

ACTIVE CONTROL FOR ENHANCING FATIGUE LIFE OF TLP PLATFORMS AND TETHERS

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Abstract. *This paper is concerned with the dynamics of a tension leg platform (TLP) under environment load effects. In deep water scenarios, large heave amplitudes caused by dynamic loads are considered as one of the most deleterious effects to the structural safety. Here it is shown that active control systems may be installed inside the hull of a TLP to attenuate dynamic amplitudes in heave motion which leads to a significant reduction of the stress levels in tendons and their links and anchorages, minimizing fatigue problems and increasing tether's service life, improving the production system performance. The control devices may be located inside the columns. The uncontrolled and controlled dynamic behaviors of a TLP prototype are investigated by using simplified mathematical model where the TLP is considered as a tridimensional rigid body, with six degrees of freedom and the tendons as elastic springs. The dynamic behavior is described by nonlinear coupled second order differential equations of motion by using the Hamilton Principle. The numerical results lead to the conclusion that active systems are effective in reducing and controlling the heave displacement amplitudes, for a given low mass ratio, and consequently the stress variations in tendons and risers of a TLP.*

Keywords. Offshore Structure, Active Control, Fatigue Analysis

1. Introduction

Tension Leg Platforms (TLP) have been considered the most promising hydroelastic systems intended for deep water oil exploitation, especially because of its economical viability. Like any other floating offshore structure, the TLP displays large response amplitudes in heave motions to wave disturbances, which vary in severity with sea/wind conditions. These large heave amplitudes present a serious drawback to the required dynamic behaviour and in service life of both prestressed tethers and hanging risers because early fatigue cracks may develop in these structural components.

The active control of heave motions of these huge compliant offshore structure is made feasible by a logical control applied to servo-hydraulic/pneumatic actuators that accelerate reaction masses counteracting the TLP's floating hull movements. Other conceptions of active and semi-active control of floating structures may be found in the literature (Hrovat, 1983; Reinhorn, 1987; Sirlin, 1980). The active system optimal control laws keep response amplitudes under pre-defined performance measures. Under severe environmental forces, the structural components of uncontrolled systems may develop inelastic deformations in a manner for dissipating energy. Alternatively, the response amplitudes may be attenuated by an active control device, through which they may be continually monitorated and corrected by added inertia forces. The actuator stroke and the developed hydraulic pressure are among the most important factors to be considered in the design of active control systems (Alves, 1997; Battista, 1993).

Control systems application to attenuate the heave motion in TLP's may be more efficient and less costly than pneumatic/telescopics devices. The TLP heave motion control leads to a significant reduction of the stress levels in risers tendons and their links and anchorages, besides minimising fatigue problems, then improving the production system performance and increasing tether's service life.

The uncontrolled and controlled TLP dynamic behaviour was investigated by using simplified mathematical models for the hydroelastic structure under irregular wave loads. The goal of this paper is to demonstrate analytically the effectiveness of active control to reduce the undesirable amplitudes of heave motion and minimising fatigue problems. Wave loads were considered dynamically, while wind and current were considered statically. The performance of the active controlled TLP is checked against the dynamic response of uncontrolled TLP.

Under environmental load, the TLP is displaced from the static vertical position to a neighbouring inclined one as showed in Fig. (1), where the platform will oscillate under wave action. The inclination angle of tendons in a design factor; lateral displacement is controlled by tendon's stiffness and might not exceed a limit value.

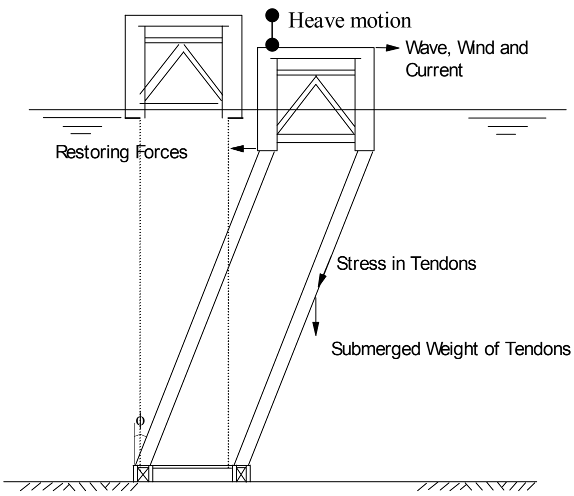

Figure 1. Environmental load under TLP

2. Mathematical Model For The TLP Motion

The TLP was considered as a tridimensional rigid body, with six degrees of freedom, and the tendons as elastic springs. The dynamic behaviour is described by second order differential equations of motion.

