Theory and Practice of Rotor Dynamics
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Module - 9
Condition Monitoring of Rotors
Lecture - 41
Common Faults and Vibration Signatures

Previous lectures, we have been dealing with one kind of common fault in rotating machinery. That is the unbalance. Apart from this unbalance, there are several other faults, which may appear in rotating machinery. Those obviously reflect in a vibration signature, because whatever the dynamics is taking place inside the machine that reflects as a vibration. If we measure the vibration onto the machine body or any other component, then we can able to have some idea about the condition of that particular machine component. So, in the today's lecture, we will be dealing with various kind of a common fault in the rotating machinery.

How it affects the vibration signature, we will try to analyze. These vibration signatures are there in time domain. Generally, we capture through some sensor. Then, we or as, generally these whatever the single comes the analog form, then we convert into the digital form using analog to digital convertor, so that we can able to store the data in a computer. Then, we can able to do the processing of those data. Either we can convert that time domain data into frequency domain or any other domain, in which these signals can be interpreted.

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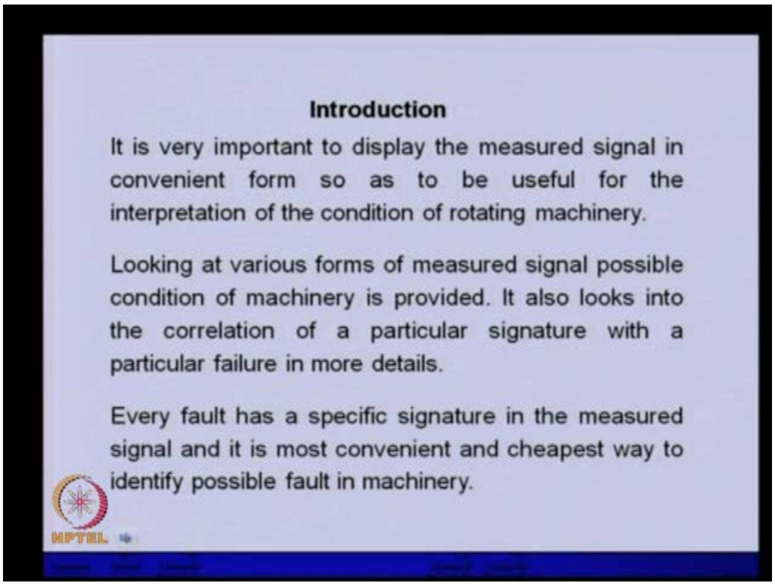

So, we will start the introduction to this particular topic. So, it is very important to display the measured signal in convenient form so as to be useful for the interpretation of the condition of the rotating machinery. As I mention the measured signal, we need to display in a convenient form which will be useful for the finding the condition of the rotating machinery. Thus, we will go into the deep into the lecture.

We will find how these signals can give condition of rotating machinery. Now, looking at various forms of measured signal, possible condition of machine is provided. It also looks into the correlation of a particular signature or the vibration signature with a particular failure in more details. Every fault has a specific signature in the measured signal and it is most convenient and cheapest way to identify possible fault in machinery.

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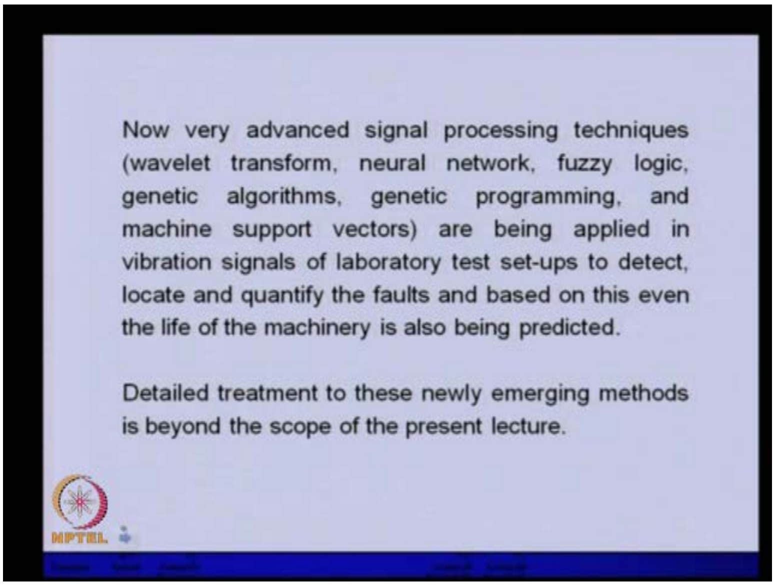

Now, very advanced signal processing techniques like; wavelet transform, neural network, fuzzy logic, genetic algorithm, genetic programming and machine support vector SVM, are being applied in vibration signals of laboratory test setup to detect, locate and quantify the faults. Not only the detection of the fault, now with these techniques we can able to pin point, where is the fault and even we can able to quantify the fault.

Based on this even the life of the machinery is also, being predicted. How much we can able run the machine without much danger. So, detailed treatment of these newly emerging methods is beyond the scope of the present lecture. But mainly, we will be focusing on based on the signature based on the vibration signature. We will try to correlate various kinds of faults in the present lecture.

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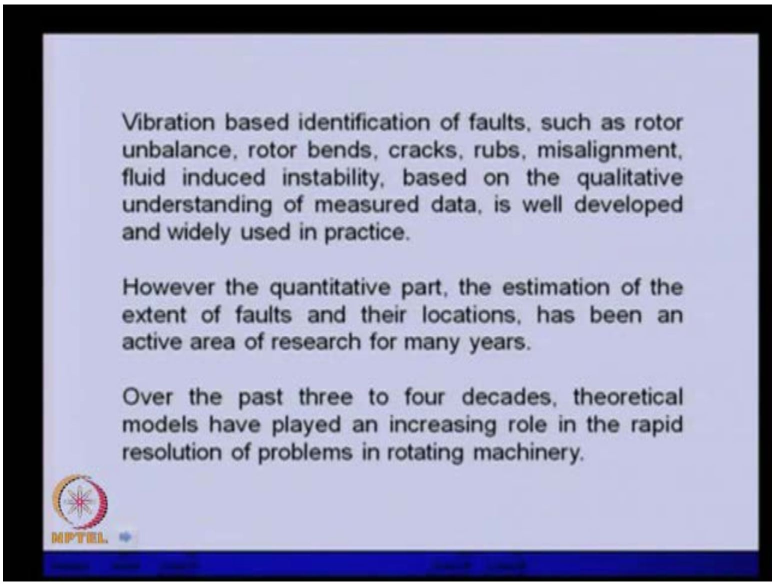

So, vibration based identification of fault such as rotor unbalance, rotor bends, cracks, rubs, misalignment a fluid induced instability, based on the qualitative understanding of major data is well developed and widely used in practice. So, even this is being applied in actual machinery in industry also. However, the quantitative part the estimation of the event of fault or the extent of the fault and their location has been an active research for many years. So, qualitative we can able to detect the fault, but quantifying that is still a research area. Over the past three to four decades, theoretical models have played an increasing role in the rapid resolution of problems in rotating machinery.

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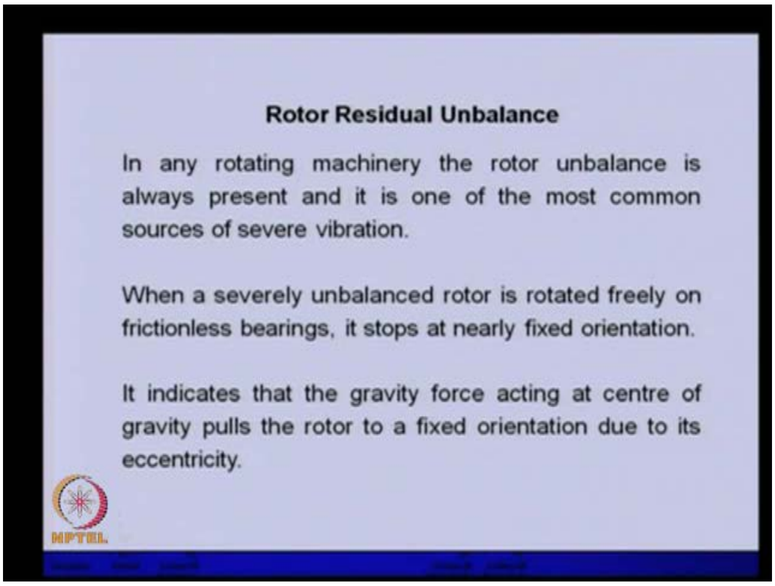

So, the most common fault, which we already sees the unbalance. So, let us see slightly more detail on this. So, in any rotating machinery the rotor unbalance is always present. It is one of the most common sources of severe vibration in rotating machinery. When a severely unbalanced rotor is rotated freely on frictionless bearing, it stops at nearly fixed orientation. So, this is nothing but the hysteretic balancing of the rotor. It indicates that the gravity force acting at the centre of the gravity pulls the rotor to a fixed orientation due to its eccentricity. So, this we already discussed regarding the hysteretic balancing of a practical rotor, how we can able to do it.

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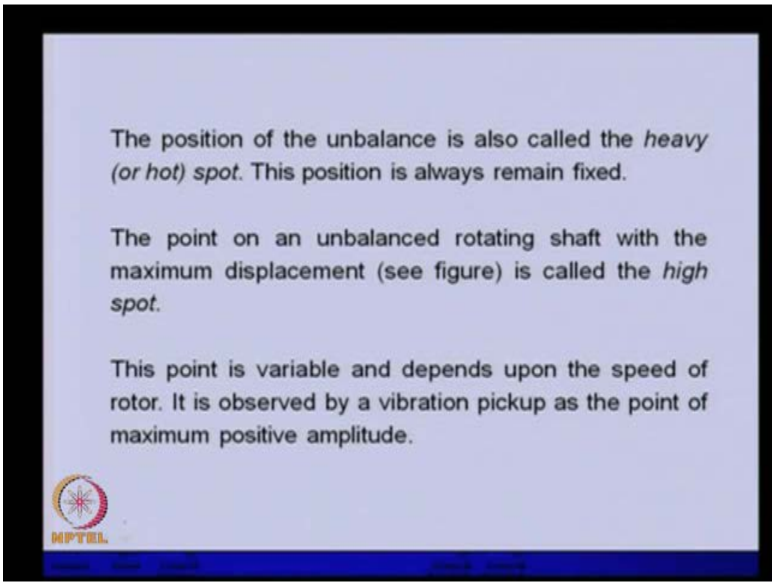

So, there were two terminologies. One is the position of the unbalance that is we call it as heavy or hot spot. This position is always remaining fixed onto the shaft. The point on an unbalanced rotating shaft with the maximum displacement is called the high spot. So, that is heavy spot. High spot is the where the maximum displacement of the rotor take place in the axial position of the rotor.

This point is a variable and depends upon the speed of rotor. It is observed by a vibration pickup sensor as the point of maximum positive displacement or amplitude of vibration. Once we are rotating the rotor in the flexible mode, we know that this high spot may change. So, that is why it is depend upon this particular speed.

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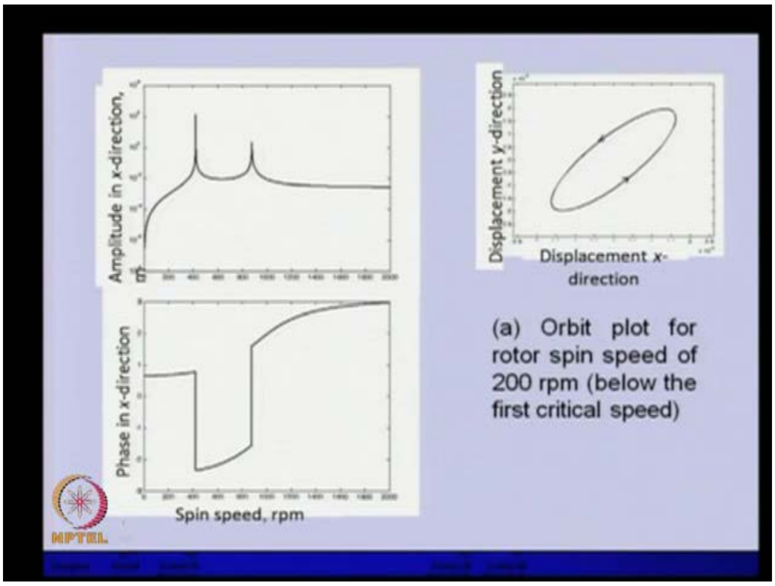

So, this is a typical unbalance response. This is you can able to see there are two critical speeds of the rotor. This is the phase plot. So, phase is also changing wherever there is a peak. So, this is another way of identifying, where are the critical speed. This is the orbit of the shaft. That means in vertical direction or horizontal direction displacement can be plotted like this. We can able to see even in some instruments. We can able to see the sense of rotation of the particular orbit using a z direction.

Even in some oscilloscope you can able to see this particular orbit the sense of rotation also using a third axis. So, x and y and another axis that is z axis in that, we can able to if it is a reference signal. With the help of that we can able to show the sense of the orbit also. This is below the critical speed. So, critical speeds are there. You can able to see at. So, 200 r p m is below the first critical speed. So, sense of rotation is in this particular case counter clockwise direction.

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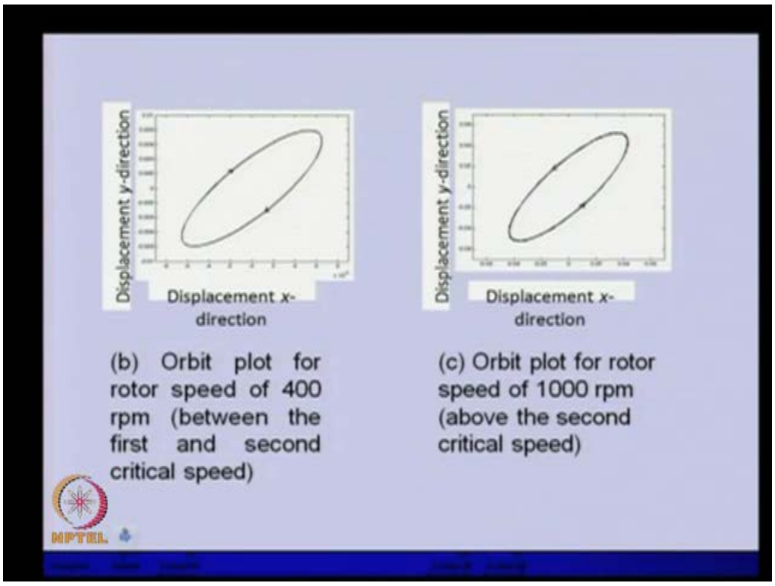

But if we are between two critical speeds, the sense of rotation is change. That is clockwise. If we cross the second critical speed, again it rotates in the counter clockwise direction. So, these also indicate how in which a reason of the critical speed range, we are there. Even the shape of the orbit helps us in understanding the especially what kind of fault may come into the rotating machinery. So, as we will see in the subsequent slides that the shape also gives indication of an enciphering fault in the rotating machinery.

