PASSIVE VIBRATION CONTROL FOR FATIGUE DAMAGE MITIGATION IN STEEL JACKET PLATFORMS

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Abstract Considering the stress cycles in the joints and members due to wave induced forces on offshore platforms, fatigue analysis is therefore one of the most important analyses in the offshore platforms design. Although most of the steel jacket type platforms are designed and located in areas with relatively high ratios of operational sea-states, for maximum environmental events, would have acceptable safety margin in an in-place and seismic analyses; but for fatigue analyses it faces critical condition. Therefore it seems utilizing control mechanisms with the aim of increasing fatigue life in such platforms will be more preferable, to merely deck displacement control. Investigation of tuned mass damper parameters optimality for vibration control of wave excited systems, implies that optimum tuning and damping ratios are strongly dependent on sea-state, in addition to system parameters. The efficiency of optimally designed tuned mass damper for fatigue damage mitigation in real steel jacket platforms has been evaluated using full stochastic spectral analysis method and the results have shown a great performance for TMDs in this application. Sea-state conditions are predictable through weather forecasting methods and this can be benefited also to increase the efficiency of TMDs with variable tuning parameters which have been optimally adjusted for each sea-state. In a case study, utilization of this auxiliary device resulted in 26.6 % reduction in maximum fatigue damage. This efficiency can be increased to 36.7 % using different tuning parameters for each sea-state.

Keywords Steel Jacket Platforms, Tuned Mass Damper, Fatigue Damage, Vibration Control

چکیده در حوزه های دریایی با لرزه خیزی پایین که در معرض طوفان های شدید قرار نداشته باشند، بسیاری از سکوها در برابر زلزله و بارهای ناشی از امواج شدید در حاشیه اطمینان مناسبی قرار دارند درحالی که نتایج تحلیل خستگی بویژه در طراحی اتصالات تعیین کننده است. بنابراین استفاده از سیستم کنترلی به منظور افزایش عمر خستگی در این سکوها نسبت به کنترل ارتعاشات صرفا" به منظور محدود کردن دامنه جابجایی ها، از اهمیت بیشتری برخوردار می‌باشد. بررسی عملکرد میراگر جرمی برای کنترل ارتعاشات سازه تحت اثر بارهای ناشی از امواج نشان می‌دهد که پارامترهای بهینه برای میراگر جرمی، شدیدا" تابع شرایط جوی و پارامترهای مشخص کننده طیف موج ناشی از آن می‌باشند. در این مطالعه کارآیی میراگر جرمی بهینه برای کاهش آسیب خستگی در سیستمی با مشخصات ارتعاشی یک سکوی واقعی بررسی شده است. عمر خستگی اتصالات به عنوان تابع هدف از روش تحلیل طیفی محاسبه شده است. همچنین از قابلیت پیشبینی شرایط جوی دریا برای طراحی مکانیزم کنترلی با پارامترهای بهینه تنظیم شده برای هر شرایط جوی خاص استفاده شده و استفاده از این ویژگی امواج دریا سبب افزایش کارآیی میراگر جرمی در مقایسه با کاربرد آن برای کنترل تحریکات زمین لرزه ای شده است. در مدل تحلیلی از یک سکوی واقعی استفاده از میراگر جرمی بهینه منجر به ۲۶/۶٪ کاهش در بیشترین آسیب خستگی شده است که می‌توان بازدهی آن را با استفاده از پارامترهای بهینه برای هر شرایط جوی تا ۳۶/۷٪ افزایش داد.

1. INTRODUCTION

In recent years various types of control plans have widely been studied and some have even been utilized to improve the behavior of structures under natural hazardous conditions to avoid large forces and/or deflections in excited structures. However a comprehensive review of these efforts which has been reported by Spencer, et al [1] shows that these researches have been focused on the protection of tall buildings and long span bridges against seismic or wind exsertions, and by comparison only a very small part of them are related to fixed offshore platforms.

The first study on this topic have been published by Abdel-Rohman, et al [2], who investigated the efficiency of some active and passive control mechanisms to moderate the dynamic responses of a steel jacket platform due to wave-induced forces. Terro, et al [3] employed active tuned mass damper with velocity feedback as a closed loop control scheme to minimize deck displacements in a sample steel jacket platform. Suhardjo, et al [4] tried some active control mechanisms such as active mass damper and active tendon singly and in combination with tuned mass. Subsequently Li, et al [5] made some improvements in their methodology.

Zribi, et al [6] used Lyapanov theory and a state feedback robust control to design an active tuned mass damper.

Hui Ma, et al [7] considered AMD mechanism and designed control law with feedforward and feedback optimal control algorithm to reduce the displacement and velocity responses of steel jacket offshore platform.

All aforesaid studies have taken active control methods into consideration. In passive control approach Hsien Hua Lee [8] utilized viscoelastic dampers as bracings to improve the dynamic performance of an offshore platform and Patil, et al [9] compared the efficiency of viscoelastic, viscous and friction dampers as energy dissipating devices to reduce the dynamic response of steel jacket platform to sea wave excitations.

Foregoing studies aimed to provide active or passive control systems to reduce the deck displacement or velocity of jacket type offshore platforms, but according to the experiences in analysis and designs of offshore steel jacket platforms, fatigue damage is the most important criterion in joints design and although most of the steel jacket type platforms located in low hazardous seismic zones fall in acceptable safety margin in in-place and seismic analyses; but in fatigue analysis they face critical condition. Therefore it seems that utilizing control mechanisms with the aim of increasing fatigue life in such platforms will be more preferable to merely deck displacement and acceleration control.

