Design of a Test Rig for Vibration Control of Oil Platforms using Magnetorheological Dampers

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Abstract

Offshore steel structures are subjected to hydrodynamic forces. These forces induce vibration in the structure, which in turn has a great effect on the safety and durability of the structure and the manpower operating it. Magnetorheological (MR) dampers are proven as a feasible alternative to reducing structural vibrations. This paper presents the initial investigation of integrating MR dampers in Offshore Steel Jacket Platforms to decrease the vibrations discussed. A Steel Jacket Structure with MR damper is simulated using Simulink and the results are presented. In addition, an experimental model is designed to verify the simulation responses.

1. Introduction

Offshore platforms have been developed over the years from traditionally shallow water structures to modern deep sea structures. They are commonly used for oil and gas extraction. Although there are many different types of offshore structures the steel jacket type platform on a pile foundation is the most commonly used [1]. These structures, normally anchored to the sea bed, are affected by wave forces, unlike onshore structures which incur vibrations due to seismic excitations and excessive random shocks [1, 2]. The flexibility of the steel jacket platforms generates self-excited non-linear hydro dynamic forces in addition to their non-linear response to large deformations [3]. Although the risk of failure or damage in such structures is higher than experienced in other structures the safety of these can usually be insured by increasing their stiffness so as to shift the natural frequencies away from the resonant range of frequencies. Such technique is costly and requires high maintenance for large offshore structures [4].

Several models for offshore structures are available in the literature; the majority of which are derived using finite elements methods. The mass matrix in finite element analysis is usually calculated using the consistent mass method or the lumped mass method. In either case, calculating the mass matrix is time consuming and the resultant matrix tends to be very large for control purposes. Another technique developed by Horr et al. [5] makes use of the spectral element method to derive the element matrices and uses the Timoshenko beam theory. This method calculates the distributed mass exactly and only one element needs to be placed between two nodes. The Euler-Bernoulli theory [6] was shown to be inadequate for the prediction of higher modes of vibration and also with beams where the effect of cross-sectional dimensions is not negligible; therefore, an alternate technique should be used to derive a more accurate model of the structure. Timoshenko beam theory [5] takes into account the effects of rotary inertia and shear deformations. If shear deformation is significant, the assumption that plane sections remain plane is no longer valid. Therefore, Timoshenko corrections are necessary to make more accurate predictions of high modes of vibration. In this work, the calibrated spectral Timoshenko pipe element [5] is used to model an offshore oil platform. The goal of this research is to develop an accurate model of the platform and to reduce its vibrations using MR dampers.

Traditionally, over the last several decades, MR dampers have been used extensively in reducing the vibrations of civil engineering structures [7, 8, 9, 10, 11], automobiles, and household appliances. MR dampers have a very important advantage [7]; they can provide large force outputs for a small change in current input. Moreover, these dampers can be actively used to systematically reduce or eliminate the vibrations of structures. Offshore oil platforms are known to undergo a lot of vibration during extreme weather conditions and when strong waves and currents hit the platform [8]. MR dampers are capable of providing the large damping forces [9] necessary to absorb the effect of the hydro-dynamic forces on the platform. To validate the theoretical findings, an experimental setup is built and tested in our laboratory.

2. Dynamic Models

In order to simulate the system understudy, the theoretical models of both devices – the MR damper and the Offshore Steel Jacket Platform are presented and discussed in the next section.

2.1 MR Damper Model

MR dampers are highlighted to be highly non-linear devices, and when subjected to a sinusoidal load, exhibit bi-viscous, hysteretic and non-linear behavior [9, 10, 12]. Several models are used to describe the MR damper response including the Bouc-Wen model, Bingham model and the bi-viscous hysteresis model [12]. Other models for MR dampers are also available see for example [7, 9, 12, 13]. These models are usually used in conjunction with a control scheme to dampen out the vibration of an engineering structure which has similarities to the Bouc-Wen model, and has been verified to be reliable in reducing vibrations [12]. The Bouc-Wen model in Figure 1 has been shown to accurately predict the behavior of a prototype MR damper over a broad range of inputs [9 and 14]. This model is numerically tractable and has been used extensively for modeling hysteretic systems. It is extremely versatile and can exhibit a wide variety of hysteretic behavior.

