Dynamics of Ocean Structures
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Lecture – 36
MLATs with Passive Dampers

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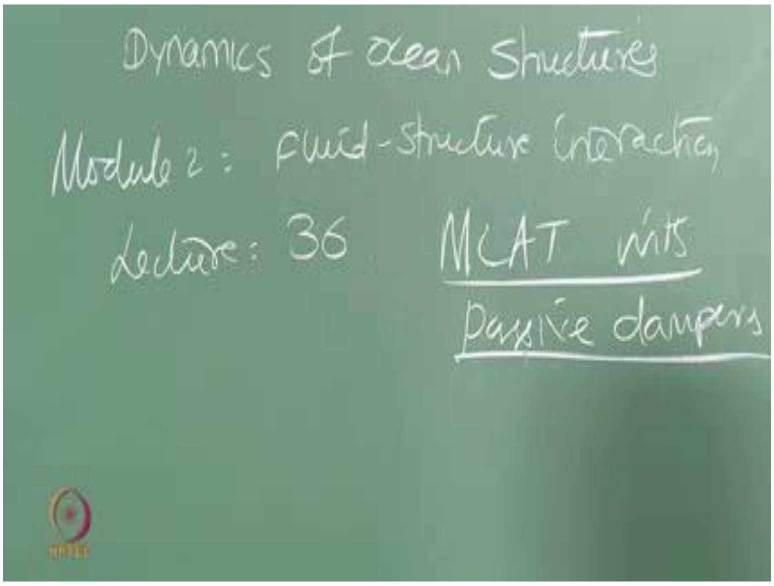

So, in the last lecture we discussed about different kinds of damping devices, which I generally used for offshore structures as a concept, but people have used it for land based structure is by enlarge. So, one of the interesting idea what people have attempted in land based structure, is do base resolution, when the structures subjected to later loading caused by wind or by earthquakes. However, an attempt on response control of structures under a wave action, on offshore structures is very limited in the literature. But there have been studies carefully done by researchers, which have been indicated in the last lecture, that one can try to use passive damping devices in offshore structures. The fundamental requirement for using a tuned damper or a passive damper in offshore structure is, the structure should remain essentially long period structures

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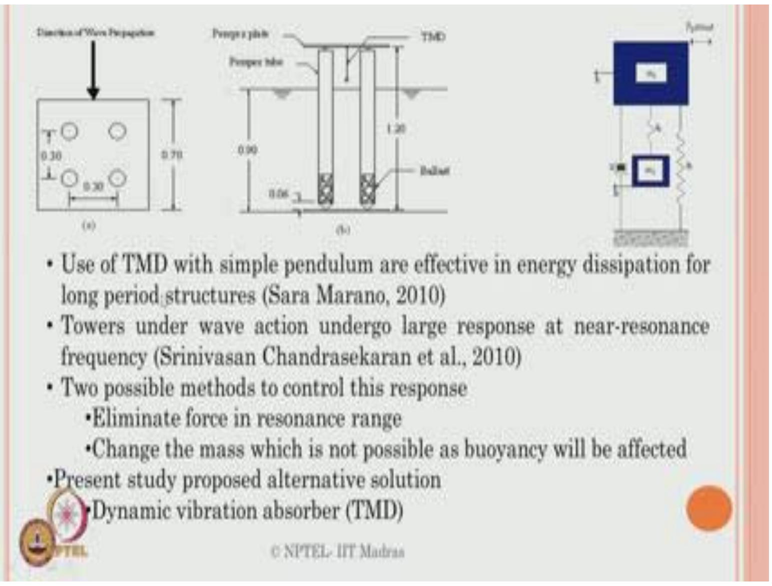

It should be remain actually a flexible system. The system has to remain for a long period, because Sara Marano 2010 showed, that if you have a long period structure the energy dissipation can be effective, by having a passive damping system which you identified, as one of the system as tuned mass damper, which is shown as t m d. The conceptual idea given by Sara Marano is shown in the right hand side in the sketch here.

The primary system m 1 remains is a primary mass, and the degree of even in the primary mass is x 1, where as the secondary system is again mass m 2, it is having k 2; however the primary system has stiffness, and see which is damper the secondary mass system, does not have a dampered, it has only a spring mass model. So, it is a 2 degree freedom system x 1 and x 2 as shown in the figure here, when x 1 is subjected to any excitation, may be in this case the general system of p zero sin omega t. One can use tuned mass damper to see effective response control of the primary system, when you are able to tune the secondary mass system, to that of the frequency or other parameters. We will see what are those parameters to that of m 1.

Now, interestingly if you are looking at the response control of the system at near resonance frequency, which is generally attempted with a researchers, by enlarge. In that case tower under wave action undergo larger response as near resonance, and it is always the good idea to attempt the response control under re resonance of the excitation frequency; that is omega bar closer to omega, a narrow band you attempt the tuning at that bandwidth. Let say recommendation generally researchers make, if you want to achieve the maximum response. Obviously, you know you really wanted to find out the tuned mass parameters, at the frequency matched to that of the natural frequency and the excitation frequency, which we called as a resonance band; obviously, you cannot solve this equation analytically, because we would not be able to get the answer

Secondly, you will not be able to actually tune it exactly even experimentally; because you will bother, it will damage the primary system. So, you can go get along only the nearer end. So, you must get these tuned ratios, or tuned mass parameters, approximately with trial and error experimenter, then feed them in analytical problem and try to see. Now, we will see what is difficulty when you attempt to solve this problem analytically. I will come to that point now, but generally the researchers are shown in the literature, there are two methods by which we can control the response of complain towers; one is we can always eliminate the force in a resonance band, which is not in our hand, because very difficult, you cannot always filter in the force in a resonance range only, you can always filter the force by some mechanism. We have shown a mechanism in the last presentation then you can put the perforated cylinder or exterior cover, you can always reduce the forces on the structure

But if you really wanted to reduce the response only on a specific band width that of frequency band, it is not possible actually, unless service you have an active control of the system, where you instigate a response opposite to that of the excitation frequency which is an active system which requires lot of power, in passive system it is not possible. The second alternative what we can do is, you can always try to change the mass. In this case it is not possible, because I am looking for a system which is buoyant controlled. So, in offshore structures, the second option of changing the mass is difficult. So, we have got only one option, where you can control the primary mass, using a tuned system which is dynamic vibration observer mechanism, which in this case attempted in the solution here.

