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Research Paper

Dynamic Design Analysis Method (Ddam) – A Case Study Applied In Themain Combustion Engine Of A Naval Ship

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Abstract. *Since the World War II, non-contact explosives have been used as military weapons to cause severe damages in ships and submarine equipment. Shock waves generated by underwater explosions (UNDEX) produce high accelerations in the hull of naval ships that can cause the failure of the ship's structures and/or equipment. Dynamic Design Analysis Method (DDAM) has been used since a long time ago by the US Navy as a powerful tool to support theengineers to predict the response of naval equipment. DDAM was selected for a dynamicalanalysis because this method considers some singularities of the naval equipment, for example: hull or deck or shellmounted equipment; or surface or submarine ship; design values according to the shock's direction on board. Thismethod is able to compute the acceleration, velocity, displacement and stress in each discretized mass in the system. In thiswork, as a case study, the DDAM was used in the dynamic analysis of a Main Combustion Engine (MCE), installed at a surface ship and another at a submarine. This equipment is essential for the ship's survivabilityin the battle, and it is mandatory to predict the possibilities of any equipment failure. Although nowadaysthere are a lot of commercial software that can be used to perform dynamic analysis, mainly based on Finite Elements Methods (FEM),the DDAM was considered a good methodology to help the designers in the first step of a dynamic analysis of a navalsystem's equipment, when the system has a few degrees of freedom, taking a low computational time.*

Keywords: *DDAM, UNDEX, Dynamic Analysis, Shock Response Spectra.*

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I. INTRODUCTION

Since the World War II (WWII), non-contact explosives have been used as military weapons to cause severe damages in ships and submarine equipment. The high pressure generated by the shock wavesin the underwater explosions (UNDEX) can cause structural failuresin the ship and/or the severe damages in an essential equipment on board. The shock waves produce high accelerations in the naval equipment, and, if the equipment is not designed to withstand this, it can impact in the failure of the mission in a battle.

This has stimulated a lot of technical effort to improve the *survivability* of the naval (combatant) ship, regarding the shell plates reinforcements, structural members and the application of the equipment shock mounts. *Survivability* is an important military term that, according to OPNAV INSTRUCTION 9070.1B (2017), it is a measure of the capability of the ship, mission-critical systems andcrew, to perform assigned warfare missions and of the protection provided to the crew to preventserious injury or death. It is a very important characteristic of a naval ship and it is covered by another three concepts: *susceptibility, vulnerability, and recoverability*. *Susceptibility* is a measure of the capability of the ship, mission-critical systems, andcrew to avoid and or defeat an attack and is a function of operational tactics, signature reduction,countermeasures, and self-defense system effectiveness. The *vulnerability*is a measure of the capability of the ship, mission-critical systems, and crew to withstand the initial damage effects, and to continue to perform assigned primary warfare missions and protectthe crew from serious injury or death. The concept of *recoverability* is the capability of the ship and crew, after initial damageeffects, whatever the cause, to take emergency action to contain and control damage, prevent lossof a damaged ship, minimize personnel casualties, and to restore and sustain primary missioncapabilities.

In this context, in the early of 1950 it was conceived the Dynamic Design Analysis Method (DDAM), that is a powerful tool to support the engineers to predict the response of naval equipment. This procedure helps to improve the concept of vulnerability of the ship, making the essential equipment able to withstand the initial damage effects, and to continue to perform assigned primary warfare missions.

As informed in Brenner *et al* (2007), the DDAM is primarily used as a design tool, to guide contractors in designing com ponents that should pass the shock test, and is also used to qualify components that are too large to test.

Based on the results of the DDAM, the engineers can get the accelerations, the pseudo-velocity (the relative velocity of the discretized equipment element), displacement in the frequency domain and the stress at the specific points. The results are a guide for the designers, in the first step, of performing a dynamic analysis of a naval equipment. The methodology is very useful and, if the system is considered as one with few degrees of freedom, the DDAM takes a low computational time if compared with a FEM.

II. MATHEMATICAL MODEL AND PHYSICAL DESCRIPTION

In the sequence, a simple mathematical model, the UNDEX phenomena and the DDAM methodology are presented. This is an important step to understand from the basic concepts until the main considerations to perform the complex analysis and interactions involving dynamic analysis of naval systems.

Although in this work the formulation of the UNDEX phenomena is not directly employed, the authors inserted this topic to disseminate the empirical formulation of this very important phenomenon that have caused a lot of damages and sinking in the naval ships since the WWII.

