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Structural control of a steel jacket platform

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Abstract. This paper deals with the application of certain active and passive control mechanisms to control the dynamic response of a steel jacket platform due to wave-induced forces. The forces are estimated using the nonlinear Morison equation which provides nonlinear self-excited hydrodynamic forces. The influence of these forces on the response of a structure without and with vibration control mechanisms is demonstrated using a steel jacket platform as a simple example.

Key words: active control; control mechanisms; hydrodynamic forces; nonlinear dynamics; offshore structures; structural control

1. Introduction

Offshore structures have evolved from very stiff and relatively shallow-water structures in 1940s to very flexible deep-water structures in recent years (Patel 1989). Nowadays, offshore structures are built in a water depth of more than 1000 feet. These modern structures are large, flexible, and equipped with a helicopter pad, drilling derrick, cranes, offices, and accommodations. They are typically subjected to severe loads due to winds, waves, and also currents. Further, their flexibility generates self-excited nonlinear hydrodynamic forces in addition to the nonlinear response due to large deformations (Chakrabarti 1987). Therefore, the risk of failure in these structures is not only higher than other structures, but also the possibility of local or major damage and considerable human discomfort due to vibrations are more likely. The safety of such structures against failure is of principal interest to many researchers (e.g. Karunakaran *et al.* 1993, Rajagopalan 1993).

The safety of structures can usually be ensured by increasing their stiffness so as to shift the natural frequencies away from the resonant range of frequencies. However, for large offshore structures, this approach is usually very costly, requiring excessive construction materials. An alternative solution is to implement passive and/or active control mechanisms to regulate the structural motion as desired. This approach is now of current concern to many researchers and there are several attempts exploring its application to offshore structures (Gosh and Meirovitch 1985, Reinhorn, Manolis and Wen 1987, Kawano 1993, etc.).

This paper presents certain control mechanisms which can be implemented to control steel jacket platforms. The control technique is demonstrated using a 300 ft steel jacket platform as an example. The wave-induced forces are estimated using the Morison equation which contains nonlinear drag and self-excited terms. The dynamic response of the structure in the presence of the control mechanism is compared with the uncontrolled response. Three control mechanisms

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are considered, namely a passive tuned mass damper (TMD), an active TMD, and an active tendon mechanism.

2. Wave forces

The horizontal wave force acting at joint P is modeled in the well-known Morison Eqs. (1)-(2). For uni-directional plane waves in the presence of wind-induced water currents, the equation is given as

$$F_p=\frac{1}{2}\rho C_D A_P |U'_{px}| U'_{px}+\rho C_I B_p a_{px}-\rho(C_I-1)B_p\ddot{U}_p \tag{1}$$

where U'_{px}=the relative water velocity$=U_{px}-\dot{U}_p$; U_{px}=the horizontal water velocity; $\dot{U}_p$=the horizontal joint velocity; A_p=the projected area at joint P; B_p the lumped volume at joint P; C_D=the drag coefficient; C_I=the inertia coefficient; a_{px}=the horizontal water acceleration at joint P; ρ=water density; and $\ddot{U}_p$=the acceleration of the joint.

There are various sources of inaccuracies in Eq. (1) arising, e.g., from the use of constants for C_D and C_I which are in fact frequency and the depth dependent. The equation considers the nonlinear self-excited and drag forces. However, designers sometimes use certain simplified linearized forms of the Morison equation, which provide linear time-invariant self-excited forces (Kawano 1993, Haritos and Karadeniz 1993, Gudmestad and Connor 1983). One such equation is given by

$$F_p=\frac{1}{2}\rho C_D A_p \hat{U}_{px} U_{px}+\rho C_I B_p a_{px}-\rho C_D A_p \hat{U}_{px}\dot{U}_p-\rho(C_I-1)B_p\ddot{U}_p \tag{2}$$

where $\hat{U}_{px}$ is a constant dependent on U_{px}.

The horizontal water velocity and acceleration depend on the wave field characteristics. For simplicity of presentation, consider a monochromatic wave as shown in Fig. 1. From the linear

