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NAVSEA DESIGN PRACTICES AND CRITERIA MANUAL

SHOCK DESIGN CRITERIA
FOR
SURFACE SHIPS

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FOREWORD

The purpose of Design Practices and Criteria (DPC) Manuals is to provide ship design practices and criteria to personnel involved with the design, conversion, or modernization of U.S. Navy ships. This Manual is therefore part of a library of DPC Manuals. A listing of all DPC Manuals and other ship design documents can be found in the DPC Index (NAVSEA T9070-AE-DPC-010/001-1).

This document is intended to provide design guidance for surface ship equipment and foundations subject to underwater explosions. The design process defined in this document, used for contractual shock qualification, involves the use of confidential data, T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1), which must be manipulated within a proper secure environment. This document is organized as follows:

Chapter 1, Introduction

Chapter 2, Applicable Documents

Chapter 3, Dynamic Design Analysis Method

Chapter 4, Foundation Shock Design

Chapter 5, DDAM of Grade B Items

Chapter 6, Allowable Stress Criteria

Chapter 7, Dynamic Shock Analysis Review and Approval Procedures

Appendix A, Sample Computation of Normal Modes of a Structure

Appendix B, Finite Element Method for DDAM Analysis

Appendix C, Transient Analysis Methods

Appendix D, Oblique Directional Shock Inputs

Appendix E, Sample Finite Element DDAM Analysis – Format and Content

Appendix F, Example Application of the Allowable Stress Criteria

Appendix G, Bolt and Bolt Joint Design

Appendix H, List of Abbreviations, Acronyms, Symbols, and Definitions

All errors, omissions, discrepancies, and suggestions for improvement to NAVSEA and SPAWAR technical manuals (TMs) shall be submitted as a Technical Manual Deficiency/Evaluation Report (TMDER) via the NAVSEA/SPAWAR TMDER form, NAVSEA 4160/1, included at the back of this TM.

L. C. SELBY
NAVSEA CHENG

CHAPTER 1
INTRODUCTION

1.1 PURPOSE.

The primary purposes of this manual 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 manual 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 manual does not provide information sufficient to permit full and efficient satisfaction of all specified dynamic shock analysis requirements, the cognizant contracting officer shall be contacted for additional information.

The requirements indicated by this manual are subject to modification by applicable specifications. Users of this manual shall carefully review applicable specifications to determine whether any of the provisions of this manual have been modified. The contents of this manual are founded upon dynamic analysis procedures originally developed by the Naval Research Laboratory (NRL), Washington, D.C.

CHAPTER 2
APPLICABLE DOCUMENTS

2.1 INTRODUCTION.

The documents listed in this chapter are specified in chapters 1-7 of this publication. This chapter does not include documents cited in other sections of this publication. While every effort has been made to ensure the completeness of this list, document users are cautioned that they must meet all specified requirements documents cited in chapters 1-7 of this publication, whether or not they are listed.

2.1.1 Government Publications.

DEFENSE SPECIFICATIONS

MIL-DTL-1222 - Studs, Bolts, Screws and Nuts for Applications where a High Degree of Reliability is Required; General Specification for

(Copies of this document are available online at http://quicksearch.dla.mil.)

DEPARTMENT OF DEFENSE ISSUANCES

DoDI 5000.61 - DoD Modeling and Simulation (M&S) Verification, Validation, and Accreditation (VV&A)

(Copies of this document are available online at www.dtic.mil/whs/directives/.)

GUIDANCE MANUALS

Report No. SUPSHIP 280-1 - Mathematical Model and Dynamic Shock Analysis Guide for Main Propulsion Shafting

Report No SUPSHIP 280-2 - Mathematical Model and Dynamic Shock Analysis Guide for Rudders, Rudder Stock and Bearings

Report No. SUPSHIP 280-3 - Mathematical Model and Dynamic Shock Analysis Guide for Main Reduction Gear

Report No. SUPSHIP 280-6 - Mathematical Model and Dynamic Shock Analysis Guide for Masts

(Copies of these documents are available from Naval Surface Warfare Center Philadelphia Division (NSWCPD) Code 333, WPN STA Earle, Bldg C38, 201 State Route 34 South, Colts Neck, NJ 07722.)

NAVSEA TECHNICAL PUBLICATIONS

T9070-AJ-DPC-010/(C) 072-1 - Shock Design Values

T9070-AN-DPC-040/100-4 - Strength of Structural Members

(T9070-AJ-DPC-010/(C) 072-1 is a classified document with controlled distribution. Requests for this document, supported by a verifiable need-to-know, shall be submitted to commandstandards@navy.mil.)

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2.2 ORDER OF PRECEDENCE.

In the event of a conflict between the text of this document and the references cited herein, the text of this document takes precedence. Nothing in this document, however, supersedes applicable laws and regulations unless a specific exemption has been obtained.

CHAPTER 3
DYNAMIC DESIGN ANALYSIS METHOD

3.1 INTRODUCTION.

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 motions 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.

3.1.1 Shock Hardening Design Process Figures. 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 manual. 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 manual.

3.1.2 Five-Phase Approach. 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:

a. Problem formulation phase

b. Mathematical modeling phase

c. Coefficient computation phase

d. Dynamic computation phase

e. 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 the design of foundations and Grade B items are described in Chapters 4 and 5 of this manual, respectively. When the DDAM was first implemented in the 1960s, 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 manual.

3.1.3 Limitations of the DDAM. The limitations of the DDAM must be clearly recognized by the users of the methods 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 persists for only a few cycles of vibratory motion. For very low frequency systems (less than 5 Hz) 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 shall be used to overcome the limitation. Similarly, analyses of foundations for very lightweight 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.

3.1.4 Limitations of the Structure Being Modeled. There are limitations to the extent of the structure to be modeled using the DDAM. For instance, the modeling of an entire subdivision of a ship or an entire deckhouse may not be appropriate for the DDAM. 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 Technical Authority to be inapplicable to the analysis of a particularly critical item of equipment. In order to support the limited low frequency data as well as the lack of whipping data during the bubble regime in the T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1) shock spectrum, a minimum shock design value of 6 g is imposed on all modes.

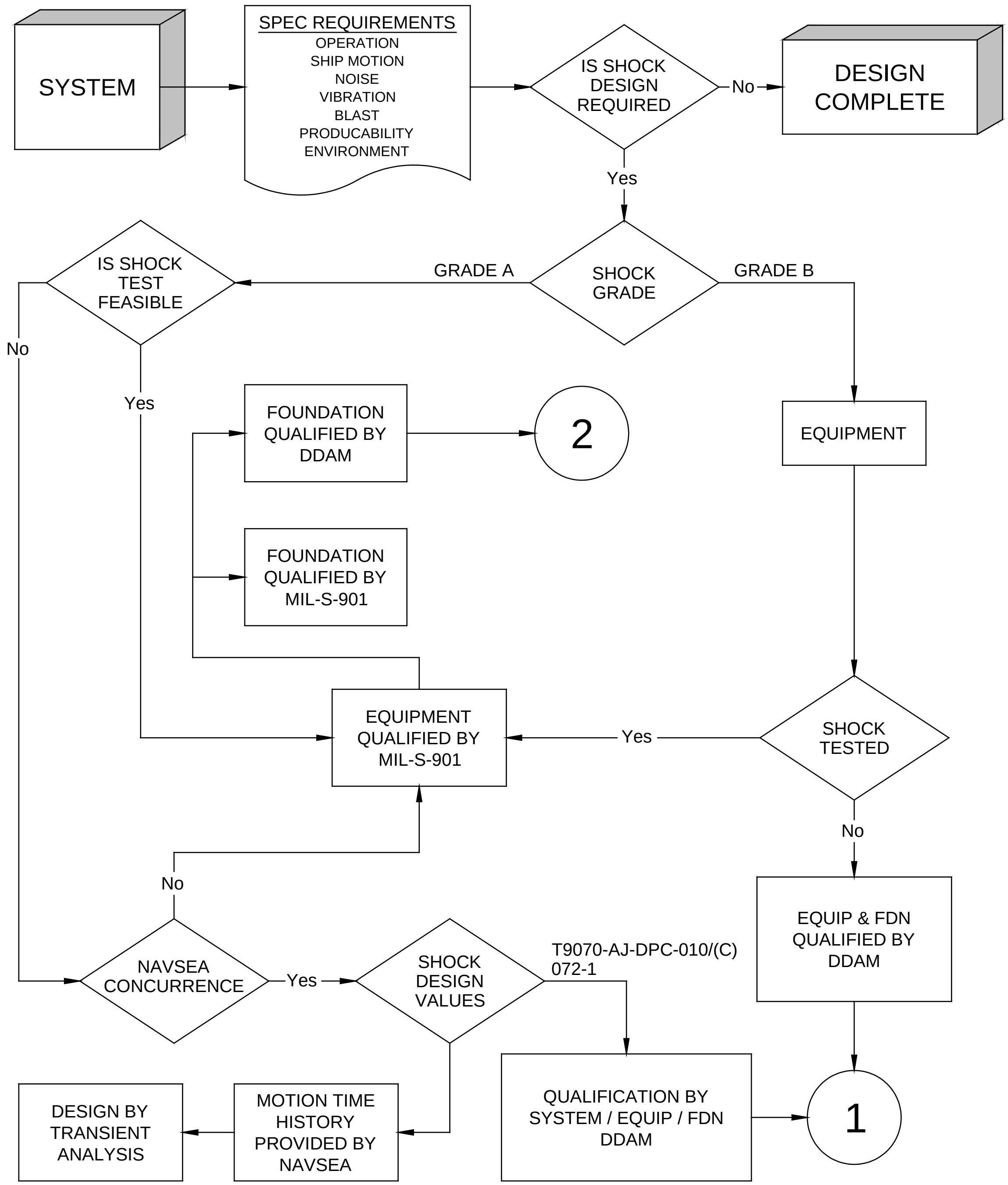


Figure 3-1. Shock Qualification Process – Overview.

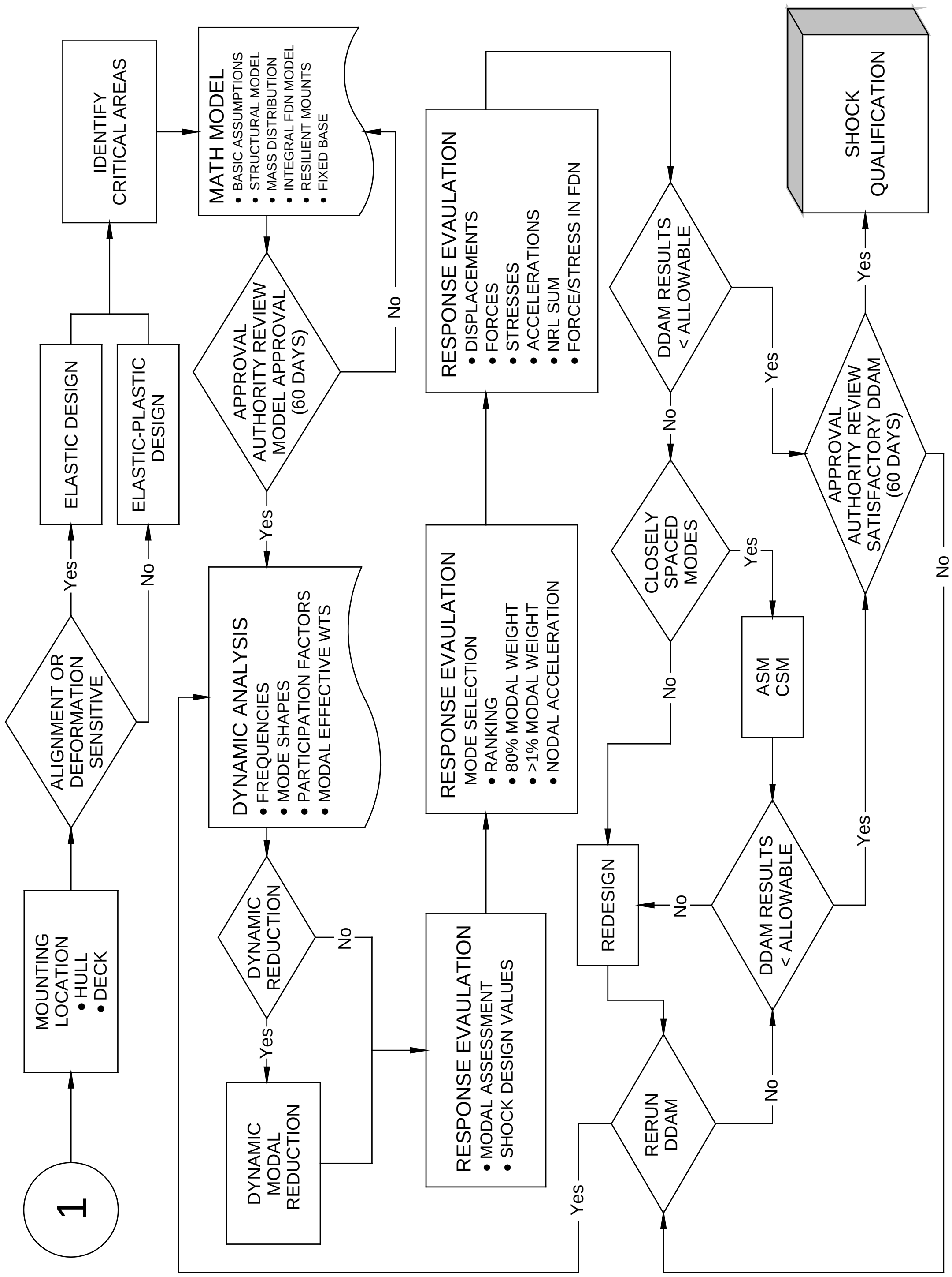


Figure 3-2. Equipment Shock Qualification by the Multi-DOF DDAM (see Chapter 3).

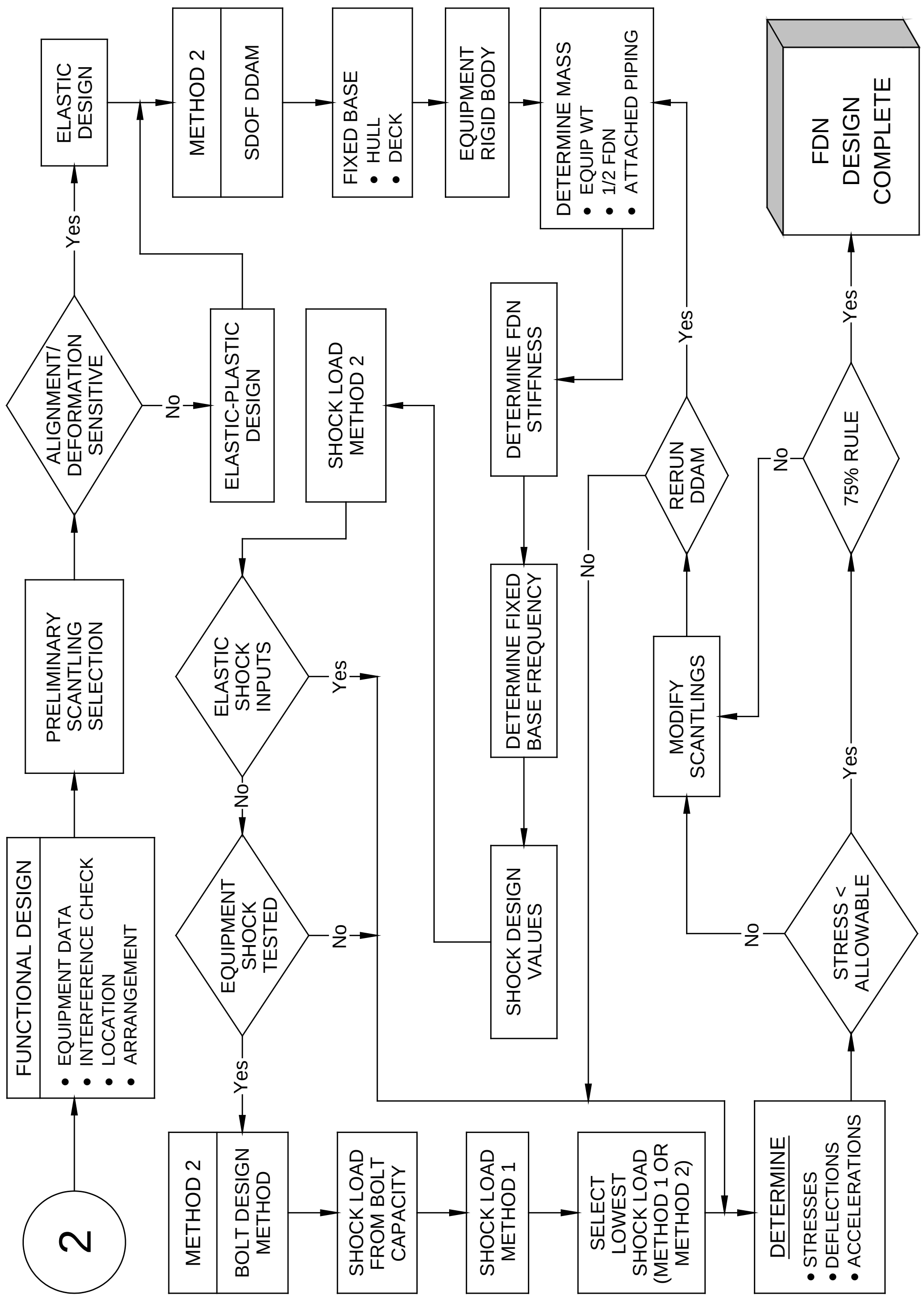


Figure 3-3. Foundation Shock Design by SDOF DDAM (see Chapters 4 and 5).

3.1.4.1 Assessing Structural Elements of Equipment. In cases where the DDAM is used to assess the structural elements of equipment, the DDAM must clearly define the subsidiary components and subassemblies which are to be evaluated via other methods (e.g., testing in accordance with MIL-DTL-901), and these items must be properly accounted for in the mathematical model representation of the equipment in the DDAM.

3.1.4.2 Limitations in Evaluating Various Hazards. The DDAM's primary function is to evaluate the mechanical failure modes associated with the "coming adrift" type hazards. It cannot evaluate electrical short or electrical fire hazards, and may not be able to evaluate the release of hazardous material hazards.

3.2 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:

3.2.1 Shock Grades. The shock grades (A and B) are defined by the contract specifications in accordance with MIL-DTL-901. 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 Appendix H, are applicable to the items which are required for the performance or to the direct and vital support of mission-essential functions aboard shock hardened ships. The following are often specified as mission-essential functions:

a. Ship control and propulsion
b. Command and control
c. Navigation
d. Communications
e. Surface, air, and underwater surveillance
f. Countermeasures
g. Launching, retrieving, fueling, defueling, rearming, and handling of aircraft and small surface craft
h. Essential checkout and maintenance of aircraft and ordnance
i. Fire control, firing or launching, and guidance of missiles and other weapons
j. Stowage, handling, and reloading of weapons
k. Replenishment-at-sea (stowed configurations)
l. Mine-hunting and sweeping
m. Transporting and landing troops, and combat payload (assault ships)
n. Casualty and damage control
o. Collective protection system

Grade B shock criteria, as defined in Appendix H, 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.

3.2.2 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 T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1). Figure 3-4 describes various mounting locations with respect to the level of shock design input that shall be applied. Proper identification of the mounting location (see 4.4) is important as this will determine the proper shock design value to use for dynamic analysis (see 3.2.3 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 discussions "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 structure 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 the DDAM assumptions.

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

a. "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; for carriers, the basic hull structure, including frames, structural bulkheads below the waterline (or limiting draft, as defined by the ship specification), and shell plating above the waterline.

b. "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); for carriers, decks, non-structural bulkheads, or structural bulkheads which are above the waterline (or limiting draft, as defined by the ship specification).

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

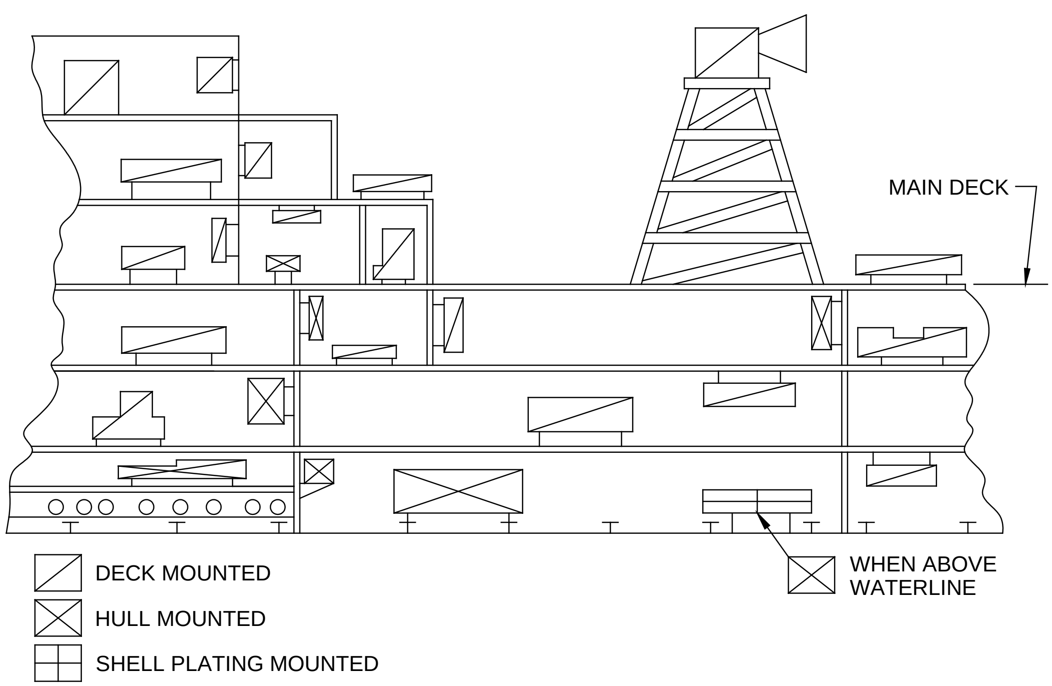


Figure 3-4. Mounting Locations for Surface Ships.

3.2.2.2 Item Mounted to Two Different Parts of the Ship. 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.

3.2.2.3 Evaluating Specific Characteristics of Deck Structure. 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.

3.2.2.4 Items Not Mounted Directly on Deck or Basic Hull Structure. The following considerations shall apply for items not mounted directly on a ship's deck or on the basic hull structure:

3.2.2.4.1 Shock Design Values for Items Mounted on Structural Bulkheads. As indicated by T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1), hull mounted shock design values are to be used in the design of foundation 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:

a. Main subdivision bulkheads.

b. Main longitudinal bulkheads.

c. Bulkheads that replace stanchions, web frames, or any other load-carrying members.

d. Bulkheads located or constructed such that they must be considered capable of transmitting shock loads, regardless of their function. These would include any bulkheads below the bulkhead deck which is thicker than ⅛ inch (3.175 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.

3.2.2.4.2 Criteria for Lightweight Items Mounted on Machinery Space Upper Levels. In analyzing lightweight items such as heating ventilation and air conditioning (HVAC) ducts 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 4.4.2 for applicable criteria.

3.2.3 Shock Design Values. Elastic and elastic-plastic shock design values are contained in T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1). Criteria for selections of elastic versus elastic-plastic shock design values are as follows:

3.2.3.1 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 can be 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 this 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 brakes	Main thrust bearing
Main controllable pitch (CP) servo pump	Gyroscopic compass
Radar antenna	Radio antenna

Missile directors	Gun directors
Steering gear (ram unit)	Steering rudder system
Ammunition hoist	Elevators and elevator machinery
Sonar transducers	Catapult machinery
Arresting gear	Missile launchers
Guns	Torpedo tubes
CP propeller	

3.2.3.2 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.

3.2.3.3 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.

3.2.3.4 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 bases of elastic shock design values for purposes of analysis of bolting, dowels, and similar hold-down or locating devices (such as clamps, brackets, straps, etc.), as well as local pull through and tearout membrane forces in way of hold-down/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 arranged such that they tend to resist foundation or subbase deformation. Hold-down or locating devices which will not see increased loading as a result of foundation or subbase deformation shall be designed to suit the same criteria as the other structural elements of the equipment in question.

3.2.3.5 Minimum Acceleration Value. The minimum modal acceleration value to be used in a DDAM analysis is 6 g.

3.2.4 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 analysts 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 guidance manuals (see 2.1.1).

3.3 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. Standard finite element modeling techniques must be used in development of the mathematical model. Non-standard finite element modeling techniques shall be reviewed with the Technical Authority prior to execution of the computational phase of the DDAM. 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)

3.3.1 Major Steps. To simplify discussions of the mathematical model phase, the following major steps will be considered separately:

a. Basic modeling assumptions (see 3.3.2)

b. Frequency calculations (see 3.3.3)

c. Mass lumping (see 3.3.4)

d. Mass locations (see 3.3.5)

e. Designation of structural model (see 3.3.6)

f. Special modeling criteria (see 3.3.7)

3.3.2 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 specification to be dynamically analyzed, the SUPSHIP guidance manuals (see 2.1.1) may be used as guidance for basic assumptions for specific items; however, calculation methods shall still be as specified in this document. 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. The character of the fixed base at different shipboard mounting locations is described in 3.2.2. 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 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.

3.3.3 Frequency Calculations. As stated in 3.2.4, 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 3.6.3. 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.

3.3.4 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 (frequency 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 deflections 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 (see 2.1.1). 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.

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

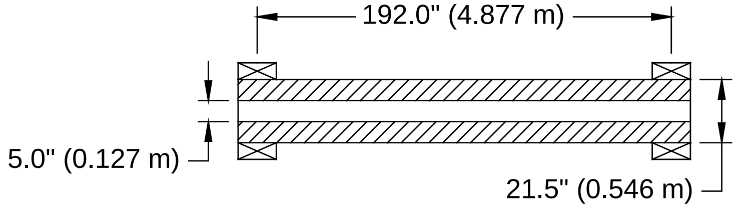


Figure 3-5. Simply Supported Shaft.

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

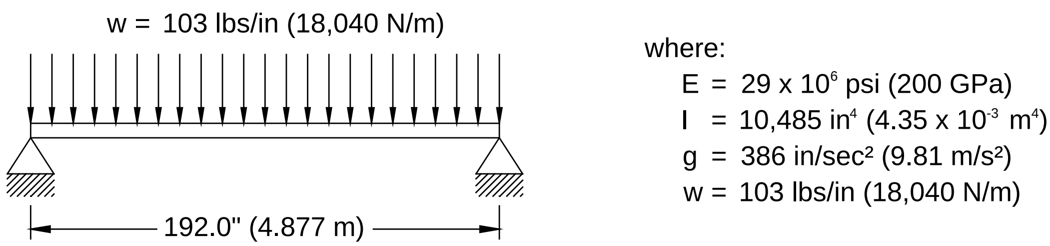


Figure 3-6. Schematic Representation of Simply Supported Shaft.

(3) 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:

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

(4) 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 Hz

f_2 = 181.96 Hz

f_3 = 409.4 Hz

(5) 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-DTL-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.

3.3.5 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:

$$\overline{X} = \frac{\Sigma Wx}{\Sigma W}$$

Where:

$\overline{X}$ = 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.

3.3.6 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.

3.3.7 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.

a. 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.

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

c. 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.

3.4 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 (T9070-AJ-DPC-010/(C) 072-1 [formerly 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 equation of motion requires the formulation of the associated coefficient matrices. Damping is not considered in 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} = Deflections 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.

3.4.1 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 the 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:

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

b. 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.

3.5 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 DDAM are available or are developed external to commercially available general purpose structures programs. A sample computation for extracting characteristic values (frequency and mode shapes) is shown for a three degree of freedom system in Appendix A.

3.5.1 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 influences coefficient vector {r} is defined to represent displacement 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:

$\overline{M}_a = \{\Phi\}_a^T [M]\{\Phi\}_a$ Generalized mass of the a[th] mode

$\overline{M}_a = \sum_{i=1}^{N} \Phi_{ia}^2 M_i$ Where Φ_{ia} is the a[th] mode shape for a lumped mass system represented by a diagonal mass matrix

$P_a = \dfrac{\{\Phi\}_a^T [M]\{r\}}{\overline{M}_a}$ Participation factor for the a[th] mode

$P_a = \sum \dfrac{\Phi_{ia} M_i r_i}{\overline{M}_a}$ Participation factor for a lumped mass system represented by a diagonal mass matrix

$M_a = P_a^2 \overline{M}_a$ 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 acceleration 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 T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1) (see 3.6.2) but in no case shall the design acceleration be less than 6 g.

3.5.2 <u>Dynamic Reduction Techniques</u>. 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.

3.5.2.1 <u>Verification of Adequacy</u>. 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

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

ω^a = 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.

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

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

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

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

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

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

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

As a check, $[\overline{K}]$ and $[\overline{M}]$ 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:

$$\sum_{a=1}^{NOM} \frac{\left(\sum_{i=1}^{N} \Phi_{ia} M_i\right)^2}{\sum_{i-1}^{N} \Phi_{ia}^2 M_i} = \text{Total modal effective weight}$$

3.5.2.2 Reducing Number of Dynamic Degrees of Freedom. At least three general approaches have been used effectively to reduce the number of dynamic degrees of freedom:

a. Kinematic condensation (Guyan reduction): 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.

b. Generalized dynamic reduction (Rayleigh-Ritz): 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.

c. Component mode synthesis (sub-structuring): 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.

3.5.2.3 Guidance for the Number of Master Degrees of Freedom. 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.

3.6 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 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:

a. Modal assessment

b. Shock design values to apply

c. The number of modes to use

d. Combining stresses within each mode

e. Summing stresses across the modes

f. Combining operating and shock stresses

g. Response assessment

3.6.1 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 basic DDAM assumptions are resolved or specifically approved by the Technical Authority.

3.6.1.1 Closely Spaced Modes. One of the critical areas where the results of an analysis could exceed the limitations of the basic DDAM assumptions is the existence of out of phase closely spaced modes. Closely spaced modes are defined as two modes whose frequencies are within 10 percent 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.

3.6.1.2 Treatment of Closely Spaced Modes. The following outline describes the basic approach for the treatment of closely spaced modes:

a. 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.

b. Identify closely spaced modes which are defined as modes which are separated by less than 10 percent 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 3.6.3 can be used to identify modes that are likely to be significant.

c. 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 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 boundaries of the mathematical model so that it includes more of the supporting ship structure.

d. 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.

e. Determine if damaging effects of closely spaced modes cannot be eliminated by remodeling or redesigning (detuning). If this cannot be done, the analyst shall request Navy approval of application of alternate technique such as the methods described in the remainder of this section. The Algebraic Summation Method (ASM) and the Closely Spaced Modes (CSM) techniques used to evaluate closely spaced modes are discussed in 3.6.7. The ASM analysis submittal and approval requirements with regard to supplementary information to be supplied in the corrective action report are discussed in 3.6.7 and 7.2.2.h.

3.6.2 Shock Design Values to Apply. As noted in Appendix H, the shock design values to apply when performing a DDAM analysis are contained in T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1). The shock design values are given in the form of frequency-dependent and modal weight-dependent equations of pseudo-velocity or acceleration. The minimum shock design value to be used in any mode shall be 6 g.

3.6.2.1 <u>Derivation</u>. 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 T9070-AJ-DPC-010/(C) 072-1 (formerly 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.

3.6.2.2 <u>Multi-Directional Response Analysis</u>. 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 T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1), special combinations of the shock design values may be appropriate. See Appendix D for discussion of oblique shock design values.

3.6.2.3 <u>Special Design Criteria</u>. While the T9070-AJ-DPC-010/(C) 072-1 (formerly 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 non-ferrous, 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 foundation 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.

3.6.3 <u>Number of Modes to Use</u>. 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 contribute to the localized high responses. See Figure 3-7 for an overview description of the mode selection process.

3.6.3.1 <u>Set Cut-Off Frequency</u>. A cut-off frequency may be selected in the mode calculation phase of the analysis which is sufficiently high to guarantee the selection requirements are complete. This cut-off frequency is to be consistent with the frequency of the system, and the level of refinement of the mathematical model used to represent it. Nominally, 250 Hz may be taken as an upper bound on the frequencies of interest. Frequencies beyond this level are, for most equipment items aboard ship, of lesser importance in a shock environment in which the ship structure filters the input motions. Alternatively, a number of modes may be selected in the mode calculation phase of the analysis which is sufficiently high to guarantee the selection requirements are satisfied. Iterations may be required if the number of modes to be extracted is specified too low to guarantee compliance with the selection requirements.

3.6.3.2 <u>Sort Modes by Modal Effective Weight</u>. The calculated modes shall be sorted by modal effective weight, in descending order, prior to the mode selection process. The number of modes considered shall be sufficient so that their total modal effective weights shall not be less than 80 percent of the total weight of the system.

In this sorting process it is useful to construct a graph of the modal effective weight versus frequency. The graph will provide an overview of the system modal responses and will provide early identification of the existence of closely spaced modes. See 3.6.7 for further discussion of closely spaced modes.

3.6.3.3 Modal Effective Weight in Excess of Minimum Percent of Total Weight. All calculated modes contributing a modal effective weight in excess of a minimum percent of the total weight of the system analyzed shall be included in the selection. The value of the minimum percent of the total weight of the system shall be the greater of 1 percent or 20 divided by the number of dynamic degrees of freedom (NDOF) in the model expressed as a percent. The value 20/NDOF, expressed as a percent of the total weight, is intended to exclude the least massive modes of small dynamic systems. However, for a two-degree-of-freedom model both modes are to be considered regardless of this minimum percent of total weight criteria. Similarly, for a three-degree-of-freedom model, at least two modes must be used. When a system consists of a series of repeated cells or modules, the minimum percent of total system weight criterion shall be based on the weight of a single cell or module, not the total weight of the system. This will reduce the chance of omitting a mode which is primarily responsible for the movement of a given cell.

3.6.3.4 Modal Effective Weights Less than Minimum Percent of Total Weight. All additional modes of systems with modal effective weights less than the minimum percent of the total weight of the system which are deemed likely to produce critical stresses within the model are to be included in the selection. Specifically, relatively lightweight sub-components may derive a significant portion of their localized response to shock from a seemingly insignificant mode. Examples of such critical areas include antennas on yardarms, control panels, and gages. The additional modes to be included shall be those in which the nodal acceleration exceeds 10 percent of the maximum nodal acceleration (of a corresponding node) from any previously selected mode. Only the responses of those node representing critical areas or components need be considered. Alternative mode selection criteria may be used if approved by the Technical Authority.

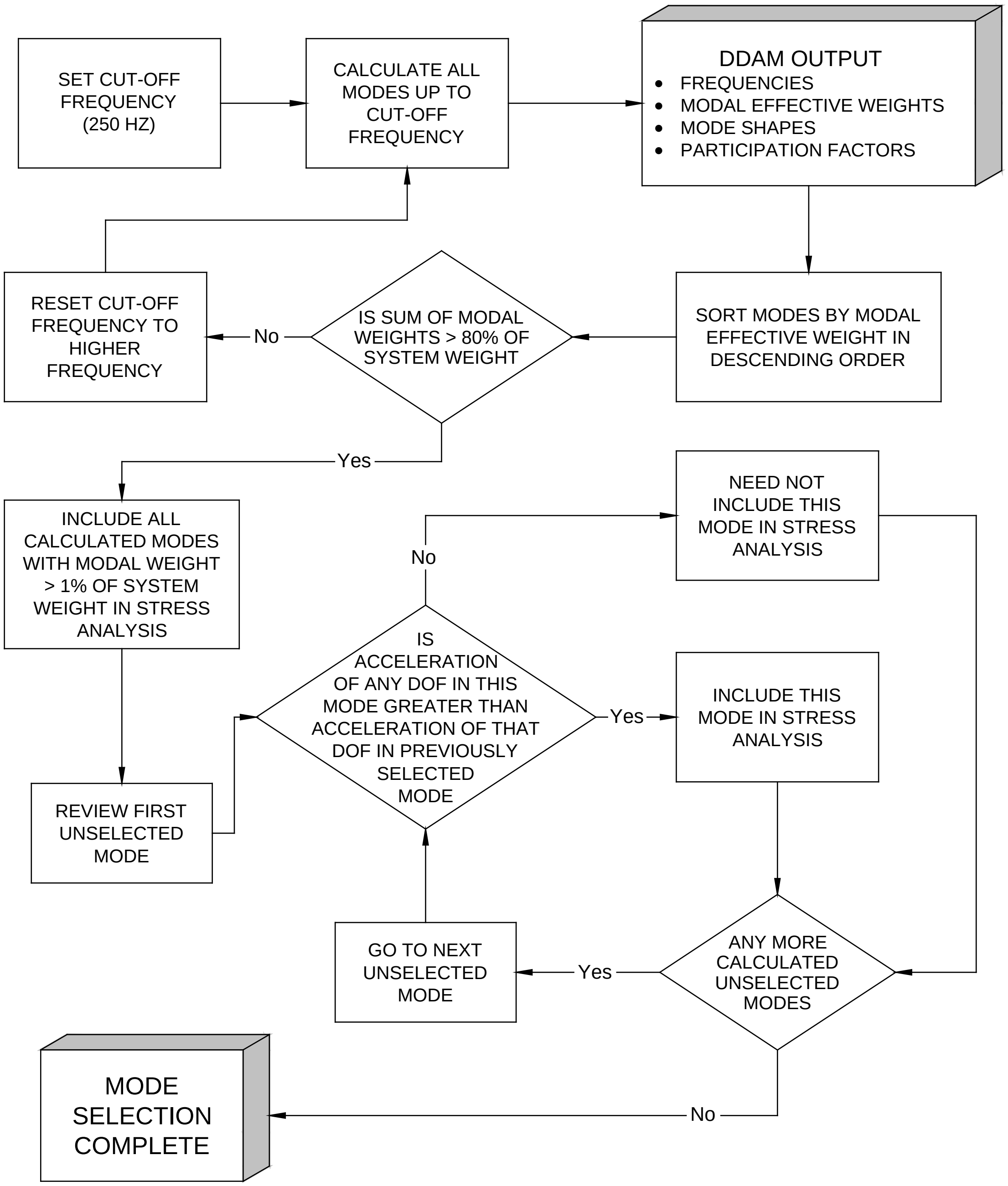


Figure 3-7. Mode Selection Process.