The formulation of Lagrange equations of motion was made by using the Hamilton Principle, and leads to the following system of coupled non linear equations of uncontrolled motion of the TLP, in a matrix form (Alves, 1997; Hrovat, 1983), where, **M** is the mass matrix, **C** is the damping matrix, **K** is the stiffness matrix, **Q** the external forces vector and $\ddot{\mathbf{q}}(t), \dot{\mathbf{q}}(t), \mathbf{q}(t)$, are acceleration, velocity and displacement, respectively.

$$\mathbf{M}\ddot{\mathbf{q}}(t) + \mathbf{C}\dot{\mathbf{q}}(t) + \mathbf{K}\mathbf{q}(t) = \mathbf{Q}\,(q, t) \qquad (1)$$

The matrix construction was based in Hooft (1971) , that showed a method to determine hydrodynamic and excited forces in each element of the TLP hull. The resultant is obtained by the summation of the force components in all elements.

3. Optimal Active Control Of Heave Motion

Control systems may be applied in TLP's in other to attenuate the response amplitude due to external loads . It's made by applying control forces in heave displacement phase opposition, to guarantee the structural system safety. The basic conception of active control of heave motion of a TLP by using activated masses (AMC) is illustrated in Fig. (2), together with a block diagram of active control.

The structure is instrumented for obtaining the real response signal in terms of acceleration. This real signal after conversion is called "FEEDBACK signal" (in terms of velocity) and is compared to the desired structure response or the "REFERENCE signal". The difference between them is the "error signal" that is used to control the system, keeping the response in the desired value. The control force is transferred to the structure by auxiliary masses that has been accelerated by a servo-hydraulic/pneumatic actuators, counteracting the TLP's floating hull heave motion.
For sensored system, the control force is a function of response amplitudes, and is automatically regulated. The sensors located in the structure allow that the error measurement could be reanalysed.

For active motion control of the modal mass, optimal control principles are used to determine the added control force **u***(t) (Meirovitch, 1990).

The dynamics of 6DOF system (active heave control force) is governed by the following coupled second order differential equations (Alves, 1997):

$$m_{11}\ddot{x} + m_{15}\ddot{\theta} + m_{16}\ddot{\beta} + c_{11}\dot{x} + c_{15}\dot{\theta} + c_{16}\dot{\beta} + k_{11}x + k_{15}\theta + k_{16}\beta = f_1$$

$$m_{22}\ddot{y} + m_{24}\ddot{\alpha} + m_{26}\ddot{\beta} + c_{22}\dot{y} + c_{24}\dot{\alpha} + c_{26}\dot{\beta} + k_{22}y + k_{24}\alpha + k_{26}\beta = f_2$$

$$m_{33}\ddot{z} + m_{34}\ddot{\alpha} + m_{35}\ddot{\theta} + c_{33}\dot{z} + c_{34}\dot{\alpha} + c_{35}\dot{\theta} + k_{33}z + k_{34}\alpha + k_{35}\theta = f_3 - \mathbf{u(t)}$$

$$m_{44}\ddot{\alpha} + m_{42}\ddot{y} + m_{43}\ddot{z} + m_{45}\ddot{\theta} + m_{46}\ddot{\beta} + c_{44}\dot{\alpha} + c_{42}\dot{y} + c_{43}\dot{z} + k_{44}\alpha + k_{42}y + k_{43}z = f_4$$

$$m_{55}\ddot{\theta} + m_{51}\ddot{x} + m_{53}\ddot{z} + m_{54}\ddot{\alpha} + m_{56}\ddot{\beta} + c_{55}\dot{\theta} + c_{51}\dot{x} + c_{53}\dot{z} + k_{55}\theta + k_{51}x + k_{53}z + k_{54}\alpha = f_5$$

$$m_{66}\ddot{\beta} + m_{61}\ddot{x} + m_{62}\ddot{y} + m_{64}\ddot{\alpha} + m_{65}\ddot{\theta} + c_{66}\dot{\beta} + c_{61}\dot{x} + c_{62}\dot{y} + k_{66}\beta + k_{61}x + k_{62}y = f_6 \qquad (2)$$

Where, apart from the variables already defined, for i=1,6: $m_{i,j}$, are the coefficients of **M**; $k_{i,j}$, are the coefficients of **K**; $c_{i,j}$, are the coefficients of **C,** and $f_{i,}$, are the components of external applied force; x, y, z, α, θ, β, are relative to six degrees of freedom: surge, sway, heave, roll, pitch and yaw, respectively.