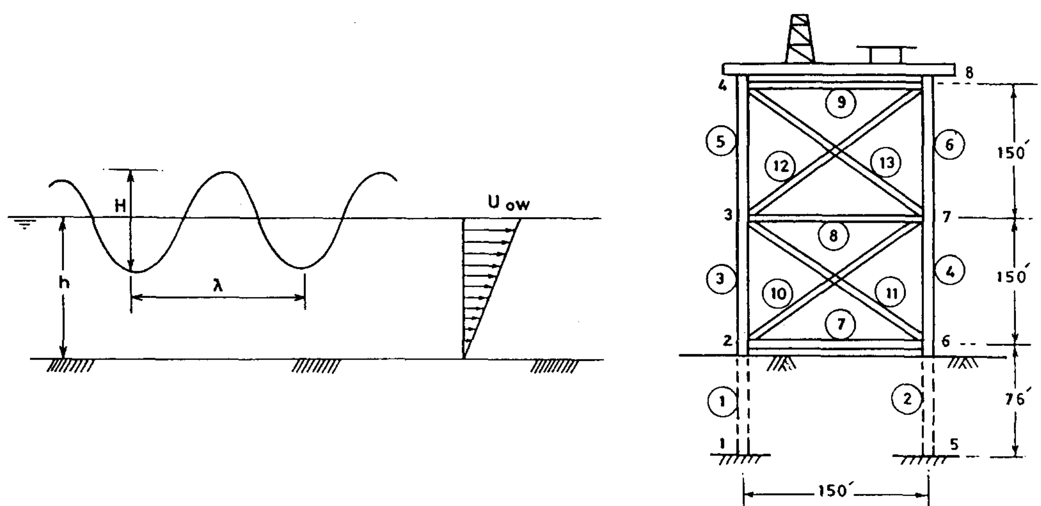

Fig. 1 Steel jacket platform dimensions

Table 1

Member	Outside Diam. (ft)	Inside Diam. (ft)	A (ft^2)	I (ft^4)
Vertical	4	3.75	1.5217	2.85
Horizontal	2	1.92	0.2463	0.11832
Diagonal	2	1.92	0.2463	0.11832

Table 2

Member	Area (ft^2)	Volume (ft^3)	Mass (Slug)
1	–	–	–
2	–	–	–
3	600	1884.956	6720.657
4	600	1884.956	6720.657
5	400	1256.637	5924.53
6	400	1256.637	5924.53
7	–	–	1418.42
8	–	–	1418.42
9	–	–	554.175
10	300	471.239	2005.95
11	300	471.239	2005.95
12	200	314.159	1598.54
13	200	314.159	1598.54

wave theory (Patel 1989, and Chakrabarti 1987), the corresponding horizontal water particle velocity and acceleration at joint P are defined as

$$U_{px}=E_p\cos(k\,x_p-\Omega t)+U_{ow}\left(\frac{y_p}{h}\right) \tag{3}$$

$$a_{px}=-E_p\,\Omega\sin(k\,x_p-\Omega t) \tag{4}$$

$$E_p=\frac{\Omega H}{2}\,\frac{\cosh k\,y_p}{\sinh k\,h} \tag{5}$$

where y_p=the elevation of joint P from the sea bed; h=the water depth; Ω=the wave frequency; H=wave height; k=the wave number=$(2\pi/\lambda)$; λ=wave length; and U_{ow}=the current velocity at the water surface, usually taken as 1% of the wind speed at a height of 33 ft above the water surface. It is clear from Eqs. (1)-(5) that the wave forces depend on the wave parameters Ω, λ, H, the local wind speed, the drag and inertia coefficients of the members, the water depth, and location (x_p, y_p) of the joint with respect to a fixed reference.

3. Example

The simple steel jacket platform shown in Fig. 1 is used as an illustrative example for which H=40 ft, h=250 ft, λ=600 ft and U_{ow}=0.4 ft/sec. The structure consists of cylindrical steel tube members with the dimensions shown in Table 1. The density of steel is taken as 15 slug/ft^3,

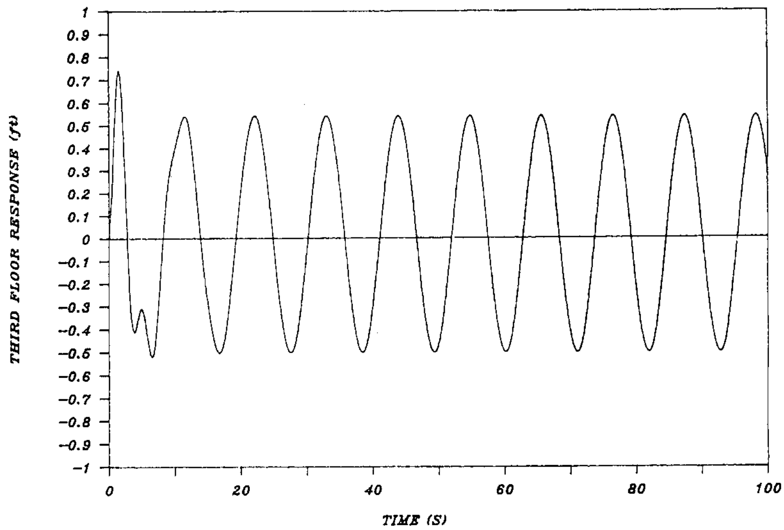

Fig. 2 Lateral response considering linearized Morison equation

the water density as 1.99 slugs/ft^3, and the steel modulus of elasticity as $E=30\times10^6$ lb/in^2. The weight of the concrete deck carried by the steel members is 1500 kips. The projected areas, volumes, and masses of each member in the structure are given in Table 2. From these data the wave force parameters at each joint can be calculated.