2.1 DYNAMIC ANALYSIS

The Eq. (1) shows the well-known equation of motion of a SDOF (single degree of freedom) system:

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = 0 \quad (1)$$

where M, C, and $K$are the mass, damping, and stiffness, respectively, and$\ddot{x}$,$\dot{x}$ and x are the acceleration, velocity and displacement, respectively.

According to Rao (2018), the solution of the Eq. (1) can be given by Eq. (2).

$$x(t) = \mathrm{e}^{-\zeta\omega_n \mathrm{t}}\left\{x_0\, cos(\omega_d \mathrm{t}) + \frac{\dot{x}_0 + x_0\zeta\omega_n}{\omega_d} sin(\omega_d \mathrm{t})\right\} \quad (2)$$

where$x(t)$ is the response (displacement) of the system, ζ is the damping ratio, ω_n is the natural frequency and$\omega_d$is the frequency of damped vibration.

The natural frequency ω_n and the frequency of damped vibration ate defined by the Eq. 3 and Eq. 4.

$$\omega_n = \sqrt{\frac{K}{M}} \quad (3)$$

$$\omega_d = \omega_n\sqrt{1-\zeta^2} = \sqrt{\frac{K}{M} - (\frac{C}{2M})^2} \quad (4)$$

where$C$ is the viscous damping coefficient.

When it is applied an external force in a mechanical system in a short time it is called an impulse.The Figure 1 presents the system subjected to an impulse and the response of the system.

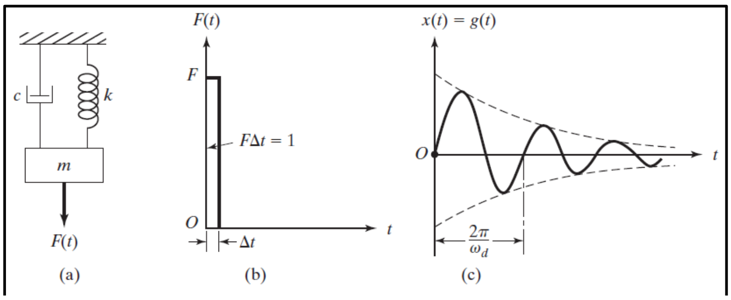

Figure 1. Basic elements of a SDOF subjected to an impulse (a), the impulsive force in the time (b) and the response (c).
Source: Rao (2018).

Equation(5) bellow considers the impulsive force interacting with a SDOF system:

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = \mathrm{F}(t) \tag{5}$$

where $\boldsymbol{F}(t)$is the impulsive force (on the time).

If the mass is at rest before the unit impulse is applied x(t=0) = x0 =0 and, from the impulse-momentum relation, $F = 1 = M\dot{x}(t = 0) = M\dot{x}_0$, the initial conditions are given by Eq. (6) and Eq. (7).

$$\mathrm{x(t = 0)} = x_0 = 0 \tag{6}$$

$$\dot{x}\ \mathrm{(t = 0)} = \dot{x}_0 = 1/\mathrm{M} \tag{7}$$

Equation (8) presents the solution of Eq. (2) considering the initial conditions presented in Eq. (6) and Eq. (7), as mentioned in Rao (2018).

$$x(t) = \frac{\mathrm{F\ e}^{-\zeta\omega_n \mathrm{t}}}{M\omega_d}.\sin(\omega_d \mathrm{t}) = \mathrm{F.g}(t) \tag{8}$$

Rao (2018) also writes that when a force is applied for short duration, usually for a period of less than one natural time period, it is called a shock load. A shock causes a sudden increase in the displacement, velocity, acceleration and stress in a mechanical system. A shock may be described by a pulse shock, velocity shock, or a shock response spectrum (SRS). The shock response spectrum describes the way in which a machine or structure responds to a specific shock instead of describing the shock itself.

According to AFNOR (1993) *apud* Silva (2005), the shock occurs when a force, a position, a velocity or an acceleration is abruptly modified and creates atransient state in the system considered. The modification is normally regarded as abrupt if it occurs ina time period that is short compared with the natural period concerned. Regarding the shape of the shock signal, it can be represented exactly in simple mathematical terms and standards generally specify three main patterns: half-sine, terminal peak sawtooth (or triangular pulse) and rectangular pulse.Silva (2005) also define that the half-sine is a simple shock for which the acceleration time curve has the form of a half-period (partpositive or negative) of a sinusoid. The triangular pulse is a simple shock for which the acceleration – time curve has the shape of a triangle, where acceleration increases linearly up to a maximum value and then instantly decreases to zero. The rectangular pulse is a simple shock for which the acceleration time curve increases instantaneously up to a givenvalue, remains constant throughout the signal, and decreases instantaneously to zero. It is important to define which pulse shape will be used, because the response of the system depends directly on the chosen pulse shape, like the ones presented in the Figure 2.

