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The Efficacy of Tuned Mass Damper in Mitigating Vibration on Steel Jacket Structure Using Matlab

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***Abstract*—Offshore Steel Jacket Structures are designed to withstand harsh environmental forces such as wind, and wave induced forces, along with dead loads. All of these forces are random and dynamic in nature; therefore the structural response is also dynamic leading to unsafe and uncomfortable conditions. One of the approaches to reduce excessive vibration on steel jacket structure due to dynamic forces is by installing a passive mechanical device called Tuned Mass Damper (TMD). This research presents a study on the effectiveness of TMD in reducing the response of structures that are subjected to wind induced excitation. A two-dimensional linear-elastic model with TMD on the top is used in performing dynamic analysis, the governing equations of motion of the jacket platform with TMD are derived and their result in the amplitude and frequency domain is presented. The response of the jacket platform with TMD is compared with the corresponding response without dampers in order to investigate the effectiveness of the passive control systems using MATLAB. It is discovered that the presence of the TMD add significant damping to structure, thereby control the response of the platform favorably. Subsequent research should consider the motion of the structure in two dimensions (i.e. the horizontal and torsional effect on the structure), instead of the horizontal movement alone as evaluated in this research.**

Keywords—Vibration Mitigation; Steel Jacket Structure; Mass Damper; MATLAB

I. INTRODUCTION

The exploration of crude oil initially was concentrated on land. Due to the high demand of oil at an explosive rate as well as communal crisis where these companies operate especially in the Niger Delta region of Nigeria, the need for new discoveries was eminent, leading to the discovery of crude in near shore and medium range of water depth to beef-up production and to curb away from the excessive demands from these communities.

The discovery of crude off the shore led to several installations of offshore platforms. The offshore platforms are usually built in a harsh ocean environment; where it is subjected to all kinds of loads like; loads from wave, wind induced loads, ice loads and sometimes earthquake loads, also loads caused by equipment and machines on the platform. The platforms pulsate very harshly under the corporate action of these loads, bringing much more inconvenience to the platform's performance. The acute exterior loads, such as earthquake, wave, wind or ice loads, can destroy the whole platform [3]. In order to boost the reliability and safety of the steel jacket platform, many vibration control methods have been used, among them is the nonviolent vibration mitigation method which is widely used because it doesn't need additional energy and has low cost, good control effect and easy actualization (ibid). The extensively used nonviolent vibration mitigation method is TMD (Tuned Mass Damper). So to advance the vibration mitigation performance, the excellent specifications of the TMD are studied below based on the help of MATLAB program.

Common dynamic problems in offshore platforms are associated with vibrations induced by ambient and machinery excitation on the platform.

Vibration can be characterized as the regular, harmonic or recurring motion of a machine or machine component from its position of rest [1]. It is the fluctuation of a particle or system of linked bodies displaced from its equilibrium position. Vibration is an unwanted phenomenon that is always found in a system, undesirable in machines and structures because it promote stresses, produce energy losses which accompany them, induce fatigue, causes wear, increase bearing loads etc. Excessive vibration has been a common problem throughout engineering history, it makes the weakened life of structures shorter and induces uncomfortable noise [2]. Complex steel structures are usually lightly damped as such are prone to resonant vibration problems. The most effective way to reduce unwanted vibration is to suppress the source of vibration. However, this is not often possible for practical reasons. The source of vibration is distinctively associated with the machine's primary function, for example the rotating imbalances in a running engine. Moreover, the vibration may be induced by natural sources like winds or earthquakes.

Vibration is one of the outstanding challenging problems in offshore structures; it cannot be eliminated, but can be mitigated. Dynamic problems associated with vibrations in offshore structure are possibly one of the highest effort-intensive tasks faced by the engineering profession; these structures have added difficulty of being installed in an ocean environment where hydrodynamic synergy effects and forceful feedback become dominant consideration in their design over the normal conditions encountered by land-based structures.

Other problems associated with the construction of offshore platforms are; fatigue, creep, buckling etc.

This paper aimed at investigating the efficacy of Tuned Mass Damper (TMD) in mitigating vibration response of structures using MATLAB simulation.

II. LITERATURE REVIEW

Offshore platforms are stupendous steel structures that are basically used for the extraction of oil and gas from the earth's crust, and they are installed in the open sea environment. These platforms are made of different grades of steel such as mild steel, high-strength steel etc. Also there are varieties of structures depending on their use and water depth in which they will function. Offshore structures are exposed to very unfriendly marine environments and must function reliably for a very long of time. For most platforms, the major potential damage and risks to personnel are caused by the activities such as; vibration resulting from drilling and crane operations and other rotating equipment on the topside of the platform. The study of offshore structures is intricate due to various uncertainty sources like; uncertainty of dynamic strength of materials used, dynamic strength of joints, environment; load, etc. Furthermore, the analysis process embroils various fields of sciences and engineering. With the swift advancement of offshore structures for oil and gas fields since 1973, a considerable number of researchers have developed interest in the analysis of these structures up to this moment. One of the underlying hitches in analyzing an offshore platform structure is the determination of environmental and machinery forces acting on it.

The most effective and economic way to mitigate vibration is to apply an additional dynamic system at a discrete point of the existing structure to change the system dynamics in a desired way. TMDs fall into this category of devices. They are basically simple mass spring damper systems connected to a selected point of the vibrating structure[4]. Tuned Mass Dampers are basically utilized on machines running at a steady or uniform speed, also they are exploited in transient vibration control, wind-induced vibrations control as well as sound radiation control [5]. It is essential to note that the TMD is only effective over a small frequency band. The effective band can be selected by design parameters, and it is typically designed to match with a problematic vibration mode.

A. *Types of Offshore Platforms*

Types of offshore platforms ranges from those fixed (i.e platforms that are fixed to the sea bed) to that considered to be free floating. The particular type to be installed in a given offshore environment depends on a number of factors that also reflect the use and water depth in which it will be located.

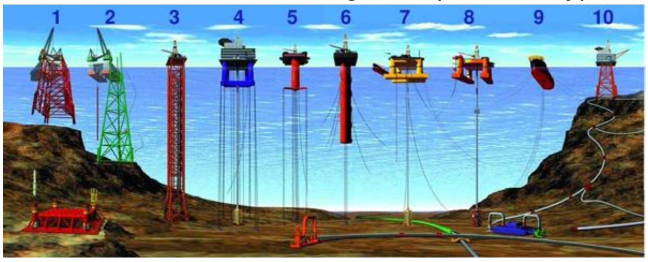


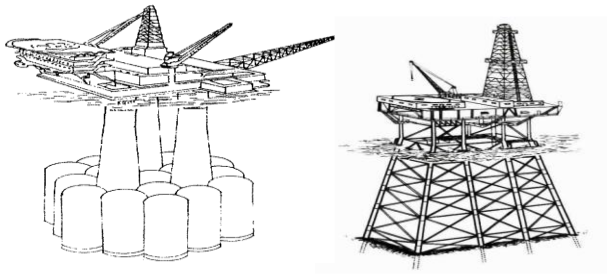
Fig.1. Profile of different types of offshore platform [6].