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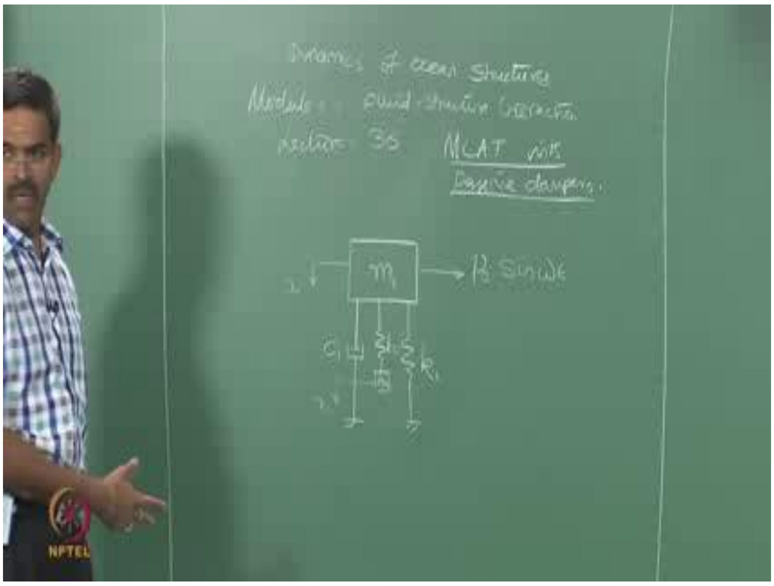

So, we will now pick up this particular case and see, how this can be mathematically modeled. So, I have as a mass system here, which is having a damper, which has of course, a stiffness which can be evaluated, this is k 1, this is c 1, and this is of course, m 1. And now I have a system, which is m 2 and q 2; that is what the systems suggested be Marano also. So, this is now subjected to and excitation frequency, let us call this has a p naught sin omega t, where omega is the excitation frequency of the system, and this is of course, a two degree system model, I have this has x 1, and of course, this is x 2 and both of them are independent, please understand because the k 2 will control the relative response between m 1 and m 2. If m 2 is attached m 1, or you put m 2 is exactly on the m 1 itself, then x 2 will not have any value, but in this case x 2 will have a meaning, because the response of mass, secondary mass will be different from that of the primary mass, because of the k 2 adjusted; that is a actually the whole problem, which makes an interesting; that is why I want to tune m 2 to the top m 1. So, if you do not have this concept, then I will not able to tune it.

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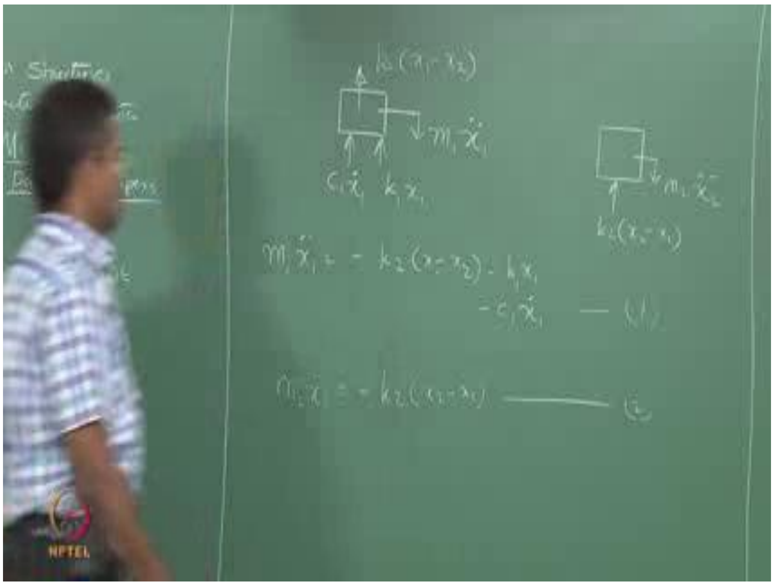

Let us pick up this problem and see, how I can write the equation of motion for this first and see the complication. So, which is now when you try to pull this mass down this force will apply in the opposite direction. So, that is going to be k 2 of x 1 minus x 2, the same concept what we use in the first module for writing the creative displacement. Similarly, the second mass will have a system like this, which can be simply m 2 x 2 double dot, and this is going to be let us say k 2 of x 2 minus x 1. I am using the same concept and same method by which we wrote the equation (Refer Time: 7:06) earlier. So, m 1 x 1 double dot should be equal to minus of k 2 x 1 minus x 2 minus k x 1 minus c 1 x 1 dot that is the first equation of motion (Refer Time: 7:32) as second m 2 x 2 double dot x minus of k 2 of x 2 minus x 1 which will give me a second equation of motion.

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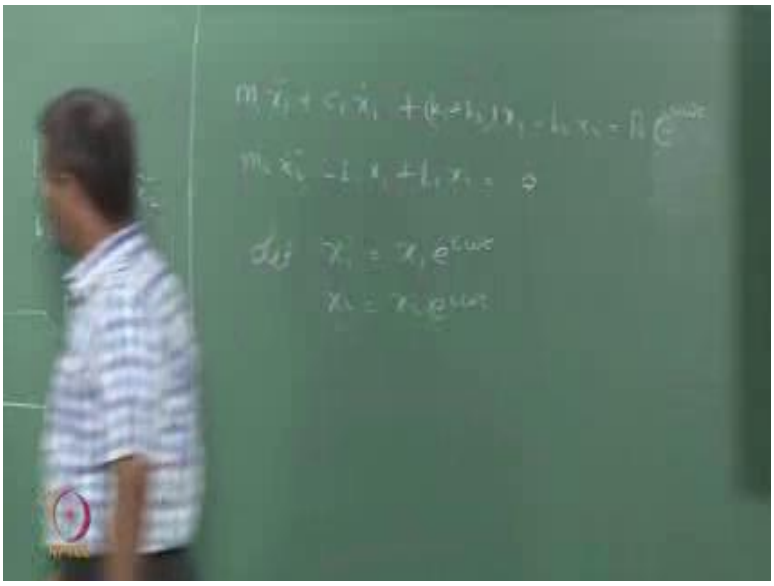

Let us expand this m 1 x 1 double dot plus c 1 x 1 dot plus k 1 plus k 2 of x 1, because k 1 here and k 2 here in both are negative, take it with the other way it becomes positive, and k 2 x 2 is positive here, goes there minus k 2 x 2 is let us say p naught sin omega t, is an excitation value available to me. The second one will be m 2 x 2 double dot minus k 2 x 1 plus k 2 x 2. It is zero because secondary mass does not have an external force. So, let us make it more general I want to include both the components. So, I will make this has p naught e I omega t. So, now, even you try to solve this, I will actually get the response as x 1 and x 2. So, let us say one of the interesting in basic method in solving this equations of motion is, you assume this and write iterate.

One can now ask me a question that is why the responses should be as same as that of the bandwidth of the excitation frequency. I can take this frequency band of the any value, why I am looking this, because I am looking for the maximum response. I can always looked at this frequency by any content, I am looking for the maximum response, because I am trying to see what would be that maximum response which needs to be controlled in the system. So, I say generically (Refer Time: 9:59). Now once I have x 1 I can always derive x 1 dot and x 1 double dot, substitute back in equation one and two, get the components in the equation, let us see how the equation looks like.