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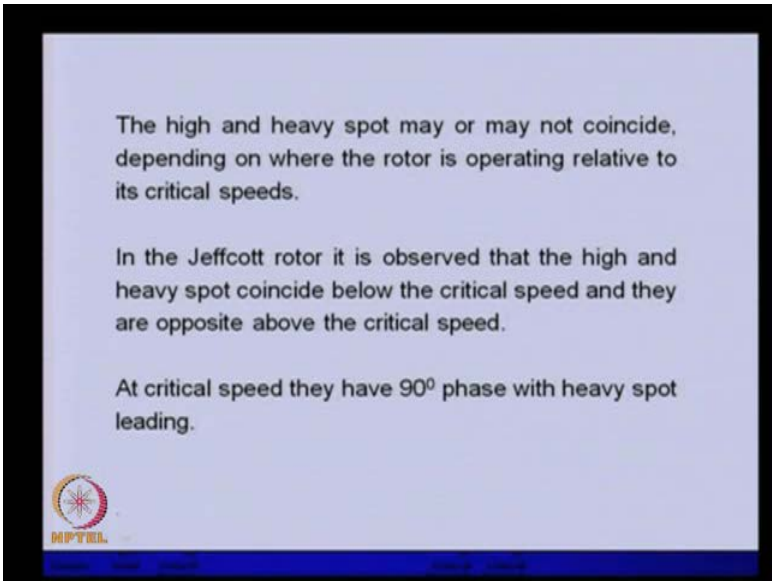

So, the high and heavy spots may or may not coincide, depending upon where the rotor is operating relative to the critical speed. So, in a Jeffcott rotor, the example which I showed in the previous slide, it is observed that the high and heavy spot coincide below the critical speed. They are opposite above the critical speed. At critical speed they have 90 degree phase with heavy spot. These kinds of a phase, we have already seen in very beginning of these lectures, when we considered the Jeffcott rotor.

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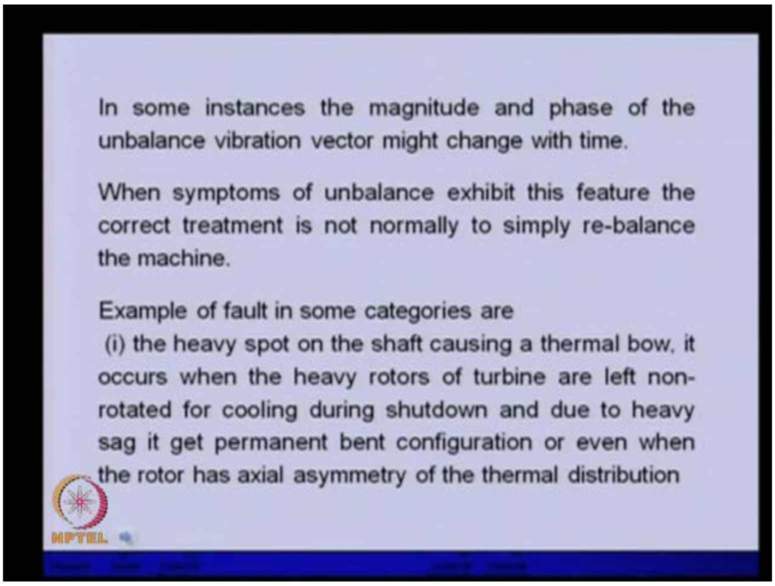

In some instances the magnitude and the phase of the unbalance vibration vector might change with time. When symptom of unbalance exhibit this feature the correct treatment is not normally to simply re-balance the machine. So, if magnitude and phase of the unbalance vibration is changing with time then that is not a balancing problem. That is a different problem. Example of fault in such category is the heavy spot on the shaft causing a thermal bow.

It occurs when the heavy rotors of turbine are left non-rotated for cooling during shutdown. Due to heavy sag it get permanent bend configuration, even when the rotor has axial asymmetry of the thermal distortion. So, because of this we may find that the response may change over period of time.

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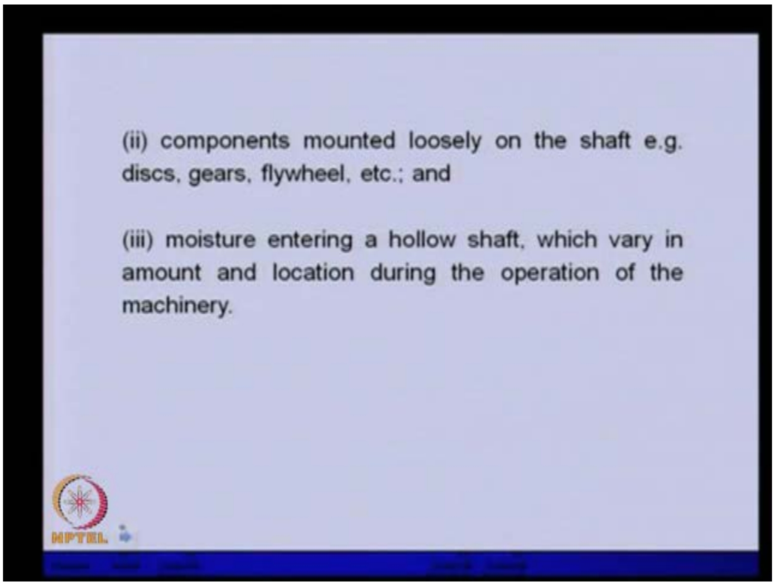

Components mounted loosely on shaft like discs gear flywheel etcetera. If that is the case then also we find this particular kind of a behavior in the response. Moisture enters in a hollow shaft, which vary in amount and location during the operation of the machinery. So, this is another potential cause of that we may find that the response may change over time.

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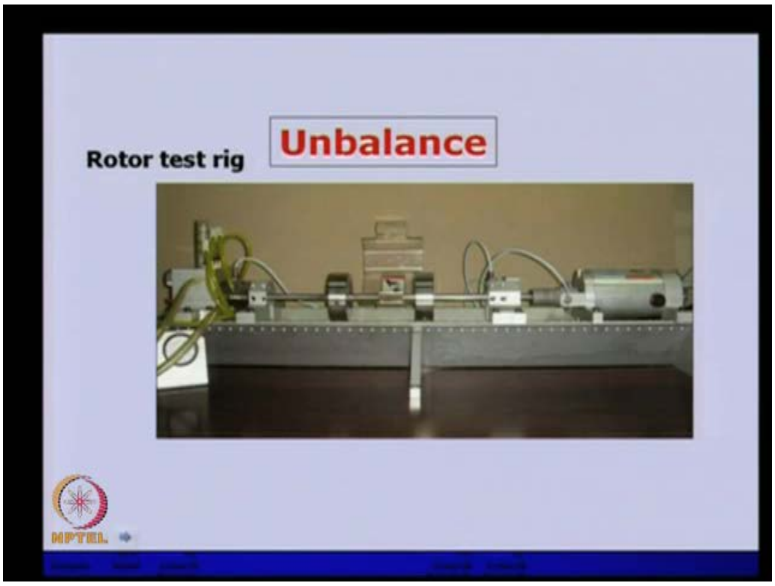

Now, coming to the unbalance, we already seen how the procedure how to balance a rotor. So, it is a typical rotor in which this is a motor the bearings are there here. It is a fluid film bearing and this is a bush bearing this is a two discs and this is mounted on flexible shaft. They represent to measure the vibrations at several locations. So, we can able to use proximately displacement force to measure the displacement.

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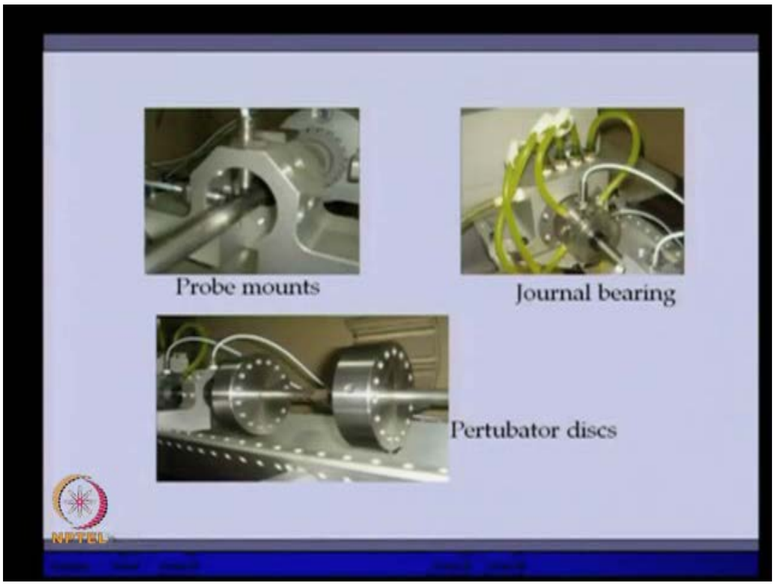

So, you can able to see. This is the shaft. These are the proximately probe. So, there is a gap between the shaft and the probe and when gap changes it produces the current proportional to that. We can able to measure the displacement. This is a fluid film bearing in which we can able to measure the pressure also at various location. This is a disc and you can in this particular disc. You can able to see there are threaded holes, in which we can able to put some kind of screw, so that we can put some kind of trial mass in this. So, basically we can call this as a Pertubator disc.

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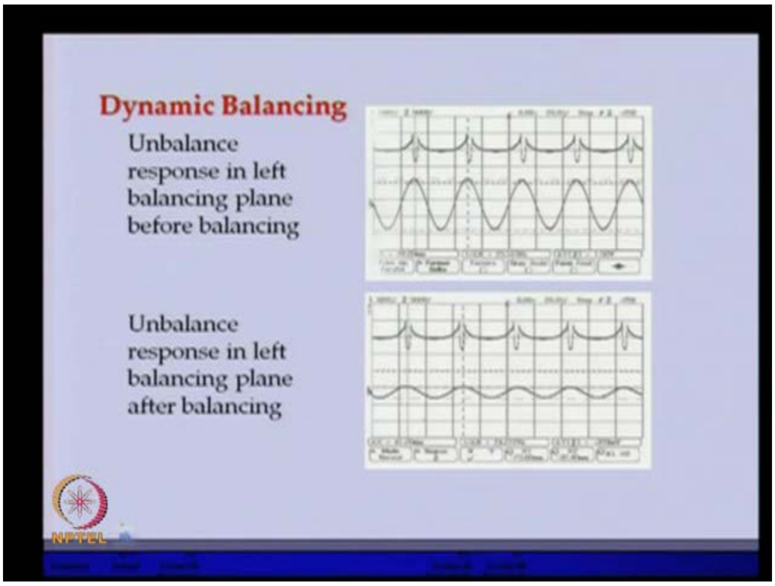

Now, this is a typical response. So, this is without unbalance. So, you can able to see this is a response of the rotor. This has been filtered at the rotational speed of the rotor. That is why it is so perfectly sinusoidal in nature. This is a reference signal which we, generally keep at a on to the shaft or by putting some kind of a tape photo reflecting tape. So, these spikes help us in finding the phase of this with respect to this reference angle.

So, it has been balanced by the procedure described earlier. Now, you can able to see after putting the correction mass again we measure the displacement. You can able to see. Compare this drastic decrease in the amplitude of the vibration took place after balancing.

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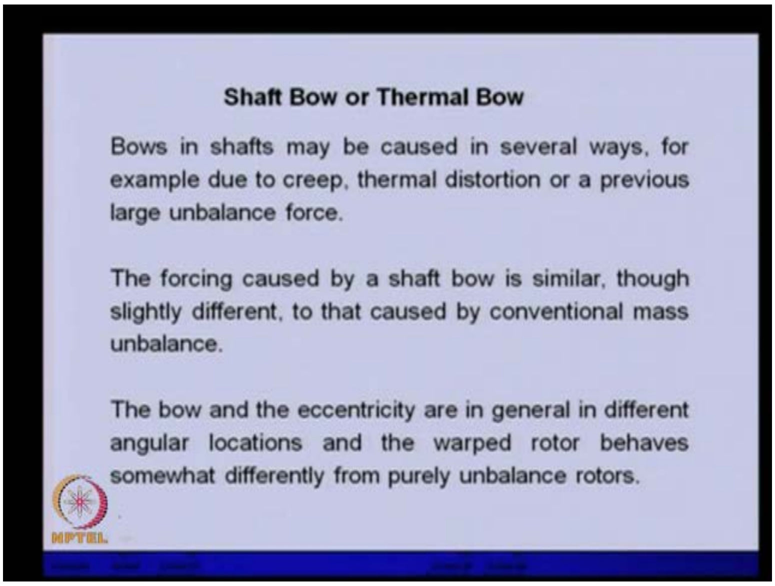

Next, category of the fault is the shaft bow or thermal bow. As we have seen in the previous discussion. This bows in shafts may caused in several ways. For example due to creep thermal distortion or a previously large unbalance force. The forcing caused by a shaft bow is similar slightly different to that caused by conventional mass unbalance. The bow and the eccentricity are in general in different angular location. Warped rotor behaves somewhat differently from purely unbalance rotor. So, unbalance and the bend rotor they have slightly different behavior.

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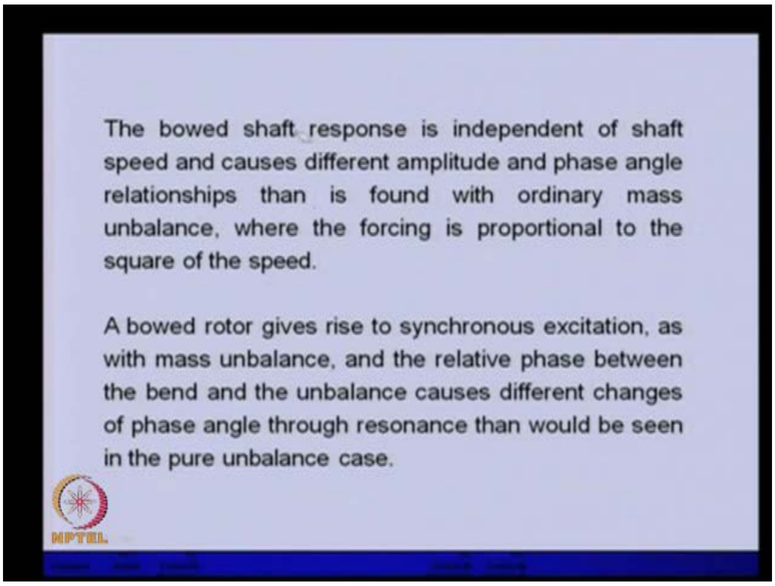

Bowed shaft response is independent of the shaft speed. This is very important to observe this particular thing. It is independent of the speed of the shaft and causes different amplitude and phase angle relationship. Then, it is found with the ordinary mass unbalance, where the forcing is proportional to the square of the speed. We know for unbalance the force is proportional to the square of the speed. But for the both shaft it is not the case. Both shaft gives rise to synchronous excitation as with the as the mass unbalance. The relative phase between the bent and the unbalance causes different changes of phase angle through resonance, and then would be seen in the pure unbalance case.