3.6.4 <u>Calculating Stresses Within Each Mode</u>. The following stress formula shall be used in each mode to determine the maximum modal stress. The NRL summation procedure outlined in 3.6.5 is then applied to obtain a total shock stress summed across the modes.

The Von Mises Theory of Failure is used to determine the modal stress σ_a in a structural member subjected to both normal and shear stresses. Modal stresses may require modification before summing across the modes (see 6.3.2 and 6.4). The formulas are as follows:

For the uni-directional case the modal stress σ_a for the a[th] mode is given by:

$$\sigma_a = \sqrt{\sigma_{norm}^2 + 3\tau_{shear}^2}$$

Where σ_{norm} is the total normal stress produced by axial and bending loads and τ_{shear} is the total shear stress produced by either shearing or torsional loads.

For two-dimensional analysis:

$$\sigma_a = \sqrt{\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2}$$

Where σ_x is the normal stress in the x direction of the element coordinate system, σ_y is the normal stress in the y direction of the element coordinate system and τ_{xy} is the shear stress.

For the three-dimensional case:

$$\sigma_a = \sqrt{\sigma_x^2 + \sigma_y^2 + \sigma_z^2 - \sigma_x\sigma_y - \sigma_y\sigma_z - \sigma_x\sigma_z + 3(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{xz}^2)}$$

$$\text{Mean axial stress} = \frac{\sigma_x + \sigma_y + \sigma_z}{3}$$

Where σ_x, σ_y, and σ_z are the normal stresses in the x, y, and z directions and τ_{xy}, τ_{yz}, and τ_{xz} are the shear stresses.

Consider an element of a mathematical model of a multi-degree of freedom system that has the following stresses in a particular mode of response:

$$\sigma_x = 20.0 \text{ ksi } (137.9 \text{ x } 10^6 \text{ N/m}^2)$$

$$\sigma_y = -15.0 \text{ ksi } (-103.4 \text{ x } 10^6 \text{ N/m}^2)$$

$$\tau_{xy} = 10.4 \text{ ksi } (71.7 \text{ x } 10^6 \text{ N/m}^2)$$

The combined shock stress for this element is:

$$\sigma_a = \sqrt{20^2 - (-15)(20) + (-15)^2 + 3(10.4)^2}$$

$$= 35.3 \text{ ksi}$$

$$\begin{pmatrix} \sigma_a = 10^6\sqrt{137.9^2 - (-103.4)(137.9) + (-103.4)^2 + 3(71.7)^2} \\ = 243.7x10^6 N/m^2 \end{pmatrix}$$

3.6.5 Summing of Stresses and Deflections Across the Modes by NRL Method. The following NRL sum formula developed by NRL shall be used when calculating the total shock stress or total relative deflection at point i:

$$R_i = |R_{ia}| + \sqrt{\left(\sum_{b=1}^{N} R_{ib}^2\right) - R_{ia}^2}$$

Where R_{ia} is the value of the largest modal stress or deflection (for all the modes selected) at the point i and R_{ib} represents each member of the complete set of stress or deflection contributions at the same point under consideration. Unless the stresses or deflections under consideration are directly proportional to the forces, this formula is never to be used to combine modal force on a mass(es) where these resultant forces are then used to calculate stresses or deflections.

Example: Suppose the following modal stresses were calculated for a point on an element of a two dimensional model (Note: The combined stress in each mode is determined as described in 3.6.4):

Mode Number A	σ_x ksi (x 10^6 N/m^2)	σ_y ksi (x 10^6 N/m^2)	τ_{xy} ksi (x 10^6 N/m^2)	σ_a ksi (x 10^6 N/m^2)
1	10.0 (68.95)	3.59 (24.75)	1.32 (9.10)	9.1(62.74)
2	20.0 (137.89)	-15.0 (-103.42)	10.4 (71.70)	35.3 (243.70)
3	3.0 (20.38)	2.0 (13.79)	1.63 (11.24)	3.9 (26.89)
4	1.2 (8.27)	-0.2 (-1.38)	2.03 (13.99)	3.8 (26.20)
5	8.2 (56.54)	1.0 (6.89)	1.92 (13.24)	8.4 (57.92)

Then R_{ia} = 35.3 ksi (243.7 x 10^6 N/m^2) and the formula is applied as follows:

$$R_i = |35.3| + \sqrt{9.1^2 + 35.3^2 + 3.9^2 + 3.8^2 + 8.4^2 - 35.3^2}$$

$$= 35.3 + 13.5 = 48.8 \text{ ksi}$$

$$\begin{pmatrix} R_i = |243.7(10^6)| + 10^6\sqrt{62.7^2 + 243.7^2 + 26.2^2 + 57.9^2 - 243.7^2} \\ = 10^6(243.7 + 93.27) = 336.97x10^6\, N/m^2 \end{pmatrix}$$

Therefore the value of the total shock stress summed across the five modes is σ_{shock} = 48.8 ksi (336.97 x 10^6 N/m^2).

3.6.6 Combining Operating and Shock Stresses (Total Stresses). In order to compare the stresses produced by shock loading to a specified failure criterion, the analyst shall combine the Von Mises stresses derived by dynamic analysis with the continuous Von Mises operating stresses present in the area under consideration. Continuous operating stresses are defined as those stresses, present in the system due to the system's operating characteristics (e.g., rotating elements, steam pressure, etc.), which will not be relieved by minor yielding. An example of a continuous operating stress is that which is produced by the torsional effect of a rotating element. Non-continuous operating stresses, such as thermal stresses, shall be ignored. Gravity loads need not be considered. The portion of the bolt pre-load tensile stress to be added to the shock stress shall be consistent with the relative stiffness of the joint elements to the bolt. For rigid joints bolt pre-load stress shall not be added to the shock stress. For joints with flexible elements (e.g., gaskets, Mechanical Adjustable Chocks, etc.), the pre-load shall be accounted for.

For dynamic analysis purposes, the total stress shall be the combination of the shock stress summed across the modes by the NRL method described in 3.6.5 and the continuous operating stress. The total stress at a point shall be calculated by the following formula:

$$\sigma_{total} = |\sigma_{shock}| + |\sigma_{oper}|$$

The total stress σ_{total} is compared to the allowable stress of the material to determine whether failure will occur. Allowable shock stress criteria are contained in Chapter 6 of this manual.

As an example of the method used to combine operating stresses at right angles to each other, assume a 20,000 HP (14.91 MW) shaft in an equipment is rotating at 2,000 RPM (209.3 rad/s) (continuous operating load). This rotation yields a continuous operating torque of:

$$T = \frac{33{,}000(HP)12}{2\pi(RPM)}$$

$$= \frac{33{,}000(20{,}000)12}{2(3.14)2{,}000}$$

$$= 630.57 \text{ inch-kips}$$

The maximum torsional stress on the shaft surface is:

$$\tau_{tor} = \frac{Td}{2J}$$

Assume the shaft diameter d = 7 inches (177.8 mm) and the shaft cross-section polar moment of inertia J = 236 in^4 (9.82 x 10^{-5} m^4). Then:

$$\tau_{tor} = \frac{630.57(7)}{2(236)} = 9.35ksi(64.4x10^6 MPa)$$

This operating torsional stress is now added to the operating axial stress at the point of maximum stress. Assume the axial operating stress to be 34,700 psi (239.25 x 10^6 MPa) in compression. The total operating stress is:

$$\sigma_{oper} = \sqrt{\sigma_{axial}^2 + 3\tau_{tor}^2}$$

Therefore the total operating stress is:

$$\sigma_{oper} = \sqrt{34.7^2 + 3(9.34)^2} = 38.3ksi$$

$$(\sigma_{oper} = 10^6\sqrt{239.25^2 + 3(64.4)^2} = 264x10^6 MPa)$$

The total stress is a combination of the Von Mises shock stress and the Von Mises operational stress. With the total operating stress of 38.3 ksi (264 x 10^6 MPa) and the result shown previously (see 3.6.5) for the shock stress, the total stress becomes:

$$\sigma_{total} = 48.8 + 38.3 = 87.1ksi$$

$$\left(\sigma_{oper} = (336.97 + 264.0)x10^6 = 600.97x10^6 MPa\right)$$

3.6.7 Response Assessment. The basic method of determining the acceptability of a design is by DDAM using the NRL method of combining the responses over the modes. Where the NRL method produces responses that are within the allowable limits, the requirements of this section do not apply. Where the NRL method produces results significantly greater than the allowable failure criteria, the analyst shall conduct further analysis of the equipment to determine if the responses can be reduced to levels within the allowable limits. In these cases, the following three options are available to the analyst.

a. If the high responses are not caused by closely spaced modes, the item shall be redesigned to reduce the responses to acceptable limits. If the overstress results from a closely spaced modes condition, the analysis should show the extent of detuning necessary to eliminate the overstress condition. If damaging effects of closely spaced modes cannot be eliminated by remodeling or redesigning (detuning), the analyst should request the Technical Authority approval of application of an alternate assessment in accordance with the CSM method or by using the ASM. Both methods consider the effects of modal phasing. These methods can only be presented as a supplemental calculation to the NRL summation method of 3.6.5, and should only be used as a cost effective alternative to redesigning the foundation or equipment.

b. The Closely Spaced Modes (CSM) method combines two or more closely spaced modes into one mode. This method is restricted to mode pairs which have frequencies within 10 percent of the common mean frequency, and have amplitudes which are opposite in sign. The contributions of these closely spaced modes are then included in the NRL sum as a single effective mode. The method can be easily applied by using Figure 3-10 to account for the combined effect of two modes. Refer to 3.6.7.1 for the details of the CSM method and for an example calculation. Computer software has been developed to implement the CSM method automatically.

c. The Algebraic Summation Method (ASM) is an alternate method of combining modal responses that preserves the phase relationships among the modes. The set of modes required to be used in the ASM calculation is the same as those selected in accordance with 3.6.3 for the NRL summation. Refer to 3.6.7.2 for the details of ASM and for an example calculation.

Application of the CSM method or ASM will produce more credible results if closely spaced modes are the primary cause of the high shock responses. If closely spaced mode phenomena are not the cause of the high calculated response, then application of CSM method or ASM will not have a significant effect on the results. The phenomenon known as closely spaced modes is an artificial amplification of the response of a system. It occurs when the phase relationship between individual modes with very close natural frequencies is ignored in the NRL method of summing modal shock responses.

When the responses calculated by the ASM or CSM method are significantly less than the responses calculated by the NRL method, the ASM or CSM method responses provide a technical basis for determining the acceptability of a design. However, the ASM or CSM method shall only be used in cases when the NRL method cannot produce a cost effective design.

The Technical Authority will determine the extent to which the results of the ASM or CSM method supplementary analysis will influence the final decision to accept the lower stress values as the shock response levels in the item. NAVSEA will decide whether or not to modify the structure to withstand the loads associated with the NRL summation results. This decision will depend on, among other things, the criticality of the item, the reliability or the mathematical model, and the relative impact of implementing design modifications.

Descriptions and example calculations of the CSM method and ASM are shown in 3.6.7.1 and 3.6.7.2, respectively. It must be pointed out that these examples represent the peak response at only one location and serve only to illustrate a sample calculation procedure for the CSM method and ASM. In practice, the calculations must be performed at all points that are being assessed for closely spaced modes. When performing calculations for beam elements, multiple points of the cross-section must be checked to ensure that the most critical location is evaluated.

3.6.7.1 Closely Spaced Modes (CSM) Method. The analysis method described below provides a method for combining responses from two closely spaced modes. The method does not eliminate the need to calculate a response which includes all significant participating modes, but it does provide a method for calculating the combined effect of closely spaced modes. Once this combination is determined, it may be used in the NRL sum of responses as a single effective mode.

In a DDAM shock analysis, the normal practice is to combine the responses from individual modes using the NRL sum. This practice does not explicitly treat either the relative phasing of the individual modes or the effects of damping.

For finite element models which have significant responses in modes which are close in frequency and for which the modal responses are nearly equal in amplitude and are opposite in sign (180 degrees out of phase), damping becomes very important in determining the combined response. Since they are initially out of phase, these modal responses tend to cancel each other during early portions of the response. As time passes, the frequency difference causes the responses to shift in phase so that the magnitudes eventually add. For close frequencies, this time will be large enough so that the combined amplitude can be significantly reduced by the effects of damping. See Figures 3-8 and 3-9 for examples of the superposition of two modes with and without damping.

The associated amplitude reductions are most significant where the responses of the closely spaced pair of modes are about equal in amplitude. An explicit, closed-form method for determining the reduction that can be achieved, as well as alternate numerical and graphical methods for determining the amplitude reduction, is provided in 3.6.7.1.1.

The treatment given below, and associated derivations, assume that the phasing is that associated with a velocity step input. The justification for the method, however, is based on comparison of analysis to full scale ship shock test data. Therefore, no restriction relative to step velocity is included in the method. However, the method is limited to closely spaced modes, which are defined here as having frequencies within 10 percent of the common mean frequency of the modes considered.

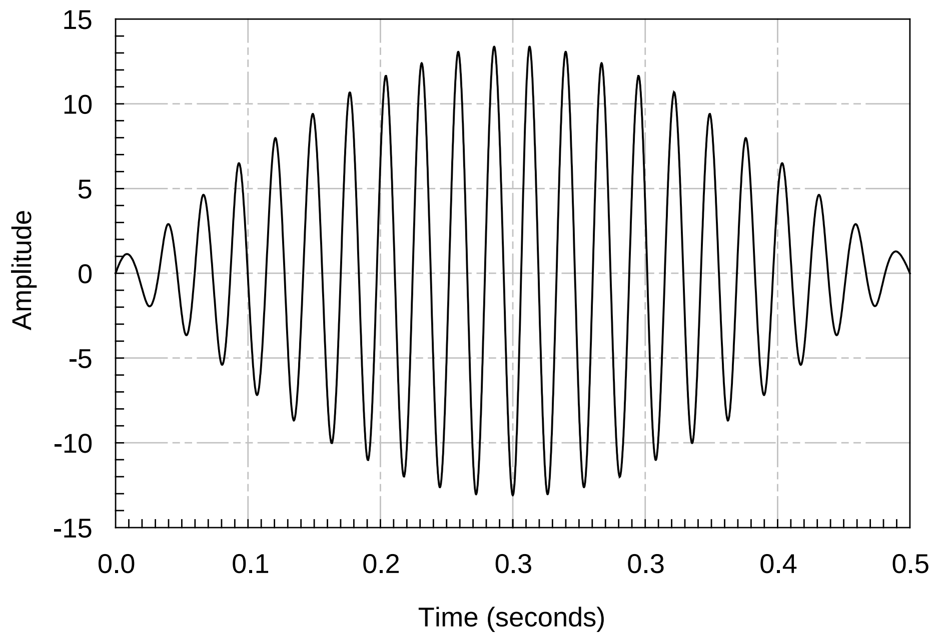


Figure 3-8. Combined Responses of Two Undamped, Closely Spaced Modes.

Damping has been set at 2 percent of critical as a lower bound estimate of the damping normally associated with the shock response of welded structures.

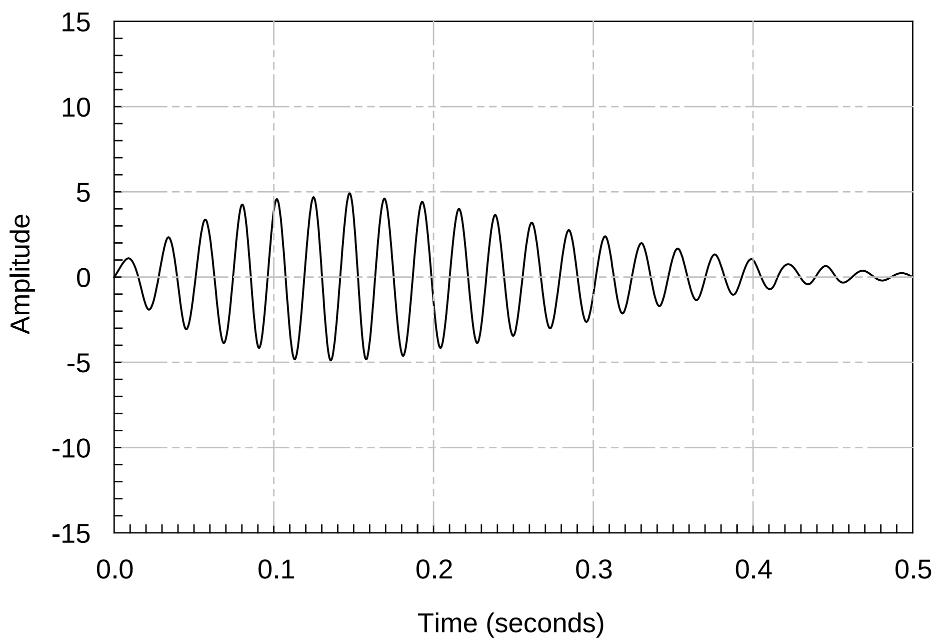


Figure 3-9. Combined Response of Two Damped (2 Percent), Closely Spaced Modes.

3.6.7.1.1 Analysis Methods. This section presents the closed form, numerical, and graphical procedures for evaluating closely spaced modes using the CSM method. The closed form may be slightly more conservative because it is based on determining the peak of the envelope rather than the peak of the superposed values. The numerical procedure, while more tedious, provides an alternative method which might be (in the future) extended to a cluster of several modes. The graphical approach is the simplest to apply, but the graph in Figure 3-10 is strictly limited to 2 percent damping.

Damping is not associated with a particular mode because application of the procedure to date has included only cases with uniform damping.

All three procedures require an amplitude correction from the DDAM-determined modal values in order to account for the effect of damping during the first quarter cycle. Omission of this correction will result in lower modal amplitudes (about a 3 percent error for 2 percent damping).

3.6.7.1.1.1 Modal Amplitude Correction. The DDAM response spectra do not explicitly include damping. However, the values for relative amplitude or acceleration implicitly include any damping forces which act during the time from shock arrival to the maximum component response. For a step velocity model of the input, this would imply damping had been acting for one quarter of a cycle when the peak acceleration or displacement is reached. C_j accounts for damping during this time:

$$C_j = A_j e^{\xi \frac{\pi}{2}} \quad \textbf{(1)}$$

Where:

A_j = mode algebraic amplitude from DDAM for the j^{th} mode.

C_j = mode algebraic amplitude for the j^{th} mode with quarter cycle correction.

ξ = damping as a fraction of critical = 0.02.

The effect of the correction is not large. For ξ = 2 percent, C_j/A_j=1.032. For larger damping values, the correction would be larger.

3.6.7.1.1.2 Closed Form Treatment. The envelope of the sum of two decaying sinusoids (modes j and k) may be written as a function of the algebraic amplitude and damping for each sinusoid:

$$E(t) = e^{-at}\sqrt{(C_j + C_k)^2 - 4C_j C_k \sin^2(dt)} \quad \textbf{(2)}$$

Where:

$E(t)$ = combined effect of two modes

C_j, C_k = mode algebraic amplitudes with quarter cycle correction

a = $\xi\Omega_m$

d = $0.5\sqrt{1-\xi^2}\left|\Omega_j - \Omega_k\right|$

Ω_m = average undamped natural frequency in radians per second

ξ = damping in fraction of critical damping

t = time in seconds

The times at which this function is an extreme (a minimum or a maximum) are t = 0 and the times given by:

$$t_n = \frac{\sin^{-1}\left[\dfrac{-a(C_j^2 + C_k^2)}{2C_j C_k \sqrt{a^2 + d^2}}\right] - \theta}{2d} \qquad \textbf{(3)}$$

Where:

$$\theta = \tan^{-1}(a/d) \text{ and } 0 \le \theta \le \pi/2$$

Equation 3 has multiple solutions only if:

$$S = \frac{-a(C_j^2 + C_k^2)}{2C_j C_k \sqrt{a^2 + d^2}} \le 1 \qquad \textbf{(4)}$$

If S is greater than one, than E_{max} is at t=0. If S is equal to one, there is one solution, t_1, to equation 3 and E_{max} is the greater of E(0) and E(t_1). If S is between 0 and one, E(t) must be calculated at t = 0 and at the first two positive values of t_n from equation 3. E_{max} is then greatest of the three values. If S is less than zero, then the modes are additive and the CSM sum cannot be used.

Once E_{max} is determined, the modified NRL (or CSM) sum may be written:

$$\sum_{CSM} = \max\left|A_m, E_{\max}\right|_{m\neq j,k} + \sqrt{E_{\max}^2 + \sum_{m\neq j,k} A_m^2 - (\max\left|A_m, E_{\max}\right|_{m\neq j,k})^2} \qquad \textbf{(5)}$$

Where the index m ranges from one to the highest mode considered, excluding the closely spaced modes, and A_m is the unsigned amplitude.

3.6.7.1.1.3 Numerical Treatment. The individual modal contribution may also be combined using the numerical procedure described below. If the corrected mode algebraic amplitude for an individual component is C_j at a natural frequency, Ω_j in radians per second with damping, ξ as a fraction of the critical damping, the amplitude at any time may be written:

$$D_j(t) = C_j e^{-\xi\Omega t} \sin(\sqrt{1-\xi^2}\,\Omega_j t) \qquad \textbf{(6)}$$

Thus, for two modes:

$$D(t) = D_j(t) + D_k(t) \qquad \textbf{(7)}$$

may be calculated to identify the maximum amplitude, $E_{max} = D(t)_{max}$.

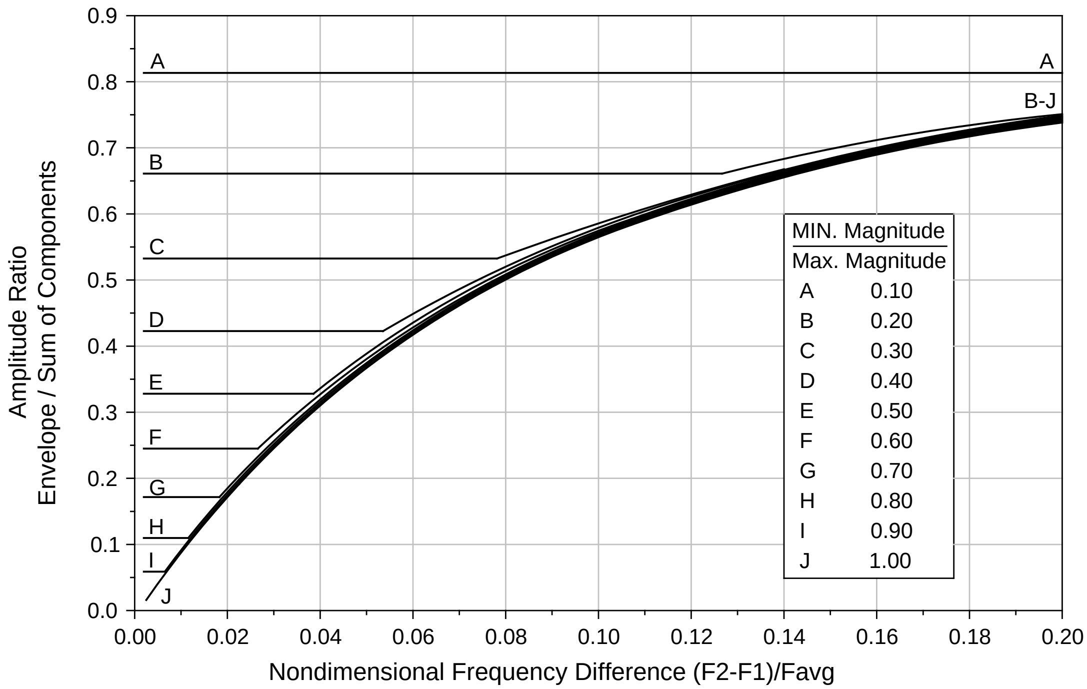
Equation 5 may then be used to determine the CSM sum.

The accuracy of the above procedure is dependent upon the time step used in the numerical procedure. If the time step is too large, an unconservative sampling error will result. The time step shall be, as a minimum, 1/32 of the shorter period of the two frequencies to keep the error in any mode due to time resolution below 2 percent.

As a minimum, D(t) should be calculated to one half the “beat cycle” of the combined frequencies. That is, for:

$$0 \langle\, t \,\langle \frac{0.5}{\left|f_k - f_j\right|}$$

3.6.7.1.1.4 Graphical Treatment. As another alternate to evaluation of the equations of 3.6.7.1.1.2, Figure 3-10 provides a graphical representation for the combined effect of two modes. This figure allows determination of the combined effect to two modes without direct calculation. The ratio of the envelope magnitude to the sum of the unsigned magnitudes of the original modes may be read from the figure given a magnitude ratio (smaller divided by the larger) and a non-dimensional frequency difference $2(f_k-f_j)/(f_k+f_j)$.



(This figure was generated for a damping of 2 percent)

Figure 3-10. Envelope/Sum of components for use with Closely Spaced Modes Sum.

3.6.7.1.2 Example Problems. Assume that a DDAM analysis has resulted in the following fixed base frequencies and modal responses of some point P, on the structure or equipment being analyzed.

Mode	Frequency (Hz)	Acceleration of P (g)
1	27	5.0
2	43	7.0
3	45	-6.0
4	87	-3.0
5	91	-2.0

$$\sum_{NRL} = 7.0 + \sqrt{5.0^2 + 6.0^2 + 3.0^2 + 2.0^2} = 15.6g's$$

Modes 2 and 3 are close in frequency and the acceleration responses at P have the opposite sign. The relative difference in frequency for modes 2 and 3 is calculated:

$$\frac{2(f_3 - f_2)}{f_3 + f_2} = 0.045 = 4.5\%$$

Since the difference is less than 10 percent, one may proceed.

Note that the frequencies of modes 4 and 5 are also within 10 percent of their common mean frequency of 89 Hz. A reduction cannot be achieved by combining these modes, however, since the modal accelerations have the same sign.

The modal amplitude correction is then applied to both modes 2 and 3. From equation 1:

$C_j \; = \; A_j e^{\pi\xi/2}$

$C_2 \; = \; 7.0 \; e^{\pi 0.02/2} \; = \; 7.22$ g

$C_3 \; = \; -6.0 \; e^{\pi 0.02/2} \; = \; -6.19$ g

3.6.7.1.2.1 Approach 1: Closed Form Treatment (Example Problem).

For an analytical solution, the equations from 3.6.7.1.1.2 may be evaluated directly. First, the preliminary calculations:

$$a = \xi\Omega_m = 0.02(2\pi)44 = 5.5292\,\text{sec}^{-1}$$

$$d = \sqrt{1-\xi^2}\,\frac{(\Omega_3 - \Omega_2)}{2} = \sqrt{1-\xi^2}\,\pi(f_3 - f_2)$$

$$= \sqrt{1-(0.02)^2}\,\pi(45-43) = 6.2819\,\text{sec}^{-1}$$

Use equation 4 to check that a solution exists:

$$S = \frac{-a(C_2^2 + C_3^2)}{2C_2C_3\sqrt{a^2 + d^2}}$$

$$= \frac{-5.5292(7.22^2 + (-6.19)^2)}{2(7.22)(-6.19)\sqrt{5.5292^2 + 6.2819^2}} = 0.6685 \le 1$$

Because S is less than one, multiple solutions to equation 3 exist. Only the first two solutions are of interest as they are potential absolute maximums of the envelope.

Equation 3 gives the times at which the envelope of the sum of the damped sinusoids is at a relative extreme (minimum or maximum). The first two solutions are given by the following expressions:

$$t_1 = \frac{\sin^{-1}(S) - \tan^{-1}(\frac{a}{d})}{2d}$$

$$= \frac{\sin^{-1}(0.6685) - \tan^{-1}\left(\frac{5.5292}{6.2819}\right)}{2(6.2819)} = 0.00083\,\text{sec}$$

and

$$t_2 = \frac{\pi - \sin^{-1}(S) - \tan^{-1}\left(\frac{a}{d}\right)}{2d}$$

$$= \frac{\pi - \sin^{-1}(0.6685) - \tan^{-1}\left(\frac{5.5292}{6.2819}\right)}{2(6.2819)} = 0.1343\,\text{sec}$$

The inverse trigonometric functions in the above expressions were evaluated to yield results in radians. Substituting into equation 2 with t = t_2 gives:

$$E(t_2) = e^{-at_2}\sqrt{(C_2 + C_3)^2 - 4C_2C_3\sin^2(dt_2)}$$

$$= e^{-5.5292(0.1343)}\sqrt{(C_2 + C_3)^2 - 4C_2C_3\sin^2 dt}$$

$$= 0.4759\sqrt{(7.22 + (-6.19))^2 - 4(7.22)(-6.19)\sin^2(6.2819(0.1343))}$$

$$= 4.8 \text{ g}$$

The height of the envelope at the other times, t = 0 and t = t_1, must also be calculated. The results of those calculations are:

E(0) = 1.0 g and

E(t_1) = 1.0 g

Therefore E_{max} = E(t_2) = 4.8. The CSM sum may now be calculated from the following modal contributions:

Mode	Acceleration
1	5.0 g
2	4.8
3	–3.0
4	–2.0

$$\sum = 5.0 + \sqrt{4.8^2 + 3.0^2 + 2.0^2} = 11.0 g's$$

Comparing the closely spaced modes sum with the NRL sum for point P in this example, a reduction of (15.6-11.0)/15.26 or 29 percent is obtained. This is slightly more reduction than the graphical solution.

3.6.7.1.2.2 Approach 2: Numerical Treatment (Example Problem).

This treatment (described in 3.6.7.1.1.3) requires calculation of equation 6 at many times for each mode. The time step must be less than 1/32 of the shorter period.

$$\left(\frac{1}{32}\right)\left(\frac{1}{45}\right) = 0.0006944\,\text{sec}$$

For convenience, choose Δ t = 0.000667 seconds. The total time considered must be for:

$$0 \langle t \langle \frac{0.5}{f_3 - f_2} = \frac{0.5}{45 - 43} = 0.25\,\text{sec}$$

Thus 0.25/0.000667 or 375 solutions of equation 6 are required for each mode. This obviously requires a computer even for this simple example.

For modes 2 and 3 of this sample, equation 6 becomes

$$D_2(t) = 7.22e^{-5.4035t} \sin(270.123t)$$

$$D_3(t) = -6.19e^{-5.6549t} \sin(282.687t)$$

$$D(t) = D_2(t) + D_3(t)$$

The calculation is not reproduced here. Figure 3-9 shows a typical plot of D(t) as a function of time. The maximum value determined at 136.068 msec is:

$$D(t)_{max} = |\, D(0.136068)| = 4.8 \text{ g}$$

The CSM sum may now be calculated from the "modal contributions":

Mode	Acceleration
1	5.0 g
2 & 3	4.8
4	-3.0
5	-2.0

$$\sum_{CSM} 5.0 + \sqrt{4.8^2 + (-3.0)^2 + (-2.0)^2} = 11.0g's$$

Comparing this CSM sum with the NRL sum of 15.6, a reduction of (15.6-11.0)/15.6 or 29 percent is achieved.

3.6.7.1.2.3 Approach 3: Graphical Treatment (Example Problem).

The nondimensional frequency ratio calculated above is 0.045. The amplitude ratio is 6.19/7.22 = 0.857. Examination of Figure 3-10 gives:

E/(sum of magnitudes) = 0.37

or

E = 0.37(6.19 + 7.22) = 5.0 g

The CSM sum is then calculated from the following contributions:

Mode	Acceleration
1	5.0 g
2 & 3	5.0
4	-3.0
5	-2.0

$$\sum_{CSM} 5.0 + \sqrt{5.0^2 + (-3.0)^2 + (-2.0)^2} = 11.2 g's$$

Comparing the CSM sum with the NRL sum for this example shows a reduction of (15.6-11.2)/15.6 = 28 percent.

3.6.7.2 The Algebraic Summation Method (ASM). ASM uses repetitive calculations that are not practical for manual calculation but can be easily programmed for any computer. ASM can be applied to any response characteristic, for example stress, member force, acceleration, velocity, displacement, or relative displacement.

3.6.7.2.1 Example ASM. As an example, the ASM is applied to a beam element from a mathematical model in the following manner:

a. Step 1: A set of discrete times at which to calculate the stress time history is selected. The calculation should be made over a time interval beginning at time zero and continuing until the lowest natural frequency mode of the summation (first mode) has been damped by 50 percent or until the envelope of any closely spaced pairs reaches a maximum, whichever is greater. The fraction of critical damping should be 2 percent. The discrete times should be evenly distributed over the interval at a spacing of one tenth of the period of the highest mode in the summation. Larger time steps are not allowed.

$$e^{-2\pi\xi f_1 T_{\max}} = 0.50$$

$$T_{\max} = \frac{\ln(2)}{2\pi\xi f_1}$$

$$= \frac{5.516}{f_1}$$

and

$$t_{inc} = 1000 \text{ msec/sec x } \tfrac{1}{10} \text{ x } 1/f_n$$

$$= 100/f_n$$

where:

T_{max} = duration of time interval, in seconds

f_1 = natural frequency of the first mode, in Hz

f_n = natural frequency of highest mode in summation

t_{inc} = time step increment, in milliseconds

b. Step 2: A set of points of interest on the periphery of the cross-section of the beam is selected. These are the points of possible maximum stress at which the NRL stresses were determined. It should be noted that the maximum ASM stress may not occur at the same point on the cross-section as did the maximum NRL stress. Therefore, all potential locations on the cross-section must be evaluated. For each of the points of interest steps 3 through 6 are performed.

c. Step 3: At each discrete time the equivalent static force vector and/or moment vector in each mode at the end of each beam element under consideration is multiplied by the damping factor and the wave amplitude of the corresponding mode to give the ASM modal force at time (t). The wave amplitude of each mode at time t is equal to the sine of the product of the natural frequency (in radians/sec) of the mode and the time (in seconds):

$$M_c^t = M_c e^{-2\pi\xi f_n t} \sin(2\pi f_n \sqrt{1-\xi^2}\, t)$$

where:

M_c^t = member force at time t for a given mode

M_c = maximum member force for a given mode

$\exp(-2\pi\xi f_n t)$ = damping factor

$\sin(2\pi f_n (1-\xi^2)^{\frac{1}{2}} t)$ = wave amplitude

c = subscript which indicates the plane in which the member force acts

f_n = frequency of the mode

t = the discrete time

d. Step 4: At each discrete time, an algebraic (vector) summation of the ASM modal forces is performed over all the modes (n) considered to be acting at each point of interest.

$$M_c^t = \sum_n M_c^t$$

e. Step 5: The sum of the forces at each discrete point and time is used to calculate the resultant normal and shear stresses acting at the point by the conventional methods of strength of materials.

f. Step 6: At each discrete point and time, the Von Mises stress is calculated from the resultant normal and shear stresses. For each point of interest the maximum combined stress is the maximum response calculated at all of the discrete times at that point. The ASM stress for the beam element is the greatest Von Mises stress of all the points at the cross-section of the element.