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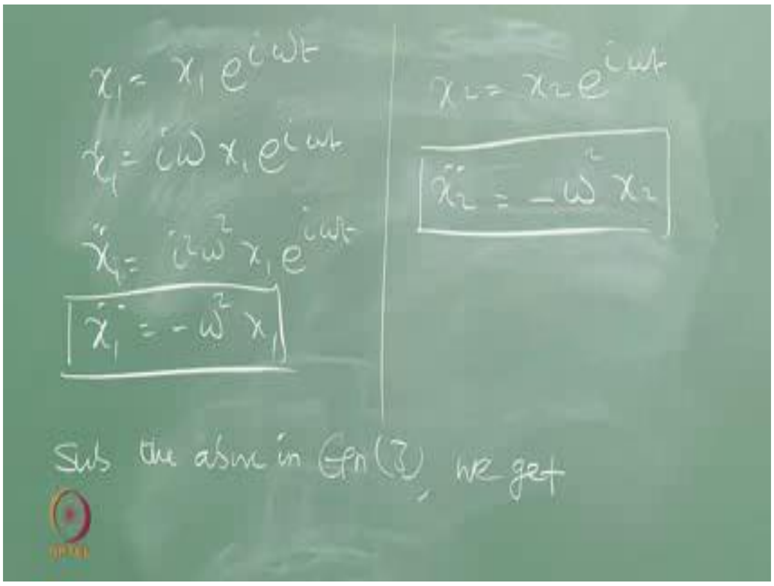

The classical relationship which we all know, similarly now substituting the above in equation three we get, let us see what do we get.

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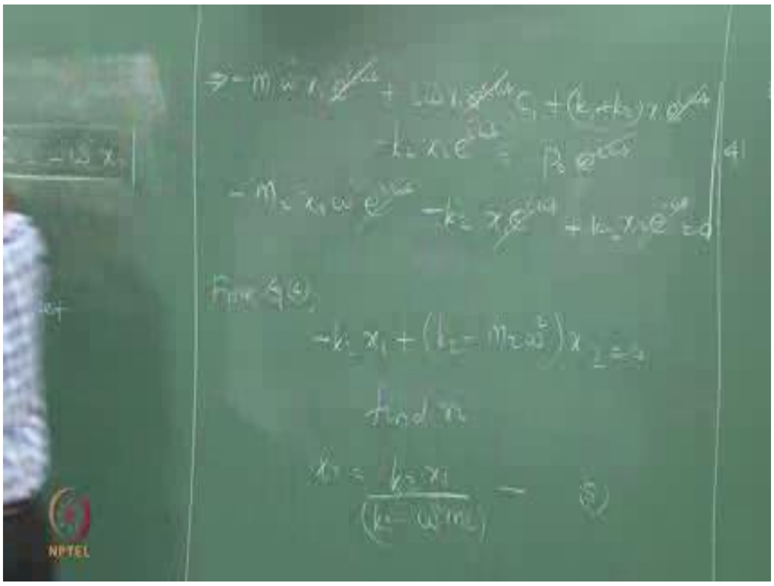

So, let us say minus m 1 omega square x 1 p I omega t, because x 1 is actually x 1 as omega 2 plus I omega x 1 p I omega t c 1 plus k 1 plus k 2 of x 1 e I omega t minus k 2 of x 2 e I omega t is p naught e I omega p minus m 2 x 2 omega square e I omega t minus k 2 x 1 e I omega t plus k 2 x 2 e I omega t set to zero. So, once you do that even I omega t here also goes away, I simply the equation. So, I can re write the second equation, I call this as equation set four, this is a set four. So, from set four let us club the x 2 part and x 1 part separately. So, minus k 2 of x 1 plus k 2 minus m 2 omega square of x 2 except to zero. So, from this I will know the x 1. I will get to k 2 x 2. So, find x 2, where x 2 is going to be k 2 x 1 by k 2 minus omega square into, and call this as equation number five, substitute five in four a. This is actually four a and this is four b, second equation we get.

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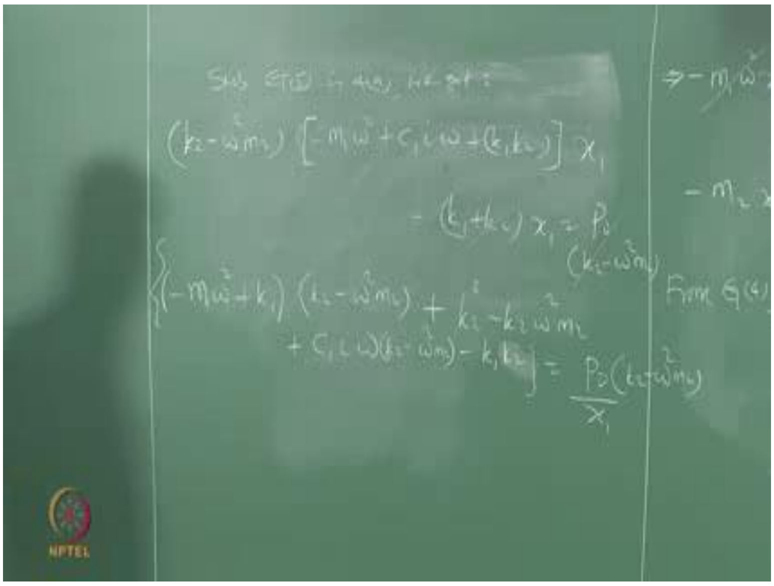

I will write the step exactly what it is. So, k 2 x 1 minus omega square m 2 minus m 1 omega square plus c 1 I omega plus k 1 plus k 2 of x 1 minus k 1 plus k 2 of x 1, will be actually equal to p naught; of course, the denominator p naught of k 2 minus omega square m 1. So, k 2 omega square m 2 minus m omega square I c 1 omega 1 k 1 plus k 2 of x 1 minus k 1 k 2 of x 1 and the denominator gets multiplied, because as an x 2 here, it is multiplied with p zero and get this value.

We simplify this, what I will do is. I multiply minus m 1 omega square plus k 1; I am separating the secondary mass and primary mass system from this. This is m 2 here this is m 1 here minus m 1 omega square k 1 plus k 1 with k 2 minus m 2 minus omega square m 2 plus k 2 square minus k 2 omega square m 2 minus k 1 k 2. This is going to be a product, the whole, because now you have no variable x at x 1 of course, plus this term c 1 I omega k 2 minus omega square m, will be equal to, let us say p naught x 1 with the multiplier of this k 2 minus omega square. This is an advantage of writing like this; I will come to the point now. Let me call this equation number let us say six.

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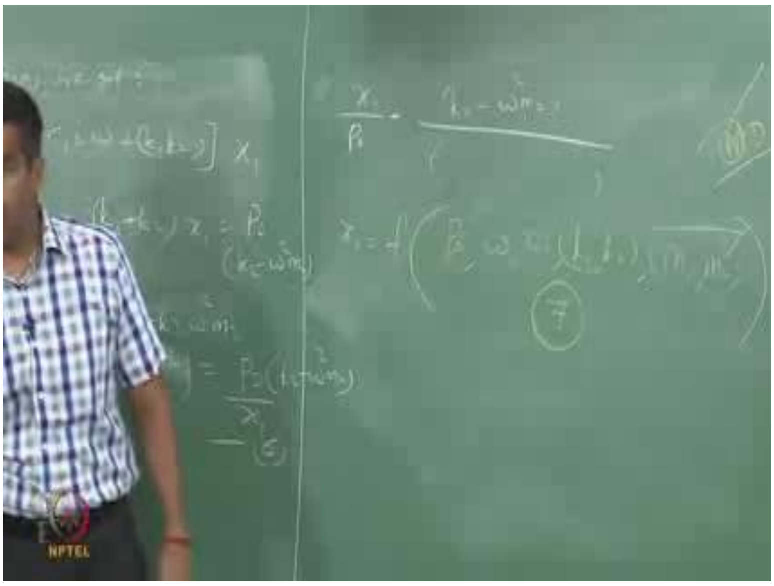

I want to write this as a function of static displacement which is x 1 by p naught; yes, p naught is the amplitude of the system and x 1 is a vibration or displacement of the system. So, this should give me equivalence static displacement of the primary mass, so x 1 by p naught. So, it becomes this value divided by the whole in the denominator. I simply say it is k 2 minus omega square m 2 by this function. Now interestingly x 1 depends on the following, let see what are the dependency of x 1, because x 1 is what I actually want, and my aim is to control x 1 not x 2. My aim is to control the response of the primary mass, not the secondary mass at all. So, it is it should be the functions of the following, let say it should be the function of; of course, the amplified excitation p naught, this is available in the equation itself. It can be a frequency of the exciting force, it can be the damping of the primary system, it can be stiffness of the primary and secondary system, and more importantly this is now a function of the primary and secondary mass itself.