KEY: 1, 2) Conventional fixed platforms;
3) Compliant tower;
4, 5) Vertically moored tension leg and mini-tension leg platform;
6) Spar;
7, 8) Semi-submersibles;
9) Floating production, storage, and offloading (FPSO) facility;
10) Sub-sea completion and tie-back to host facility.

Below are some of the main types of oil production platforms that are used:

i. *Fixed Platforms*

The conventional fixed platforms are assembled on steel or concrete legs, or even both. They are fastened directly to the ground under the sea, assisting a deck for drilling rigs, facilities for production



and quarters for the work force (i.e crew). Owing to their immobile nature, they are economically designed to last long and are installed in water depths of about 520 m (1,710 ft).

(a) (b)

Fig. 2 Fixed platform support systems (a) Concrete gravity-based structure (b) Steel jacket structure [7]

It is suggested that the de-commissioning or deactivation of large offshore substructures, such as the concrete gravity-based platform, should be conducted on a case-by-case basis because each substructure has its own design features and performance history [8]. However, they maintained that the first generation concrete gravity-based structures installed in the 1970s were not designed to be decommissioned thereafter and that only the second generation offshore concrete gravity platforms have been constructed for decommissioning (ibid). Furthermore, it is eminent that small lateral displacements which occur at frequencies above those of the incident waves can be associated to concrete gravity based platform.

ii. Compliant Towers

Compliant Towers are composed of narrow, pliable towers and a piled substructure or base sustaining a deck for drilling and production operations. They are constructed to support sideway deflections and loads, and are situated in water depths ranging from 370 to 910 meters (1,210 to 2,990 ft).

Fig. 3 Compliant Tower (Ali Seyedalangi, "Types of Drilling Rigs and Structures" LinkedIn www.linkedin.com).

iii. Jack-up Platformss

These are Offshore Drilling Units that are able to move around and it has three or four 'legs' that are lowered to rest on the seabed like jacks, allowing the working platform to rest above the water surface.

They are constructed to migrate from place to place, and are supported on legs that move in the vertical direction. These rigs are typically safer to operate as their working platform is elevated above the water level. The platform as presented in figure 4 is anchor by positioning the legs to the seabed using a rack and pinion gear system on each leg. They are suitable only for shallower waters ranging from 120-170 meters (390 - 560 feet) depth

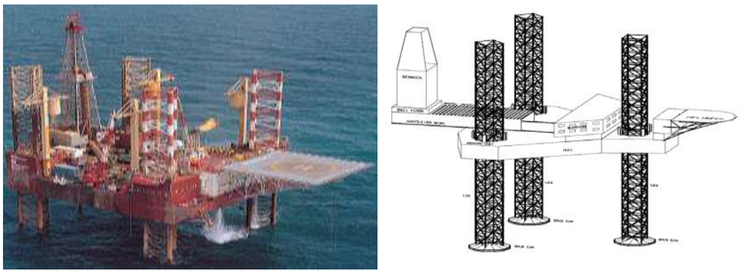


Fig. 4. Jack-up platform [9]

iv. Tension Leg Platform (TLP)

These are floating production platforms that are anchored to the seabed through the tendons fixed vertically to the platform such that they eliminate vertical movement of the structure. They are operated in water depths upto about 2,000 meters (6,600 feet). These platforms undergo a lot of stresses horizontally due to waves, compared to the vertical stresses that are restricted by the tendons fastened to the base at the ocean floor.

Fig. 5 Tension Leg Platform

v. Spar Platform

This platform has a hollow cylindrical hull that can go down to a water depth of 200 meters and it is fastened to the ocean bed by a convoluted network of cables and tendons. The weight of the cylindrical hull balances the drilling platform and coddle for the risers to descend down to the well on the sea bed. They are used to conveniently drill wells above 10,000 feet.

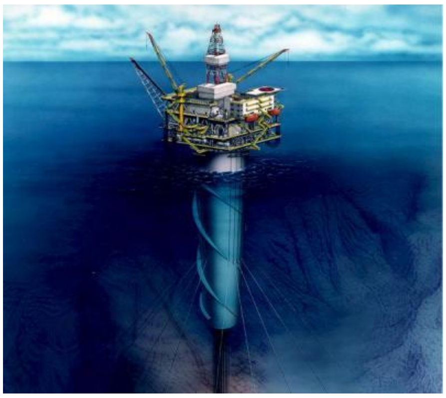


Fig. 6 Spar Platform

Unlike the TLP, the spar has more intrinsic stability due to its large counterweight at the bottom and it does not rely on the mooring to sustain it uprightly. Also, the mooring line tensions can be adjusted to move horizontally and be positioned over wells at some distance from the location of the main platform.

vi. Submerible Rig Platforms

Like jack-up rigs, submersible rigs are appropriate for shallow water, as such they come in contact with the sea or lake floor [10]. They are made up of platforms with two shell or hulls situated on each other; the upper shell contains drilling platform and the living quarters for the crew, while the lower shell is more like the outer shell or hull in a submarine that is filled with air – making the entire rig buoyant so as to enable the platform move from one point to another (ibid). As soon as the rig is positioned over the drill site, the air is discharged from the hull, and the rig submerges to the floor of the lake. This rig has the merit of moving in the water; however, its use is limited to shallow water areas.

vii. Semi – Submersible Rig Platform

This type of platform has hulls of adequate buoyancy to enable the structure float, but its weight must be enough or ample to maintain upright structure. It can be transported from one point to another and be counterbalanced by modifying the amount of overflow in buoyancy tanks. The platform works on the same principle like that of the submersible rigs; by way of inflating and deflating its lower hull [10].

During drilling, the hull below when filled with water proffers balance to the rig. Semi-submersible rigs are usually positioned in place by massive anchors, each having a weight of about ten tons, and are attached with the submerged segment of the rig, to ensure stable platform and safe use of the structure in violent offshore waters (ibid). Dynamic positioning can also keep it in place. Now with the advent in technology, depths up to 6,000 feet (1,800 m) can easily and safely be achieved. Semi-submersible rig platform can drill a hole in the seabed and quickly be moved to new locations.

