Tension Leg Platform Design Optimization for Vortex Induced Vibration

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Abstract- **Tension Leg Platform design is a challenging and popular area of research in the offshore oil industry. In order to compete in the International Student Offshore Design Competition (ISODC), a Tension Leg Platform (TLP) was designed. Our TLP design addresses five fundamental areas of technical competency (General Arrangement and Overall Hull/System Design, Weight, Buoyancy and Stability, Global Loading, General Strength and Structural Design, Risk Assessment) and three specialized areas of technical competency unique to a Vortex Induced Vibration (VIV) optimized design (Hydrodynamics of Motions and Loading, Fatigue Strength, and Structural Analysis: global and local strength).**

Our design optimization process begins with a four-caisson, four-pontoon tension leg platform, operating at a depth of 3,000 ft. Hydrostatic and hydrodynamic analysis for design iterations are performed by our own MATLAB script, which calculates the effects of motions due to Vortex Induced Vibration (VIV). Structural analysis addresses fatigue loading from VIV. Our design includes risk-based analysis and conforms to class society rules and regulations. VIV phenomena cause uncontrollable motions of offshore platforms, as well as fatigue damage and failure of components such as cables and risers. The effects of VIV need to be addressed early in the design process to avoid costly platform damage and costly retrofits, such as hydrodynamic strakes for platform tendons.

I. INTRODUCTION

The offshore industry encompasses those structures which are engineered specifically for the deeper ocean, as opposed to those marine structures, like boats, which are used in any body of water. An oil rig is a primary example of such a structure. Because the environment for which offshore engineers are designing can be so hostile, the constraints and safety measures which govern the design are crucial. These structures are located in the mid Gulf of Mexico where dangerous hurricanes and rogue current eddies are a constant menace, and for the North Atlantic and Pacific where wave heights and sea states are so extreme that often the structure must be designed to operate autonomously because it is too dangerous to risk the personnel. The offshore industry, although challenging and often stressful, is a very exciting and cutting-edge field to be a part of.

Offshore drilling began over 50 years ago, and the challenges that engineers working in this area are presented with are extremely complex and difficult. Because of this, companies who exist in this sector of our economy, require highly skilled engineers and scientists. It is therefore in the best interests of these companies, mostly oil companies, to encourage young professionals and engineering students to get involved with offshore design. The International Student Offshore Design Competition (ISODC), an offshore platform design competition sponsored by the Society of Naval Architects and Marine Engineers (SNAME) as well as the American Society of Mechanical Engineers (ASME), is a means to achieve this goal.

A team from the Department of Ocean Engineering of the Massachusetts Institute of Technology is entering a design of a Tension Leg Platform (TLP) that is optimized for Vortex Induced Vibration (VIV).

II. PRELIMINARY RESEARCH PHASE: CHOOSING AN EXISTING VESSEL AND FIELD

The starting point for the design of our TLP, nicknamed “Tim,” was to determine which field and what kind of production field we were targeting. This process led us to an understanding of the range of water depths and operating conditions in which the TLP is found to be economically and operationally viable. The choice was made to model our design, at least preliminarily, after an existing larger TLP. Shell Deep Water Development’s “Brutus” was chosen. Brutus encompasses two leases approximately 265 kilometers (165 miles) southwest of New Orleans in water depths ranging from 838 to 1,005 meters (2750 to 3300 feet). The estimated gross recovery from the development is 230 million barrels of oil equivalent with a 70:30 oil to gas ratio. The project cost the company approximately $750 million with ¾ of that going to the fabrication and installation of the TLP, and the rest is associated with drilling. Brutus went into service in August of 2001 [1].

III. FUNDAMENTAL DESIGN AREAS

A. General Arrangement and Overall Hull System Design

A Tension Leg Platform (TLP) concept was selected for our offshore platform design because it has cost and station keeping properties that make it an appropriate and viable design for deep water applications [2]. A TLP is a compliant, free-floating offshore platform concept. Unlike fixed offshore platforms, compliant platforms respond to external effects with motions. Mooring systems control these motions. A TLP is compliant in the horizontal degrees of freedom, surge and sway. In the vertical degrees of freedom, a TLP is fixed. The feature that distinguishes a TLP from other moored platform concepts

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is its reserve buoyancy. Because the buoyancy of a TLP exceeds its weight, vertical moorings called “tendons” keep the TLP vertically stable and control heave motions. The cost of TLPs does not significantly increase with depth, because most of the steel in the structure is in the hull, which only extends to a finite depth. This is not the case with offshore structures such as towers, piled towers and jackets [2].

The “Tim” TLP design is based on Shell’s Brutus TLP in the Gulf of Mexico. The main components of both Tim and Brutus are the deck, hull, and mooring system. The deck supports accommodations, working area, processing equipment, derricks, cranes, pumps, the helideck, and control room of the TLP. The hull consists of four hollow cylindrical caissons and horizontal pontoons. The hull houses bilge and ballast systems, drilling and potable water, diesel fuel, pumps, and machinery. Caissons and pontoons provide buoyancy for the hull, and caisson spacing influences platform motions response. A four-caissoned square TLP is simpler to build in a shipyard than other geometric configurations, allows for a large deck area, and good stability features [2].

The mooring system consists of three thin walled, tubular steel tendons on each caisson, for a total of twelve tendons. Foundations (tension piles in our design, gravity base structures in some other TLP designs) anchor each tendon in place. The foundations, and subsequently the mooring, are permanent [2].