Moreover, considering the excessive cost of underwater fabrication and welding; using a control mechanism to improve the fatigue life of existing offshore platforms, which are overloaded with extra piping and equipments would be an attractive idea.

Since the control mechanism used for fatigue damage mitigation shall be interacting with the main structure, during a considerable part of platform's life span; therefore implementation of an active control technique with a permanent power source would not be practical in such cases, and instead utilizing some of passive or semi-active control mechanisms that can be installed on existing platforms are often preferable.

Aprile, et al [10] propounded the usage of passive control techniques to mitigate the fatigue damages, and investigated the efficiency of viscoelastic dampers for prolonging the fatigue life of an aluminum lattice tower subjected to stochastic wind loading. Li, et al [11] proposed a theoretical framework for TMD design to minimize the displacements of a single degree of freedom system under sea wave excitations [12].

In the current article optimality of TMD adjustable parameters have been studied to minimize the fatigue damages in realistic jacket type platforms. Considering that these optimal values are dependent on sea-state condition, variable TMD has been proposed for this application and its efficiency has been compared with constant TMD.

2. SCOPE OF THE CURRENT STUDY

Sea waves in long term sense can be considered as non-stationary random process which can be decomposed into short term stationary random processes. On the other hand the assumption of stationarity limits the validity of spectral analysis approach to short time periods. For this reason in spectral fatigue analysis of marine structures a set of power spectral density functions with different probability of occurrence for each one is used in the shape of wave scatter diagram. Each one of these power spectral density functions is related to a specific sea-state.

Obviously an auxiliary control system which is designed to minimize the fatigue damages due to a specific wave height, power spectral density function will not necessarily be the optimum one, considering different spectrums in the wave scatter diagram, and a passive control system with constant parameters can not supply its optimality for all sea-states. Therefore it is expected that using a passive control mechanism with variable parameters which are optimally designed for each sea-state will improve its performance in the fatigue damage degradation.

Although earthquake excitations have unpredictable nature in duration and intensity, yet sea waves loading can be predicted based on climatologic studies. Based upon climatological parameters such as wind speed and wind duration, prediction models for sea-state forecast are available in texts. Also, wave statistic parameters such as significant wave height H_s and mean zero crossing wave period T_z can be estimated for each sea-state [13].

In the current study the efficiency of optimally designed tuned mass damper, for fatigue damage mitigation in offshore steel jacket platforms has been evaluated, and it is demonstrated that using the optimal mass for each sea-state will cause the improvement in control system performance.

3. FATIGUE DAMAGE EVALUATION

In many areas with relatively high ratios of operational sea-states to extreme environmental events, fatigue is a major design consideration. Although the spectral analysis approach is more complicated in comparison with deterministic method, it yields more realistic and reliable results and therefore it is recommended in most offshore engineering standards. Linear time invariant response of the structure and zero mean stationary Gaussian random excitation are the principal assumptions in this approach. Based on the first assumption the response of the structure to stationary Gaussian random waves can be characterized by the following expression in the frequency domain:

$$Y(\omega) = T(\omega)X(\omega) \tag{1}$$

Where $Y(\omega)$, $X(\omega)$ are Fourier transforms of the response and excitation respectively and $T(\omega)$ is the frequency response function which defines the ratio of the structural response to the environmental excitation. In the spectral fatigue analysis $T_{sp}(\omega)$ is defined as transfer function from wave height to hot spot stress which is the stress in the immediate vicinity of the discontinuities on tubular connections and can be obtained by increasing the nominal stress by a stress concentration factor.

Considering the wave height as a zero mean stationary random process with power spectral density function $S_p(\omega)$, the PSD function of the hot spot stress $S_s(\omega)$ can be obtained as:

$$S_s(\omega) = \left|T_{sp}(\omega)\right|^2 S_p(\omega) \tag{2}$$

As it is shown in standard references [14], the variance of a zero mean stationary random process can be computed by integrating its PSD function then:

$$\sigma_{RMS}^2 = \int_0^\infty \left|T_{sp}(\omega)\right|^2 S_p(\omega)d\omega \tag{3}$$

Also the mean zero up crossing period for the same stationary Gaussian process can be calculated as follow:

$$T_Z = \frac{\sigma_{RMS}}{\sqrt{\int_0^\infty \omega^2 \left|T_{sp}(\omega)\right|^2 S_p(\omega)d\omega}} \tag{4}$$

This can be interpreted as the average period of a narrow band process; hence the expected number of cycles during the a fraction of design fatigue life L is equal to:

$$n = \frac{aL}{T_z} \tag{5}$$

Linear systems excited by Gaussian random process will show a Gaussian response whose spectrum is confined to a narrow band of frequencies in the vicinity of the resonance frequency and for such a narrow band Gaussian random process the probability distribution of peak values complies with the Rayleigh distribution then the probability density function of the stress range is equal to:

$$p(s) = \frac{s}{\sigma_{RMS}^2} e^{-\frac{s^2}{2\sigma_{RMS}^2}} \qquad (6)$$

and the expected number of stress cycles whose amplitudes are between s and s + ds can be estimated as:

$$n(s) = \frac{aL}{T_z} p(s)ds \qquad (7)$$

If $N(s)$ is the number of cycles for which the given stress range, s would be allowed by the appropriate S-N curve, the fatigue damage caused by those cycles of stresses is equal to:

$$d(s) = \frac{n(s)}{N(s)} \qquad (8)$$

According to the Palmgren-Miner theory for fatigue damage accumulation in the materials subjected to different stress ranges, total fatigue damage due to a sea-state with a specific PSD function can be computed as:

$$D = \sum_{s=0}^{\infty} d(s) = \frac{aL}{T_z \sigma_{RMS}^2} \int_0^{\infty} \frac{s}{N(s)} e^{-\frac{s^2}{2\sigma_{RMS}^2}} ds \qquad (9)$$