Fig.7. Marine_7000 fourth generation semi-submersible [11]

viii. Floating Production, Storage and Offloading (FPSO) Systems

FPSO units are floating vessel used for the production and processing of hydrocarbons, as well as the storage of oil. These vessels are produced to store hydrocarbons extracted from subsea template, process them, and store the processed oil until it is offloaded to a tanker or conveyed through a pipeline. FPSOs are platforms that are fixed to a particular location for a long time, and they consist of huge hull structures, mainly ship-shaped, furnished with processing facilities. They are isolated at the boundary of the shore because they are simple to install, and do not require a local pipeline infrastructure to export oil.

Fig. 8. FPSO system. [9]

ix. Drillships

Drillships are ships or mobile vessels designed with drilling facilities used for exploration and research. They are designed specifically to convey the drilling facilities to oil well locations in the sea. Some drillships in addition to the equipment normally found on it, have a drilling platform and derrick situated at the middle of its deck. This offshore oil rig can drill in very deep waters. Drillships are fitted with the dynamic positioning system to provide stability during operation. They are furnished with electric motors under the hull of the ship, able to propel the ship in any direction. The electrical motors are inserted into the computer system of the ships, which employ satellite positioning technology, alongside with sensors situated on the drilling template, to ensure that the ship is directly above the drill site at all times [10]. They are applicable for sea depths upto 3700 m (12,100 ft).

B. The Concepts of Vibration

Vibration is defined as a motion which repeats after equal interval of time and is also a periodic motion [12]. It occurs in all bodies that has mass and elasticity. A lot of factors lead to the cause of vibration, some them are; the presence of unbalanced force in rotating machines, elastic nature of the system, external application of forces. In most engineering systems, vibration is not needed while in some they are desirable.

Mechanical vibrations can be classified based on degrees of freedom. This is the number of kinematically Independent variables required to effectively describe the motion of every particle in the system [13].

Based on degrees of freedom, vibrations can be classified as:

i. Single Degree of freedom Systems (SDOF)
ii. Two Degrees of freedom Systems (MDOF)
iii. Multi-degree of freedom Systems (MDOF)
iv. Systems with vast or infinite degrees of freedom.

Vibration can also be classified as follows:

i. Damped and Undamped vibration
ii. Linear and nonlinear vibration

C. Forms of Vibraiton

i. Free vibration

This is the natural feedback of a structure to some shock or movement in the absence of an external excitation. The reaction is totally dependent on the structural properties, and its excitation can be perceived by surveying the mechanical properties of the structure.

ii. Forced vibration

This is the reaction of a structure to a recurring forcing function that makes it to vibrate at the frequency of the stimulation.

iii. Sinusoidal vibration

This is an exceptional form of vibration where the structure is excited by a forcing function with a single frequency.

iv. Random Vibration

Random vibration is universal in nature. For instance, when a car moves on a rough road, the vibration resulting from a combination of the surface of the rough road, vibration of the engine and the wind buffeting on the car's exterior are all referred to as random vibration.

v. Rotating Imbalance

This is the movement of a machine part that is not balanced thereby causing the vibration of an entire structure. Examples are as follows; an automobile engine, shafts, pumps and turbines, washing machine etc.

D. Sources of Vibration

i. Earthquakes

Earthquake generates seismic waves which cause buildings or structures to shake and vibrate in diverse ways depending on the frequency and bearing of the ground motion.

ii. Wind

The action of wind force on tall structures can cause the topmost part of structures to sway above one meter. This displacement can be in form of swaying or twisting, and as such can cause the upper parts of buildings to move. Specific angles of wind and aerodynamic properties of a structure can highlight the movement [12].

iii. Mechanical Sources

Mechanical equipment (static and rotating) on offshore structures can pose serious vibration problem to the platform.

E. Consequences of Vibration

i. Damages to safety-related equipment
ii. Adverse human response
iii. Fatigue fracture
iv. Overstressing and collapse of structures
v. Cracking and other damages

F. Mechanical Equipment used in Offshore Platform

i. Rotating Equipment

Rotating equipment require a rotor system that add kinetic energy to a process in order to execute its operation. The list of common rotating equipment are; Pumps, Compressors, Turbines, Motors, Agitators, Blowers, Fans, drill pipes and bits, generators on platform, etc..

ii. Static Equipment

Static equipment are fixed or rooted to the members of the platform and have no rotating shaft, but the liquid content that flows through or comes into it often give rise to Vortex Induced Vibration (VIV). Examples of static equipment are; Piping, Pressure vessels, Heat Exchangers, Columns, Reactors, Filters, Drums, Tanks, bullet, others are manifold, Header, flow lines, etc.

G. *Steel Jacket Offshore Platform*

The Steel Jacket structures have been in used in petroleum activity for years. They are mostly adopted for shallow and intermediate water depths below 150m [14]. The steel jacket offshore platform is a large steel structure mainly used for the exploration and extraction of oil and gas from the earth's crust, constructed for installation in the open sea. This platform is made of various grades of steel and it is exposed to very unfriendly marine environments and designed to function reliably for a long duration of time. Some significant considerations during construction are peak loads generated by wind and waves, fatigue loads conceived by waves over the lifetime of the platform as well as static loads and vibration induced load initiated by some rotating equipment on the platform [10].

The conventional steel jacket platforms are constructed on either concrete or steel legs, or even both. They are fixed directly to the seabed, aiding a deck for drilling rigs, production equipments and quarters for workforce. Due to their immobility, they are designed for long term use and are economically viable for installation in water depths of about 520 m (1,710 ft). They either have concrete gravity based structures or are made of steel jacket.

The steel jacket structure is mainly design to be safe, functional, economical, as well as its ability to withstand environmental forces for a prescribed period of time. One of the underlying hitches in analyzing an offshore platform structure is the determination of environmental and machinery forces acting on it. Safety of thee structure is generally assumed to be obtained by design according to established standrds and methods, for an expected desgn life, and for a structure is stay beyond its design life, a thorough control of the structural safety must be executed [14].

H. *Loads and Responses on the Offshore Steel Jacket Platform*

Loads on the structure are distinguished between static and dynamic. The static loads come from gravity loads, deck loads, hydrostatic loads and current loads, while the dynamic loads emanate from the variable wind and waves [11]

The various loads acting on offshore platform are listed below:

a. Gravity Loads

Gravity loads are loads that arise from dead weight of structure, operating equipment weights, live and buoyancy loads. Examples of gravity loads are:

Structural Dead Loads, Facility Dead Loads, Fluid Loads, Live Loads

b. Current Loads on Structures

Current is often considered time-invariant in the construction of offshore structures; it is represented by its mean value. Though the strength of the current may have a fluctuation with water depth. The current proposes a varying pressure circulation around a member of the structure producing a steady drag force on the structure in the flow direction [11]. Also, a transverse force is produced on the structure due to asymmetric distribution of the pressure about the flow direction.