The equations governing the force f predicted by this model are given by:

$$\boldsymbol{f = c_1\dot{y} + k_1(x - x_0)} \tag{1}$$

The evolutionary variable z is governed by:

$$\boldsymbol{\dot{z} = -\gamma|\dot{x} - \dot{y}|z|z|^{n-1} - \beta(\dot{z} - \dot{y})|z|^n + A(\dot{x} - \dot{y})} \tag{2}$$

and

$$\boldsymbol{\dot{y} = \frac{1}{(c_0 + c_1)}\{\alpha z + c_0\dot{x} + k_0(x - y)\}} \tag{3}$$

The accumulator stiffness is represented by k_{d1}, and the viscous damping for large velocities is denoted by c_{d0i}. A dashpot, represented by $c_{\,d1i}$, is included in the model to introduce the nonlinear roll-off in the force-velocity loops at low frequencies. Similarly, a spring with constant k_{d0} is included in the model to control the stiffness at large velocities. It is assumed that the initial deflection of the spring, whose constant is k_{d1}, is x_{d0}. Consider the modified Bouc-Wen model depicted in Figure 1. The displacement of the i^{th} damper, (i =1, 2), can be expressed in terms of the displacement w_i and x_{ti} as follows:

$$\boldsymbol{w_i = x - b_i\theta \qquad i = 1,2} \tag{4}$$

$$\boldsymbol{\dot{e}_i = \frac{1}{c_{d0i} + c_{dli}}\{\alpha_i\varphi_i + c_{d0i}\dot{w}_i + k_{d0}(w_i - e_i) + c_{dli}\dot{x}_{ti}\}, \quad (i = 1.2)} \tag{5}$$

where the evolutionary variable φ_i is governed by [7]:

$$\boldsymbol{\dot{\varphi}_i = -\gamma\left|(\dot{w}_i - \dot{e}_i)\varphi_i^{n-1}\right|\varphi_i - \beta(\dot{w}_i - \dot{e}_i)|\varphi_i|^n + \delta(\dot{w}_i - \dot{e}_i), \qquad (i = 1,2)} \tag{6}$$

and w_i and x_{ti} are the displacement of the front and back nodes of the structure where the i^{th} damper is placed. The constants γ, β, n, and δ are given in [9]. Referring to the schematic diagram shown in Figure 1, the i^{th} damper force, F_i, for i =1, 2, can be written as:

$$\boldsymbol{F_i = \alpha_i\varphi_i + c_{d0i}(\dot{w}_i - \dot{e}_i) + k_{d0}(w_i - e_i) + k_{d1}(w_i - x_{ti}) - k_{d1}x_{d0}} \tag{7}$$

where

α_i= α_a+ α_b V_i (8a)

c_d0i= c_d0a+ c_d0b V_i i =1, 2 (8b)

c_d1i= c_d1a+ c_d1b V_i (8c)

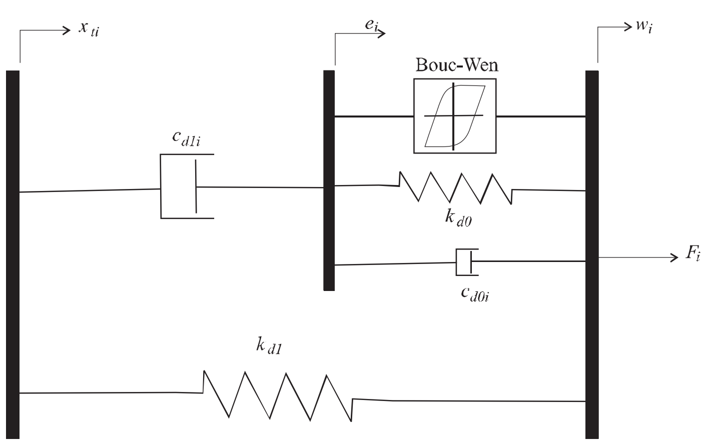


Figure 1. Simple Mechanical Models of the MR Damper.

The values of the constants α_a, α_b, c_{d0a}, c_{d0b}, c_{d1a}, and c_{d1b} are listed in [9]. The variables α_i, c_{d0i}, and c_{d1i} are functions of the current driver voltage output, V_i as it is indicated in (8a,b, and c). However, for control purposes, it is better to express these variables in terms of the applied voltage v_i. The voltage applied to the i^{th} current driver, v_i, is related to the output voltage, V_i, as follows:

$$\dot{V}_i = -\eta\,(V_i - v_i),\; i = 1,2 \tag{9}$$

Equations (8)-(9) are used to obtain the variables α_i, c_{d1i} and c_{d0i}, as a function of the applied voltage v_i. The selection of the voltage v_i is described in the next section. The reader can refer to [7] for more details. To take full advantage of the unique features of the MR damper for the offshore platform, the Bouc-Wen model has been used to accurately reproduce the behavior of the MR damper.