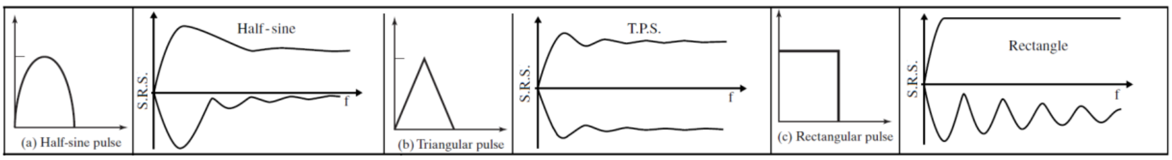

Figure 2.Response for three types of shock pulses.
Source: Adapted from Silva (2005).

A useful tool to analyze the severity of the shock in a mechanical system is the Shock Response Spectrum (SRS) that, according to Silva (2005) is a curve representative of the variations of the largest response of a linear SDOF system subjected to a mechanical excitation, plotted against its natural frequency, for a given value of its damping ratio. All the axes are in a logarithmic scale. Horizontally, the natural frequency is plotted and, vertically, there is the "pseudovelocity". The graph shows lines rotated 45º anti-clockwise from the horizontal position. These are lines of constant relative displacement. On the other hand, the graph also shows lines rotated 45° clockwise from the horizontal position. These are lines of constant absolute acceleration of the masses.

Figure 3 shows an example of a SRS, in which appears the NATO (*North Atlantic Treaty Organization*)standard shock levels limits, cited by the Bureau Veritas (2020), Rules for the Classification of Naval Ships. This is and important reference source to guide the engineers to design the naval equipment, if not presented another requirement.

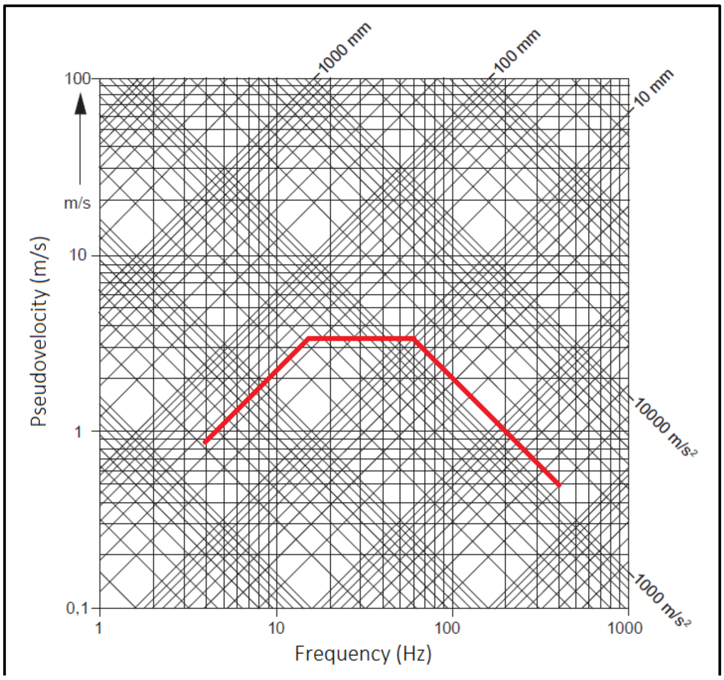

Figure 3.Shock Response Spectrum (SRS).
Source: Adapted from Rules for the Classification of Naval Ships - Bureau Veritas (2020).

2.2 UNDEX PHENOMENA

According to Keil (1961) and Reid (1996) *apud* Lu and Brown (2019), UNDEX events can be categorized as near-field and farfield.In near-field explosions, the explosive is sufficientlyclose to the ship's hull and sufficiently large that its effectsresult in large plastic deformation. Since the energy ismostly absorbed in the ship's hull by the deformation, thedestructive effect is typically local, but severe.Far-field UNDEX occurs when theexplosive is small or relatively far from the ship's hull, andthe shock wave released by the explosive has decayed to alinear acoustic wave with a discontinuous pressure profileas it reaches the ship. Onboardequipment and people can be exposed to dangerous accelerations, and the structural effects on the ship hull is mostly or entire elastic.