Oceans currents with wave action induce drag loading and generate dynamic forces on offshore structures. They are classified into; wind driven current, tidal current and induced current. The wind driven currents varies linearly with depth whereas tidal currents vary nonlinearly with depth. Then the induce currents due to ocean circulation vary nonlinear with depth and can have a velocity up to 5 ms^{-1}. The current variation with depth can be expressed in equations 1 and 2 shown below [15].

$$V_T = V_{oT}\left(\frac{y}{h}\right)^{-7} \qquad 1$$

Where;

V_T = tidal current at any height from sea bed,

V_{oT} = tidal current at the surface,

y = distance measure in m from seabed and

h = water depth.

$$V_w = V_{ow}\left(\frac{y}{h}\right) \qquad 2$$

Where;

V_w is the wind driven current at any height from sea bed,

V_{ow} is the wind driven current at the surface,

y is the distance measure in m from seabed and h is the water depth.

c. Dynamic Wind Loads on Structures

Wind is considered as a time-invariant environment, it has a mean value that is equal to its turbulent velocity [11]. With this clarification, the outcome of the wind on the superstructure (i.e. portion above the water) of an offshore platform is denoted with a mean force. In this case the wind load is given by an expression similar to the current force in terms of a wind drag coefficient as shown below (ibid):

$$f = \frac{1}{2}\rho C_D A U_w^2 \qquad 3$$

Where:

ρ = density of air,

A = structure projected area normal to the wind flow,

C_D= the wind drag coefficient, and

U_w = mean wind velocity, generally taken at an elevation of 10 m from the water surface.

The wind speed (V_o) at 10m above LAT (Lowest Astronomical Tide) is often provided. This wind speed shall be extrapolated to the height above for the calculation of wind speed (V). The extrapolation shall be calculated using the equation shown in equation 4 [15].

$$V = V_o\left(\frac{y}{10}\right)^{-8} \qquad 4$$

Where;
y = the elevation of point in consideration above LAT (m)
V = the velocity at that point.
Wind loads shall be calculated as per API RP2A guidelines.

d. Marine Growth

Marine growth increases the loads on offshore structures. The marine algae increase the diameter and coarseness of structural members which in turn increase the wave or current loading. The stratum of marine growth usually reduces with depth from the mean sea level and it is maximum at the splash zone. In the splash zone the size of marine growth can be as much as 0.2m and will reduce below to 0.05m. In deeper zones, it may be negligible.
Splash Zone is an area where the water levels alternate between low to high. The actual altitude of the bottom and top of these vary from location to location due to different tidal conditions. In general terms, the splash zone will vary from -3m to +5m [15].
The increased diameter of the member ($D = d + t_m$) is often added so as to calculate the loads of wave and current accurately during analysis. D and d are the diameter of increased member and original member respectively while t_m is the thickness of marine growth. The roughness of the marine growth is an essential parameter in determining the drag and inertia coefficients.

e. Wave Loads on Structures

The forces on offshore structure induced by waves are computed by two different methods depending on the size of the structure. At this point, the structure is stratified as small or large. The forces on small structures will be discussed first.

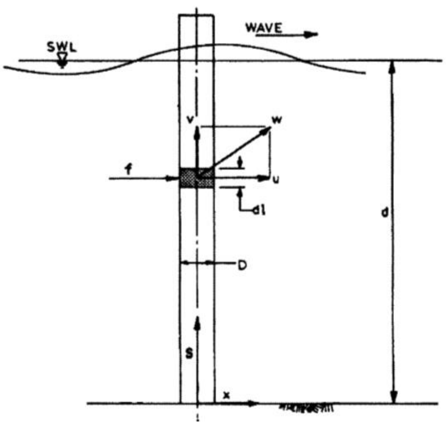


Fig. 9 Morison force on a vertical pile [11]

J.E. Morison, predicted wave forces on an exposed vertical pile. He intelligently foists the linear inertia force (from potential theory and periodic flows) and a modified quadratic drag force (from real flows and steady currents) to arrive at a total force shown in equations 5 and 6 [11]

$$F_T(t) = F_{inertia}(t) + F_{drag}(t) \qquad 5$$

Or;

$$F(t) = \frac{\pi D^2}{4} C_M \rho_w a(t) + \frac{1}{2} C_D \rho_w D V(t)|V(t)| \qquad 6$$

Where:
F= Horizontal Force per unit length
C_D and C_M = Drag and Inertia Coefficient respectively (Typical values of coefficients for a cylinder: C_D = 1 and C_M = 2)
ρ_w = Density of water
D = Pile Diameter including marine growth,
V$|V|$ = Wave particle Velocities in the horizontal and vertical directions respectively
a = Acceleration
dl = Unit elemental length.

A moving structure will oscillate under the influence of environmental loads, thus an altered form of the Morison equation is displayed to describe the force per unit length undergone by the structure due to its motion through the water by the following equation (ibid):

$$F_T = m\ddot{x} + \frac{\pi D^2}{4} C_A \rho \ddot{x} + \frac{1}{2} C_D \rho D \dot{x}|\dot{x}| \qquad 7$$

Equation 7 is exclusively for a cylinder, where;
m = the mass of the cylinder per unit length,
C_A = the added mass coefficient,
x = the known harmonic displacement of the cylinder and
$\dot{x}$ and $\ddot{x}$ = first and second derivatives of x.

The first term on the right-hand side of the equation represents the cylinder inertia, while the last two terms are the drag forces due to the motion of the cylinder in water and the hydrodynamic inertia respectively.
Since equation 7 is empirical, the values of the coefficients C_A and C_D, are determined experimentally. The coefficient values are assumed constant over a cycle for a given frequency of vibration.

f. Mud Loads

Mud in coastal areas like the Niger Delta region of Nigeria is mainly found within the intertidal zone (i.e. the shore region that lies between the highest standard high tide and the lowest standard low tide). It is a mixture of fine-grained sediments (clays, silt and sand), organic matter and water.
Shallow water platforms often experience mud loads, as the river flows it brings sediment transport and nearby mud towards the platform which may slide through the area.

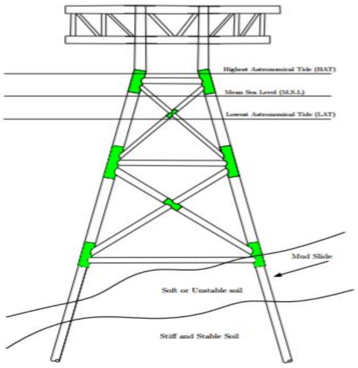


Fig. 10. Mud Loading on a structure

Sometimes over time sediment that have settled at the area of the platform may have sloping surface leading to mud slides which can as well generate mud loads. These loads can be estimated using the formula below [15].

$$F_{mud} = C_{mud}\ \tau\ D \qquad 8$$

Where:

C_{mud} = Force Coefficient vary from 7 to 9
T = Shear strength of soil 5kPa to 10kPa
D = Diameter of pile or member.