2.2 Offshore Steel Jacket Platform

Several models for offshore structures have been developed where mostly they were derived using finite element method. Another technique that is less time consuming has been developed by [5]. This method is based on using the spectral element method to derive the element matrices and uses the Timoshenko beam theory. The spectral element method concerns the synthesis of wave forms from the superposition of many frequency components [6, 15, 16, 17]. The spectral approach used was based on the exact solution of the governing differential equation for the dynamic motion of structures which differs from the classical method by using Fast Fourier Transform for conversion purposes. In the work, it is proposed to theoretically and experimentally verify the design and control of the vibrations of steel jacket platforms caused by hydrodynamic forces.

Both the linear and non-linear forms of the Morison's equations have been used in the literature and comparative studies are available in [2]. It was shown that the non-linear Morison equation provides superior results in particular when the wave frequencies are near the first mode of the offshore structure [2]. Motivated by the work of [5] a dynamic model based on Timoshenko Pipe Element is developed for an offshore structure.

The dynamic response of a structure is a general function of space and time. If the time variation of the solution is focused on at a particular point in space, then the spectral representation of field variable is shown in (10):

$$u_1(x,t) = \sum \hat{u}_n(x,\omega_n)e^{i\omega_n t} \tag{10}$$

where ω represents the angular frequency and $\hat{u}_n$ are the Fourier coefficients. The governing differential equations in structural dynamics are given in term of both space and time derivatives.

The general form of the one dimensional, homogeneous differential equation can be expressed as [17]:

$$u + c_1\frac{\partial u}{\partial x} + c_2\frac{\partial u}{\partial t} + c_3\frac{\partial^2 u}{\partial x^2} + c_4\frac{\partial^2 u}{\partial t^2} + c_5\frac{\partial^2 u}{\partial x \partial t} + ... = 0 \tag{11}$$

where c_1, c_2, … can be functions of position but not time. The differential equation can be represented by:

$$\left[\sum_n \left\{\hat{u}_n + c_1\frac{d\hat{u}_n}{dx} + c_2 i\hat{u}_n + c_3\frac{d^2\hat{u}_n}{dx^2} + c_4(i\omega_n)^2\hat{u}_n + c_5(i\omega)\frac{d\hat{u}_n}{dx} + ...\right\}e^{j\omega_n t} = 0\right] \tag{12}$$

This summation must be true for each value of *n*, and hence there are n equations as:

$$\begin{aligned} &\left[1 + (i\omega_n)c_2 + (i\omega_n)^2 c_4 + ...\right]\hat{u}_n + \left[c1 + (i\omega_n)c_5 + ...\right]\frac{d\hat{u}_n}{dx} \\ &+ \left[c3 + ...\right]\frac{d^2\hat{u}_n}{dx^2} + ... = 0 \end{aligned} \tag{13}$$

which can be written as [18]:

$$C_1(x,\omega_n)\hat{u}_n + C_2(x,\omega_n)\frac{d\hat{u}_n}{dx} + C_3(x,\omega_n)\frac{d^2\hat{u}_n}{dx^2} + ... = 0 \tag{14}$$

where coefficients C_1, C_2, … are complex values. The summation of n frequency components reconstructs the time dependency for the response. For a linear differential equation with constant coefficients, the spectral solution can have the form Ce^{ikx}, where k is called the wave number and is obtained by solving the characteristic equation,

$$C_1 + C_2 ik + C_3 (ik)^2 + ... = 0 \tag{15}$$

The complete spectral solution of any differential equation is obtained by summing all modes for each frequency values [6, 19, 20, 21]:

$$u(x,t) = \sum_n (C_{1n} e^{ik_{1n}x} + C_{2n} e^{ik_{2n}x} + ...C_{mn} e^{ik_{mn}x}) e^{i\omega_n t} \tag{16}$$

Therefore, the general solution for a differential equation with non constant coefficient can be expressed as [22]:

$$u(x,t) = \sum_n A_n T(k_{mn} x) e^{i\omega_n t} \tag{17}$$

where A_n is an amplitude spectrum and T is the system transfer function.

