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Rev.1

SHOCK DESIGN CRITERIA FOR SURFACE SHIPS

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Chapter 1. INTRODUCTION

The primary purposes of this report are as follows:

a. Provide technically oriented shock design criteria for Navy review and approval of shock design calculations.

b. Provide a limited amount of general background/educational material concerning application of the Dynamic Design Analysis Method (DDAM).

This report is intended to convey Navy dynamic shock analysis requirements to engineers who possess an educational or experience background in the fields of vibration analysis, structural dynamics and stress analysis. If the user finds that this report does not provide information sufficient to permit full and efficient satisfaction of all specified dynamic shock analysis requirements, the cognizant contracting officer should be contacted for additional information.

The requirements indicated by this report are subject to modification by applicable specifications. Users of this report should carefully review applicable specifications to determine whether any of the provisions of this report have been modified.

The contents of this report are founded upon dynamic analysis procedures originally developed by the Naval Research Laboratory, Washington, D.C. These procedures were originally reported in the following reports:

NAVSHIPS Report "Shock Design of Shipboard Equipment, Dynamic Design Analysis Method", R.O. Belsheim and G. O'Hara 250-423-30 dated May 1, 1961,

BUSHIPS Report "Shock Design of Shipboard Equipment, Interim design Inputs for Submarines and Surface Ship Equipment"(U), 250-423-31, dated January 1, 1961 (Confidential), and

Naval Research Laboratory Report, "Interim Design Values for Shock Design of Shipboard Equipment", G.J. O'Hara and R.O. Belsheim NRL 1396, dated February 1, 1963

This report is a revision of NAVSEA 0908-LP-000-3010 which was prepared by the Supervisor of Shipbuilding, Conversion and Repair, USN, Brooklyn under the direction of the Naval Sea Systems Command. Portions of this report are directly derived from the reports referred to above. Where the requirements of this document are in conflict with previous DDAM guidance, this document shall take precedence.

Sections 3, 4 and 5 of this report contain example engineering calculations that illustrate

Shock Design Value (D) - Numerical representation of shock response (acceleration or velocity) used for each mode in a dynamic analysis. The values depend on the mounting location of the equipment, structure, or foundation, the direction of shock response (vertical, athwartship, or fore and aft) and the item's design requirements (elastic or elastic-plastic). Formulas for the computation of shock design values are contained in "Shock Design Values", Design Data Sheet DDS 072-1 (CONFIDENTIAL).

Shock Grade - Classification category of required system or equipment performance (operability) levels in a combat environment. Items classified as Grade A are systems or equipment which are essential to the safety and continued mission capability of the ship. Accordingly they must remain operable and not create a hazard when exposed to combat environment corresponding to full shock design levels. Grade B items are items whose operation is not essential to the safety or mission capability of the ship but could become a hazard to personnel, Grade A items or the ship as a whole as a result of exposure to design level shock loading.

Shock Input - Refers to the shock design values as an input to the DDAM or to the physical shock loading due to an underwater explosion.

Shock Response - The dynamic behavior of an equipment, structure or foundation due to shock loading. Shock response generally refers to the displacement, velocity, acceleration, force, stress or strain experienced by an item.

Split Modes - A closely spaced modes phenomenon where, for example, a normal mode of the mathematical model is divided into two modes, close in frequency with each mode containing approximately equal portions of modal effective mass. The sum of the modal effective mass of the two split modes is approximately equal to that of the original single normal mode. Since the shock design values are inversely related to the modal effective mass, this artificial splitting of a mode results in a potentially erroneous increase in shock loading to the system.

Uncombined Modes - A closely spaced modes phenomenon where similar portions of the system are prevented from combining into a single mode.

Uni-Directional Response Analysis - Shock analysis which evaluates system response in only the direction of shock input. The model may be linear, planar or three dimensional.

Chapter 3. DYNAMIC DESIGN ANALYSIS METHOD

A shipboard equipment or structure, when subjected to a specified shock motion, will experience stresses and deflections in excess of those present under static conditions. The Dynamic Design Analysis Method (DDAM), developed to supersede the static G design method, is used to evaluate the shock capability of various shipboard equipment and structures. A static G analysis does not constitute an alternative to a dynamic response analysis. The first step in the evaluation process involves representing the item in question by a mathematical model. DDAM models essentially reduce an equipment or structure to an equivalent mass-elastic system which is used to design the system to sustain dynamic stresses induced by shock response motions. The desired strength levels are specified in terms of spectral values which are frequency and mass dependent. By setting up and solving the equations of motion of a mass-elastic system, forces and displacements associated with each mass and structural element in the system are determined. These forces and displacements are used to determine the stresses and/or deflections of various components of the equipment, the foundation and the hold-down means. These forces, stresses, or deflections are then compared with specified allowable values to determine the acceptability of the analyzed items from a shock standpoint.

As part of Total Quality Management (TQM) a graphical description of the total shock hardening design process is provided (as Figures 3-1, 3-2 and 3-3) to aid in understanding the material contained in this report. The process description covers the analysis methodology, evaluation considerations, applicable resources and interaction between the Navy approval agency and the analyst's organization.

Figure 3-1 is an overview of the process showing the relation of testing and analysis for Grade A and Grade B equipment and their foundations. Figure 3-2 describes the process of equipment shock qualification by DDAM and Figure 3-3 shows the procedural steps associated with foundation shock qualification. The details of the flow charts are presented throughout the text of this report.

In order to simplify discussion of the shock analysis procedure mentioned above, it will be divided into five distinct, yet interrelated, phases. These five phases will be called:

(1) Problem formulation phase
(2) Mathematical modeling phase
(3) Coefficient computation phase
(4) Dynamic computation phase
(5) Evaluation phase

Each of these phases is discussed on the following pages. The analysis criteria presented are applicable to all dynamic analyses, unless otherwise stated herein. Special considerations which apply to design of foundations and Grade B items are described in Chapters 4 and 5 of this report, respectively.

When the DDAM was first implemented in the 1960's, only manual calculation methods or simplistic computer codes were available. The calculations were performed strictly in the five-phase approach described above. With the advent of the powerful finite element computer programs, the distinctions between the various phases have become less clear. For example, current finite element programs generally permit the user to perform the coefficient computations and dynamic computations (phases (3) and (4)) in one step. The DDAM, in conjunction with finite element analysis is described throughout this report.

The limitations of the DDAM must be clearly recognized by the users of the method so that, if necessary, they can initiate a request for approval of an alternate approach or approval of special modeling considerations. First of all, the procedure is based on the presumption that the equipment being analyzed can be represented as a linear, elastic system with discrete modes. Second, except as inherent in the shock design values, damping is neglected in the DDAM which, for most shipboard equipment, is a reasonably valid assumption since shock-induced motions persist for only a few cycles of vibratory motion. For very low frequency systems (less than 5 Hertz) the DDAM may not be appropriate. Finally, where closely spaced modes exist in an analysis, DDAM may produce excessive responses. For these cases, as well as cases of non-linear or non-elastic systems, appropriate modeling assumptions must be developed or a NAVSEA approved alternate analysis method should be used to overcome the limitation. Similarly, analyses of foundations for very light weight equipment and analyses of equipment external to the hull will require appropriate modeling assumptions or alternate analysis/qualification methods. The specified shock spectrum design acceleration and velocity values are general in nature. While they have been derived from large scale model tests as well as data recorded in past full ship shock tests, they make no distinction between sizes and types of ships (e.g. cruisers, frigates, and aircraft carriers). Although DDAM defines shock design values for various mounting locations, for a given model, DDAM assumes that the shock design value is the same at every point where that mathematical model is attached to the fixed base. This may not be strictly applicable for widely distributed systems. The shock design values also do not distinguish between the motion differences expected at various decks within a given type of ship. Where such distinctions are expected to be critical in the evaluation of the equipment under consideration, inclusion of portions of the ship's structure in the analysis may be essential. Alternate methods of analysis, if specified by the Navy, employing motion inputs measured in a test of an identical or similar ship may be used when the general DDAM inputs are judged by the cognizant Navy approval authority to be inapplicable to the analysis of a particularly critical item of equipment.