The natural frequencies and mode shapes for the undamped free vibration system can be determined using any structural dynamic software. Although one may consider as many modes as possible, for simplicity of presentation, only the first two modes shall be considered here. The natural frequencies of the first two modes are $\omega_1=1.818$ rps and $\omega_2=10.87$ rps. The equations of motion of the structure can be expressed in a matrix form as

$$\underline{\ddot{A}}+\underline{C}\underline{\dot{A}}+\underline{\Lambda}\underline{A}=\underline{\phi}^T\underline{F} \tag{6}$$

where C=diagonal damping matrix, assumed to provide 0.5% damping ratio in each mode; $\underline{\phi}$=normalized model matrix; $\underline{\Lambda}$=diagonal stiffness matrix, containing the squared natural frequencies of the modes; $\underline{A}$=generalized coordinate vector; and $\underline{F}$=wave force vector.

The horizontal response of the deck calculated using the linearized equation, Eq. (2), at a wave frequency $\Omega=0.577$ rps is shown in Fig. 2. The response at the same wave frequency using the Morison equation, Eq. (1), is shown in Fig. 3. It is clear that there is a difference in the response depending on the wave force model. The nonlinear self-excited hydrodynamic forces cause a larger amplitude and more oscillation cycles than the linearized forces do. For this reason, the rest of the paper will use the nonlinear form of the Morison equation Eq. (1). Figs. 4 and 5 show, respectively, the influence of increasing damping and stiffness on the nonlinear response of the structure. It is apparent that the amplitudes of vibration and the cycles of oscillations have been decreased. On this basis, the vibration control mechanism is designed to introduce more damping and stiffness to the structure.

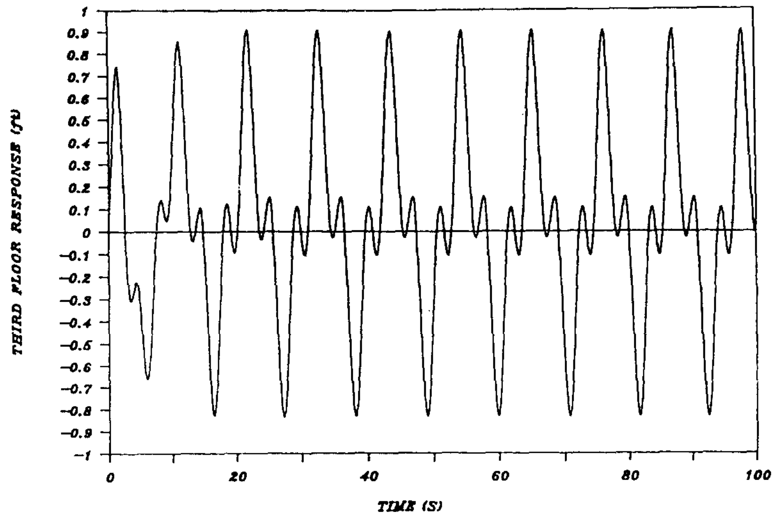

Fig. 3 Lateral response considering nonlinear Morison equation

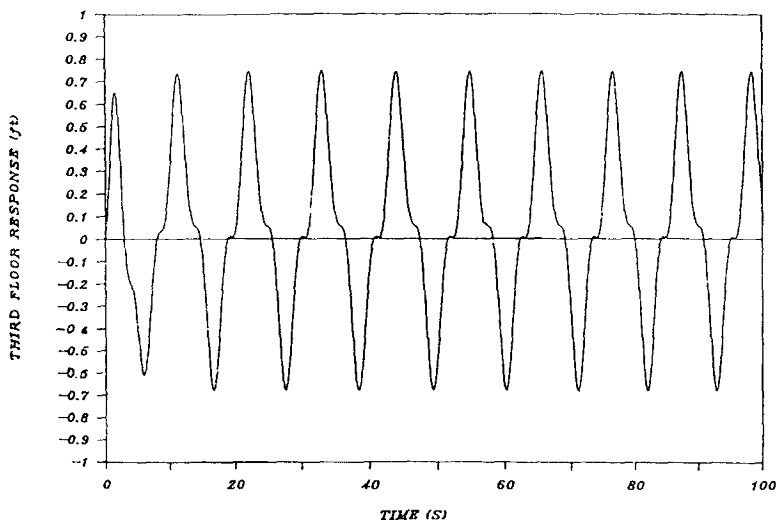

Fig. 4 Effect of increasing the damping ratio to 10% on the nonlinear response

4. Controlled response using passive TMD

A tuned mass damper (TMD) is a device consisting of a small mass, m_T, a spring with constant K_T, and a viscous damper with the coefficient C_T. By connecting this mass to joint 8 of the structure as shown in Fig. 6, the equations of motion of the coupled system become:

$$\underline{M}\,\underline{\ddot{U}}+\underline{C}\,\underline{\dot{U}}+\underline{K}\,\underline{U}=\underline{F}-\underline{F}_{TMD} \tag{7}$$

$$m_T\ddot{x}+C_T(\dot{x}-\dot{U}_8)+K_T(x-U_8)=0 \tag{8}$$

where $\underline{F}_{TMD}$ represents the interacting force vector due to the passive TMD. It is a zero vector