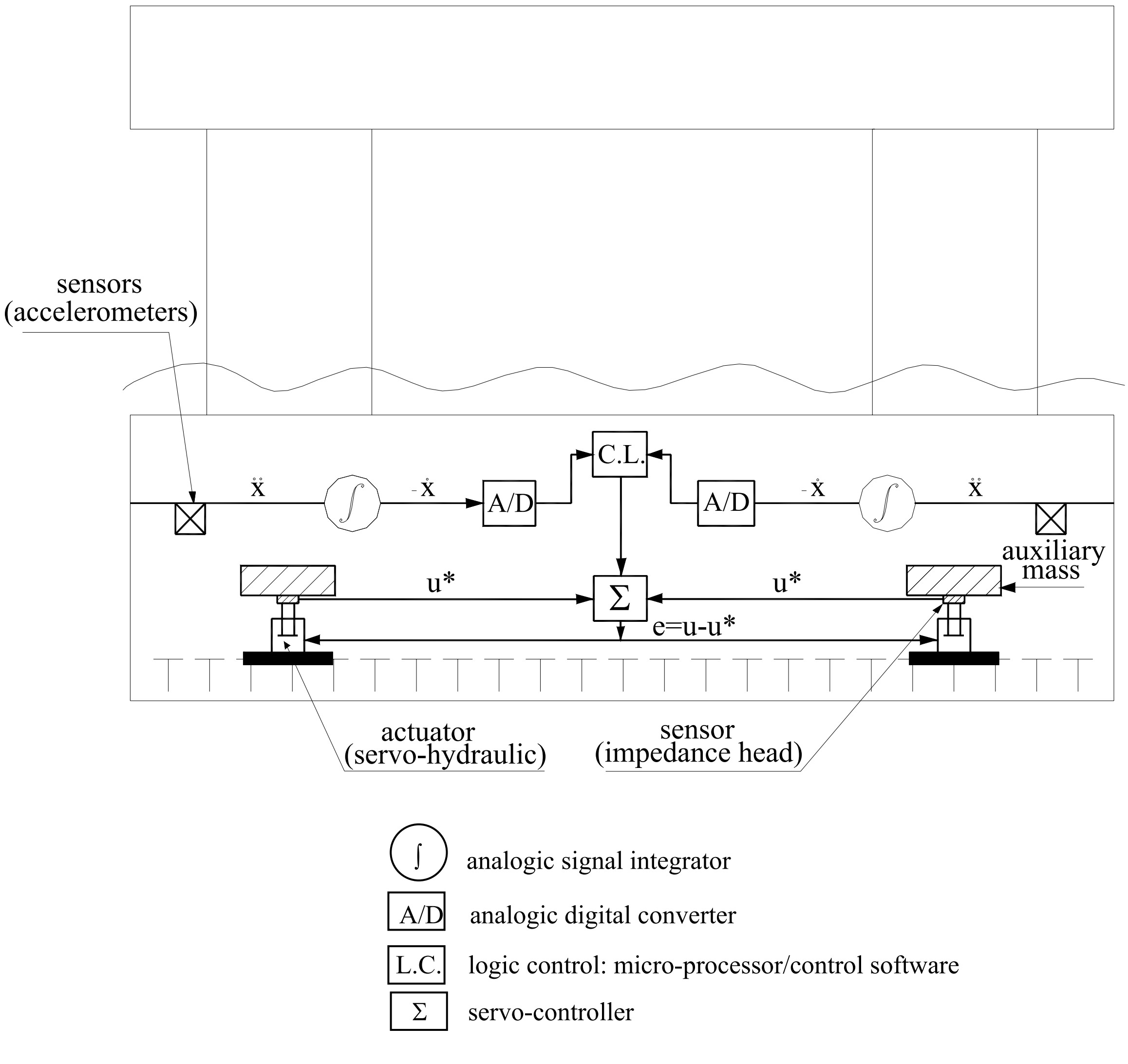

Figure 2 - Schematic of active control in TLP

3.1. TLP's Equation of motion with active control

The equations of motion of the controlled system is written in matrix form as :