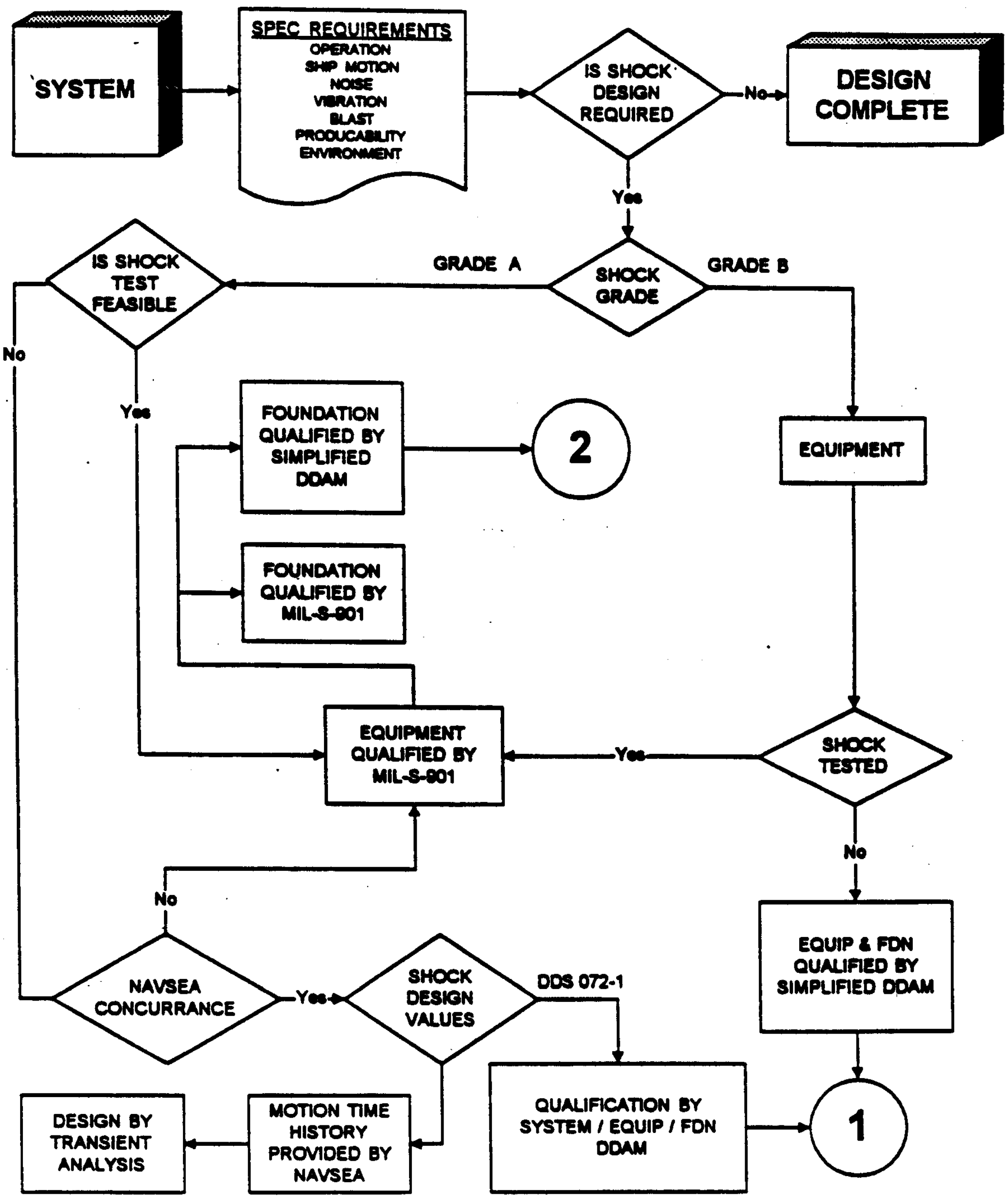


Figure 3-1 Shock Qualification Process - Overview

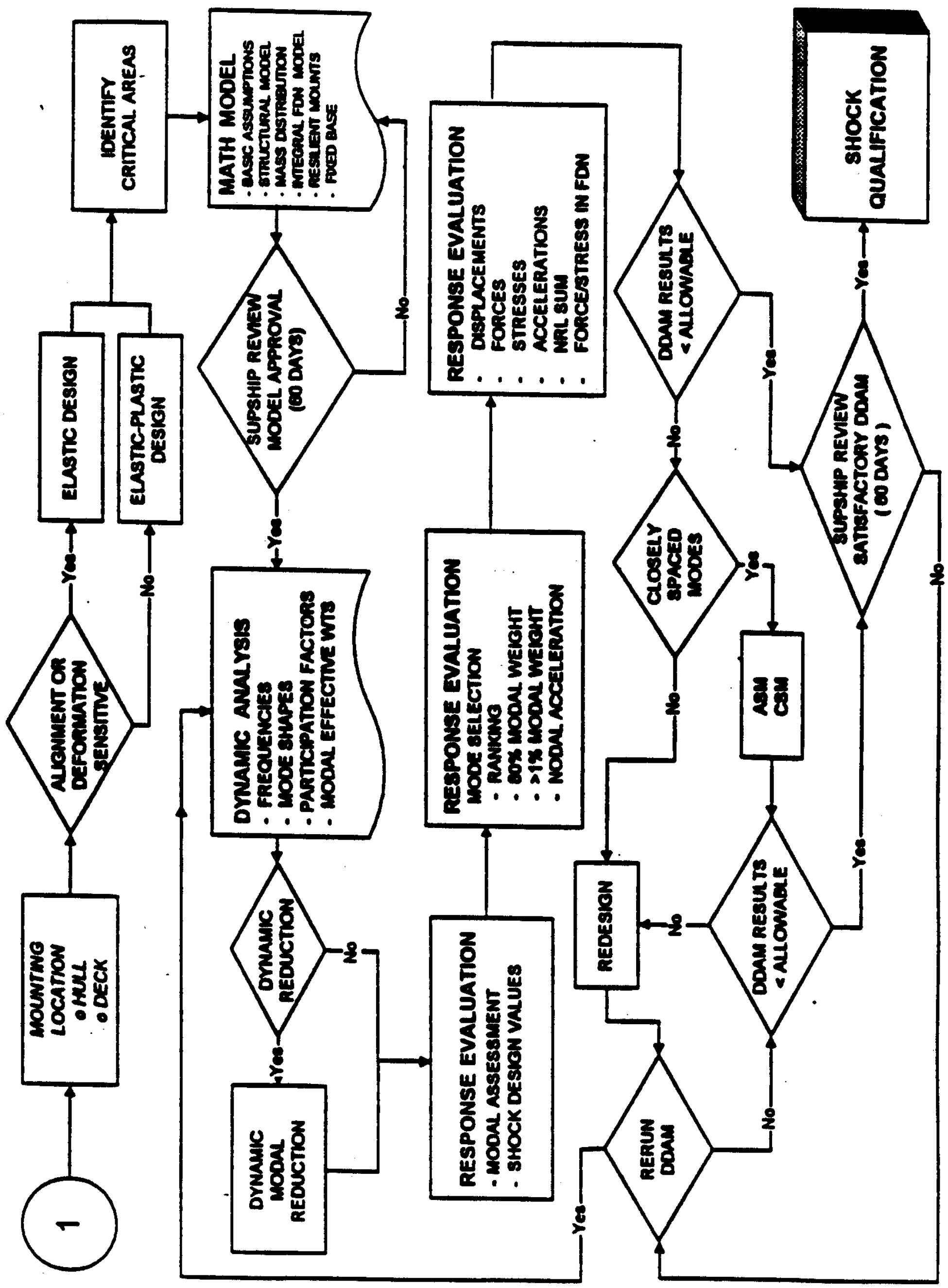


Figure 3-2 Equipment Shock Qualification By DDAM

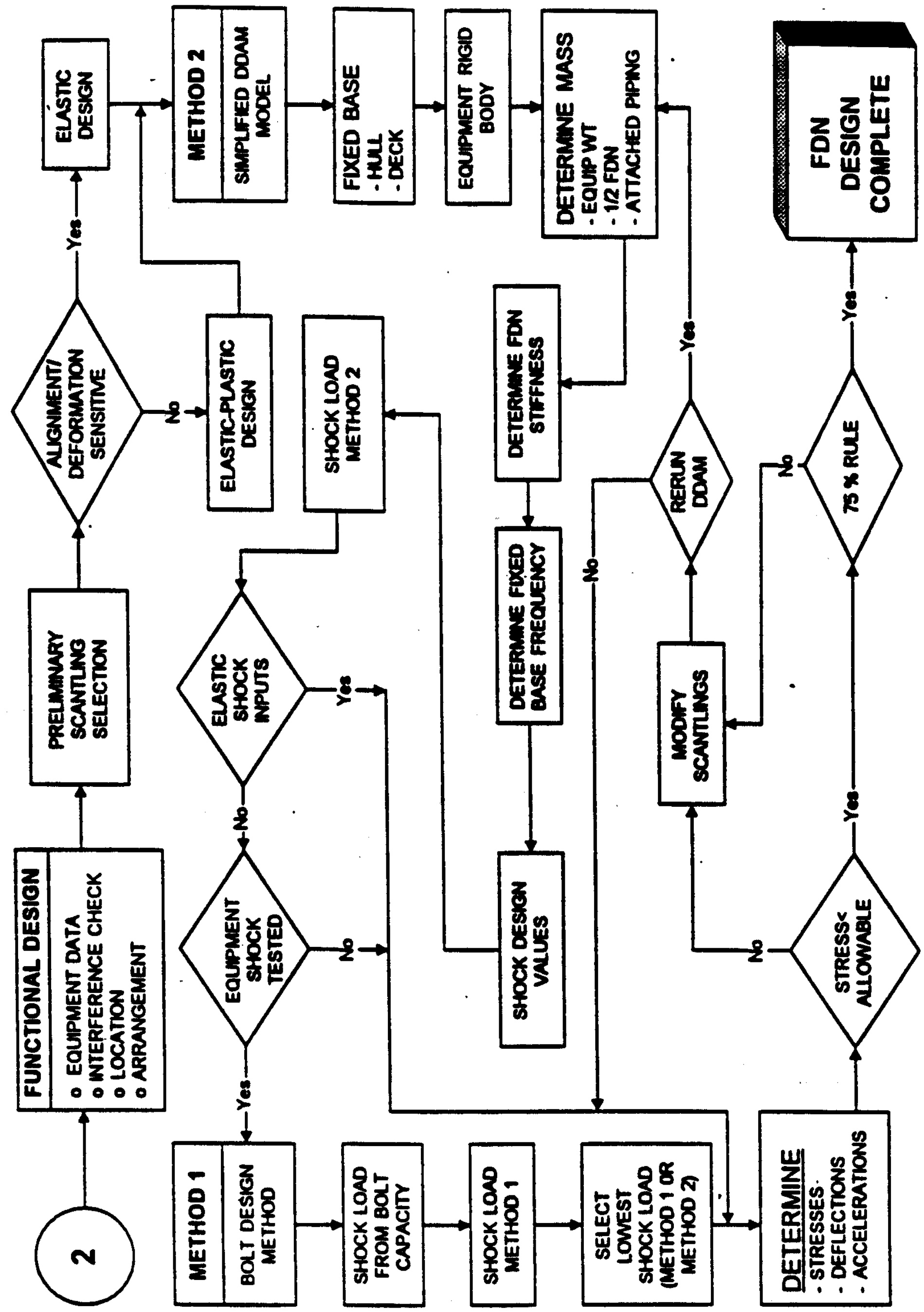


Figure 3-3 Foundation Shock Design By DDAM

3.1 Problem Formulation Phase

This phase involves a detailed study of the equipment or structure under consideration by the analyst. The analyst must determine the shock grade of the equipment or structure, the mounting location of the foundation, the shock design value to be used and the critical areas of the system which may require specific modeling considerations. For these determinations, the following requirements apply:

3.1.1 Shock Grades

The shock grades (A and B) are defined by the contract specifications. Criteria for determining shock grade requirements for an item are provided below. Grade A items are identified as such by the ship contract specifications. The specifications also designate certain Grade B items and provide general criteria for determining the shock grade of items which are other than Grade A.

Grade A shock criteria, as defined in Chapter 2, are applicable to the items which are required for the performance or direct and vital support of mission-essential functions aboard shock hardened ships. The following are often specified as mission-essential functions:

(1) Ship control and propulsion
(2) Command and control
(3) Navigation
(4) Communications
(5) Surface, air and underwater surveillance
(6) Countermeasures
(7) Launching, retrieving, fueling, defueling, rearming, and handling of aircraft and small surface craft
(8) Essential checkout and maintenance of aircraft and ordnance
(9) Fire control, firing or launching and guidance of missiles and other weapons
(10) Stowage, handling and reloading of weapons
(11) Replenishment-at-sea (stowed configuration)
(12) Mine-hunting and sweeping
(13) Transporting and landing troops and combat payload (assault ships)
(14) Casualty and damage control
(15) Collective protection system