When the calculated ASM stress is less than the NRL stress, it may be compared to allowable values given in Chapter 6 to determine the adequacy of a design for shock. If the ASM combined stress for any member exceeds the allowable values, the design should be modified to eliminate the over-stress determined by the NRL method. If the ASM combined stress is less than the allowable value, the design may be accepted by the Navy as adequate for shock.

In the event that parameters other than stresses are used to determine the shock adequacy of a design, the above calculation procedure may still be applied. Likely alternatives to the stresses are forces, relative displacements and accelerations, etc. For modal forces, the above procedure should be amended by omitting the stress calculations (steps 5 and 6) and substituting the vector quantity of the desired response characteristic (modal force) in step 3 above. The vector sum determined in step 4 will be the value of the response characteristic time history at the particular point and time.

The ASM value of the response characteristic would be the greatest magnitude achieved by the response-time history during the time interval considered. When stresses are used to determine acceptability, the algebraic sum of the forces (and moments) is used to determine the stresses rather than calculating a stress contribution for each mode and summing them as is done in the NRL method.

As an example of DDAM-ASM, consider a hypothetical beam element with an arbitrary cross-section in bending and shear (see Figure 3-11). Suppose the mathematical model contains the following cross-sectional properties in some consistent system of units (the subscripts ‘c’ and ‘d’ refer to the two transverse directions about which the member bends):

Sectional modulus for bending in two directions:

$Z_c = 1.0$ $Z_d = 2.0$

Shear areas for transverse shear in two directions:

$A_c = 0.1$ $A_d = 0.2$

Assume an allowable stress of 100 (in consistent units) and assume that DDAM has resulted in the following modal forces and frequencies:

MODE	BENDING MOMENTS		SHEAR FORCES		FREQ
	PLANE c M_c	PLANE d M_d	PLANE c V_c	PLANE d V_d	Hz
1	10	-20	3	2	30
2	-12	18	-4	-2	31
3	5	5	1	1	45

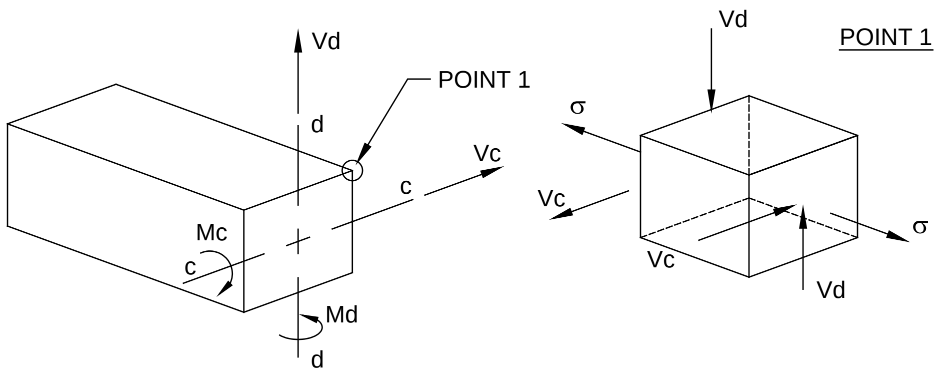


Figure 3-11. Bending and Shear Forces on Beam Element.

The NRL method of 3.6.5 and the method of combining stresses of 3.6.4 would result in the following stresses:

Mode 1:

$$\sigma_{bend} = M_c / Z_c + M_d / Z_d$$

$$\sigma_{bend} = \frac{10}{1} + \left(-\frac{20}{2}\right) = 0$$

$$\tau_c = V_c / A_c = 3/.1 = 30$$

$$\tau_d = V_d / A_d = 2/.2 = 10$$

$$\sigma = \sqrt{\sigma_{bend} + 3(\tau_c^2 + \tau_d^2)}$$

$$\sigma = \sqrt{0 + 3(30^2 + 10^2)} = \sqrt{3000} = 54.8 \text{ (consistent units assumed)}$$

Mode 2:

$$\sigma_{bend} = -12/1 + 18/2 = -3$$

$$\tau_c = -4/.1 = -40$$

$$\tau_d = -2/.2 = -10$$

$$\sigma = \sqrt{3^2 + 3(40^2 + 10^2)} = 71.5$$

Mode 3:

$$\sigma_{bend} = 5/1 + 5/2 = 7.5$$

$$\tau_c = 1/.1 = 10$$

$$\tau_d = 1/.2 = 5$$

$$\sigma = \sqrt{7.5^2 + 3(10^2 + 5^2)} = 20.8$$

The NRL sum of the Von Mises stresses is:

$$\sum_{NRL} = 71.5 + \sqrt{54.8^2 + 20.8^2}$$

$$= 71.5 + 58.6$$

$$= 130.1 \rangle 100 \text{ (greater than the assumed allowable)}$$

Note: This NRL summed stress is for only one point on the member, i.e., "point 1" shown in Figure 3-10. By the usual methods of strengths of materials, the same calculations would be repeated for all points of interest on the periphery of the cross-section.

From the frequencies above (shown in the previous table) it is seen that the first two modes are closely spaced and the NRL summed stress may be too conservative.

3.6.7.2.2 Examining ASM Results. Since the calculated NRL result above exceeds the allowable, the ASM results are examined as a basis for further technical evaluation as follows:

a. Step 1:

   (1) Find the total time interval from the lowest frequency:

   T_{max} = 5.516/30 Hz = 0.1848 sec

   (2) Find the time step spacing from the highest frequency:

   T_{inc} = 100/45 = 2.2 msec.

b. Step 2: For this example procedure only one point, the same one considered in NRL summation above, will be used.

c. Step 3: The following calculations (step 3 through 6) would be repeated for each of the 84 discrete times in the set {2.2, 4.4, 6.6, …184.8}. In this example calculations for all times were conducted, but only the details for t = 116.6 milliseconds, which was the worst case, are shown here.

   (1) Find the product of the damping factor and the wave amplitudes in each mode at the discrete time:

$$\sqrt{1-\xi^2} = \sqrt{1-0.02^2}$$

$$= 0.99980$$

$$e^{-2\pi\xi f_n t}\sin(2\pi\sqrt{1-\xi^2}\, f_n t) = e^{-2\pi(0.02) f_n (0.1166)}\sin\left[2\pi(0.9998) f_n (0.1166)\right]$$

$$= e^{-0.01465 f_n}\sin(0.73247 f_n)$$

$$e^{-0.01465(30)}\sin\left[0.73247(30)\right]=0.6444(0.01705)=0.0110$$

$$e^{-0.01465(31)}\sin\left[0.73247(31)\right]=0.6340(-0.65594)=-0.4159$$

$$e^{-0.01465(45)}\sin\left[0.73247(45)\right]=0.5172(0.99967)=0.5170$$

Note: The frequency f_n is in Hertz, and the time t is in seconds, so that the argument of the sine function is in radians. The products of the damping factor and the wave amplitude are dimensionless.

(2) Multiply the modal member force components by the corresponding wave amplitude for that mode at the selected time (the superscript ‘t’ is used to denote “at time t”). For example, the bending moment in plane c in mode 1 is calculated as follows:

(a) $M_c^t = M_c e^{-2\pi\xi f_n t}\sin(2\pi f_n\sqrt{1-\xi^2}\,t) = (10)(0.0110) = 0.11$

(b) Repeating the calculation for each force component in each mode gives:

MODE	BENDING MOMENTS		SHEAR FORCES	
	PLANE c M_c^t	PLANE d M_d^t	PLANE c V_c^t	PLANE d V_d^t
1	0.110	-0.220	0.033	0.022
2	4.991	-7.486	1.664	0.832
3	2.585	2.585	0.517	0.517

d. Step 4: Calculate the algebraic (signed) sum over the modes of the force components at the selected time (the superscript ‘T’ is used to denote “total over all the modes at time t”):

(1) $M_c^T = \sum_N M_c^T$

= 0.110 + 4.991 + 2.585

= 7.686

(2) Repeating the summation for each of the force components gives:

SUM OF THE BENDING MOMENTS		SUM OF THE SHEAR FORCES	
PLANE c M_c^t	PLANE d M_d^t	PLANE c V_c^t	PLANE d V_d^t
7.686	-5.121	2.214	1.371

e. Step 5: Based on the algebraic sum of the force components (the vector sum of the modal member forces), calculate the normal and shear stresses at the selected time:

$$\sigma_{bend} = \frac{7.686}{1} + \frac{-5.121}{2} = 5.126$$

$$\tau_c = \frac{2.214}{0.1} = 22.14$$

$$\tau_c = \frac{1.371}{0.2} = 6.855$$

f. Step 6: Based on the normal and shear stresses, calculate the Von Mises stress at the selected time.

$$\sigma = \sqrt{\sigma_{bend}^2 + 3(\tau_c^2 + \tau_d^2)}$$

$$= \sqrt{5.126^2 + 3(22.14^2 + 6.855^2)}$$

$$= 40.5$$

Note: The calculation indicates the Von Mises stress at time t = 0.1166 seconds only. Steps 3 through 6 must be repeated at each point of interest at each of the 84 discrete times. The greatest value of the Von Mises stress obtained is the ASM stress.

The above results represent the peak response at one location and serve only to illustrate a sample calculation procedure for DDAM-ASM. Although the NRL summed stress above was evaluated at only one point on the periphery of the cross-section of the beam element, it may be larger at another point on the beam cross-section.

The ASM summed stress in step 6 is for only one point and at only one time. However, for this one point, a complete ASM stress-time history was calculated and the largest stress did occur at 116.6 milliseconds. Therefore, it is appropriate to compare the NRL summed stress to the DDAM-ASM stress.

Assuming the example above resulted in a final NRL summed stress of 130.1 for the member and an ASM stress of 40.5 for the member, the member should be designed for a shock induced stress of 40.5. The relative responses reflected in this example indicate that the closely spaced modes phenomenon acts to artificially amplify the stress results when using the NRL summation method.

CHAPTER 4
FOUNDATION SHOCK DESIGN

4.1 INTRODUCTION.

All foundations which support Grade A and B equipment shall be assigned the same shock grade as the supported equipment. For foundations which require shock qualification, shock testing as described in the contract specifications or the design methods described herein shall be employed to demonstrate that the foundation is adequate from a shock standpoint. In general, possible shock damage shall be minimized. If misalignment would not interfere with operation of equipment, energy dissipation through permanent deformation of the foundation is preferable to damage to the equipment or the hull. In any case, deformation should take the form of buckling or bending of local structure, rather than permitting the equipment to tear loose from its attachment. Accordingly, joints shall develop the ultimate strength of the weakest member of the connection. Foundation deformation shall not act to compromise or invalidate the grade of shock for which the supported equipment was qualified. Foundation structures shall be proportioned to give approximately uniform stress distribution, permitting maximum absorption of energy through elastic deformation. Structural attachments of connections which minimize stress concentrations shall be used where possible. In general, brittle materials, with low ductility, as defined in 6.10, shall not be used. Where practical, under vertical shock, bolts should be loaded in tension rather than shear.

The designer should not assume that a heavier/stiffer foundation is required when developing the design of shock resistant foundations. Foundations which are initially designed without regard for shock loadings will generally satisfy shock requirements specified for any ship with little or no modifications required. The procedure to follow in meeting shock requirements for foundations is to first design the foundation to meet normal operating requirements (e.g., ship motion, vibration, air blast, wave slap, etc.) and then check the foundation to determine its adequacy from a shock standpoint. When the analysis indicates local over-stresses in the foundation, it is usually a simple matter to redesign the over-stressed area to meet shock stress requirements. To achieve an efficient design in cases where shock governs the design of a foundation, total stresses (shock plus operating) in at least the primary members shall exceed 75 percent (but not 100 percent) of the allowable stress (see 6.8).

See the shipbuilding or contract specifications for permissible bolt hole clearances. Applicable shock criteria for equipment hold-down bolts are cited in 3.2.3.d of this manual and are illustrated in Example 1 of Chapter 5 of this manual.

For systems suitable for modeling with a single degree of freedom, two alternate methods of designing shock resistant foundations, Method 1 and Method 2, are presented herein. For cases in which Method 1 applies, analysis shall be conducted using both methods and the lesser shock design loading shall be used. Method 1 or Method 2 may be used independently for each direction of shock.

Method 1 may only be used in cases where both of the following apply:

a. The mounted equipment has been qualified on the basis of shock testing. It is essential in such cases that the foundation designer not compromise the shock qualification of the equipment by his foundation design. The designer shall consider the type of support used in the shock testing of the equipment. For example, if a support of uniform stiffness at each mounting point was used in the testing, the foundation being designed should also have uniform stiffness,

b. The design of the foundation based on a single mass model to suit elastic-plastic shock criteria would be acceptable (see Chapter 3 for criteria pertaining to applicability of elastic-plastic shock design values).

In the procedures outlined below, the term “hold-down means” refers to hold-down bolts, dowels, keys, and any other devices which serve to locate or secure equipment to its foundation.

4.2 METHOD 1 (DESIGN METHOD FOR HOLD DOWN MEANS).

Method 1 procedures for design of foundations for a specific shock direction are as follows:

a. Determine the magnitude of the maximum shock loads which can be transmitted to the foundation by the equipment hold-down means by assuming the shock loading is applied at the center of gravity of the mounted equipment (or at the centers of gravity of each separately mounted equipment, if appropriate) and that the maximum load is developed when stress in one or more of the hold-down means equals 90 percent of ultimate strength in either shear or tension. For those cases in which the hold-down means are loaded for only one condition of a shock direction (e.g., bolts loaded in vertical downward direction but not in vertical upward direction), the analyst shall perform the Method 1 calculation for that condition in which the hold-down means are under loading.

b. Check all critical areas of the foundation except the connection to ship's structure to assure that the foundation can resist the loads determined by step 1 above.

c. Increase the magnitude of the shock loadings obtained in step 1 above by a factor equal to the ratio of foundation weight to equipment weight:

$$\text{Shock load at base of foundation} = \left[\frac{Weight\ of\ \ Foundation}{Weight\ of\ \ Supported\ \ Equipment} + 1\right] Shock\ \ Load_{step1}$$

Use these increased loadings for purposes of checking the connection between the foundation and the ship's structure.

d. Repeat the above three steps for the other two principal directions of shock loadings.

e. Calculate stresses in foundation members separately for each direction of shock loading. Allowable stresses are the same as for dynamically analyzed foundations which are designed to elastic-plastic shock design values. See Chapter 6 for allowable stresses.

f. If necessary, stiffen the foundation to achieve acceptable stress levels. Whenever practical, employ local stiffening only (such as by gussets) to reduce stresses to acceptable levels.

4.3 METHOD 2 (SDOF DDAM METHOD).

Method 2 is the conventional dynamic analysis method of foundation design, and is acceptable for all foundations. For purposes of foundation dynamic analysis, the item supported may generally be considered a single rigid mass and the foundation may be designed in accordance with procedures outlined in 4.6 and 4.7. Where components which must be kept in alignment are not mounted on a rigid sub-base, each component must be considered a separate mass for foundation design purposes. If shock will induce significant rocking (rotation) of the foundation in addition to translation in the shock input direction, a simplified multi-degree of freedom mathematical model should be used to represent the equipment, as illustrated in 4.8. In the model, that portion of the foundation weight consistent with its dynamic response characteristics shall be lumped with the equipment weight. The remainder of the foundation weight shall be ignored (assumed part of the fixed base). See 4.6 and 4.7.

Three or more masses may be required to adequately represent complicated foundation/equipment arrangements. In general, any major mass whose deflection under shock can be expected to differ significantly from the deflection of other portions of the structure must be separately represented by a mass point in the dynamic model.

Foundations for which multi-mass equipment representation is known to be required are listed below. Omission of equipment from this list does not relieve the contractor from his responsibility to properly model other equipment for purposes of foundation dynamic analysis.

a. Main propulsion gas turbine

b. Main propulsion reduction gear

c. Ship service diesel generator

d. Air conditioning compressor

e. Air conditioning chiller, condenser, and receiver

f. Ship service diesel engine heat exchanger

g. Lube oil cooler

h. Weapon systems (missile launchers, gun systems, torpedo tubes, etc.)

4.4 EXTENT OF FOUNDATION.

For shock design purposes, foundations shall generally be considered to end at the point where primary ship structure begins (decks, longitudinals, web frames, structural bulkheads, etc.). If the primary ship structure must be altered in any way to interface with the equipment being supported (i.e., structural element thicknesses, overall depth or width of the structural element) then this structural element has now become part of the foundation until it reaches the next primary ship structure. The primary ship structure is considered to act as a fixed base (see 3.3.2). Shock design values shall be applied at the assumed fixed base (the interface of the foundation and the primary ship structure) in accordance with 3.2.2. Since basic ship structures designed in accordance with Navy Standard Details are inherently robust and no additional shock design requirements or verification is necessary, a clear definition of the interface between ship structure and the foundation is required. Ships designed without the use of Navy Standard Details that have shock requirements may require additional design requirements or verification of structural shock hardness. The Navy Program Office will notify the Shock TWH via the SDM in such cases for review and approval of additional design and verification needs for the non-Navy Standard Detail structure. The design requirements for that interface (structural continuity) are such that the ship’s structure defining the fixed base must be able to transmit the membrane (axial, shear, and torsion) reactions at the fixed base and meet membrane acceptance criteria. Care must be taken to avoid any sudden structural discontinuity between foundation and ship structure. Chocks, brackets, or local strengthening of ship structure shall be used to provide structural continuity where necessary and checked for strength, but this added structure need not be included in the foundation mathematical model. It should be noted that deck inputs of T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1) include the effect of energy dissipation from plastic action of the deck, therefore, ship structure (deck) cannot be included in the mathematical model if deck inputs are used. If deck structure is needed in the mathematical model, then hull inputs must be used and the flexibility of the ship structure (deck) will reduce the acceleration.

4.4.1 Equipment Mounted on Shell Framing. Shell framing is not normally considered as part of the foundation, although local strengthening may be required to ensure structural continuity.

4.4.2 Equipment Mounted on Upper Levels of Machinery Spaces. Machinery space upper levels which are provided solely as a support for auxiliary machinery shall be considered as foundations, grounded on ship’s structural web frames, transverse structural bulkheads, and bottom framing (or inner bottom) through stanchion connections. The shock response and design of these levels shall consider all equipment and piping or other distributed weights supported thereon. The upper levels shall be analyzed using DDAM multi-mass techniques with hull inputs.

4.4.3 Equipment Mounted on Decks. Deck mounted equipment fall into two categories distinguished by the alignment sensitivity of the equipment. For non-alignment sensitive installations, only the structure between the deck and equipment mounting surfaces need be considered in the foundation analysis. If necessary, to ensure structural continuity or adequacy, local headers or pads shall be added to stiffen the plating or framing in way of the equipment. Beams added in the plane of the deck to suit the arrangement of foundations and to provide points for attachment of foundations, shall be designed to transmit shear forces (associated with shock loadings) to primary ship structure (longitudinals and transverse web frames).

For alignment sensitive installations, all structure expressly added for support of the equipment (including additional headers, pads, and “normal” structural members whose size has been locally increased specifically to suit the installation) shall also be demonstrated suitable from a shock standpoint. This is accomplished by imposing foundation reaction loads upon the ship structure to determine whether additional stiffening of the added structure is required. Structural continuity shall be provided between this added structure and “normal” ship structure.

4.4.4 Equipment Mounted on Structural Bulkheads. Local stiffening should be used, where necessary, to ensure structural continuity between the foundation and the supporting structural bulkhead. No general strengthening of the bulkhead should be considered solely for shock purposes.

4.4.5 Equipment Supported by Stanchions. Stanchions which are provided primarily to support heavy equipment shall be treated as an extension of the foundation and designed accordingly. Local stiffening of the interface between the stanchion and the structure upon which the stanchion falls must be provided to ensure structural continuity. Stanchions that are part of a foundation must be capable of supporting tensile as well as compressive shock loads. Stanchions which are part of the basic ship structure are designed primarily as compression members for dead, live, and sea loads and shall not be used to support equipment except as approved by the Technical Authority. Technical Authority approval of stanchion supported equipment will be limited to lightweight equipment and based on contractor submitted technical information. The contractor shall provide baseline design stress levels for the subject stanchion under operating loads, arrangement of all equipment mounted to the stanchion being considered, weight of equipment mounted to the stanchion, eccentricity of equipment center-of-gravity to center-of-stanchion, and additional shock load being introduced into the stanchion.

4.4.6 Equipment Supported by Pallets. Pallet type structures utilized for support of electronic equipment or other Grade A or B equipment shall be considered as foundations and shall be designed accordingly. Structural continuity between the pallet and the ship structure must be checked as part of the foundation shock design.

4.4.7 Equipment Mounted on Nonstructural Bulkheads. Nonstructural bulkheads include joiner, non-load bearing and non-tight, lightweight bulkheads. Where shock Grade A and B equipment are mounted to non-structural bulkheads, it is required that the bulkhead panels be considered as foundations and designed to withstand design shock loads. Bulkhead foundation systems for Grade A and B equipment should have top, bottom, and inter-panel connections designed to support design shock loads. Deflection connections and/or additional reinforcements shall be provided as required. For equipment mechanically fastened to nonstructural bulkheads considered as foundations for Grade A and B equipment, the designer should ensure that shock loads at local attachment points can be sustained by the fastener/bulkhead configuration.

4.4.8 Mechanical Attachments for Non-Metallic Hulls. The mechanical attachment of foundations to nonmetallic structure requires the designer/engineer to consider potential foundation instabilities which could occur if the design of bolted foundation attachments cannot sustain shock design loads. These attachments are typically provided by through-bolted connections attaching foundation structure or bearing brackets to ship structure. Consideration should be given to the effects of local crushing of ship structure in way of bolt attachments due to significant bolt bearing loads under shock conditions. This localized distortion of bolt openings may account for loss of equipment alignment. For alignment sensitive equipment, the local effect of bolt bearing loads should be considered in the foundation design.

Under dynamic shock load conditions the bearing strength of wood or composite structure in way of local attachments shall be considered in order to minimize the number and size of bolts required to attach foundations to ship structure.

4.5 REQUIREMENTS FOR SUPPORTING SHIP STRUCTURE.

Shock tests of ships, in which bulkheads, decks, etc., were not specifically designed for shock, have shown that structure designed for normal ship dynamic loads is generally adequate for shock loading. Nonetheless, attention shall be given to shock consideration when planning installations of certain weapon system components and any other items which are known to be alignment-critical sensitive. Structure (below foundations) supporting such items should possess the following characteristics:

a. Supporting ship structure should be “balanced” from the standpoint of resistance to deflection in the vertical direction to minimize tilting (angular misalignment) due to vertical shock. For instance, alignment-sensitive deck mounted items should be mounted squarely over bulkheads or squarely between framing members, other factors permitting. It is usually advantageous to have uniform stiffness at each mounting point of the equipment to avoid load concentrations at any one point during shock. Numerous equipment failures during ship shock testing have been traced to a disregard for this principle.

b. Plating of web frames should not be depended upon to resist angular deflections. Ensure that full structural continuity exists between alignment-critical equipment foundations and adjacent structural bulkheads or structural framing.

c. In order to avoid high lateral shock loading of stanchions and to avoid eccentric loading of stanchions (due to vertical shock), equipment shall not be attached directly to structural stanchions without Technical Authority approval.

4.6 DYNAMIC ANALYSIS OF A FOUNDATION – SINGLE DEGREE OF FREEDOM SYSTEM.

The simplest model of a foundation structure is a single degree of freedom system in which the foundation forms the spring and the equipment itself is the major portion of the mass. A schematic model of this type of system is shown in Figure 4-1. The shock loads, the total stress, and the displacements of such a system can be determined by Method 2 using the following steps:

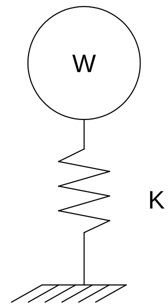


Figure 4-1. Schematic Representation of a Single Degree of Freedom System.

a. Step 1: Compute the spring constant K for a direction of shock loading. The spring constant is a measure of the stiffness of the structure and is equal to the load causing unit deflection. For the vertical shock model, the spring constant in lbs/in is numerically equal to the amount of force (lbs) acting down through the center of gravity of the equipment foundation system required to deflect the center of gravity down one inch. Simultaneous deflections of the center of gravity in other directions are ignored. Generally there will be a different spring constant in each shock direction.

b. Step 2: Determine the modal effective weight W. For an item of equipment mounted on a foundation which is to be represented as a single mass, W may be assumed equal to the equipment weight plus one-half of the weight of the foundation.

c. Step 3: Calculate the angular frequency, ω, by the following equation:

$$\omega = \sqrt{K\,g\,/\,W}$$

where: g is the gravitational constant in consistent terms

d. Step 4: Using the shock design value formulas contained in T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1), determine the design velocity value V and the design acceleration value A based on mounting location, direction of shock loading, and type of design category (elastic or elastic-plastic).

e. Step 5: Calculate the design acceleration of the system D (in gravity units) in accordance with T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1) by using:

$$D = (V)\omega\,/\,g \text{ or}$$

D = A whichever is less

D shall not be less than 6 g

f. Step 6: Determine the effective static force F applied to the equipment at its center of gravity by use of the formula:

$$F = WD$$

g. Step 6a (Optional): Where appropriate, forces resulting from application of Method 1 (see 4.2) may be compared with those derived from Method 2 (see step 6 above). Assessment of the foundation design would then be based on the shock loads which result in the least foundation weight.

h. Step 7: Apply the shock load calculated in step 6 or 6a, plus any continuous operation loads (as defined in Chapter 3). Analyze the structure using conventional static analysis procedures to determine the total stresses. If the equipment hold-down bolts are to be shock qualified by dynamic analysis, repeat step 6 with D derived from elastic shock design values and with W in step 6 equal to equipment weight only.

i. Step 8: If required for displacement-sensitive items, the maximum relative displacement of the center of gravity of the equipment with respect to the fixed base may be determined by the formula:

$$X = \frac{F}{K}$$

F is determined on the basis of elastic shock design values in all cases.

j. Step 9: Repeat the above steps for the other principal directions of shock loading.

4.6.1 Example – Single Degree of Freedom System. To illustrate the aforementioned procedure for determining the shock load on a single degree of freedom system, consider the equipment-foundation system shown in Figure 4-2. Assume that this shock tested equipment is rigid and symmetrical and that a single mass is sufficient to represent it. The shock adequacy will be determined for the vertical shock direction for upward motion of the ship (i.e., web in compression). The equipment shown in this example is not considered to be alignment sensitive; therefore, the foundation is not required to remain within the elastic range and the use of elastic-plastic shock design values is considered acceptable.

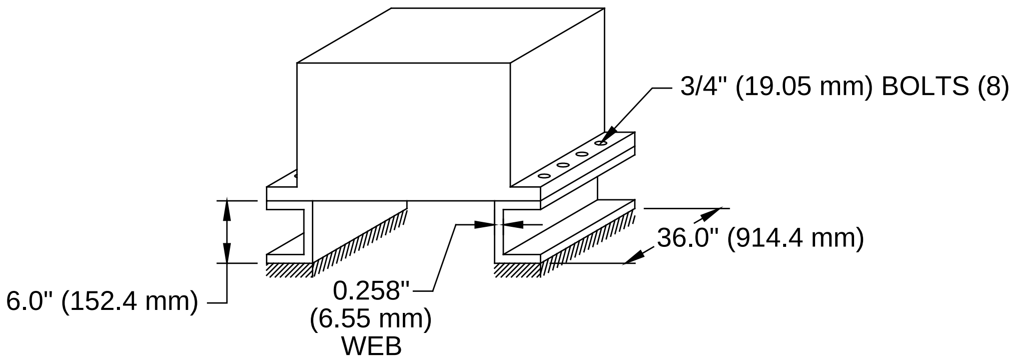


Figure 4-2. Single Degree of Freedom Foundation Model.

4.6.1.1 Characteristics. For the system shown in Figure 4-2, assume the following characteristics:

a. Equipment weight - 5000 lbs (22.241 kN)

b. Foundation weight - 720 lbs (3.202 kN) each beam

c. Equipment location - Deck

d. Category of shock design value - Elastic-plastic

e. Foundation material - Steel, $E = 30x10^6$ psi ($210\ x\ 10^6\ Pa$)

f. For the system shown in Figure 4-2, the center of gravity of the equipment is equidistant from the supports. The supports land on the fixed base (rigid frame of reference) throughout their length.

4.6.1.2 Procedural Steps.

a. Step 1: Spring Constant K

For shock in the upward direction (web in compression).

$$K_1 = \frac{AE}{L} \text{ (for one channel)}$$

$$= \frac{36(0.258)30x10^6}{6}$$

$$= 46.44x10^6 \text{ psi}$$

$$\left(\begin{aligned} K_1 &= \frac{0.9114(6.55x10^{-3})2.068x10^{11}}{0.1524} \\ &= 8.127x10^9\ N/m \end{aligned} \right)$$

$$K_2 = K_1$$

$$K = K_1 + K_2 \text{ (springs in parallel)}$$

$$= 2(46.44x10^6) = 92.88x10^6 \text{ psi}$$

$$\left(K = 2(8.127x10^9) = 1.625x10^{10}\ N/m\right)$$

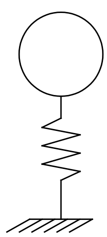
b. Step 2: Weight W

W = weight of equipment + ½ weight of foundation

$$= 5{,}000 + \frac{720 + 720}{2}$$

= 5720 lbs

Using the values obtained in steps 1 and 2 above, the system shown in Figure 4-2 is schematically represented in Figure 4-3.



W = 5,720 lbs (25,443.82 N)

K = 92.88 x 10^6 lb/in (1.625 x 10^{10} N/m)

Figure 4-3. Schematic Representation of a Single Degree of Freedom System.

c. Step 3: Angular Frequency

$$\omega = \sqrt{\frac{Kg}{W}}$$

$$= \sqrt{\frac{92.88x10^6(386)}{5{,}720}} \quad \left(\sqrt{\frac{1.625x10^{10}(9.81)}{25{,}443.82}}\right)$$

$$= 2{,}504 \frac{rad}{\sec}$$

e. Step 4: Design Velocity (V) and Design Acceleration (A)

T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1) contains formulas that give the shock design values as a function of the modal effective weight in kips. From those formulas, for a system with a modal effective weight of 5,720 lb (or 5.72 kips), vertical shock loading, deck mounted, and elastic-plastic design, the shock design values are:

V = 22.68 in/sec (0.5765 m/sec)

A = 40.7 g

f. Step 5: Absolute Acceleration D

Based on velocity:

$$D = \frac{V\omega}{g}$$

$$= \frac{22.68(2504)}{386} \qquad \left(\frac{0.5765(2504)}{9.81}\right)$$

= 147.12 g

Based on acceleration, D = A = 40.7 g

The shock design value to use is the lesser of these values, but not less than 6 g. Therefore, use D = 40.7 g

g. Step 6: Effective Static Force F

F = WD

= 5,720(40.7) (25,443.82(40.7))

= 232,804 lbs (1.036×10^6 N/m)

h. Step 6a (Optional): Computation of Effective Static Force F by Method 1

For the system shown in Figure 4-2, it is assumed that a load applied at the center of gravity of the equipment in the downward shock direction (bolts in loaded condition) will load the eight hold-down bolts equally. Therefore:

Area/bolt = 0.3340 in^2 ($2.155\ x\ 10^{-4}$ m^2)

Area(8 bolts) = 8(0.3340) = 2.672 in^2

(8($2.155\ x\ 10^{-4}$) = $1.724\ x\ 10^{-3}$ m^2)

Ultimate strength (Grade 5) = 120,000 psi (MIL-DTL-1222)

($827.37\ x\ 10^6$ N/m^2)

90 percent ultimate strength = 108,000 psi ($744.64\ x\ 10^6$ N/m^2)

Force F = 108,000 (2.672) = 288,576 lb

($744.64\ x\ 10^6$ ($1.724\ x\ 10^{-3}$) = $1.284\ x\ 10^6$ N

i. Step 7: Structural Analysis (Stresses)

Use the force F calculated in step 6 above since that value is less that the corresponding force determined by Method 1 in step 6a.

Due to the symmetry of the system, each support will experience a loading of 232,804/2 or 116,402 pounds ((1.036×10^6)/2 or 5.18×10^5 N). This is schematically represented in Figure 4-4. Note that these loadings would be increased by continuous operating loads (defined in Chapter 3), if any are present.

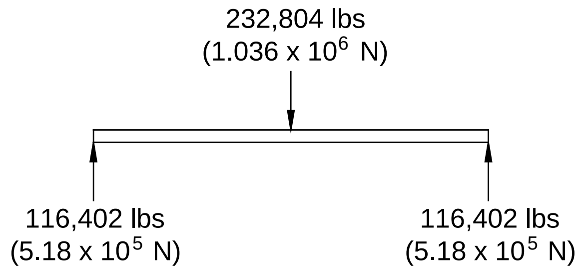


Figure 4-4. Schematic Representation of a Simply Supported Beam Loaded at the Center.

Using standard stress formulas, the compressive stress in each web of the foundation is equal to:

$$\sigma = \frac{P}{A}$$

$$= \frac{116{,}402}{36(0.258)}$$

$$= 12{,}532 \text{ psi}$$

$$\left(\begin{aligned} &\sigma = \frac{5.18x10^5}{0.9144(6.55x10^3)} \\ &= 86.49x10^6\, N/m^2 \end{aligned} \right)$$

j. Step 8: Structural Analysis (Deflection)

$$X = \frac{F}{K}$$

$$= \frac{232{,}804}{92.88x10^6}$$

$$= 0.0025 \text{ inches}$$

$$\left(\begin{aligned} &X = \frac{1.036x10^6}{1.625x10^{10}} \\ &= 0.06375 mm \end{aligned} \right)$$

The values calculated in steps 7 and 8 above shall be compared to the allowable criteria cited in Chapter 6 of this manual to determine the shock adequacy of the foundation in the upward shock direction.

k. Step 9: Steps 1 through 8 shall be repeated for the athwartship, fore and aft, and vertical downward (web in tension) directions of shock loading, if required, using the appropriate spring constant values for those particular directions. For the downward shock direction (ship moving down), the foundation flanges will be in bending and the equipment hold-down bolts and webs will be in tension.

4.7 EXAMPLE – UNI-DIRECTIONAL RESPONSE ANALYSIS OF A FOUNDATION – MULTI-MASS SYSTEM.

Foundations for Grade A, alignment-sensitive equipment such as those listed in 3.2.3.1, have, as a rule, been modeled as multi-degree of freedom systems. Analysis of multi-degree of freedom foundation systems generally require the use of computer solutions.

4.7.1 Characteristics. Multi-degree of freedom models used to analyze foundations have the following characteristics:

a. The model is three-dimensional and represents the equipment and foundation.

b. The model should minimize the complexity of the analysis i.e., sound engineering judgement should be used in the preparation of the model. It is not necessary to model the supported equipment with the same degree of refinement as is used in an equipment analysis. However, it is necessary to model the equipment such that the overall mass distribution of the equipment and its flexibility are properly represented.

4.7.2 Procedural Steps. The basic steps necessary to analyze a multi-mass system are as follows:

a. Step 1: Divide the system into N regions that adequately describe the system and calculate the mass of each; i.e., M_1, $M_2 \ldots M_n$, where:

$$\sum_{i=1}^{N} M_i = \text{Total Mass}$$

These masses represent the dynamic degrees of freedom of the system and are located at nodes in accordance with 3.4.

b. Step 2: Calculate the influence (or stiffness) coefficients for these nodes and form the influence (or stiffness) coefficient matrix.

c. Step 3: Using the method shown in Appendix A, or other suitable methods, find a number of mode shapes and natural frequencies necessary to satisfy the mode selection criteria of 3.6.3. The frequency of the highest mode calculated need not exceed 250 Hertz unless it is determined that the cumulative modal effective weight requirement of 80 percent, noted in 3.6.3, will not be satisfied at that frequency.

d. Step 4: For the first mode, mode "a", complete the following table:

MODAL COMPUTATION TABLE (MODE "a").

Mass Number, i	Mass, M_i	Mode Shape Φ_{ia}	$M_i\Phi_{ia}$	$M_i\Phi_{ia}^2$
1	M_1	Φ_{1a}	$M_1\Phi_{1a}$	$M_1\Phi_{1a}^2$
2	M_2	Φ_{2a}	$M_2\Phi_{2a}$	$M_2\Phi_{2a}^2$
-	-	-	-	-
n	M_n	Φ_{na}	$M_n\Phi_{na}$	$M_n\Phi_{na}^2$
		$\sum$	$M_i\Phi_{ia}$	$M_i\Phi_{ia}^2$

e. Step 5: Calculate the participation factor*, P_a

$$P_a = \frac{\sum_{i=1}^{N} M_i \Phi_{ia}}{\sum_{i=1}^{N} M_i \Phi_{ia}^2}$$

f. Step 6: Calculate the modal effective mass*, M_a

$$M_a = P_a\left(\sum_{i=1}^{N} M_i \Phi_{ia}\right) = \frac{\left(\sum_{i=1}^{N} M_i \Phi_{ia}\right)^2}{\sum_{i=1}^{N} M_i \Phi_{ia}^2}$$

g. Step 7: Multiply M_a by g to get the modal effective weight and divide this value by the total model weight to obtain the percent modal effective weight.

h. Step 8: Using the shock design value formulas in T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1), with the modal effective weight as W (in kips), determine the design velocity value V and the design acceleration value A.