B. *Weight, Buoyancy and Stability*

As with any naval architectural project, keeping running tabs on the weight and placement of the systems being incorporated into the design is critical. Being that we weren’t able to do the detailed design of the components of the superstructure, we resorted to asking the creators of Brutus for an outline of the major weights which make up the TLP hull, deck and topsides. Our gracious ‘resource’ at Shell, Peter Young, provided us with the abbreviated weight and balance spreadsheet for Brutus. The major weight contributors were the hull structural components, including the tendon system, weighing in at 12,247 metric tons (~13,000 long tons). The next largest components were the Drilling (1,927 metric tons), Power (1,927 metric tons), Process (2,494 metric tons), Quarters (1,973 metric tons), Wellbay modules (3,220 metric tons), and the Drilling Packages (2,585 metric tons). We took these numbers for granted as the same for Tim. Table 1 outlines the centers of gravity and flotation (buoyancy) in the transverse (North-South, East-West) directions as well as the vertical direction.

TABLE 1
VESSEL CENTERS OF BUOYANCY AND FLOTATION

	East-West	North-South	Vertical
CG (m)	-0.27	0.21	45.1
CB (m)	0.64	0.49	13.1

With the following basic geometric parameters listed in Table 2, we generated the displacement and buoyancy characteristics of the vessel. The first few times the figures were determined, they were done by hand, after that, a Matlab script file was written to perform the calculations automatically. The main structural members which contribute to the buoyancy of the vessel were modeled as the geometric prism which looked most similar. The caissons were fully displacing hollow cylinders; the pontoons were hollow rectangular prisms, and the tendons were flooded hollow cylinders. The vessel total weight/displacement is 42,421 metric tons (41,752 long tons). The displacement and buoyancy of the vessel, as predicted by the Matlab script we wrote, is 52,052 metric tons (51,230 long tons). Another useful parameter with respect to weight and balance is, of course, the waterplane area. In Tim’s case the waterplane area is 1,290 square meters (13,892.9 square feet). The stability of the vessel is discussed in detail in the “Dynamic Response Estimates” section.

C. *Global Loading, Strength, and Structural Design*

The global loads on the structure are weight, buoyancy, and wave and current loading. The structural components of the TLP are made of steel. The critical structural components of a TLP are the tendons, foundations, caissons and pontoons, connections between columns and pontoons, deck girders, and connections between the deck and pontoons. Because they are long columns, the tendons are subject to buckling. Tendon pre-tension is a static, permanent load on the TLP foundations. Environmental loads such as wave loads and currents are variable loads, and lateral inclination of the tendons causes lateral loads on the foundations. The TLP caissons and pontoons are orthogonally stiffened shells. The caisson shells have a cylindrical cross-section and the pontoon shells have a rectangular cross section. The stiffened shells are subject to buckling failure under compressive loads and yielding under tensile loads. The stringers and attached shell plate may buckle together, the panels themselves may buckle, or the shell plating may buckle locally, while the stiffeners remain stable. The deck girders, like the stiffened shells, may buckle or yield, but are not subject to external water pressure [3].

TABLE 2
WEIGHT AND GEOMETRIC DATA FOR BRUTUS: INITIAL DESIGN PARAMETERS

<table>
<tr><th colspan="2">Height (m)</th><th colspan="2">Weight (ton)</th><th colspan="2">Diameter (m)</th></tr>
<tr><td>Superstructure</td><td>37</td><td>Hull</td><td>55,577</td><td>Cylinder</td><td>20</td></tr>
<tr><td>Deck</td><td>12</td><td rowspan="2">Deck/ Superstructure</td><td rowspan="2">22,353</td><td>Tendon</td><td>0.81</td></tr>
<tr><td>Pontoon</td><td>51</td><td colspan="2"></td></tr>
<tr><td>Cylinder</td><td>7</td><td>Tendons</td><td>76,203</td><td colspan="2" rowspan="2">Design Draft (m)</td></tr>
<tr><td>Tendons</td><td>884</td><td>Pilings</td><td>996</td></tr>
<tr><td>Pilings</td><td>104</td><td colspan="2"></td><td colspan="2">19</td></tr>
</table>

D. Risk Assessment

Given the scale of engineering time and capital investments that are involved with a functioning offshore production platform, managing risk and reliability from the start is imperative to the success of the project. Assessing the risk associated with a system allows the project manager to select the most cost-efficient design based upon considered facility risks. The first step in managing risk is identifying the most prevalent sources of uncertainty, and, in many cases, associating probabilities of occurrence and costs with the various failure or near-failure situations. In the offshore industry, managing risk is very nearly enforced by whichever classing agency you are employing to certify your production vessel. As production projects in the Gulf of Mexico (GoM) move into deeper and deeper waters, costs and complexity have increased. Therefore, the current industry standards and practices for identifying and mitigating risk to the facilities, personnel and the environment are becoming insufficient. The conventional sources for risk assessment guidance in the GoM are the API (American Petroleum Institute) RP 14C, 14J, and 2A WSD [4].

Classification and inspection organizations, such as Det Norske Veritas (DNV) and the American Bureau of Shipping (ABS), are developing new tools, and enhancing the existing ones, to extend coverage over new sources of risk associated with deeper water projects and, specifically, with the design-accidental-loads and performance standards for the safety of critical elements. They hope to expand the use of more detailed risk-assessment techniques in order to provide a sufficient method of considering hazard scenarios and impact on personnel and facilities, thereby ensuring better documentation of design performance and improving future projects. The Minerals Management Service (MMS) estimates approximately $1 million extra dollars in additional costs as a result of executing the proper hazard analyses for new floating production systems. However, the use of risk-based reliability is extremely cost effective when you adequately consider the cost of a major catastrophe [5].