So, there are one two three four five six seven, seven parameters. So, there are seven unknowns, or seven factors which control actually the response of x 1. Let us see out of which what we can control; that is how you can tune it. Now we have no control on p naught, because that was actually shown here, exciting force cannot be controlled in a given bandwidth, you have no control on excitation amplitude p naught. Omega actually is a forcing frequency which we are trying to capture, but the forcing frequency can change. You cannot actually say my system is tuned to a specific frequency of omega only. For example, if the exciting frequency changes from this band, your tuning will not be effective. So, this is also cannot be actually a controlling parameter, which you can depend on controlling x 1.

Of course damping a 2 zeta omega m which is actually a natural period of the system, or which you have got alter k and m of the primary system, which you cannot, because if you alter k or k 1, let us say specific, if you alter k 1 then your buoyancy will be affected. If you alter m 1 your top side details will be effected; therefore, you cannot play much with c 1, but you cannot always find out the ratio between m 1 and m 2, because m 2 is what your giving to the system, m 1 is present in the system, but m 2 versus m 1 can be a ratio; that is why it is called tuned mass damper.

So, we are taking only about the tuning off mass property one can also try to say a scanning of ratio of k 1 and k 2, compare to m 1 versus m 2 to that of k 1 and k 2, k 1 and k 2 actually is a bending stiffness in a given system, because is a tower, you know tower is like a cantilever it bends. So, it is going to be e a by l. So, if you want to invoke k 2 to that of k 1, because k 2 actually is an axial spring. So, nothing, but a e by l. So, spring actually whereas k 1 can be a system of a bending nature, because this is a tower. So, you cannot actually calibrate k 2 versus k 1 ratio effectively to get the response control, number one. Number two this ratio can be more effective and targeted, because m 1 is a larger in number in terms of it is magnitude, because this is a top side detail in the given platform. So, m 2 can be tuned to that.

Now, one can ask me a questions sir if I have five percent of m 1 as m 2, five percent of m 1 as m 2, where m 1 is 2500 tones which we saw in the last slide. So, twenty five in a tones means approximately about 2.5 or twenty five tones let us say we can go, where will have hang twenty five tone mass in a given suspended system. I will come to that question; I will answer this question at the end of a presentation. Now there is a doubt here that, if you arrive at a percentage of m 2 versus m 1, can I really physically post that m 2 in a given system, because m 1 is relatively large in number. We are not talking about a model, we are talking about prototype. So, can where can I hang this, I will answer this after completing the representation. Now complexity in solving equation is you have got seven parameters you want to tune x 1, you are not interested in finding x 1, you are finding the control and maximum x 1; that is a problem, it is an optimization problem.

So, you have got parameters, you have to tune them. Now; obviously, if you want to tune a find a result of this parameter, you must try with different omegas, closer to omega n, where analytical method will not help you. We have to tune different m 1 m 2 ratios analytically, for different omega two, because the moment you change m 1 versus m 2, you will see that omega versus omega n will change, and it may enter into a non soluble bandwidth in a problem. So, you cannot analytically solve this. So, you have to try for different ratios of m 1 m 2 by hanging it experimentally. Try to measure the responses and see, for what ratios of m 1 m 2 the responses decreasing or increasing the tower. Then for that band of values, you want again solve, you must again solve, naturally experiments will not be done at omega equals omega n, it is not possible. But you will closely go two mega equals omega n, and for different attempts of m 1 versus m 2 ratio, you can try to find x 1 decrease with that of original x 1. You pick up those values, feed in the optimization problem, solve it analytically, and now check what would be in a governing curve.

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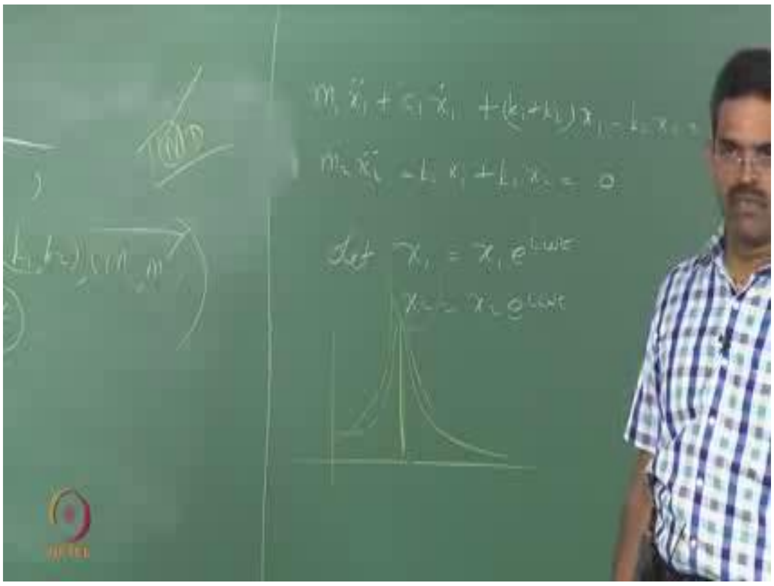

Now interesting you will see, and you will remember when I have a dynamic amplification factor, even in a damped system at omega equals omega bar, you will see that it is bounded, but the value is very high. If it is not damped, it is going to be unbounded. So, you will always notice in a given system, the dynamic amplification factor or the curve, daf curve, will always have a dominant peak closer to the resonance band. Now, interestingly when I show you the result of this, you will see that at omega close equal to omega n, this band will go to zero; that is a beauty here. Just see here how it is happening. So, this has been solved for optimization problem.