I. Waveload on a member

In reality, the members of the offshore structure may be horizontal or inclined in space and cannot use without modification.

a. Water Wave Kinematics

Airy wave theory is considered in the estimation of water wave kinematics. Consider a continuous wave with water surface elevation represented by cosine curve;

$$\zeta = \frac{H}{2} \text{Cos}\ (kx - \omega t) \qquad 9$$

And the relative velocity potential is given by

$$\phi = -\frac{H}{2}\frac{\omega}{k}\frac{\cosh k(h+z)}{\sinh kh} \sin\ (kx - \omega t) \qquad 10$$

The horizontal and vertical velocity and acceleration of water particle is calculated using the following equations [11].

$$V_h = -\frac{\partial\phi}{\partial x} = \frac{H}{2}\ \omega\ \frac{\cosh k(h+z)}{\sinh kh} \cos(kx - \omega t) \qquad 11$$

$$V_v = -\frac{\partial\phi}{\partial z} = \frac{H}{2}\ \omega\ \frac{\text{sinsh}\ k(h+z)}{\sinh kh} \sin(kx - \omega t) \qquad 12$$

$$a_h = \frac{\partial V_h}{\partial t} = \frac{H}{2}\ \omega^2\ \frac{\cosh k(h+z)}{\sinh kh} \sin(kx - \omega t) \qquad 13$$

$$a_v = \frac{\partial V_v}{\partial t} = -\frac{H}{2}\ \omega^2\ \frac{\cosh k(h+z)}{\sinh kh} \cos(kx - \omega t) \qquad 14$$

Where:

k = wave number (defined by $\frac{2\pi}{\lambda}$),
ω = wave circular frequency (defined by $\frac{2\pi}{T}$),
λ = wave length and x is the distance of the point in consideration from origin.

b. Maximum Load on a vertical member

Consider a cylinder breaking through the surface of the sea as in the case of the leg of a jacket structure, the total force (i.e. sum of the drag and inertia force) varies with time and will be maximum only at one occasion.

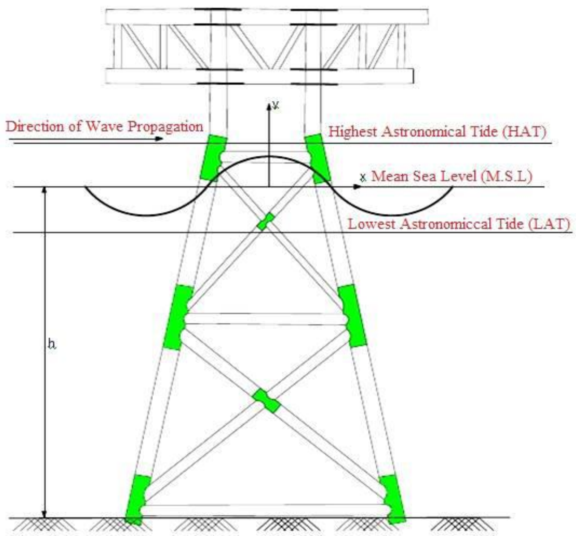


Fig. 11. Wave Loads on Jacket Structure ([15]

Let the effective force on the pile be express by substituting the velocity and acceleration components and integrating between the limits (from the sea surface to the seabed, i.e., 0 to h) .

$$F_T = \frac{1}{2} C_D\ \rho\ D \frac{\pi^2 H^2}{T^2} \frac{\cos\theta|\cos\theta|}{\sinh^2 kh} \left[\frac{\sinh(2kh)}{4k} + \frac{h}{2}\right] - C_M\ \rho \frac{\pi D^2}{4} \frac{2\pi^2 H}{T^2} \frac{\sin\theta}{k} \qquad 15$$

The total force will be maximum when, $\frac{\partial F_T}{\partial\theta} = 0$

Substituting the values of velocity and acceleration components in to the total force equation above and differentiating with respect to **θ**, then rearranging the terms, we have;

$$\theta_{max} = \cos^{-1}\left[-\frac{\pi D}{H}\frac{C_M}{C_D}\frac{2\sinh^2 kh}{(\sinh 2kh + 2kh)}\right] \qquad 16$$

c. Maximum Load on a horizontal member

In the case of a horizontal cylinder like the brace of a jacket structure, the total force also varies with time and will be maximum only at one occasion. Like the maximum load on a vertical member, to determine the maximum force, phase angle at which this maximum force occurs shall be found first, and to achieve that let the total force on the pile be express by substituting the velocity and acceleration (ibid).

$$F_T = \frac{1}{2} C_D \rho D \frac{\omega^2 H^2}{4} \cos\theta \left|\cos\left[\frac{\cosh^2 k(z+h)}{\sinh kh}\right]\right| - C_M \rho \frac{\pi D^2}{4} \frac{\omega^2 H}{2} \sin\left[\frac{\cosh^2 k(z+h)}{\sinh kh}\right] \quad 17$$

The total force will be maximum when, $\frac{\partial F_T}{\partial \theta} = 0$

Substituting the values of velocity and acceleration components in to the total force equation above and differentiating with respect to **θ**, then rearranging the terms, we have

$$\theta_{max} = \sin^{-1}\left[-\frac{\pi D}{2H}\frac{C_M}{C_D}\frac{\sinh kh}{\cosh k(h+z)}\right] \quad 18$$

d. *Maximum Load on an Inclined Member*

The effective force on a rapidly aligned round cylinder in water waves can be premeditated using vector analysis integrated with Morison equation. It has two components; component normal to the cylinder axis F_n and a component along the axis of the cylinder (a tangential component) F_t. Thus, the total load per unit length of the cylinder can be written as (ibid);

$$\vec{F} = \overrightarrow{F^n} + \overrightarrow{F^t} \quad 19$$

Each of these components can be expressed as functions of the fluid particle motions by using Morison's equation. The force in normal direction can be expressed as;

$$\overrightarrow{F^n} = \overrightarrow{F_D^n} + \overrightarrow{F_I^n} \quad 20$$

Where $\overrightarrow{F_D^n}$ and $\overrightarrow{F_I^n}$ are the drag and inertia forces respectively. These forces can be expressed as;

$$\overrightarrow{F_D^n} = \frac{1}{2} C_D^n D \rho \overrightarrow{V_n} \left|\overrightarrow{V_n}\right| \quad 21$$

$$\overrightarrow{F_I^n} = = \frac{1}{4} \pi C_M^n \, I \, D^2 \rho \, \overrightarrow{a_n} \quad 22$$