Grade B shock criteria, as defined in Chapter 2, are applicable to items whose operation is not essential to the safety of the ship or to the direct and vital support of mission-essential functions identified above but which, due to either location or function, could become a hazard to personnel, to Grade A items or to the ship as a whole as a result of exposure to shock.

3.1.2 Mounting Locations

All shipboard equipment and structures are, for purposes of DDAM analysis, considered to be either hull mounted, deck mounted or shell mounted through their foundations. Shock inputs for each of these types of mounting locations are defined in DDS 072-1. Figure 3-4 describes various mounting locations with respect to the level of shock design input that should be applied. Proper identification of the mounting location (See Section 4.4) is important as this will determine the proper shock design value to use for dynamic analysis (See Section 3.1.3 below). This is particularly important in the case of major items of equipment mounted on decks or on bulkheads above the main deck. In the context of the following discussion "main deck" is used to indicate the "bulkhead deck" or the uppermost deck up to which the transverse or longitudinal watertight bulkheads and shell are carried. Major equipment items are often directly connected to the keel through structural bulkheads or stanchions and may thus be subjected to hull-mounted, rather than deck-mounted shock design values. The influence of the particular ship's structure supporting such items must therefore be carefully considered prior to initiating the analysis. The symmetry of the ship's structure supporting an item of equipment must also be considered. Severe asymmetry may cause undesirable rocking motions and uneven structural loading. Since the shock design values are predicated on uniform translational motion of the fixed base and rotation of the fixed base is not considered, sufficient ship structure must be considered in the development of the mathematical model such that the location of the fixed base conforms to that DDAM assumption.