The hydrodynamics of the explosions inside the water are complex and it is not the scope of this work. More details on this topic can be found in Cole (1948).

Two types of loadings have fundamentally different physicsand consequences on the ship structure. The first is the shock wave and the second is the pressure wave. The duration of the shock wave is extremely small (decreases exponentially and lasts a few milliseconds) and affects the local hull structure and equipment on board, while the duration of the pressurepulses (generated by the pressure waves)is longer and can induce the global hullgirder vibrations. Tthis phenomena can be graphically represented as shown in Figure 4.

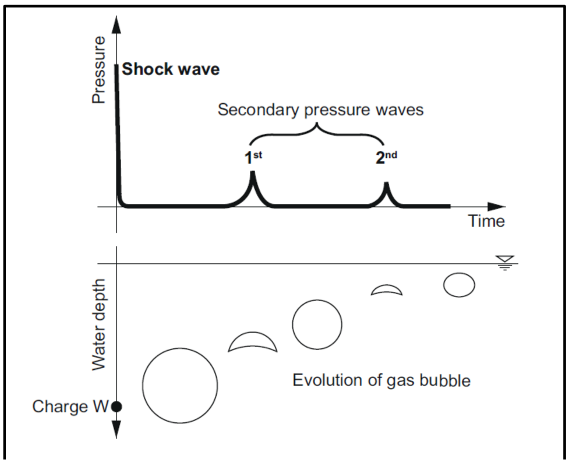

Figure 4. Evolution of the pressure and gas bubble in the UNDEX phenomena.
Source:Rules for the Classification of Naval Ships - Bureau Veritas (2020).

Another important definition is the Shock Factor (SF), or in ships, Keel Shock Factor (KSF). According to Reid (1996), it is the quantity used to characterize the severity of the shockwave, defined by the Eq. (9).

$$\mathrm{KSF} = \frac{W^n}{R} . \frac{1+\sin(\theta)}{2} \tag{9}$$

where$W$ is the mass of the explosive (Kg), R is the standoff distance from the charge to the hull (m), θ is the shock wave angle between a horizontal line and distance from the measured point of the charge and n is a value that depends on the experimental conditions (½ is usually adopted).

The shock wave is modeled according to Rules for the Classification of Naval Ships - Bureau Veritas (2020), the empirical formulations for modeling the pressure are showed in the Eq. (10), Eq. (11) and Eq. (12).

$$\mathrm{P}_{max} = K_1 (\frac{\sqrt{W}}{R})^{A_1} \tag{10}$$

$$\mathrm{P}_0 = \mathrm{P}_{max}{}^{(-\frac{t}{\eta})} \tag{11}$$

$$\eta = K_2 \sqrt{W} (\frac{\sqrt[3]{W}}{R})^{A_2} \tag{12}$$

whereP_{max} is the maximum pressure (MPa) at the distance R from thedetonation point, t is time (milliseconds), ηis adecay constant that depends on the type of charge. The constants K_1, K_2, A_1 and A_2 are showed in Table 1. These coefficients are function of the kind of explosive, as informed in Reid (1996).

Table 1. Equivalent coefficients for different materials.
Source: Reid (1996).

Explosive/ Coefficient	HBX-1	TNT	PENT	NUCLEAR
K_1	53.51	52.12	56.21	1.06E+04
K_2	0.092	0.092	0.086	3.627
A_1	1.144	1.18	1.194	1.13
A_2	-0.247	-0.185	-0.257	-0.22

The effect of the high energy and pressures generated by a UNDEX in the naval ship cause high accelerations, velocity and displacements on all systems and components onboard. But these motions are not of the same value on the ship components; they have different values around the ship position. This difference can be visualized in the Figure 5, from the Keil (1961). It is showed the response of a destroyer (a class of military surface ship). It is possible to see the high peak acceleration (A) in a very short time – around 15 milliseconds and at (B) the different part responses of the destroyer – keel, main deck and superstructure.