The mathematic form of S-N curve has been recommended in standards [15] as:

$$N(s) = c\left(\frac{s}{s_c}\right)^{-m} \qquad (10)$$

In which c is the permissible number of cycles for applied cyclic stress range s_c and m is termed as negative inverse slope of logarithmic S-N curve. Substituting in Equation 9 and integrating leads to:

$$D = 2^{\frac{m}{2}} \sigma_{RMS}^m c^{-1} s_c^{-m} \frac{aL}{T_z} \Gamma\left(1 + \frac{m}{2}\right) \qquad (11)$$

This damage index is to be calculated for all sea-states in the wave scatter diagram and should be aggregated as per the Palmgren-Miner theory:

$$D = \sum_{seastates} D_i \qquad (12)$$

The resultant damage index has been considered as objective function in this study aiming to optimize the adjustable parameters of TMD to minimize the fatigue damage in tubular connections of jacket. To increase the efficiency of TMD variable tuning parameters have been tried for different sea-states.

Two standard forms for sea waves PSD often found in the literature are used here. Most popular wave spectrum which is especially suited for open sea areas is well known as Pierson-Moskowitz wave spectrum. The next PSD function which is developed from the North Sea wave data analysis and is advised for costal wind generated seas is JONSWAP wave spectrum [16].

4. TMD EQUIPPED SDOF SYSTEM UNDER WAVE EXCITATIONS

TMDs are simple passive control devices which have been installed on a large number of civil structures some have been listed by Holmes [17]. TMDs have been found reliable and effective at reducing vibration response of excited structures. They exhibit good performance to control a single mode of structural vibration that can be demonstrated with a single degree of freedom system. Therefore derivativation of closed form expressions for the optimum tuning and damping ratios which are available in literature has been based on TMD utilized SDOF systems equation of motion as shown in Figure 2.

$$\begin{bmatrix} 1+\mu & \mu \\ \mu & \mu \end{bmatrix}\begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} + \begin{bmatrix} 2\xi_x\omega_x & 0 \\ 0 & 2\xi_y\omega_y \end{bmatrix}\begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} +$$

$$\begin{bmatrix} 2\xi_x\omega_x & 0 \\ 0 & 2\xi_y\omega_y \end{bmatrix}\begin{Bmatrix} x \\ y \end{Bmatrix} = \begin{Bmatrix} f_x/m_x \\ 0 \end{Bmatrix}$$

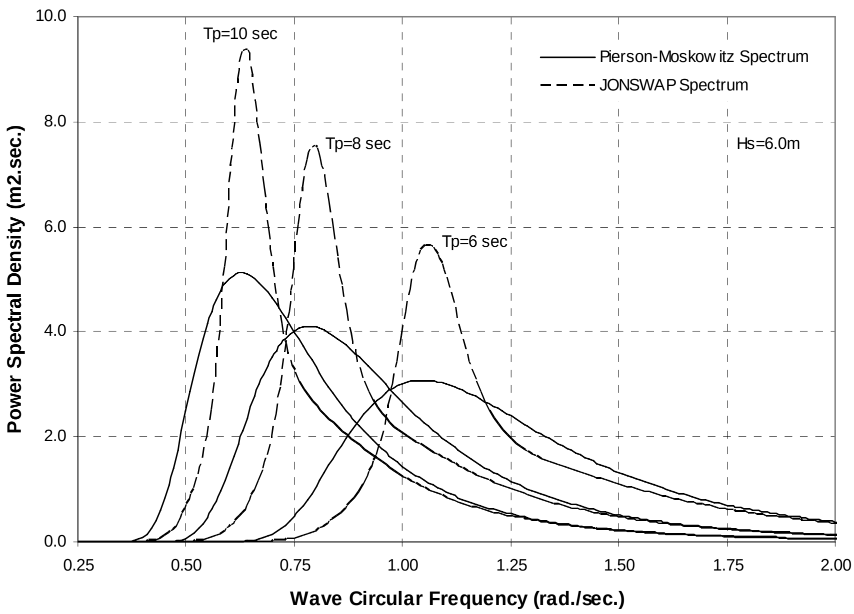

Figure 1. Comparison of two spectral formulations.

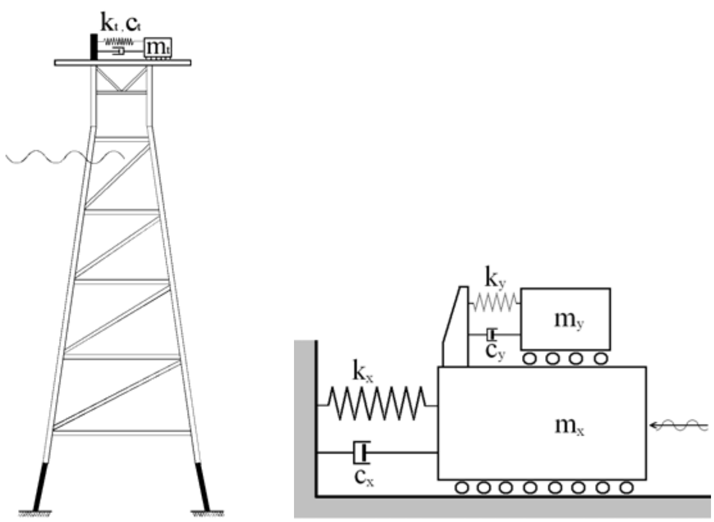

Figure 2. Steel jacket platform utilized with a TMD and equivalent SDOF system.