* The definition of participation factor and modal effective mass shown herein apply only to the uni-directional models. See 4.8 for general definition of these parameters.

i. Step 9: Calculate the values of $V\omega_a$ and Ag. Determine the modal shock design value D_a to be the lesser of the two:

$$D_a = V\omega_a$$

or

$$D_a = Ag$$

but, not less than 6 g

j. Step 10: Calculate the effective static force applied at each mass:

$$F_{ia} = M_i \Phi_{ia} P_a D_a$$

k. Step 11: Apply the effective external static forces calculated in step 10 to their respective nodes and calculate the desired response (e.g., stresses, reaction forces, bending moments, deflections, etc.) by the usual methods of structural analysis of static structures.

l. Step 12: Repeat steps 4 through 11 for modes "b", "c", etc., as necessary (see 3.6.3). The values obtained in step 11 for all calculated modes shall be summed across the modes by the NRL summing method described in 3.6.5. The resultant value (combined with continuous operating stresses, if present) shall be compared to the failure criteria given in Chapter 6 of this manual.

If required, the following quantities may be determined from the information obtained above:

(1) Relative displacement between any two nodes, within a mode,

$$X_{ia} - X_{ka} = (\Phi_{ia} - \Phi_{ka}) P_a (D_a / \omega_a^2)$$

(2) Relative displacements between any node and the fixed base, within a mode,

$$X_{ia} = \Phi_{ia} P_a (D_a / \omega_a^2)$$

Relative displacements can also be summed across the modes using the NRL summing method described in 3.6.5. The NRL summing method shall not be used to sum absolute deflections across the modes unless total displacement of a point on the structure with respect to the final base is required.

4.8 DYNAMIC ANALYSIS OF A FOUNDATION – MULTI-DIRECTIONAL RESPONSE ANALYSIS.

The analytical technique for an MDR analysis is analogous to that for uni-directional analysis. The basic principles are derived from normal mode theory and are valid for a maximum of six directions of response motion at each node. The full theory, for rotations as well as translations, is considered too involved for presentation here. Most three-dimensional systems can be adequately described by translational motions alone. Therefore, the analysis procedure for three directional response motions, as given below, is applicable in most cases.

4.8.1 Modal Analysis for Multi-Direction Response. Some of the basic concepts of modal analysis for multi-direction response are:

a. Stiffness matrix:

k_{ij} = The reaction force at the i^{th} degree of freedom due to a unit deflection at the j^{th} degree of freedom, with all other degrees of freedom restrained.

$k_{ij} = k_{ji}$ for linear elastic structures,

where i and j are arbitrary degree of freedom indicators.

b. Influence coefficient vector: The influence coefficient vector {r} is a vector of direction cosines between the direction of shock input and the direction of response for each degree of freedom.

c. Participation factor:

P_a = participation factor for mode a:

$$P_a = \frac{\sum_{i=1}^{N} m_i \Phi_{ia} r_i}{\sum_{i=1}^{N} m_i \Phi_{ia}^2}$$

where: m_i = mass associated with the i^{th} degree of freedom

Φ_{ia} = mode shape for i^{th} degree of freedom in mode a

r_i = direction cosine for the i^{th} degree of freedom

d. Modal effective mass:

$$m_a = \frac{\left(\sum_{i=1}^{N} M_i \Phi_{ia} r_i\right)^2}{\sum_{i=1}^{N} M_i \Phi_{ia}^2}$$ the modal effective mass (acting in the direction of shock input) for the a^{th} mode.

4.8.2 Shock Input in One Selected Direction. For shock input in one selected direction (vertical, athwartship, or fore/aft), the basic steps for evaluating the dynamic response for a particular mode, mode a, are given below. (Steps a, b, and c are generally done once and apply to the system for all three directions of shock input. Steps d through g are repeated for each mode and for the other two directions of shock input.) The steps outlined are illustrative of the DDAM procedure, however, numerically equivalent steps may be substituted for calculation efficiency.

a. Determine the stiffness and mass matrices for the mathematical model.

b. Calculate the modal characteristics Φ_{ia} and ω_a.

c. Determine vector $\{r_i\}$, the direction cosines for each degree of freedom with respect to the direction of shock input considered.

d. Calculate the participation factor and modal effective mass as shown above.

e. Determine the design velocity value V and the design acceleration value A from T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1) using the calculated modal effective weight W_a (in kips). Calculate the values of $V\omega_a$ and Ag. Determine the modal shock design value D_a to be the lesser of the two:

$D_a = V\omega_a$

D_a = Ag but in no case shall D_a be less than 6 g

f. Calculate the effective static forces applied for each degree of freedom:

F_{ia} = force at node i in mode a:

$$F_{ia} = m_i \Phi_{ia} P_a D_a$$

g. Apply the effective static forces calculated above at their respective nodes. Since these forces F_{ia} occur simultaneously, the ensuing stress analysis will properly consider the concurrent effects of the forces in all directions. The modal displacements may be calculated directly:

$$X_{ia} = \Phi_{ia} P_a D_a / \omega_a^2$$

4.8.3 Example – Multi-Directional Response Analysis. This example is provided to demonstrate the application of DDAM for MDR analysis. Consider a simply supported structure as shown in the figure below. This model may represent a mast yardarm with mounted antennas (masses M_1, M_2, and M_3). The vertical members below the masses represent the antenna foundations. In the context of this manual an MDR analysis is defined as an analysis that uses a model which allows response degrees of freedom in all directions including directions other than the direction of input motion. Thus, under vertical shock, masses M_1, M_2, and M_3 will have lateral as well as vertical shock responses. It is obvious that under vertical shock (shock input motion at the supports in the Y direction), bending of the vertical members cannot be evaluated unless an MDR DDAM analysis is conducted. Omission of lateral degrees of freedom for each mass in the vertical mathematical model will significantly alter the results and conclusions of the analysis.

The shock inputs for an MDR model are applied independently as they are for a uni-directional model. A separate analysis is conducted for each direction of shock input.

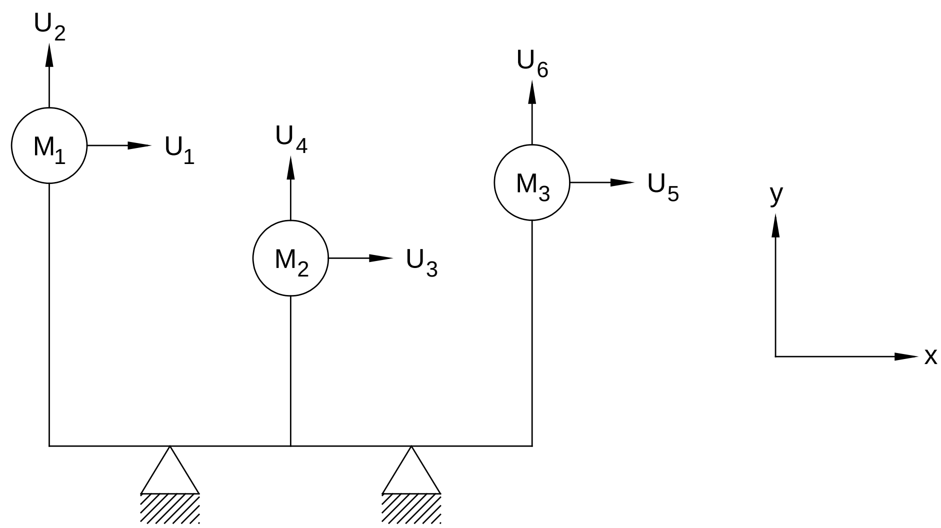


Figure 4-5. Schematic Representation of a Mathematical Model for an MDR Analysis.

a. Mass and stiffness matrices:

$$[K] = \begin{bmatrix} k_{11} & k_{12} & k_{13} & k_{14} & k_{15} & k_{16} \\ k_{21} & k_{22} & k_{23} & k_{24} & k_{25} & k_{26} \\ k_{31} & k_{32} & k_{33} & k_{34} & k_{35} & k_{36} \\ k_{41} & k_{42} & k_{43} & k_{44} & k_{45} & k_{46} \\ k_{51} & k_{52} & k_{53} & k_{54} & k_{55} & k_{56} \\ k_{61} & k_{62} & k_{63} & k_{64} & k_{65} & k_{66} \end{bmatrix} \qquad [M] = \begin{bmatrix} m_{11} & 0 & 0 & 0 & 0 & 0 \\ 0 & m_{22} & 0 & 0 & 0 & 0 \\ 0 & 0 & m_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & m_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & m_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & m_{66} \end{bmatrix}$$

Using quantities from the mass matrix above:

$M_1 = m_{11} = m_{22}$

$M_2 = m_{33} = m_{44}$

$M_3 = m_{55} = m_{66}$

b. Frequency response:

DOF	MODE / FREQ	1	2	3	4	5	6
		ω_1	ω_2	ω_3	ω_4	ω_5	ω_6
u_1		Φ_{11}	Φ_{12}	Φ_{13}	Φ_{14}	Φ_{15}	Φ_{16}
u_2		Φ_{21}	Φ_{22}	Φ_{23}	Φ_{24}	Φ_{25}	Φ_{26}
u_3		Φ_{31}	Φ_{32}	Φ_{33}	Φ_{34}	Φ_{35}	Φ_{36}
u_4		Φ_{41}	Φ_{42}	Φ_{43}	Φ_{44}	Φ_{45}	Φ_{46}
u_5		Φ_{51}	Φ_{52}	Φ_{53}	Φ_{54}	Φ_{55}	Φ_{56}
u_6		Φ_{61}	Φ_{62}	Φ_{63}	Φ_{64}	Φ_{65}	Φ_{66}

c. The influence coefficient vector {r} for shock in the Y direction is:

$$r = \begin{Bmatrix} 0 \\ 1 \\ 0 \\ 1 \\ 0 \\ 1 \end{Bmatrix}$$

d. Modal composition (shown for mode 1):

Degree of Freedom	Mass m_{ii}	Mode Shape Φ_{ia}	r	$\{\Phi\}_a^T[M]\{r\}$	$\{\Phi\}_a^T[M]\{\Phi\}_a$
u_1	M_1	Φ_{11}	0	$M_1\Phi_{11}=0$	$M_1(\Phi_{11})^2$
u_2	M_1	Φ_{21}	1	$M_2\Phi_{21}$	$M_2(\Phi_{21})^2$
u_3	M_2	Φ_{31}	0	$M_3\Phi_{31}=0$	$M_3(\Phi_{31})^2$
u_4	M_2	Φ_{41}	1	$M_4\Phi_{41}$	$M_4(\Phi_{41})^2$
u_5	M_3	Φ_{51}	0	$M_5\Phi_{51}=0$	$M_5(\Phi_{51})^2$
u_6	M_3	Φ_{61}	1	$M_6\Phi_{61}$	$M_6(\Phi_{61})^2$

For shock input in the “Y” direction, the participation factor for mode a is:

$$P_a = \frac{M_1\Phi_{2a} + M_2\Phi_{4a} + M_3\Phi_{6a}}{M_1(\Phi_{1a})^2 + M_1(\Phi_{2a})^2 + M_2(\Phi_{3a})^2 + M_2(\Phi_{4a})^2 + M_3(\Phi_{5a})^2 + M_3(\Phi_{6a})^2}$$

The modal effective mass for mode a in the direction of shock input is:

$$m_a = \frac{(M_1\Phi_{2a} + M_2\Phi_{4a} + M_3\Phi_{6a})^2}{M_1(\Phi_{1a})^2 + M_1(\Phi_{2a})^2 + M_2(\Phi_{3a})^2 + M_2(\Phi_{4a})^2 + M_3(\Phi_{5a})^2 + M_3(\Phi_{6a})^2}$$

e. Shock design value:

The shock design values to be applied in each mode are obtained from T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1). These values are a function of modal effective weight (in kips) and the modal frequency in radians. In no case shall the shock design value be less than 6 g.

f. Effective static forces:

The effective static forces in mode a for each degree of freedom are:

(1) Mass 1:

$$F_{1a} = M_1\Phi_{1a}P_aV_a\omega_a \text{ or } M_1\Phi_{1a}P_aA_ag$$

$$F_{2a} = M_1\Phi_{2a}P_aV_a\omega_a \text{ or } M_1\Phi_{2a}P_aA_ag$$

(2) Mass 2:

$$F_{3a} = M_2\Phi_{3a}P_aV_a\omega_a \text{ or } M_2\Phi_{3a}P_aA_ag$$

$$F_{4a} = M_2\Phi_{4a}P_aV_a\omega_a \text{ or } M_2\Phi_{4a}P_aA_ag$$

(3) Mass 3:

$$F_{5a} = M_3\Phi_{5a}P_aV_a\omega_a \text{ or } M_3\Phi_{5a}P_aA_ag$$

$$F_{6a} = M_3\Phi_{6a}P_aV_a\omega_a \text{ or } M_3\Phi_{6a}P_aA_ag$$

g. Stress analysis:

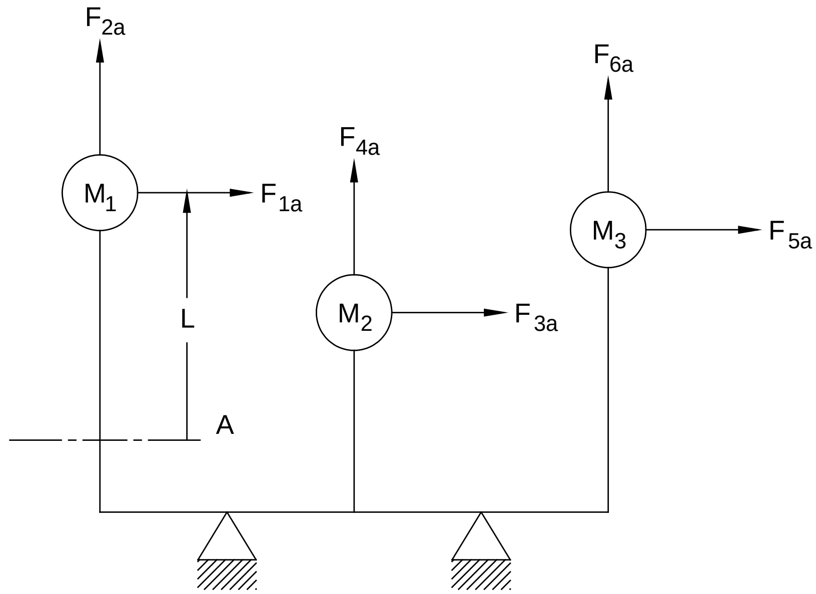


Figure 4-6. Force Schematic for an MDR Analysis.

Stress at section A-A:

$$\sigma_a = \frac{Mc}{I} + \frac{F}{A} = \frac{(F_{1a}L)c}{I} + \frac{F_{2a}}{A}$$

where I, A, and c are the member section properties:

I = Section Moment of Inertia

A = Section Area

c = Distance from the neutral axis to the fiber carrying the greatest stress

4.9 FINITE ELEMENT APPLICATION OF DDAM.

To illustrate the finite element modeling of a complicated structure, consider the example shown in Figure 4-7 below. Appendix E provides details for the format and content of a finite element mathematical model and dynamic analysis. The model used for this example is a typical finite element representation for a rack type foundation. Each equipment mounted in the rack is modeled with its weight concentrated at its center of gravity. The weight of the rack structure, associated cooling water piping, cabling, mounting hardware, and other distributed weight is included in the model. The flexibility of the equipment should be included if known. Otherwise, the equipment can be considered rigid bodies.

This model is used to design the foundation structure and can be used to check the shock loading in the equipment hold-down bolting. The foundation model and analysis is not used to evaluate the equipment itself since the equipment is normally qualified for shock by testing in accordance with MIL-DTL-901. If the equipment is a Grade B item, its shock adequacy can be demonstrated by analysis in lieu of testing. The results of application to the equipment and equipment appendages of acceleration values derived from the DDAM analysis of this model can be evaluated in accordance with 6.4 to determine whether the item meets Grade B shock requirements.

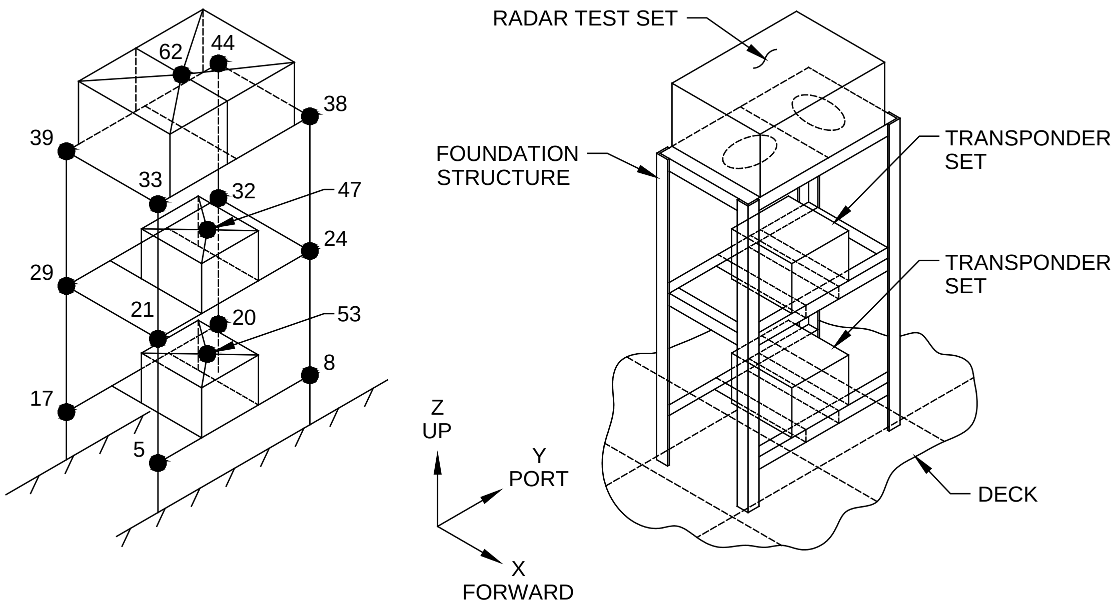


Figure 4-7. Schematic Representation of a Multi-Mass Finite Element Foundation Mathematical Model.

CHAPTER 5
DDAM OF GRADE B ITEMS

5.1 INTRODUCTION.

Ship specifications may permit shock qualification of Grade B items by dynamic analysis (in lieu of shock testing). Permission to shock qualify Grade B equipment via analysis is subject to demonstrating to the Technical Authority that potential hazards can be adequately assessed via analysis (i.e., equipment contains structural failure modes of limited complexity, such that dynamic analysis is capable of demonstrating compliance with Grade B acceptance criteria).

In cases where the dynamic model of a Grade B item would be relatively complex or where analysis is incapable of evaluating the equipment's operation, potential hazards and/or failure modes, the item shall be shock qualified by shock testing instead of by dynamic analysis. Examples of hazards requiring assessment that may result in the exclusion of dynamic analysis to qualify the item include, but are not limited to: equipment causing impairment or malfunction to Grade A equipment, equipment causing an electrical shock hazard or fire hazard, etc.

Dynamic analysis criteria contained in Chapter 3 apply to analysis of Grade B items, with the exception that low frequency components need not be modeled as separate masses unless they are items which can cause a hazard. For example, if an item could cause a shock hazard by coming adrift external to the equipment, it should be considered as a separate mass.

Generally elastic-plastic shock design values apply to dynamic analysis of non-alignment critical items. However, hold-down means must be designed based on elastic inputs. Elastic shock design values shall also be used for Grade B equipment where a hazard can arise by overstressing a component which releases a toxic material from a bolted joint or where a hazard can arise as a result of excessive deformation or fracture of a brittle container. Allowable stress criteria are contained in Chapter 6.

5.2 EXAMPLE PROCEDURES FOR DYNAMIC ANALYSIS OF GRADE B ITEMS.

The following two examples illustrate procedures for dynamic analysis of Grade B items.

5.2.1 Example 1. Consider the deck mounted, Grade B, equipment shown in Figure 5-1. The analysis for this item is required to show that it will not come adrift under shock. This is accomplished by ensuring that failure will not occur in the equipment legs or the hold-down bolts under shock loading. (Only vertical shock is shown in this example.)

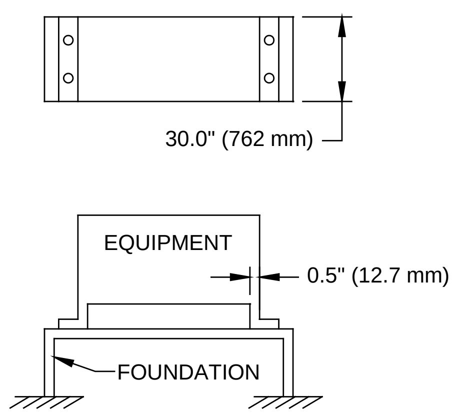


Figure 5-1. Single Degree of Freedom Foundation Model for Dynamic Analysis of Grade B Item.

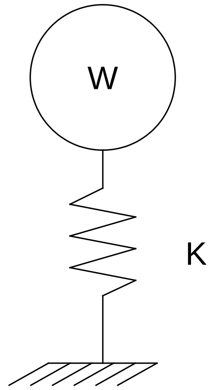


Figure 5-2. Schematic Representation of a Single Degree of Freedom Foundation.

The system can be analyzed as a single degree of freedom system as shown in Figure 5-2.

$$W = \text{equipment weight} + \frac{\textit{foundation weight}}{2}$$

$$= 3{,}000 + \frac{450}{2} \quad \left(13{,}345 + \frac{2001.7}{2}\right)$$

$$= 3{,}225 \text{ lb} \qquad (14.346 \times 10^6 \text{ N})$$

Assume K = 1.92 x 10^6 lb/in (336.24 x 10^6 N/m) and assume the angular frequency of the system is derived as follows:

$$\omega = \sqrt{\frac{K\,g}{W}}$$

$$= \sqrt{\frac{1.92\,x10^6(386)}{3{,}225}} \qquad \left(\sqrt{\frac{336.2\,x10^6(9.81)}{14{,}346}}\right)$$

$$= 479\ \frac{rad}{\text{sec}}$$

From T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1), for a system with a modal effective weight of 3.225 kips vertical shock loading, deck mounted, and elastic-plastic design, the shock design values are:

V = 24.8 in/sec (0.63 m/sec)

A = 49.0 g

Therefore,

$$D = \frac{V\omega}{g}$$

$$= \frac{24.8(479)}{386} \quad \left(\frac{0.63(479)}{9.81}\right)$$

$$= 30.8 \text{ g}$$

$$D = A = 49.0 \text{ g}$$

$$D = 6 \text{ g minimum}$$

$$\text{Use } D = 30.8 \text{ g}$$

To analyze the stress in each foot, a force of

$$F = (3{,}000)(30.8) = 92{,}400 \text{ lb} \qquad (13{,}346(30.8) = 411.1 \text{ kN})$$

would be divided between the two legs. It will be noted that 3,000 lb (13,345 N) was used instead of 3,225 lb (14,346 N) to calculate the force. This was done because only the weight of the equipment effectively acts on the legs (and bolts).

$$\sigma_{foot} = \frac{F}{A}$$

$$= \frac{92{,}400}{2(0.5)(30)} \qquad \left(\frac{4.111x10^5}{2(0.0127)(0.762)}\right)$$

$$= 3{,}080 \text{ psi (axial)} \qquad (21.24 \text{ x } 10^6 \text{ N/m}^2)$$

In the interest of expediency for this problem, bending stresses in the legs will not be examined. To stress analyze the four hold-down bolts, the force of 92,400 lb (4.097 x 10^6 N) is not appropriate because all bolts, dowels, pins, and similar hold-down means must be designed for shock on the basis of elastic inputs.

Thus, for elastic inputs, the shock design values for this system would be:

$$V = 49.6 \text{ in/sec } (1.26 \text{ m/sec})$$

$$A = 49.0 \text{ g}$$

and $D = \frac{V\omega}{g}$

$$= \frac{49.6(479)}{386} \qquad \left(\frac{1.26(479)}{9.81}\right)$$

$$= 61.6 \text{ g}$$

$$D = A = 49.0 \text{ g}$$

To determine bolt stresses, the shock force is

$$F = 3{,}000\ (49) = 147{,}000 \text{ lb}$$

$$(13.346(49) = 654.0 \text{ kN})$$

and

$$\sigma_{bolt} = \frac{147{,}000\ lbs}{4A_{bolt}} \qquad \left(\frac{654.0\ kN}{4A_{bolt}}\right)$$

The stress values determined above for the legs and the bolts shall be compared to the allowable stress criteria in Chapter 6 of this manual to determine if the design criteria is met.

5.2.2 Example 2. Consider that the equipment shown in Figure 5-1 has a 200 lb (889.6 N) motor attached to it as shown in Figure 5-3.

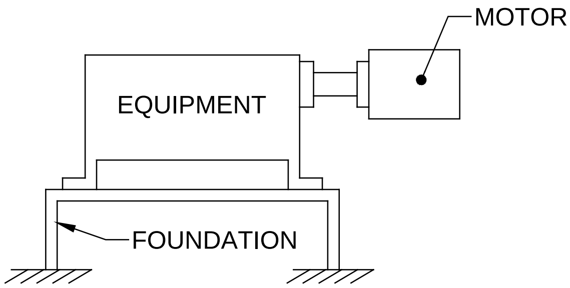
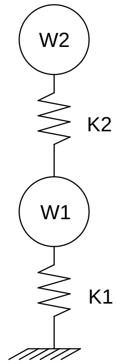


Figure 5-3. Equipment/Foundation Configuration with a Cantilevered Motor.

To determine whether this Grade B system will create a hazard by coming adrift under shock loading, the analyst must check that neither the legs, hold-down bolts, nor the motor attachment will fail under shock loading because any one of them would cause the equipment or motor to come adrift. To analyze this system, a two mass model such as the one shown in Figure 5-4 is required.



Where:

W1 = weight of equipment + ½ foundation weight

W2 = weight of motor, portion of motor shaft, and attachment

K1 = foundation spring constant

K2 = motor attachment spring constant

Figure 5-4. Schematic Representation of a Two Degree of Freedom System.

The method used for the dynamic analysis of a two mass system has been discussed in 4.7 and 4.8. The analyst shall determine whether failure of the motor mount bolting will occur by using the forces developed in Spring 2 (K2). The feet and bolt stresses are determined from the forces developed in Spring 1 (K1). In keeping with criteria presented in Chapter 3, elastic shock design criteria would apply to the design of the hold-down means which secure the equipment to the foundation, but not to the bolting which secures the motor to the equipment.

CHAPTER 6
ALLOWABLE STRESS CRITERIA

6.1 GENERAL CRITERIA.

Each principal direction of shock loading (vertical, athwartship, and fore and aft) shall be considered separately. Continuous operating stresses (as defined in Chapter 3) shall be added to calculated shock stresses. Allowable stress criteria presented herein shall be calculated based on the Von Mises Failure Theory. Comparison of combined shock and operating stresses to allowable stresses will generally determine design acceptability. The allowable stress described in 6.2 through 6.5 apply to Navy standard metal materials (e.g., steel, aluminum, K-Monel, etc.). Allowable stresses for other non-standard materials (e.g., GRP, composite, epoxy chock, titanium, wood, etc.) shall be provided by the contract specifications. If the shipbuilding specifications do not address the material design properties of these materials, the contractor shall propose material properties for Navy approval. Other failure criteria, as discussed in 3.2.4 and specified in the approved model report, shall also be considered. In addition, it shall also be assured that column buckling for those items designed to elastic shock design values will not occur and the deflection of foundations must not lead to overloading of flexible couplings or other displacement-critical components. Figure 6-1 is a summary table for the allowable stress criteria reflected in this manual for Grade A and B systems.

Design stresses are categorized as general or local, and as membrane or membrane plus bending. Definitions of these categories are provided below with examples for their application provided in Appendix F. (Note: stresses derived from one-dimensional beam elements are limited solely by the general stress categories.) In finite element analyses, local high stresses, analogous to stress concentrations, may be reported. Examples of regions of local high stress include inadequate mesh refinement in areas of complex stress gradients, loading and geometry, or modeling distributed connections where the results are in terms of a point load rather than the true distributed load. In these cases, engineering judgement must be applied to the results to properly determine the allowable stress requirements.

6.1.1 General Stress. General stress is the average (normal and/or shear) stress resulting from global deformation of the structure under consideration.

6.1.2 Local Stress. Local stress (normal and/or shear) occurs in regions of load application or structural discontinuity. Stresses which exceed the general stress allowables may be considered local if the area over which the stress exceeds the general stress allowable does not exceed 10 percent of the effective area. Definitions of the effective area are shown in Table F-1 of Appendix F. The 10 percent limit can be waived if it can be demonstrated that the load carrying capacity of the structure is adequate.

6.1.3 General Membrane Stress. General membrane stress is calculated for the average normal and/or shear stress across the thickness or depth of a section under evaluation. For one-dimensional beam elements, this includes mean axial, shear, and torsional shear stresses. The mean axial stress is the normal stress averaged over the effective cross-section under evaluation. It should be noted that for Grade A, elastic, case 2; Grade A, elastic-plastic; and Grade B elastic, the membrane stresses (normal, shear), defined as the average stress components through the load carrying section, must be separated for the total stress prior to Von Mises stress combination (see step i of 6.1.8.2).

6.1.4 General Membrane Plus Bending Stress. General membrane plus bending stress is calculated from stresses at the outermost fibers of the subject section. The bending stress is the variable component of the stress (normal and/or shear) across the thickness or depth of a section, but excludes peak stresses caused by geometric discontinuities. The variation may or may not be linear across the thickness or depth of a section. The depth of a section may be that of a composite section made up of effective plate elements of a finite element model or the thickness of a single plate element. General membrane plus bending stress includes membrane stress categorized as local in the evaluation of the adequacy of the cross-section. The consideration of local membrane stresses may result in lower magnitudes of general bending stresses being considered acceptable.

6.1.5 Local Membrane Stress. Local membrane stress is calculated from the total membrane stress produced by mechanical loads, including the effects of constraint of adjacent material or self-constraint of the structure. It can occur in regions of gross or local structural discontinuities and at locations of intersecting structural members. Stress concentrations as shown in F.14 are not limited.

6.1.6 Local Membrane Plus Bending Stress. Local membrane plus bending stress is calculated from the total stress evaluated at the outermost fibers of the subject section produced by mechanical loads including self-limiting stresses developed by the constraint of adjacent material or self-constraint of the structure. It can occur in regions of gross or local structural discontinuities and at locations of intersecting structural members. Stress concentrations as shown in F.14 are not limited.

6.1.7 Adjacent Local Stresses Regions. Table F-2 of Appendix F provides examples of adjacent local stresses regions. Adjacent areas of local stress due to the introduction of concentrated loads may not overlap. The centers of adjacent local stressed regions cannot be closer than 2.5 times the average of the dimensions of the two locally stressed areas. The length of each locally stressed region shall be based on the limit of local stress exceeding general stress limits, and shall be measured along a line of action between the center of each pair of adjacent locally stressed areas.

6.1.8 Stress Evaluation and Classification. A procedural outline is presented for finite element stress evaluation and classification to help ensure consistent application of the criteria to structural evaluations. The outline is specific to thin plate/shell stress evaluation, which represents a significant portion of the structural evaluations. Considerations for one-dimensional beam elements are also presented.

GRADE	ALLOWABLE DEFORMATION	SHOCK INPUT	TYPE OF STRESS	ALLOWABLE STRESS		SECT
				GENERAL	LOCAL	
GRADE A	CASE 1. ALIGNMENT SENSITIVE, NO PERMANENT DEFORMATION ALLOWED	ELASTIC	$\sigma_{MEMBRANE}$	1.0 σ_{YIELD}	1.0 σ_{YIELD}	6.2.1
			$\sigma_{MEMBRANE} + \sigma_{BENDING}$	1.0 σ_{YIELD}	1.0 σ_{YIELD}	
			σ_{COLUMN}	BUCKLING	BUCKLING	
			$\sigma_{BEARING}$	1.6 σ_{YIELD}	1.6 σ_{YIELD}	
	CASE 2. ALIGNMENT SENSITIVE, SLIGHT PERMANENT DEFORMATION ALLOWED	ELASTIC	$\sigma_{MEMBRANE}$	1.0 σ_{YIELD}	1.5 σ_{YIELD}	6.2.2
			$\sigma_{MEMBRANE} + \sigma_{BENDING}$	1.0 σ_{EFF}	2.0 σ_{EFF}	
			σ_{COLUMN}	BUCKLING	BUCKLING	
			$\sigma_{BEARING}$	1.6 σ_{YIELD}	1.6 σ_{YIELD}	
	NON-ALIGNMENT SENSITIVE, EQUIPMENT AND FOUNDATIONS SLIGHT PERMANENT DEFORMATION ALLOWED	ELASTIC PLASTIC	$\sigma_{MEMBRANE}$	1.0 σ_{YIELD}	1.0 σ_{YIELD}	6.3.1
			$\sigma_{MEMBRANE} + \sigma_{BENDING}$	1.0 σ_{YIELD}	1.0 σ_{YIELD}	
			σ_{COLUMN}	NO LIMIT	NO LIMIT	
			$\sigma_{BEARING}$	NO LIMIT	NO LIMIT	
	NO PERM DEFORM ALLOWED; BOLTS	ELASTIC	$\sigma_{MEMBRANE}$	1.0 σ_{YIELD}	1.0 σ_{YIELD}	6.5
	NON-ALIGNMENT SENSITIVE, EQUIPMENT AND FOUNDATIONS CONSIDERABLE PERMANENT DEFORMATION ALLOWED	ELASTIC PLASTIC	$\sigma_{MEMBRANE}$	1.0 σ_{YIELD}	1.0 σ_{YIELD}	6.3.2
			$\sigma_{MEMBRANE} + \sigma_{BENDING}$	2.0 σ_{YIELD}	2.0 σ_{YIELD}	
			σ_{COLUMN}	NO LIMIT	NO LIMIT	
			$\sigma_{BEARING}$	NO LIMIT	NO LIMIT	
GRADE B	NO PERMANENT DEFORMATION ALLOWED	ELASTIC	$\sigma_{MEMBRANE}$	1.0 σ_{YIELD}	1.5 σ_{YIELD}	6.2.2
			$\sigma_{MEMBRANE} + \sigma_{BENDING}$	1.0 σ_{EFF}	2.0 σ_{EFF}	
			σ_{COLUMN}	BUCKLING	BUCKLING	
			$\sigma_{BEARING}$	1.6 σ_{YIELD}	1.6 σ_{YIELD}	
	NON-ALIGNMENT SENSITIVE, PERMANENT DEFORMATION ALLOWED (EQUIPMENT AND FOUNDATIONS)	ELASTIC PLASTIC	$\sigma_{MEMBRANE}$	1.0 σ_{YIELD}	NO LIMIT	6.4
			$\sigma_{MEMBRANE} + \sigma_{BENDING}$	NO LIMIT	NO LIMIT	
			σ_{COLUMN}	NO LIMIT	NO LIMIT	
			$\sigma_{BEARING}$	NO LIMIT	NO LIMIT	
	NO PERM DEFORM ALLOWED; BOLTS	ELASTIC	$\sigma_{MEMBRANE}$	1.0 σ_{YIELD}	1.0 σ_{YIELD}	6.5

Figure 6-1. Allowable Stress Criteria and Application Design Levels.

6.1.8.1 Stress Evaluation Procedure for Solid Elements. Compute finite element stress components of the solid element. Stresses may be evaluated at integration points of the element or extrapolated to nodal points for joint averaging. Stresses may be computed in the principal coordinate system or the local coordinate system as long as failure criteria definitions are consistent with the specific coordinate system used. Significant differences (an order of magnitude) in unaveraged nodal stresses from adjacent elements indicate a stress concentration or an inadequate mesh size. Joint averaging should not be performed at thickness discontinuities at material modulus changes, or at geometric discontinuities such as the intersection of two plates.

a. Compute the finite element stress components of the solid element within each mode.

b. Compute the mean normal stress of the solid element within each mode (see 3.6.4).

c. Compute the maximum principal shear stress of the solid element within each mode.

d. Compute the Von Mises (bending) stress of the solid element within each mode.

e. Compute NRL summed total stress as defined in 3.6.5.

f. Compare the NRL summed mean axial stress of the solid element to the membrane allowable stress.

g. Compare the NRL summed maximum principal shear stress of the solid element to 60 percent of the material yield stress.

h. Compare the NRL summed Von Mises stress to the membrane plus bending allowable stress.