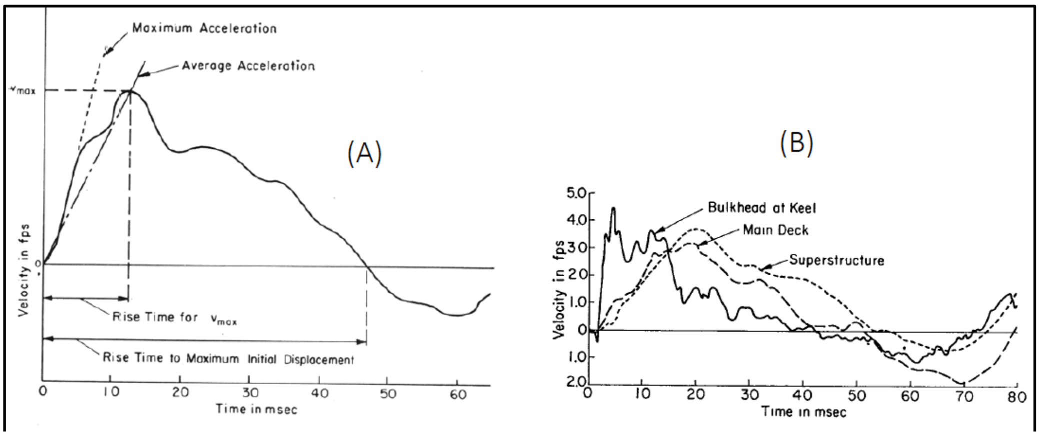

Figure 5. The response of a destroyer caused by UNDEX.
Source: Keil (1961).

2.3 DDAM

As covered in the UNDEX Phenomena session above, the ship response due to the shock and the pressure waves is not the same along the ship, in other words, it depends on where the equipment is placed onboard. For this reason, according to NAVSEA-0908-LP-000-3010 (1995), the position of the equipment is defined in three categories: hull mounted, deck mounted and shell mounted. Hull mounted is the equipment installed on basic framing, tank tops, inner bottom, shell plating above the water line and on the structural bulkheads bellow the main deck. Deck mounted is the equipment mounted on decks, platforms, non-structural bulkheads and structural bulkheads above the main deck. Shell mounted is the equipment mounted directly to the shell plating bellow the water line.

O'Hara and Belsheim (1963) indicate that the equipment shock design is categorized in two classifications: elastic response and elastic-plastic response. Elastic response is the equipment-and-foundation system that any local plastic deformation is not allowed. Elastic-plastic response is the equipment-and-foundation system that several times the maximum elastic deflection is permissible.

According to NEi Nastran (2009), analytically, DDAM works like a normal response spectrum analysis. First, the model is run to determinethe fixed base natural frequencies and mode shapes. These are combined with the mass matrix to formthe participation factors and modal effective masses. In a classical response spectrum analysis, theexcitations would then be read off of the spectrum at each of the natural frequencies, and then usedthese for the excitation portion of the analysis.

The principle of DDAM is to add all mass and the stiffness coefficient contributions on the analyzed mechanical system to compute the modal shapes, to after that, obtain acceleration and velocity. The Figure 6 illustrate the DDAM principle.

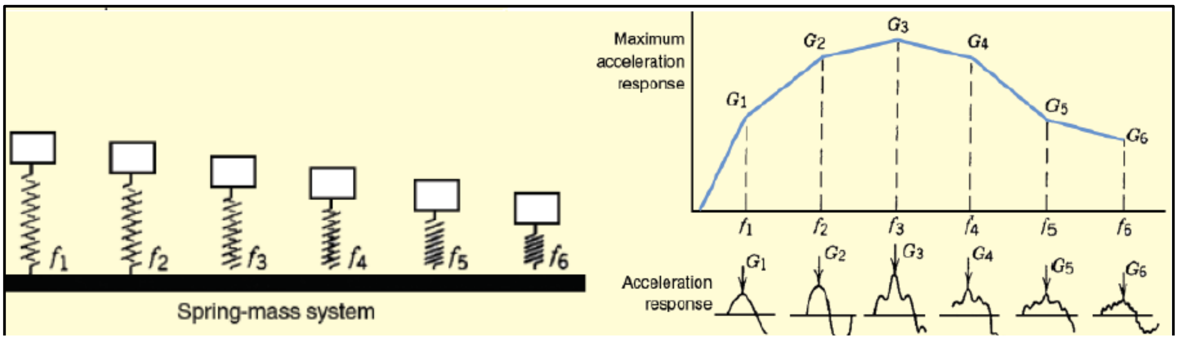

Figure 6. The SRS of the Spring-Mass system as output in DDAM.
Source:Tasdelen (2018).