The shear-deflection effect is a significant parameter in the flexural vibration of a short member of a tubular structure, thus it must be included. This requires some modifications in the conventional finite element formulation. This is accomplished by introducing a shear coefficient, defined as the ratio of the actual beam cross-sectional area to the effective area resisting shear deformation [23], in the stiffness matrix. The element shear stiffness generally decreases with increasing value of the shear coefficient. As the ratio of the radius of gyration of the pipe member cross-section to its length (r/L) becomes small compared to unity, the significance of this coefficient decreases.

In the conventional beam theory, it is assumed that the cross-sectional dimensions of a beam element were small in comparison with its length. For the purpose of including the effects of the cross section dimensions on the frequency, [23] has included some corrections to the theory given by Timoshenko. This addition of shear deformation leads to the invalidity of the Euler Bernoulli theory that plane sections remain plane. This means that the slope ф of any section along the length of the beam simply cannot be obtained by differentiation of the transverse displacement v. This then creates two independent motions ф , v. These corrections are important when studying the modes of vibrations of higher frequencies [22] when a vibrating beam is subdivided into comparatively short length portions. Timoshenko [23] gave the equation of motion for a beam with tubular section as:

$$\rho A \frac{\partial^2 v}{\partial t^2} - KAG\left(\frac{\partial^2 v}{\partial x^2} - \frac{\partial \phi}{\partial x}\right) = 0 \tag{18}$$

$$E_1 \frac{\partial^2 \phi}{\partial x^2} + KAG\left(\frac{\partial v}{\partial x} - \phi\right) - I\rho \frac{\partial^2 \phi}{\partial t^2} = 0 \tag{19}$$

where K is a factor depending on the shape of the cross section and G is the modulus of shear rigidity. The shear force and bending moments acting on any section of beam are:

$$\boldsymbol{V} = \boldsymbol{KAG}\left(\frac{\partial v}{\partial x} - \emptyset\right) = -\boldsymbol{El}\frac{\partial^2 \emptyset}{\partial x^2} + \boldsymbol{l\rho}\frac{\partial^2 \emptyset}{\partial t^2} \tag{20a}$$

$$\boldsymbol{M} = \boldsymbol{El}\frac{\partial \emptyset}{\partial x} \tag{20b}$$

The spectral solution may be found as [15, 24, 25]:

$$\begin{aligned} v(x,t) &= \hat{v}_1(x,\omega_1) e^{-i(kx-\omega_1 t)} + \hat{v}_2(x,\omega_2) e^{-i(kx-\omega_2 t)} + ... \\ &+ \hat{v}_n(x,\omega_n) e^{-i(kx-\omega_n t)} = \sum v_0 \, e^{-i(kx-\omega t)} \end{aligned} \tag{21}$$

$$\begin{aligned} \phi(x,t) &= \hat{\phi}_1(x,\omega_1) e^{-i(kx-\omega_1 t)} + \hat{\phi}_2(x,\omega_2) e^{-i(kx-\omega_2 t)} + ... \\ &+ \hat{\phi}_n(x,\omega_n) e^{-i(kx-\omega_n t)} = \sum \phi_0 \, e^{-i(kx-\omega t)} \end{aligned} \tag{22}$$

These solutions may be substituted into the equations of motions,

$$(GAKk^2 - \rho A\omega^2) v_o - iGAKk\emptyset_0 = 0 \tag{23a}$$

$$iGAKkv_0 + (Elk^2 + GAK - \rho l\omega^2)\emptyset_0 = 0 \tag{23b}$$

which result in the following relationship between the motions for each mode as:

$$v_0 = \left[\frac{GAKki}{GAKk^2 - \rho A \omega^2} \right] \phi_0 \tag{24}$$

where

$$k = \pm \left[\frac{1}{2}\omega^2 \left\{ \left(\frac{1}{C_3}\right)^2 + \left(\frac{C_1}{C_2}\right)^2 \right\} \pm \sqrt{\left(\frac{\omega}{C_2}\right)^2 + \frac{1}{4}\omega^4 \left\{ \left(\frac{1}{C_3}\right) - \left(\frac{C_1}{C_2}\right)^2 \right\}^2 } \right]^{1/2} \tag{25}$$

and

$$C_1 = \sqrt{\frac{\rho I}{\rho A}}; \quad C_2 = \sqrt{\frac{EI}{\rho A}}; \quad C_3 = \sqrt{\frac{GAK}{\rho A}}$$