6.1.8.2 Stress Evaluation Procedure for Thin Plate/Shell Elements. Compute finite element stress components at the bottom, middle, and top surfaces of the plate element. Stresses may be evaluated at integration points of the element or extrapolated to nodal points for joint averaging. Significant differences in unaveraged nodal stresses from adjacent elements indicate a stress concentration or an inadequate mesh size. Joint averaging shall not be performed at thickness discontinuities, at material modulus changes, or at geometric discontinuities such as the intersection of two plates. Stress components should be oriented such that predominant stress states (e.g., beam bending, axial stress, hoop stress, radial stress, etc.) can be evaluated.

a. Compute the stress within each mode based on the Von Mises Failure Theory as defined in 3.6.4. Membrane stresses are computed from mid-surface stress components. Membrane plus bending stresses are computed at extreme fibers of the plate.

b. Compute NRL summed total stress as defined in 3.6.5.

c. Review stresses computed in step b (i.e., through use of fringe/contour stress plots) noting which intensities exceed the established stress allowable (general membrane) for the particular shock grade of the structure. General membrane stress limits apply to mean axial and shear stress states over the member cross-sectional area.

d. In cases where the general membrane allowable is exceeded, further investigation of component level stresses in each mode will be required to classify the stress component as general membrane, general membrane plus bending, local membrane, or local membrane plus bending. A deformed plot for each mode of the subject structure can aid in classifying stresses.

e. To classify an outer fiber stress as general membrane plus bending, a variable component of stress through the thickness or depth of the section must be present. If general bending of the structural member is present, use of the general membrane plus bending limit is permitted.

f. To classify an outer fiber stress as local membrane plus bending, the stress must exist at a location of load introduction or structural discontinuity. The bending stress variation is predominantly through the thickness of a plate and limited to 10 percent of the effective area. Average shear stresses derived from plate punch-through and plate tear-out calculations shall be limited to the general membrane allowables.

g. The stress at a load introduction or structural discontinuity identified in step f may exceed the local membrane plus bending stress limit if it is confined to less than 5 percent of an effective area. No limits are established within this area. Average shear stresses derived from plate punch-through and plate tear-out calculations shall be limited to the general membrane allowables.

h. Classification of local membrane stresses is similar to classification of local membrane plus bending stresses except that local membrane stresses are evaluated at the plate mid-surface.

i. For those elements classified with general membrane plus bending or local stress, re-evaluate the Von Mises stress as follows. Re-compute the combined stress within each mode using component level stresses adjusted by the factor of the general membrane stress allowable over the stress allowable applicable to each respective component stress. Re-compute the NRL summation of stresses. Compare the NRL summation of combined stresses to the membrane stress allowables.

j. Failure to meet the specified allowables is cause for structural modification and re-analysis in accordance with 3.6.7 or in cases of local stresses further demonstration that the load carrying capacity of the structure is adequate.

Note: General bending of a cross-section may result in membrane stresses at the element level (i.e., for I-beam in strong axis bending, flanges will be predominantly membrane). It is not the intent of these criteria to limit element level membrane stresses to membrane allowables. However, such limitation would be conservative.

6.1.8.3 Stress Evaluation Procedure for Beam Elements. Stress evaluation for one-dimensional beam element models is limited to the general membrane and general membrane plus bending stress categories. Member mean axial and shear stresses are limited to the membrane stress allowables. Stresses evaluated at the extreme fibers of a beam cross-section that includes bending stresses are limited to general membrane plus bending stress category. It should be noted that the maximum bending stress and the maximum shear stress do not occur at the same location on the cross-section of a beam. Many finite element analysis post processors have programs available to evaluate Von Mises stresses on a complex beam cross-section, if available. If not, transverse shear distributions may be averaged for use in Von Mises stress calculations.

6.2 ALLOWABLE DESIGN STRESSES FOR GRADE A AND GRADE B ITEMS DESIGNED TO SUIT ELASTIC SHOCK DESIGN VALUES.

6.2.1 CASE 1. Where deflection is critical, combined operating and shock stresses shall not exceed the material static yield strength (0.2 percent offset).

6.2.2 CASE 2. Where slight permanent deformation of a cross-section can be tolerated, general membrane stresses (average normal and/or shear stress) are limited to the material static yield stress. The criterion of failure for general membrane plus bending stresses is the effective yield strength of the material. This effective yield strength is defined by:

$$\sigma_{EFF} = \sigma_y + f(\sigma_u - \sigma_y)$$

In this equation σ_y is the 0.2 percent offset yield strength, elastic limit, or other accepted definition of material yield strength.

σ_u is the conventional definition of material ultimate strength. All strengths are the values at the expected operating temperature.

The symbol f represents a factor which takes account of the efficiency with which the material in the member being analyzed is utilized. Examples of f are given below. The efficiency is computed by comparing the load required to just initiate yielding of the member with the load required to have the member completely yielded. In this computation it is assumed that the stress-strain curve of the material is bi-linear, with no strain hardening. The factor f (the efficiency minus one) is thus dependent on the kind of loading (i.e., tension, bending, etc.) and on the cross-section of the member. For example, the factor is zero for any member in pure tension and 0.5 for a rectangular section in pure bending.

In general, brittle materials, as defined in 6.10, may not be used. However, where exceptions are granted the following applies: for any brittle material (one which has less than 10 percent elongation before fracture in a tension test) the factor f is always zero. This is often true for ultra-high strength steels and cast material (steel or aluminum). The factor f must be taken as zero for any application where a slight plastic set cannot be permitted.

The value of the factor f is taken from limit design theory, in which the existence of a “plastic hinge” is postulated. The plastic hinge occurs when the member’s cross-section is fully yielded, as described above, in bending. Limit design theory may be used to define allowable component stresses under shock loading provided that the operability of any Grade A equipment is not compromised by the permanent distortion associated with yielding. For example, limit design theory permits the use of multiple plastic hinges under certain limited conditions.

Sample factors f:

Consider a rectangular bar subject to pure bending. The ratio of the fully plastic moment obtained by limit analysis to the bending moment at yield is well known to be 1.50.

So, f = 1.5 – 1 = 0.5 and the allowable stress is:

$$\sigma = \sigma_y + 0.5(\sigma_u - \sigma_y)$$

For a typical I section:

f = A/(6 + 2A)

$$\text{Where A} = \frac{(web\ width)(depth\ of\ \sec tion)}{2(\ flange\ width)(\ flange\ thickness)}$$

For a solid shaft in bending, f = 0.7

For a hollow shaft in bending, f = 0.913 – 0.638(R^1/R) where R^1/R is the ratio of the inner to the outer radius and R^1/R is equal to or greater than 0.6.

If bending is combined with torsion, shear, tension, or compression, then the analyst should compute the ratio of the maximum load to the yield load, and subtract one, to obtain the factor f.

For CASE 2, local stresses have higher limits than general stresses. The local membrane stress limits are 1.5 times the general membrane stress allowables. The local membrane plus bending stress limits are twice the general membrane plus bending stress allowables.

6.2.3 CASE 1 and CASE 2. For CASE 1 and CASE 2, combined continuous operating and shock loads shall not exceed allowable column loads. Allowable bearing stresses are 160 percent of the material static yield strength.

6.2.4 Special Design Criteria. Special design criteria must be considered in the case of equipment of foundation structures fabricated from aluminum or incorporating bimetallic (steel to aluminum) elements. Tabulated nominal yield stresses from contract specifications for welded aluminum alloys should be used to determine allowable design stresses. Manufacturers' specified yield strength should be used as the basis for shock design evaluations of bimetallic elements. Consideration must be given in such evaluations to the increased width of the elements in comparison to the thicknesses of the steel or aluminum structural members adjacent to the bimetallic elements. In general, the design of the bimetallic elements should be such that their strength in shock is greater than that of adjacent structural members.

6.3 ALLOWABLE DESIGN STRESSES FOR GRADE A ITEMS DESIGNED TO SUIT ELASTIC-PLASTIC SHOCK DESIGN VALUES.

6.3.1 Limiting Permanent Deflection. In cases where it is necessary to limit permanent deflection to approximately twice the maximum elastic deflection at yield, the calculated stresses (from elastic-plastic analysis) shall not exceed the material static yield strength (0.2 percent offset). The limiting elastic-plastic deflection used for evaluation is twice the deflection that occurs at yield. Where deflections are critical, elastic-plastic analysis cannot be used.

6.3.2 Where Considerable Plastic Bending Can be Tolerated. In cases where considerable plastic bending can be tolerated (as is usually the case with foundations designed to suit elastic-plastic shock design values), membrane plus bending stresses not exceeding 200 percent of the material static yield strength will be considered acceptable. Membrane stresses shall not exceed the material static yield strength.

a. Where 200 percent allowable stress criteria apply, continuous operating stresses (if present) shall be doubled before combining with shock stresses.

b. Combined stresses, calculated as described in 3.6.4 and 3.6.6, shall not exceed the material static yield strength. (Calculated bending stresses subject to 200 percent allowable stress criteria shall be halved before inserting into the combined stress formula.)

6.3.3 Areas of Foundations or Equipment in Way of Hold-Down Bolts. Allowable stress criteria for areas of foundations or equipment in way of hold-down bolts are the same as for other areas of the foundation and equipment. However, average shear stresses derived from plate punch-through and plate tear-out calculations shall be limited to general membrane stress allowables. Higher loadings resulting (in some cases) from special criteria applied for purposes of hold-down bolt design are applicable solely to hold-down bolting and shall not be transferred for design purposes to foundations or equipment.

6.3.4 Column Buckling and Bearing Stresses. Column buckling and bearing stresses need not be considered.

6.4 ALLOWABLE DESIGN STRESSES FOR GRADE B ITEMS DESIGNED TO SUIT ELASTIC-PLASTIC SHOCK DESIGN VALUES.

Allowable design stresses for Grade B items are the same as those which apply to Grade A items, except that bending stresses need not be considered in cases where it is evident that plastic bending of the members in question will not lead to violation of Grade B criteria. There are no limits placed on local stresses. In cases where the above cannot be assured, the allowable stress criteria described previously for Grade A items shall apply.

6.5 ALLOWABLE BOLT STRESSES.

For bolts, where MIL-DTL-1222 applies, the elastic proof stress may be considered as the yield stress. For bolts fabricated from materials other than the materials included in MIL-DTL-1222, the material static yield strength is the allowable stress. See 3.2.3.4 of this manual for related criteria. If not shock qualified with the equipment, fasteners used as hold-down devices under shock loading shall be designed for axial and shear loads so that the stress measure does not exceed the static yield strength of the material. The Von Mises Failure Theory shall be used to combine the normal and shear stresses. Typically, fastener bending stresses are not considered under shock loading. However, where consideration for fastener bending is required, the maximum value of stress measure at the periphery of the fastener resulting from direct tension, shear, and bending, but excluding stress concentration, shall not exceed the static yield strength.

6.6 ALLOWABLE STRESSES FOR WIRE ROPE.

a. For Grade A systems in which no permanent deformation can be tolerated, 60 percent of the specified nominal breaking strength used in conjunction with elastic inputs shall be the basis for shock design of wire rope.

b. For Grade A and B systems in which permanent deformation can be tolerated, 75 percent of the breaking strength in conjunction with elastic inputs shall be the basis for the design of wire rope. Elastic-plastic inputs will not be used in conjunction with the design of wire rope.

c. Reduction of effective breaking strength due to wear, abrasion, lubrication, corrosion, etc. are included in the determination of the preceding values. The fact that wire rope does not possess the same degree of energy absorption (beyond the elastic limit) as a solid steel bar is also included in the 60 percent and 75 percent values noted above.

6.7 ALLOWABLE STRESS FOR NON-METALLIC MATERIAL.

For material where the creep strength is low in relation to the yield strength and where pre-load is an important factor in shock design, the allowable stress for joint design shall be creep strength rather than yield.

6.8 SPECIAL STRESS CRITERIA FOR FOUNDATIONS.

In order to minimize weight, maximum shock stresses on foundation members whose size is governed by shock shall exceed 75 percent (but not 100 percent) of allowable tensile, compressive, or shear stresses in at least one primary member for all foundations supporting Grade A and B machinery and equipment systems weighing more than 125 pounds. A primary member is any main structural supporting member. Foundations for machinery and equipment systems that weigh less than 125 pounds are not covered by this requirement. Shock design values to be used for foundation dynamic analysis are specified in T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1). Allowable stresses for foundations designed by Method 1 (see Chapter 4) are the same that apply to foundations which are designed to suit elastic-plastic shock design values.

6.9 SPECIAL CRITERIA FOR PIPING CONNECTIONS.

When determining the stress in nozzles due to restraint of attached piping, maximum shock motion of mounts shall be considered or the nozzles shall be designed to withstand the fully plastic moment of the attached piping.

6.10 DUCTILITY.

In developing the allowable stress criteria presented in this chapter, it was assumed that the material under consideration has adequate ductility (expressed, for example, as percent elongation measured in a tensile test). Adequate ductility means that the material is not subject to a brittle fracture failure, but will yield plastically before fracturing. Many types of cast materials do not exhibit adequate ductility and thus cannot be analyzed with the criteria contained herein. Elements with less than 10 percent ductility shall not be used in structural applications which are intended to withstand shock loading.

6.11 SPECIAL CRITERIA FOR DESIGN OF HOLD-DOWN BOLTS.

When a bolted joint is loaded in tension (pre-load), shock loads do not directly increase the stress in the bolt, but decrease the clamping force between the bolt flange and the foundation. If the bolt load exceeds the clamping force, the flanges will separate and the bolts will begin to stretch. Acceptability criteria are exceeded when the load exceeds the yield strength or proof load of the bolt. The adequacy of the joint in a quasi-static condition (when the load is gradually applied) depends more on the bolt material strength than the tightness of the joint. Under dynamic loads, however, the stiffness of the joint decreases radically when the flanges separate and the system goes through a part of its cycle at a reduced frequency, with correspondingly increased deflection, until the gap closes with associated hammering of the joined parts. The initial tightness of the bolted joint, therefore, is of vital importance for system shock resistance since this hammering may be a more significant damage mechanism for the equipment than direct acceleration associated with the shock motion. In shock design calculations a bolted connection may not be adequate if the pre-stress is exceeded regardless of the bolt strength to prevent joint separation.

Bolts subjected to shock loading shall be tightened to a preload greater than the shock design force to eliminate chatter, but no greater than ⅔ of their yield strength to avoid hydrogen embrittlement conditions. Significant variation from this value may violate the fundamental assumptions of bolted joint behavior.

To prevent separation of the equipment flange and its foundation, shipbuilding specifications require that threaded fasteners, which are used to hold down machinery and equipment to sub-bases and foundations, shall be of the self-locking type. In connection with this requirement, the pre-load torque necessary to achieve the desired clamping force for hold-down bolts of Grade A machinery and equipment must be determined in the associated foundation shock design calculations and specified on the applicable installation drawings. See Appendix G for sample bolting calculations.

6.12 WETTED SURFACE DYNAMIC ANALYSIS.

A wetted surface dynamic analysis is performed on equipment that is mounted on surface ships external to the ship's shell and below the waterline, or in a normally filled tank adjacent to the shell. Equipment mounted in this fashion is subject to the direct impingement wave from the blast as well as the dynamic inertial effects through its attachment to the ship. All of these design loads must be considered. Because the timing of these two phenomenon are out of phase, each phase of this analysis will be considered and evaluated separately:

6.12.1 Phase I. A direct impingement wave front must be calculated from the design geometry as specified by the Technical Authority consistent with the shock factors contained in T9070-AJ-DPC-010/(C) 072-1. A transient analysis of the system using this pressure time history or a static analysis based on stagnation pressure is then performed to evaluate the stresses and displacements of the early time direct response.

a. For air-backed structures: first, the local plating stresses resulting from the pressure wave must be evaluated. Second, the potential crushing effects of the shock pressure wave must be evaluated on the structure. Finally, the structure must be evaluated for the flow effects of the pressure wave using the stagnation pressure.

b. For fluid-backed structures (i.e., solid plate or a tube filled with fluid): the local plating stresses and crushing effects can be ignored since the amount of reflection at the fluid structure boundary will be small and can be neglected. Only the flow effects of the pressure wave using the stagnation pressure need to be considered.

6.12.2 Phase II. A DDAM analysis is performed including the effects of added fluid mass, drag, and boundary layer effects. This may be accomplished by the use of "fluid" elements which include the effects of the fluid-structure interaction. It is not recommended to just incorporate the added mass as a lumped mass on the diagonal of the mass matrix. Although adding mass as a lumped mass on the diagonal will tend to reduce the frequencies, it will not account for the reduction in modal amplitudes which will tend to reduce the stress in the system. Adding mass as a lumped mass on the diagonal will be conservative and may not produce weight optimized designs.

6.13 SPECIAL CRITERIA FOR USE OF MECHANICAL ADJUSTABLE CHOCKS (MAC). A mechanical adjustable chock must be used at each bolt location. Angular misalignment between equipment and foundation must not exceed four (4) degrees. The ratio of the maximum height of the MAC to the bolt diameter must not exceed 3.0. For retrofit installations, the original configuration must have a valid shock qualification.

6.13.1 Allowable Stresses for the MAC Body and Associated Structure. Using elastic inputs for T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1):

a. The maximum compressive force on the mechanical adjustable chock (MAC) must not exceed 90% of the maximum element load of the mechanical adjustable chock as determined by compressive testing of the MAC and documented in the approved Navy source control drawing. (Note: the pre-load in the bolt will effect the overall compressive stress in a manner analogous to the tensile effect above.)

b. The bearing stress between the MAC and the equipment flange or the MAC and the foundation flange must not exceed the bearing allowable of the respective materials. (Note: the pre-load in the bolt will affect the overall compressive stress in a manner analogous to the tensile effect above.)

c. The pull-through and tear-out stresses of the equipment flange and the foundation flange must not exceed the allowable of the respective flange materials.

d. The bending stress of the equipment or foundation flanges must not exceed the allowable bending plus membrane stress as defined in this chapter.

6.13.2 Allowable Stress for Hold-Down Bolts in Mechanical Adjustable Chocks (MAC). Using elastic inputs from T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1), the Von Mises stress of the hold-down bolts shall not exceed the proof stress. The Von Mises stress shall be calculated using axial stresses and shear stresses as determined below to account for the effects of the MAC.

The bolt axial stress shall be set equal to P/A where P is the larger of Pmax or Pext as defined below:

$$P\max = P_i + \left(\frac{K_b}{(K_b + K_j)}\right)P_{ext}$$

Where:

P_{max}	=	Maximum Bolt Axial Load
P_i	=	Bolt Pre Load
P_{ext}	=	Bolt External Tensile Load (Shock and Operational)
K_b	=	Stiffness of the Bolt (AE/L)
K_j	=	Stiffness of the Joint (includes the effects of the flanges, MAC, gaskets, washers etc. Kj must be as determined by compressive testing of the MAC and documented in the approved Navy source control drawing.

The bolt shear stress shall be:

F(Sext / Smax)/A

Where:

S_{ext}	=	Bolt External Shear Load (Shock and Operational)
S_{max}	=	Bolt Maximum Shear Load (.6 x proof stress x A)
F	=	Bump-up factor to account for bending across the MAC and may have two values, one for clearance holes and one for fitted. The factor “F” must be obtained by an approved method.

CHAPTER 7
DYNAMIC SHOCK ANALYSIS REVIEW AND APPROVAL PROCEDURES

7.1 BACKGROUND.

Because of the specialized nature of shock design requirements and in particular the extreme importance of consistent and qualified determination of compliance, the need for a responsible centralized review activity was recognized by the Navy. Shipbuilding specifications generally indicate that review and approval of the mathematical model and the dynamic analysis will be made by the Technical Authority. To meet this need, a special group was established and trained with the office of Supervisor of Shipbuilding, Brooklyn, NY. In 1965, the Dynamic Shock Analysis Division, Code 280, Supervisor of Shipbuilding, Brooklyn, NY, was assigned responsibilities to provide centralized technical support in review/approval of dynamic analysis. Currently the responsibility resides with the Technical Authority.

The mathematical model report and dynamic shock analysis review and approval requirements described in this chapter shall be considered to apply unless specifically modified by applicable contract specifications.

7.2 REPORT FORMAT AND CONTENT.

The format and content required by the Navy for mathematical model reports and dynamic shock analysis reports are as follows:

7.2.1 Mathematical Model Report Format and Content. The mathematical model report describes the structural and functional characteristics and the mathematical model of a shipboard equipment or structure, with its foundation, for purposes of dynamic analysis. The report is used to provide assurance that the equipment or structure will be properly modeled prior to submittal of the dynamic shock analysis report. The mathematical model report shall have a document number, revision, and date that can be referenced, and contain the following information as a minimum:

a. An introductory description of the equipment or structure being analyzed and its normal function or operation.

b. The planned location and orientation of the equipment or structure with respect to the ship's axes.

c. The shock Grade (A or B) to which the equipment is to be qualified.

d. For Grade B items, all shock hazards as defined in the ship specifications need to be addressed and shock qualification methodology defined.

e. Mounting location (hull, deck, or shell) of the equipment.

f. Type of shock design value (elastic or elastic-plastic) to be used in the analysis.

g. Procurement specification(s) under which the equipment is procured.

h. Description of proposed method of analysis.

i. A list of specific areas of concern of the equipment or structure which might be subject to high stresses or deflections under shock loading. Particular attention should be given to the proposed failure criteria for each area. Yield stress or effective yield stress criteria (at normal equipment operating temperatures) shall be described. The consequences of failure in each critical area shall be considered. The effects of a postulated failure on equipment operability or on potential personnel hazards must be included.

j. Assumptions which have been made in the preparation of the model and justification for such assumption.

k. An estimate of the weight and location of center of gravity of the equipment of structure. A listing of weights of components which are used to arrive at the equipment weight shall also be included.

l. A description of the proposed breakdown of the equipment or structure for analysis. The description must indicate how the proposed mass breakdown permits determination of stresses or deflections in the previously defined areas of concern.

m. A separate list of all lumped masses considered in the mathematical model shall be provided. This list shall specify the location with respect to a specified coordinate system and the composition, magnitude, and direction of associated degrees of freedom for each lumped mass. The model report shall discuss the extent and magnitude of computer generated distributed mass used in the problem.

n. If dynamic reduction techniques are to be used in the shock analysis, the mathematical model report shall fully describe the controls that will be applied to ensure the important response characteristics will not be overlooked. The center of gravity of the mathematical model masses of the original and reduced model shall be determined and identified in the model report (see 7.2.1.k). The model report shall also provide a list of the master degrees of freedom. The planned dynamic reduction process and associated criteria for reducing the problem size must be specifically approved by the Navy.

o. A description of the extent and structural characteristics of the foundation. Sketches or drawings are required as part of the model report to indicate the arrangement of the equipment and its foundation.

p. Properly labeled figures and text to describe the model for each direction of shock shall be provided. The text shall discuss:

   (1) Formulation of the model.

   (2) Representative element properties.

   (3) Details associated with combining shock stresses with continuing operating stresses.

   (4) When the model is prepared for finite element computer analysis, the following information shall also be included:

      (a) A description of the applicable portion of the computer program and the characteristics of the elements to be used.

      (b) A neutral file (viewable by the Technical Authority reviewer) and complete electronic input file and description of the input data used.

      (c) The node and element numbering system and plots of the model to help the reviewer correlate specific nodes, elements and lumped mass locations with the input data.

      (d) Boundary conditions used in the model.

   (5) Where special modeling techniques are used such as mesh generation routines, sub-structuring, etc., additional information shall be furnished to clearly describe the process including objectives and limitations.

q. A map of the finite element model (figures or sketches) shall be provided showing grid point (or node) numbers, element numbers and lumped mass locations (this information can be provided by separate figures or neutral file). Computer generated mathematical model figures (graphics) are often difficult to read. Care should be taken so that the material is legible and clear.

r. Fixed-base natural frequency calculations of suspected low frequency system components (e.g., shafts, cantilevered equipment, yardarm), shall be provided. A comparison of these frequency values to the cut-off frequency of the system shall be made and the components modeled accordingly.

s. References to the source of analysis method, formulas, constants, curves, and all other sources used. Shock tested items which are a part of the equipment or structure to be analyzed must be included in the model but need not be modeled in detail. Wherever qualification of components is to be through MIL-DTL-901 testing, rather than through analysis, the mathematical model report shall contain information on the status of testing. If testing has been completed, references shall be given to the test report and applicable approvals by the Technical Authority or its representatives. If testing is to be done in the future, schedules and planned test facilities should be described.

t. Equipment outline and assembly drawings, support, sub-base, and foundation plans. The report shall include preliminary drawings when final drawings are not available. If no drawings are available, sketches shall be provided. These drawings or sketches shall disclose a level of design detail commensurate with the analysis. Detailed working drawings are not required.

u. A simplified bench-mark model, including all input and output, shall be provided separately or with the model report if requested by Navy. The purpose of this bench-mark problem is to ensure that the DDAM criteria are correctly applied. The characteristics and parameters of the bench-mark model shall be as specified by the Navy (or a simple three degree of freedom model that can easily be verified by hand calculations). Stress calculations in the bench-mark problem should be limited to beam-type stresses. The bench-mark problem shall also demonstrate pre- and post-processing routines and any special modeling procedures or capabilities that are planned for the shock design analysis.

7.2.2 Dynamic Analysis Report Format and Content. The dynamic analysis report demonstrates the ability of equipment, structures, and systems to resist shock as defined by the DDAM. The report is used in conjunction with the mathematical model report when an item's shock resistance cannot be determined by shock testing or extension from a previously qualified item. The dynamic analysis report shall have a document number, revision, and date that can be referenced and contain the following information:

a. An electronic file of the input data used in the analysis. This data shall include all nodal point locations, element connectivity, material properties, element properties, and mass distribution. The DDAM report shall include a full description of the mathematical model used. The approved mathematical model report may be submitted as an appendix to the final DDAM report. Any differences between the approved mathematical model and the model presented in the DDAM shall be noted, fully explained and justified. When computer output on large finite element analyses is too large for inclusion in the dynamic analysis report, the Technical Authority, shall be consulted to obtain a precise definition of the data which may be excluded.

b. A list of all calculated modal frequencies, modal effective weights, and participation factors for all modes of the system including what's not considered in the stress analysis process. The input accelerations should NOT be included in the final report but submitted to the Technical Authority reviewer on a separate CD. This list shall also identify the modes which are used in the stress or deflection calculations. Mode shapes and associated forces and deflections for all modes considered in the stress or deflection calculations shall be included in the report. If computer output is used directly, adequate references and sufficient explanatory detail must be provided to facilitate review.

c. A graph showing modal effective weight versus modal frequency. Closely spaced modes occurring in a DDAM analysis can produce misleading results. The existence of closely spaced modes can best be determined by a graph showing the modal effective weight versus modal frequency for all the modes chosen for analysis. This representation will show potential closely spaced modes. All DDAM analyses reports must contain this graph in order to show that the assumptions of the DDAM with respect to closely spaced modes have not been violated. Where closely spaced modes exist an additional graph representation is required to evaluate the effect of the closely spaced modes on the system design. This second graph shows the modal response versus node point for the modes which are considered to be closely spaced.

d. Calculations of stresses and deflections at those specified areas of concern on the equipment or structure under shock loading, as defined in the mathematical model report. References to the source of data used in these calculations shall be provided. If no drawings are available, sketches shall be provided.

e. Tabulated summaries of calculated and allowable stresses and deflections. These summaries shall include the sources of the tabulated stresses and deflections (for example, tensile, shear, and operating loads). NRL sum of stresses for all elements in the mathematical model shall be provided in the DDAM report. Where the element is an equivalent elastic member, such as a spring or a uniform beam rather than a comprehensive finite element description, the effective forces or stresses on the actual structural element shall be derived and presented in separate calculations.

f. A list of any elements with a negative margin of safety. Where an over-stress is indicated (and is not considered a stress concentration or local stress by Appendix F), a proposed remedy for the condition is required. The effect of any such change on the overall analysis shall be provided. A re-analysis may be required by the Navy. If re-analysis is required, a formal plan of action and milestones (POAM) must be submitted which defines the dates by which necessary NAVSEA approvals for the design change must be obtained, as well as dates for completion of detail design and installation of the change.

g. A comprehensive analysis of the foundation, when such foundation is supplied by the equipment vendor. When the foundation is provided by the shipbuilder, the vendor shall provide a summary of the shock forces into the foundation for use by the shipbuilder in this analysis.

h. A full description of the application of ASM shall be submitted if ASM is used to evaluate responses as part of a corrective action recommendation report. This discussion shall provide the following information as a minimum:

   (1) description of the response characteristics under investigation

   (2) time step used

   (3) period of duration of the ASM

   (4) lowest modal frequency

   (5) highest modal frequency considered in the analysis

(6) the suspected closely spaced modes for each member evaluated

i. A list of modal accelerations for sub-component appendages (such as on mast yard arms). This list shall include all modes of response and shall be sorted in decreasing order by magnitude of the acceleration. The DDAM analysis shall include, in addition to the normal mode selection, the modal stresses or deflections for at least the two most severe responses associated with each appendage.

j. When plate finite elements are used in the mathematical model, for which forces and stresses are calculated at each node point in the plate element, the values at high stress areas may not be averaged between elements unless it can be demonstrated that the variations in unaveraged stresses in the region of interest are within acceptable limits. A contour plot of the unaveraged node stresses in the region of interest can be used to supplement contour plots with averaged stresses. The evaluation of adequacy of mesh discretization will be based on the relative magnitudes of stress among adjacent elements. Typically, in an adequately refined mesh, the contour plots of Von Mises effective stresses will reveal “Stress Bands” which are slightly discontinuous across element boundaries. Large discontinuities indicate a mesh which is too large.

7.3 REVIEW AND APPROVAL AUTHORITY.

Mathematical model reports, dynamic shock analyses, and extension requests based upon dynamic analyses shall be forwarded to the Technical Authority by ship contracts and/or specifications. All dynamic analysis submittals not covered by this section shall be forwarded to the Technical Authority for review and approval.

7.3.1 Equipment, Weapons, and Systems Analyses. Mathematical models, dynamic shock analyses, and extension requests based upon approved dynamic analyses developed to satisfy contractual requirements shall be forwarded to the Technical Authority for approval action.

7.3.2 Foundation Analyses. Where required, foundation dynamic analyses shall be subject to review and approval by the Technical Authority unless otherwise stated by applicable contract documents or specifications.

7.3.3 Ten Sample Foundations. Where the ship’s specifications/contracts require the shipbuilder to submit sample foundation calculations to the Navy for review, these calculations shall be forwarded to the Technical Authority. The shipbuilder shall prepare sample shock calculations for a series of at least ten foundations covering all elements noted below. This selected set of calculations will constitute a diverse and representative sample describing the application of shock design requirements by the shipbuilder. The math model and analysis may be submitted together. Calculations for additional foundations shall be provided if requested.

All the following categories shall be included in the sampling. One foundation model may be used to address more than one of the categories listed below. Foundations associated with equipment DDAM analysis shall be prepared with the equipment DDAM analysis and shall not be included in the list of sample foundations.

a. Foundation for hull mounted equipment

b. Foundation for deck mounted equipment

c. Equipment foundation including a sway brace configuration

d. Foundation for resiliently mounted equipment

e. Foundation for overhead mounted equipment

f. Foundation for bulkhead mounted equipment (structural bulkheads)

g. Foundation for bed-plate, raft, or pallet mounted equipment (items with two or more mounted components)

h. Foundation for a typical electrical power distribution switchboard

i. Foundation for bulkhead mounted equipment (joiner bulkhead)

j. Foundation with an upper support in addition to a base mount

k. Foundation for Grade A alignment sensitive equipment

l. Foundation for typical Grade B equipment

m. Foundation for a fire pump

n. Foundation for equipment with critical clearance requirement

o. Typical deck-to-deck foundation
p. Foundation for free standing tank
q. Stanchion mounted foundation
r. Foundation using Mechanical Adjustable Chocks

7.4 NAVY REVIEW AND APPROVAL/DISAPPROVAL CYCLE.

a. Unless modified by the shipbuilding or contract specification, the Navy will complete action on math model reports within 60 days of receipt of same. Provisional approvals may be granted to permit proceeding with the analyses in cases where only minor corrections and/or additional reference material are required. In such cases the cognizant design approval agency will ensure that supplemental material is forwarded promptly.

b. For mathematical models which are disapproved, the forwarding letter will indicate the basis for disapproval. The cognizant design approval agency is expected to follow up the rejection to ensure that the shipbuilder or contractor is aware of the need for timely response.

c. Unless modified by the shipbuilding or contract specification, the Navy will complete action on dynamic analysis reports within 60 days of receipt of same.

d. For dynamic analysis reports which are not approved, the forwarding letter will indicate the basis for disapproval. The cognizant design approval authority is expected to follow up the rejection to ensure that the shipbuilder or contractor is aware of the need for timely response.

e. Re-submittals of model reports and dynamic analyses which involve the review of extensive modifications shall be treated as new submittals and subject to the applicable Navy review times stated above.

f. The allotted time for Navy review and approval/disapproval of all other dynamic analysis submittals shall be determined by the Technical Authority on a case-by-case basis.

7.5 GUIDELINES AND REQUIREMENTS.

a. A list of all equipment requiring dynamic shock analysis shall be prepared by the shipbuilder or contractor and forwarded within 60 days of the signing of the contract, unless otherwise indicated by appropriate specification or contract.

b. A planned schedule of submittals of mathematical models and dynamic shock analysis shall be prepared by the shipbuilder or contractor and forwarded within 30 days of item a above. The schedule shall be updated at 30-day intervals unless otherwise indicated in the appropriate specification or contract. This schedule shall be based on realistic vendor information and shall reflect the shipbuilder’s or contractor’s requirements for orderly plan development and production/delivery schedules.

c. Each mathematical model report and dynamic analysis report for an equipment being analyzed must provide sufficient information and detail to permit timely review. Items indicated in 7.2.1 and 7.2.2 of this chapter are needed to establish the suitability of these reports. The cognizant design approval agency will screen all mathematical model reports and dynamic analysis reports for conformance with guidelines of this chapter, prior to submittal to the Technical Authority. In order to expedite review, the local design approval agency may authorize direct liaison between the Technical Authority and the shipbuilder or contractor.

d. Since it is the responsibility of the cognizant design approval agency to ensure that characteristics of the equipment are in conformity with the applicable ship or equipment specifications, modifications to equipment or foundations which are indicated by the analysis shall be monitored by the cognizant design approval agency to ensure that the equipment installation complies with the analyzed system. Responsibilities for approval of plans and installations are not transferred to the Technical Authority.

e. The shipbuilder (or his design agent or the prime contractor for Government furnished material) shall ensure that all model reports and analyses are acceptable and shall indicate in the forwarding letter that such documentation satisfies all of the requirements of the applicable specifications.

f. In order to provide for timely submittals and reviews, all local design approval agencies shall incorporate the reporting and review actions of this document in all contracts involving dynamic shock design requirements and on outstanding contracts where applicable and permissible under existing provisions.

g. For items approved on the basis of dynamic analysis, the following information shall be added on the assembly drawings:

(1) Shock Grade (as defined in MIL-DTL-901)

(2) Equipment Class (as defined in MIL-DTL-901)

(3) Mounting location aboard ship (as defined in MIL-DTL-901)

(4) Mounting plane aboard ship (as defined in MIL-DTL-901)

(5) Mounting orientation aboard ship (as defined in MIL-DTL-901)

(6) Applicable DDAM report number

(7) Approval letter reference

APPENDIX A
SAMPLE COMPUTATION OF NORMAL MODES OF A STRUCTURE

A.1 INTRODUCTION.

The purpose of this appendix is to illustrate, by a simple numerical example, the computation of required modes of a structure. Consider the following system in Figure A-1:

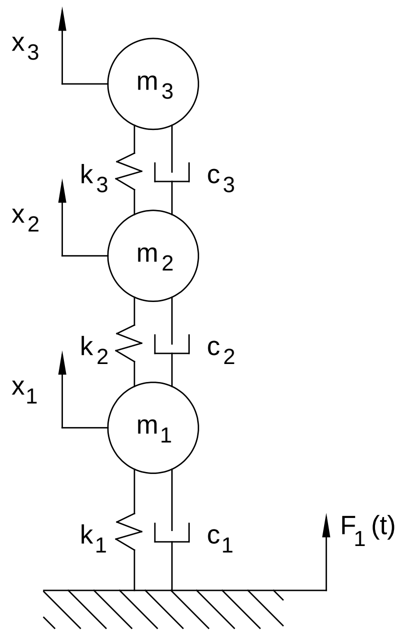


Figure A-1. Mathematical Model – Three Degree of Freedom System.