$$\mathbf{M\ddot{q}} + \mathbf{C\dot{q}} + \mathbf{Kq} = \mathbf{F} - \mathbf{F}_C \tag{3}$$

where: $\mathbf{M} = \sum_{i=1}^{N} \mathbf{M}_i^{el} \quad \mathbf{C} = \sum_{i=1}^{N} \mathbf{C}_i^{el} \quad \mathbf{K} = \sum_{i=1}^{N} \mathbf{K}_i^{el} \quad \mathbf{F} = \sum_{i=1}^{N} \mathbf{F}_i^{el} \quad \mathbf{F_C} = \sum_{i=1}^{N} \mathbf{F_C}_i^{el}$ (4)

being $\mathbf{F}_C$ the control force.

The state vector x(t) is defined as:

$$\mathbf{x}(t) = \begin{bmatrix} \mathbf{q}(t) \\ \dot{\mathbf{q}}(t) \end{bmatrix} \quad , \quad \dot{\mathbf{x}}(t) = \begin{bmatrix} \dot{\mathbf{q}}(t) \\ \ddot{\mathbf{q}}(t) \end{bmatrix} \tag{5), (6}$$

From Eq. (3): $\ddot{\mathbf{q}} = \mathbf{M}^{-1}\left(-\mathbf{C}\dot{\mathbf{q}} - \mathbf{Kq} + \mathbf{F} - \mathbf{F}_C\right)$ (7)

and substituting Eq. (7) in (6), leads to :

$$\begin{bmatrix} \dot{\mathbf{q}}(t) \\ \ddot{\mathbf{q}}(t) \end{bmatrix} = \begin{bmatrix} \dot{\mathbf{q}}(t) \\ \mathbf{M}^{-1}\left(-\mathbf{C}\dot{\mathbf{q}} - \mathbf{Kq} + \mathbf{F} - \mathbf{F}_C\right) \end{bmatrix} \tag{8}$$

In state-space notation the dynamic behavior of the system is described in the vector form by the following first order Hamiltoniam differential equations :

$\dot{\mathbf{x}}(t) = \mathbf{Ax}(t) + \mathbf{Bu}(t) + \mathbf{F}$, **A** and **B** are coefficient matrixes (9)

where: $\mathbf{Bu}(t)$,is related to the introduction of active control in the system

$$\dot{\mathbf{x}}(t) = \left[x_1, x_2, x_3, x_4, x_5, x_6, x_7, x_8, x_9, x_{10}, x_{11}, x_{12}\right]^T \tag{10}$$

or in another notation,

$$\dot{\mathbf{x}}(t) = \left[u, v, w, \gamma, \xi, \psi, \dot{u}, \dot{v}, \dot{w}, \dot{\gamma}, \dot{\xi}, \dot{\psi}\right]^T \tag{11}$$

and :

- $\mathbf{A}_{(2n \times 2n)} = \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\mathbf{M}^{-1}\mathbf{K} & -\mathbf{M}^{-1}\mathbf{C} \end{bmatrix}$
- $\mathbf{B}_{(2n \times 2n)} = \begin{bmatrix} \mathbf{0} \\ -\mathbf{M}^{-1}\mathbf{K} \end{bmatrix}$
- $\mathbf{u}(t) = \mathbf{F}_C$
- $\mathbf{F}_{(2n \times n)} = \begin{bmatrix} \mathbf{0} \\ \mathbf{M}^{-1}\mathbf{Q} \end{bmatrix}$ (12)

For a sufficiently large controlling time interval (i.e. $t_f \to \infty$), it follows from modern control theory that the optimal linear feedback control law for **u***(t) is given by (Meirovitch, 1990) :

$$\mathbf{u}^*(t) = -\mathbf{R}^{-1}\mathbf{B}^T\mathbf{P}(t)\,\mathbf{x}^*(t) \tag{13}$$

in which Γ is the control gain matrix: $\Gamma = \mathbf{R}^{-1}\mathbf{B}^T\mathbf{P}(t)$ (14)

and the symmetric matrix P satisfies the algebraic matrix Ricatti equation :

$$\mathbf{PA} + \mathbf{A}^T\mathbf{P} - \mathbf{PBR}^{-1}\mathbf{B}^T\mathbf{P} + \mathbf{Q} = \mathbf{0} \tag{15}$$

being the control force **u***(t), in this linear regulator problem, the best linear function of **x**(t) for minimizing the quadratic objective function J (Meirovitch, 1990):

$$\text{Min}\left\{ J = \frac{1}{2} \int_0^{t_f} \left[\mathbf{x}^T(t)\, \mathbf{Q}\mathbf{x}(t) + \mathbf{R}\, \mathbf{u}^*(t)^2 \right] dt \right\} \tag{16}$$

In eqns. (15-16) **Q** is a positive semi-definite weighting matrix associated with the controlled components of the state vector **x**(t), and **R**, is a weighting positive parameter related to the required level of $u^*(t)$.