Since both motions are independent, it is possible to consider one of them as unknown:

$$\begin{aligned} \hat{\phi}(x) &= Ae^{-ik_1x} + Be^{-ik_2x} + Ce^{ik_1x} + De^{ik_2(x} \\ \hat{v}(x) &= P_1Ae^{-ik_1x} + P_2Be^{-ik_2x} - P_1Ce^{ik_1x} - P_2De^{ik_2x} \end{aligned} \tag{26}$$

where A, B, C, D, P_1 and P_2 are constants. Equations (26) are the exact solutions to the governing Equations (18) and (19), thus it can be considered as a shape function. This results in the possibility of writing the four unknown coefficients in terms of nodal displacements. For the element of length L, with no load applied between the nodes, the spectral shape function can be written as [25]:

$$\begin{aligned} \hat{\phi}(x) &= Ae^{-ik_1x} + Be^{-ik_2x} + Ce^{ik_1(L-x)} + De^{ik_2(L-x)} \\ \hat{v}(x) &= P_1Ae^{-ik_1x} + P_2Be^{-ik_2x} - P_1Ce^{ik_1(L-x)} - P_2De^{ik_2(L-x)} \end{aligned} \tag{27}$$

$$\boldsymbol{v_1} = \boldsymbol{v}(\boldsymbol{0});\ \emptyset_1 = \emptyset(\boldsymbol{0}); \boldsymbol{v_2} = \boldsymbol{v}(\boldsymbol{L});\ \emptyset_2 = \emptyset(\boldsymbol{L}) \tag{28}$$

It can be shown that the equation of motion of the structure using the Timoshenko beam elements can be written as follows:

$$\boldsymbol{M\ddot{x}} + \boldsymbol{C\dot{x}} + \boldsymbol{Kx} = \boldsymbol{Bu} + \boldsymbol{W_F} \tag{29}$$

where M , C, and K represent the global mass, damping and stiffness coefficients, respectively. u is the MR damper force and W_F is the hydrodynamic force applied on the structure. Or in state space format:

$$\boldsymbol{\dot{x}} = \boldsymbol{Ax} + \boldsymbol{Bu} + \boldsymbol{g}(\boldsymbol{x}, \boldsymbol{t}) \tag{30}$$

where x is the state vector containing the nodal displacements and velocities.

2.3 Hydrodynamic Forces

The difficulties in the dynamic response analyses of such structures arise from the random nature of loading, the non linearity of the drag-dominated wave forces, and the nonlinear relation between such forces and the response structure. The wave induced nonlinear model proposed by Morison [26] was linearized by [27]. This linearization reduced drastically the analytical and conceptual solution time. Stavros [28] investigated the use of Morison's equation in estimating the response of an offshore structural model and concluded that amplified equivalent damping factors should be used on the results from the linearized displacement to match the nonlinear response. Other researchers [2, 29, 30, 31] who studied the wave induced forces derived from the linear form of Morison's equation. But only the latter has studied the validity and accuracy of the linear form of Morison's in estimating the wave induced forces as compared to non linear form.

The Morison equation for a horizontal wave induced force acting at a joint p in an offshore structure, using Morison's equation for a unidirectional plane waves in the presence of wind-induced water currents is:

$$\boldsymbol{Fp} = \boldsymbol{0.5\rho C_D A_p}\left|\boldsymbol{\dot{U}_x} - \boldsymbol{\dot{U}_p}\right| + \boldsymbol{\rho C_I B_p \ddot{U}_x} - \boldsymbol{\rho(C_I - 1)B_p\ddot{U}_p} \tag{31}$$

in which $U_p, \dot{U}_p, and\ \ddot{U}_p$ are, respectively, the horizontal displacement, velocity and acceleration at joint p; $U_x, \dot{U}_x, and\ \ddot{U}_x$, are respectively the horizontal displacement, velocity and acceleration of water at joint p; A_p is the lumped area dedicated to joint p; B_p is the lumped volume dedicated to joint p; C_D is the drag coefficient; C_I is the inertia coefficient and ρ is the water density.