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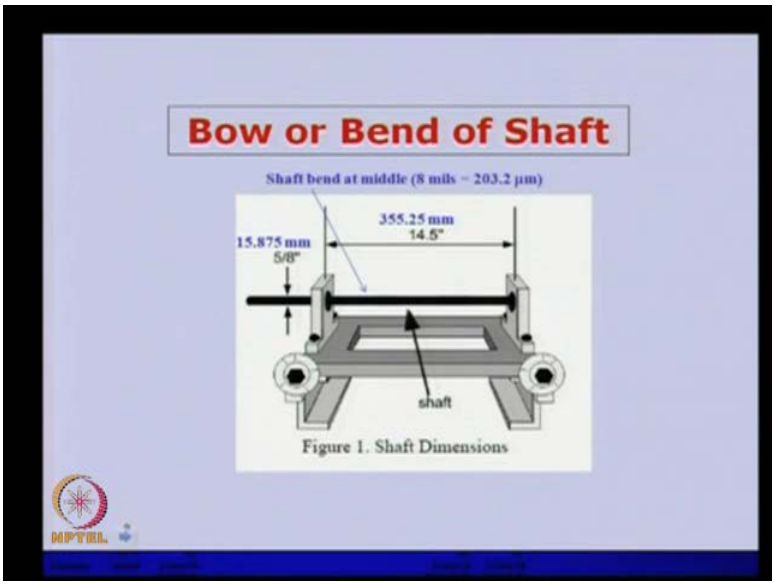

So, this is a typical very interesting in which the bearing blocks are there. There is a shaft and we can see. This we can able to introduce a bow in this particular shaft that means permanent bent we can able to introduce. We can able to rotate the shaft by some motor. Then, we can able to measure the vibrations at some location on to the shaft.

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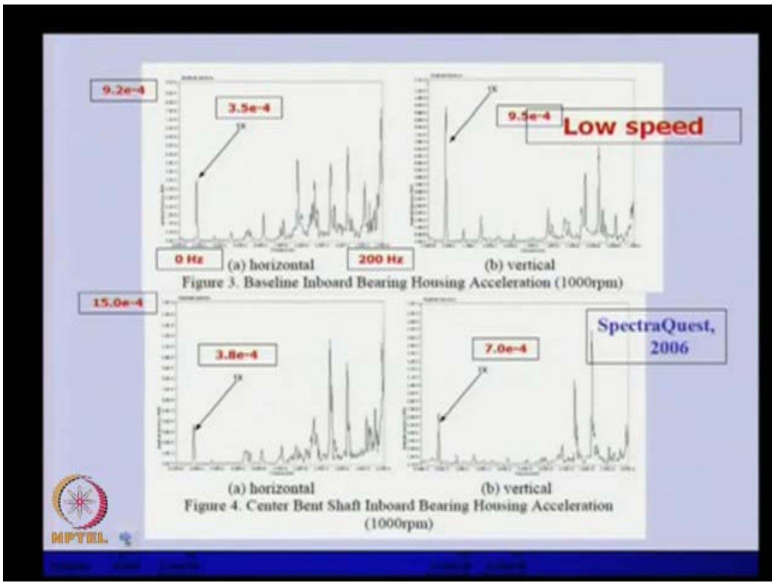

This is a particular this vibration we have been measured at the bearing block through accelerometer. Now, you can able to see in this basically it is a horizontal and vertical response. This is in the frequency domain. From 0 to 200 hertz is the frequency span in this. This is the amplitude of vibration. This is in horizontal. This is in the vertical direction; this is without any bow into the shaft at one particular speed 1000 RPM that has been rotated. Now, we can able to see this is corresponding to the 1 X. That is corresponding to this rotating speed of the shaft.

This is also corresponding to the rotating speed of the shaft. Speed is 1000 RPM, which is relatively low. This is the response, when we introduce the bend into the shaft. We are measuring the vibration from the bearing housing at the same speed we are operating as this one. In this we can able to see, this is still the spikes are there. But the amplitudes are different. You can able to see here it is 9.2, but it is 15.0. So, amplitudes are different. But not much appreciable amount of difference was found between these two responses for slow speed of vibration. So, this is discussed.

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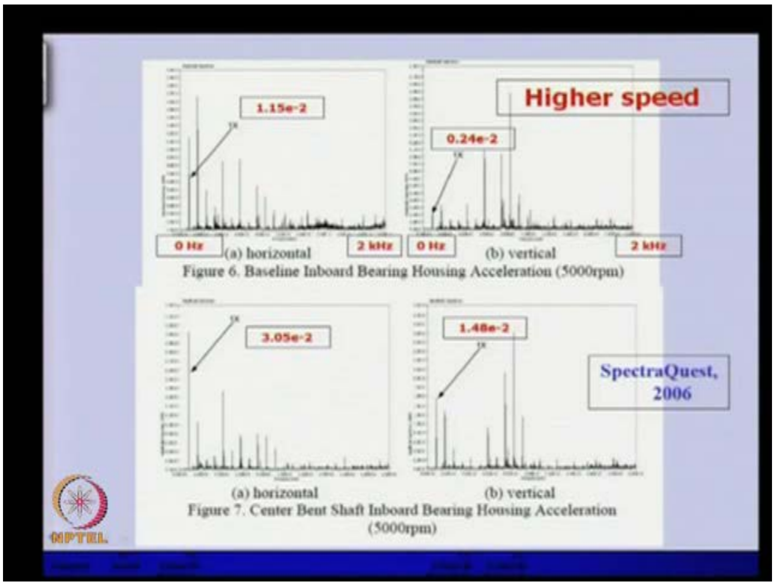

This is the same experiment with slightly higher speed at 5000 RPM. This has been taken from spectra quest case studies. In this you can able to see that this is the corresponding to the rotational speed of the shaft. This is in the vertical horizontal direction vertical direction. This is a baseline that means without bent. This is with bent. With bent we can able to see the there is a change in the amplitude of vibration. Now, quite drastic change in the amplitude of vibration of at the rotational speed the frequency we observed.

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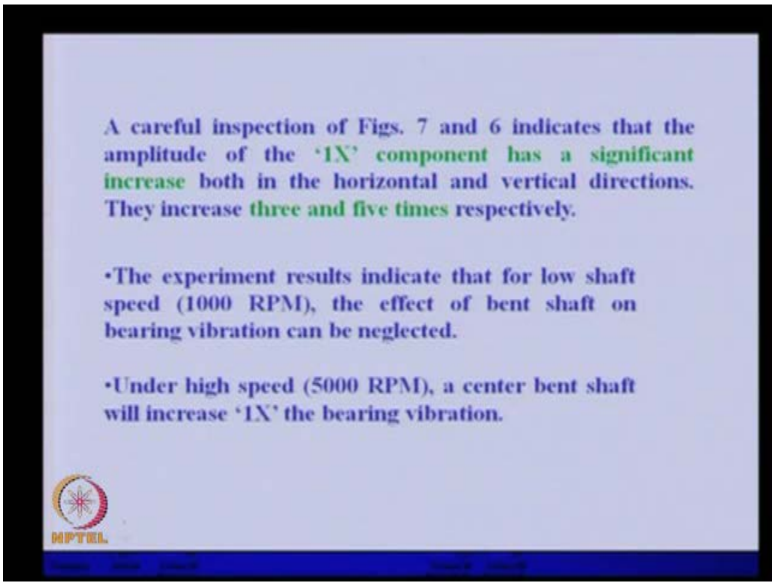

So, in this a careful inspection of the previous two figures, indicate that the amplitude 1 X that is a rotational speed has a significant increase both in the horizontal and vertical directions. They increase three to four times respectively. The experimental results indicate that for low speed the effect of bent shaft on bearing vibration can be neglected. Under high speed 5000 RPM a central bent a center bent shaft, the bent was there at the centre of the shaft, will increase 1 X at the bearing vibration. So, we are measuring the vibration of bearing. So, it has been observed in the previous figures that for high speed the bent shaft is having significant effect onto the 1 X component of the vibration.

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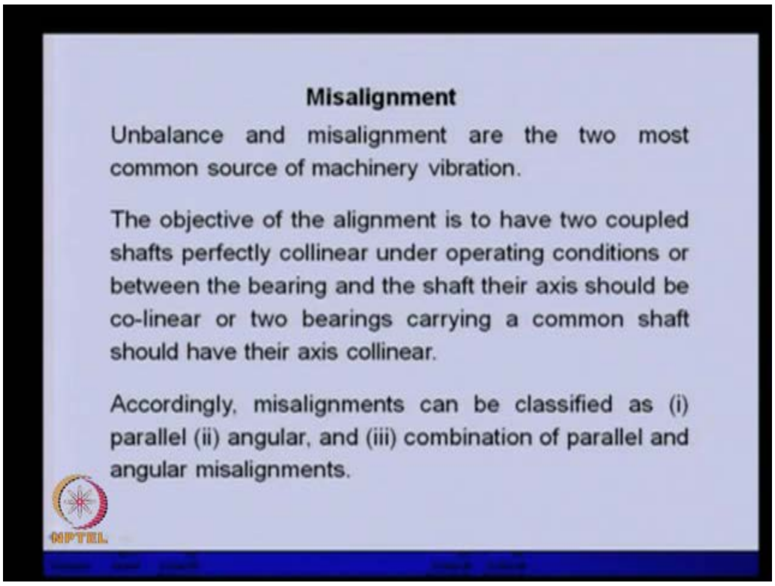

So, next fault is the misalignment. Unbalance and misalignment are the two most common source of vibration in the machinery. Objective of the alignment is to have two coupled shaft perfectly collinear under operating condition. Or between the bearing and the shaft their axis should be collinear. Or the two bearing carrying a common shaft should have their axis collinear. So, these are the condition this is the perfect alignment. But if it deviates, then we will be having misalignment in the system in the rotating components. So, accordingly misalignment can be classified as parallel angular or combination of parallel and angular misalignments.

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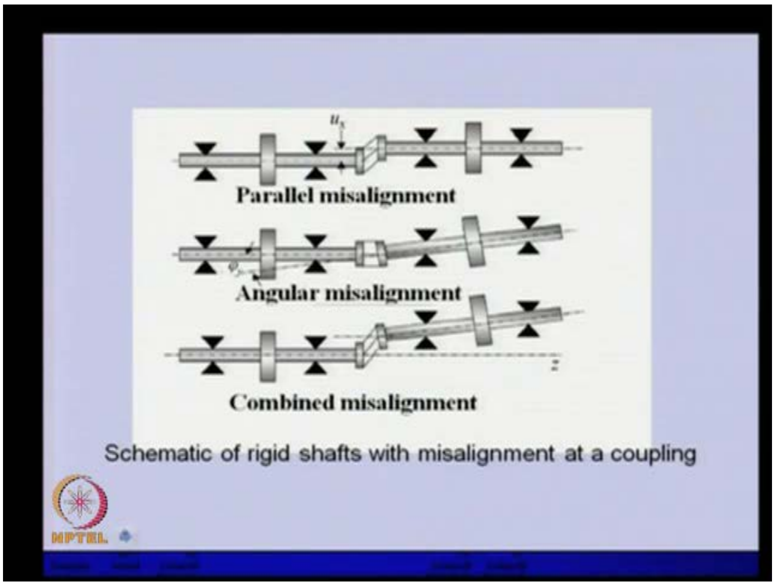

So, this we can able to see here there are two shafts rigid shafts and their offset by some amount. So, this is parallel this shaft axis are parallel, but they are offset. So, this is a parallel misalignment. Then, another is in which two shafts are having some inclination. So, this is an angular misalignment. But there is no parallel offset and the third is the combination of both parallel as well as angular. So, in this both the parallel and angular misalignments are present. So, we can able to expect that there will be large forces, which will be coming out in the bearing because of this misalignment. That will reflect the vibration signatures like the unbalance.

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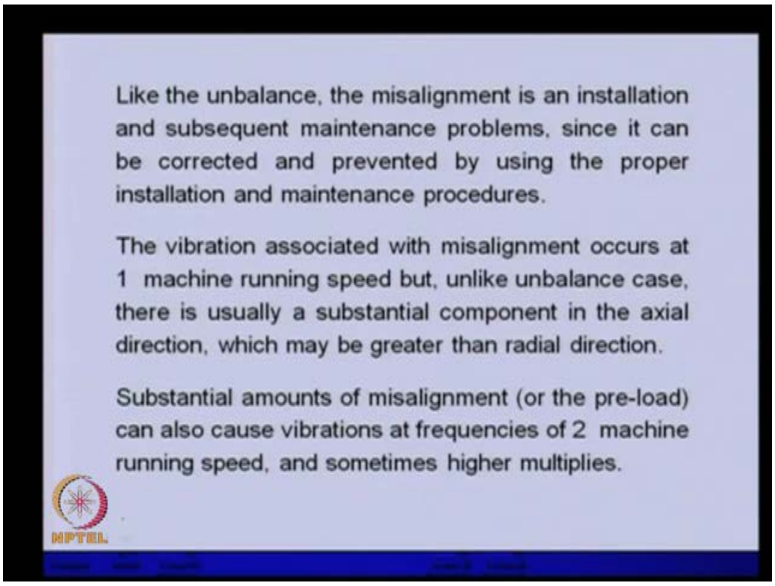

The misalignment is an installation and again I am repeating this like the unbalance the misalignment is an installation and subsequent maintenance problem. Since, it can be corrected and prevented by using the proper installation and maintenance procedures. The vibration associated with misalignment occurs at one machine running speed, but unlike unbalance case, there is a usually substantial component in the axial direction, which may be greater than the radial direction.

So, apart from the radial vibration we can have axial vibration in this particular case, when the misalignment is present, which is not there in the case of unbalance. In unbalance only pure radial vibrations will take place. Substantial amount of misalignment or the preload can also cause vibrations at frequencies of two machine running machine, twice of the machine running speed and sometimes higher multiples also three four also.