The following definitions, used in the context of DDAM, are provided for the purpose of determining the category of shock inputs to apply:

"Hull Mounted" shock design values are used for equipment mounted on basic hull framing, tank tops, inner bottom, shell plating above the water-line and structural bulkheads below the main deck (bulkhead deck). Where a structural bulkhead (grounded on the inner bottom) ends at the Main Deck, or a deck below, an item attached to the deck at that location shall be considered hull mounted.

"Deck Mounted" shock design values are used for equipment mounted on decks, platforms, non-structural bulkheads and structural bulkheads above the main deck (bulkhead deck).

"Shell Mounted" shock design values are used for equipment mounted directly to the shell plating below the water line.

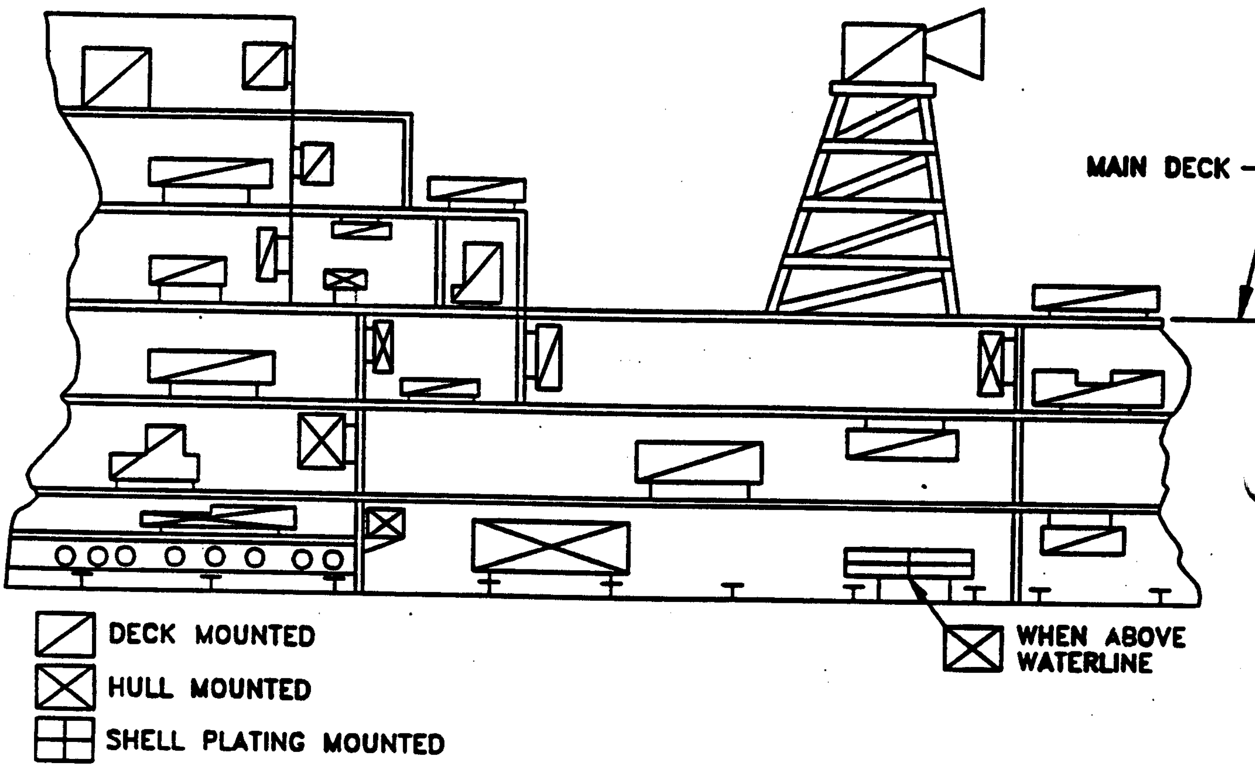


Figure 3-4 Mounting Locations for Surface Ships

In the event that an item is mounted to two different parts of the ship, for which different shock design values are specified, the larger shock design value shall be used for the analysis of the item.

Where it is necessary to evaluate specific characteristics associated with the deck structure such as load path within the ship structure or relative deflections of independent items mounted on the deck, the deck structure shall be included as part of the mathematical model. Where this is done, the fixed base of the mathematical model should extend to the structural bulkheads, stanchions, or hull framing. In these cases hull level shock inputs shall be used for design.

The following considerations shall apply for items not mounted directly on a ship's deck or on the basic hull structure:

a. <u>Shock Design Values for Items Mounted on Structural Bulkheads</u> - As indicated by DDS 072-1, hull mounted shock design values are to be used in the design of foundations mounted on structural bulkheads below the bulkhead deck. For this purpose, structural bulkheads are defined as any main transverse or longitudinal bulkhead that carries ship's loading and other bulkheads which, if removed, would require the addition of a stanchion to carry these loads. These are:

   (1) Main subdivision bulkheads.
   (2) Main longitudinal bulkheads.
   (3) Bulkheads that replace stanchions, web frames, or any other load-carrying members.
   (4) Bulkheads located or constructed such that they must be considered capable of transmitting shock loads, regardless of their function. These would include any bulkhead below the bulkhead deck which is thicker than 1/8 inch (31.75 mm) and which attaches directly to the shell or inner-bottom, or which is aligned with bulkheads, floors, or stanchions which are attached to the inner-bottom.

   For the design of foundations mounted on all other bulkheads below the bulkhead deck, and structural bulkheads above the bulkhead deck, deck inputs shall be used.

b. <u>Criteria for Lightweight Items Mounted on Machinery Space Upper Levels</u> - In analyzing lightweight items such as HVAC duct or piping systems which are supported from upper levels, the levels may be treated as decks and deck-mounted inputs applied. These criteria do not apply to analysis of the upper levels themselves. See Section 4.4.2 applicable criteria.

3.1.3 Shock Design Values

Elastic and elastic-plastic shock design values are contained in DDS 072-1. Criteria for selection of elastic versus elastic-plastic shock design values are as follows:

a. Elastic Shock Design Values - Elastic shock design values shall be used in cases where it is necessary to preserve the original physical dimensions after exposure to shock. All foundations which support rotating elements in the propulsion train (turbines, reduction gear and propeller shafting), and foundations for other alignment-critical components shall be designed to perform elastically. Foundations for rotating auxiliary equipment shall be designed elastically unless it canbe shown that plastic deformation or tilting of the equipment mounting surface will not occur or will not result in impaired equipment performance. (Note that standoff chocks may often be used to eliminate prying effects resulting from distortion of equipment mounting surfaces). Shipboard items which are known to be alignment sensitive (for purpose of shock design) are listed below. Omission of alignment sensitive items from this list does not relieve the contractor of his responsibility to assure proper selection of shock design values for all applicable items.