The DDAM do not consider damping in the mechanical system, then the Eq. (1) is reduced to Eq. (13).

$$M\ddot{x}(t) + Kx(t) = 0 \tag{13}$$

The Eq. (13) is transformed into an eigenvalue problem, as showed in the Eq. (14). According to NEi Nastran (2009), each solutionproduces a unique ω called the eigenvalue, and a corresponding unique {x} called the eigenvector,usually designated as ϕ. Each eigenvalue is a natural frequency of the system, and there is one for each degree of freedom (or row) in the M (mass) and K (stiffness) matrices.

$$\{[K] - \omega^2[\mathrm{M}]\}\phi = 0 \tag{14}$$

The modal mass M_i of each mode can be computed by the Eq. (15).

$$M_i = [M]\phi_a^T \tag{15}$$

where ϕ_a^T is the transposed matrix of the eigenvector ϕ at the a^th^ mode and $[M]$ is the mass matrix.

The Participation Factor is defined as per the Eq. (16).

$$P_a = \frac{\sum_{i=1}^{n} M_i}{\sum_{i=1}^{n} \phi_{ia}^2 . M} \tag{16}$$

where n is the number of degrees of freedom (DOF) in the model and ϕ_{ia} is the eigenvector component of the a^{th} mode at the i^{th} degree offreedom.
The modal effective weight can be computed as showed in Eq. (17).

$$W_a = \frac{g}{1000} * P_a * M_i \tag{17}$$

where P_a is the participation factor, M_i is the modal mass and g is the gravity acceleration (386.4 in/sec^2). The output of W_a is in kips (thousands of pounds).

As showed in O'hara and Belsheim (1963) *apud*Tasdelen (2018), the input acceleration and the input velocity at certain mode can be calculated by the Eq. (18) and Eq. (19) respectively.

$$A_0 = A[\frac{BW_a^2 + CW_a + D}{(E + W_a)^F}] \tag{18}$$

$$V_0 = A[\frac{B + W_a}{C + W_a}] \tag{19}$$

where W_a is the modal effective weight (in kips) of thatmode and the coefficients A, B, C, D, E and F are given on Table 2. These coefficients depend on the naval vessel (surface ship or submarine) and where the system is installed, as mentioned in the beginning of this session.

Table 2. Input Acceleration and Velocity Shock Value Coefficients.
Source: O'hara and Belsheim (1963) *apud*Tasdelen (2018).

Input Acceleration Shock Value Coefficients		A	B	C	D	E	F
Submarine	Hull mounted system	10.4	0	1	480	20	1
	Deck mounted system	5.2	0	1	480	20	1
	Shell mounted system	5.2	0	1	480	20	1
Surface Ship	Hull mounted system	20	1	49.5	450	6	2
	Deck mounted system	10	1	49.5	450	6	2
	Shell mounted system	10	1	49.5	450	6	2

Input Velocity Shock Value Coefficients		A	B	C
Submarine	Hull mounted system	20	480	20
	Deck mounted system	10	480	20
	Shell mounted system	100	480	20
Surface Ship	Hull mounted system	60	12	6
	Deck mounted system	30	12	6
	Shell mounted system	120	12	6

Then, according to O'hara and Belsheim (1963) *apud*Tasdelen (2018), input acceleration and the input velocity at certain mode as showed in Eq. (19) and Eq. (20) varies depending on the direction onboard and if the design allow plastic deformation (elastic-plastic) or if it is forbidden (elastic), preserving the original physical dimensions after the shock. These directional coefficients are presented in Table 3.

Table 3. Directional Coefficients for DDAM Analysis.
Source: O'hara and Belsheim (1963) *apud*Tasdelen (2018).

SHIP	KIND OF ASSEMBLY	V_1 (X)	V_1 (Y)	V_1 (Z)	A_1 (X)	A_1 (Y)	A_1 (Z)
Surface	Deck/elastic	0.4 V_0	0.4 V_0	1 V_0	0.4 A_0	0.4 A_0	1 A_0
Surface	Hull/elastic	0.2 V_0	0.4 V_0	1 V_0	0.2 A_0	0.4 A_0	1 A_0
Surface	Shell/elastic	0.1 V_0	0.2 V_0	1 V_0	0.1 A_0	0.2 A_0	1 A_0
Surface	Deck/elastic-plastic	0.2 V_0	0.2 V_0	0.5 V_0	0.4 A_0	0.4 A_0	1 A_0
Surface	Hull/ elastic-plastic	0.1 V_0	0.2 V_0	0.5 V_0	0.2 A_0	0.4 A_0	1 A_0
Submarine	Deck/elastic	0.8 V_0	2 V_0	1 V_0	0.8 A_0	2 A_0	1 A_0
Submarine	Hull/elastic	0.4 V_0	1 V_0	1 V_0	0.4 A_0	1 A_0	1 A_0
Submarine	Shell/elastic	0.08 V_0	0.2 V_0	1 V_0	0.08 A_0	0.2 A_0	1 A_0
Submarine	Deck/ elastic-plastic	0.4 V_0	1 V_0	0.5 V_0	0.8 A_0	2 A_0	1 A_0
Submarine	Hull/ elastic-plastic	0.2 V_0	0.5 V_0	0.5 V_0	0.4 A_0	1 A_0	1 A_0
(X) = Fore/aft (Y) = Athwart-ship (Z) = Vertical							