Where:

$\mathbf{m_i}$ = Mass Value

$\mathbf{k_i}$ = Stiffness Coefficient

$\mathbf{c_i}$ = Damping Coefficient

$\mathbf{x_i}$ = Displacement Coordinate

The equations of motion for the system, which are obtained by considering the dynamic equilibrium of each mass, are shown in Figure A-2.

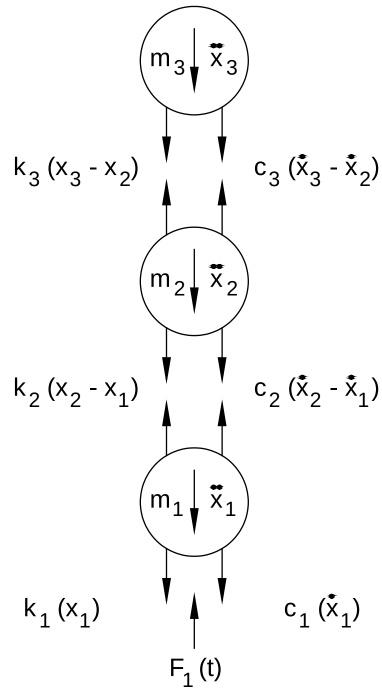


Figure A-2. Free Body Diagram – Three Degree of Freedom System.

Where:

$$\dot{x}_i = \text{Velocity}$$

$$\ddot{x}_i = \text{Acceleration}$$

$$m_1\ddot{x}_1 + k_1x_1 - k_2(x_2 - x_1) + c_1\dot{x}_1 - c_2(\dot{x}_2 - \dot{x}_1) = 0$$

$$m_2\ddot{x}_2 + k_2(x_2 - x_1) - k_3(x_3 - x_2) + c_2(\dot{x}_2 - \dot{x}_1) - c_3(\dot{x}_3 - \dot{x}_2) = 0$$

$$m_3\ddot{x}_3 + k_3(x_3 - x_2) + c_3(\dot{x}_3 - \dot{x}_2) = 0$$

These equations may be conveniently written in matrix form as:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F(t)\} \qquad \textbf{(1)}$$

where:

$$[M] = \begin{vmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m_3 \end{vmatrix}$$

$$[C] = \begin{vmatrix} c_1 + c_2 & -c_2 & 0 \\ -c_2 & c_2 + c_3 & -c_3 \\ 0 & -c_3 & c_3 \end{vmatrix}$$

$$[K] = \begin{vmatrix} k_1 + k_2 & -k_2 & 0 \\ -k_2 & k_2 + k_3 & -k_3 \\ 0 & -k_3 & k_3 \end{vmatrix}$$

$$\{F(t)\} = \begin{Bmatrix} 0 \\ 0 \\ 0 \end{Bmatrix}$$

For undamped free vibration, the damping matrix [C] and the forcing vector {F(t)} are equal to zero and equation (1) reduces to:

$$[M]\{\ddot{x}\} + [K]\{x\} = \{0\} \qquad \textbf{(2)}$$

These equations are solved by substituting:

$$x_1 = a_1 \sin(\omega t)$$

$$x_2 = a_2 \sin(\omega t)$$

$$x_3 = a_3 \sin(\omega t)$$

$$\ddot{x}_1 = -a_1 \omega^2 \sin(\omega t)$$

$$\ddot{x}_2 = -a_2 \omega^2 \sin(\omega t)$$

$$\ddot{x}_3 = -a_3 \omega^2 \sin(\omega t)$$

into equation (2), and canceling the factor $\sin(\omega t)$ to obtain:

$$-m_1 a_1 \omega^2 + k_1 a_1 - k_2 (a_2 - a_1) = 0$$

$$-m_2 a_2 \omega^2 + k_2 (a_2 - a_1) - k_3 (a_3 - a_2) = 0$$

$$-m_3 a_3 \omega^2 + k_3 (a_3 - a_2) = 0$$

In matrix form:

$$\begin{bmatrix} k_1+k_2-m_1\omega^2 & -k_2 & 0 \\ -k_2 & k_2+k_3-m_2\omega^2 & -k_3 \\ 0 & -k_3 & k_3-m_3\omega^2 \end{bmatrix} \begin{Bmatrix} a_1 \\ a_2 \\ a_3 \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \\ 0 \end{Bmatrix} \qquad \textbf{(3)}$$

For a non-trivial solution, it is required that the determinant of the coefficient matrix be equal to zero (eigenvalue problem), that is:

$$\begin{vmatrix} k_1+k_2-m_1\omega^2 & -k_2 & 0 \\ -k_2 & k_2+k_3-m_2\omega^2 & -k_3 \\ 0 & -k_3 & k_3-m_3\omega^2 \end{vmatrix} = 0 \qquad \textbf{(4)}$$

The expansion of the determinant gives a cubic equation in ω^2, namely:

$$m_1m_2m_3\omega^6 - \left[k_3m_1m_2 + (k_1+k_2)m_2m_3 + (k_2+k_3)m_1m_3\right]\omega^4 +$$

$$\left[k_2k_3m_1 + (k_2k_3+k_1k_3)m_2 + (k_1k_2+k_2k_3+k_1k_3)m_3\right]\omega^2 - k_1k_2k_3 = 0$$

Substituting the values for m_1, m_2, m_3, k_1, k_2, and k_3:

$$m_1 = 7.764 lb - \sec^2/in.$$

$$m_2 = 5.176 lb - \sec^2/in.$$

$$m_3 = 2.588 lb - \sec^2/in.$$

$$k_1 = 8.4804x10^6\, lb/in.$$

$$k_2 = 5.6536x10^6\, lb/in.$$

$$k_3 = 2.8268x10^6\, lb/in.$$

the cubic equation becomes:

$$y^3 - 4.551133x10^6\, y^2 + 5.36876x10^{12} - 1.303144x10^6 = 0$$

Where: $y = \omega^2$

The roots of the cubic are:

$$\omega_1^2 = 326{,}722$$

$$\omega_2^2 = 1{,}424{,}591$$

$$\omega_3^2 = 2{,}799{,}875$$

Therefore, the natural frequencies of the system are:

$$\omega_1 = 571.60 \frac{rad}{\sec}$$

$$\omega_2 = 1193.56 \frac{rad}{\sec}$$

$$\omega_3 = 1673.28 \frac{rad}{\sec}$$

or in cycles per second:

$f_1 = 90.97$ Hz

$f_2 = 189.96$ Hz

$f_3 = 266.31$ Hz

The modal shapes are then determined by substituting each of the values for the natural frequencies into equation (3), deleting one of the equations, and solving the remaining two equations for two of the unknowns in terms of the third unknown. The first parameter a_1 is set to 1.00. Performing these operations, the following values are obtained for the modal shapes:

$$\begin{array}{lll} a_{11} = 1.00 & a_{12} = 1.00 & a_{13} = 1.00 \\ a_{21} = 2.05 & a_{22} = 0.54 & a_{23} = -1.34 \\ a_{31} = 2.93 & a_{32} = -1.79 & a_{33} = 0.86 \end{array}$$

The graphical form is shown in Figure A-3.

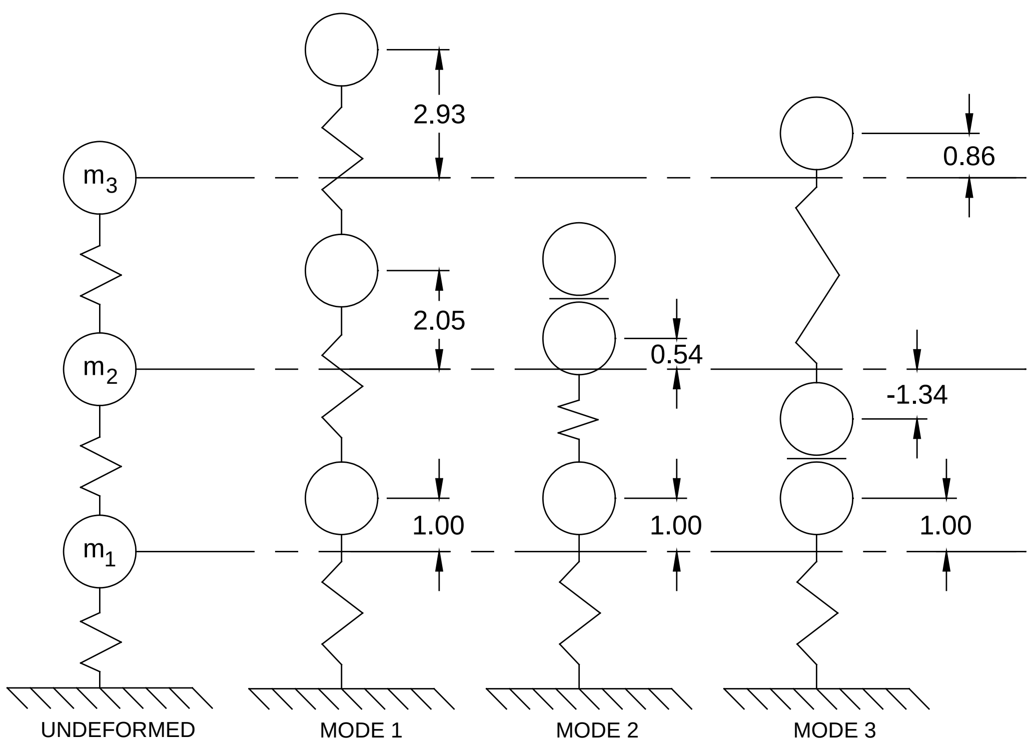


Figure A-3. Mode Shapes, Three Degree of Freedom System.

The eigenvalue problem may also be solved by numerical techniques. There are many methods which can be used to solve the eigenvalue problem. However, the inverse iteration technique is demonstrated here, since it is employed in various important iteration procedures including the determinant search and subspace iteration. The following discussion is presented to illustrate a typical computer analysis method rather than provide the reader with a manual computation approach which would rarely be used. The method presented below converges to the lowest eigenpair; however, shifts may be applied to obtain the higher order eigenpairs.

In the solution, a starting iteration vector $\{X_1\}$ is assumed and then equation (4) is evaluated in each iteration step k = 1, 2…:

$$[K]\{\overline{X}_{k+1}\} = [M]\{X_k\} \qquad \textbf{(4)}$$

After convergence, equations (5) and (6) are evaluated:

$$X_{k+1} = \frac{\overline{X}_{k+1}}{\sqrt{\overline{X}^T_{k+1}[M]\overline{X}_{k+1}}} \qquad \textbf{(5)}$$

$$\rho(X_{k+1}) = \frac{\overline{X}^T_{k+1}[K]\overline{X}_{k+1}}{\overline{X}^T_{k+1}[M]\overline{X}_{k+1}} \qquad \textbf{(6)}$$

As k goes to infinity, X_{k+1} goes to ϕ_1 (eigenvector) and $\rho\{X_{k+1}\}$ goes to ω_1 (eigenvalue).

The solution for the first eigenpair using this technique will be demonstrated for the sample problem. The higher order pairs may be obtained by imposing a shift on the original matrices and proceeding in the same fashion.

$$[K] = \begin{bmatrix} 14.134 & -5.6536 & 0 \\ -5.6536 & 8.4804 & -2.8265 \\ 0 & -2.8265 & 2.8265 \end{bmatrix} x10^6$$

$$[M] = \begin{bmatrix} 7.76 & 0 & 0 \\ 0 & 5.176 & 0 \\ 0 & 0 & 2.588 \end{bmatrix}$$

To solve equation (4), it is first necessary to decompose the stiffness matrix [K] into its triangular factors [D] and $[L]^T$. The general equations for the decomposition are as follows:

$$g_{i,j} = k_{i,j}$$

$$g_{ij} = k_{ij} - \sum_{r=1}^{i-1} l_{ri}\, g_{rj} \,..i = 2,............., j-1$$

$$l_{ij} = \frac{g_{ij}}{d_{ii}} ...i = 1,............., j-1$$

$$d_{jj} = k_{jj} - \sum_{r=1}^{j-1} l_{rj}\, g_{rj}$$

The particular solution is:

$$d_{11} = k_{11} = 14.134 \text{ x } 10^6$$

$$g_{12} = k_{12} = -5.6536 \text{ x } 10^6$$

$$l_{12} = g_{12}/d_{11} = (-5.6536 \text{ x } 10^6)/(14.134 \text{ x } 10^6) = -0.4$$

$$d_{22} = k_{22} - l_{12}g_{12} = (8.4804 \text{ x } 10^6) - (-0.4)(-5.6536 \text{ x } 10^6) = 6.226 \text{ x } 10^6$$

$$g_{23} = k_{23} = -2.8268 \text{ x } 10^6$$

$$l_{23} = g_{23}/d_{22} = (-2.8268 \text{ x } 10^6)/(6.226 \text{ x } 10^6) = -0.454$$

$$d_{33} = k_{33} - l_{23}g_{23} = (2.8268 \text{ x } 10^6) – (-0.454)(-2.8268 \text{ x } 10^6) = 1.543 \text{ x } 10^6$$

The resulting decomposed matrices are:

$$[D] = \begin{bmatrix} 14.134 & 0 & 0 \\ 0 & 6.226 & 0 \\ 0 & 0 & 1.543 \end{bmatrix} x10^6$$

$$[L]^T = \begin{bmatrix} 1 & -0.4 & 0 \\ 0 & 1 & -0.454 \\ 0 & 0 & 1 \end{bmatrix}$$

Equation (4) may now be written as:

$$10^6 \begin{bmatrix} 14.134 & 0 & 0 \\ 0 & 6.226 & 0 \\ 0 & 0 & 1.543 \end{bmatrix} \begin{bmatrix} 1 & -0.4 & 0 \\ 0 & 1 & -0.454 \\ 0 & 0 & 1 \end{bmatrix} \begin{Bmatrix} \overline{X}_{12} \\ \overline{X}_{22} \\ \overline{X}_{32} \end{Bmatrix} = \begin{bmatrix} 7.76 & 0 & 0 \\ 0 & 5.176 & 0 \\ 0 & 0 & 2.588 \end{bmatrix} \begin{Bmatrix} 1 \\ 1 \\ 1 \end{Bmatrix}$$

Multiplying through:

$$10^6 \begin{bmatrix} 14.134 & -5.6536 & 0 \\ 0 & 6.226 & -2.827 \\ 0 & 0 & 1.543 \end{bmatrix} \begin{Bmatrix} \overline{X}_{12} \\ \overline{X}_{23} \\ \overline{X}_{32} \end{Bmatrix} = \begin{Bmatrix} 7.76 \\ 5.176 \\ 2.588 \end{Bmatrix}$$

Reducing the right side vector:

$$q_i = Q_i - \sum_{r=m_i}^{i-1} l_{ri} q_r$$

$$q_1 = 7.764$$

$$q_2 = 5.176 - l_{12}(V_1) = 5.176 - (-0.4)(7.76) = 8.282$$

$$q_3 = 2.588 - l_{23}(V_2) = 2.588 - (-0.454)(8.28) = 6.340$$

Solving for $\{\overline{X}_2\}$

$$10^6\begin{bmatrix}14.134 & -5.6536 & 0\\ 0 & 6.226 & -2.827\\ 0 & 0 & 1.543\end{bmatrix}\begin{Bmatrix}\overline{X}_{12}\\ \overline{X}_{22}\\ \overline{X}_{32}\end{Bmatrix}=\begin{Bmatrix}7.764\\ 8.282\\ 6.340\end{Bmatrix}$$

$$\overline{X}_{32} = 6.340/1.543x10^6 = 4.10894x10^{-6}$$

$$\overline{X}_{22} = (8.282+(2.827x10^6)(4.1089x10^{-6}))/6.226x10^6 = 3.1956x10^{-6}$$

$$\overline{X}_{12} = (7.764+(5.6536x10^6)(3.1956x10^{-6}))/14.134x10^6 = 1.828x10^{-6}$$

Dividing $\{\overline{X}_2\}$ by the first component $\overline{X}_{12}$ gives the first iteration $\{X_2\}$ approximation in the lowest eigenvector.

Therefore X_2 = 1.0

1.75

2.25

Continuing the iteration process using the resulting vector $\{X_i\}$ from the previous iteration, at the starting vector in equation (4), the resulting iterations are:

Vector X_3	Vector X_4	Vector X_5	Vector X_6
1.0	1.0	1.0	1.0
1.99	2.04	2.05	2.05
2.76	2.89	2.92	2.93

Evaluating equation (3), to generate a normalized mode shape:

$$\sqrt{[\overline{X}_{k+1}]^T[M][\overline{X}_{k+1}]}$$

$$\begin{bmatrix}1.0 & 2.05 & 2.93\end{bmatrix}\begin{bmatrix}7.76 & 0 & 0\\ 0 & 5.176 & 0\\ 0 & 0 & 2.588\end{bmatrix}\begin{bmatrix}1.0\\ 2.05\\ 2.93\end{bmatrix}$$

Multiplying through and taking the square root:

= 7.1923

The normalized mode shape is:

0.1391

0.2853

0.4071

Now evaluating the eigenvalue from equation (6):

$$\frac{\begin{bmatrix}0.1391 & 0.2853 & 0.4071\end{bmatrix}\begin{bmatrix}14.134 & -5.6536 & 0 \\ -5.6536 & 8.4804 & -2.8268 \\ 0 & -2.8268 & 2.8268\end{bmatrix}x10^6\begin{bmatrix}0.1391 \\ 0.2853 \\ 0.4071\end{bmatrix}}{\begin{bmatrix}0.1391 & 0.2853 & 0.4071\end{bmatrix}\begin{bmatrix}7.764 & 0 & 0 \\ 0 & 5.176 & 0 \\ 0 & 0 & 2.588\end{bmatrix}\begin{bmatrix}0.1391 \\ 0.2853 \\ 0.4071\end{bmatrix}}$$

$$\rho = \omega^2 = 326{,}719.83(\frac{rad}{\sec})^2$$

$$\omega = 571.59\frac{rad}{\sec} \qquad f = 2\pi\omega = 90.97Hz$$

Now imposing a shift, the eigenvalue problem becomes:

$$[K - \mu M]\phi = \eta M\phi$$

Where:

$$\eta_i = \lambda_i - \mu$$

Assuming a shift of 1 x 10^6, the $[K - \mu M]$ matrix becomes:

$$[K] = \begin{bmatrix}6.374 & -5.6536 & 0 \\ -5.6536 & 3.304 & -2.8265 \\ 0 & -2.8265 & 0.2385\end{bmatrix}x10^6$$

The decomposition and iteration may now proceed as before.

APPENDIX B
FINITE ELEMENT METHOD FOR DDAM ANALYSIS

B.1 INTRODUCTION.

When performing a dynamic analysis for any system with two or more degrees of freedom, it is necessary to create a flexibility or stiffness coefficient matrix (see 3.4). For complicated mass-spring systems, hand calculations are impractical. The finite element method is currently being used by many analysts to perform this analysis. The method is described below in this appendix. For dynamic analyses required by the shipbuilding specifications, the finite element method is acceptable.

Finite element codes provide the user with a library of element types which represent distinct patterns of structural response reflected by rods, beams, plates, continuum, etc. These finite elements are derived from the principle of Minimum Potential Energy based on assumed shape functions and are therefore approximate. However, sufficiently refined assemblages of finite elements can be constructed to represent the behavior of structural systems. At element intersections, displacements and rotational compatibility may be enforced or released by the user. The finite element method is a systemized method for assembling sophisticated mass elastic systems and therefore must conform to the guidelines provided with this document.

The following is a list of the type of information that the analyst must assemble for a discrete element type model:

a. Type of material – steel, aluminum, etc.

b. Type of structure – frame or truss

c. Type of loading

d. Degrees of freedom – description of all releases and constraints

e. Description of each finite element

f. Mass distribution

The above type of information, when entered into an appropriate computer program, will produce the stiffness matrices necessary for the performance of the dynamic analysis.

State-of-the-art finite element programs are capable not only of producing the stiffness matrices, but also of calculating natural frequencies and mode shapes in one step. Such programs tend to eliminate the distinctions made in Chapter 3 between the coefficient computation phase and the dynamic computation phase. Certain proprietary versions of finite element programs calculate the DDAM motion inputs, modal stresses in beam or plate elements, NRL stress, and margins of safety relative to allowable design stresses. The evaluation phase described in Chapter 3 can therefore largely be done in conjunction with the coefficient computations phase with such programs.

It must be emphasized that the use of large finite models for DDAM analyses does not relieve the analyst from obligations to exercise judgement and to properly interpret the analytical results. For example, shock stresses calculated directly by finite element models are often only gross approximations. In many instances, complicated geometrical parts are represented by simple constant-section beam elements for purposes of generating system flexibility or stiffness properties. The program-calculated stresses in such elements must be checked by means of manual calculations which account for the true geometry of the parts being evaluated. Alternatively, secondary finite-element analyses with more modeling detail in the areas in question may be conducted. These secondary analyses may be static ones, with the applied loads being the DDAM-calculated inertia loadings.

The capabilities available in modern finite element programs tend to encourage the use of large mathematical models for DDAM analysis. Figures B-1 and B-2 illustrate a finite element mathematical model of moderate complexity. Included in the model are both beam and plate elements. The tendency to use models of ever-increasing complexity should be discouraged. Overly complicated models have the following disadvantages:

a. Difficulty in performing review and check. Extremely voluminous input/output data sets make checking of the analytical results difficult for both the contractor and the Navy and thus reduces the overall level of confidence in the shock hardness of the design.

b. Misleading accuracy of results. Since the dynamic analysis by DDAM of most large complicated models generally requires the use of reduction techniques, the accuracy of the results may not be as reliable as expected. Since the solution of the dynamic problem has been obtained from a reduced mathematical model, the accuracy has not been increased by excessive refinement of the model. In fact, if the reduction process is improperly applied, a lower level of accuracy will be achieved for the more complicated model.

c. The larger the model, the higher the probability of producing closely spaced modes.

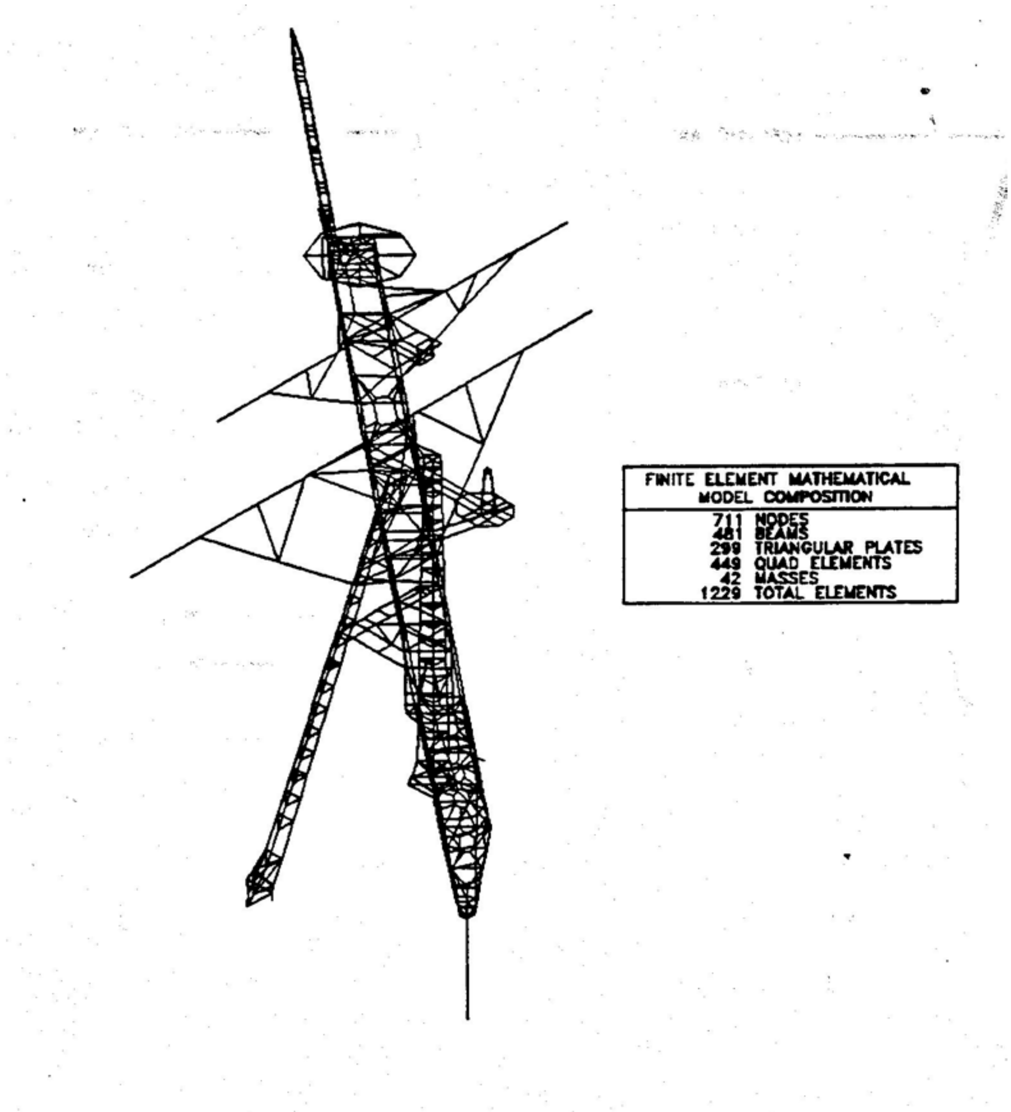


Figure B-1. Mathematical Model Representation of a Mast – Isometric View.

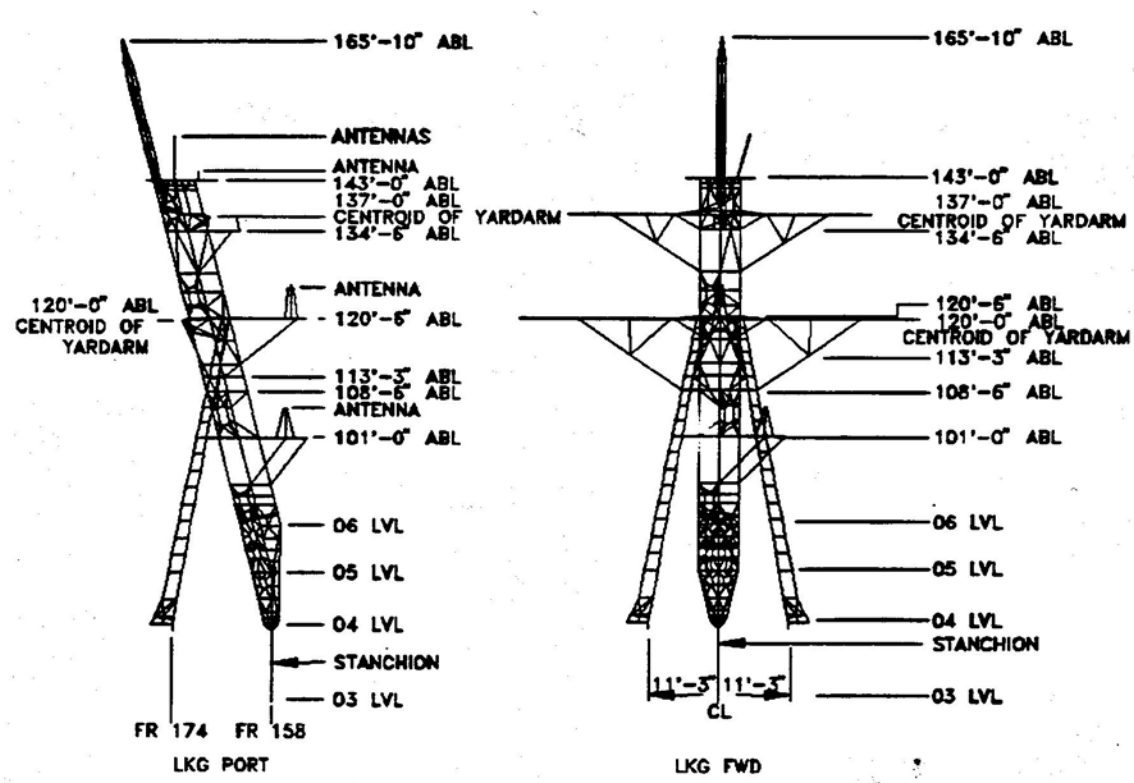


Figure B-2. Mathematical Model Representation of a Mast – View Looking Forward and View Looking Port.

APPENDIX C
TRANSIENT ANALYSIS METHODS

C.1 INTRODUCTION.

As noted throughout this manual, the DDAM is considered appropriate for use on linear, elastic shipboard systems for which the T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1) shock design values are considered applicable and appropriate. DDAM is approved by the navy as the validated standard analysis method. Therefore, DDAM shall be the first choice for shock qualification by analysis unless elements of the system under consideration invalidate the assumptions of the DDAM. In cases where the DDAM's limitations are violated, other procedures, such as a transient analysis method or an energy method, may be substituted for DDAM if approved by the Technical Authority. It should be noted however, that transient analysis or some other energy method will require verification, validation, and accreditation (VV&A) prior to approval of the analysis process being substituted for the DDAM. The VV&A process is defined in DoDI 5000.61.

C.2 TRANSIENT ANALYSIS METHOD.

The transient analysis method, also referred to as the Transient Shock Analysis (TSA) method, similar to the modal summation technique of DDAM, requires a mathematical model to be developed which represents both the elastic and inertial properties of the system. Whereas the DDAM assumes an undamped solution by combining the maximum responses of each modal contribution regardless of the times at which these modal maxima occur, transient analyses determine the phased responses within a finite response interval. It is not likely, in the presence of structural damping, that the peak modal contributions will constructively combine as assumed in the DDAM. The high frequency responses will likely diminish very rapidly and many of the analytical difficulties within the DDAM associated with closely spaced modes will not be present in a transient analysis. Furthermore, lightweight equipment mounted on low frequency structures in tune with the ship's hull girder frequencies will be subjected to vibratory motion rather than a shock loading, sometimes referred to as shock induced resonance.

Differential base motions can be supplied for larger equipment items with multiple supports to reflect variation in support structure. Non-linear effects and the ability of redundant structures to redistribute forces can also be accounted for in transient analyses. However, unlike the modal methods in which enveloped spectral response motions can be supplied to the analyst, transient inputs depend on the specific test parameters being simulated. As yet no set of general inputs have been determined by the Navy which constitute a transient design environment.

C.3 TWO LEVELS OF TRANSIENT RESPONSE CALCULATIONS.

Transient response calculations can be conducted on two distinct levels of analytical rigor:

a. Local equipment responses can be determined by subjecting the equipment to transient base motions. Care must be exercised in selecting boundaries for the equipment model and the application of the input motions. Gage records must be chosen prudently to best represent the characteristics of the equipment structure interaction. This is the least complex, and, thus, preferred method.

b. Fully coupled responses in which a three-dimensional model of the test vehicle is loaded via a fluid structure interaction algorithm. The pressures and motions within the fluid, resulting from a postulated attack geometry and charge weight, load the test vehicle and the response of the internal equipment is calculated interactively with the test vehicle response. This methodology allows the analyst to consider the effects of shock, cavitation, and bubble pulsations on the test vehicle, thereby providing the most complete representation of the three-dimensional response. Transient analysis techniques may also be applied to the analysis of external appendages. The fully coupled transient approach is potentially an expensive method to apply and is not a practical substitute for DDAM in a production mode for most, if not all, equipment foundation design.

The limitations inherent in any of the transient analyses approaches discussed above must be clearly understood. The transient analysis approach requires an accurate definition of the base input motion. As explained in Chapter 3, test data have shown the great importance of the spectrum dip, or equipment feedback, effect on base motions. Determination of this effect requires that the equipment under consideration must be accounted for when deriving the input motions. Errors in the determination of the spectrum dip effect could cause the transient analysis to over-predict equipment and foundation responses to shock. Similarly, responses caused by multiple resonant conditions within the data set used to generate input motions will generally lead to over-prediction.

C.4 TRANSIENT ANALYSIS (TRANSIENT SHOCK ANALYSIS) PROCESS.

a. Request approval from Technical Authority to shock qualify the system by transient analysis, in lieu of test. (The transient analysis approval request must include an explanation why the DDAM is not appropriate for the subject system.)

b. Obtain transient input for use within the transient analysis process from the Navy.

c. Prepare VV&A approval process documentation.

d. VV&A approval process should include:

(1) Assumptions:

(a) Shock Grade

(b) Equipment Class

(c) Shock Test Type

(d) Mounting Location

(e) Mounting Plane

(f) Mounting Orientation

(2) Basis for transient input loading provided by the Navy.

(3) Discussion of dynamics program of choice.

(a) Analysis type – modal superposition or direct integration(implicit or explicit)

(b) Free Surface Effects (if appropriate)

(c) Bulk Cavitation Effects (if appropriate)

(d) Bubble Effects (if appropriate)

(4) Discussion of mathematical model.

(a) Mass Distribution

(b) Element Types

(c) Boundary Conditions

(d) Damping Model Definition

(e) Resilient Element Definition

(f) Material Model Definition

(5) Discussion of acceptance criteria.

(6) Identification of quantities of interest.

(7) Identification of specific intended use.

e. Submit VV&A approval process to the Navy for review and approval.

f. Submit validation analysis to the Navy for review and approval.

g. Perform transient analysis on subject system.

h. Prepare final analysis report using Chapter 7 as guidance.

i. Submit final analysis report to EM for review and approval.

APPENDIX D
OBLIQUE DIRECTIONAL SHOCK INPUTS

D.1 INTRODUCTION.

Components of the design spectrum levels can be used to solve for equipment response to an oblique shock or for redefining the shock design values into equipment oriented axes. Consider that the three specified design spectrum values, D_v, D_a, and D_f form an ellipsoid (not of revolution). The octant of space occupied by this ellipsoid intersects the X, Y, and Z axes at values which correspond to the maximum (or principal) ship oriented design shock spectrum inputs. Figure D-1 shows the relationship between the three axes of a hypothetical damage surface. If the Y axis corresponds to the ship's vertical direction and its principal design spectrum value is D_v, the Z axis corresponds to the athwartship direction with its principal design spectrum value as D_a, and the X axis corresponds to the fore/aft ship direction with its principal design spectrum value of D_f, parametric equations can be developed for any angle of attack. The point P on the surface of this ellipsoid represents the components of the design shock spectrum values to be used for oblique angles of attack or to determine responses along axes other than the principal ship axes.

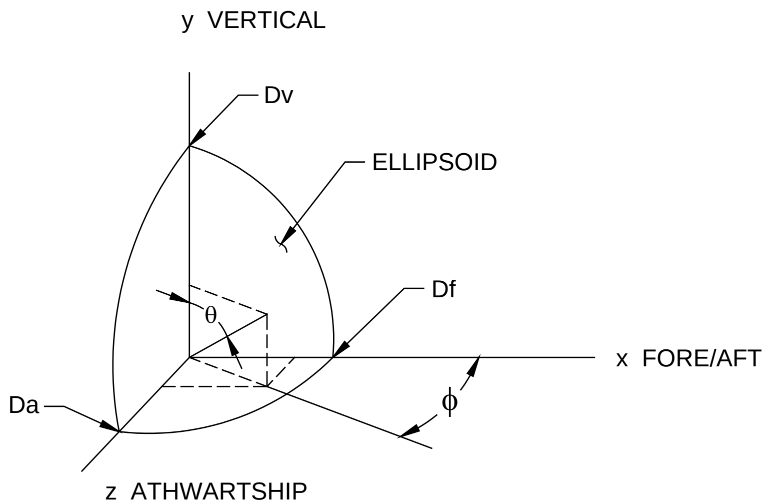


Figure D-1. Hypothetical Damage Surface.

D.2 OBLIQUE EQUIPMENT ORIENTATION.

In a similar manner, if the response axis is rotated to correspond with the equipment axis, rather than principal ship axes, the analysis method requires determination of three coordinate input values for each individual direction of design input. That is, components of the specified vertical design shock input are required to be determined along each of the three equipment axes. These component inputs are to be applied simultaneously and the solution combined on a mode-by-mode basis.