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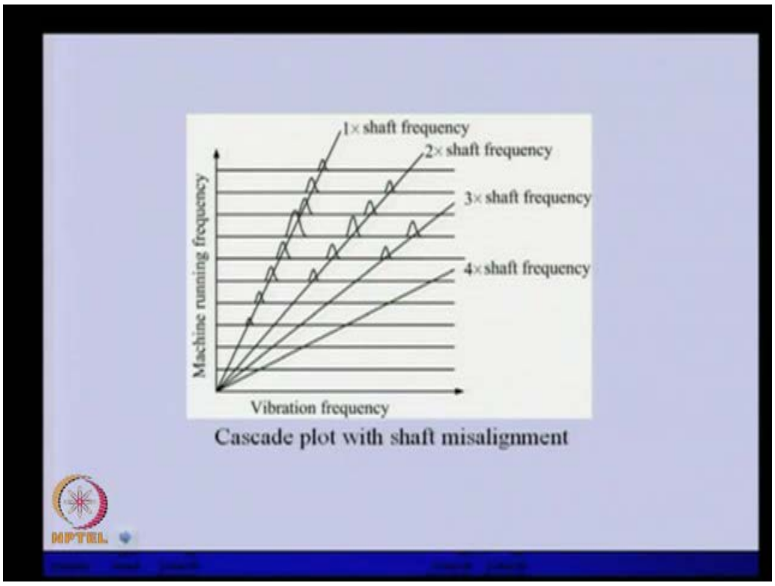

So, this is a typical cascade plot with shaft misalignment. So, this is the vibration frequency. This is the shaft running frequency. So, you can able to see, this is the 1 X. So, amplitudes are present in this even in 2 X machine. Some of the peaks are present as we are increasing the speed 3 X few are there. So, this kind of behavior you can able to get when the misalignment is present in the system.

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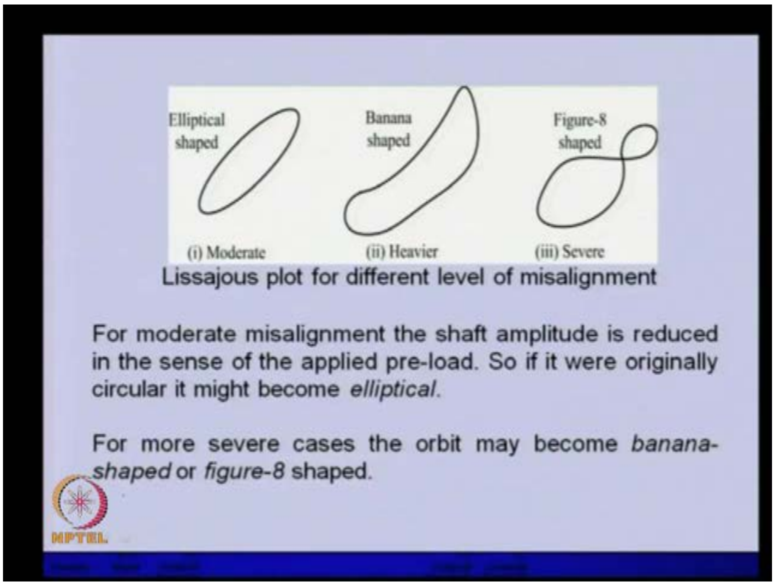

So, this is the Lissajous plot for different level of misalignment. For moderate misalignment, we will be having elliptical shape of the orbit or Lissajous plot, which is nothing but the x and y plot as we have shown earlier. But for heavier misalignment we have a banana shaped or for very severe misalignment. We will be having shape of eight or maybe multiple loops we may find because of severe misalignment. So, by looking into the plots, we can able to pin point that whether the rotor system is having some kind of misalignment converted the degree of misalignment we can able to identify with this.

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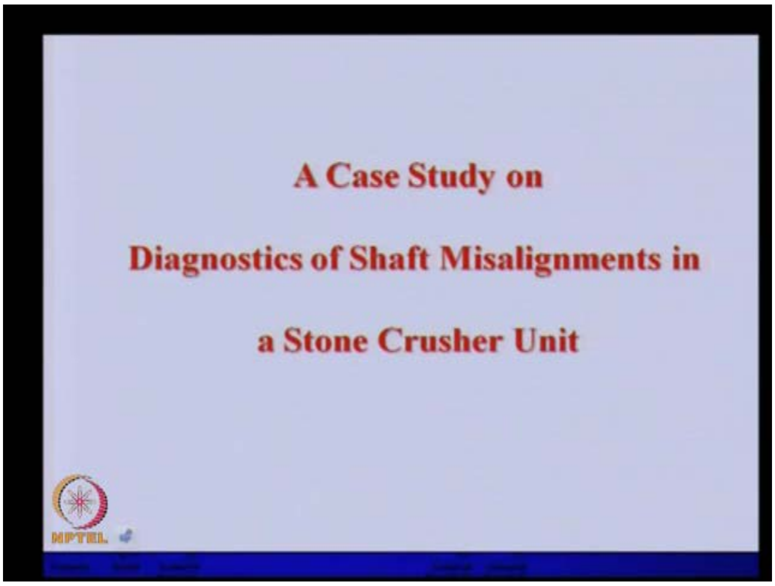

Now, we will take up very one case study, which is of a stone crusher unit in a cement plant.

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So, in this is a typical industrial stone crusher. Basically, here motor is there. Drive is there. Then, this is the coupling unit, which we can able to see it. We have then this side. There was one crusher unit, in which the stones used to crush from bigger to smaller size. In this particular case they had lot of problem of the vibration. Basically this whole unit was mounted on a plate form. If we…So, basically it was in the first floor kind of thing.

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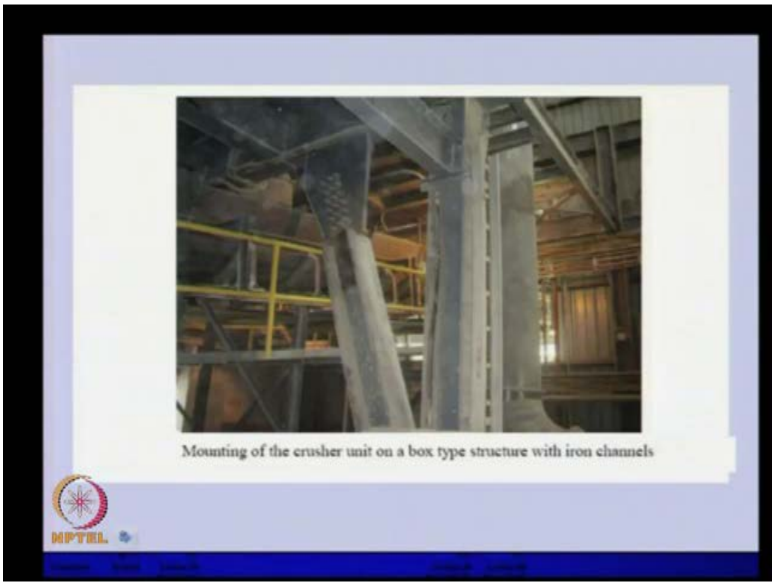

So, if we see from bottom. So, this was the support to mount that machinery at the top. They had lot of vibration in this of order of vibration was very large because of that even in the columns. So, whatever this kind of columns, they had the cracks in the columns. The machine was in very bad condition. They did not know what the actual cause of such vibration was.

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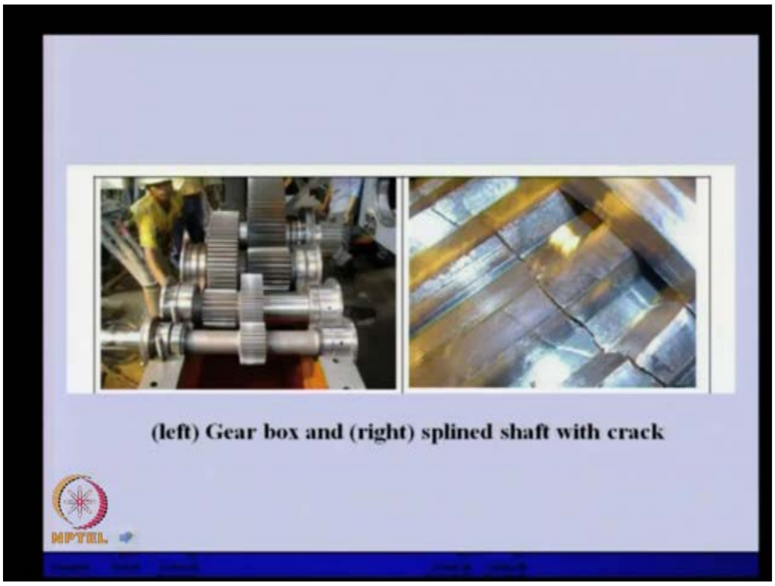

So, we took measurement of that particular vibration and then after one week of that there was total shutdown of the plant. This is the typical gear box, they took out and try to find out what is where is the failure of the machine took place. So, basically at the rear end of the gear box, this is a splinted shaft, which connects this particular gear box to the stored rollers. So, this splinted shaft is shown here in more close view of that. So, that splinted shaft has had crack. This is the crack in which is very severe crack is present in this. This was after the measurement we took. So, that was after one week this happened. This is a typical vibration measurement which we took to the failure.

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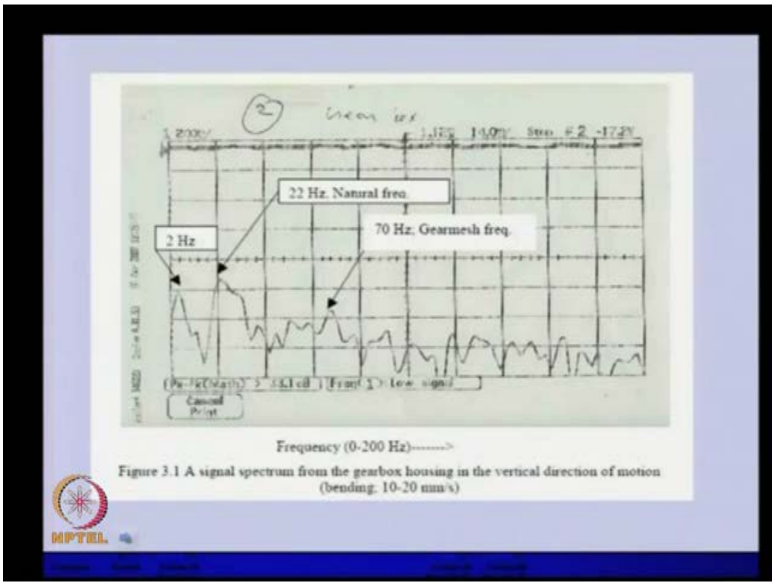

This we could able to find out some of the frequencies, which were natural frequency of the system because in several plots they were appearing and disappearing. So, whenever there was excitation close to this they were appearing and otherwise it was not appearing. The gearless frequency one of them was here. This was a typical low frequency peak which was otherwise should not have present.

We found this was because of the misalignment between this splinted shafts from the roller shaft. That misalignment when we calculated because the speed of that was very low. This is the typical misalignment behavior, we you could able to identify from this vibration signature. So, this is a typical case study on the misalignment, how we can able to find out through vibration signature directly.

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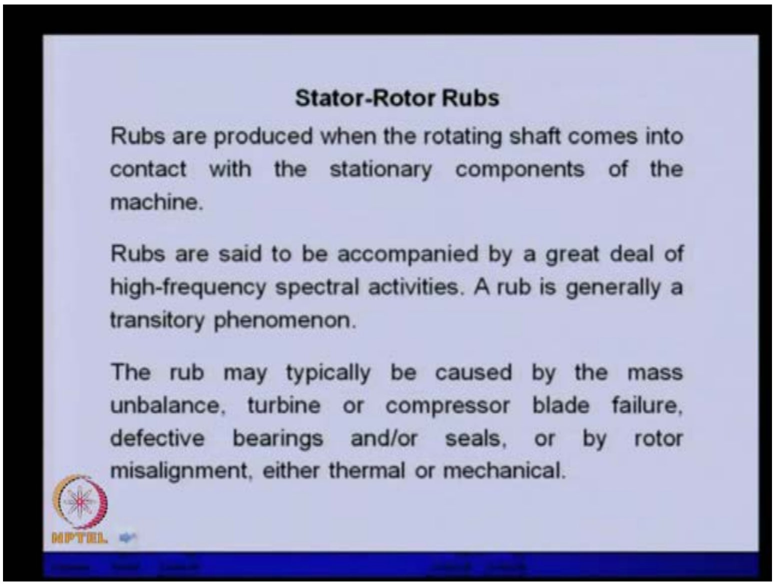

Next kind of fault is the stator rotor rubs. So, when the rotor touches the stator there will be rub. This may reflect the vibration signal. So, rubs are produced when the rotating shafts comes into contact with the stationary component of the machine. Rubs are said to accompany by a great deal of high frequency spectral activity. A rub is generally transitory behavior transient behavior is there. The rub may typically be caused by the mass unbalance, heavy mass unbalance, turbine or compression blade failure, defective bearing seal, or the rotor misalignment, either thermal or mechanical.

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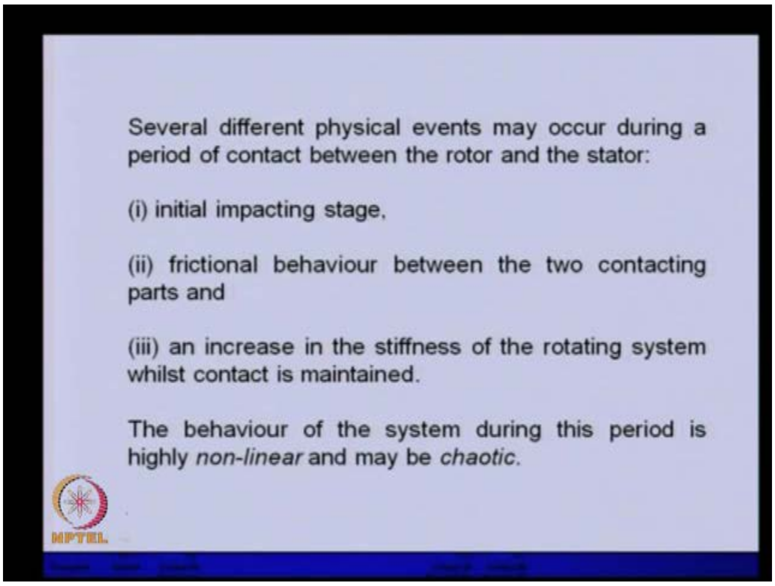

Several different physical events may occur during a period of a contact between the rotor and the stator. The initial impacting stage, frictional behavior between the two contacting parts, an increase in the in the stiffness of the rotating system while contact is maintained. The behavior of the system during this period is highly non-linear and may be chaotic also in the nature. It is one of the research have been reported that this may be chaotic behavior of the system.