Where:

C_D^n= Drag coefficient for flow normal to the cylinder
C_M^n = Inertia coefficient for flow normal to the cylinder
D = Diameter of cylinder
ρ = Density of seawater
$\overrightarrow{V_n}$ = Velocity of fluid particle normal to the cylinder axis
$\overrightarrow{a_n}$= Acceleration of fluid particle normal to the cylinder axis

Only a drag force exist at the direction of the tangent, while inertial component along the axis of the element does not exist unless the specification of an axial inertia coefficient. Thus the equation for tangential force can be written as;

$$\overrightarrow{F^t} = \overrightarrow{F_D^t} = \frac{1}{2} C_D^t D \rho \overrightarrow{V_t} \left|\overrightarrow{V_t}\right| \quad 23$$

Where
C_D^t= Drag coefficient for flow tangential to the cylinder
$\overrightarrow{V_t}$ = Velocity of fluid particle tangential to the cylinder axis

These forces can be summed and expressed in terms of cylinder local axis as follows:

$$\overrightarrow{F_x} = \frac{1}{2} C_D^t D \rho \overrightarrow{V_t} \left|\overrightarrow{V_x}\right| \quad 24$$

$$\overrightarrow{F_y} = \frac{1}{2} C_D^n D \rho \overrightarrow{V_n} \left|\overrightarrow{V_y}\right| + \frac{1}{4} \pi C_M^n \, I \, D^2 \rho \, \overrightarrow{a_y} \quad 25$$

$$\overrightarrow{F_z} = \frac{1}{2} C_D^n D \rho \overrightarrow{V_n} \left|\overrightarrow{V_z}\right| + \frac{1}{4} \pi C_M^n \, I \, D^2 \rho \, \overrightarrow{a_z} \quad 26$$

III. METHODOLOGY

The wind induced vibration on the steel-jacket structure can be mitigated by adjusting rigidities, masses, damping, or shape, and by providing passive counter forces, of which tuned mass damper is considered due to its efficiency, compactness and weight, capital cost, operating cost, maintenance requirements and safety. The tuned mass damper made with welded steelwork is attached on the top of the structure as shown below.

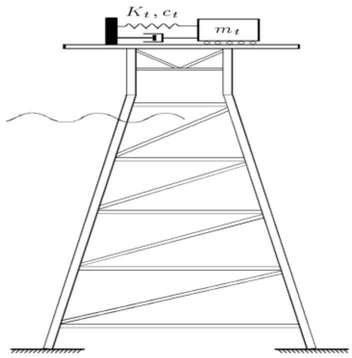


Fig. 12. Systemic Modeling of Steel Jacket platform with a tuned-mass damper (TMD).

It is made up of a secondary mass (mass damper) with satisfactorily tuned spring and damping elements, providing a frequency-dependent device that reduces response in the primary (main) structure. In this study, the motion of the structure is considered only in one dimension (i.e. the horizontal direction). Furthermore, the torsional effect on the structure is assumed to be negligible. When wind load induced vibration, the damper will absorb the vibration so that the amplitudes of the vibration will be diminished.

A. *Analytical Formulation of Tuned Mass Damper*

The steel jacket structure is idealized as a multi degree-of-freedom system, and the TMD device consists of a frame, a mass, two springs, four wheels, and two tracks, as demonstrated in figure 13

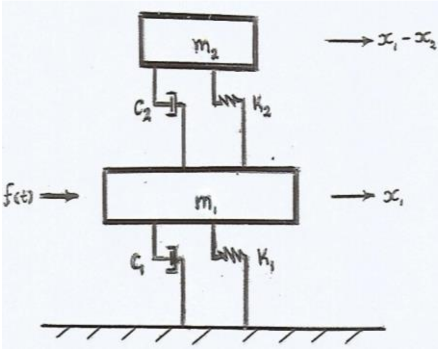


Fig. 13. Model of platform with TMD system

Where:
m_1 = mass of the structure;
m_2 = mass of TMD
k_1 = spring stiffness of the structure
k_2 = TMD spring stiffness
c_1 = Damping coefficient of the structure;
c_2 = TMD damping

In figure 13 above, the spring system m_2; k_2; c_2 is the damping oscillator. (i.e. TMD attached to the structure), and m_1; k_1; c_1 is the oscillator to be damped (i.e. the platform)
Assuming that the damping force is proportional to velocity and there is a periodic force $F\sin(\omega t)$ on m_1 it is easy to work out the differential equations governing the motion of the system.
The governing equations of motion of the system under consideration are expressed as follows

$$m_1\ddot{x}_1 + c_1\dot{x}_1 + k_1x_1 + c_2(\dot{x}_1 - \dot{x}_2) + k_2(x_1 - x_2) = F(t) \qquad 27$$

$$m_2\ddot{x}_2 + c_2(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) = 0 \qquad 28$$

Presenting equations 27 and 28 in matrix form gives;

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} c_1 + c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix}\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} F(t) \\ 0 \end{bmatrix} \qquad 29$$

The presence of damping in equations 28 and 29 induces a phase shift between periodic excitation and response. Therefore, it is convenient to utilize complex quantities. Thus let:

$$F(t) = F_0 e^{i\omega t} \qquad 30$$

The excitation will be composed of real and imaginary components, and since the system is linear, it will be justified to use complex input.
Therefore let:

$$\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} A_1 \\ A_2 \end{bmatrix} e^{i\omega t} \qquad 31$$

The time derivate vectors are expressed as:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = i\omega e^{i\omega t}\begin{bmatrix} A_1 \\ A_2 \end{bmatrix} \qquad 32$$

And

$$\begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} = -\omega^2 e^{i\omega t}\begin{bmatrix} A_1 \\ A_2 \end{bmatrix} \qquad 33$$

Substituting equations 31, 32 and 33 into equation (29)

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\left\{-\omega^2 e^{i\omega t}\begin{bmatrix} A_1 \\ A_2 \end{bmatrix}\right\} + \begin{bmatrix} c_1 + c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix}\left\{i\omega e^{i\omega t}\begin{bmatrix} A_1 \\ A_2 \end{bmatrix}\right\} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}\left\{\begin{bmatrix} A_1 \\ A_2 \end{bmatrix} e^{i\omega t}\right\} = \begin{bmatrix} F_0 e^{i\omega t} \\ 0 \end{bmatrix} \qquad 34$$