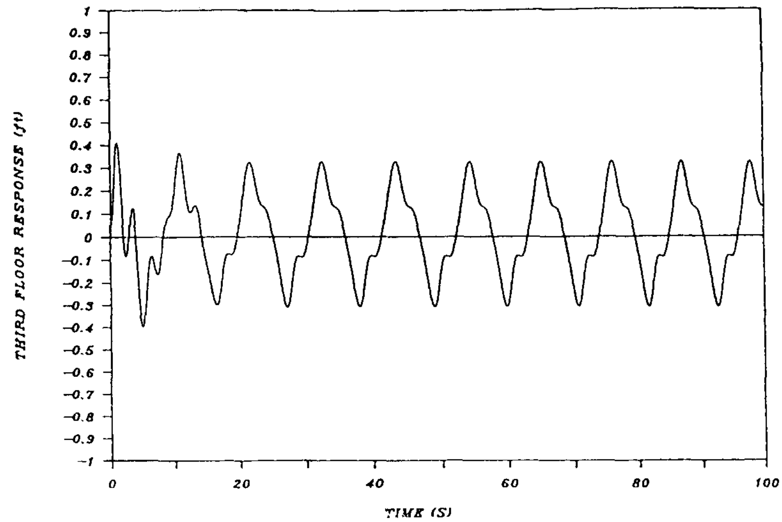

Fig. 5 Effect of increasing the stiffness 100% on the nonlinear response

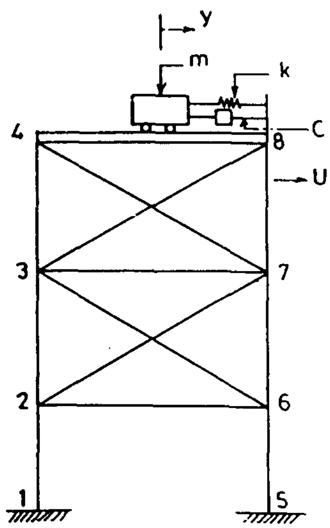

Fig. 6 Passive tuned mass damper mechanism

except at node $8 = C_T(\dot{x} - \dot{U}_8) + K_T(x - U_8)$; U_8 is the displacement of joint 8; $\underline{M}$, $\underline{C}$, and $\underline{K}$ are the structural mass, damping, and stiffness matrices; and x is the TMD response.

The tuned mass damper parameters are selected such that the damper natural frequency is close to 0.98 times the first mode natural frequency of the structure, and the damping ratio ξ_T is 15%. In order to determine the optimal value of the ratio between TMD mass and the frist modal mass, denoted by ρ, the locus of the eigenvalues for various values of ρ is plotted as shown in Fig. 7. From this locus, the damping ratio introduced into the structure can be determined, as shown in Fig. 8. It is obvious that taking a mass ratio $\rho \geq 0.02$ provides the structure with the highest damping. The nonlinear response using passive TMD with the mass ratio $\rho = 0.02$ is shown in Fig. 9. It is apparent that the structural response has decreased slightly