Determining risks and managing risks are two separate processes, once aware of your potential hazards, it is imperative that offshore engineer has a system which monitors the vessel operations so as to warn against impending problems. To ensure that a vessel, TLP in our case, is performing satisfactorily during operation, operators make use of barrier diagrams, Bow-Tie analysis and criticality reviews. Bow-tie analyses are where one connects a primary event with its potential consequences, threats, preventative measures and recovery measures. The operator must monitor the mechanical integrity of the vessel as well as the SHE (safety, health and environment) systems. Control measures, to prevent occurrences or mitigate problems, are linked to something called a platform SMS (safety management system). Most all operating platforms have one of these systems, in one form or another, and through them, they manage the key barriers to failure and the performance standards of the vessel [5].

In order to get a sense for the risk associated with operating the TLP "Tim," we researched a private risk-management consultancy firm named Noble Denton. The firm claims to have a quality team of analysts who are adept at implementing Failure Modes and Effects Analyses, HAZOP studies, fault and event tree analysis and cause consequence analysis. They also have an extensive database of offshore accidents which supports there analyses. Through their technical expertise, they can identify risks stemming from fire or explosions, stability, structural reliability, dropped objects, evacuations, escape and rescue procedures, pollution and smoke dispersion and to the personnel. From information provided on their web site [6], we were able to identify the following risks.

Collision Risk includes the physical arrangements for bringing on board or offloading supplies, etc. Installation Risk examines the potential threats to the assets and personnel which can arise as the vessel is being transported and/or installed. Heavy Lift Risks are related to the installation or maintenance to the superstructure and other systems. Other areas include Loss of Stability and Structural Reliability Risks, Dropped Object Risks, and Optimizing Subsea Engineering, Tow Risks, Lifeboat and Evacuation Risks, and Mooring System Reliability.

IV. SPECIALIZED DESIGN AREAS

A. Hydrodynamics of Motions and Loading

1). Tendon Design: VIV Analysis and VIVA Runs

Some of the more fatigue sensitive areas of an offshore structure are the mooring and production systems. The forces that this collection of tendons and risers are exposed to are understood and controlled to a much lesser extent than those in the hull and superstructure or pilings. The forces they see are related to the random set of currents and environmental situations that will occur over the life of the system, and have nothing to do with the engineering or construction of the vessel.

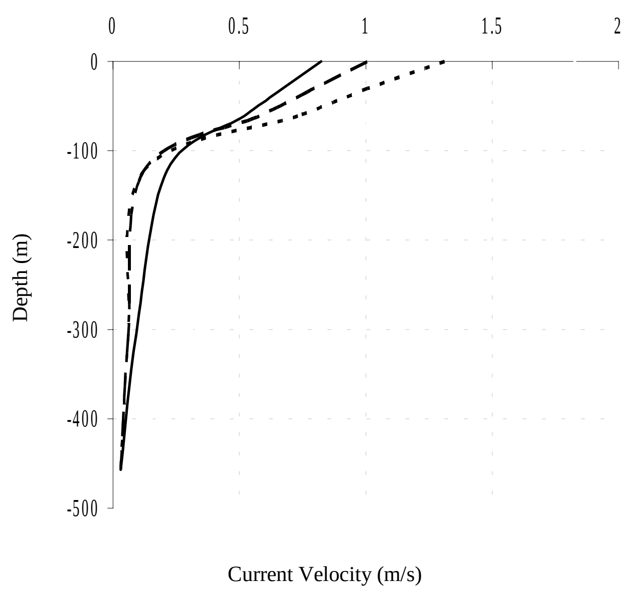

Fig. 1. Current profiles as a function of depth

TABLE 3
VIVA INPUT AND OUTPUT FILES AND DESCRIPTIONS

Input Files	Description
Risdyn.dat	Tendon Tension, current profile, damping coefficients
Rispre.dat	Riser Properties: length, number of segments, diameters, etc.
Test.dat	Boundary conditions
Risfat.dat	Fatigue constants (A,B) and stress concentration factor
Mode-def	Tells VIVA whether to calculate natural frequencies or if they are specified
Output Files	
Bend.dat	Stress response and bending moments as a function of depth
Bend-mm.dat	Multi-frequency stress response and bending moments as a function of depth
Fat.out	Fatigue life calculations
Out.dat	Motions response
Out-mm.dat	Multi-frequency motions response
Sum-sta	Summary of all motions, stress, and bending moment response

The ability to withstand these random forcing functions however, is directly linked to the quality and thoroughness of the engineering design beforehand. These structures must be designed against rogue currents and storms which might only occur once every 100 years or more, but pose serious environmental, safety and economic threats.

Engineers are faced with the challenge of first understanding and modeling the full range of possible environmental characteristics, and then being able to model the response of the system in these situations and ensuring proper safety factors and fatigue lives. The TLP after which this design was preliminarily modeled, Shell Oil's "Brutus", is known to have had fairings retrofitted onto its risers based on problems that it did in fact experience with VIV induced fatigue. Subsequent Shell testing programs have proven the superior performance of fairings over helical strakes with both smooth and rough (i.e. barnacles and marine growth etc) surface conditions. For the design of the TLP "Tim," computational analyses of the response of the tendons in varying currents will be carried out using two sets of commercially viable VIV codes: VIVA and Shear7.