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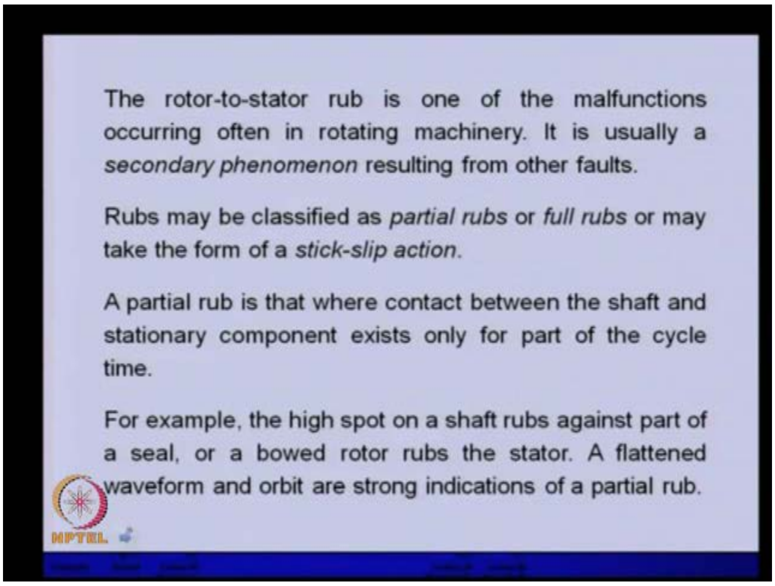

The rotor to stator rub is one of the malfunctions occurring often in rotating machinery. it is usually a secondary phenomenon resulting from other faults. So, it is because of the other fault this comes, that is why it is a secondary phenomenon. Rubs may be classified as a partial rub full rub or take a form of a stick slip action. A partial rub is that where contact between the shaft and the stationary component exists only for a part of the cycle. For example, the high spot on the shaft rubs against part of the seal or a bowed rotor rubs the stator. The flattened waveform and orbit are strong indication of the partial rub. So, from there waveform we can able to identify, whether rub is taking place if it is getting flattened.

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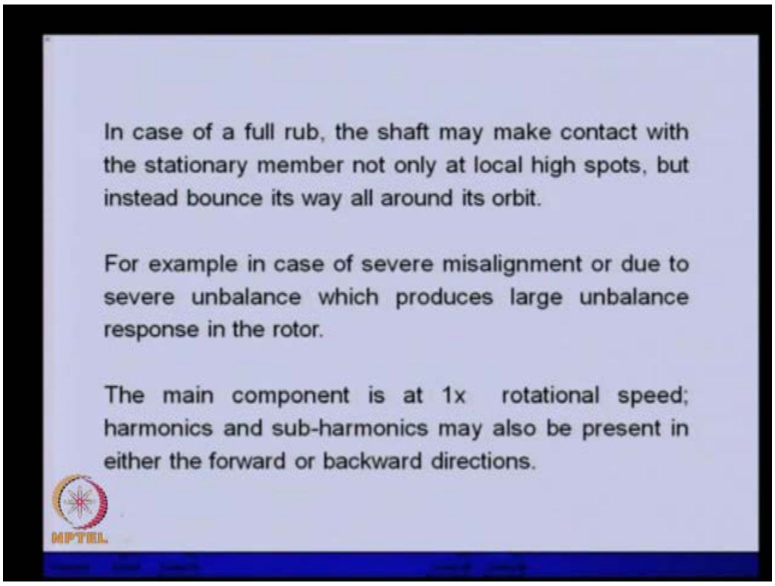

In the case of full rub the shaft may take may make contact with stationary member not only at local high spot, but instead bounce its way all around its orbit. For example in case of severe misalignment or due to severe unbalance which produces large unbalance response in the rotor. The main component is at 1 X rotational speed; harmonics and sub-harmonics may also be present in either the forward or backward directions because of the rub full rub.

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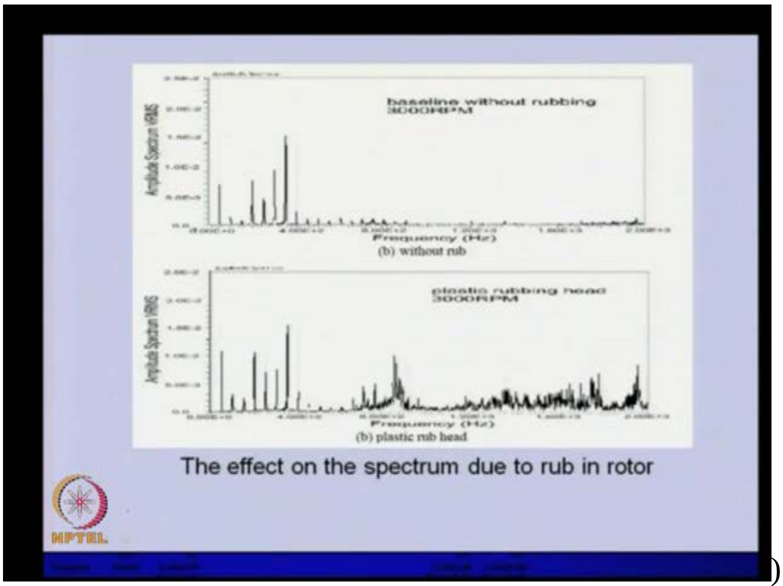

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This is a typical plot of the vibration which takes place due to the rub. So, this is a without rub and with rub you can able to see various high higher order components have appeared. So, this could be the indication of the rub in the rotor system. Mechanical looseness of component, this is another kind of fault, which is which comes into the system because of the over maintenance. So, mechanical looseness…

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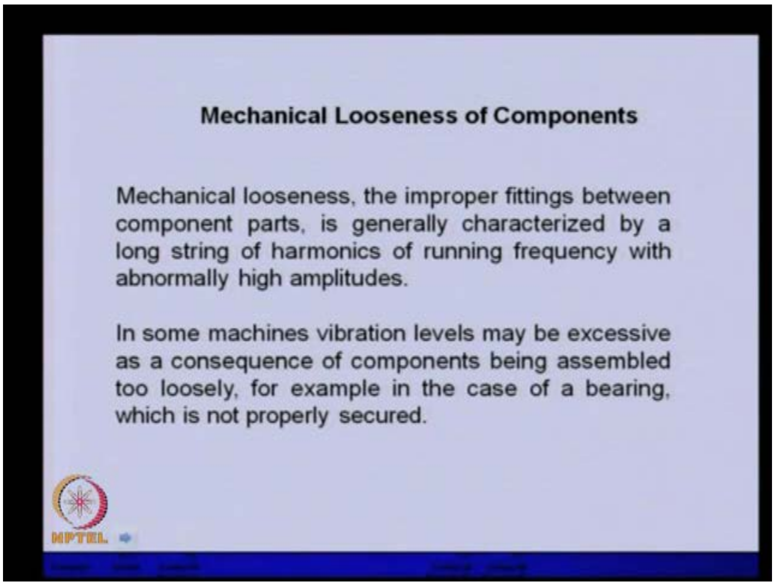

The improper fitting between component parts is, generally characterized by a long string of harmonics of running frequency with abnormally high amplitudes. In some machines vibration levels may be excessive as a consequence of component being assembled too loosely. For example, in the case of bearing, this is not properly secured.

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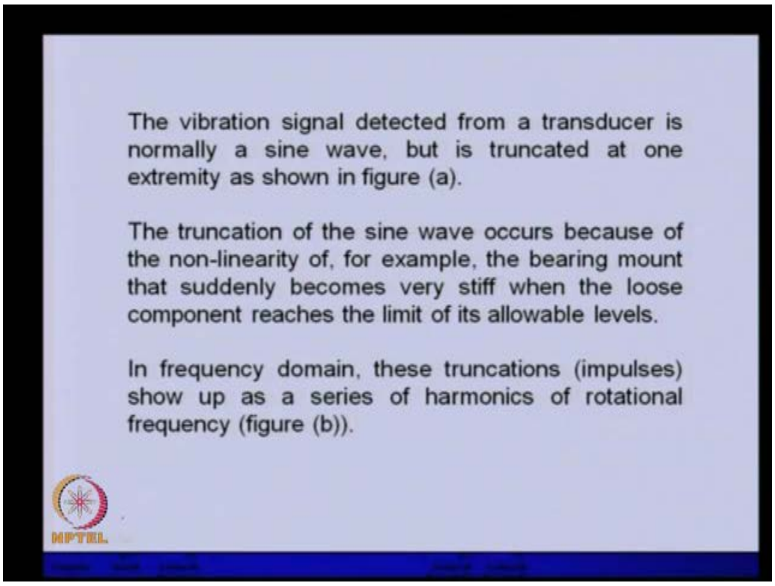

The vibration signal detected from a transducer is normally a sine wave, but is truncated at one extremity as shown in the figure.

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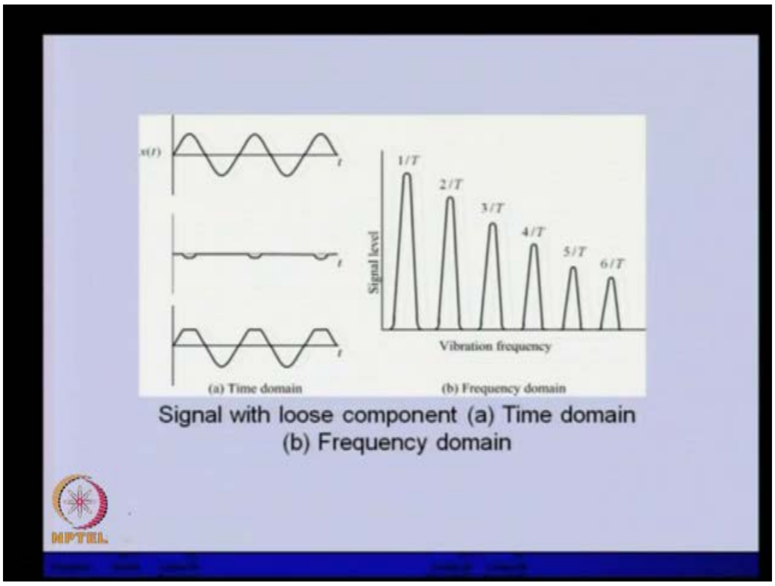

So, that means vibration signal should be sinusoidal. But sometimes we will see that at one extent it is getting flattened because of this looseness.

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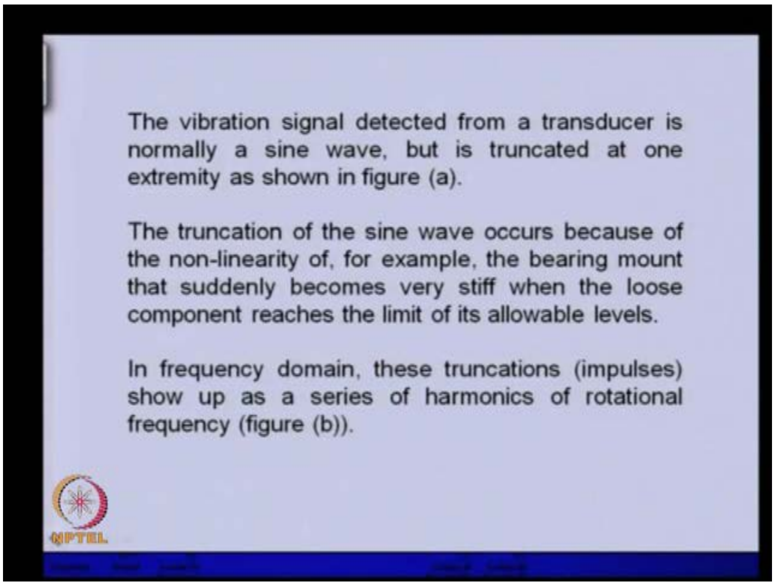

So, a truncation of sine wave occurs because of non-linearity of, for example, the bearing mount that suddenly becomes very stiff when the loose component reaches the limit of its allowable level. The frequency domain in frequency domain, these truncation impulses show up as a series of harmonics of rotational frequency. So, if you see this particular signal in frequency domain.

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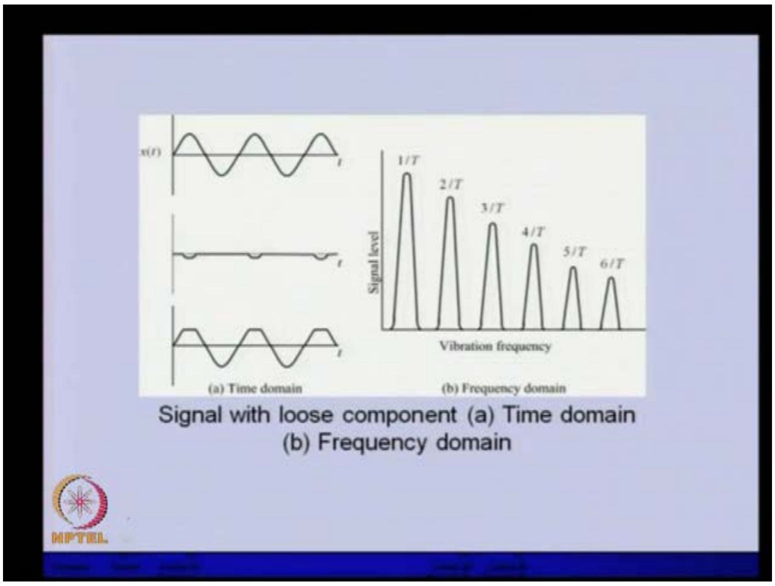

There will be a fundamental harmonics and then higher harmonics will be there for this. In fact without looseness we expect only single peak, but because of the truncation, now we will be having multiple harmonics of the signal will be present.

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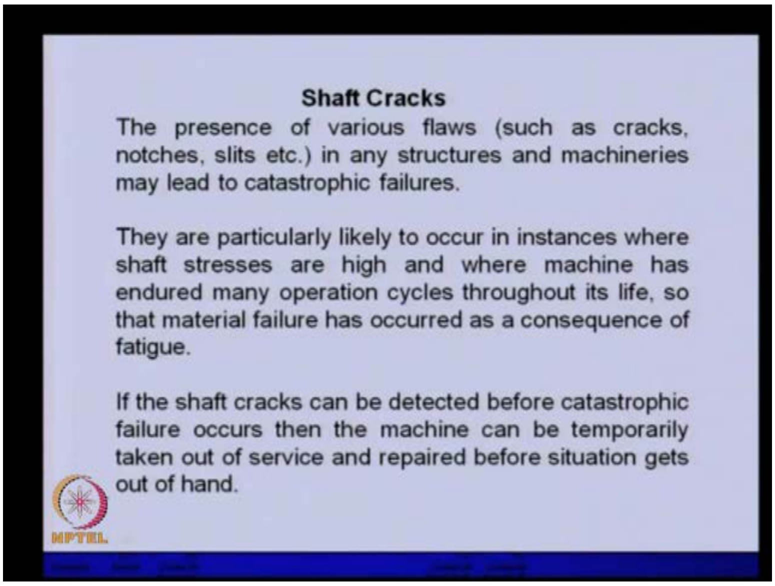

Next common fault in the shaft especially where the high power transmission is there. The there is a shaft cracks the presence of various flaw such as cracks notches slits etcetera. In any structure and machinery may lead to catastrophic failures. They are particularly likely to occur in instances, where shaft stresses are high and where machines have endured many operation cycles throughout its life.