Expanding and grouping common terms:

$$\begin{bmatrix} -m_1\omega^2 & 0 \\ 0 & -m_2\omega^2 \end{bmatrix}\begin{bmatrix} A_1 \\ A_2 \end{bmatrix} e^{i\omega t} + \begin{bmatrix} (c_1 + c_2)i\omega & -c_2 i\omega \\ -c_2 i\omega & c_2 i\omega \end{bmatrix}\begin{bmatrix} A_1 \\ A_2 \end{bmatrix} e^{i\omega t} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}\begin{bmatrix} A_1 \\ A_2 \end{bmatrix} e^{i\omega t} = \begin{bmatrix} F_0 e^{i\omega t} \\ 0 \end{bmatrix} \qquad 35$$

Further grouping and cancelling the common term "$\boldsymbol{e^{i\omega t}}$" yields:

$$\begin{bmatrix} (-m_1\omega^2 + k_1 + k_2) + (c_1 + c_2)i\omega & -k_2 - c_2 i\omega \\ -k_2 - c_2 i\omega & (-m_2\omega^2 + k_2) + c_2 i\omega \end{bmatrix}\begin{bmatrix} A_1 \\ A_2 \end{bmatrix} = \begin{bmatrix} F_0 \\ 0 \end{bmatrix} \qquad 36$$

The amplitudes A_1 and A_2 can be solved using Cramer's Rule. The determinant;

$$D = \{ [(-m_1\omega^2 + k_1 + k_2) + (c_1 + c_2)i\omega][(-m_2\omega^2 + k_2) + c_2 i\omega] - [-k_2 - c_2 i\omega][-k_2 - c_2 i\omega]\} \qquad 37$$

$$D = \{ [-m_1\omega^2 + (c_1 + c_2)i\omega + k_1 + k_2][-m_2\omega^2 + c_2 i\omega + k_2] - (c_2 i\omega + k_2)^2\} \qquad 38$$

And the amplitudes:

$$A_1 = \frac{\begin{bmatrix} F_0 & -k_2 - c_2 i\omega \\ 0 & (-m_2\omega^2 + k_2) + c_2 i\omega \end{bmatrix}}{D} = \frac{(-m_2\omega^2 + k_2) + c_2 i\omega}{D} F_0 \qquad 39$$

$$A_2 = \frac{\begin{bmatrix} (-m_1\omega^2 + k_1 + k_2) + (c_1 + c_2)i\omega & F_0 \\ -k_2 - c_2 i\omega & 0 \end{bmatrix}}{D} = \frac{-k_2 - c_2 i\omega}{D} F_0 \qquad 40$$

Equations 39 and 40 show that the displacements are linearly dependent on the excitation amplitude F_0. The steady state displacements of the system due to the harmonic excitation are:

$$|A_1(i\omega)| = \left|\frac{(-m_2\omega^2 + k_2) + c_2 i\omega}{D(i\omega)} F_0\right| \qquad 41$$

And;

$$|A_2(i\omega)| = \left|\frac{-k_2 - c_2 i\omega}{D(i\omega)} F_0\right| \qquad 42$$

In order to write equations 41 and 42 in a more convenient form, a set of new terms is defined as shown in table 1.

Table 1. Definition of Variables for TMD Optimization

Natural frequencies:	$\omega_1 = \sqrt{\frac{k_1}{m_1}}$, $\omega_2 = \sqrt{\frac{k_2}{m_2}}$
Mass ratio:	$\mu = \frac{m_2}{m_1}$
Frequency ratios:	$\gamma = \frac{\omega}{\omega_1}$, $\beta = \frac{\omega_2}{\omega_1}$
Damping ratios:	$\zeta_1 = \frac{c_1}{2\omega_1 m_1}$, $\zeta_2 = \frac{c_2}{2\omega_2 m_2}$
Primary system static displacement:	$\delta_{st} = \frac{F_0}{k_1}$

Substitution of these parameters to equation 41 and 42 leads to a useful non-dimensionalized form:

$$\left|\frac{A_1}{\delta_{st}}\right|(\gamma) = \frac{\sqrt{(\beta^2-\gamma^2)^2+(2\zeta_2\gamma\beta)^2}}{\sqrt{[(\gamma^2-\beta^2)(1-\gamma^2)+\mu\gamma^2\beta^2+4\zeta_1\zeta_2\gamma^2]^2+[2\zeta_2\beta\gamma(\gamma^2+\mu\gamma^2-1)+2\zeta_1\gamma(\gamma^2-\beta^2)]^2}} \quad 43$$

Equation 43 above only involves general parameters that are independent on the size of a TMD. Therefore, it is general and provides a good qualitative overview of the characteristics of the TMD and the specifications are as shown in table 2.

Table 2. Specifications for Optimization of TMD

m_2 = 150,000 kg	m_1 = 7500 kg
k_2 = 200,000 kN/m	k_1 = 10,000 kN/m
ω_2 = 36.5158 rad/s	ω_1 = 11.547 rad/s
0.00 ≤ t ≤ 100	0.5≤ f ≤1.5 rad/s

The values of the parameters above are taken from Datta, T.K. (2010).

B. Analytical Modeling of the system using Matlab Simulink

The schematic of the sub-sections of Main Sstructure and TMD are shown in figure 14

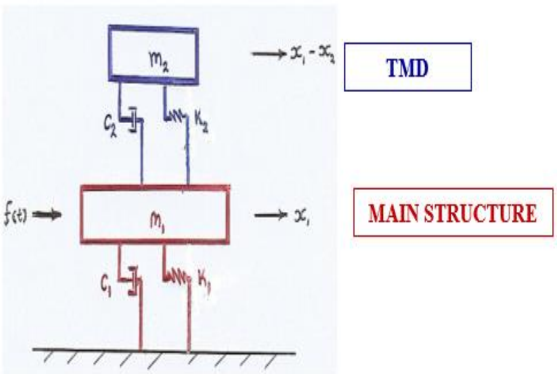


Fig. 14 Schematic of the Sub-sections of Main Structure and TMD

i. First analyzing the Main Structure

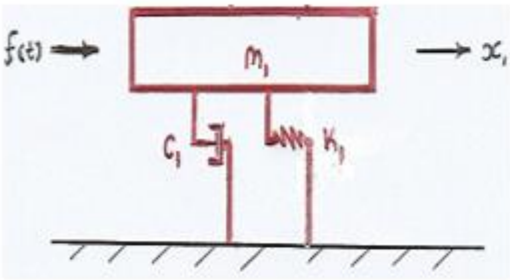


Fig. 15. Schematic of Main Structure

The equation of motion is expressed as:

$$m_1\ddot{x}_1 + c_1\dot{x}_1 + k_1x_1 = f(t) \quad 44$$

$$m_1d^2x_1/dt^2 + c_1dx_1/dt + k_1x_1 = f(t) \quad 45$$

ii. Analyzing Main Structure and TMD

In the second analysis, the vibration absorber (TMD) is added to the main structure as shown in figure 17