The first and most important part of the process is to obtain quality current profiles for varying current events. This was difficult in that much of the data available is highly proprietary and the researchers were only able to obtain profiles with, on average, 6 data points. Most often the industry uses profiles with upwards of 40 data points; however they also spend large quantities of money on the equipment necessary to take these measurements and, thus, are highly protective of them. Eight current profiles were used in our analyses. The first four currents used: the 100 year storm, reduced extreme storm, normal operation, and eddy current event, all came from an Offshore Technology Conference (OTC) proceedings source. The next four, OTC 8606, OTC 8405, Typhoon and Non Typhoon, came from varying sources, all of which were found in past years OTC proceedings [10]. Obviously the typhoon and non-typhoon current events have an extremely low probability of occurrence in the Gulf of Mexico (GoM).

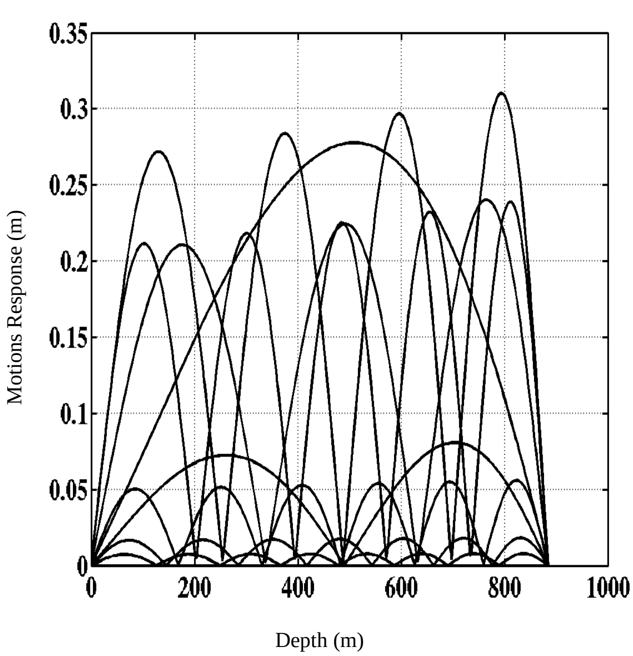

Fig. 2. Motions response as a function of depth for Normal Operation current conditions

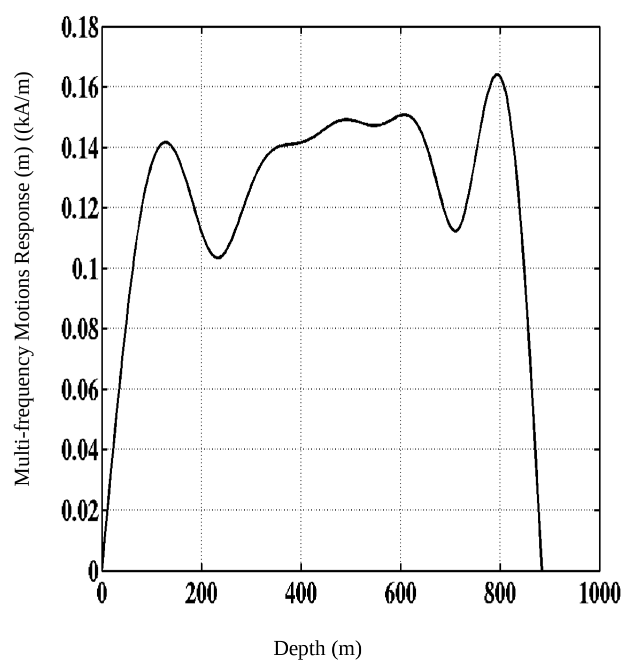

Fig. 3. Multi-frequency motions response as a function of depth for Normal Operation current conditions

TABLE 4
MAXIMUM DISPLACEMENTS AND PROBLEM NODES FOR EACH CURRENT CONDITION

Current	Multi-modal Displacement (cm)	Modal Displacement (cm)	Problem Mode
Normal Operation	31	16	4
100 Year Storm	24	45	6
Eddy Current Event	25	43	12
Reduced Extreme Storm	21	40	5
OTC 8045	25	49	7
OTC 8606	19	35	11 and/or 5
Typhoon	30	55	12
Non-Typhoon	27	63	2

However, the researchers did not feel it would hurt the design process to see the dynamic response of the tendons in the largest cross-section of environments possible. Fig. 1. shows the first four current profiles. For profiles where maximum depths did not coincide with the design depth for our vessel, the last available speed value was simply extrapolated to depth. The vortex shedding frequency off of the tendon scales with velocity given by the following equation:

$$0.2 \cdot U(z)/D \quad (2.1)$$

'U(z)' is the current velocity at a given depth 'z,' and 'D' is the diameter of the tendons (approximately 1 meter). After obtaining a sufficient array of current conditions in which to analyze the tendon system, the next step is to prepare the input files for the respective hydrodynamic codes. To date, the only code which has been utilized is VIVA. VIVA requires a set of input files which describe the physical and material properties of the tendons, the boundary conditions, and the currents [9]. Table 3 outlines the file names and descriptions which were generated for each run.