Ricatti equation (15) may be solved for **P** by using any appropriate iterative technique, but in systems with numerous degrees of freedom the solution become cumbersome. Alternatively, efficient optimal control algorithms such as the instantaneous algorithm may be used in this closed-loop control problem, as indicated in Fig.(2). The control force **u***(t) is regulated only by the feedback response state vector **x**(t), which is measured by means of sensors located on the TLP's hull; one sensor at each AMC location. In the instantaneous optimal approach the time dependent quadratic function J(t), is used as a concurrent performance index (Battista, 1993) .

$$J(t) = \underbrace{\mathbf{x}^T(t)\, \mathbf{Q}\, \mathbf{x}(\mathbf{t})}_{\textit{safety}} + \underbrace{\mathbf{u}^T(t)\, \mathbf{R}\, \mathbf{u}(\mathbf{t})}_{\text{economy}} \tag{17}$$

that being minimized under the constraint given by the state equation (9) leads to the following closed-loop control law:

$$\mathbf{u}^*(t) = -\frac{\Delta t}{2} \mathbf{R}^{-1}\, \mathbf{B}^T\, \mathbf{Q}\, \mathbf{x}(\mathbf{t}) \tag{18}$$

in which Δt is the numerical integration step for the solution of the state equation of motion (9) and J(t) is a minimum in each interval (t, t+Δt). A standard fourth-order Runge-Kutta scheme was used for the numerical solution of all equations of uncontrolled and controlled motions (Carnaham, 1969).

3.2. Random Analysis In Time Domain

Time domain analyses of the uncontrolled and controlled TLP were performed under random dynamic load. The wave load was simulated with the ISSC spectrum (Chakrabarti, 1987), that is based in two parameters: significative height (H_s) and zero cross period (T_z). The spectrum was subdivided in 50 frequencies.

4. Uncontrolled And Controlled Dynamic Responses

The performance of the active control of heave motions of a TLP intended for deep waters scenarios was examined numerically using the base-line parameters shown in Tab (1). As depicted in Fig.(2), the active control of heave motions may be well accomplished by using servo-hydraulic actuators, mounted vertically, to accelerate masses totaling 1.9% of the whole mass of the floating structure, that in the present case example reads: 59.0 t, to be divided by 4, i.e. mounted on each of the envisaged 4 actuators, located at the four corner legs of a TLP's hull.

Table (2) shows the obtained activelly controlled dynamic responses for heave motion. It is seen that control is accomplished by an active mass (ACM) for an accelerated mass equal to 59.0t, that was enough to reduce the heave motion in 75% .

After parametric analysis, (Alves, 1995), the values of Q and R for that the heave displacement amplitudes had significant reduction were $Q=10^{11}$ and R=8.

The actuator stroke is given by:

$$\delta_a = \frac{F_C}{m_a\, \omega_h^2} \tag{19}$$

where, m_a is the accelerated mass which is a fractional of the structure mass (m_e); F_C is the control force and ω_h the heave frequency.

Figure (4.a) presents the obtained uncontrolled and actively controlled dynamic responses for heave displacements. The required total active control force and the resulting actuator stroke time histories are shown in figures (4.b) and (4.c), respectively.

The servo system time response was not considered in simulations, but, in a future work it may be easily introduced.

5. Fatigue Analysis

A fatigue analysis was performed to show that active control systems minimize fatigue problems increasing tether's service life.

Fatigue damage is a cumulative effect of stresses histories. The development of fatigue damage under stochastic or random loading is in general termed cumulative damage. The fatigue life calculation is founded on the assumption of linear cumulative damage (Miner-Palmgren rule) combined with S-N curves.