$$\mu = \frac{m_y}{m_x} \quad \xi_x = \frac{c_x}{2m_x\omega_x} \quad \xi_y = \frac{c_y}{2m_y\omega_y} \quad \omega_x = \sqrt{\frac{k_x}{m_x}}$$
$$\omega_y = \sqrt{\frac{k_y}{m_y}} \tag{13}$$

Where μ is the mass ratio between TMD and the SDOF system, ξ_x and ω_x are the damping ratio and the natural frequency for the SDOF respectively and ξ_y and ω_y are the same for the auxiliary control mechanism. Solution of this coupled set of ODE's for a harmonic external excitation $f_x(t) = f_0 e^{i\omega t}$ leads to the response function of the combined system in the frequency domain:

$$H_{xf}(\omega) = \frac{-B_2\omega^2 + B_1\omega i + B_0}{m_x[A_4\omega^4 - A_3\omega^3 i - A_2\omega^2 + A_1\omega i + A_0]} \quad (14)$$

$$A_0 = \mu\omega_x^2\omega_y^2$$

$$A_2 = \mu\omega_x^2 + \mu(1+\mu)\omega_y^2 + 4\mu\xi_x\xi_y\omega_x\omega_y$$

$$A_1 = 2\mu\omega_x\omega_y(\xi_x\omega_y + \xi_y\omega_x)$$

$$A_3 = 2\mu\xi_x\omega_x + 2\mu(1+\mu)\xi_y\omega_y$$

$$A_4 = \mu \qquad B_0 = \mu\omega_y^2 \qquad B_1 = 2\mu\xi_y\omega_y \qquad B_2 = \mu$$

The mean square value of the main mass displacement when it is subjected to a random excitation with power spectral density function $S_f(\omega)$ can be found as:

$$E[x^2] = \int_{-\infty}^{+\infty} \left|H_{xf}(\omega)\right|^2 S_f(\omega)d\omega \quad (15)$$

Assuming the external excitation force as a Gaussian white noise random process with an intensity of S_0, $E[x^2]$ can be evaluated as:

$$E[x^2] = \frac{\pi S_0}{m_x^2} \frac{B_0^2(A_2A_3 - A_1A_4) + A_0A_3(B_1^2 - 2B_0B_2) + A_0A_1B_2^2}{A_0A_1(A_2A_3 - A_1A_4) - A_0^2A_3^2} \quad (16)$$

The efficiency of TMDs for reduction of vibration amplitude was first investigated by Den Hartog,et al [18]. Warburton,et al [19] followed his procedure to find the optimum parameters for TMDs attached to undamped SDOF systems subjected to various types of excitations such as harmonic force, harmonic support motion or white noise random excitations. In the case of harmonic excitation, the object is to bring the resonant peak of the amplitude $H_{xf}(\omega)$ down to its lowest possible value which leads to optimum parameters as:

$$\omega_{y\ opt} = \omega_x \frac{1}{1+\mu} \qquad \xi_{y\ opt} = \sqrt{\frac{3\mu}{8(1+\mu)}} \quad (17)$$

In the case of white noise excitation, setting the derivatives of Equation 16 with respect to ω_y and ξ_y equal to zero, results in the optimal tuning and damping parameters for TMD device:

$$\omega_{y\ opt} = \omega_x \frac{\sqrt{1+\mu/2}}{1+\mu} \qquad \xi_{y\ opt} = \sqrt{\frac{\mu(1+3\mu/4)}{4(1+\mu)(1+\mu/2)}} \quad (18)$$

All above mentioned optima’s have been derived under simplifying assumptions for primary structure and imposed excitations so they are only dependent on the mass ratio while practical applications confront other types of response and excitation which leads to more complicated optimization problems that can be solved by numerical schemes. Offshore platform which is under sea waves excitation are one of these cases.

Using linear Airy theory for sea water waves, horizontal components of the resultant velocity and acceleration of water particles can be expressed in a complex form as:

$$\varsigma = \varsigma_a e^{(kx-\omega t)i} \quad (19)$$

$$u = \varsigma_a \omega \frac{\text{Cosh } k(h+z)}{\text{Sinh } kh} e^{(kx-\omega t)i} = \varsigma_a H_{u\zeta}(\omega) e^{(kx-\omega t)i} \quad (20)$$

$$\dot{u} = -\varsigma_a \omega^2 \frac{\text{Cosh } k(h+z)}{\text{Sinh } kh} e^{\left(kx-\omega t-\frac{\pi}{2}\right)i} = \varsigma_a H_{\dot{u}\zeta}(\omega) e^{\left(kx-\omega t-\frac{\pi}{2}\right)i} \quad (21)$$

In which k is the wave number (rad/m) and ω is the circular wave frequency (rad/s) and ς_a is the wave amplitude. Knowing the water particle velocity and acceleration in the vector notation, the wave induced distributed forces on a cylindrical member can be calculated by Morrison's equation [16]. Derivation of the force PSD function considering the nonlinear drag component in Morrison's equation is not possible. Using a stochastic linearization technique the Morison equation is thus approximated as [20]:

$$p(t) = \rho C_m V \dot{U}_n(t) + \frac{1}{2}\rho C_d A \sigma_u \sqrt{\frac{8}{\pi}} U_n(t) \qquad (22)$$