Main Propulsion Machinery	Auxiliary Propulsion Machinery
Ship Service Generators	Propulsion Shafting
Propulsion Shaft Bearings	Main Propulsion Reduction Gear
Propulsion Clutches	Propulsion Couplings
Turbine Brake	Main Thrust Bearing
Main CP Servo Pump	Gyroscopic Compass
Radar Antenna	Radio Antenna
Missile Directors	Gun Directors
Steering Gear (Ram Unit)	Steering Rudder System
Ammunition Hoists	Elevators and Elevator Machinery
Sonar Transducers	Catapult Machinery
Arresting Gear	Missile Launchers
Guns	Torpedo Tubes
Controllable Pitch Propeller	

b. Elastic-Plastic Shock Design Values - If elastic design is not required for the reasons stated above, elastic-plastic shock design values shall be used in cases where design by dynamic analysis is required.

c. Special Criteria for Displacement-Critical Items - In cases where deflections (rather than stresses) are critical from a shock standpoint, deflection calculations shall be based upon elastic design values.

d. Special Criteria for Hold-down/Locating Devices - In cases where equipment and/or foundations are designed to suit elastic-plastic, velocity limited shock design values, shock loadings shall be redeveloped on the basis of elastic shock design values for purposes of analysis of bolting, dowels, and similar hold-down or locating devices if shock qualification of these items by dynamic analysis is intended. Applicability of this criterion shall be limited, however, to hold-down or locating devices which are directly attached to the shipboard foundation. Hold-down or locating devices which are not at the equipment/foundation interface shall be designed to suit the same criteria that apply to other structural elements of the equipment in question. no!

3.1.4 Critical Areas

The critical areas of an equipment or structure are defined as those areas or components which are most likely to exceed failure criteria under shock loading. For purposes of these requirements, "failures" in a Grade A system are those which could cause functional impairment of the system. "Failures" in a Grade B system are those which will constitute a hazard as defined for Grade B items in the applicable contract specifications. The analyst shall construct the model so that necessary information (stresses, deflections) can be obtained for these critical areas. Typical critical areas of investigation for major systems normally required by the shipbuilding specifications to be designed by DDAM are contained in the SUPSHIP Brooklyn guidance manuals referred to in Section 3.6.1. The systems include: the rudder and rudder stock, main propulsion shafting system (excluding propeller), masts, and main reduction gear.

The intent of these SUPSHIP Brooklyn manuals is to provide the analyst with guidance in modeling and dynamically analyzing a specific system or equipment. Besides critical areas of investigation, the aforementioned manuals also contain information on basic assumptions used in modeling, frequency calculations for modeling purposes, sample mathematical models, and mass lumping procedures. For features of components not specifically treated by the aforementioned guidance manuals, the analyst should rely on the following means to determine which areas of an equipment or structure shall be considered critical:

(a) Frequency calculations
(b) Previous analyses
(c) Damage history
(d) Shock test information for similar equipment

In relation to the four factors listed above, engineering judgement must be used. For example, under vertical shock loading high stresses would be expected in an equipment's foundation. High stress would also be expected in bolting between an upper and lower housing.

Fixed base natural frequency calculations of individual system components are useful in determining regions which should be explicitly modeled. It is known that relatively low frequency items are likely to undergo relatively large displacements under shock. Therefore, low frequency structural components should be included in the mathematical model.

Previous analyses of similar equipment, damage history, and shock test information for similar equipment may provide useful information concerning critical areas.

3.2 Mathematical Modeling Phase

The mathematical modeling phase consists of constructing a system of masses and structural elements (beams, springs, plates, etc.) to represent the significant dynamic characteristics of the system under consideration. In the case of a reduction gear, for example, the system under consideration will include the reduction gear, its foundation, a portion of the line shafting, connections to the turbines and any other piece of attached equipment which will affect the response of the gear under shock loading. A separate dynamic analysis shall be performed for each principal direction of shock loading (e.g. vertical, athwartship, and fore and aft), and the shock resistance of the item to each direction of loading shall be evaluated separately. For uni-directional response analyses a separate mathematical model is required for each of the three directions of shock input. If a Multi-Directional Response (MDR) analysis is performed, a single mathematical model may suffice for analysis in each of the three directions of input. An MDR analysis is required where the structure or equipment is such that an input motion in a specified direction produces significant responses in other directions. Examples of such structures are:

(a) Flexible structure subject to whipping (e.g. masts)
(b) Structures oriented in directions oblique to the ships axes (e.g. radar arrays)
(c) Structures with large unbalanced masses (e.g. air conditioning plants)

To simplify discussion of the mathematical modeling phase, the following major steps will be considered separately:

(1) Basic modeling assumptions
(2) Frequency calculations
(3) Mass lumping
(4) Mass locations
(5) Designation of structural model
(6) Special modeling criteria

3.2.1 Basic Modeling Assumptions

Basic modeling assumptions must be formulated to permit reduction of a real structure to a simplified linear system of lumped masses and elastic structural elements. For certain major items required by contract specifications to be dynamically analyzed, the SUPSHIP Brooklyn guidance manuals referenced in Section 3.6.1 of this chapter describe typical basic assumptions for specific items.

a. Selection of the Fixed Base -A fundamental assumption necessary in the application of DDAM is the selection of the fixed base. A fixed base acts as a rigid stationary boundary in the direction of shock motion through which the shock motion is transmitted to the mounted equipment or structure. Inherent in the selection process is the determination of important characteristics of the fixed base. The fixed base is assumed to be at the interface of the system foundation and the basic ship structure. Section 3.1.2 describes the character of the fixed base at different shipboard mounting locations. Proper selection of the fixed base for a system, whether hull or deck mounted, also defines the proper choice of shock design values to be applied. It is necessary for the mathematical model to reflect local flexibilities of the interface which can affect the system response. For example, if rocking of the supporting ship structure is a dominant response characteristic for the system, the mathematical model should include this feature of the interface.

3.2.2 Frequency Calculations

As stated in Section 3.1.4, fixed base natural frequency calculations are used to determine those components which may be critical. These components may require a separate mass or masses to properly model them. The cut-off frequency is defined as the frequency of the highest mode of vibration to be considered in the dynamic analysis corresponding to conditions specified in Section 3.5.3. Those components whose frequency (which may be approximated by the fixed base frequency) falls below the cut-off frequency of the system shall be modeled.

3.2.3 Mass Lumping

Having determined critical areas and frequency values, the analyst can now proceed to model the equipment or structure. To aid the analyst in this task, the following guidelines are given:

a. The model should be as simple as possible. The analyst should strive for the simplest model which yields all the information required for a complete analysis of the equipment or structure.

b. High frequency components should be lumped together. The analyst is justified in combining adjacent high frequency (frequencies above the cut-off frequency) components into one mass. This justification is based upon the fact that adjacent high frequency components tend to move as a single rigid mass under shock loading, and so may be analyzed as a unit. Some high frequency components, however, may require separate modeling. This may be the case where it is required to know the relative deflection between two components of the system.

c. Low frequency components shall be represented as separate masses. A critical component whose frequency is below cut-off frequency shall be represented by one or more masses in the mathematical model. Non-critical low frequency components shall be represented by one or more masses if the weight of the component is such that it will significantly influence the shock response of a critical part of the system. To illustrate this situation the analyst is referred to the main reduction gear guidance manual referred to in Section 3.6.1. In this manual it can be seen that in modeling the main reduction gear for vertical and athwartship shock loading, the relatively low frequency line shafting adjacent to the gear is represented. Even though the shafting is not required to be stress analyzed with the reduction gear, its effect on the critical bull gear bearing requires that it be included in the gear model. The number of masses needed to model a component depends on fixed base natural frequency and the distribution of the component mass. For example, if the second mode fixed base frequency of a component is below the system cut-off frequency, then at least two masses are required to adequately model it.