The horizontal water velocity and acceleration depend on the wave field characteristics. Considering single fixed wavelength, the corresponding horizontal water velocity and acceleration at joint p in a structure are defined as:

$$\dot{U}_x = E_p \cos\left(kx_p - \omega t\right) + U_{ow}\left(\frac{y_p}{h}\right) \tag{32}$$

$$\ddot{U}_x = E_p \cos\left(kx_p - \omega t\right) \tag{33}$$

$$E_p = 0.5\omega H x \frac{\cosh(ky_p)}{\sinh(kh)} \tag{34}$$

where y_p is the elevation at joint p from the sea-bed; h is the water depth; ω is the wave frequency; H is the wave height; k is the wave number (2π/L) where L is the wavelength; and U_{ow} is the current velocity at the water surface level which is usually 1% of the wind speed at a height of 10 m above the water surface. Using the linear assumption derived by [32], and with some substitutions, the force can be written as:

$$F_p = F_{op}\sin\left(kx_p - \omega t + \varphi_p\right) - \left(\frac{8}{3\pi}\right)\rho A_p E_p \dot{U}_p + \left(\frac{4}{3\pi}\right)\rho A_p E_p U_{ow}\left(\frac{y_p}{h}\right) - \rho B_p (C_I - 1)\ddot{U}_p \tag{35}$$

This equation clearly shows that the damping and mass of the structure contribute to the wave induced forces. The term containing the nodal vector $-\rho B_p(C_I - 1)\ddot{U}_p$ is usually added to the mass matrix *M* and the term containing $-\left(\frac{8}{3\pi}\right)\rho A_p E_p \dot{U}_p$ is added to the damping matrix *C*.

The theoretical bases of the MR damper and the Offshore Oil Platform models has been presented. A combination of the Timoshenko Pipe Element model with the MR Bouc-Wen damper model will be investigated experimentally and digitally.

3. Simulation Results

In this section, the simulation results of two open loop systems will be presented. The two systems represent the offshore steel jacket platform and the MR damper, respectively.

3.1 Offshore Steel Jacket Open Loop System

A simple steel jacket platform shown in Figure 2 is used as an illustrative example. For which the wave height H is 0.05m, water depth h is 0.03m, wave length L is 1.33m, and current velocity of the water surface U_{ow} is 0.122m/s.

The performance of the system is simulated when no control is applied to the system and with the wave frequency of 1.8 rps applied to the structure. The simulations results of the equations of motions (30) and (34) with zero input from the MR damper for the three floors are shown in Figures 3a, b, and c. These responses are the solutions of the linearized Equation (30) due to the hydrodynamic force of Equation (34) and shown in Figure 4. The steady state displacement is harmonic due to the nature of the hydrodynamic force. Figures 5a, b, and c show the displacement of the three floors due to a pulse input to represent an arbitrary shock wave on the structure. It is clear that the structure is very stiff and very lightly damped. Therefore, the oscillations of the system must be damped out through a dissipation device such as the MR damper.

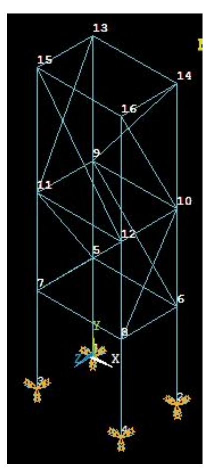
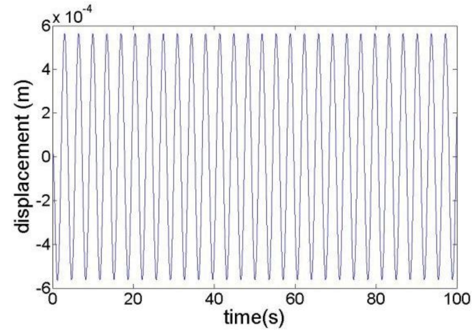
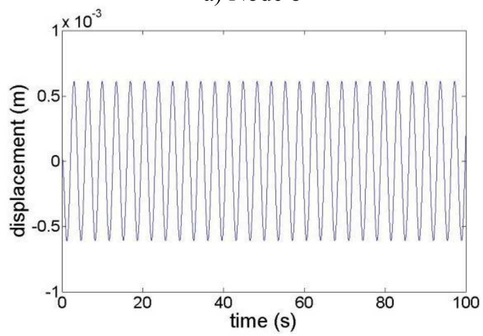


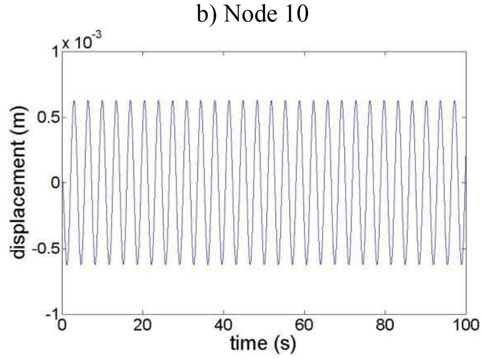
Figure 2. Diagram representing the position of the nodes.



a) Node 8



b) Node 10



c) Node 14

Figure 3. Floor displacement at nodes 8, 10 and 14.