Where $p(t)$ is wave force per unit length of the member and C_d, C_m are drag and inertia coefficients respectively. V and A are displaced volume of the cylinder and projected area normal to the member axis both per unit length of the member and ρ is sea water mass density. $U_n(t)$ infers to the velocity of the fluid normal to the structural member. σ_u is the standard deviation of the velocity. Introducing Equations 22 and 23 in this expression results in:

$$p(t) = \varsigma_a H_{f\zeta}(\omega) e^{(kx - \omega t + \varphi)i} \qquad (23)$$

$$H_{f\zeta}(\omega) = \rho\omega \frac{\text{Cosh}\, k(h+z)}{\text{Sinh}\, kh} \sqrt{(C_m V)^2 \omega^2 + \frac{2}{\pi}(C_d A \sigma_u)^2} \qquad (24)$$

$$\varphi = A\tan\left(\frac{\sqrt{\pi} C_m V}{\sqrt{2} C_d A \sigma_u}\right) \qquad (25)$$

Thus, the variance of the displacement response can be found by using the transfer function of the wave load $H_{f\zeta}(\omega)$ and the wave spectrum by:

$$E[x^2] = \int_{-\infty}^{+\infty} \left| H_{xf}(\omega) H_{f\zeta}(\omega) \right|^2 S_\varsigma(\omega) d\omega \qquad (26)$$

The optimal tuning and damping parameters of the auxiliary device which minimize this equation are also dependent on the shape of the wave spectrum in addition to the dynamic characteristics of the primary structure and this is the major concern of this study in comparison with traditional and seismic applications of TMDs as per Equations 19 and 20. From the two common parameters used to describe wave spectrum, the mean zero up crossing period T_z evidently affects the optimal values of these parameters and this is apparent in Figure 3 which demonstrates the efficiency of a TMD for suppression of the vibration in a SDOF system excited with random waves described with P-M spectrum. The natural frequency of the corresponding primary system is $\omega_x = 2.0$ rad/s. Although using the TMD with $\mu > 0.1$ has positive performance in all sea-states; but in practical range of mass ratios the functionality of TMD is very sensitive to sea-state conditions.

Zero up-crossing periods of the exciting waves also affects the optimal tuning ratio of TMD as shown in Figure 4 but according to Figure 5 it does not have considerable effect on the optimal value of TMD damping ratio.

TMD also affects the mean zero crossing period of the displacement response as demonstrated in Figure 6, but due to the minor role of the zero crossing periods in Equation 11 TMDs optimality for fatigue damage mitigation adheres to the optimal values for displacement range suppression as shown in Figure 7.

It is well known that passive control systems are only operative for structures with remarkable dynamic response, whereas for low rise platforms the quasi static response contribution becomes dominant because their effective frequency band of dynamic response stands apart from the narrow frequency band of the wave induced forces PSD function as shown in Figure 8. Therefore optimally designed TMD to moderate the peak value of the response function of such a primary structure does not necessarily reduces the variance of the displacement response under sea waves excitation and the optimal tuning of TMD is dependent on the peak frequency of the exciting spectrum as is clear in Figure 9a.

Further overlapping of the frequency response function and the exciting forces spectrum, raises the efficiency of TMD as is clear in Figure 9b; however, the dependency of the optimal tuning on

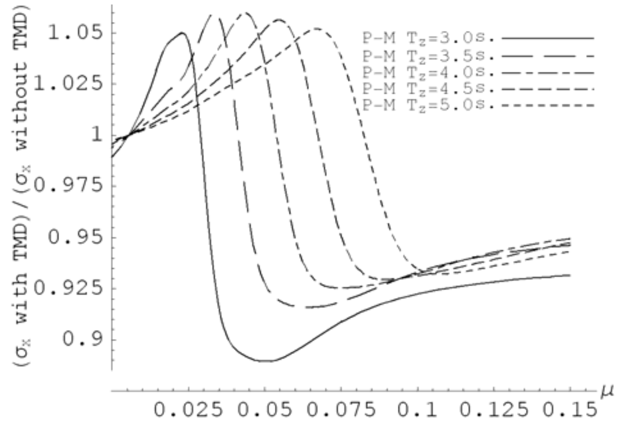

Figure 3. Suppression of the displacement range by a TMD with a constant stiffness $k_y = 0.02\ k_x$ and damping $\xi_y = 0.02$.

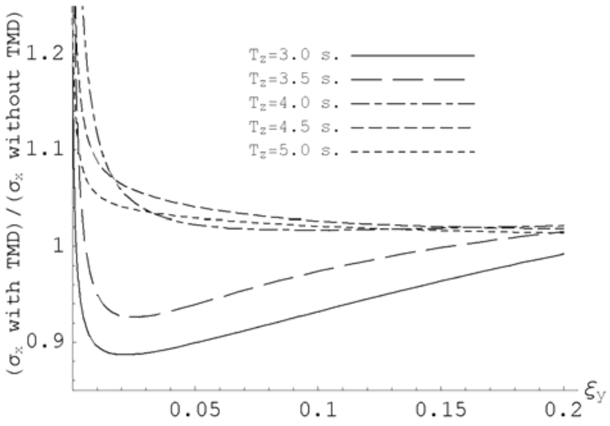

Figure 5. Suppression of the displacement range by a TMD with a constant mass ratio $\mu = 0.05$ and stiffness $k_y = 0.02\ k_x$.

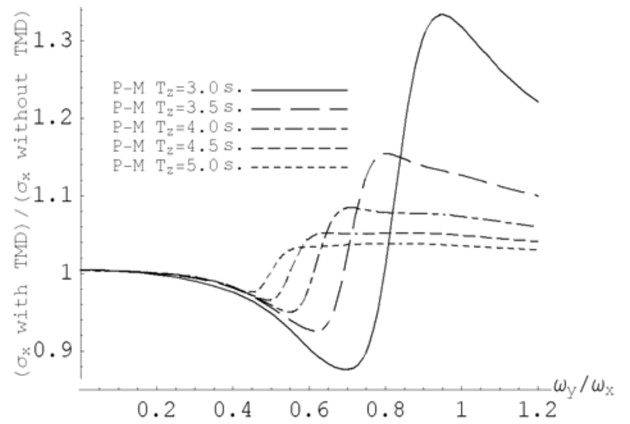

Figure 4. Suppression of the displacement range by a TMD with a constant mass ratio $\mu = 0.05$ and damping $\xi_y = 0.02$.