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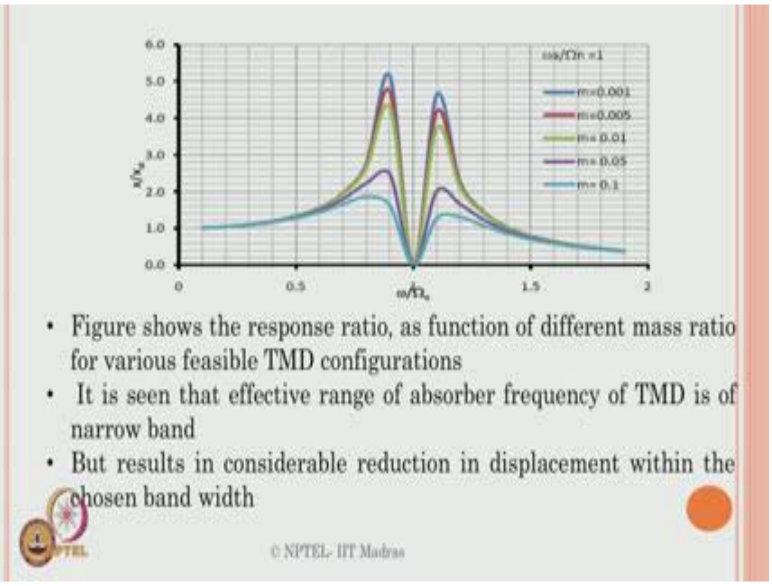

And now you see the curve here. At omega equals omega n, where I say omega and sigma n, the notation there, is this, is a views instead of omega n, just for avoiding confusion. Closely at one let us say, you see this particular peak, has been particular brought close to zero, it is not actually zero it is analytically zero, but practically it will become zero, close to zero, but it has got a demerit. The demerits are it picks up two adjacent bands nearer to omega equal to omega n. But you will see that they are bounded, they are not unbounded. So, this is a problem always whenever you have any damping present of the system, the response gets bounded, but you have got address that particular thing.

Now instead of having an unbounded response at omega equals omega n, we are able to get practically a zero response, my and we all understand that x 1 will be maximum; obviously, when omega equals omega n. So, the idea was to optimize that and x 1 was brought to practically zero at the resonance band, but it has given my counterproductive response to me, that it is giving two more bands. So, let us discuss this result now. The figure shows the response ratio, as a function of different mass ratio. This m shows the mass ratio of, let say 0.1 0.5 and 1 5 and 10 percent for different values.

So, it has been plotted with the ratio of frequency versus x versus x static; that is how it generally plotted for all our responses in dynamics. We always try to look at the response versus a static response; that is how daf is always plotted. So, the daf is plotted versus frequency ratio, and the figure gives me the difference for different mass ratio, for variable t m d configuration. Now one doubt will come in mind 0.1 percent of m is m 2. So, m is about 2500 tones, 0.1 we will calculate the value comes to be very large 2.5 25 tones let us say, where will we hang this mass, how it is possible, we will come to that point. So, one can understand very clearly from here that it is seen that the effective range of absorber frequency of the t m d is of a narrow band only.

It means only in these bandwidths, only for these bandwidths, and this bandwidth a t m b will be effective. So, you will not be able to control the response, for a wider band starting from a frequency of these to the frequency of this. It is not possible to have a single tuned damper in a passive system. In active system it is possible; you have to keep on varying the frequency or the mass ratio, actively either deleting or adding a mass. In a passive damping system, since I am not giving any external input of the system interest of energy, you always have a deficiency in saying; only in a specific band this is able to control the response.

Now, as you see here, let us as the mass ratio keeps on increasing, let say this corresponds to this value; though the color I do not know you have to realize, these two colors are different, this is actually corresponding to this, and this color corresponds to this, I think you will be able to. Let us look at this color this is very evident. So, if you look at this, the response actually starts, in the same value for all mass ratios, ends are the same value for all mass ratios, but closer to the resonance band, the responses control, from let us say 6 or 5.5 to that of 2.2. It is about we can say half of the response, fifty percent of the responses controlled, but it is sort only very narrow band. The results are shown to have a considerable reduction in the displacement, within the chosen bandwidth.

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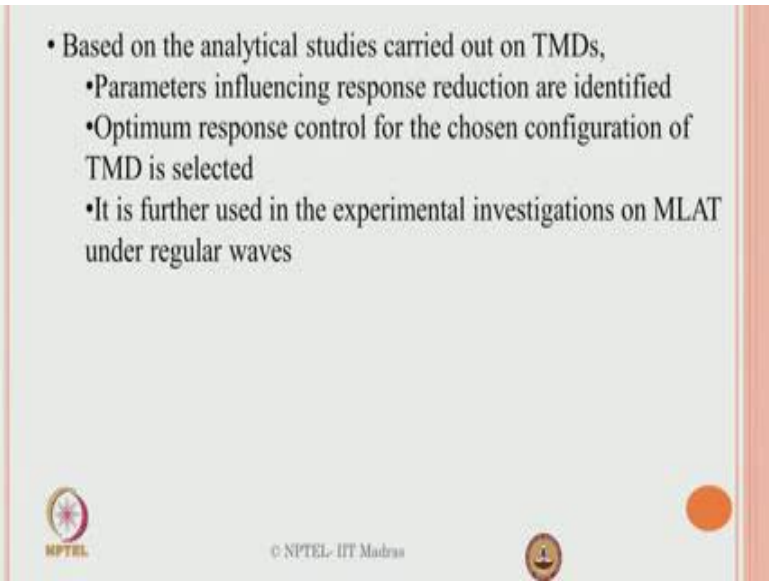

Now, based on the analytical studies now picked with these values, parameters influencing, the response reduction are identified, optimum response control has achieved for a chosen t m d configuration, then it is further investigated under regular waves, when you do that

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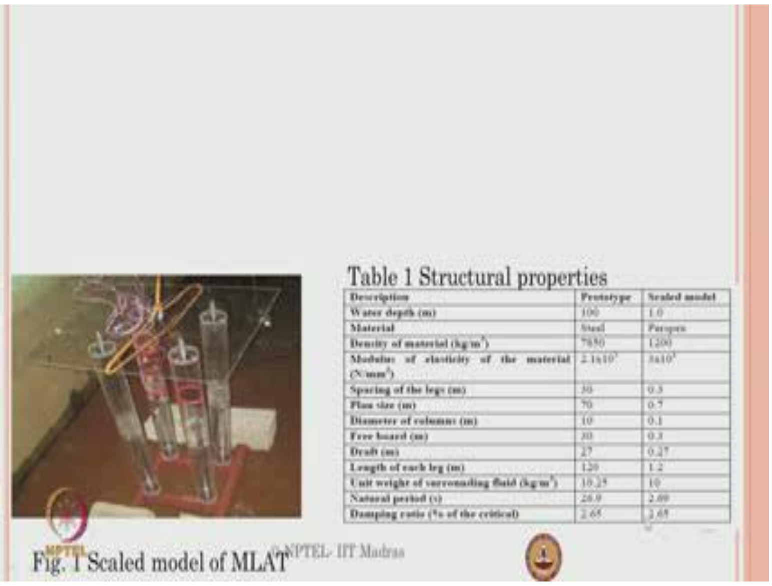

Table 1 Structural properties

Description	Prototype	Scaled model
Water depth (m)	100	1.0
Material	Steel	Perspex
Density of material (kg/m^3)	7850	1200
Modulus of elasticity of the material (N/mm^2)	$2.1x10^5$	$3x10^3$
Spacing of the legs (m)	30	0.3
Plan size (m)	70	0.7
Diameter of columns (m)	10	0.1
Free board (m)	30	0.3
Draft (m)	27	0.27
Length of each leg (m)	120	1.2
Unit weight of surrounding fluid (kg/m^3)	10.25	10
Natural period (s)	20.9	2.09
Damping ratio (% of the critical)	2.65	2.65

This is the four legged, let us say hinged bottom is there, and the deck is place in the top it is without water, the flume is then filled up with water, the structural properties are shown here, the scale trusted is one is to hundred. This is a prototype, this is scaled model. And these are the particulars, what we have as per as the model and prototype equivalency are concerned. We are talking about a damping ratio approximately 2.5 percent, because it is a steel structure in reality.