To illustrate this point, assume that the simply supported shaft shown in Figure 3-5 is a part of an equipment which has an estimated cut-off frequency of 200 Hertz. Assume the shaft weight between supports is W = 19,776 lbs (87.97 kN) and that the length between supports is L = 192 inches (4.88 m).

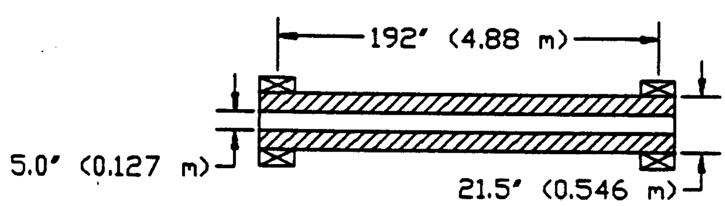
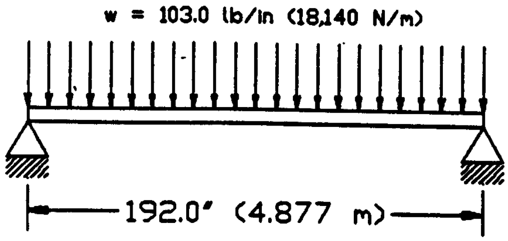


Figure 3-5 Simply Supported Shaft

The shaft shown in Figure 3-5 may be represented schematically as shown in Figure 3-6.



where:

E = 29 x 10^6 psi (200 GPa)
I = 10,485 in^4 (4.35 x 10^{-3} m^4)
g = 386 in/sec^2 (9.81 m/s^2)
w = 103 lbs/in (18,040 N/m)

Figure 3-6 Schematic Representation of Simply Supported Shaft

Using the natural frequency equation (which reflects the consistent mass nature of the beam) for a simply supported beam with a uniformly distributed weight to determine the fixed base frequency of this component:

$$f = B\sqrt{\frac{EIg}{wL^4}} = 28.96\,B\ \text{(Hz)}$$

Where B = 1.571 for the first mode, B = 6.283 for the second mode and B = 14.137 for the third mode frequency, the following results are obtained:

f_1 = 45.49 cps

f_2 = 181.96 cps

f_3 = 409.4 cps

Therefore, the shaft shown above is required to be modeled with two or more masses so that the effect of two significant modes of response on the equipment can be adequately evaluated.

d. Shock tested items shall be included in the model. Small shock testable items that are a part of a larger equipment shall be shock tested rather than being separately analyzed as part of the equipment DDAM. This applies to such items as tachometers, gauges and motors. The NAVSEA policy to test such items avoids the uncertainties involved in analyzing small mechanical components. Whether they are represented by a single mass or lumped into adjacent masses, shock tested items must be included if they are a part of the system under consideration. However, items which have been shock qualified should be stress analyzed only to the extent of determining the adequacy of their hold-down means unless these fasteners have also been previously qualified by shock testing of the items. For example, a turning gear motor which has been shock tested and found acceptable requires analysis to determine the adequacy of its hold-down means but does not require analysis of its internal parts (e.g. armature, brushes, etc.). Analysis should be limited to the structural portions of the equipment under consideration. For completeness, the mathematical model report shall contain information on the status of the MIL-S-901 testing of any components. If testing has been completed, references to the test report and approval documentation shall be provided. If testing is to be done in the future, planned test schedules shall be indicated.

3.2.4 Mass Locations

The concentrated masses having been determined, the analyst must then proceed to properly place them in relation to a fixed origin (the analyst should choose any origin convenient to the system under consideration).

The masses of high frequency components are added and considered to be a single mass located at a node. Center of gravity calculations locate each lumped mass relative to the origin of the coordinate system. This is done by determining the mass center of gravity of each component making up the mass point and locating the component's position along a principal axis relative to the origin. For example, to locate the X-coordinate of a mass point relative to the origin, the following formula is used:

$$\bar{X} = \frac{\sum Wx}{\sum W}$$

where,

$\bar{X}$ = distance between mass point and origin as measured along the X-axis

W = weight of individual component contained in the lumped mass.

x = distance between the origin and the center of gravity of the individual component as measured along the X-axis.

The same procedure is used to determine the Y- and Z-coordinates of a mass point. The overall center of gravity of the model should match the center of gravity of the actual item.

3.2.5 Designation of Structural Model

A structural model (linear, elastic, mathematical description) of an item can be a finite element description of the item or a mass-spring, lumped parameter representation. The structural model describes the item in terms of physical characteristics which when combined with the concentrated masses will produce dynamic characteristics representative of the equipment or system under investigation. All material properties used in generating the mathematical model shall be values at the expected operating temperature of the item.

3.2.6. Special Modeling Criteria

During the modeling, coefficient computation and dynamic computation phases, resilient mounts shall be assumed to be rigid in translation (in the direction of application of shock motion only) unless it can be shown that the mounts will remain linear and elastic during shock excursions. The effects of overturning characteristics of a resiliently mounted system shall be considered in determining the degree and extent to which the mount flexibility will be included in the mathematical model. Regardless of the representation of the mounts in the mathematical model, the actual mount physical characteristics shall be considered during the evaluation phase. It is noted that shock isolation or protection devices shall not be used in foundation systems without approval of NAVSEA.

For equipment with attached external piping which is not separately modeled, the analyst shall include the weight of five feet of this piping (including fluid) as mass when modeling the equipment.

Where an item is modeled as a lumped mass with rigid links, the equipment model should not provide constraint to the support structure.

Where foundations are grounded on deep frames, inner bottom structure, built-in tanks, or similar structure above the shell plating, this local structural flexibility may be included (but is not required) in the mathematical model. Incorporation of this structure in the model may serve to reduce the calculated shock response.

3.3 Coefficient Computation Phase

Having developed a mass-spring or finite element representation (structural model) of the equipment or structure under consideration, the analyst must then determine how this model reacts to a pre-determined shock design value (DDS 072-1). In order to determine this reaction, evaluation of the dynamic equations of motion are required.

$$[M]\{\ddot{X}\} + [C]\{\dot{X}\} + [K]\{X\} = \{P(t)\}$$

Solution of the equations of motion requires the formulation of the associated coefficient matrices. Damping is not considered in the DDAM and therefore the damping coefficient matrix, [C], is assumed to be null. The mass coefficient matrix [M] (called the mass matrix) is the matrix of elements m_{ij} where:

m_{ij} = Force corresponding to coordinate i due to a unit acceleration at coordinate j only.

The stiffness coefficient matrix, [K] is the matrix of elements k_{ij} where:

k_{ij} = Force corresponding to coordinate i due to a unit displacement of coordinate j (and no other coordinate displacements are permitted)

X, $\dot{X}$ and $\ddot{X}$ = Displacement, velocity and acceleration respectively of a nodal degree of freedom.

$P(t)$ = Externally applied forcing function

The inverse relation of the stiffness matrix is called the flexibility matrix [Δ] and is a matrix of elements δ_{ij} where:

δ_{ij} = deflection of coordinate i due to a unit load applied to coordinate j.

Methods of determining these matrices can be found in standard structural dynamics textbooks.