According to O'hara and Belsheim (1963), the acceleration and the modal velocity are obtained by the Eq. (21) and Eq. (22).

$$A_i = A_0.A_1 \tag{21}$$

$$V_i = V_0.V_1 \tag{22}$$

The acceleration is computed in g (gravity acceleration) and the velocity in inch/second.

Finally,O'hara and Belsheim (1963) indicate that the modal velocity V_i shall be multiplied by the ω_a to obtain the result in terms of acceleration. The result shall be compared with the A_i and the higher value will be the Shock Design Value (SDV). But there is a minimal input acceleration that is to be considered at the design in the DDAM: the value is 6 g (2,318.4 in/s^2 or 58,86 m/s^2). If the highest value found is smaller than 6g, thus the SDV will be 6g.
It is important to mention that the input and output values for DDAM method shall be in imperial systems of units, because the developers were Americans, and for this reason, the coefficients employed are in these units. But it is possible, after the output, convert normally the result for the International System of Units.

III. CASE STUDY

This work employs the DDAM methodology to analyze the acceleration and velocity of a Main Combustion Engine (MCP) installed at two any naval ships: one surface ship (Littoral Combat Ship - LCS) and one submarine. The MCP of both ships is installed at the hull mounted system, inside the engine rooms. Figure 7 shows the localization of the MCP onboard.

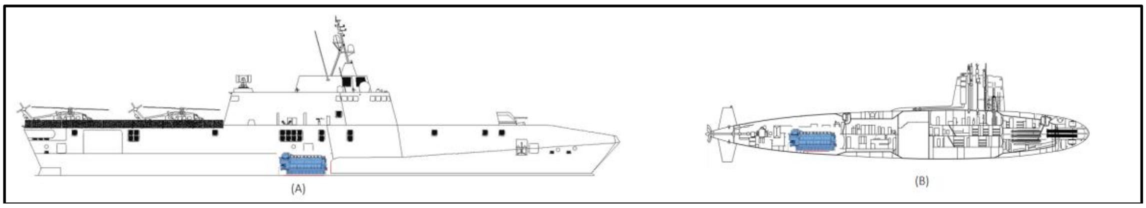

Figure 7. MCP installed at the engine rooms at the LCS (A) and the submarine (B).

The model of the MCP selected was the commercial model series 4000 12V, with the 3500 BHP output power, the mass is about 8000 Kg and the mass moment of inertia is J_0=8000 Kg.m^4. The total length of the engine is 3,840 meters, with center of gravity (CG) is in a dislocated position (l_1=1,840 m and l_2=2,000 m). It is important to mention that the MCP was selected in the first step, it was not applied the refinements, like hydrodynamic requirements among others.

For the MCP mounts, it was selected the navy mount series and the stiffness adopted (K) is 1390 kN/m. It was employed four mounts and to simplify the analysis the system was considered as 2 degrees of freedom: one at the x(t) and another the θ(t), both in relation to the vertical direction of the ship. For this simplification, it was considered the two springs in parallel (K_1=$K_{2=}$ 2780 kN/m), as visualized in the Figure 8. The characteristics of the system considered in this work are hypothetical and were taken for investigative purpose only.

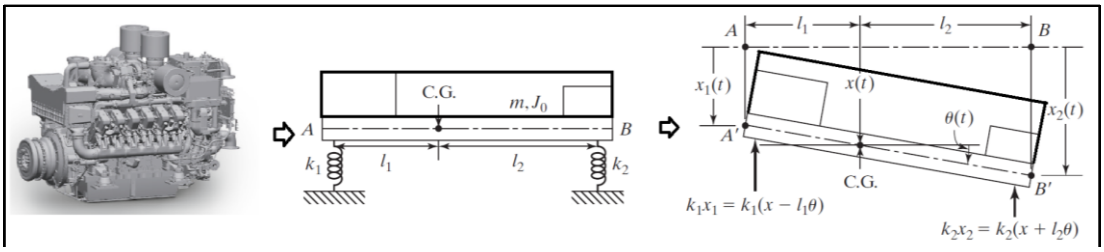

Figure 8. Mounts standing the MCP.