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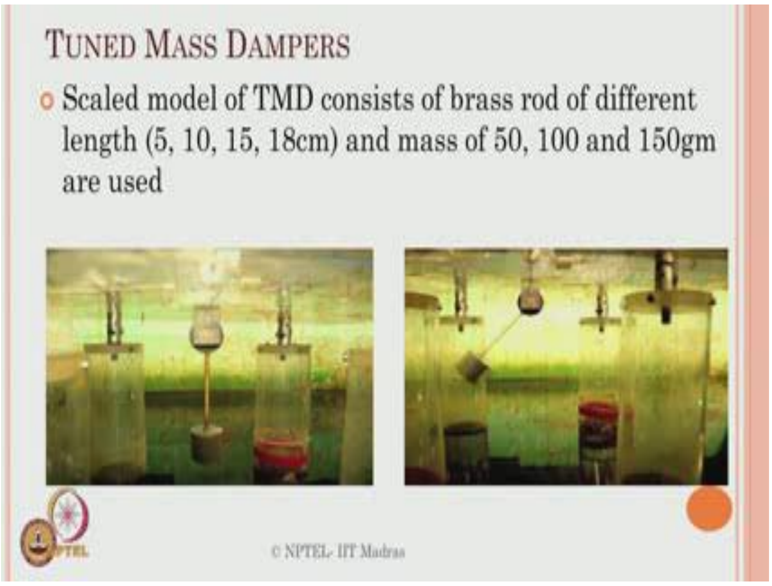

The scaled model of t m d with different extension of the length of the rods, because the pendulum system should have a rod length, and we all know the period of the rod length depends upon only the length of rod not on the mass, for a pendulum. So, a different configuration would be tested for 50 100 and 150 grams or 1 is to 100 scales, or basically they will become in tones.

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Model		Proto type		
Length (cm)	Mass (gm)	Length (m)	Mass (ton)	% of mass of TMD to prototype
11.5	100	11.5	100	1.42
18	150	18	150	2.14
23	200	23	200	2.85
--	250	--	250	3.57

If we will look at that you will see for different configuration they have been identified, and corresponded a percentage mass, you will be able to know what is the maximum percentage of 4. Let say 4 percentage is the maximum one we have, minimum is about 1.5 percent.

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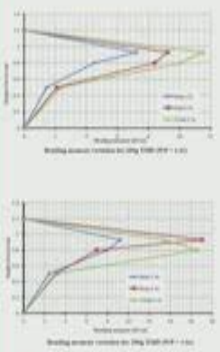
INFLUENCE OF WAVE PERIOD ON BENDING MOMENT VARIATION OF THE LEG



Maximum values of bending moment for different wave periods

T_{mass}	wave period(s)	Wave height (cm)	BM (kN-m)	Wave height (cm)	BM (kN-m)	Wave height (cm)	BM (kN-m)
100gm	1.3s	8	8.30	9	9.12	12	17.64
	1.5s		8.40		15.48		18.00
	1.7s		10.44		19.31		--
	1.9s		12.84		--		--
150gm	1.3s		6.72		16.44		20.64
	1.5s		10.44		22.92		26.68
	1.7s		16.92		24.12		26.68
	1.9s		19.65		--		--
200gm	1.3s		7.25		9.24		11.32
	1.5s		9.24		17.04		16.56
	1.7s		12.36		31.2		27.72
250gm	1.3s		8.52		9.60		17.52
	1.5s		8.52		15.12		18.72
	1.7s		10.00		15.12		--
	1.9s		13.32		--		--

The most interesting part is what response actually we are talking about. There are two response we talk about; one the bending movement on the legs, two the axial or sorry later displacement of the deck. We are taking about two responses, because these are the two responses which can have. We can say they are inter connected, you may wonder how they are, when I have a theta given to the leg; obviously, they said down effect or the push effect will be there and the deck. So, they will give me x 1 automatically, and they are related we know that, we already said in the last equation, in the last lecture

So, if we look at the influence of wave period and bending on variation, you will see that for different configuration of t m d's. So, varying form 100 grams to 250 grams, for different wave periods, for different wave heights for example, if these are all converted to meter your scale is 1 is to 100. So, these are all the operation sea states. So, you believe and you will agree, for a de portal platforms or complained systems in medium water, the water deck the wave height generally varies from 6 meter to that of 12 meter, 15 meters is the maximum height.

So, we have tested it with up to 12 centimeter, on the other hand twelve meter and prototype, and the periods are anyway vary from about six seconds to that of fifteen seconds, wave operated up to two seconds, which can be related to the value. You will see that bending movement keeps on increasing for different values of wave height, which is quite obvious. The only difficulty and interesting part here is, this bending on does not happen at the central of the leg. Now, this bending on maximum value, for all the cases is the wave heights, is shifted towards the upper end of the deck, where it shows stability of a deck, having shifted below it would give you a more moment to the leg actually. It is shifted towards the upper side; it is practically at one third of this not a 50 percent of this; that is one observation.

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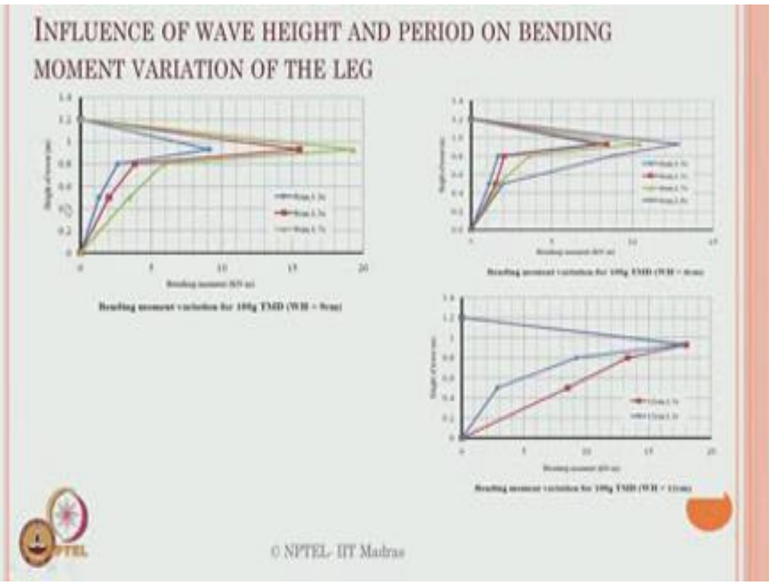

If you look at the influence of wave it and period on bending moment variation in the leg, you will see that this plot is for nine centimeter, this is for six centimeter, this for twelve centimeter. And you will see that predominantly as the wave period, is increasing for the same wave height, you will see the bending movement ordinates, do not get displaced or moved off from the point of concentration, but only magnitude changes. On the other hand this is very interesting to see, the stress concentration changes, but the area or the material at least stress concentration is there does not change. It means only at that particular segment the member has got to be strengthened, that gives the very good design perspective. If you really wanted to adopt a t m d, which of course, induces two additional bands, where which improvises additional forces in terms of bending movement on the leg, only that segment need to be strengthened, may be using a collar, may be use in a additional joint, or may be using a thicker member in terms of it is bending strength.