The design produced from shock inputs that have been re-oriented to coincide with equipment axes is the same as the design produced by inputs along the ship axes, and these alternate inputs can be used if desired for ease of calculation and design.

D.3 OBLIQUE EQUIPMENT ORIENTATION – ILLUSTRATIVE EXAMPLE.

Consider a mass – elastic model of the equipment oriented in the fore/aft – vertical plane of the ship whose local axes, X and Y, are rotated about an angle with respect to the global axes of the ship.

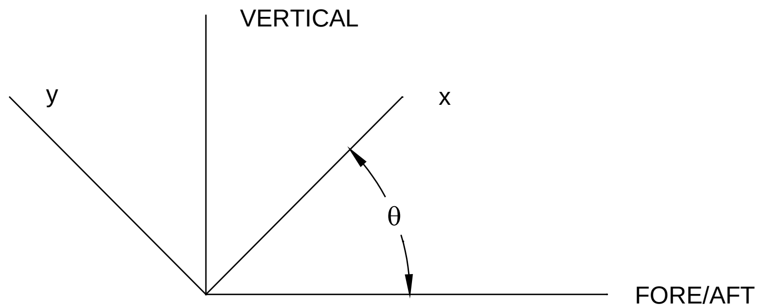


Figure D-2. Orientation of Equipment Axis with Respect to Ship Axis.

For each mode "a" of the equipment analysis a frequency ω_a and a mode shape $\{\Phi_a\}$ are defined in the local x-y coordinate system. Correspondingly, for this multi-directional response analysis, participation factors are calculated for each mode and direction of motion as:

$$P_a = \frac{\{\Phi_a\}^T [M]\{r\}}{\{\Phi_a\}^T [M]\{\Phi_a\}}$$

Where the vector {r} relates the orientation of the motion of the ship to the local coordinates of the equipment. For a simple two degree of freedom system with one degree of freedom in the local x axis and the other in the local y axis, the {r} vector will be:

$$\{r\} = \begin{Bmatrix} \cos\theta \\ -\sin\theta \end{Bmatrix} \qquad \text{for fore/aft motion}$$

$$\{r\} = \begin{Bmatrix} \sin\theta \\ \cos\theta \end{Bmatrix} \qquad \text{for vertical motion}$$

Modal masses are calculated for each mode "a" and assumed direction of ship motion and the spectral response values are obtained from T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1). For fore/aft motion the spectral value is D_{fa} and for vertical motions the spectral response value is D_{va}.

The equipment response displacements for each mode and each direction of ship motion is calculated form normal mode theory as:

$$\{d_{fa}\} = \{\Phi_a\} P_{fa} D_{fa} \qquad \text{for fore/aft ship motion}$$

$$\{d_{va}\} = \{\Phi_a\} P_{va} D_{va} \qquad \text{for vertical ship motion}$$

For the two degree of freedom example previously described, the two components of equipment response, X and Y, for a particular mode "a" will be, for fore/aft ship motion:

$$\{d_{fa}\} = \begin{Bmatrix} X_{fa} \\ Y_{fa} \end{Bmatrix}$$

$$X_{fa} = \frac{\Phi_{a1}(\Phi_{a1} M_1 \cos\theta - \Phi_{a2} M_2 \sin\theta)}{M_1 \Phi_{a1}^2 + M_2 \Phi_{a2}^2} D_{fa}$$

$$Y_{fa} = \frac{\Phi_{a2}(\Phi_{a1}M_1\cos\theta - \Phi_{a2}M_2\sin\theta)}{M_1\Phi_{a1}^2 + M_2\Phi_{a2}^2} D_{fa}$$

and for vertical ship motion:

$$\{d_{va}\} = \begin{Bmatrix} X_{va} \\ Y_{va} \end{Bmatrix}$$

$$X_{va} = \frac{\Phi_{a1}(\Phi_{a1}M_1\sin\theta + \Phi_{a2}M_2\cos\theta)}{M_1\Phi_{a1}^2 + M_2\Phi_{a2}^2} D_{va}$$

$$Y_{va} = \frac{\Phi_{a2}(\Phi_{a1}M_1\sin\theta + \Phi_{a2}M_2\cos\theta)}{M_1\Phi_{a1}^2 + M_2\Phi_{a2}^2} D_{va}$$

Alternatively, spectral response values can be prescribed in the orientation of the local coordinates N_x and N_y:

$$\begin{Bmatrix} N_{xa} \\ N_{ya} \end{Bmatrix} = \begin{bmatrix} \cos\theta & \sin\theta \\ -\sin\theta & \cos\theta \end{bmatrix} \begin{Bmatrix} D_{fa} \\ D_{va} \end{Bmatrix}$$

Participation factors P_{xa} and P_{ya} are determined as before, however, the {r} vector will now relate the new orientation of the ship motion to the local coordinate of the equipment. For the two degree of freedom example for vector {r} will now be:

$$\{r\} = \begin{Bmatrix} 1 \\ 0 \end{Bmatrix} \qquad \text{for x motion and}$$

$$\{r\} = \begin{Bmatrix} 0 \\ 1 \end{Bmatrix} \qquad \text{for y motion}$$

The equipment response displacements for each mode and each direction of ship motion is calculated from normal mode theory as before:

$$\{d_{xa}\} = \{\Phi_a\}P_{xa}N_{xa} \qquad \text{for x direction ship motion}$$

$$\{d_{ya}\} = \{\Phi_a\}P_{ya}N_{ya} \qquad \text{for y direction ship motion}$$

For the example of the two degree of freedom system, the two components of equipment response, X and Y for a particular mode will be, for x direction ship motion:

$$\{d_{xa}\} = \begin{Bmatrix} X_{xa} \\ Y_{xa} \end{Bmatrix}$$

$$X_{xa} = \frac{\Phi_{a1}^2 M_1(D_{fa}\cos\theta + D_{va}\sin\theta)}{\Phi_{a1}^2 M_1 + \Phi_{a2}^2 M_2}$$

$$X_{xa} = \frac{\Phi_{a1}\Phi_{a2}M_1(D_{fa}\cos\theta + D_{va}\sin\theta)}{\Phi_{a1}^2 M_1 + \Phi_{a2}^2 M_2}$$

Similarly for the y direction ship motion:

$$\{d_{ya}\} = \begin{Bmatrix} X_{ya} \\ Y_{ya} \end{Bmatrix}$$

$$X_{ya} = \frac{\Phi_{a1}\Phi_{a2}M_2(-D_{fa}\sin\theta + D_{va}\cos\theta)}{\Phi_{a1}^2 M_1 + \Phi_{a2}^2 M_2}$$

$$Y_{ya} = \frac{\Phi_{a2}^2 M_2(-D_{fa}\sin\theta + D_{va}\cos\theta)}{\Phi_{a1}^2 M_1 + \Phi_{a2}^2 M_2}$$

These modal response motions can be related by reorganizing the relationship between the input motions prescribed in the two coordinate systems. For a pure fore/aft ship motion the response value $D_{va} = 0$, and there will be two components of spectral response values to be applied simultaneously:

$$N_{xa} = D_{fa}\cos\theta$$

$$N_{ya} = -D_{fa}\sin\theta$$

Correspondingly, the equipment response in the x direction will be the sum of the x direction response resulting from N_{xa} and N_{ya}:

$$\{d_{fa}\} = \begin{Bmatrix} X_{fa} \\ Y_{fa} \end{Bmatrix}$$

$$X_{fa} = \frac{\Phi_{a1}^2 M_1(D_{fa}\cos\theta)}{\Phi_{a1}^2 M_1 + \Phi_{a2}^2 M_2} - \frac{\Phi_{a1}\Phi_{a2}M_2(D_{fa}\sin\theta)}{\Phi_{a1}^2 M_1 + \Phi_{a2}^2 M_2}$$

$$Y_{fa} = \frac{\Phi_{a1}\Phi_{a2}M_1(D_{fa}\cos\theta)}{\Phi_{a1}^2 M_1 + \Phi_{a2}^2 M_2} - \frac{\Phi_{a2}^2 M_2(D_{fa}\sin\theta)}{\Phi_{a1}^2 M_1 + \Phi_{a2}^2 M_2}$$

In this fashion, equivalent equipment responses can be calculated to motions in either the global ship axes or the local equipment coordinate system.

APPENDIX E
SAMPLE FINITE ELEMENT DDAM ANALYSIS – FORMAT AND CONTENT

E.1 INTRODUCTION.

This appendix is provided as an example of the format and content of a dynamic analysis report for a typical finite element model. It is not the intent of this example to provide technical guidance in the performance of DDAM.

This appendix presents the mathematical model and the vertical dynamic analysis of the foundation for a radar test set, two transponder sets, and an electronic controller (E.5 of this appendix, Figure E-1). The appendix verifies that the foundation structure shown in E.7 of this appendix, Figures E-4 through E-10, is adequate for Grade A vertical shock. Since some permanent deformation would not invalidate the design for its intended grade of shock, elastic-plastic inputs were used. The appendix also identifies and eliminates closely spaced modes from the modal analysis. It is noted that the finite element model and shock analysis for athwartship and longitudinal shock, although not presented here, will have the same format as the vertical shock analysis.

The material presented in the example problem is representative of the material that would be provided in a finite element DDAM submittal to the Navy for final approval. The following data is provided:

E.1	Introduction	E-1
E.2	Mathematical Model	E-2
E.3	Computer Analysis	E-2
E.4	Results	E-2
E.5	Sketch or Arrangement of Item	E-4
E.6	Sketch of Equipment	E-5
E.7	Sketches of Foundation	E-7
E.8	Mathematical Model Sketch (Node Numbers)	E-14
E.9	Mathematical Model Sketch (Element Numbers)	E-15
E.10	Mathematical Model Sketch (Mass Locations)	E-16
E.11	Computer Input	E-17
	a. Boundary conditions	
	b. Beam element and mass properties	
	c. Grid coordinates	
	d. Beam element, rigid body elements, and mass definitions	
E.12	DDAM Output	E-21
E.12.1	Frequency, Participation Factors, and Modal Weights for Each Mode Used in the NRL Sum	E-21
E.12.2	Modal Mass vs. Frequency and Eigenvector vs. Node Number Charts	E-22
E.12.3	Modal Output (Forces)	E-25
E.12.4	Modal Output (Stresses)	E-28

E.2 MATHEMATICAL MODEL.

The rack type foundation, shown in E.5, Figure E-1, supports a radar test set, two transponder sets and an electronic controller. The electronic controller is attached to the center transponder set. The equipment sketches for the radar test set are shown in E.6, Figure E-2. The equipment sketches for the transponder set are shown in E.6, Figure E-3. The electronic controller is a small rectangular box and the equipment sketches are not provided. Scantling drawings for the rack foundation are shown in E.7, Figures E-4 through E-10. The foundation was modeled using prismatic beam elements for the entire model. The plates shown in Figure E-8 were represented as flanges of beams using effective plate widths. The radar test set and transponder sets are represented with a rigid frame configuration. The electronic controller is modeled as a linear spring and mass. The equipments are represented using a rigid body element consisting of a master node at the equipment center of gravity and slave nodes at the rigid body element (equipment)/foundation interface. The slave nodes are dependent on the master node for the three translation directions allowing the moments to be released to simulate the effects of the bolted connections. The foundation frame is also supported by rigid body element. The slave nodes of the rigid body element are dependent on the master node for all translations and rotations to simulate a full fixed support at the deck. The master node of the rigid body element acts as a seismic mass and is fully fixed for translations and rotations. Computer generated plots for the full structural model are shown in E.8, Figure E-11 and E.9, Figure E-12.

The mass distribution for the mathematical model is shown in E.10, Figure E-13. The three equipment masses are given dynamic degrees of freedom in the three global directions (fully coupled). Because of its size, the electronic controller is given only a vertical dynamic degree of freedom. Due to the symmetry of the structural masses, and resulting small coupled motions in the horizontal plane for vertical inputs, these masses were given only vertical dynamic degrees of freedom for the vertical shock analysis.

E.3 COMPUTER ANALYSIS.

A particular computer program and dynamic solution technique has been chosen for this example. There are numerous other programs available to perform a DDAM analysis. It is not the intent of this example to restrict the finite element analysis to any one computer code. A copy of the computer input data used for this shock analysis is shown in E.11.

A system with three phases to the analysis was used for the shock analysis of the example foundation. The first phase (a general structures program) calculates the stiffness matrix, member loads, support reactions, and joint deflection. The second phase performs the dynamic analysis and determines the natural frequencies and effective static forces associated with each mode. The last phase used in conjunction with the output of the general structures program determines the forces, stresses in each member, and all joint displacements associated with the shock loading. This final phase also combines (NRL sum) the member stresses developed in the modes analyzed.

E.4 RESULTS.

The results for the foundation analysis are provided on the following pages. To demonstrate the identification and elimination of closely spaced modes, an iteration prior to the final iteration is shown for demonstration purposes only. This iteration would not normally be submitted with the final analysis report.

Section	Description	Page
E.12.1	Frequency, Participation Factors, and Modal Weights for Each Mode Used in the NRL Sum	E-21
E.12.2	Modal Mass vs. Frequency and Eigenvector vs. Node Number Charts	E-22
E.12.3	Modal Output (Forces)	E-25
E.12.4	Modal Output (Stresses)	E-28

Allowable bending stresses are twice yield for an elastic-plastic analysis.

Allowable stress is:

$$2 \times 33{,}000\ psi = 66{,}000\ psi$$

Allowable shear stresses are 60 percent of twice yield.

Allowable shear stress is:

$$0.6 \times 66{,}000\ psi = 39{,}600\ psi$$

Reviewing the modal results shown in Figure E-14, it can be seen that modes 6, 7, and 8 are closely spaced (within 10 percent of the lower mode). Further review of the eigenvectors of the three modes, Figure E-15 reveals that the 1-pound electronic controller (node 66) is out of phase and dominates in modes 6 and 7. The force that the 1-lb electronic controller is anticipated to have is excessive. When these modes are summed in the NRL procedure, the canceling effect of the small mass is lost and erroneous results occur.

The problem is eliminated by stiffening the interface so that the mass of the electronic controller may be combined with that of the transponder set. Figure E-16 shows the modified modal data. It can be seen that modes 6 and 7 have been combined into a single mode having the same modal weight as the two previous modes.

After elimination of the closely spaced modes, the critical normal stress (NRL) in member 18, at joint 21 is:

$$\sigma_{max} = 57{,}710\ psi < 66{,}000\ psi$$

After elimination of the closely spaced modes, the critical shear stress (NRL) in member 7 at joints 40/39 is:

$$\tau_{max} = 8{,}233\ psi < 39{,}600\ psi$$

All other stresses are also below allowable limits.

E.5 SKETCH OR ARRANGEMENT OF ITEM.

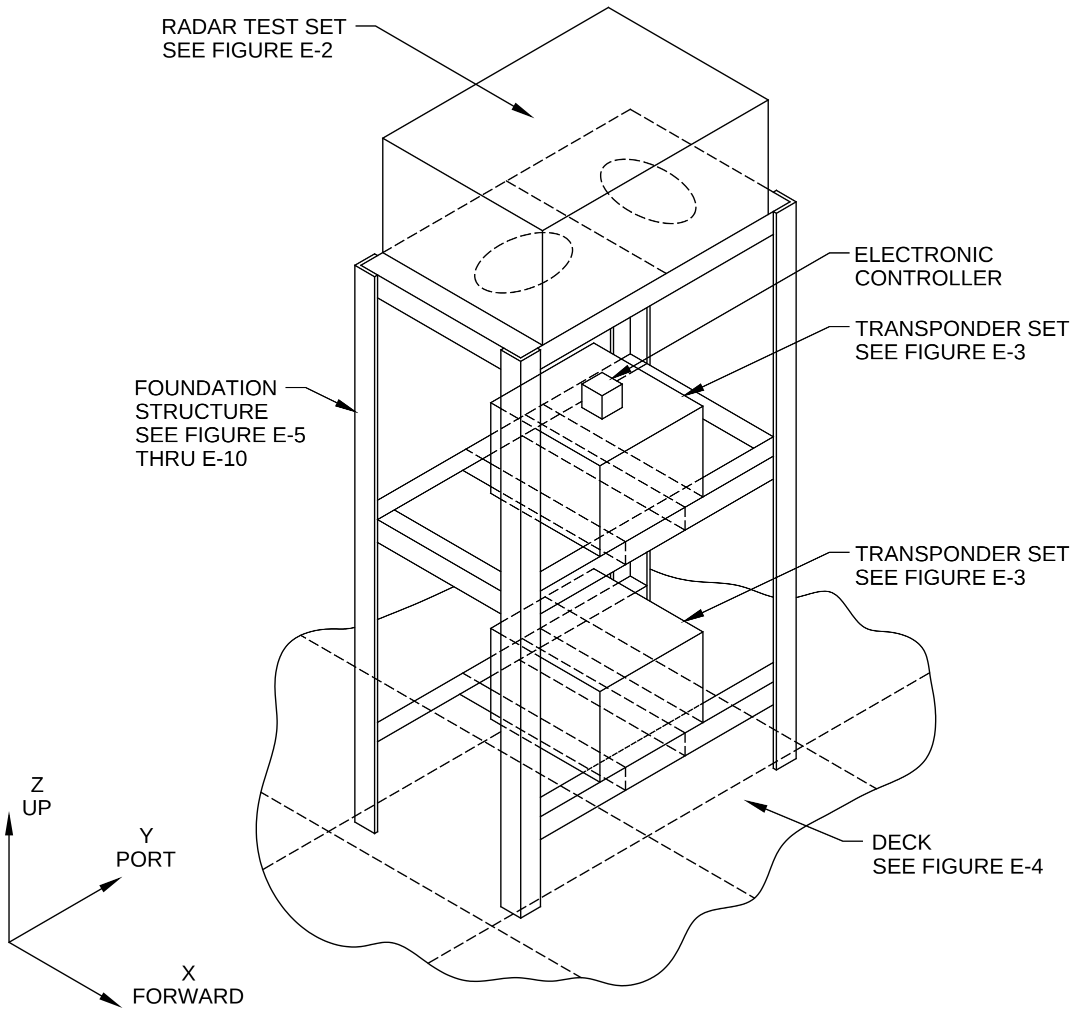


Figure E-1. General Arrangement of Foundation.

E.6 SKETCH OF EQUIPMENT.

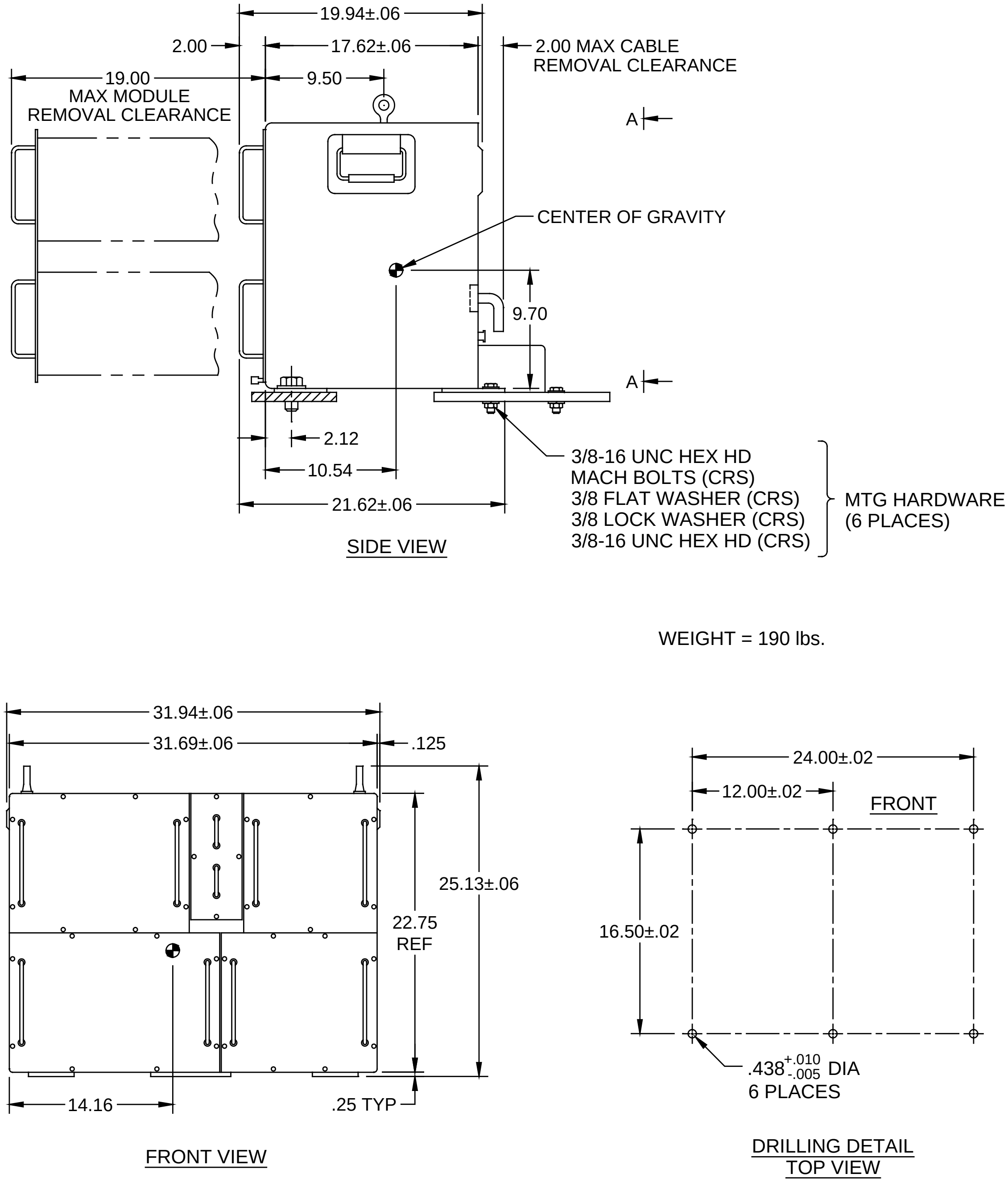


Figure E-2. Radar Test Set.

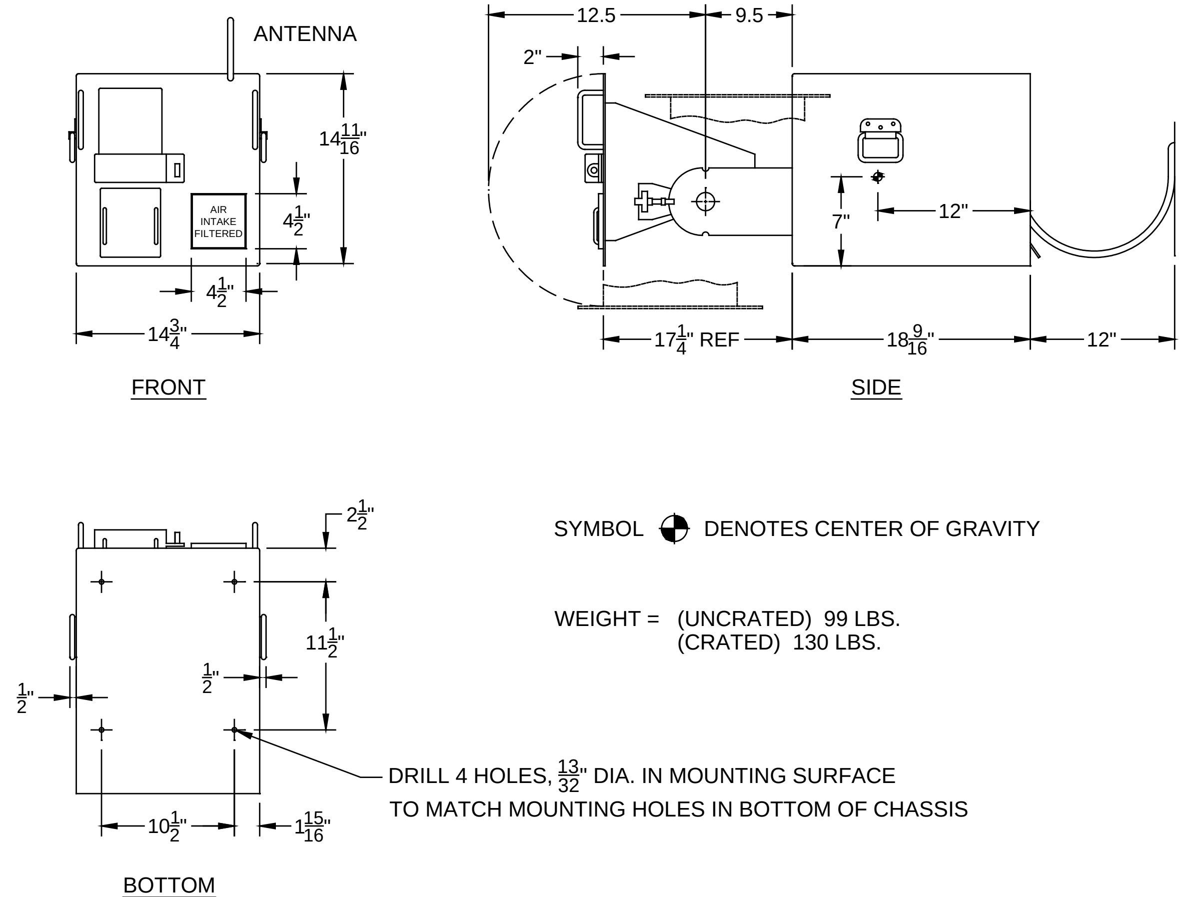


Figure E-3. Transponder Set.

E.7 SKETCHES OF FOUNDATION.

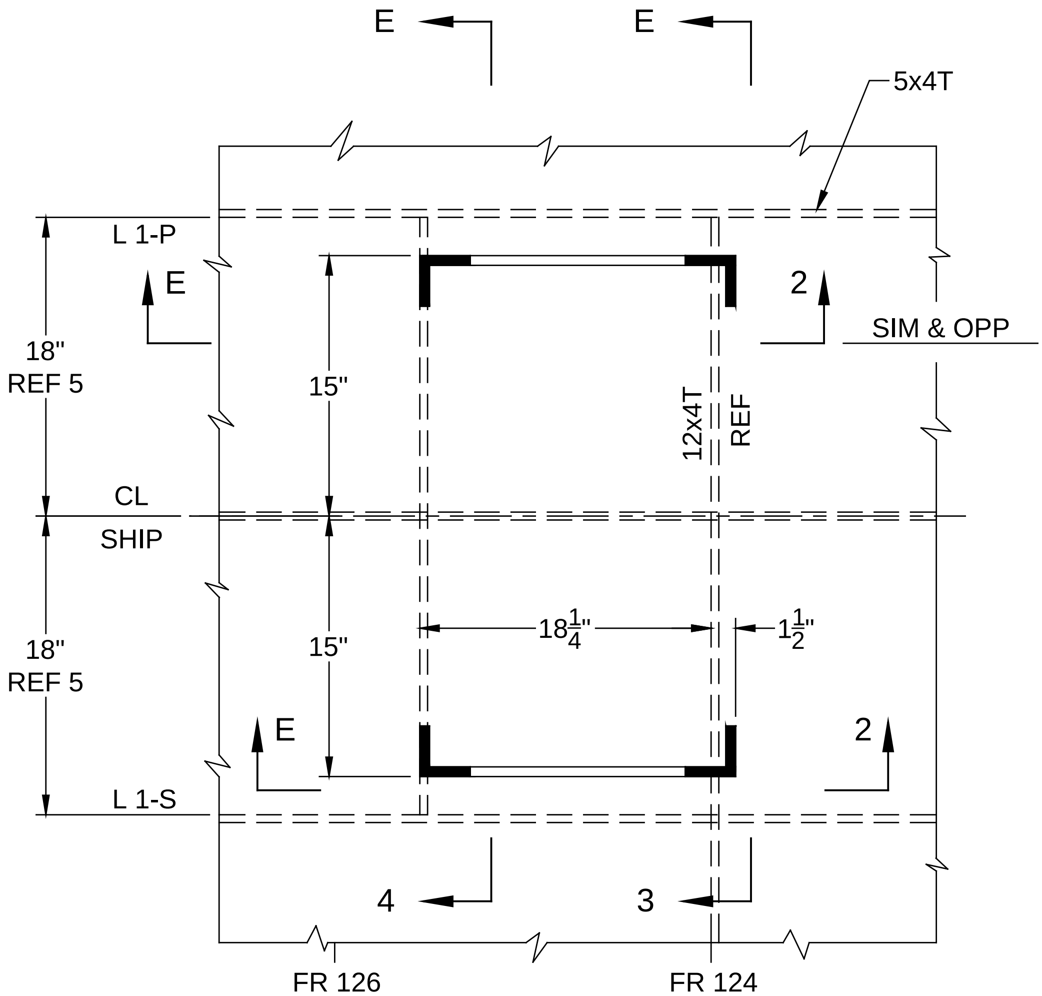


Figure E-4. Scantling Drawing for the Foundation (image 1 of 7).

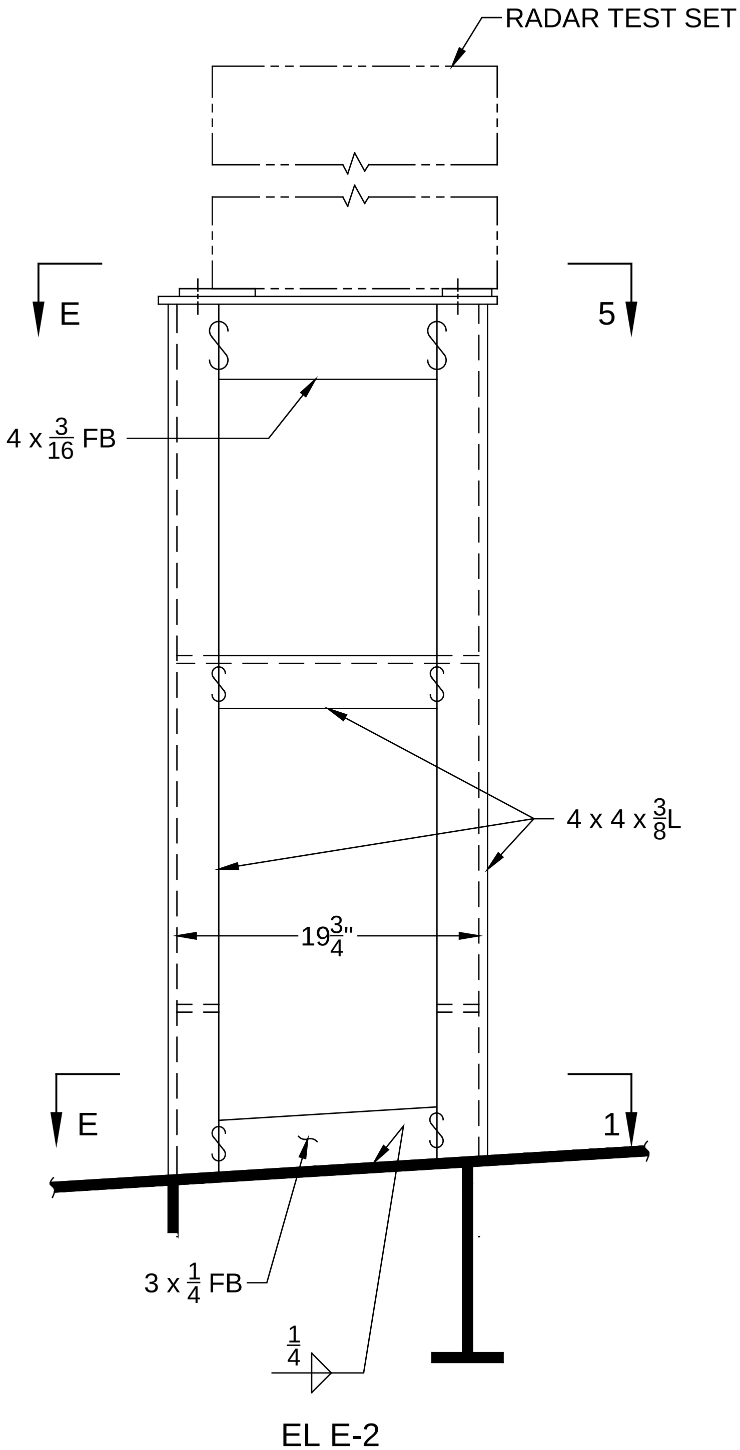
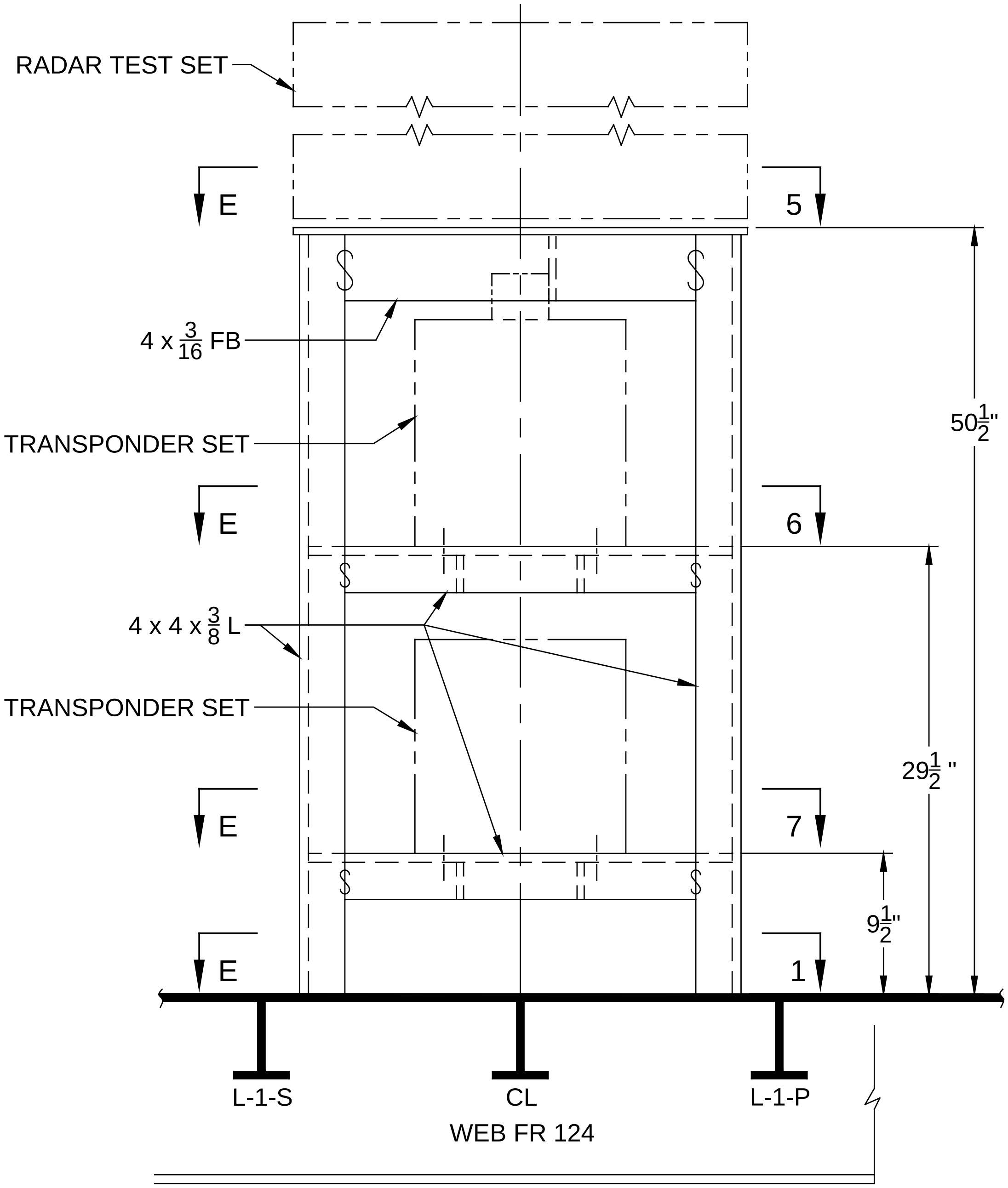


Figure E-5. Scantling Drawing for the Foundation (image 2 of 7).



SECTION E-3

Figure E-6. Scantling Drawing for the Foundation (image 3 of 7).