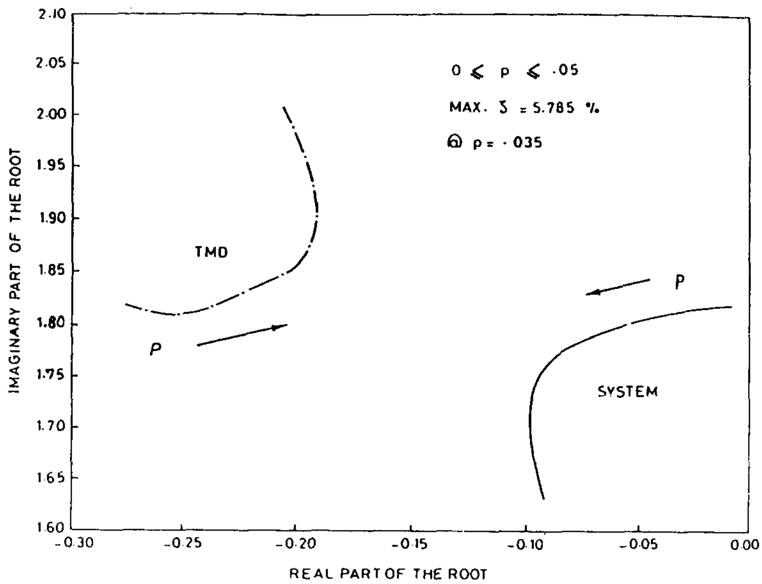

Fig. 7 Effect of mass ratio on the eigenvalues

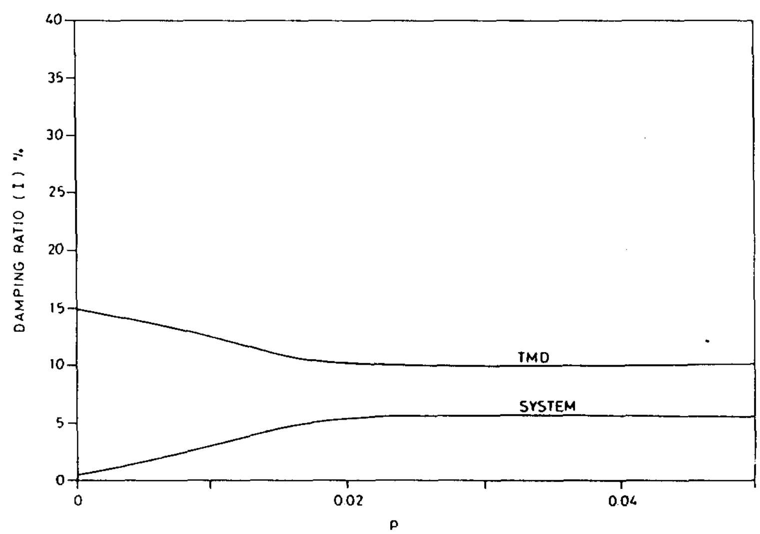

Fig. 8 Effect of mass ratio on the damping ratio

as compared with Fig. 4. However, if one assumes a wave frequency of Ω=1.8 rps, which is near the first mode natural frequency, it turns out that the controlled response, shown in Fig. 10, is much smaller than the uncontrolled response shown in Fig. 11. The reason is that at low wave frequency, and in the absence of any stiffness increase, the increase in damping does not affect the structural response significantly.

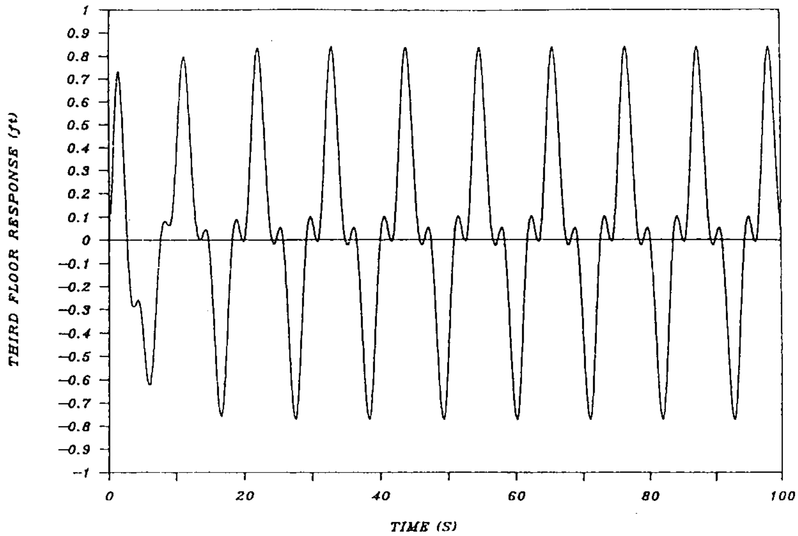

Fig. 9 Passive TMD controlled nonlinear response at $\Omega=0.577$

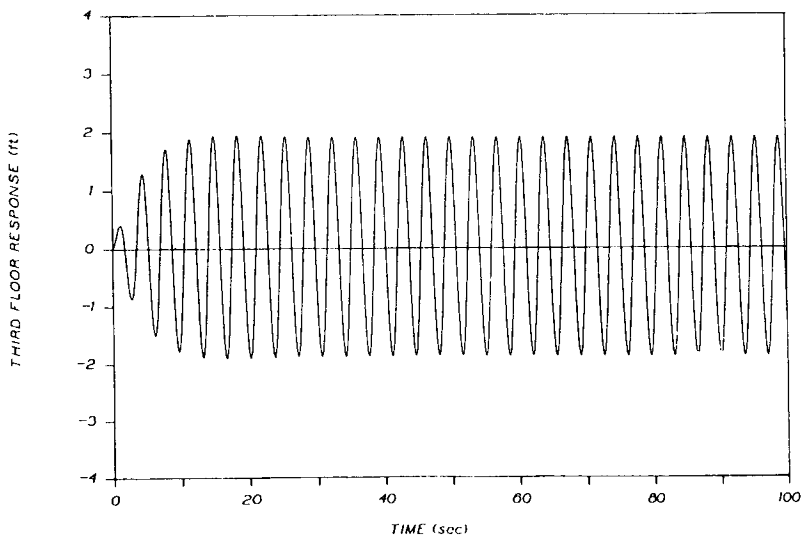

Fig. 10 Passive TMD controlled nonlinear response at $\Omega=1.80$

5. Controlled response using active TMD

An active tuned mass damper is a TMD connected with a hydraulic servomechanism. The motion of the damper is influenced by the motion of the structure (passive motion), and the operation of the hydraulic servo. The servo should be designed to regulate the motion of the damper according to a certain control law. The design of a control law for nonlinear systems is difficult. The usual practice is to design a control law for a linear or a linearized model and to test the effect of this design on the response of the nonlinear system. Several methods exist for the design of control laws (Leipholz and Abdel-Rohman 1986). The pole assignment