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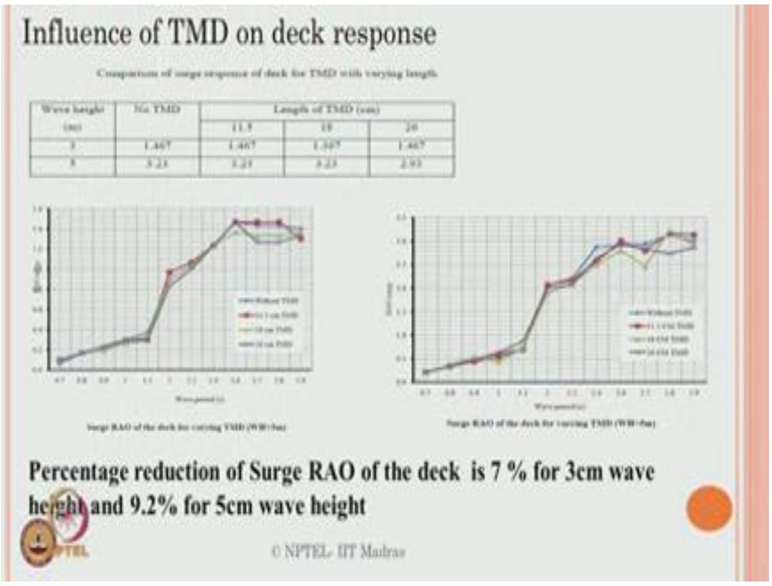

Then if you look at the influence of t m d on the deck response, for varying lengths of t m d, for different wave heights 3 meter to 5 meter, because these are prototype values. When you look at no t m d it is about 1.467. If you look at the length of t m d and it keeps on varying you will see that, for the specific value of chosen mass configuration, the response of the deck is reduced, but otherwise it is un altered, you will see in the curve also more or less they follows the same procedure, because the blue one shows without t m d, which is the blue one here, blue one here, with t m d for different lengths of the mass that is nothing, but the period variation, you will see they do not actually alter excepted one location, where they are altering and reduced, and where they reduced.

So, you do not actually get the reduction of response for all mass configuration at the same point, for wave height and wave period, it is a band. So, it is effective only on a specific band. So, the percentage reduction, is seen as about seven percent for, let say 3 meter wave, and about ten percent for let us say five meter wave in prototype. So, we are able to get a reduction of about, close to varying anywhere from 3 to 10 percent, which was attempted, and let say acknowledged by Sara Marano about fifteen years back that, yes passive damper system, can be applied to a configuration of this order, if you want to really reduce the response and complained systems like emulates.

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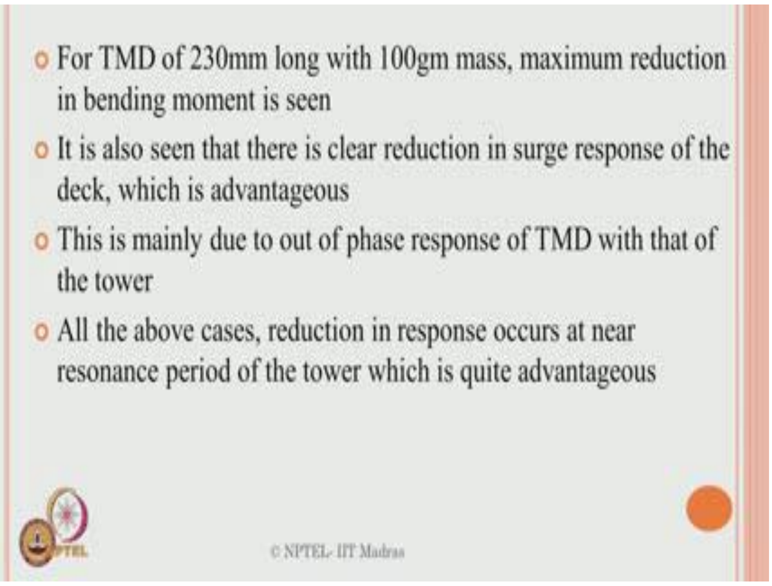

So, for t m d of 230 millimeter long with 100 m m mass maximum reduction in bending moment is seen. It is also seen that there is clear reduction in surge response, which is quite advantageous, as a deck responses concerned. Actually this has happened, because of the out of phase response of t m d with that of the system which we showed you in the last lecture. All above cases reduction response occurs at near resonance period, of the tower which is quite advantageous.

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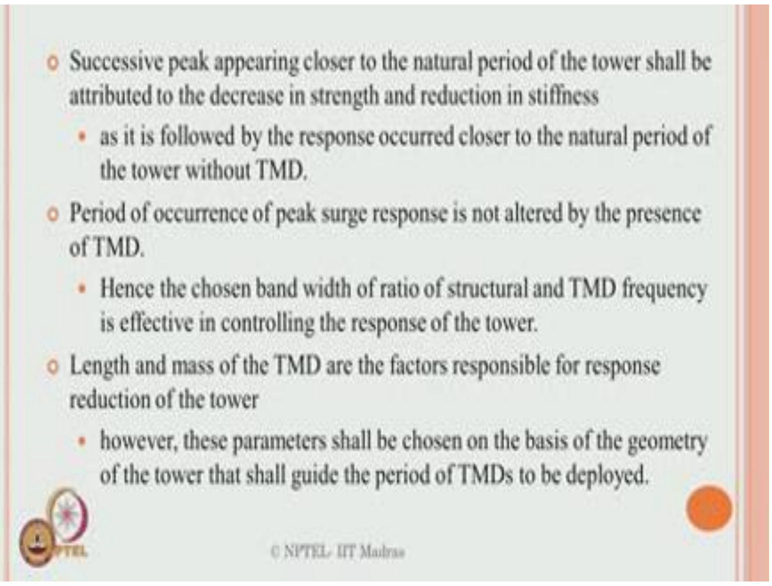

The successive peaks which occur closer to the natural period of the tower will be attributed decrease in strength reduction in stiffness, because this is to be followed, by the response occurred closer to natural period of t m d, and the tower without t m d. The period of occurrence of peak surge response is not altered; that is very important, because it is occurs at the same period, whether you have a t m d or do not have a t m d. So, hence the chosen bandwidth of ratio of structural frequency t m d frequency is effective, in controlling the response of the tower; of course, the length and mass of the t m d it should be chosen very carefully; that is nothing, the ratio of m 1 versus m 2, it has got to be chosen very carefully what we call as a mass ratio. Since the whole study is focused on choosing of ultimately the mass ratio, the whole exercise, in literature is attempted to be called as tuned mass dampers.