So, that material failure has occurred as a consequence of fatigue. In the shaft cracks again I am repeating, if the shaft cracks can be detected before catastrophic failure occur, then the machine can be temporarily taken out of service and repaired before situation gets out of hand.

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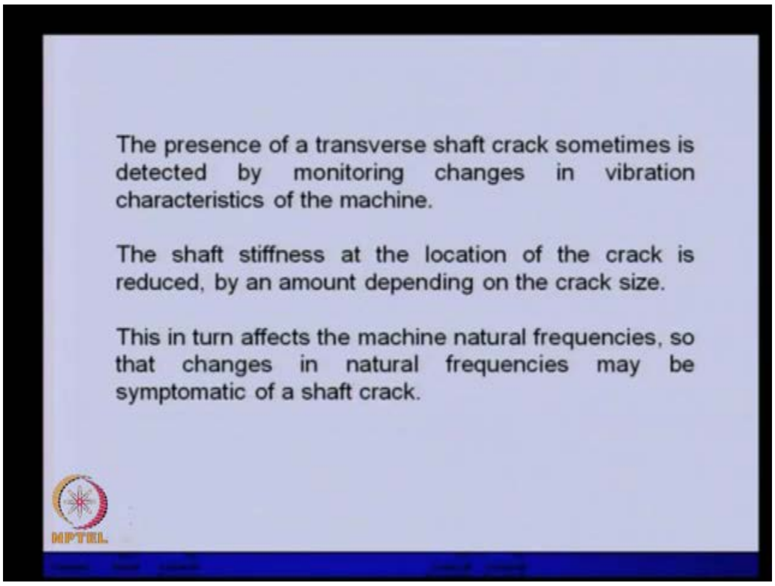

The presence of transverse shaft crack is sometimes detected by monitoring change in the vibration characteristics of the machine. The shaft stiffness at the location of the crack is reduced by an amount depending up on the crack size. That is in turn affecting the machine natural frequency, so that changes in the natural frequency may be that could be a symptom of the shaft crack. Crack introduces flexibility in the shaft that changes the stiffness. Natural frequency is nothing but root k by m. So, if the stiffness changes we expect there will be change in the natural frequency of the system. So, that could be one indication of the crack in the rotor system.

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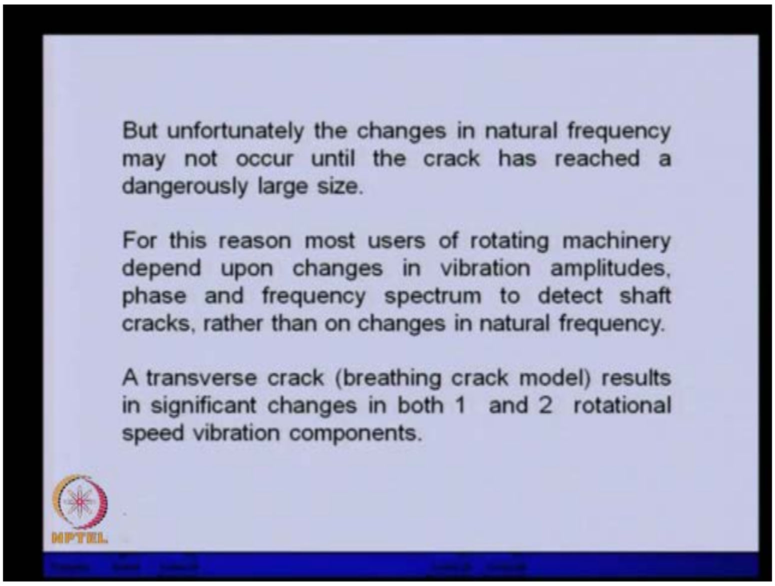

But unfortunately the changes in natural frequency may not occur until the crack has reached a dangerously large size. For this reason most users for rotating machinery depend on changes in vibration amplitude phase. Frequency spectrum to detect shaft cracks rather than the change in natural frequency. A transverse crack or the breaking crack sometimes we call it in which continuously opening and closing of the crack take place during vibration that results in significant change in both 1 X and 2 X vibration component. So, not only 1 X and 2 X component all are also present in this.

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So, as I shown earlier this is a typical crack shaft. But we measure the vibration at the housing of these, so that we could not able to locate as particular behavior in the vibration. But if you could have taken near to this shaft, we could have got some indication of the cracking in the shaft. Till now we have seen some of the very common fault in the rotating machinery. There are various components in the rotating machinery which do get failures. They have special frequency components like bearings and gears. Now, we will take up a case of bearing faults. Because in bearing we know there are various components and any of the components can fail.

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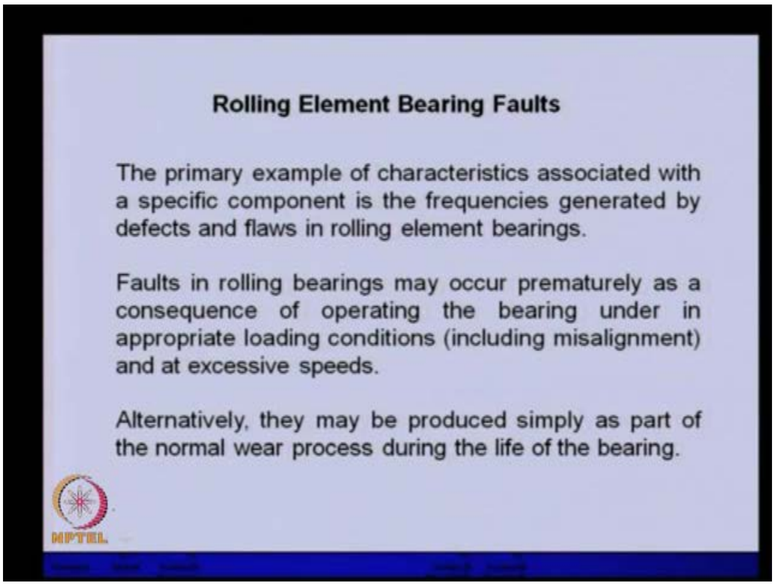

So, for rolling element bearing fault the primary example of characteristic associated with a specific component is the frequency, is generated by the defects and flaw in the rolling element bearings. Faults in rolling element bearing may occur prematurely as a consequence of operating the bearing, under appropriate loading condition in appropriate loading condition .including misalignment and excessive speeds. Alternatively, they may be produced simply as a part of normal wear process during the life cycle of the bearing.

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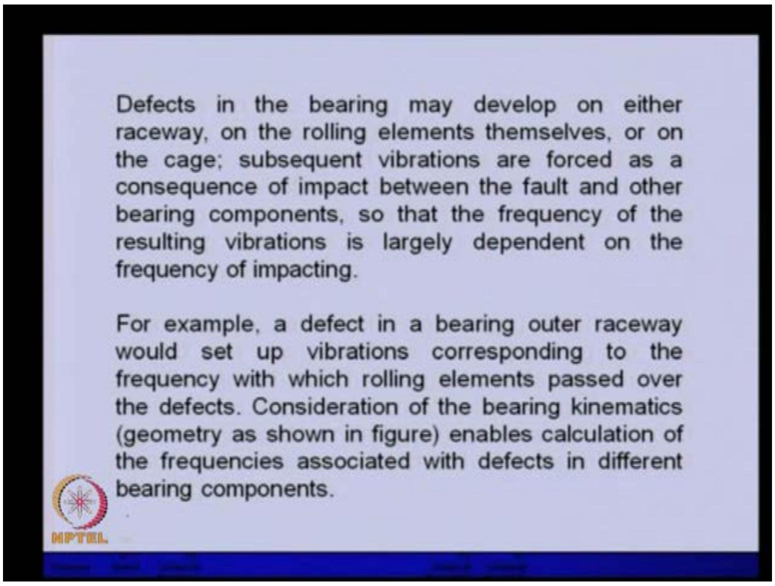

Defects in the bearing maybe developed on either raceway on the rolling element or themselves or on the cage. Subsequent vibrations are forced as a consequence of impact between the fault and the bearing components. So, that the frequency of the resulting vibration is largely depending on the frequency of impacting. For example, a defect in a bearing outer raceway would set up vibrations corresponding to the frequency with which rolling element passed over the defects. Consideration of bearing kinematics geometry enables calculation of these frequencies associated with defects in different component of the bearing.

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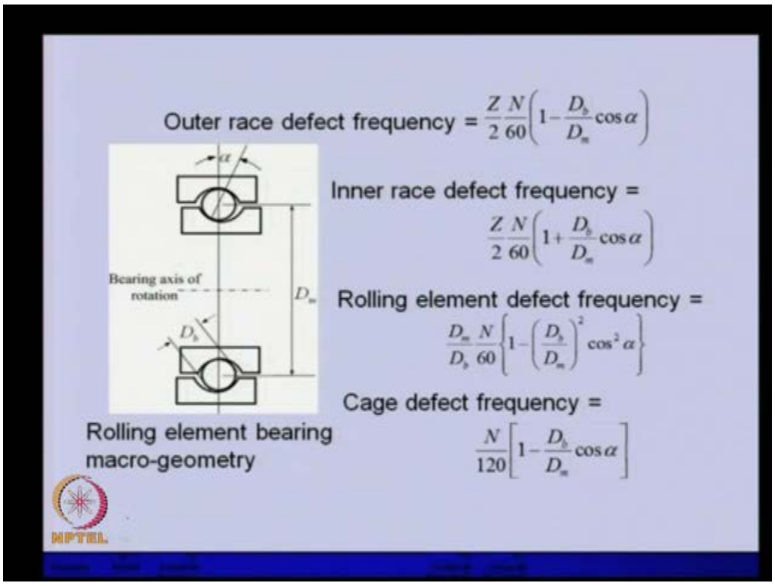

So, if we see the bearing this is a deep grove ball bearing. So, you can able to see this is a rolling element outer raceway inner raceway. We can have fault in any of these two three four components. This is the contact angle. This is the mean diameter of the bearing. This is a ball diameter. So, based on geometry we can able to calculate outer raceway defect frequencies like this, where Z is the number of rolling element, N is the speed of the shaft. These I have already defined. Similarly, inner race defect frequency can be given by this rolling element. Defect frequency cage defect frequency. So, these frequencies we can look into the vibration signal, if we are monitoring a bearing.

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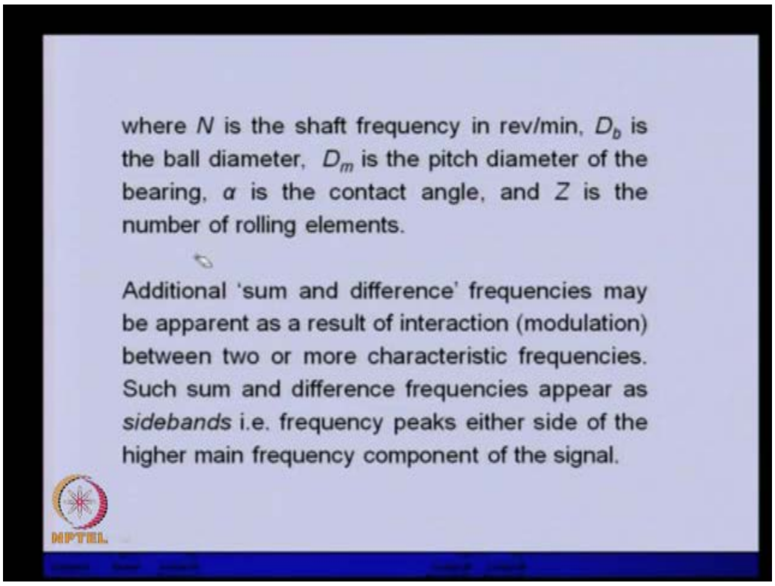

So, even addition additionally sum and difference frequencies may be apparent as a result of interaction or modulation between two or more characteristic frequencies. So, even with the rotor speed and the other defect frequency there can be sum and difference or the frequency may be present in the spectra because of the modulation. Such sum and difference frequency appear as a sidebands frequency peaks either side of the higher main frequency component of the signal. So, sidebands are nothing but frequency then be predominant and other side of that there will be two smaller peaks. So, that is called sidebands.

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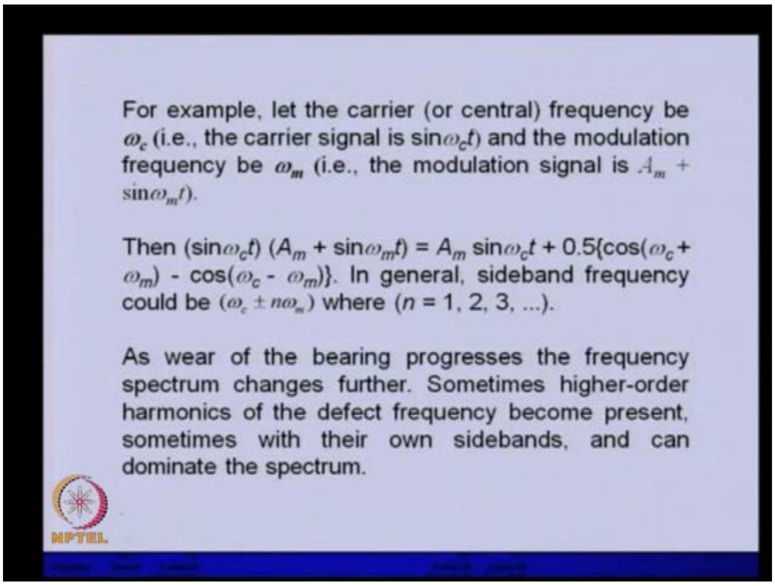

For example let the carrier or the central frequency is omega C. The carrier signal is sin omega t. The modulation frequency would be let us say, omega m n. Omega m and the modulation signal is some constant plus sin omega m t. Then, multiplication of this can be written like this and the sin omega C t and sin omega m t can be again rewritten like. So, you can able to see that basically we are getting a sum and the difference of the frequencies.

So, in general sideband frequency could be central frequency plus minus of the n times the modulated frequency. Some of the parts, we will be showing in the subsequent slides, a wear of the bearing progresses. As the wear of the bearing progresses, the frequencies spectrum changes further sometimes higher order harmonics of the defect frequency becomes present. Sometimes with their own sidebands and can dominate the spectrum. So, even the high frequency component can be there in the bearing fault.