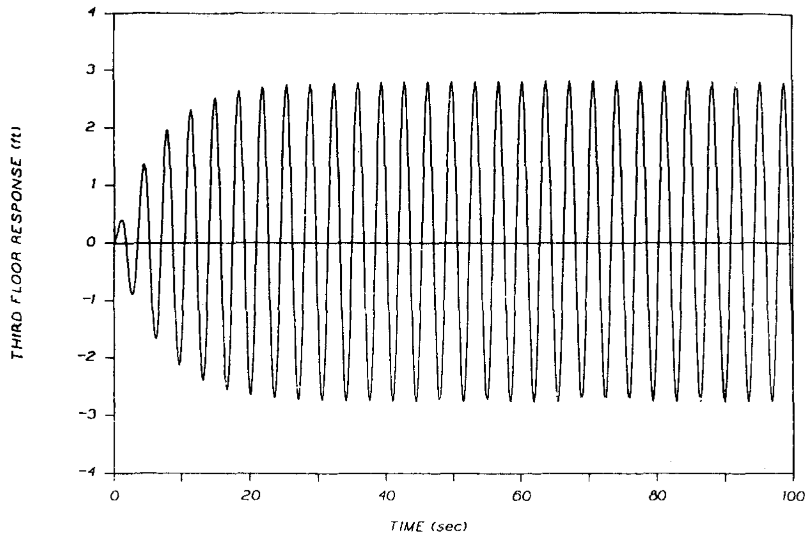

Fig. 11 Uncontrolled nonlinear response at $\Omega=1.68$

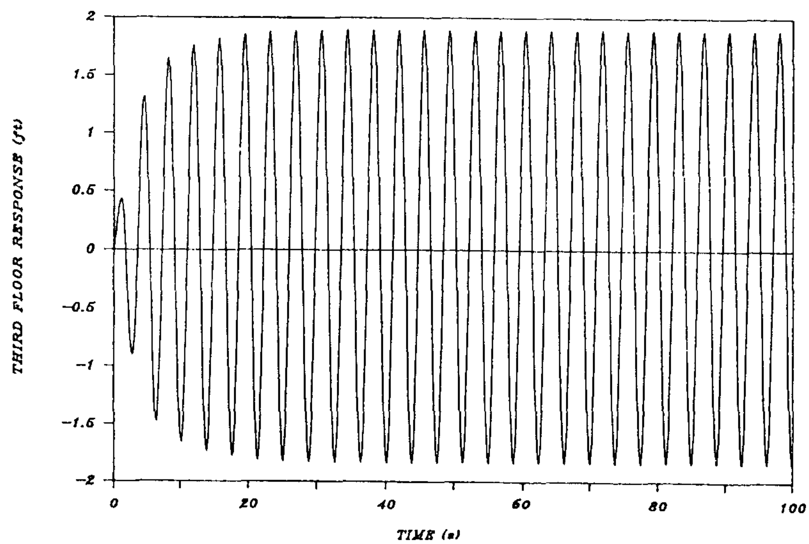

Fig. 12 Passive TMD controlled nonlinear response at $\Omega=1.68$

method (Abdel-Rohman and Leipholz 1978) is considered as a simple method in the presence of one active control force, as in the present case. The equations of motion considering the first two modes are

$$\ddot{A}_1+2\xi_1\omega_1\dot{A}_1+\omega_1^2A_1=\phi_1^T[\underline{F}-\underline{F}_{TMD}-\underline{F}_a] \tag{9}$$

$$\ddot{A}_2+2\xi_2\omega_2\dot{A}_2+\omega_2^2A_2=\phi_2^T[\underline{F}-\underline{F}_{TMD}-\underline{F}_a] \tag{10}$$

$$\ddot{x}+2\xi_T\omega_T(\dot{x}-\dot{U}_8)+\omega_T^2(x-U_8)=\frac{F_a}{m_T} \tag{11}$$

in which $\omega_1=1.818$ rps is the first natural frequency; $\omega_2=10.87$ rps is the second mode natural

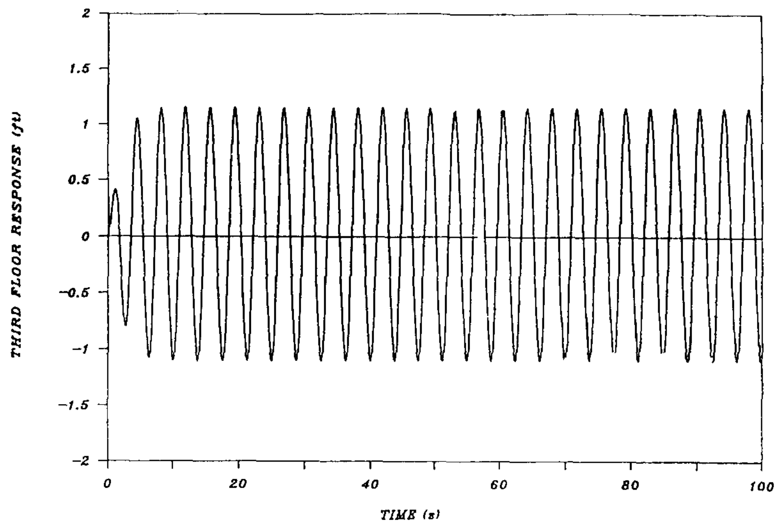

Fig. 13 Active TMD controlled nonlinear response at Ω=1.68 (Design 1)

