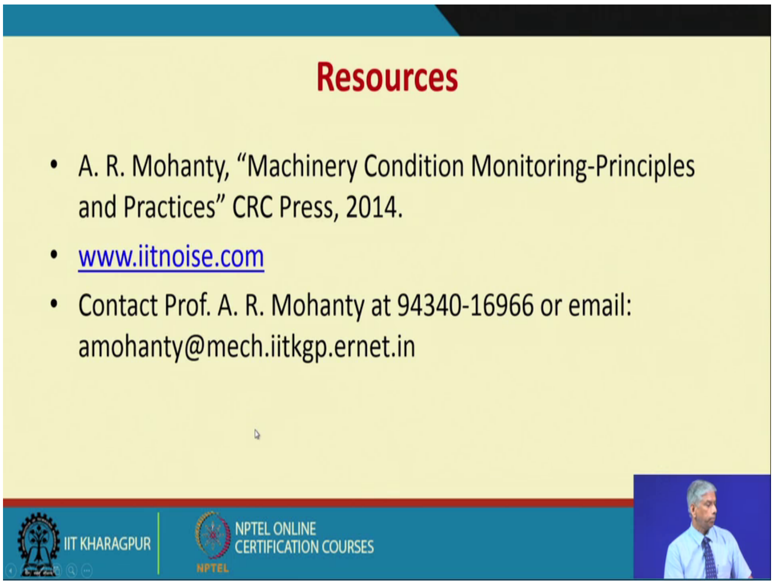

Well, more about this can be found in my book on machinery condition monitoring.

Thank you.