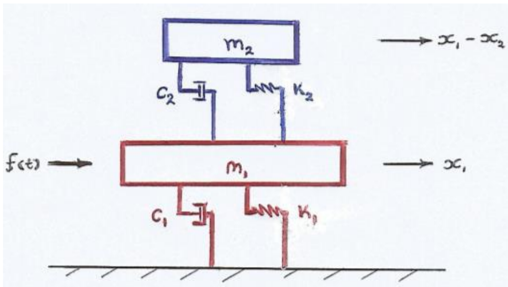


Fig. 16. Schematic of the main structure and TMD

The governing equation of motion is expressed as:

$$m_1\ddot{x}_1 + c_1\dot{x}_1 + k_1x_1 + c_2(\dot{x}_1-\dot{x}_2) + k_2(x_1-x_2) = f(t), \quad 46$$

$$m_2\ddot{x}_2 + c_2(\dot{x}_2-\dot{x}_1) + k_2(x_2-x_1) = 0 \quad 47$$

Modeling of the main structure and the TMD in Simulink.

$m_1d^2x_1/dt^2 + c_1(dx_1/dt + k_1x_1) + c_2(dx_1/dt - dx_2/dt) + k_2(x_1 - x_2) = f(t)$
$m_2d^2x_2/dt^2 + c_2(dx_2/dt - dx_1/dt) + k_2(x_2 - x_1) = 0$ 48

IV. RESULTS AND DISCUSSION

The simulation results that validate the effect of the TMD on the single degree of freedom (SDOF) and two degree of freedom systems of the frames structure subjected to wind induced excitation is presented. The dynamic response of the structure with and without the TMD is displayed to authenticate the efficacy of the TMD. Also presented is how the maximum displacement of the structure was minimized as the damper efficiently counteracts a force with a frequency slightly less than the structure's natural frequency. Figure 14 displays the MATLAB simulation of the main structure without the damper.

A. *Amplitude Responses*

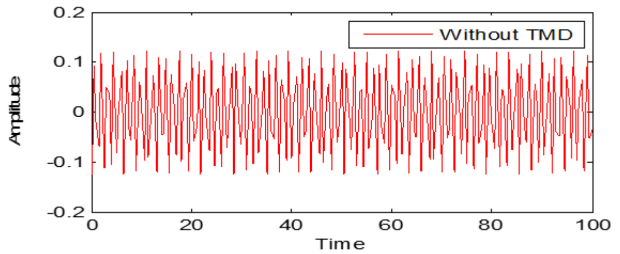


Fig. 17. Maximum Displacement of MATLAB simulation of the structure without TMD.

Figure 18 below shows the potency of the TMD when applied to reduce the effects of induced vibration. The optimal parameters of the TMD based on the specifications in Table 2 were calculated using a MATLAB program. The program performed much iteration using a larger step size.

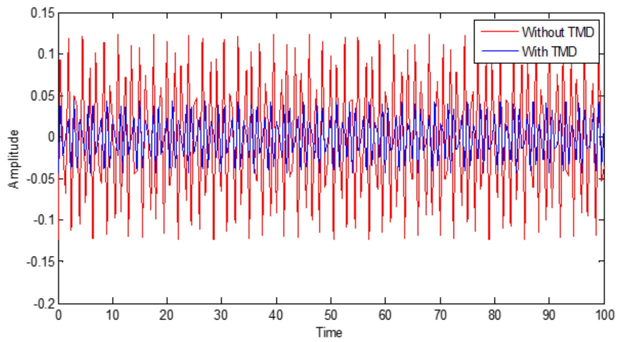


Fig. 18 Maximum Displacement of the Structure with and without TMD.

It is observed that due to the introduction of the TMD the resonance peak corresponding to the period of the structure is minimized. The peak with higher amplitude corresponds to the structure and the other corresponds to the TMD. The TMD in this simulation example is lightly damped with a damping ratio of 5%.

B. *Maximum Displacement vs Frequency*

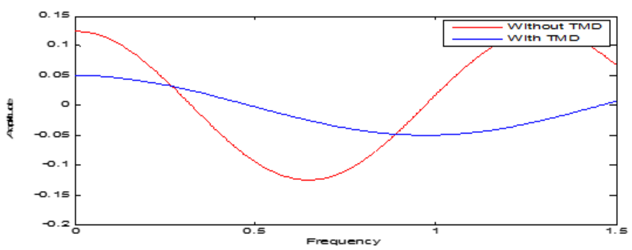


Fig. 19. Maximum Displacement of the Structure with and without TMD vs. Frequency.

As an effective harmonic device that reduces dynamic responses of the steel jacket structure by altering the fundamental frequency of the structure, the maximum displacement was plotted against the frequency as shown in figure 19.

The figure above shows the response of the maximum displacement as a function of the frequency. It is observed that when the damping ratio of the TMD increases, there is a corresponding change in the structural response.

The maximum displacement is reduced when the TMD effectively act against a force with a frequency that is slightly less than the natural frequency of the main structure. It can be seen from the graph that when f = 1, the maximum displacement of the structure occurs around 0.125 without the damper, but with the introduction of the TMD the maximum displacement occurs around 0.045.

There is a clear vibration attenuation effect of the TMD control, especially around the 1.45 Hz, because the resonance reaction of the main structure occurred around this frequency domain. Thus, with the TMD, it is remarkably effectual in mitigating vibration response on the main structure; particularly the considerable reduction of the peak responses around the 1.45 Hz frequency domain.

V. CONCLUSION

A careful study on the efficacy of Tuned Mass Damper and practical methods of mitigating dynamic response remains a work in progress. It is conspicuous that the harmonic response of a system can be adjusted by annexing one or more secondary vibrating systems to it. The Tuned Mass Damper has proven efficient in mitigating the dynamic response of structures. Numerical analysis was performed in MATLAB to better design the TMD to mitigate vibrations for the steel jacket platform.

The amplitude and time responses indicated that the displacement significantly decreased for the offshore platform while the TMD system was applied.

Tuned mass damper are designed to reduce responses on structures, this research is made to study the efficacy of using the mass damper based on MATLAB for mitigating vibration of the steel jacket structure due to wind induce excitation force. From the simulation results, it shows that the response of the structure subjected to excitation force system is relatively higher without tuned mass damper which shows the efficacy of TDM in mitigating the vibration on the structure. It is also observed that the displacement response is decreased by increasing damping ratio of TMD.

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