The basic assumption in the Miner summation method is that the "damage" on the structure per load cycle is constant to a given stress range and equal to:

$$D = \frac{1}{N} \tag{20}$$

where, N is the constant amplitude endurance at all given stress range. In a constant amplitude test this leads to the following failure criterion:

$$D_f \geq 1 \tag{21}$$

In a stress history of several stress ranges $S_{r,j}$ each with a number of cycles n_j, the damage sum follows from:

$$D = \sum_i \frac{n_i}{Ni} \quad , \tag{22}$$

with the failure criterion still given by Eq. (21) or under any other connection according the type of wave spectrum considered.

Two methods were used to peak counting: "Rainflow" and "Reservoir " .

5.1. S-N Curves And Joint Classifications

For practical fatigue design, welded joints are divided into several classes, each with a corresponding design S-N curve.

The S-N design curves are defined as the mean minus two standard deviations of log N and thus corresponds to 97.6% probability of survival and are written as:

$$\text{Log}(N) = \log a - 2.\log s - m.\log \Delta\sigma = \log \bar{a} \text{ -,m. } \log \Delta\sigma \tag{23}$$

Basic S-N curves parameters for seawater and cathodic protection, indicated for welded joints of tubular members can be find in reference (Gurney, 1976).

Assuming that active control should have the same efficiency in other sea states, the sea state variation was not considered during the TLP operation. The objective of this work is to show that active control leads to a significant reduction in dynamic amplitudes and increase the service life of the platform's tethers.

Figures (5) show a histogram with the results of number of cycles per tension for both methods and Tab.(3) the fatigue life for controlled and uncontrolled responses for Rainflow method. Fatigue life was calculated for classes E, F, F2, using standard deviation **1 (**Gurney, 1976), and stress concentration factor (SCF) **1.5**, **2.0, 2.5** and **3.0**. The recommended value for SCF is 2.5, but references (Aggerskov, 1994; Alves, 1996) , use lower values .

Table 1- Description of the idealized TLP system

Basic characteristics	Values	Comments
Hull mass (m)	3.156,68 t	Not including pay-load
Water depth (d)	938.0 m	
Tether stiffness (K)	15.674,0 kN/m	Each one
Initial tether tension (T_0)	23.838,0 kN	TLP's upright position
Initial tether length (L_0)	911,5 m	
Tethers section area	0,204 m2	3 tethers per leg
Tethers width	3,175 m	
Submerged depth (d_s)	28.0 m	
Heave period (T_h)	3,34 sec.	Undamped; f_h = 0.299 Hz
Surge and sway period (T_s)	136,0 sec.	Undamped; f_S = 0.0074 Hz
Damping ratio (ξ)	15%,15%, 1,65%	Percent of critical damping surge, sway, heave, respectively
Wave frequency	f_w = 0.0769 Hz	
Pontoons diameter	10,27 m	
Columns diameter	18,2 m	
Wind force	2.463,68 kN	45° with x direction
Current force	1.697,25 kN	45° with x direction

Table 2 – Active Control Dynamic Responses ($Q=10^{11}$, R=8) (Alves,1995)

	Peak Amplitude Reduction			**Control Force**	**Actuator Stroke**
m_a	***Heave***		**ΔF_t**	(kN)	δ (cm)
	máx	Mín	máx		
59,0 t	83,0 %	75,0 %	48,0 %	680,0	90,0

Table 3 – Fatigue Life – Rainflow Method

		FATIGUE LIFE (years)		
Classes	**SCF**	**Uncontrolled**	**Controlled**	**Ratio**
E	1.5	92	701	7.6
E	2.0	38	302	8.0
E	2.5	19	152	8.0
E	3.0	11	87	7.9
F	1.5	52	397	7.6
F	2.0	22	171	7.8
F	2.5	11	86	7.8
F	3.0	6	49	8.2
F2	1.5	36	276	7.6
F2	2.0	15	119	7.9
F2	2.5	7	60	8.6
F2	3.0	4	34	8.5