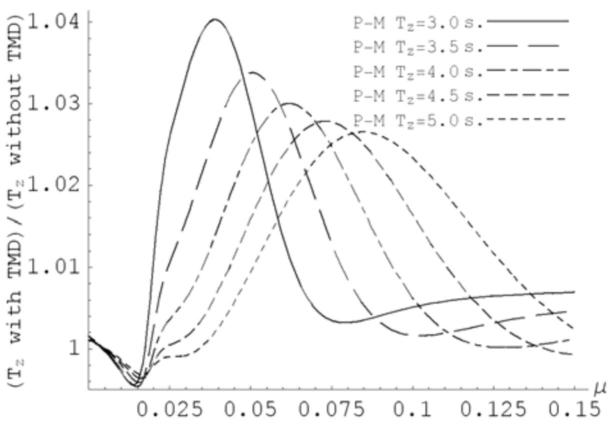

Figure 6. Variation in zero-crossing period of displacement response.

the peak frequency of the exciting forces spectrum persists up to the nearness of the natural frequency of the structure to the peak frequency of the exciting forces spectrum and this is discernible from Figure 9c. But in close vicinity of these two frequencies, suppression of the peak value of the response function finds more relevancy.

Putting aside some simplifying assumptions that have been intended in deriving the Equations 19 and 20, the optimal parameters for a TMD to minimize the displacement response of a SDOF under random excitations can be determined by numerical approaches. Figure 10 shows these optimal values in comparison with classic optima for a primary system with natural period of 3.5 seconds. It is deduced that consideration of damping for the primary structure under white noise excitation only reduces the optimal damping ξ_y of the auxiliary system and has not any remarkable effect on its efficiency and optimal tuning ratio. Yet quite different results have been observed for systems excited by wave induced random forces. As discussed before and is evident from Figure 10b, the optimal tuning ratio for such systems evidently complies with the peak frequency of the exciting forces spectrum unless the natural frequency of the primary structure fall foul of the relatively high intensity region of the

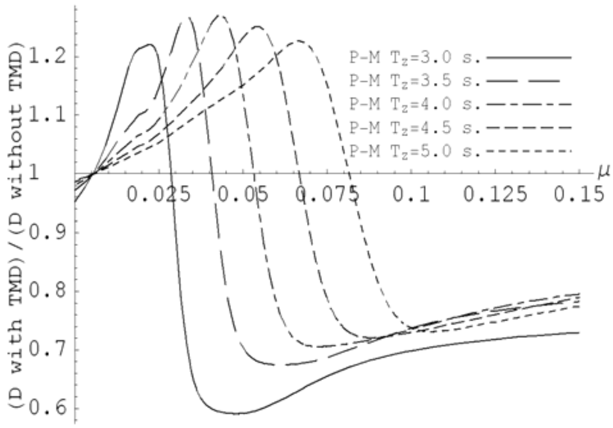

Figure 7. Fatigue damage mitigation by a TMD with a constant stiffness $k_y = 0.02\ k_x$ and damping $\xi_y = 0.02$.

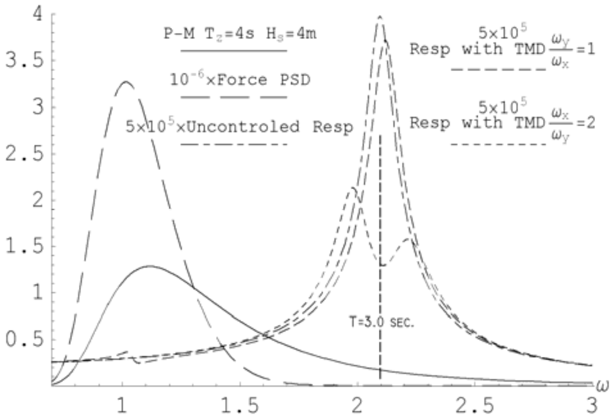

Figure 8. Wave height and resultant forces PSD functions and controlled and uncontrolled SDOF systems' frequency response functions.

exciting spectrum. In such a case, the optimum tuning ratio tends towards the classical optimum values pursuant to Equation 18. This occurs in Figure 10b for the system excited by random waves with zero crossing periods of 3.0 seconds. Similar results have been observed in Figure 10c for the optimum damping ratios for the auxiliary device in different sea-states. For practical values of mass ratio, only when the peak frequency of the exciting forces spectrum approaches the natural frequency of the primary structure, the optimum value of the damping ratio complies with the classical optima.

According to the foregoing discussions the optimum TMD parameters to moderate the displacement response of fixed offshore platforms under sea wave excitations are dependent on the sea-state. Whereas high probability of occurrence for moderate sea-states raises their participation in fatigue damage accumulation, furthermore a large amount of kinetic energy is exerted on the structure in savage sea-states, so all sea-states are decisive in fatigue damage accumulation in offshore structures and deficiency of the control system in each sea-state may taint its performance in fatigue damage mitigation. Improving the functionality of the TMD for fatigue damage mitigation in an offshore platform has been summarized in Table 1. Generalized mass associated with the principal dynamic mode of the platform and the corresponding frequency of vibration has been considered as the parameters of the primary SDOF system. P-M wave height spectrum with parameters related to open seas and north Atlantic wave scatter diagram [13] has been employed in these calculations.