Once the input files are properly generated they can be fed to VIVA which then produces an extensive set of output files. Table 3 also shows the names and descriptions of these output files. The first set of results which will be discussed is the overall motion, first with the separate modal responses graphed independently and then the full spectrum response. Fig. 2 shows the modal responses and Fig. 3 shows the multi-modal responses in the normal operation current situation. Similar plot were obtained for all eight current situations.

The features of this set of results which it is important to note include the maximum offset in the multimodal tendon motions and also those when each mode is excited independently.

TABLE 5
Modal and Multi-Modal Bending Moments and Stresses for Eddy Current Event Condition

Current	Multi-modal Bending Moment (Nm)	Multi-modal Stress Pa (N/m2)	Modal Bending Moment (Nm)	Modal Stress Atm (kgf/cm2)
Normal Operation	6.5 x 104	4.3 x 106	1.3 x 105	8.7 x 106
100 Year Storm	2 x 105	13.5 x 106	4.2 x 105	28 x 106
Eddy Current Event	4.7 x 105	32 x 106	10.8 x 105	80 x 106
Reduced Extreme Storm	1.1 x 105	7.5 x 106	2.2 x 105	14.9 x 106
OTC 8045	2.4 x 105	17 x 106	5 x 105	34 x 106
OTC 8606	3.5 x 105	24 x 106	7.6 x 105	53 x 106
Typhoon	7 x 105	52 x 106	16 x 105	110 x 106
Non-Typhoon	1.13 x 105	7.7 x 106	2.5 x 105	17 x 106

In most cases the maximum displacement in a particular mode for the given current excitation is greater than that of the multi-modal response. This is important and the stresses/strains which are correlated with these large displacements must be designed against because there is no way to ensure that a random excitation force won't drive the riser at the exact natural frequency which correlates to resonances in the problem modes. For example, in the last case, the Non-typhoon current event, the maximum displacement in the modal response graph appears to correlate with the second mode where the riser displaces almost 70 centimeters at ¼ and ¾ of its length. Yet, in the multimodal response, although the overall shape of the tendon resembles mode 2, the maximum deflection is only approximately 30 centimeters. Table 4 outlines the maximum multi-modal and modal responses for each current situation.

As discussed, the maximum modal displacement values are all greater than those expected for multi-mode excitation. The largest values occur for the typhoon and non-typhoon current events. Because these situations are very unlikely to occur in the GoM, the researchers will probably design against the bending moments induced by the next biggest problem current, the OTC 8045 current. It can also be noted that the estimated problem frequencies seem to be at around mode numbers 11 or 12, and then between 5-7. This information and the natural frequencies with which these modes are correlated is very valuable in the design process.

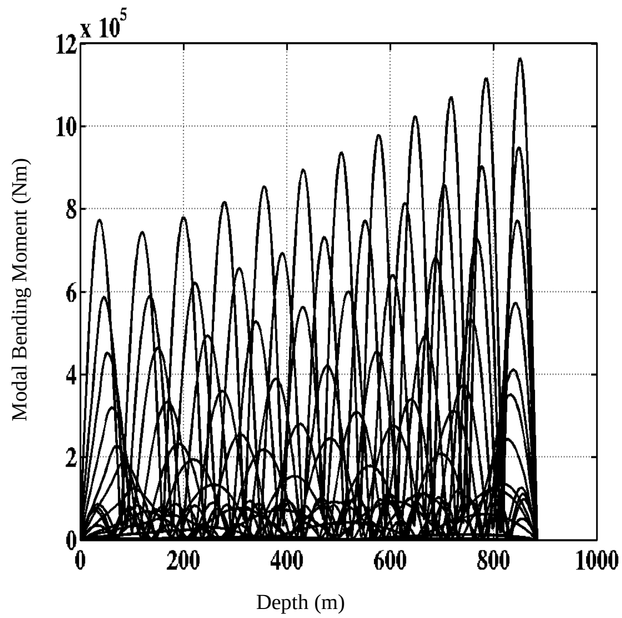

Fig. 4. Modal bending moment as a function of depth for Eddy Current Event Condition

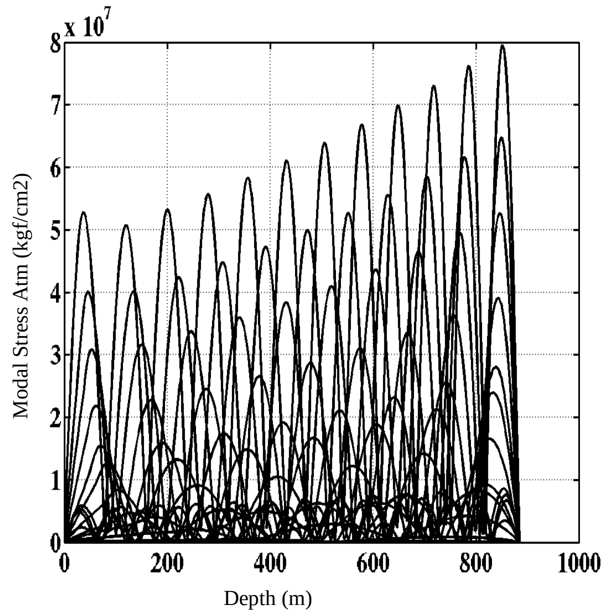

Fig. 5. Modal stress as a function of depth for the Eddy Current Event condition

The next group of data we will discuss is the bending moment and stress values in both the modal and multi-modal responses. Table 5 outlines the maximum bending moment and stress found for each current event in the two response schemes. In order to conserve space, only the graphs for the current event which produced the largest stress and moment values are shown in Fig. 4-7.