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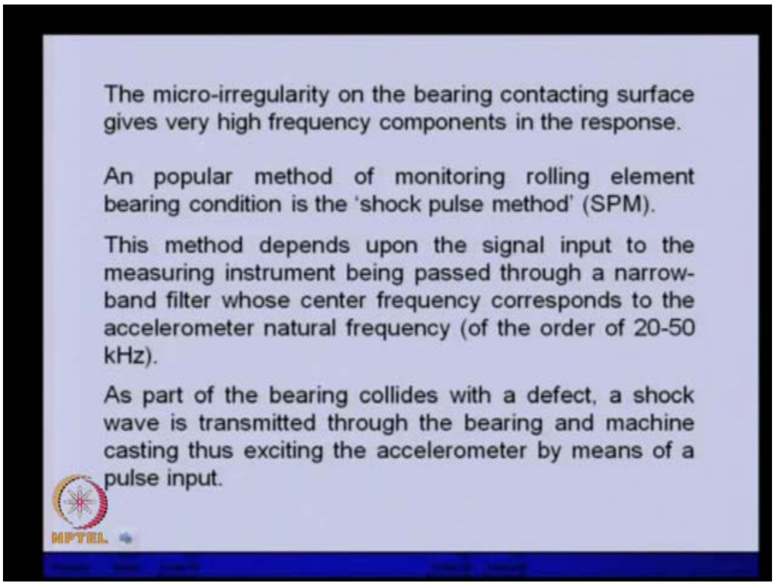

The micro irregularity of the bearing contacting surface gives very high frequency component in the response. A popular method of monitoring rolling element bearing condition is the shock impulse method. The basic principle of this is the method depends up on the signal input to the measuring instrument like accelerometer being passed through a narrow band filter, whose central frequencies is corresponding to the accelerometer natural frequency.

So, basically that is the part of the bearing collides with the defect a shock wave is transmitted through the bearing and the machine casting. Thus, exciting the accelerometer by means of a pulse input and because the frequency is close to the accelerometer natural frequency. So, we will get very high amplitude of vibration.

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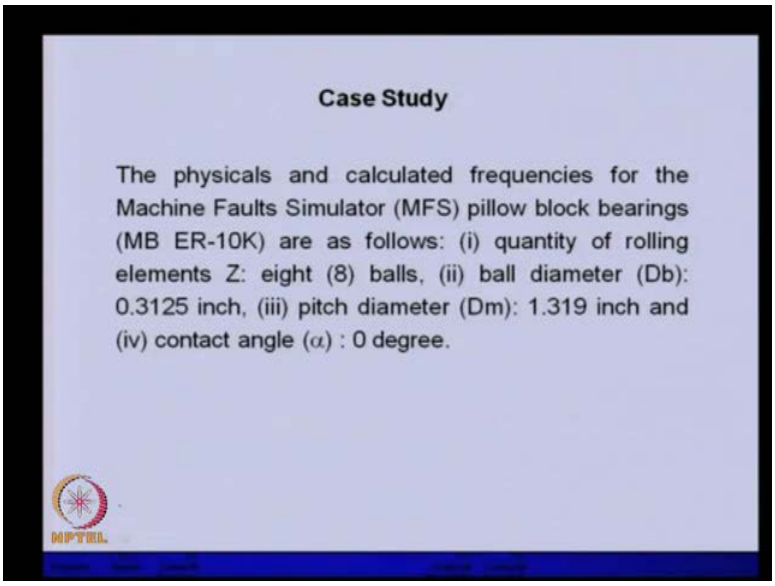

So, this is a one case study for the bearing. So, the physics and the…So, based on the geometry and the calculated frequency for the one of the machine fault simulator, we have taken. In which the bearing was of this category and various parameters like number of rolling element, ball diameter, pitch diameter, contact angles are given here.

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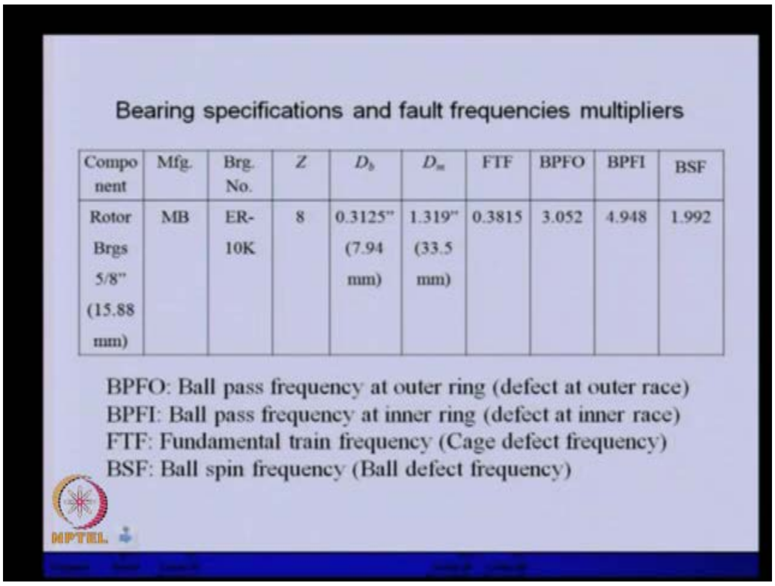

Based on this specification… So, these are the specification of the bearing. These are various frequency factors, which we have obtained from the previous formula. So, only thing is speed is not there in this, whatever the formula we have given.

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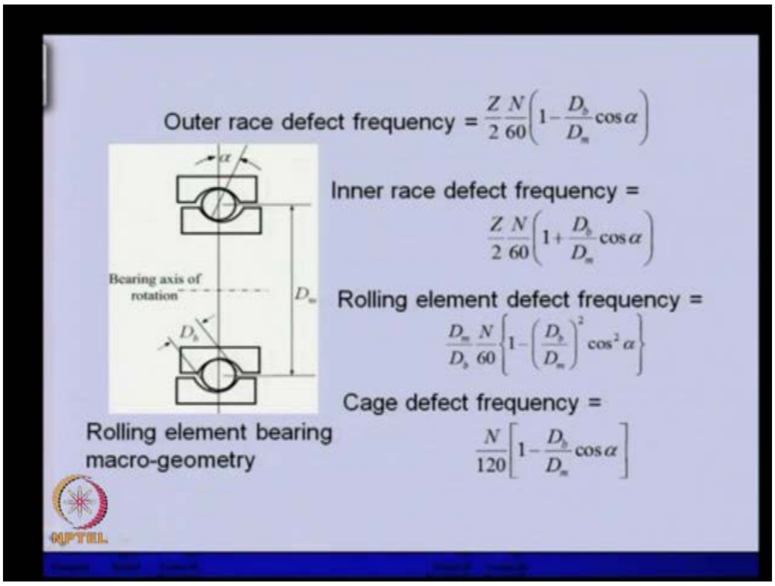

So, if you remove the speed parts N by 60.These are the factor are there.

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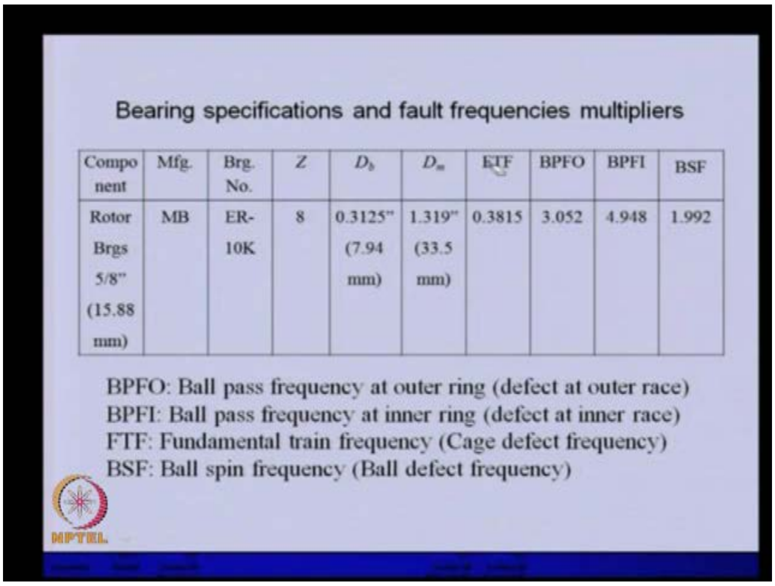
Bearing specifications and fault frequencies multipliers

Component	Mfg.	Brg. No.	Z	D_b	D_m	FTF	BPFO	BPFI	BSF
Rotor Brgs 5/8" (15.88 mm)	MB	ER-10K	8	0.3125" (7.94 mm)	1.319" (33.5 mm)	0.3815	3.052	4.948	1.992

BPFO: Ball pass frequency at outer ring (defect at outer race)
BPFI: Ball pass frequency at inner ring (defect at inner race)
FTF: Fundamental train frequency (Cage defect frequency)
BSF: Ball spin frequency (Ball defect frequency)

So, you can able to see, FTF is nothing but the cage defect frequency. BSF is ball defect frequency. Similarly, this is ball passing frequency for outer raceway and this is for inner raceway. So, this is for outer raceway defect. This is for inner raceway defect. So, these are the factor.

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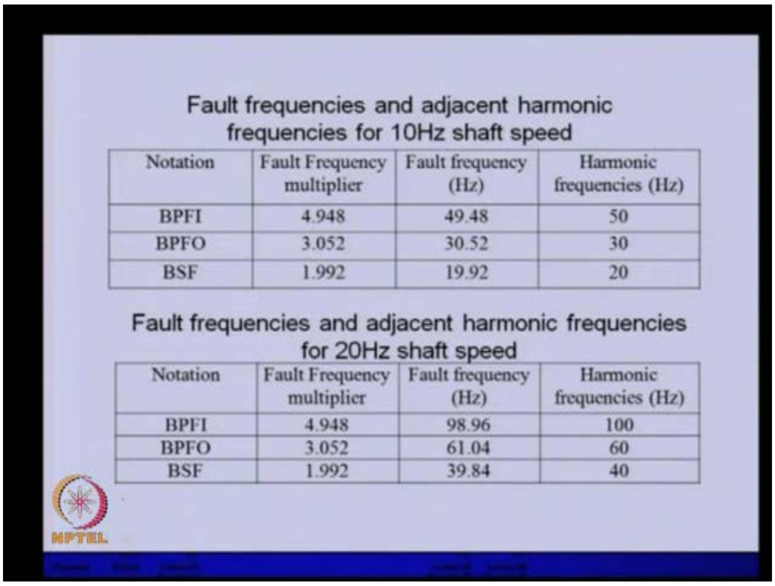
Fault frequencies and adjacent harmonic frequencies for 10Hz shaft speed

Notation	Fault Frequency multiplier	Fault frequency (Hz)	Harmonic frequencies (Hz)
BPFI	4.948	49.48	50
BPFO	3.052	30.52	30
BSF	1.992	19.92	20

Fault frequencies and adjacent harmonic frequencies for 20Hz shaft speed

Notation	Fault Frequency multiplier	Fault frequency (Hz)	Harmonic frequencies (Hz)
BPFI	4.948	98.96	100
BPFO	3.052	61.04	60
BSF	1.992	39.84	40

So, if you multiply with the speed of the rotor you will get what frequency will be coming because of these faults. So, at 10 hertz, if we are operating the rotor, we can able to see different kind of fault. These are the factor. So, we will be having the frequencies like this. Corresponding to the various faults inner raceway outer raceway like that. Similarly, for 20 hertz we can able to calculate. The main purpose of this is to find out where we should look into the signal, these frequencies.

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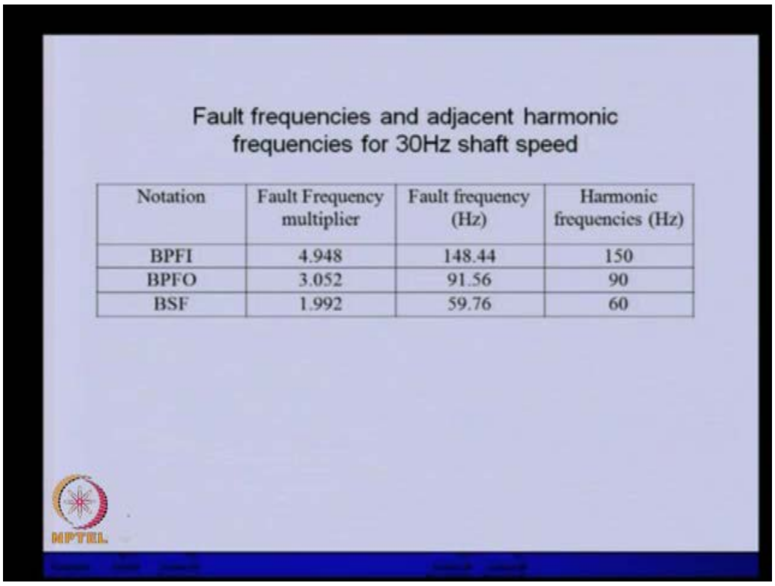

Notation	Fault Frequency multiplier	Fault frequency (Hz)	Harmonic frequencies (Hz)
BPFI	4.948	148.44	150
BPFO	3.052	91.56	90
BSF	1.992	59.76	60

So, these are the various frequencies. These are the closest value corresponding to the calculated frequency of the faults.

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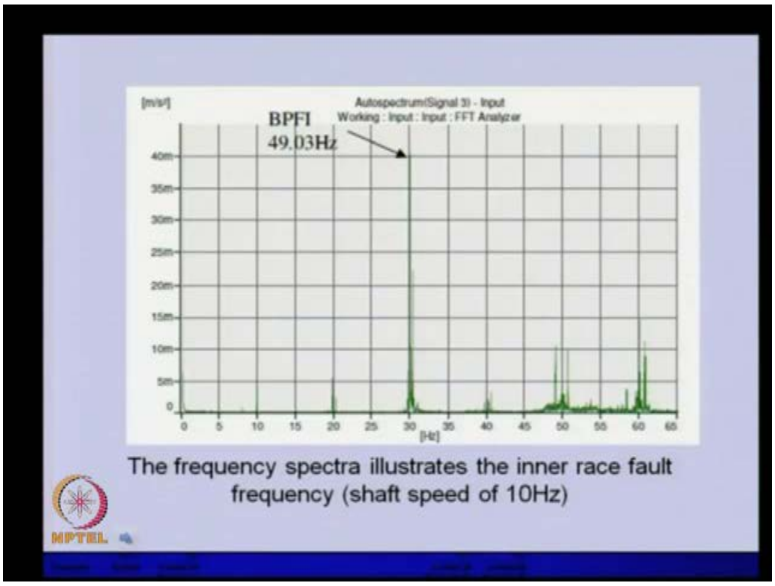

So, this is a typical signal in which the inner raceway defect is there. So, you can able to see that there is. So, in this experiment we had normal bearing and dealing with various kind of defects .So, we measure the signal with normal bearing and with various defects. We plotted in the frequency domain. So, this particular peak you can able to see corresponding to the inner race fault frequency, which is very close to the calculated once.