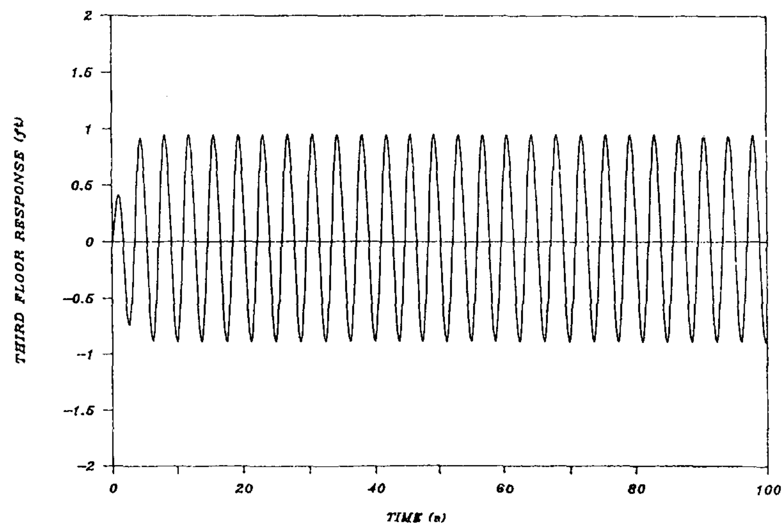

Fig. 14 Active TMD controlled nonlinear response at Ω=1.68 (Design 2)

frequency; ω_T=TMD natural frequency, $U_8=\phi_{1,8}A_1+\phi_{2,8}A_2$; $\phi_{i,8}=i^{th}$ mode shape at joint 8, and $\underline{F}_a$=the active control force vector which contains a force applied at joint 8 expressed as

$$F_a=\underline{K}\,[A_1\ \dot{A}_1\ A_2\ \dot{A}_2\ x\ \dot{x}]^T \tag{12}$$

and $\underline{K}$ is a gain matrix that needs to be determined.

To determine $\underline{K}$, one must specify the desired eigenvalues for the free vibration of the controlled system. Two different designs have been considered here. The first design assumes the eigenvalues of the controlled structure to be at the poles $(-0.438\pm j\ 1.6956)$, $(-0.05\pm\ 10.87)$, and $(-0.5\pm\ 1.9356)$. This design introduces a 25% damping ratio in the first vibrational mode. The second design corresponds to assuming eigenvalues to introduce a 40% damping in the first mode. The compari-

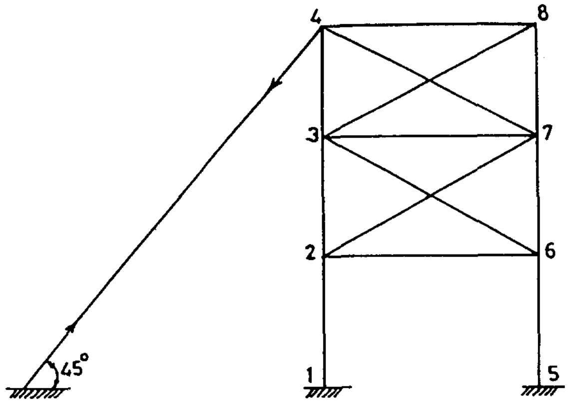

Fig. 15 A tendon control mechanism

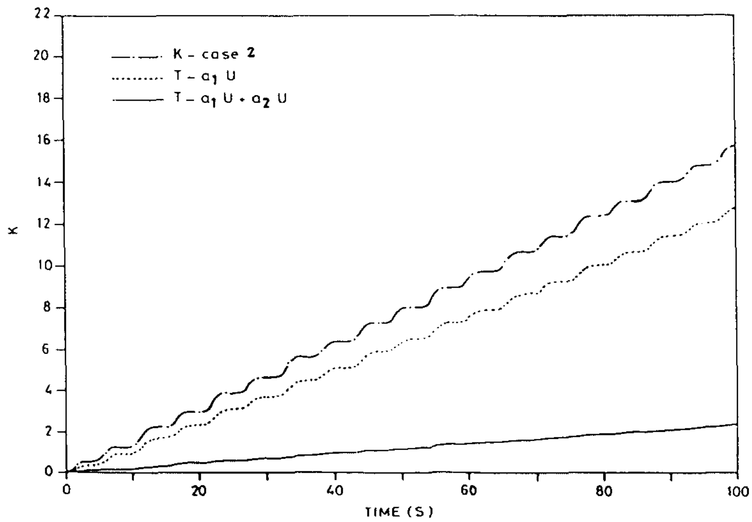

Fig. 16 Comparison of deflection indices of the controlled responses

sons between the controlled nonlinear response using the passive and the two active cases for a wave frequency of Ω=1.68 rps are shown respectively in Figs. 12-14. It is seen that as the wave frequency Ω approaches the first mode natural frequency, the increase in damping causes a decrease in the amplitude of vibration.