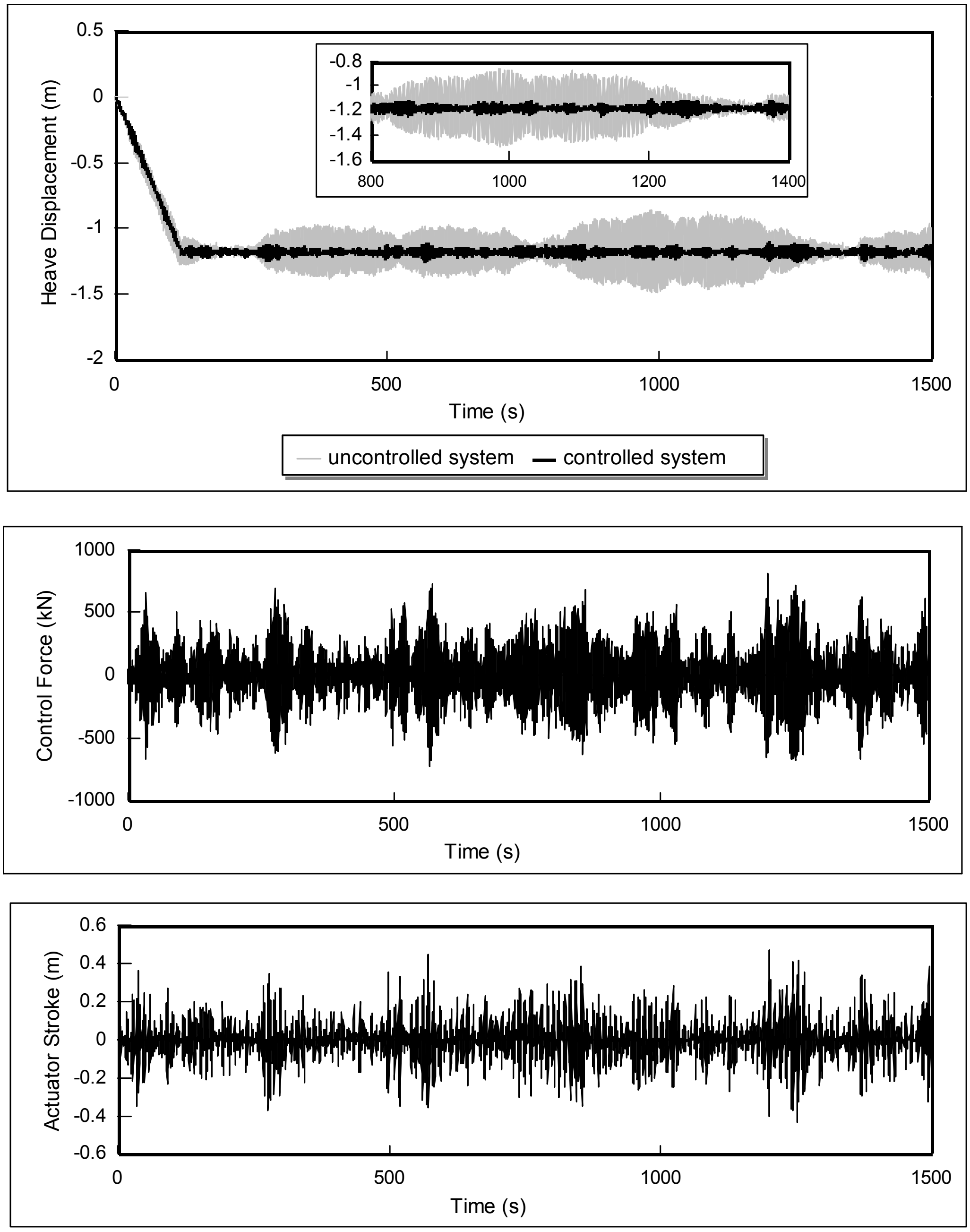

Figure 4a, 4b, 4c – Heave Displacement (controlled/uncontrolled), Control Force, Actuator Stroke
Random analysis – H_s=7 m (wave significative height) , T_z=8.8 sec(zero cross period), 30 years

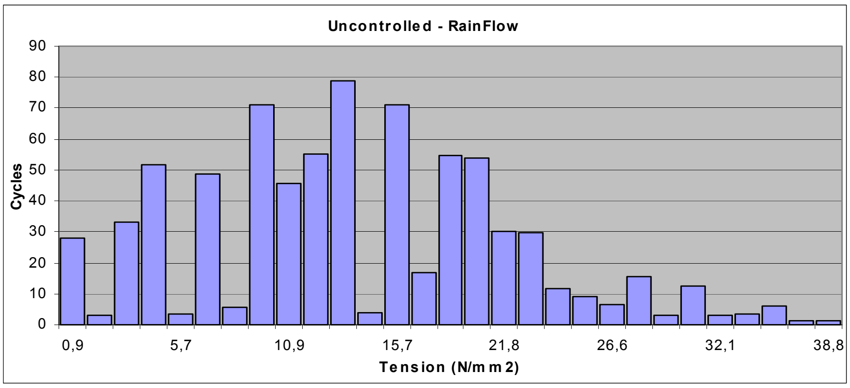

Figure 5a- Histogram – number of cycles x tension – **Rainflow** Method – **uncontrolled** system