One came under dampers do not have actually any mass system in this, because 2 zeta omega n, where as there is only percentage of frequency or structural response, where mass is coming into play here. This is actually a passive damping system, which control which has only springed mass, only the pass proportional has got been altered; that is why it is call tuned mass dampers. One can also attempt this with tuned liquid mass dampers, which is generally done in shifts. If you really wanted to avoid larger roll, and pitch in shifts, while they are sealing, people use variable buoyancy engines or variable buoyancy submergence tanks, where the sloshing effect of the buoyancy chambers will be used, to control the roll or pitch responses on the ship hull.

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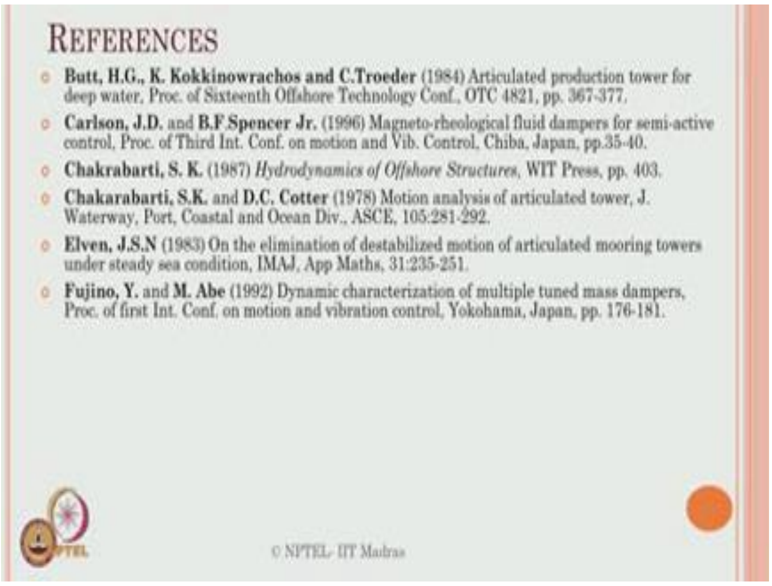

It has got very clear references, which talks about articulated towers alone. People have used different kinds of dampers, because Kalson and Spencer have tried with Magnotorial or Rheological dampers, which is a fluid damper within t m l d. So, we have used passive dampers. Chakrabarti has attempted to study the motion analysis and articulated tower, with quarter in 1978, where the study is not new. It does mean about may be thirty as an active practice, right from seventy eight people have been attempted this. Whereas Fujino and Abe has tried this kind of tuned mass dampers or vibration control in buildings in Japan.

Of course you have already studied the results from Jain and Datta. We acknowledge that studies Kirk and Jain also study the response control of towers. Of course, Lino Harilal has studied this explicitly with the m s restoration at I I T Madras which he presented here. So, we acknowledge the researchers contribution for the studies, what you are reported in this lecture today. And of course, Sarin Marano has been given and identification that they have used in recommended linear tuned mass dampers for structures in non-linear behavior, which is mean in original idea in this particular problem.

Now, the question comes here is, if you have got mass ratio of a very high proportion in terms of percentage, how can you actually suspend this kind of system in a given, because if you have got let say twenty five tons which is suspended from a spring, and as the mass oscillate like a simple pendulum, this can also have a negative impact on the given system, you can cause impact to the engine, if it is disconnected can cause disasters, damages to the top side of the platform etcetera, how this can be handled. Very interestingly, this is one extended research which I will not discuss it here for the time being. But people have attempted this in many ways. I will just give you a hint, so that this can be a research idea for people to take it forward.

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Now I am talking about only one system which is hinged here with a mass. So, this is nothing, but a spring and this is nothing, but the mass which is m 1 and k 2. Now the worry here is I am looking for a long rod with the larger m 2. Now I want to control this. Similarly I can use the system like this, I can use series. So, in that case this is l and this k 2 can be attributed to some of these springs, and this m 2 can be divided. So, it is a very interesting problem, and you can take this as an example and try to solve in equation of motion you will see that, the frequency will not drastically changed, but k 2 and m 2 can be played, this is a very interesting idea how people can adopt the recommended suggestions from these kind of studies to practical situation.

So, one can need not worry about ten percent recommended m 2, these always m 1 because the value is very high, 250 tons. You need not have to bother about that, because you can always try to use alternate system for this, and I can also use this in parallel with existing damping systems, like tuned liquid by mass dampers, magnetorheological dampers, etcetera which have been successfully used in buildings, but not a offshore structures. This has been a primary idea, which has been recommended and which has been presented to have a moderate understanding of. So, in this lecture we discussed about, how one can control the response on a complained system, because to control the response in a complain system the essentially researcher is identified that, the period of the structure should be long; otherwise you cannot.

So, stiff systems cannot be controlled. So, we have taken of a system of a in the medium water deck, like articular towers. We attempted to go for a passive damping system because the active dampers require external input energy, which may not be available at offshore end because now power is a major problem in offshore production. Therefore, people cannot advice and recommend and accept any active damping systems in an offshore, if it is a building in land it is ok, you will be able to give a power generation where as in offshore it is very difficult.

Therefore, one can look for passive damping system. We looked as literature we presented some reviews of literature interestingly. We have picked up a same problem of a single tower, we saw the single tower using a results from A K Jain and Datta, and we showed that how they are manipulated using a very interesting numerical simulation, and we understood them then we realized that why response control becomes necessary in a single tower to improved stability, and to control the response. We have done attempted to find out multi hinged towers, we solve the equations, I mean we prepared the equation of motion; we are left to solve the equation of motion.

Then we attempted to find out multi leg it is with t m d, which is a passive damper system, which can be easily attempted. We identify the parameters which are governing the response control, it becomes an optimization problem, it is cannot be solve analytical situation, because I am looking for omega equals to omega n. So, I have experimentally evaluated closer to omega n, found out the parameters, then substitute analytical and got the curves, and we identified that, though the response goes down in the daf at omega equals omega n, but it gives me a rise of two different matches, it is nothing, but have an elastic band here, instead of pulling this band top, bring this peak down you will see that the balloon will have a rise here. The energy remains same. So, that is what actually the problem is. So, it is effective for a narrow band for a chosen bandwidth only; however, the reduction has been seeing experimentally, analytically, it is about ten percent that is the idea in this structure. So, this has also showed some attempts in design of at's through dynamical analysis, which was the original idea given in this lecture, any questions?

So, in the next class we will talk about again complained towers, because we moved from fixed medium to now de plotters, we will talk about tension in a platforms, from the configuration concept, we will derived the equation of emotion, in terms of mass matrix stiffness, and Raleigh damping matrix for a TLP, then we will try to solve using a numeric’s of integration method, and show the results of TLP. Then we will move on to buoyant like structures in advance, how these hinges can be used from the deck also to control the response. Then we move on to large vessels plotting vessels like fsru's, and reclassification plants, where we use the same concept.

We show you results of experimental analytical and numerical and patents done at IIT Madras on this kind of research, where people can attempt this kind of platforms for ultra de potters. So, the conceptual development of design, which was tailed around a dynamic analysis using a FSOI fluid structure interaction. This is a focus of this particular module, which is research driven for this particular segment of lectures.

Thank you.