As Table 5 delineates, the greatest stress concentrations occur in the tendons in the Typhoon current event. Again, however, since this current is not probable in our operating environment, we instead focus on the second largest values within our operations range.

Interestingly, the second largest stress and moment values are associated with the Eddy current event as opposed to the OTC 8045 as would have been proposed given the displacement results. The explanation for this behavior is probably linked to the fact that the difference between the maximum displacements between the top 3 or so current events is not significant; therefore it is difficult to make failure expectations based solely on the displacement data. Please observe Table 5 along with the graphical representation of these values as shown in Fig. 4-7, because in the design process, it is just as important to know where the maximum stresses occur as to know what the value of those stresses are.

The purpose of completing an analysis of tendon vortex induced vibration responses in varying currents is to ultimately evaluate the integrity of the system from a structural fatigue perspective. All of the displacement, moment and stress data is generated with the aim of determining how long the structure could withstand a given environmental criterion. At this point in the discussion we will move to this topic and the results which VIVA generated for our TLP "Tim." Fatigue life determinations must be observed with caution.

It is easy to forget that the life of the tendon as quoted by whatever code the designer is utilizing represents the time it would take for the tendon or system to fail if it was continually exposed to the given current event. A TLP or other offshore structure is not going to see 10 straight years, for example, of an eddy current event. The calculations are useful none-the-less because if the fatigue life of the system in a given environment is analytically determined to be, say, 2 hours, or even worse, 5 minutes, the tendons will have to be redesigned to increase the life to within satisfactory factors of safety.

Table 6 describes the minimum fatigue life, as determine per mode, for each current event. The complete fat.out files, which are not included in this paper due to lack of space, give the fatigue lives in all modes and the associated stress contributions for the minimum.

Once again, the Typhoon event produces the least satisfactory fatigue life results with the Eddy current event being the next worse within the range of probable environmental conditions. For the eddy current, if the system is driven at the natural frequency of mode 12, 0.4747 Hertz, failure will occur after only 80.3 days. In the multi-modal response, the system could last for 63 years, however with this type of analysis you must place some sort of weight factor on the results which correlates to the reliability and accuracy of the analysis tool. In this case, if we were only 50 percent sure of our results, the minimum multi-modal fatigue life would be approximately 30 years, and for an offshore system whose design life is somewhere in that range, this might not be a satisfactory result.

A cohesive look at the displacement, bending moment, stress, and fatigue life results, as determined by VIVA, shows that the problem current event is the Eddy Current. Given the proper data, the design could then move forward to the associated probabilities of occurrence with each current event and even further establish the reliability of the structure. It is immediately obvious however, that Tim could not withstand a typhoon condition under any circumstances. It would be interesting to reevaluate these results given a tendon model which represented the faired, or even straked, retrofit. This level of complexity is simply not feasible or necessary for this type of design project.

2). *Dynamic Response Estimates*

The TLP dynamic behavior is similar to that of a pendulum. The natural period determinations were modeled as such. These calculations were also done by hand initially and then subsequently by a Matlab script. Given that the vessel behaves like a pendulum, the first value to be determined was the natural frequency of oscillation in pendulum motion which includes the swaying side to side, and associated "set-down", of the vessel. All calculations were done in English units, which carried with it significant frustration. The basic equation of motion (EOM) of the vessel in this degree of freedom (DOF) is

$$I\,(d^2\varphi/dt^2) + k\,(\varphi) = 0 \qquad (1.1)$$

Assuming there is no forcing function and no damping, 'I' represents the sum of the vessel moment of inertia and the added moment due to the entrained mass of water. It is found my multiplying these two masses by the length of the tendons (moment arm) squared. The symbol 'k' represents the stiffness of the vessel in this DOF, and it is found by multiplying the tension in the tendons by the length of the tendons. The natural period in pendulum motion for Tim was found to be 23 seconds.

The next dynamic characteristic to be determined was the vessel natural period in heave. Due to the large amount of tension in the tendons, you can imagine that the vessel oscillates quickly in this DOF.

The process followed in the hydrodynamic Matlab script was to first determine the dynamic and static stiffness coefficients, 'k_{dyn}' and 'k_{stat},' of the vessel. These values were based on the waterplane area of the vessel and have to do with the incremental buoyant force generated by a unit displacement in the vertical direction. The natural frequency in heave is found by the following equation:

$$\sqrt{((k_{stat} + k_{dyn}) / (m + m_{added}))} = 1.9\text{rad/sec}\ \ (0.3\text{Hz}) \qquad (1.2)$$

The third and final DOF that was analyzed in the dynamic analysis of Tim the TLP was the Pitch/Roll (relatively equal for a TLP) direction. In this case again, the static and dynamic stiffness coefficients in this direction of movement must be determined. The static stiffness is given by

$$K_{stat} = 2a^2 g\,\rho\,AWP \qquad (1.3)$$

where 'a' is the distance from the centerline of the vessel to the center of each caisson, 'g' is the acceleration of gravity, 'ρ' is the density of seawater, and 'AWP' is the waterplane area. The dynamic stiffness is given by

$$K_{dyn} = (\,2a^2 E\ A\ n_{tendons}\,)\,/\,L \qquad (1.4)$$

where 'E' is the Young's modulus of the tendon material, steel, 'a' is the cross-sectional area of the tendons, $n_{tendons}$ is the number of tendons, and L is the length of the tendons. The next step in the analysis is to determine the mass moment of inertia of the vessel in the pitch/roll DOF as well as the added mass moment of inertia associated with the water accelerated by the moving hull. These calculations were tedious and required us to make some assumptions about the radius of gyration for the major hull components.