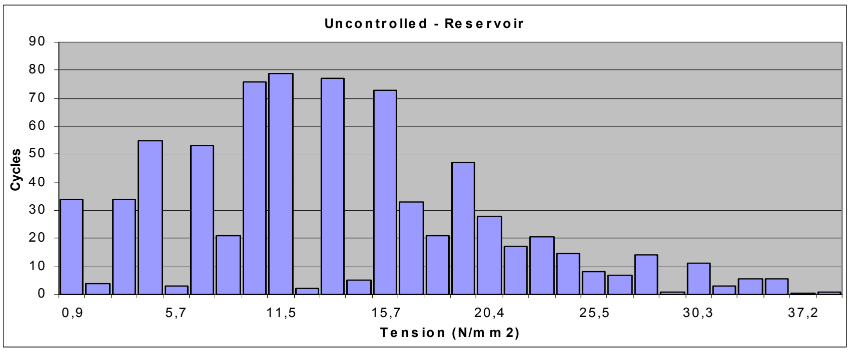

Figure 5b- Histogram – number of cycles x tension – ***Reservoir*** Method – **uncontrolled** system

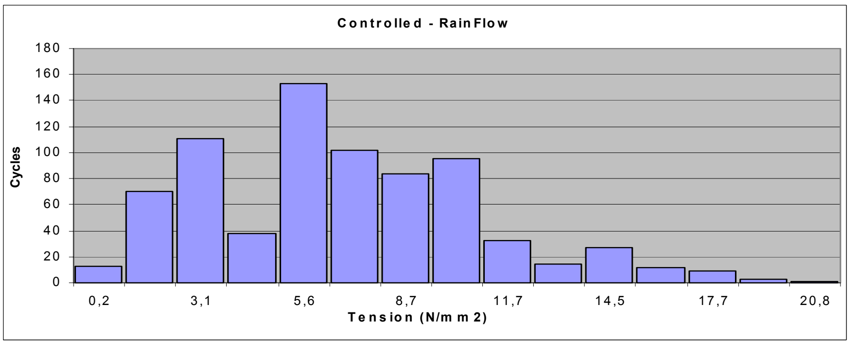

Figure 5c - Histogram – number of cycles x tension – **Rainflow** Method – **controlled** system

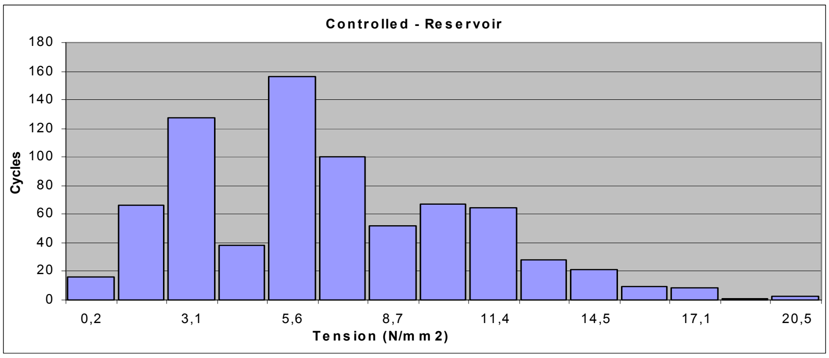

Figure 5d - Histogram – number of cycles x tension – ***Reservoir*** Method – **controlled** system

6. Concluding Remarks

The purpose of controlling the heave motion amplitudes of a TLP's hull is to ensure structural integrity and safety, to the most critical components: tethers and risers and their links and connections. The feasibility of applying active control was then examined by a weighting performance index that accounts for both response amplitudes (of TLP's hull and masses of the subsidiary control system) and the control forces.

The obtained numerical results have shown that the active control system presents high performance in terms of the adopted criteria. The active control leads to a large reduction of heave displacement amplitudes, and largely increases the fatigue life of the TLP's tethers (around 8 times more than uncontrolled), for a given mass ratio.

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