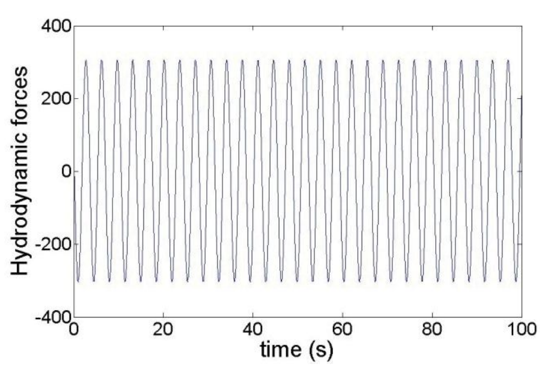
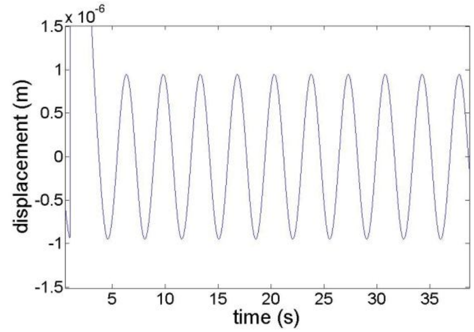
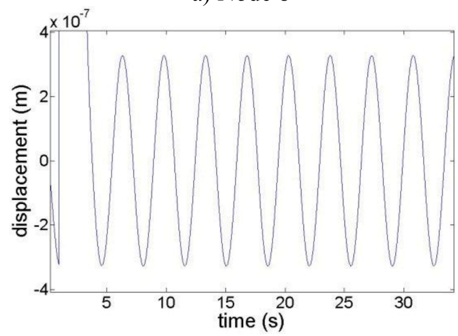


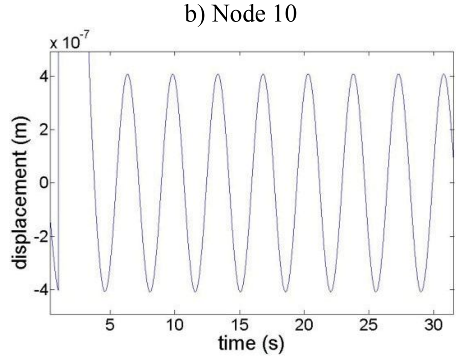
Figure 4. Hydrodynamic forces.



a) Node 8



b) Node 10



c) Node 14

Figure 5. Displacement at nodes 8, 10 and 14.

3.2 MR Damper Open Loop Model

The response of the MR damper due to a 1Hz sinusoid with an amplitude of 1 applied as a displacement to the damper and the result is shown in Figure 6. The damper is voltage is assumed to be a pulse voltage of 100V. It can be seen that the applied voltage significantly alters the characteristics of the damper force.

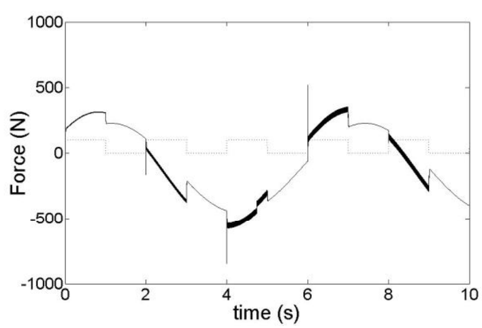


Figure 6. Response of a MR Damper.

4. Experimental Model

To validate the theoretical findings, an experimental setup is built in our laboratory (Figure 7). The setup consists of two major parts: the steel structure and the wave generator (Figure 8). A computer equipped with a data acquisition system is used to collect data from several accelerometers which are shown in Figure 9 strategically placed on the structure.

A controller is used to manipulate the accelerometers' data and output the correct voltage command for the MR damper. The controller voltage command is then transmitted to the MR damper through a voltage to current converter to produce the necessary damping force. Correct implementation of the control law will reduce the vibration of the steel structure.