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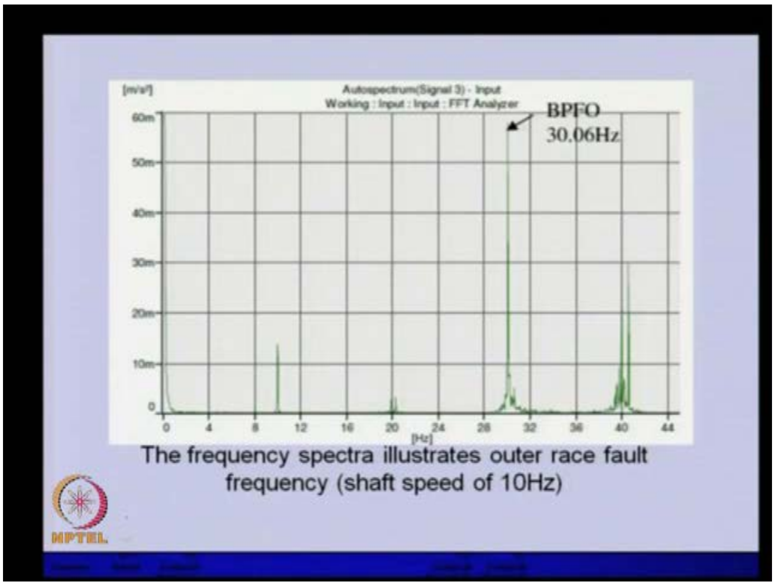

Similarly, this is for the outer raceway bond outer raceway fault frequency. So, this is very predominant and the value, which we calculated earlier, is close to this. Based on the speed of the rotor they may change, as we have seen in the previous calculation.

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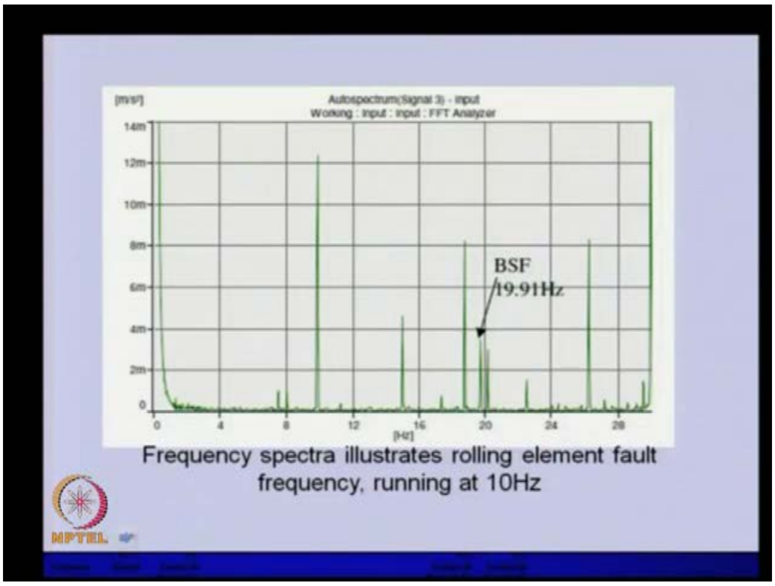

Similarly, this is for the rolling element fault at 10 hertz. Some spike we can able to see. So, we if we have baseline signal in which there is no fault. If suddenly such peak comes, then we can able to pin point, what kind of fault is there in the bearing.

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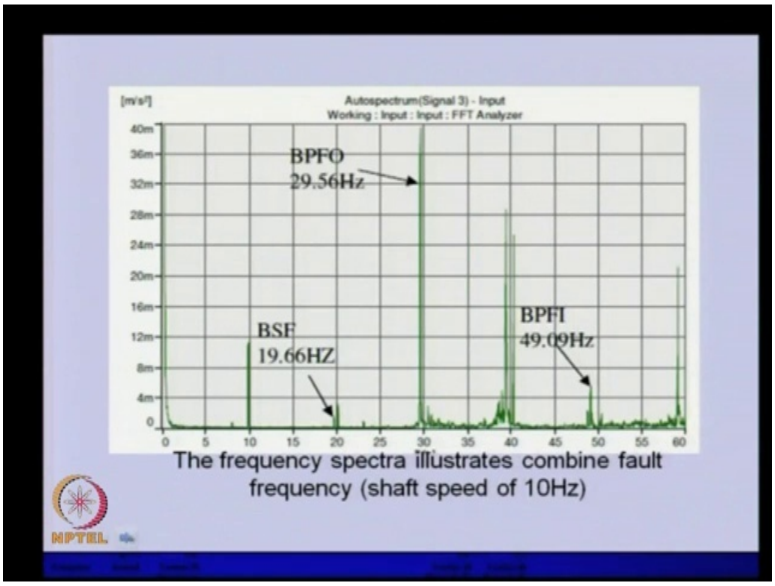

So, this is when combined fault is there. So, you can able to see outer raceway inner raceway and all kind of faults by. Fault is there. We can able to see, when simultaneously all faults are there in the bearing. So, that will reflect in the vibration signal or the frequencies. Now, we will take up another machine element that is here, in which we can have several kind of fault. So, one typical case study we will be taking up after a brief description of the gears, in which what kind of frequency it can produce in the vibration signal.

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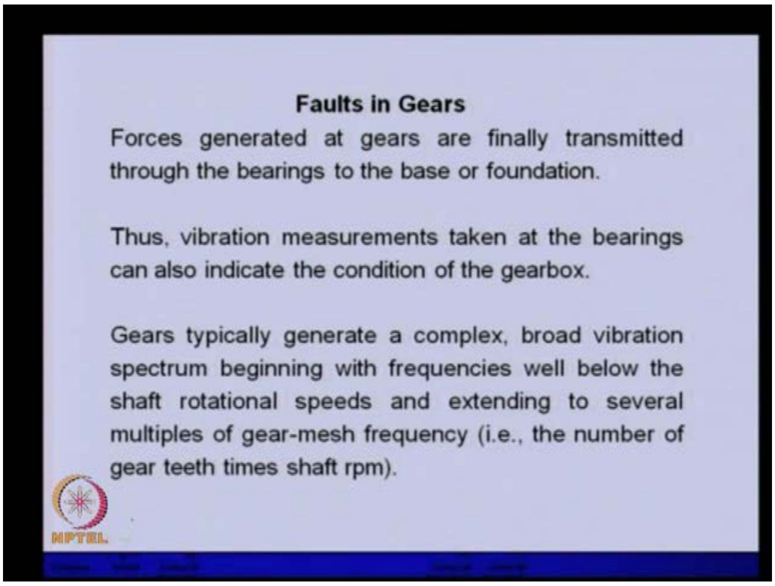

So, the forces in the fault in the gear the forces generated at gears are finally transmitted through the bearing to the base of or the foundation. Thus, vibration measurement taken at the bearing can also indicate the condition of the gearbox. So, finally whatever the dynamics is taking place at the gear, reflect at the bearing location. If you are measuring the vibration there, it will give a condition of the gear also. So, gears typically generate a complex broad vibration spectrum beginning with frequencies well below the shaft rotational speed and extending to the several multiples of gear mesh frequency. Gear mesh frequency is nothing but the number of gear teeth times shaft speed.

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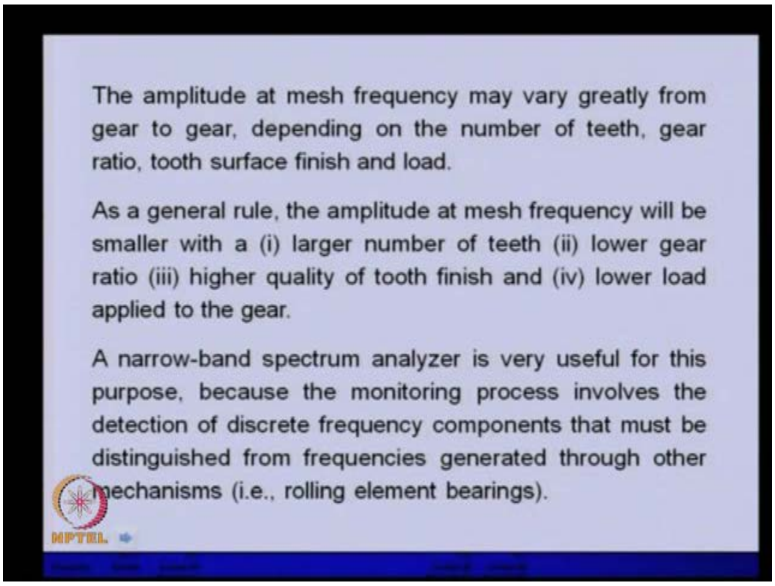

So, the amplitude of mesh frequency may vary greatly from gear to gear depending upon the number of teeth, gear ratio, tooth surface finish and load. As a general load the amplitude at mesh frequency will be smaller with; a larger number of teeth, lower gear ratio, high quality of gear finish and lower load, applied to the gear. A narrow bands spectrum analyzer is very useful for this purpose, because the monitoring processes involve the reduction of discrete frequency component. That must be distinguished from frequencies generated through other mechanisms like rolling element bearings.

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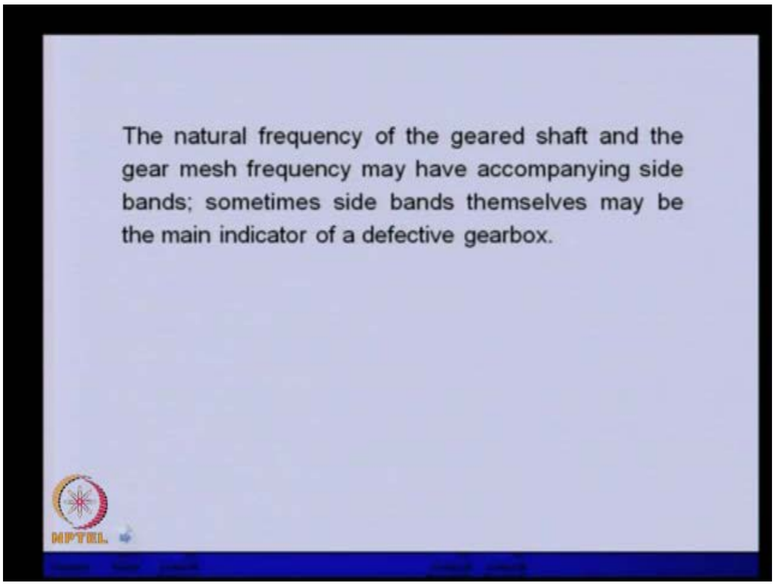

So, the natural frequency of geared shaft and the gear mesh frequency may have accompanying the side bands; sometimes side bands themselves may have main indicator of the defective gearbox. So, if there are sidebands in the in the gear to the gear mesh frequency, then we can have the fault in the gearbox.

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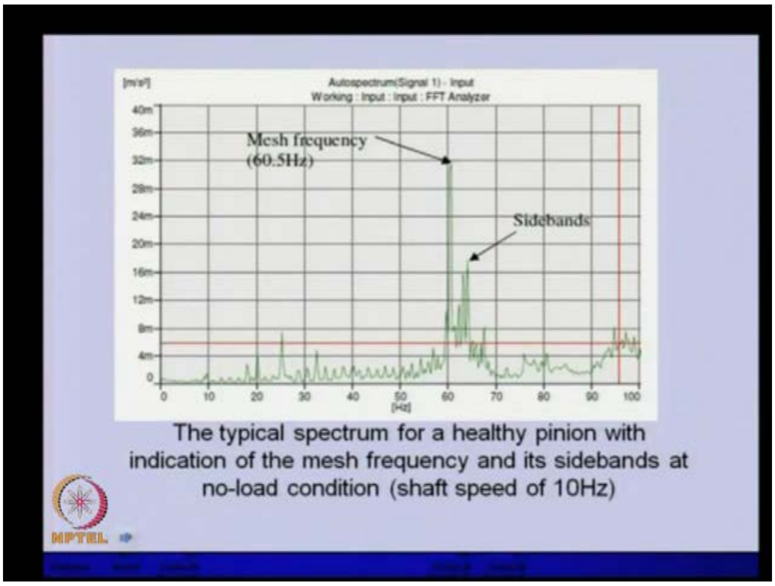

So, this is a typical spectrum for a healthy pinion with indication of the mesh frequency and its sideband at no load condition. So, you can able to see that there is a using meshes frequency for a typical gear and this kind of sideband. So, only one side is there. But if severity of the fault is more, then we can have sidebands in other side of the mean particular frequency. This mesh frequency will be there always. But if there is fault then the amplitude of this will increase.

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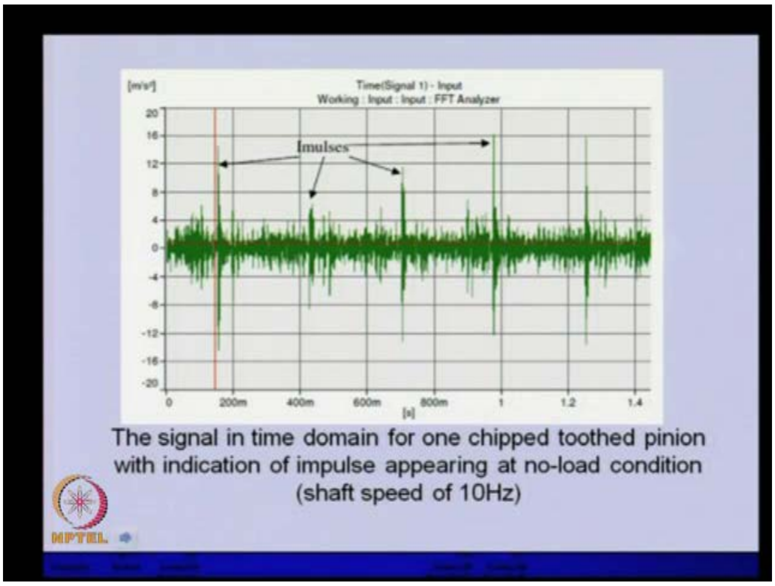

This is a typical time domain signals. Generally, time domain signal is very difficult to difficult to analyze. But in this particular case, the signal in time domain for one chipped toothed pinion. So, one tooth is being chipped or removed. In that obviously we will be having impact. So, whenever there will be that particular tooth will appear during the machine there will be impulse. You can able to see there are impulses, which are there that is indicating that the one tooth has been chipped off. This could be indication of the one tooth has been chipped.

We can have a baseline signal and we can able to compare with that. These impulses may not be presented in even healthy gear. In the today’s lecture we have seen various kinds of fault and without going too much into the mathematics, we directly try to see; this kind of faults, how they can be detected especially through vibration signatures in frequency domain. We took up various kind of fault like unbalance boat shaft bent shaft, even the rubs gears bearings. We have seen that each of these machine elements is this kind of faults. They have some unique frequency, which they give, which they reflect in the vibration signal.

As a maintenance engineer, obviously, we need to look into these frequencies continuously. If they appear suddenly, then we should able to pin point, where the fault is appearing or where the fault is starting to grow, before the failure of the whole machinery may take place. In the subsequent lecture, we will be extending the similar concept of the condition monitoring. But we will be giving broader view of the condition based model condition monitoring in which, we will be looking into the various techniques, which are available at present. What the research people are doing on that particular topic? So, just we will retry to brief them in next lecture.