The efficiency of the control system is defined as:

$$\text{Control Efficiency} = 1 - \frac{\text{Fatigue Damage with Control Mechanism}}{\text{Fatigue Damage wit hout Control Mechanism}}$$

In the studied model, using the optimum mass for TMD device in each sea-state has caused a significant improvement in its performance.

Such a mechanism is practical for offshore wellhead or production platforms which include massive oil and water storage tanks. Dual purpose tanks which are filled and discharged continually can be used as vibration absorber devices without the need for extra power sources and because of above water fabrication it is utilitarian for existing platforms.

5. CONCLUSION

An analytical study had been performed aiming to evaluate the efficiency of TMDs for fatigue damage mitigation in offshore steel jacket platforms.

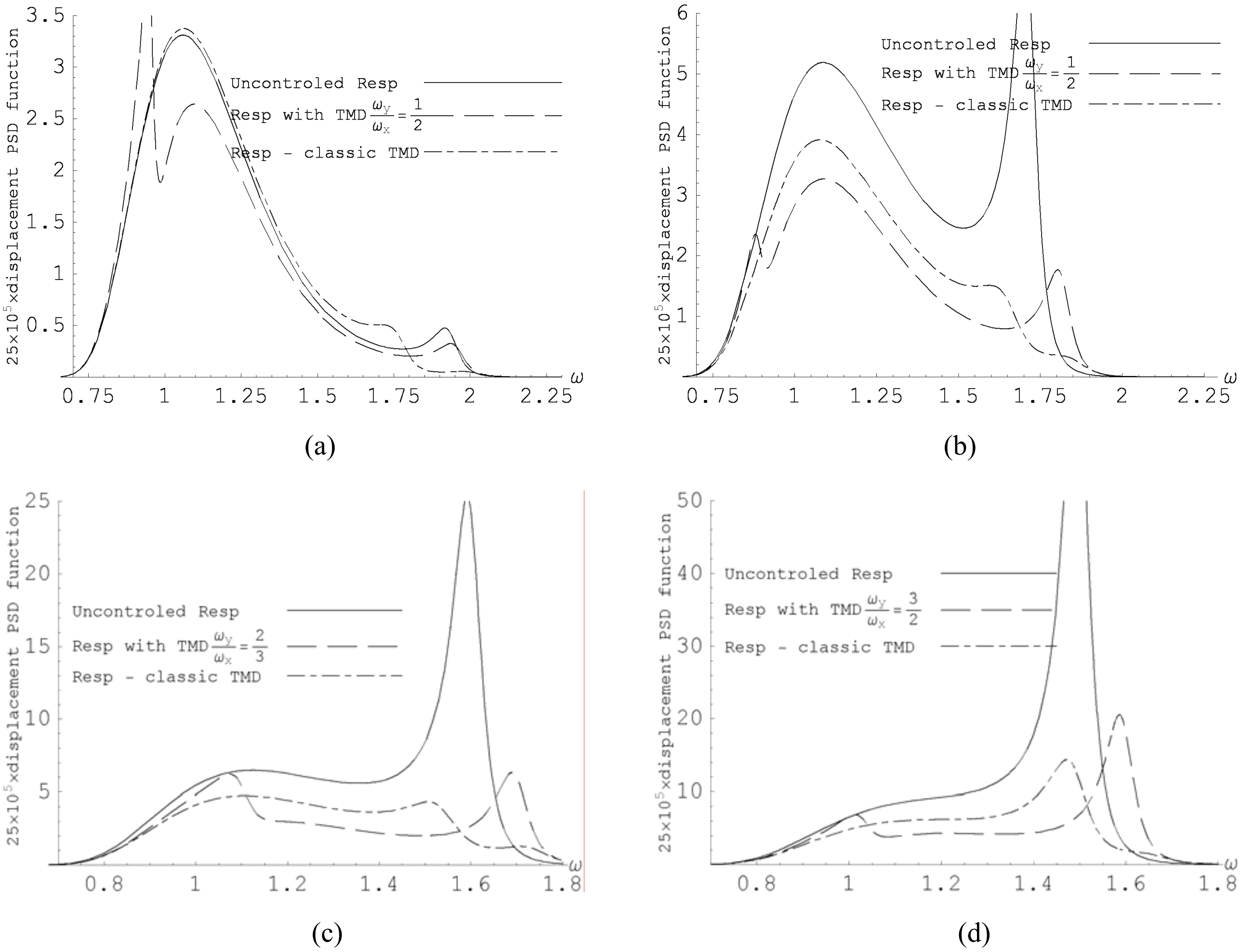

Figure 9. Displacement response PSD functions for a SDOF system considering P-M spectrum with T_z = 4.0 S. and H_s = 4.0 m for exciting waves (a) T = 3.25 S., (b) T = 3.5 S., (c) T = 3.75 S. and (d) T = 4.0 S.

An equivalent single degree freedom, corresponding to the principal mode of vibration for an offshore platform was studied under wave loadings, and the optimum values for adjustable parameters of tuned mass damper, and its performance were evaluated, considering the closed form solution of dynamic equation of motion, under full stochastic excitation.

The results exemplifies that the optimal values for the adjustable parameters of the auxiliary devices are strongly dependent on sea-state conditions defined with wave spectrum parameters; T_z and H_s. Using the optimal tuning ratio for each sea-state causes considerable increase in TMD efficiency, and considering the predictability of sea-states, this can be addressed as an advantage for offshore application of TMDs in comparison with their seismic applications. This passive type of vibration control devices are also more efficient for flexible platforms whose principal period of vibration fall within the prevailing sea waves frequency band.