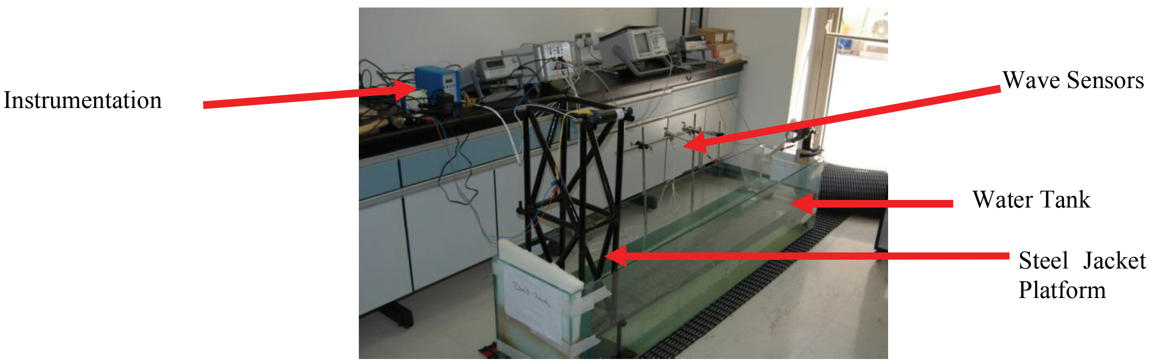


Figure 7. Experimental Model of the Overall Offshore Platform.

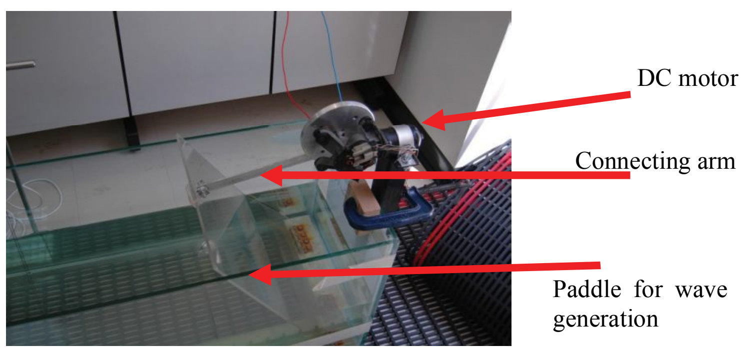


Figure 8. Experimental Wave Generator.

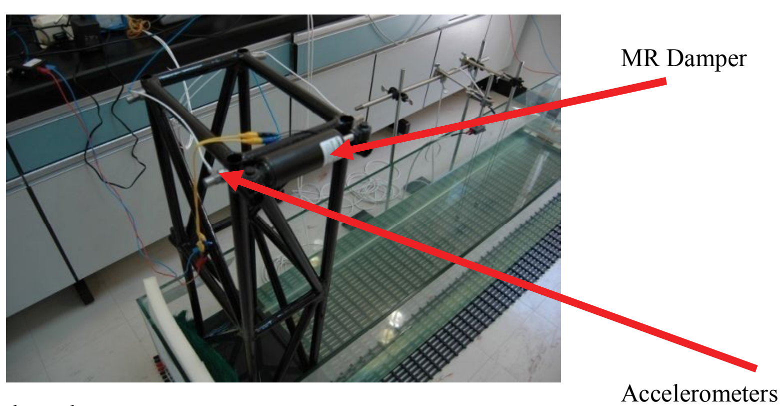


Figure 9. MR dampers and accelerometers.

5. Results and Discussion

A dynamic model for the steel jacket structure is derived using the Timoshenko beam theory. It is assumed that the vibrations of the structure are damped out using an MR damper. The model of the MR damper is presented here based on the Bouc-Wen model. The source of vibration of the structure is the effect of the hydrodynamically induced forces. The wave equations are developed based on the Morison's equation for a unidirectional plane wave. The model for the structure is simulated and the displacement responses of the three floors are presented here. An experimental setup is built in our laboratory to validate the theoretical findings. The setup is almost ready for data collection. The data will be provided in due time.

6. Conclusions

The accuracy of the dynamic models of large civil engineering structures depends on the modeling technique used, the assumptions made, and the accuracy of the parameters of the system. In this work a modeling technique known as spectral elements method is used to model the dynamic behavior of an offshore structure under the effect of hydrodynamic force. MR dampers are actively used to reduce the vibrations of the platform. These are smart actuators whose output force varies with the applied current. Moreover, MR dampers are capable of providing large forces for a relatively small input current. An experimental setup is built in our laboratory to validate the theoretical model and to implement the designed controller. The preliminary experimental results are promising and show agreement between the first four modes of vibrations.

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