6. Controlled response using active tendon

A mooring line can be used as an active tendon mechanism as shown in Fig. 15. The equations of motion of the controlled structure reads as

$$\underline{M}\,\ddot{\underline{U}}+\underline{C}\,\dot{\underline{U}}+\underline{K}\,\underline{U}=\underline{F}-\underline{F}_a \tag{13}$$

in which $\underline{F}_a$ is the active control force vector which contains a force at the location of the tendon (in this case joint 4). In the present case, the active control force is designed to be

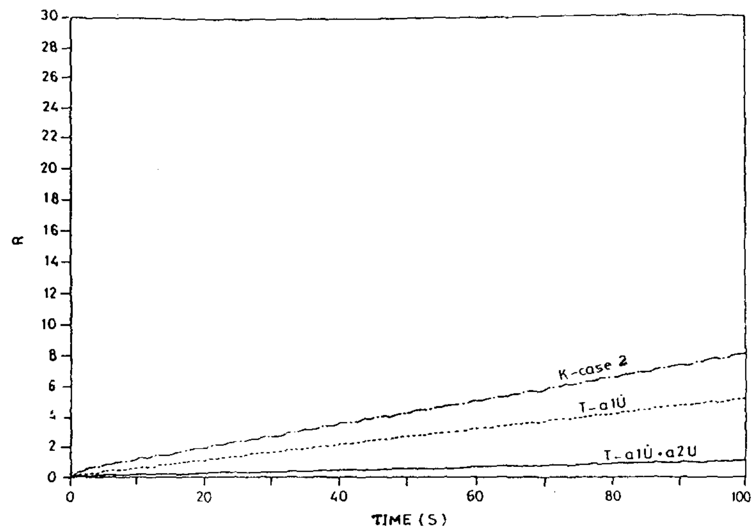

Fig. 17 Comparison of velocity indices of the controlled responses.

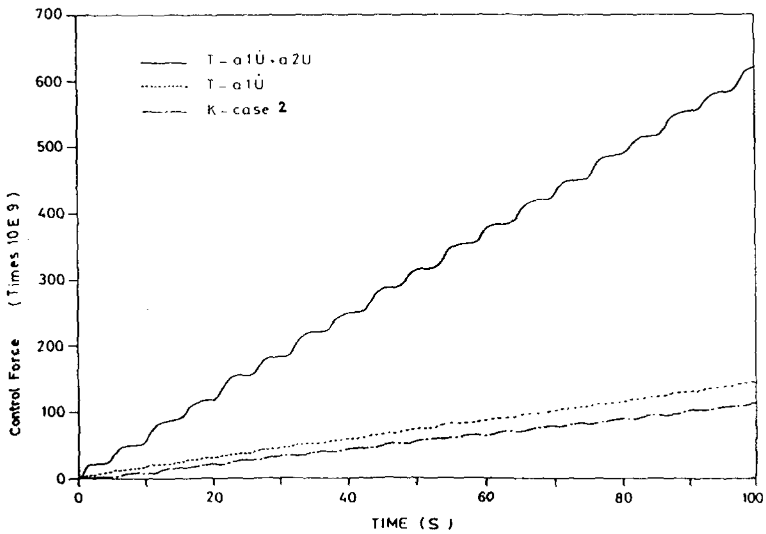

Fig. 18 Comparison of required active control force indices

proportional to lateral displacement and velocity of joint 4, in the form of

$$F_a = \alpha_1 \dot{u}_4 + \alpha_2 u_4 \tag{14}$$

in which α_1 and α_2 are gain values.

The gains values α_1 and α_2 are determined after specifying a damping ratio and stiffness for the first vibration mode. In order to introduce a damping ratio of 40% and to increase the natural frequency of the first vibrational mode, ω_1, to 2.7 rps, the values of α_1 and α_2 are determined as $\alpha_1=254780$ and $\alpha_2=486574$. The comparisons between using active TMD and active tendons in terms of displacement, velocity, and control force indices are, respectively, shown in Figs. 16-18. The deflection, velocity, and control force indices are respectively, defined as follows:

$$I_D=\int_0^T u_4^2\ dt \tag{15}$$

$$I_V=\int_0^T \dot{u}_4^2\ dt \tag{16}$$

$$I_U=\int_0^T F_a^2\ dt \tag{17}$$

From Figs. 16 and 17, it is seen that the displacement and velocity responses are reduced significantly because of the increased amount of the dissipated energy as a direct result of using the active control mechanism.

7. Conclusions

The evaluation of hydrodynamic wave forces in offshore structures involves inaccuracies due to modeling assumptions and the nature of various parameters. In order to improve the safety and serviceability of such structures one can implement various control mechanisms to decrease the lateral vibrations. The present paper has demonstrated the implementation of a passive tuned mass damper, an active tuned mass damper, and an active tendon as possible control mechanisms in offshore structures. It has been shown that the controlled response of the structure can significantly be reduced by using an active control and a suitable control mechanism.

It has also been shown that at low wave frequency excitation, the decrease in the response is achieved only by increasing the stiffness via active control force. An active tuned mass damper does not provide sufficient structural stiffness and thus can only be used to control the response at wave frequencies near to the primary natural frequency of the structure.

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