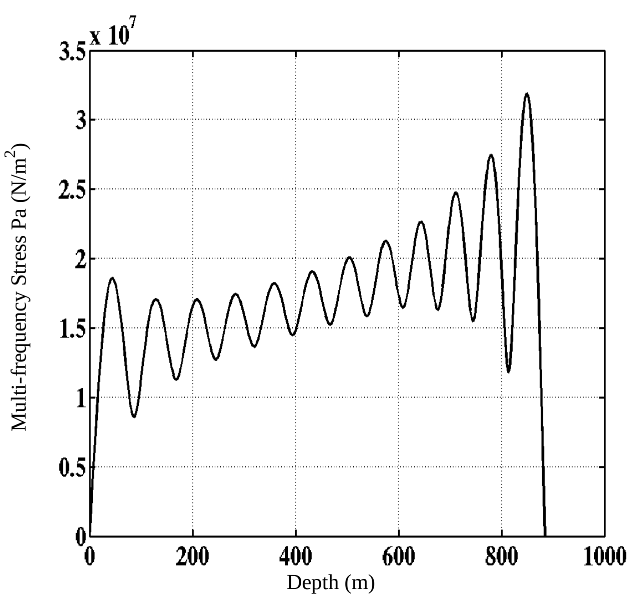

Fig. 6. Multi-frequency stress as a function of depth for the Eddy Current condition

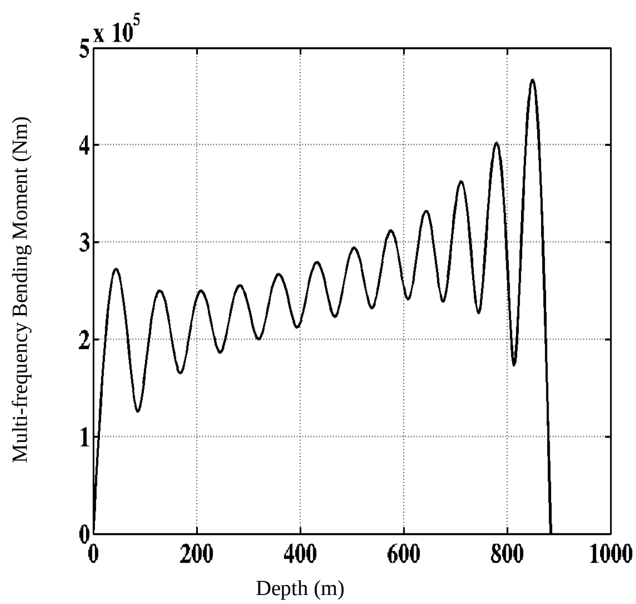

Fig. 7. Multi-frequency bending moment as a function of depth for Eddy Current condition

TABLE 6
MODAL AND MULTI-MODAL FATIGUE LIFE ESTIMATES PREDICTED BY VIVA

Current	Multi-Modal Minimum Fatigue Life (years)	Multi-Modal Location (m)	Modal Minimum Fatigue Life (years)
100 Year Storm	5100	833.7	43
Eddy Current Event	63	851.5	.22
Reduced Extreme Storm	68000	821.9	810
OTC 8045	1600	833.7	17
OTC 8606	240	848.5	1.5
Typhoon	6.6	851.5	.052
Non-Typhoon	43000	836.7	370

Current	Modal Location (m)	Problem Mode	Associated Stress Pa (N/m^2)
100 Year Storm	833.7	7	8×10^6
Eddy Current Event	851.5	12	2×10^7
Reduced Extreme Storm	824.9	6	4.6×10^6
OTC 8045	833.7	7	1×10^7
OTC 8606	848.5	11	1.4×10^7
Typhoon	854.4	13	2.9×10^7
Non-Typhoon	833.7	7	4.9×10^6

Therefore the error associated with this calculation is probably greater than for the other two DOFs. After going through all of these calculations, by hand and computationally, the final natural frequency in pitch/roll was determined to be 3.069 radians per second (0.5 Hertz).

In addition to examining the natural frequencies of Tim in pitch/roll, heave and pendulum motion, we also attempted to complete a more complex analysis of Tim's dynamics using the loading situation as a result of the currents and seas in the GoM. Again, this area of the design was preliminarily done by hand, but quickly we relied on Matlab for our solutions. Thus the Matlab script, isodc_dyn_rev1.m, was written. The major assumptions integrated into the code are that, first, the waves made by the structure are ignored, and second, that the drag and vortex shedding off the submerged components are also ignored. Basically any vortex induced vibration or forces are ignored. These assumptions are valid if the wavelength of the sea state is long compared to the diameter of the caissons, and for the most part in the GoM, this is valid.

This code must, of course, re-derive the basic weight and displacement values for the vessel, and from those assign a tension to each of the tendons (variable tension was not considered in any part of the design). Using the dispersion relation you can find the forces on the vessel with only the amplitude of the wave and the natural frequency information for the vessel. This results in a transfer function of the vessel relating the force on the vessel to the angle displacement of the vessel in pendulum motion.