As a case study, the efficiency of TMD had been examined to moderate the dynamic response of a realistic jacket type platform. Utilization of optimally tuned TMD on this platform resulted in 26.6 % reduction in the maximum fatigue damage. This efficiency can be increased up to 36.7% using variable tuning of auxiliary device in each sea-state.

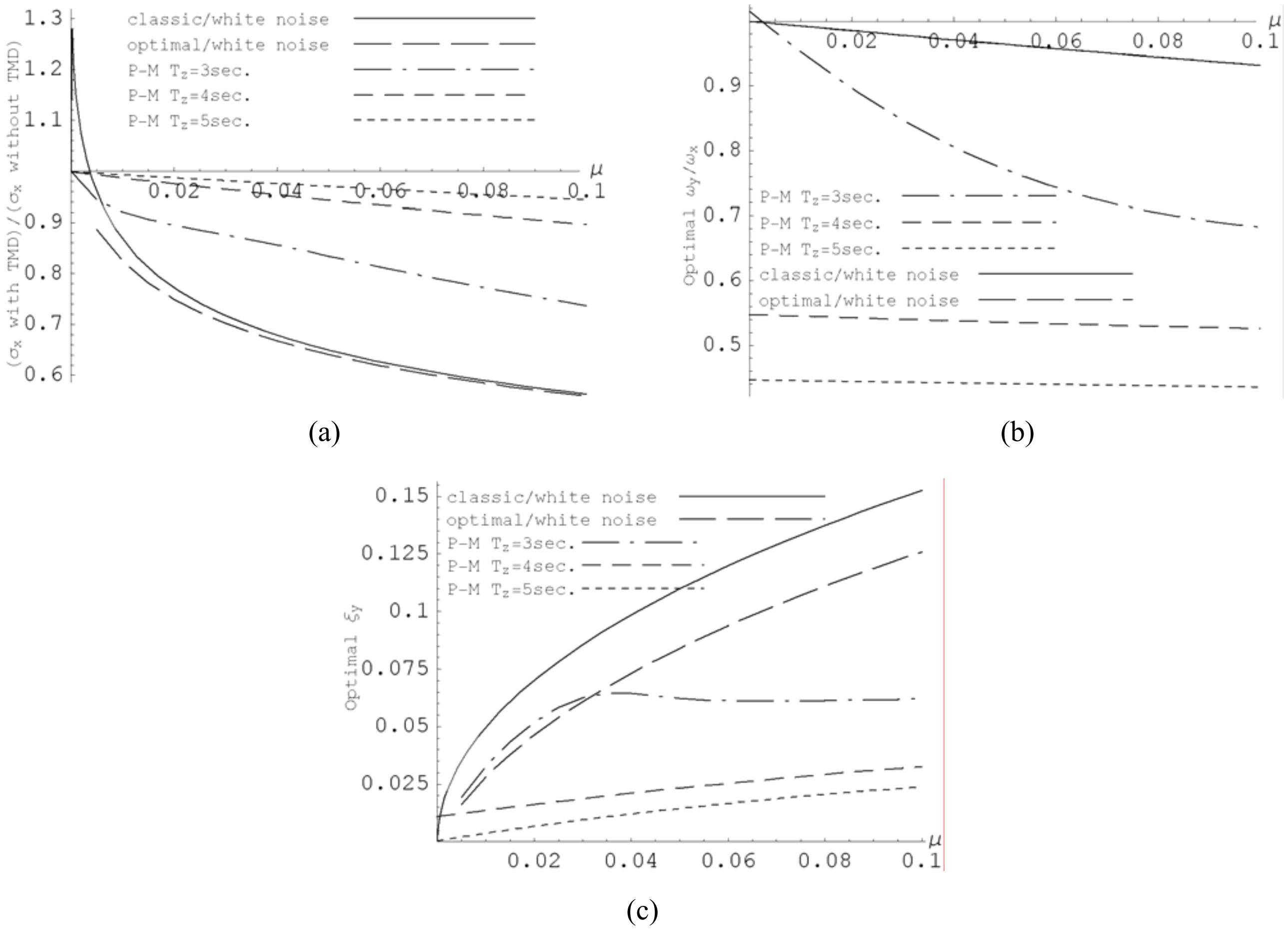

Figure 10. Optimum parameters for TMD and its efficiency to minimize the response of a SDOF with T = 3.5 S. and ξ_x = 3 % under random excitations

TABLE1. Variable Mass TMD and its Functionality in Comparison with Constant TMD for a Primary Structure with m_x = 2450 Tons, ω_x = 2.1 rad/S., ξ_x = 3 %.

Sea-State Definition		Sea-State Occurrence (%)	Efficiency of Constant TMD μ = 0.05 k_y = 0.024 k_x c_y = 0.05 c_x	Optimal TMD k_y = 0.024 k_x c_y = 0.05 c_x	Optimal TMD Efficiency
T_z S.	H_s m				
1.95	0.30	7.2	39.0 %	μ = 0.024	45.2 %
3.34	0.88	22.4	35.1 %	μ = 0.025	43.1 %
4.88	1.88	28.7	32.0 %	μ = 0.034	36.5 %
6.42	3.25	15.5	28.1 %	μ = 0.055	33.2 %
7.96	5.00	18.7	12.3 %	μ = 0.064	31.7 %
9.75	7.50	6.1	2.5 %	μ = 0.075	30.2 %
12.07	11.50	1.2	-4.5 %	μ = 0.081	29.3 %
13.32	14.00	0.2	-11.4 %	μ = 0.085	28.5 %
Aggregative:			26.6 %		36.7 %

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