IV. RESULTS AND DISCUSSION

The vibration problem can be represented in matrix form (Rao, 2018) as shows the Eq. (23).

$$\begin{bmatrix} -m\omega^2 + k_1 + k_2 & -k_1 * l_1 + k_2 * l_2 \\ -k_1 * l_1 + k_2 * l_2 & -J_0 * \omega^2 + k_1 * l_1^2 + k_2 * l_2^2 \end{bmatrix} \begin{Bmatrix} X \\ \theta \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix} \tag{23}$$

Solving the eigenvalue problem of Eq. (23), the natural vibration frequencies were obtained: ω_1=26.33 rad/s (4.19 Hz) and ω_2=50.67 rad/s (8.06 Hz). Table 4 presents the eigenvalues, the eigenvectors and the modal effective weight for the two vibration modes of the system.

Table 4. Modal coefficients calculated for the DDAM analysis.

Mode	ω_a (rad/s)	ϕ_a	M_i(lbs)	W_a(kips)
1	26.33	$\begin{Bmatrix} -0.0296 \\ 1 \end{Bmatrix}$	17114.92	6391.77
2	50.67	$\begin{Bmatrix} 1 \\ 0.0296 \end{Bmatrix}$	18159.03	7195.43

Taking the eigenvectors for first and second modes, it can be computed the modal mass M_i and the participation factor P_a trough the Eq. (15) and Eq. (16), respectively. With these two parameters computed, it is possible to obtain the modal effective weight W_a of each mode (as presented in Table 4).
Taking the two values of the modal effective weight W_a, it can be computed the input acceleration and velocity (A_0 and V_0 respectively) through the Eq. (18) and (19).
Next, it is necessary multiply the A_0 and V_0 for the values presented in the Table 3 (according the ship, kind of assembly and the direction). For this study, it was considering the MCE installed above a hull mounting, with elastic deformation restriction and, for the preference direction, it was selected the Z direction (vertical).
Finally, it is possible to obtain the A_i and V_i output values from DDAM method, shown in Table 5, using Eq. (21) and Eq. (22).

Table 5. Output values from the DDAM analysis.

Ship	Mode	W_a(kips)	A_0 (g)	V_0 (in/s)	A_i (g)	V_i (in/s)	V_i*ω_a (in/s^2)
LCS	1	6391.77	20.11	60.05	20.11	60.05	1581.37
	2	7195.43	20.10	60.05	20.10	60.05	3043.14
Submarine	1	6391.77	11.14	21.43	11.14	21.43	564.41
	2	7195.43	11.06	21.27	11.06	21.27	1078.15

As indicated in O'hara and Belsheim (1963), the result of V_i shall be multiplied by ω_a to obtain the acceleration value and, after that, compared with the A_i. The highest value will be the Design Shock Value adopted. Doing the conversions (1 g = 386.4 in/s^2) in the rightmost column of Table 5, one can note that there is no value higher than A_i. Therefore, the DSV for the LCS is 20.11 g for the first mode and 20.10 g for the second mode. For the submarine, the DSV is 11.14 g for the first mode and 11.06 g for the second mode. As an initial approach, these values can be used to verify if the naval equipment (MCE and mountings) meet the requirements specified by the purchaser/shipowner.

V. CONCLUSIONS

After the implementation of DDAM, for the case study exposed in this paper, it is possible to take some conclusions:

1 – the method employs modal concepts, as present in the dynamic analysis;

2 – the mechanical system has the same modal effective weight. The difference when installed in different ships are due to the coefficients used for computing the accelerations and velocities;

3 – for a mechanical system with a few degrees of freedom, the result can be computed analytically, spending a low computational time;

4 – for the mechanical system (MCE) considered in this study, the acceleration and the velocity of the surface ship (LCS) are about 2 times higher than in a submarine. It is due to the coefficients presents in the equations to compute accelerations and velocities. They represent the physical distinguished effects in the ships regarding buoyancy and density between the water and the air; and

5 – the stiffness and mass of the ship structure, as well the equipment foundations, should be considered in the DDAM method to increase the accuracy of the equipment accelerations and velocities determined by the method.

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