3.3.1 Mass Matrices

The mass coefficient matrix can be determined by either the lumped mass or the consistent mass formulation. In the lumped mass method the mass properties of a component or model element are typically associated only with the translational degrees of freedom at the nodes of that element. However, this does not preclude the use of rotational inertia where desired. The simplest procedure for defining the lumped mass properties of any structure is to assume that the nearby distributed mass is concentrated at the nodes where translational displacements are defined. The usual procedure for defining the magnitude of mass to be located at each node is to assume that the structure is divided into regions or elements with nodes serving as connection points. The mass of each element is assumed to be concentrated as point masses at its node points. The distribution of the element mass to the node points is determined by geometric relations. The total mass concentrated at any node point is the sum of all the nodal contributions of the elements attached to that node. For the lumped parameter system the mass matrix has a diagonal form.

A consistent mass matrix is defined using a consistent shape function for both the potential and kinetic energies. Unlike the lumped mass matrix, the consistent mass matrix includes off-diagonal coefficients that couple related degrees of freedom.

The dynamic analysis of a consistent mass system generally requires considerably more computational effort than a lumped mass system does, for the following reasons:

(1) The lumped mass matrix is diagonal while the consistent mass matrix has many off-diagonal terms (leading to what is called mass coupling).

(2) Unmassed degrees of freedom can be eliminated from a lumped mass analysis by static condensation, whereas all rotational and translational degrees of freedom must be included in a consistent mass analysis.

As the lumped mass model is refined, the influence of the missing off-diagonal terms will diminish and the calculated response will converge to that of the consistent mass model.

3.4 Dynamic Computation Phase

The dynamic computation phase usually involves placing the pertinent data developed in the previous phases into a suitable computer program in order to obtain the modal characteristics present in the system. Many computer programs which perform the computations associated with the Dynamic Design Analysis Method are available or are developed external to commercially available general purpose structures programs. A sample computation for extracting characteristic values (frequencies and mode shapes) is shown for a three degree of freedom system in Appendix A.

3.4.1 Modal Analysis

The dynamic analysis of a mathematical model representation of a system or structure initially involves the definition of the modal (frequency) equations of motion for that system. The undamped free-vibration, modal equations of motion for a multi-degree of freedom system in matrix notation become:

$$-\omega^2 [M]\{\Phi\}_a + [K]\{\Phi\}_a = \{0\}$$

Solution of the equations (the eigenvalue problem) produces natural frequencies ω_a and mode shapes $\{\Phi\}_a$.

N	=	Number of degrees of freedom within the mathematical model
[M]	=	Mass matrix of the system
$\{\Phi\}_a$	=	Mode Shape for the a^{th} mode

For the purpose of the following discussions an influence coefficient vector {r} is defined to represent displacements of all degrees of freedom resulting from a unit support translation. The influence coefficient vector {r} has the following characteristics:

(a) For a uni-directional response analysis, {r} is a column of ones.

(b) For a multi-directional response analysis in which the orientation of ship input motion coincides with the orthogonal axis of the model, {r} is a column of ones and zeros.

(c) For a multi-directional response analysis in which the orientation of the input motion is arbitrary with respect to an orthogonal axis of the model, {r} is a column of direction cosines and zeros.

Given the above characteristics (i.e. N, [M], ω_a and $\{\Phi\}_a$) the following quantities are determined for each mode and each direction of motion:

$$\bar{M}_a = \{\Phi\}_a^T [M] \{\Phi\}_a$$

Generalized mass of the a^{th} mode.

$$\bar{M}_a = \sum_{i=1}^{N} \Phi_{ia}^2 M_i$$

Where Φ_{ia} is the a^{th} mode shape for a lumped mass system represented by a diagonal mass matrix

$$P_a = \frac{\{\Phi\}_a^T [M] \{r\}}{\bar{M}_a}$$

Participation factor for the a^{th} mode

$$P_a = \sum_{i=1}^{N} \frac{\Phi_{ia} M_i r_i}{\bar{M}_a}$$

Participation factor for a lumped mass system represented by a diagonal mass matrix

$M_a = P_a^2 \bar{M}_a$ Modal effective mass for the a^{th} mode

$\{F\}_a = D_a P_a [M] \{\Phi\}_a$ Nodal forces for the a^{th} mode

$\{A\}_a = D_a P_a \{\Phi\}_a$ Nodal accelerations for the a^{th} mode

D_a is the design acceleration of the a^{th} mode and is equal to the lesser of $V\omega_a$ or Ag as obtained from DDS 072-1 (See Section 3.5.2)

3.4.2 Dynamic Reduction Techniques

The number of dynamic degrees of freedoms used in DDAM mathematical models has increased dramatically over the years since DDAM was first introduced. As a consequence of this increase in model complexity, reliance on matrix reduction techniques has also increased. Matrix reduction techniques allow the use of a large number of static degrees of freedom while reducing the number of dynamic degrees of freedom to a fraction of the static.

There is an inherent risk in using dynamic reduction techniques as a means of simplifying complicated models. Reduction techniques attempt to convert extremely detailed models into smaller models for computational efficiency. However, these reduced models are difficult to review in detail and they may not satisfy all the requirements of Chapter 3. It is preferable to rely on engineering judgement rather than an automatic selection process available in various dynamic reduction techniques as a means of creating simplified structural models. Certain criteria must be met where dynamic reduction is used. Consider the following procedure as a minimum verification of the adequacy of any reduction technique considered within the DDAM:

Assume that the original dynamic system, with N degrees of freedom, has mass matrix [M] and stiffness matrix [K]. By any reduction method this system is reduced to a system with mass matrix [M*] and stiffness matrix [K*] with N* master degrees of freedom. This reduced dynamic system is then solved for:

NOM = Reduced number of modes

$\{\Phi^*\}$ = Mode shapes of reduced set

ω^* = Natural frequencies of the reduced set

Transform back to the original system and obtain each mode shape $\{\Phi\}$ in the original degrees of freedom.

Determine whether these mode shapes, obtained by the back transformation process, are orthogonal with respect to the original mass and stiffness matrices.

$$[\Phi]^T [M] [\Phi] = [\bar{M}]$$

$$[\Phi]^T [K] [\Phi] = [\bar{K}]$$

$$[\bar{K}] [\bar{M}]^{-1} = [\omega^2]$$

where $[\Phi]$ = A mode shape matrix with the number of columns equal to the number of degrees of freedom and the number of rows equal to the number of modes

$[\bar{M}]$ = Generalized mass matrix

$[\bar{K}]$ = Generalized stiffness matrix

$[\omega^2]$ = A diagonal matrix with the diagonal equal to the squared natural frequencies of the original system

As a check, $[\bar{K}]$ and $[\bar{M}]$ should be diagonal matrices and hence the mode shapes are orthogonal with respect to the mass and stiffness matrix. ω^* should be the same as ω and the modal masses should add up to the total modal effective weight of the system. For lumped parameter systems:

$$\sum_{a=1}^{NOM} \frac{\left(\sum_{i=1}^{N} \Phi_{ia} M_i\right)^2}{\sum_{i=1}^{N} \Phi_{ia}^2 M_i} = \text{Total Modal Effective Weight}$$

At least three general approaches have been used effectively to reduce the number of dynamic degrees of freedom:

(1) Kinematic Condensation (Guyan Reduction)
(2) Generalized Dynamic Reduction (Rayleigh-Ritz)
(3) Component Mode Synthesis (Sub-structuring)

Kinematic Condensation is based on the assumption that inertia forces are associated with only certain selected degrees of freedom of the original idealization. The remaining degrees of freedom are not explicitly involved in the dynamic analysis and can be condensed from the dynamic matrix. In the Generalized Dynamic Reduction approach, the number of dynamic degrees of freedom are limited by assuming that the displacements of the structure are combined in selected patterns, the amplitudes of which become generalized coordinates of the dynamic analysis. Component Mode Synthesis reduces the problem by dividing the solution into a series of substructures, solving the reduced substructure and combining the substructure analyses into a single reduced analysis.