To describe any sea state you need environmental data, from which you can estimate the natural frequency (ωn) and the significant wave height (ξ). The wave amplitudes were estimated using the Bretschneider Spectrum and the values of ωn and ξ for which we were concerned (GoM sea state 5 values: $\omega n=0.65$, $\xi=5m$). From there the non-time-dependant forces on the vessel were estimated, and a transfer function relating wave amplitude to the angle displacement of the pendulum model (or excursion of the hull of the TLP) was determined.

Finally, the static transfer function of the vessel is combined with dynamic wave-loading transfer function to find the overall transfer of input frequency to output excursion. The code calls the Bretschneider spectrum $S(\omega)$, the static transfer function $H(\omega)$, and to find the overall $\theta(\omega)$ the following equation was used:

$$\theta(\omega) = S(\omega) * abs[H(\omega)]2 \qquad (1.5)$$

To find the maximum excursion of the vessel, which is important for fatigue issues on the mooring system among other things, you integrate (the code uses the trapezoidal rule) the $\theta(\omega)$ curve over a reasonable range of frequencies and get the RMS2 value of the angle, the significant angle will be four times the RMS angle. These values being in radians, the only step required to find the excursion in meters, is to employ the small angle assumptions and multiply by the length of the tendons.

The code was determined to be very sensitive to geometric design parameter changes, however, the last estimate for θsig and the amplitude of surge (excursion), were 0.0011 radians and 3.16 m. [This is for a tendon tension of 157.8 tons.] Graphs of all three transfer functions which give a good visual representation of the movement of the vessel, can be found in any output of the Matlab script.

B. *Structural Analysis: Finite Element Methods*

The design team was not able to complete the solid model and finite element analysis for Tim due to time constraints and limited personnel. However, familiarity with Abaqus, a commercial finite element method software package, was achieved and preliminary structural analysis was commenced. The input, output, and calculation procedure of Abaqus is described by Yingbin Bao in “Introduction to Abaqus [7].” The user enters parameters into the CAE pre-processor, which outputs a .inp file. The .inp file is loaded into a standard solver, which outputs a .odb file. The .odb file is loaded into the CAE post-processor. Abaqus uses finite element method algorithms to calculate the displacement, stress, strain, and reaction force.

The pre-processor has eight user interface menu options. The Part feature allows the user to sketch two dimensional profiles and create part geometries. The Property feature allows the user to define material properties and section properties. The Assembly feature allows the user to assemble models from sets of parts. The Step feature allows the user to configure analysis procedures and output requests. The Load/BC/IC allows the user to apply loads, specify boundary conditions and initial conditions of the part or assembly. The Mesh feature allows the user to choose from triangular or rectangular elements and create a mesh. The Job feature submits the mesh assembly for analysis. The Visualization feature displays the results [7].

In order to calculate displacements and loads on the structure, Abaqus uses finite element methods. In finite element analysis, as described by Thomas J. R. Hughes [8], a continuous structure such as a plate or beam is divided into discrete elements, and continuous loads are divided into discrete nodal point loads. The elements are connected at nodes. The most common elements are triangular and rectangular elements. Elements can be the same size throughout the structure, or a “graded mesh” where the elements are smaller in the region where a more detailed modeling is desired. The advantage of triangular elements is a constant stress value within the element. Finite element analysis always predicts deflections that are less than the deflections predicted by elastic beam theory. To satisfy compatibility, a displacement function is assumed, which causes the finite element model to be stiffer than the actual structure.

V. CONCLUSIONS

The hydrostatic and hydrodynamic analysis of the TLP “Tim” is valid. However, the TLP design is weakest in the structural design and analysis, and is lacking in riser design. More detailed structural design, including all buckling modes of structures, needs to be done. An Abaqus solid model and finite element calculations need to be done. Riser design and analysis needs to be done, including VIVA runs for motions response, stress, bending moments, and fatigue, and analysis of lock-in phenomena. The extent and effects of limitations of the vortex-induced vibration analysis, such as current profile data points, need to be examined. Other concerns that need to be addressed for a more complete design are cost, component fabrication, and system assembly.

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REFERENCES

[1] Shell Exploration and Production Company, “Brutus Tension Leg Platform,” 13 June 2003. <http://www.shellus.com/sepco/where/offshore_shell/brutus.htm>.

[2] Z. Demirbilek, Ed. *Tension Leg Platform: a state of the art review.* New York, NY: American Society of Civil Engineers, 1989.

[3] Det Norske Veritas, *Det Norske Veritas Guidelines for Offshore Structural Reliability*, 13 June 2003. <http://research.dnv.com/skj/OffGuide/SRAatHOME.pdf>.

[4] Det Norske Veritas, “Mandatory risk assessment of floating offshore platforms.” 13 June 2003. <http://www.dnv.com/publications/oilgas_news/by_subject/Health_safety_and_environment/Mandatory_risk_assessment_of_floating_offshore_platforms.>

[5] Minerals Management Service, “Offshore Minerals Management.” 13 June 2003. <http://www.mms.gov/offshore/>

[6] Noble Denton, 13 June 2003. <http://www.nobledenton.co.uk/html/about/about.html>

[7] Y. Bao, “Introduction to Abaqus,” unpublished.

[8] T. J. Hughes, *The Finite Element Method: linear static and dynamic finite element analysis*, Prentice-Hall, Englewood Cliffs, NJ: 1987.

[9] M.S. Triantafyllou, “A Description of the Programs and their Use VERSION 4.3” unpublished.

[10] *Offshore Technology Conference Proceedings*