When considering the number of master degrees of freedom, the following should be used as guidance:

(a) The model should be kept as simple as possible.

(b) High frequency components should be considered as acting together.

(c) Low frequency critical components shall be represented as separate degrees of freedom.

(d) The number of master degrees of freedom selected should be at least two to three times the number of modes of interest.

(e) Include master degrees of freedom at locations having relatively large mass and/or rotary inertia.

(f) Master degrees of freedom should not be defined where the structure has an insignificant mass.

(g) Retain a uniform spatial distribution, such that the center of gravity of the master degrees of freedom closely represents that of the system modeled.

(h) Retain critical items as master degrees of freedom.

3.5 Evaluation Phase

The evaluation phase of DDAM is essentially one of determining the stresses and deflections in the equipment, structure and/or foundation and comparing them to specified failure criteria established by material and operational considerations. Having obtained the deflections of and forces on the masses of the mathematical model, the analyst may then proceed with the analysis of the equipment. The analysis at this point becomes a static analysis, i.e. within each mode the system is in equilibrium. Presented below are requirements for:

(1) Modal assessment
(2) Shock Design Values to apply
(3) The number of modes to use
(4) Combining stresses within each mode
(5) Summing stresses across the modes
(6) Combining operating and shock stresses
(7) Response assessment

3.5.1 Modal Assessment

The mathematical model used to define the equipment, system or structure is the fundamental tool by which satisfactory shock performance can be demonstrated by analysis. A modal analysis of the system generates dynamic response characteristics (frequencies and mode shapes). The results of this analysis should be examined for credibility before proceeding with subsequent steps in the design process. The results of the analysis should demonstrate that the basic requirements of DDAM are satisfied and that the model does not produce conditions that exceed the limitations of DDAM. The following are potential conditions wherein the requirements or limitations of DDAM may be exceeded:

(a) very low frequency systems (less than 5 Hz)
(b) closely spaced modes

The analyst should not continue with the analysis until the conditions which do not agree with the basic DDAM assumptions are resolved or specifically approved by the cognizant Navy acceptance authority.

One of the critical areas where the results of an analysis could exceed the limitations of the basic DDAM assumptions is the existence of closely spaced modes. Closely spaced modes are defined as two modes whose frequencies are within 10% of the common mean frequency. Closely spaced modes can become a problem when their modal effective masses are significant and are approximately of the same order of magnitude. Closely spaced modes will frequently occur in a dynamic analysis without resulting in any notable amplification of the component responses. These cases are generally associated with modes which have relatively low modal effective mass.

When closely spaced modes involve modes with large modal masses, they can produce significant responses which indicate a shock hazard to the equipment. Therefore, some preliminary assessment must be conducted to determine whether closely spaced modes that have been identified will have any significant effect on the design loading.

The following outline describes the basic approach for the treatment of closely spaced modes:

(1) Prepare a bar graph of modal effective mass versus modal frequency. This graph provides an overview of the system dynamic response and permits early identification of closely spaced modes.

(2) Identify closely spaced modes which are defined as modes which are separated by less than 10% of the common mean frequency. Potentially hazardous closely spaced modes are usually two or more modes close in frequency, each with significant modal mass of relatively significant magnitude. Selection criteria of Section 3.5.3 can be used to identify modes that are likely to be significant.

(3) Compare the mode shape (shape function times the participation factor) of the closely spaced modes suspected of being potentially damaging. The comparison should be conducted for each node point. An indication of a potentially hazardous closely spaced mode condition exists where the maximum response of similar magnitude and opposite sign occurs for the two closely spaced modes. This is indicative of a split modes phenomenon. Under these conditions it is concluded that either the model is incorrect or the design of a local component will result in an apparent resonance and should be detuned. Another indication of potentially hazardous closely spaced modes condition exists when the modal masses of each of the apparent closely spaced modes is contained in distinctly different sets of degrees of freedom. This is indicative of an uncombined mode phenomenon. Uncombined modes may occur for either of two reasons: each portion may have been modeled with independently fixed bases and are too lightly coupled, or, one of the portions may have been modeled so as to become a split mode. Under these conditions it is concluded that the fixed base may be inappropriately selected. An acceptable change would be to extend the boundries of the mathematical model so that it includes more of the supporting ship structure.

(4) Show the extent of detuning necessary to eliminate the split mode condition. Similarly, where uncombined modes exist, the analysis should show what quasi fixed base is needed to eliminate this condition.

(5) Determine if damaging effects of closely spaced modes cannot be eliminated by remodeling or redesigning (detuning). If this cannot be done, the analyst should request Navy approval of application of an alternate techniques such as the methods described in the remainder of this section. Section 3.5.7 discusses the ASM and the CSM techniques used to evaluate closely spaced modes. Sections 3.5.7 and 7.2.2.8 discuss the ASM analysis submittal and approval requirements with regard to supplementary information to be supplied in the corrective action report.

3.5.2 Shock Design Values to Apply

As noted in Chapter 2, the shock design values to apply when performing a DDAM analysis are contained in Design Data Sheet DDS 072-1 (Confidential). The shock design values are given in the form of frequency-dependent and modal weight-dependent equations of pseudo-velocity or acceleration.

The shock design values were derived from data recorded in full ship shock tests. The data were first converted into conventional response spectra and discrete points were extracted from the spectra at the known fixed-base natural frequencies of equipment (for which the modal masses had been calculated) mounted aboard the ships. In this way, a series of tests were used to generate the design shock spectra contained in DDS 072-1. At the fixed based natural frequencies, the various items of equipment tend to act as vibration absorbers and suppress to some degree the motions of the basic ship structure. It is these fixed-base natural frequencies which give rise to the major equipment and foundation responses to shock. Because the test shock spectra tend to show minima at these fixed-base frequencies rather than peaks, the phrase "spectrum dip effect" is often used to describe the derivation of the design shock spectra. For more information on the derivations of the design values, see the reports cited in Section 3.6.2.

Although the DDAM shock design values are to be applied in each of the three translational directions (vertical, athwartship, and fore/aft) separately, responses may be calculated in all three directions (multi-directional response analysis). For cases in which the equipment or foundation's principal axes do not coincide, even approximately, with the directions of shock design values defined in DDS 072-1, special combinations of the shock design values may be appropriate. See Appendix D for discussion of oblique shock design values.

While the DDS 072-1 shock design values have been derived from test data on steel hulls, the inputs are also considered to be the best available data for analyses of equipment on wood and glass reinforced plastic (GRP) hulls. However, special design criteria must be applied in these cases. While it is usually not considered necessary to check the stresses in basic ship structure for steel hulls, such calculations are necessary for wood and GRP hulls. Special attention must be paid to the strength of interface connections, such as bolted connections between steel foundations and non-metallic hull structure. The wood frames must be checked for continuity to ensure that local failure of the ship structure under the loads transmitted by the bolts will not occur.

3.5.3 Number of Modes to Use

The number of modes to be calculated prior to the selection process shall be sufficient to satisfy the modal weight requirement listed below and the additional modes likely to