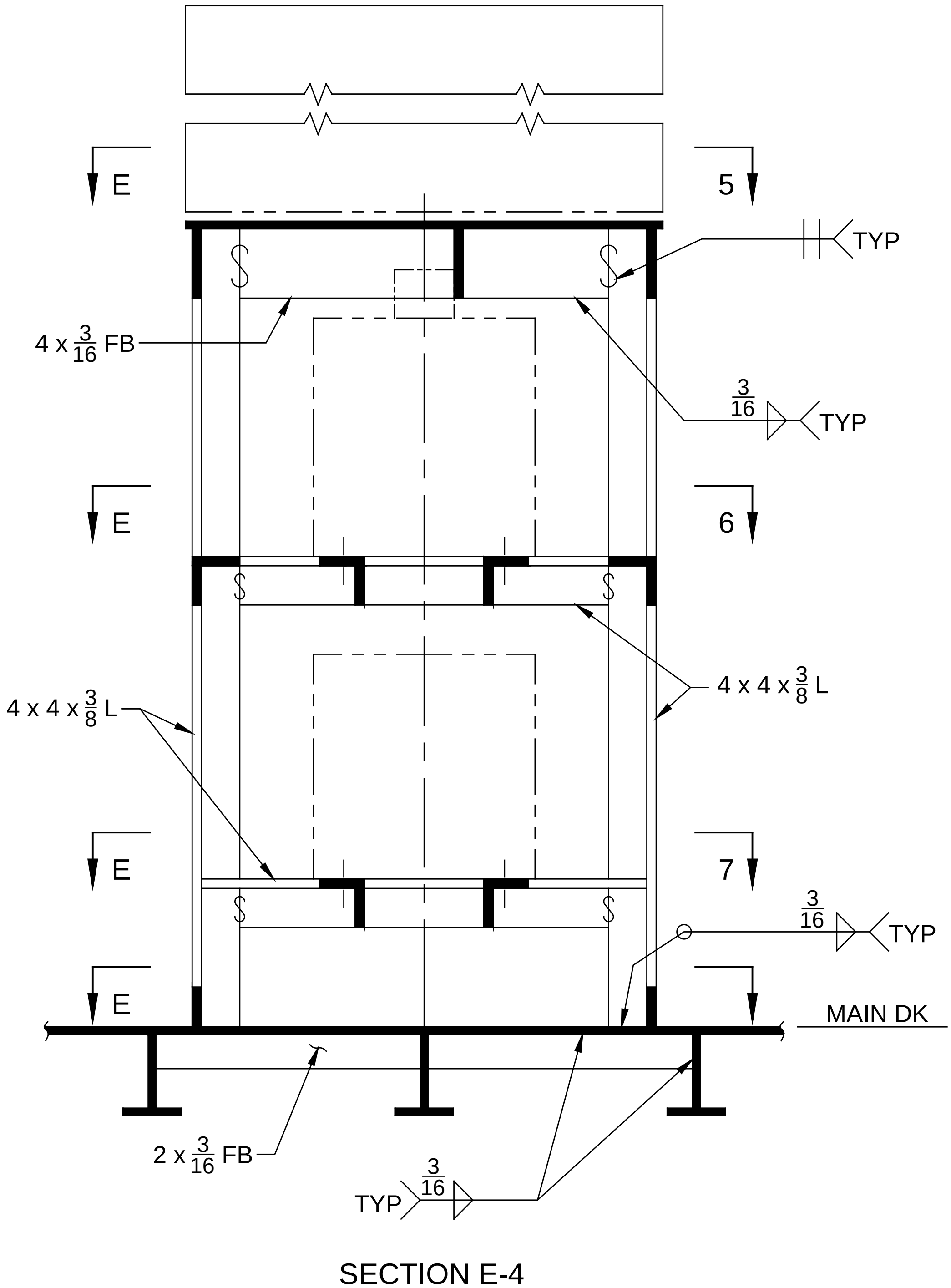


Figure E-7. Scantling Drawing for the Foundation (image 4 of 7).

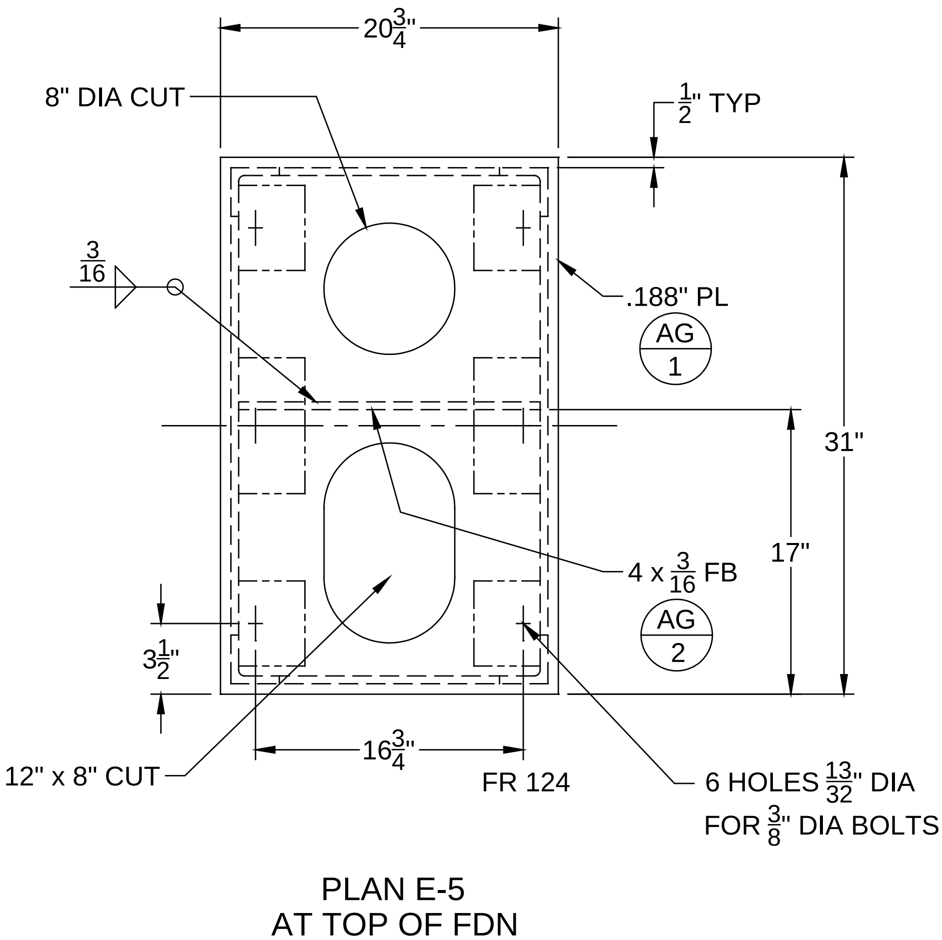


Figure E-8. Scantling Drawing for the Foundation (image 5 of 7).

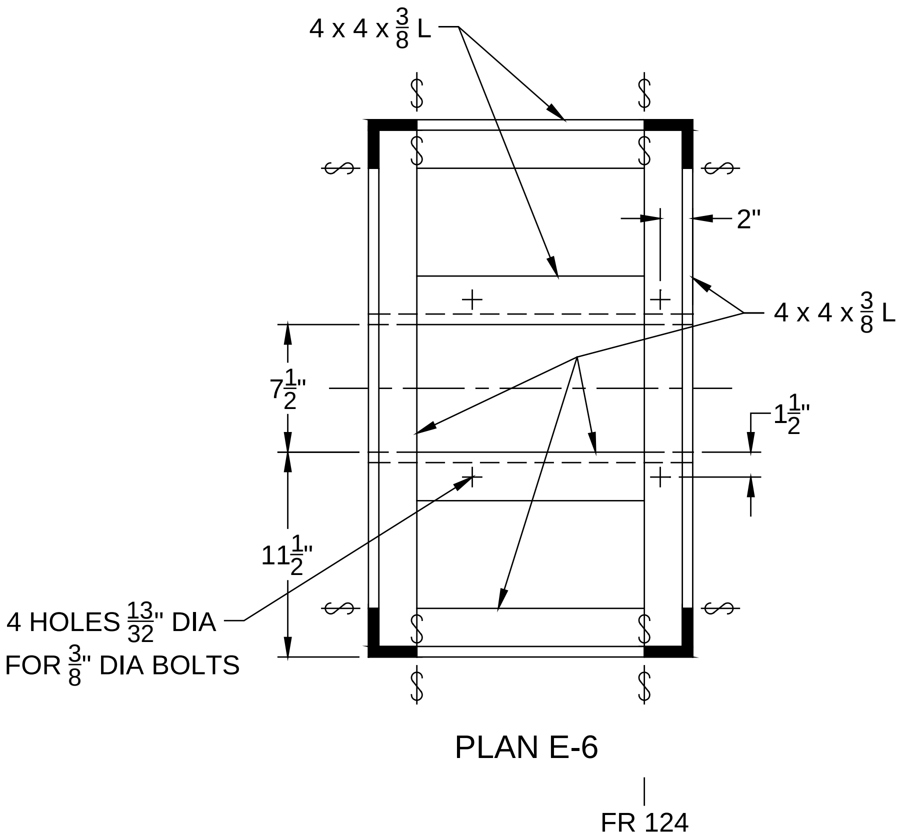


Figure E-9. Scantling Drawing for the Foundation (image 6 of 7).

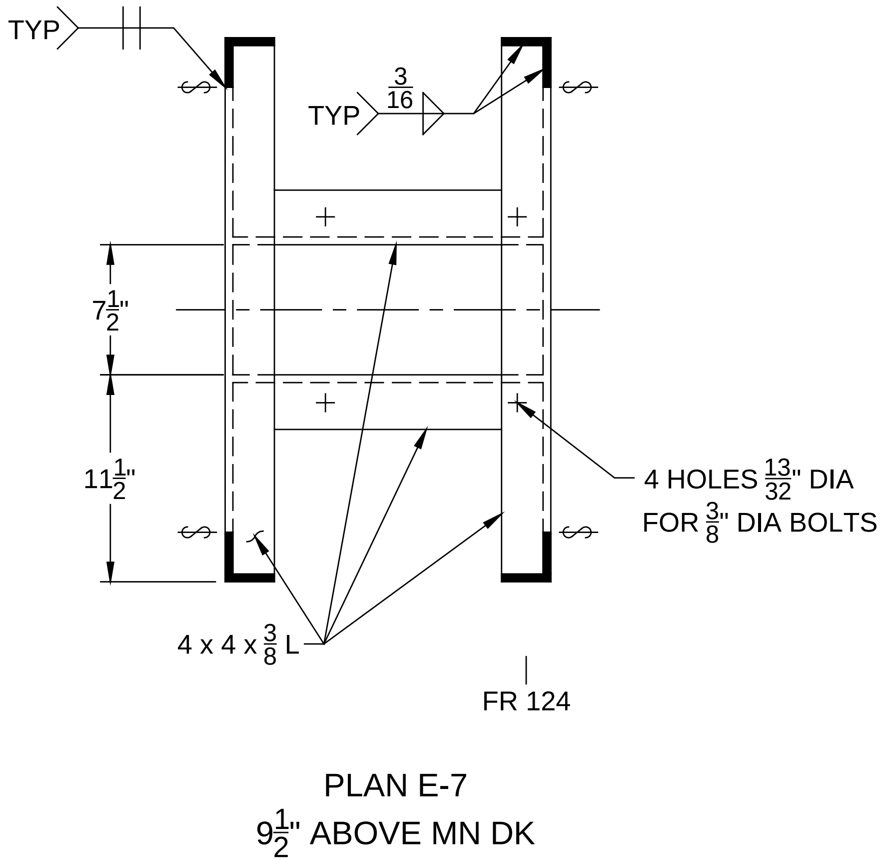


Figure E-10. Scantling Drawing for the Foundation (image 7 of 7).

E.8 MATHEMATICAL MODEL SKETCH (NODE NUMBERS).

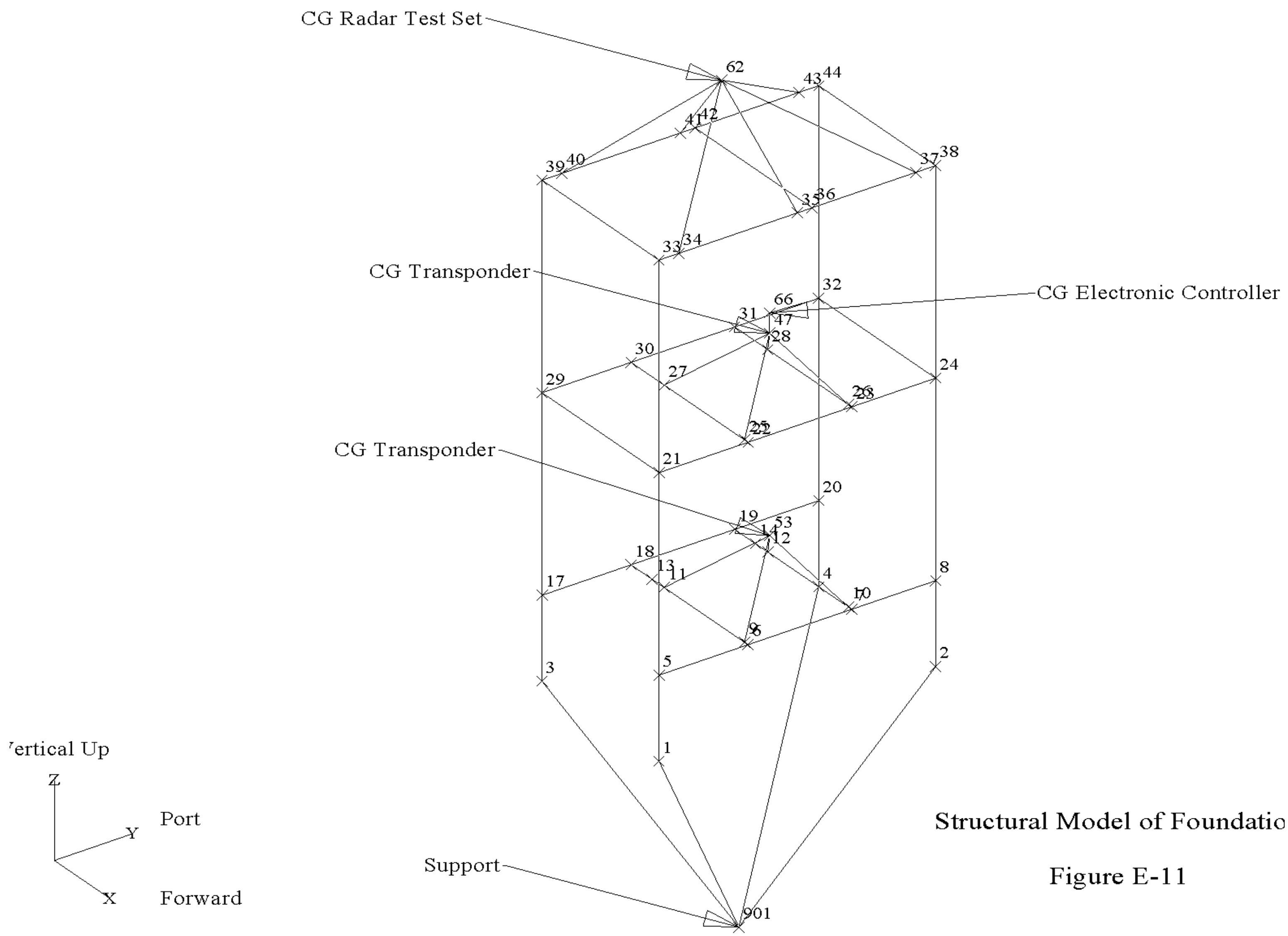


Figure E-11. Structural Model of Foundation.

E.9 <u>MATHEMATICAL MODEL SKETCH (ELEMENT NUMBERS)</u>.

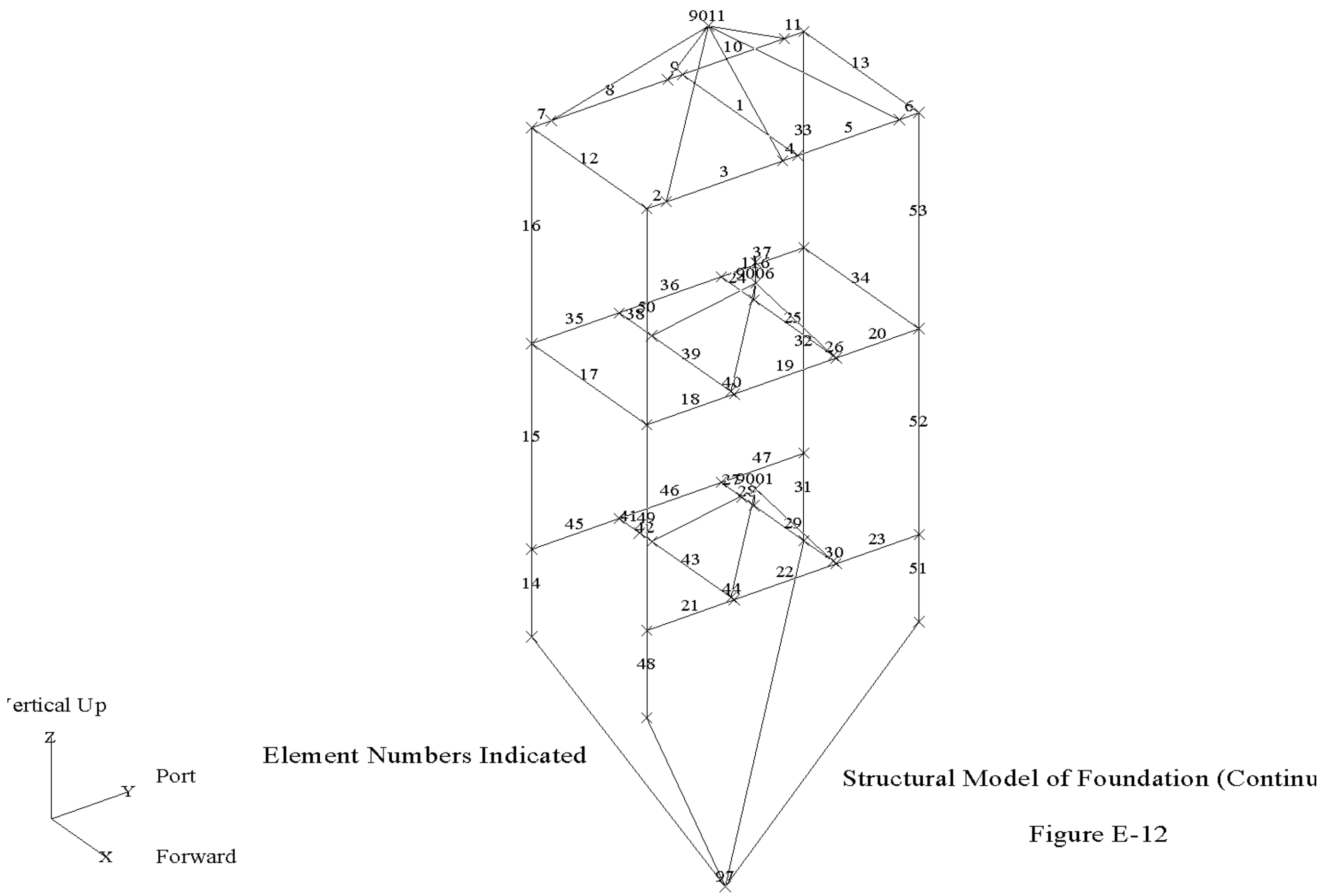


Figure E-12. Structural Model of Foundation.

E.10 MATHEMATICAL MODEL SKETCH (MASS LOCATIONS).

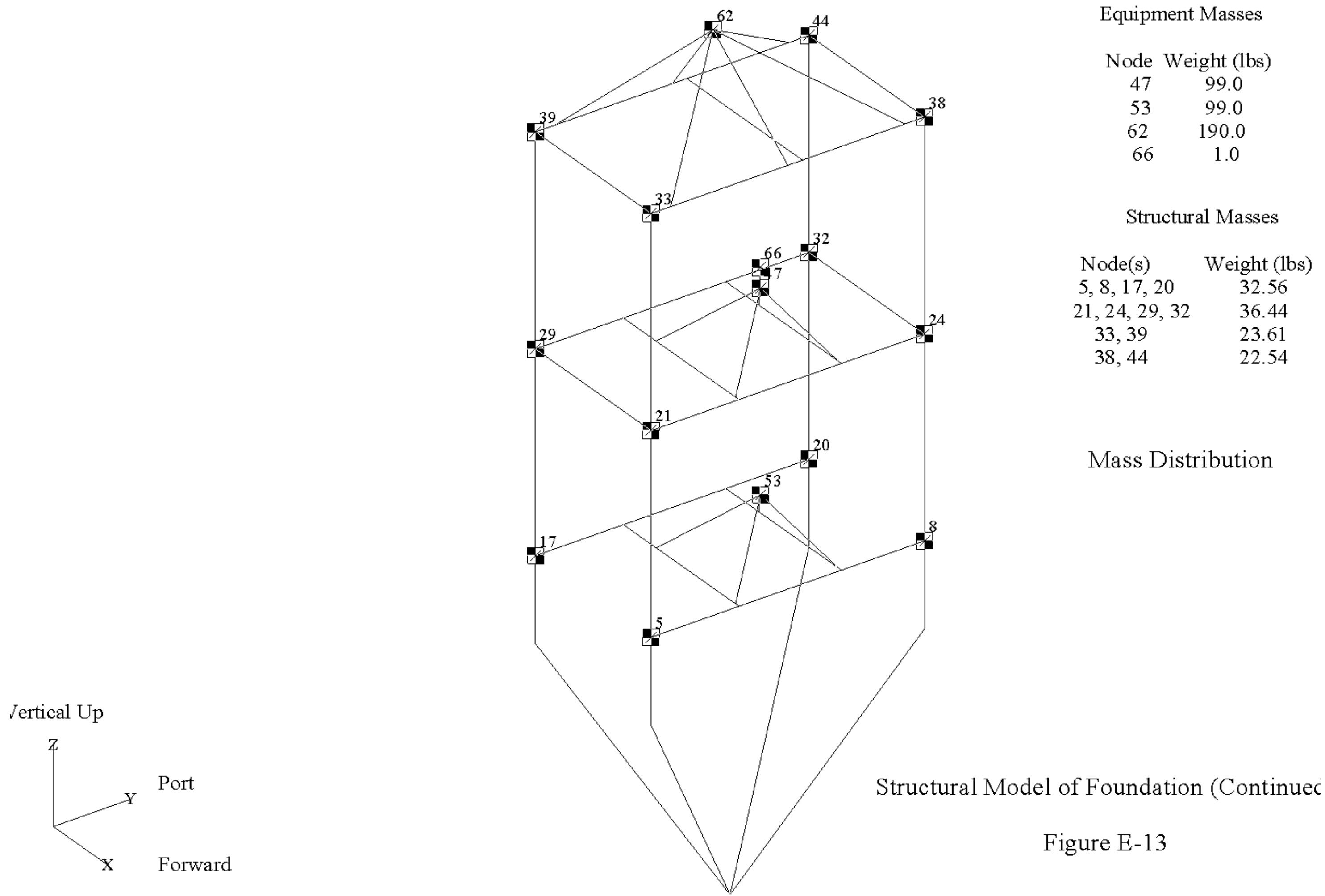


Figure E-13. Structural Model of Foundation.

E.11 COMPUTER INPUT.

```
ID C:\DLR\3,NE/Na
SOL SEMODES
TIME 10000
CEND
METHOD = 1
  ECHO = NONE
  DISPLACEMENT(PLOT) = ALL
  OLOAD(PLOT) = ALL
  SPCFORCE(PLOT) = ALL
  FORCE(PLOT,CORNER) = ALL
  STRESS(PLOT,CORNER) = ALL
  SPC  = 901
BEGIN BULK
$ ***************************************************************************
$   Written by : NE/Nastran for Windows
$   Version    : 8.30
$   Translator : NE/Nastran
$   From Model : C:\DLR\3010\3010Sample.MOD
$   Date       : Wed Jul 14 08:20:24 2004
$   Output To  : C:\DLR\3010\3010Sample
$ ***************************************************************************
$
PARAM,OGEOM,NO
PARAM,AUTOSPC,YES
PARAM,GRDPNT,0
EIGRL          1                      50                            MASS
CORD2C         1       0      0.      0.      0.      0.      0.      1.+NE/NAC1
+NE/NAC1      1.      0.      1.
CORD2S         2       0      0.      0.      0.      0.      0.      1.+NE/NAC2
+NE/NAC2      1.      0.      1.
$ NE/Nastran for Windows Constraint Set 901 : Global Support
SPC          901      66   12456      0.
SPC          901     901  123456      0.
$
$ Element Property syntax is defined as follow:
$     first line, first value is the property identification number.
$     first line, second value is material identification number.
$     first line, third value is the cross section area of the element
$     first line, fourth and fifth values are the moments of inertia about planes 1 and 2
$     first line, sixth value is the torsional constant
$     first line, seventh value is nonstructural mass
$     third line, second and third values are shear areas
$
$ NE/Nastran for Windows Property 1 : 4"x3/16"FB on 5" of 3/16"
PBAR           1     101  1.6875 100000. 2.829320.020111      0.        +PR    1
+PR    1      0.      0.      0.      0.      0.      0.      0.      0.+PA    1
+PA    1 0.46014 0.34765      0.
$ NE/Nastran for Windows Property 2 : 4"x3/16"FB on 6" of 3/16"
PBAR           2     101   1.875 100000.   2.9760.022574      0.        +PR    2
+PR    2      0.      0.      0.      0.      0.      0.      0.      0.+PA    2
+PA    2 0.49901 0.29933      0.
$ NE/Nastran for Windows Property 3 : 4"x3/16"FB on 3" of 3/16"
PBAR           3     101  1.3125 100000. 2.410720.015567      0.        +PR    3
+PR    3      0.      0.      0.      0.      0.      0.      0.      0.+PA    3
+PA    3 0.35581 0.49019      0.
$ NE/Nastran for Windows Property 4 : 4"x3/16"FB on 4" of 3/16"
PBAR           4     101     1.5 100000. 2.646120.017784      0.        +PR    4
+PR    4      0.      0.      0.      0.      0.      0.      0.      0.+PA    4
+PA    4 0.41279 0.40937      0.
$ NE/Nastran for Windows Property 5 : 4"x4"x3/8"L
PBAR           5     101 2.85938 4.35862 4.35862 0.13221      0.        +PR    5
+PR    5      0.      0.      0.      0.      0.      0.      0.      0.+PA    5
+PA    5 0.42947 0.42946-2.58504
$
$ Material Property Syntax is defined as follows:
$     first value is the material identification number.
$     second value defines the modulus of elasticity.
$     third value defines Poisson’s ratio.
$     fourth value defines material density.
$     fifth value defines the thermal expansion coefficient.
$     sixth value defines the reference temperature.
$
```

```
$ NE/Nastran for Windows Material 101 : Steel
MAT1         101 2.96E+7             0.3      0.      0.      0.
$
$ Global Cartesian Coordinate System Origin:
$ +x = 4-3/4 in. fwd of Frame 126
$ +y = 15 in. off ship's CL (stbd)
$ +z = Main Deck
$
$ Grid Coordinate Syntax is defined as follows:
$     first value identifies the grid identification number.
$     second value defines the coordinate system in which the grid point is defined.
$     third value defines the x coord., where +x if forward.
$     fourth value defines the y coord., where +y is port.
$     fifth value defines the z coord., where +z is vertical up.
$     sixth value defines the coordinate system in which the displacements are defined.
$
GRID           1       0   16.75      0.      0.       0
GRID           2       0   16.75     28.      0.       0
GRID           3       0      0.      0.      0.       0
GRID           4       0      0.     28.      0.       0
GRID           5       0   16.75      0.     8.5       0
GRID           6       0   16.75      9.     8.5       0
GRID           7       0   16.75    19.5     8.5       0
GRID           8       0   16.75     28.     8.5       0
GRID           9       0   16.25      9.     8.5       0
GRID          10       0   16.25    19.5     8.5       0
GRID          11       0    4.75      9.     8.5       0
GRID          12       0    4.75    19.5     8.5       0
GRID          13       0      3.      9.     8.5       0
GRID          14       0      3.    19.5     8.5       0
GRID          17       0      0.      0.     8.5       0
GRID          18       0      0.      9.     8.5       0
GRID          19       0      0.    19.5     8.5       0
GRID          20       0      0.     28.     8.5       0
GRID          21       0   16.75      0.    28.5       0
GRID          22       0   16.75      9.    28.5       0
GRID          23       0   16.75    19.5    28.5       0
GRID          24       0   16.75     28.    28.5       0
GRID          25       0   16.25      9.    28.5       0
GRID          26       0   16.25    19.5    28.5       0
GRID          27       0    4.75      9.    28.5       0
GRID          28       0    4.75    19.5    28.5       0
GRID          29       0      0.      0.    28.5       0
GRID          30       0      0.      9.    28.5       0
GRID          31       0      0.    19.5    28.5       0
GRID          32       0      0.     28.    28.5       0
GRID          33       0   16.75      0.    49.5       0
GRID          34       0   16.75      2.    49.5       0
GRID          35       0   16.75     14.    49.5       0
GRID          36       0   16.75    15.5    49.5       0
GRID          37       0   16.75     26.    49.5       0
GRID          38       0   16.75     28.    49.5       0
GRID          39       0      0.      0.    49.5       0
GRID          40       0      0.      2.    49.5       0
GRID          41       0      0.     14.    49.5       0
GRID          42       0      0.    15.5    49.5       0
GRID          43       0      0.     26.    49.5       0
GRID          44       0      0.     28.    49.5       0
GRID          47       0  12.438   14.25    35.5       0
GRID          53       0  12.438   14.25    15.5       0
GRID          62       0    8.33  12.287    59.2       0
GRID          66       0  12.438   14.25    37.5       0
GRID         901       0   8.375     14.    -25.       0
$
$ Element definition syntax is defined as follows:
$     first value defines the element identification number
$     second value defines the property identification number
$     third value defines the "start grid" of the element
$     fourth values defines the "end grid" of the element
$     fifth, sixth, and seventh values defines the x1, y1, z1, respectively, of the element vector
```

```
$
CBAR           1       1      42      36      0.      1.      0.
CBAR           2       2      33      34     -1.      0.      0.
CBAR           3       2      34      35     -1.      0.      0.
CBAR           4       2      35      36     -1.      0.      0.
CBAR           5       2      36      37     -1.      0.      0.
CBAR           6       2      37      38     -1.      0.      0.
CBAR           7       2      40      39      1.      0.      0.
CBAR           8       2      41      40      1.      0.      0.
CBAR           9       2      42      41      1.      0.      0.
CBAR          10       2      43      42      1.      0.      0.
CBAR          11       2      44      43      1.      0.      0.
CBAR          12       3      39      33      0.      1.      0.
CBAR          13       4      38      44      0.     -1.      0.
CBAR          14       5       3      17      1.      0.      0.
CBAR          15       5      17      29      1.      0.      0.
CBAR          16       5      29      39      1.      0.      0.
CBAR          17       5      29      21      0.      0.     -1.
CBAR          18       5      21      22      0.      0.     -1.
CBAR          19       5      22      23      0.      0.     -1.
CBAR          20       5      23      24      0.      0.     -1.
CBAR          21       5       5       6      1.      0.      0.
CBAR          22       5       6       7      1.      0.      0.
CBAR          23       5       7       8      1.      0.      0.
CBAR          24       5      31      28      0.      0.     -1.
CBAR          25       5      28      26      0.      0.     -1.
CBAR          26       5      26      23      0.      0.     -1.
CBAR          27       5      19      14      0.      0.     -1.
CBAR          28       5      14      12      0.      0.     -1.
CBAR          29       5      12      10      0.      0.     -1.
CBAR          30       5      10       7      0.      0.     -1.
CBAR          31       5       4      20      0.     -1.      0.
CBAR          32       5      20      32      0.     -1.      0.
CBAR          33       5      32      44      0.     -1.      0.
CBAR          34       5      32      24      0.     -1.      0.
CBAR          35       5      29      30      1.      0.      0.
CBAR          36       5      30      31      1.      0.      0.
CBAR          37       5      31      32      1.      0.      0.
CBAR          38       5      30      27      0.      0.     -1.
CBAR          39       5      27      25      0.      0.     -1.
CBAR          40       5      25      22      0.      0.     -1.
CBAR          41       5      18      13      0.      0.     -1.
CBAR          42       5      13      11      0.      0.     -1.
CBAR          43       5      11       9      0.      0.     -1.
CBAR          44       5       9       6      0.      0.     -1.
CBAR          45       5      17      18      1.      0.      0.
CBAR          46       5      18      19      1.      0.      0.
CBAR          47       5      19      20      1.      0.      0.
CBAR          48       5       1       5      0.      1.      0.
CBAR          49       5       5      21      0.      1.      0.
CBAR          50       5      21      33      0.      1.      0.
CBAR          51       5       2       8     -1.      0.      0.
CBAR          52       5       8      24     -1.      0.      0.
CBAR          53       5      24      38     -1.      0.      0.
RBE2          97     901  123456       1       2       3       4
CONM1        101       5       0      0.      0.      0.      0.      0.+EL   2T
+EL   2T 0.08435      0.      0.      0.      0.      0.      0.      0.+EA   2T
+EA   2T      0.      0.      0.      0.      0.      0.      0.      0.
CONM1        102       8       0      0.      0.      0.      0.      0.+EL   2U
+EL   2U 0.08435      0.      0.      0.      0.      0.      0.      0.+EA   2U
+EA   2U      0.      0.      0.      0.      0.      0.      0.      0.
CONM1        103      17       0      0.      0.      0.      0.      0.+EL   2V
+EL   2V 0.08435      0.      0.      0.      0.      0.      0.      0.+EA   2V
+EA   2V      0.      0.      0.      0.      0.      0.      0.      0.
CONM1        104      20       0      0.      0.      0.      0.      0.+EL   2W
+EL   2W 0.08435      0.      0.      0.      0.      0.      0.      0.+EA   2W
+EA   2W      0.      0.      0.      0.      0.      0.      0.      0.
CONM1        105      39       0      0.      0.      0.      0.      0.+EL   2X
+EL   2X 0.06117      0.      0.      0.      0.      0.      0.      0.+EA   2X
+EA   2X      0.      0.      0.      0.      0.      0.      0.      0.
CONM1        106      33       0      0.      0.      0.      0.      0.+EL   2Y
```

```
+EL   2Y 0.06117      0.      0.      0.      0.      0.      0.      0.+EA   2Y
+EA   2Y      0.      0.      0.      0.      0.      0.      0.      0.
CONM1        107      44       0      0.      0.      0.      0.      0.+EL   2Z
+EL   2Z  0.0584      0.      0.      0.      0.      0.      0.      0.+EA   2Z
+EA   2Z      0.      0.      0.      0.      0.      0.      0.      0.
CONM1        108      38       0      0.      0.      0.      0.      0.+EL   30
+EL   30  0.0584      0.      0.      0.      0.      0.      0.      0.+EA   30
+EA   30      0.      0.      0.      0.      0.      0.      0.      0.
CONM1        109      21       0      0.      0.      0.      0.      0.+EL   31
+EL   31  0.0944      0.      0.      0.      0.      0.      0.      0.+EA   31
+EA   31      0.      0.      0.      0.      0.      0.      0.      0.
CONM1        110      24       0      0.      0.      0.      0.      0.+EL   32
+EL   32  0.0944      0.      0.      0.      0.      0.      0.      0.+EA   32
+EA   32      0.      0.      0.      0.      0.      0.      0.      0.
CONM1        111      29       0      0.      0.      0.      0.      0.+EL   33
+EL   33  0.0944      0.      0.      0.      0.      0.      0.      0.+EA   33
+EA   33      0.      0.      0.      0.      0.      0.      0.      0.
CONM1        112      32       0      0.      0.      0.      0.      0.+EL   34
+EL   34  0.0944      0.      0.      0.      0.      0.      0.      0.+EA   34
+EA   34      0.      0.      0.      0.      0.      0.      0.      0.
CONM2        113      47       0  0.2565      0.      0.      0.        +EL   35
+EL   35      0.      0.      0.      0.      0.      0.
CONM2        114      53       0  0.2565      0.      0.      0.        +EL   36
+EL   36      0.      0.      0.      0.      0.      0.
CONM2        115      62       0  0.4922      0.      0.      0.        +EL   37
+EL   37      0.      0.      0.      0.      0.      0.
CELAS2       116 11615.8      47       3      66       3      0.
CONM1        117      66       0      0.      0.      0.      0.      0.+EL   39
+EL   392.591E-3      0.      0.      0.      0.      0.      0.      0.+EA   39
+EA   39      0.      0.      0.      0.      0.      0.      0.      0.
RBE2        9001      53     123       9      10      11      12
RBE2        9006      47     123      25      26      27      28
RBE2        9011      62     123      34      35      37      40      41+
+             43
ENDDATA
```

E.12 DDAM OUTPUT.

E.12.1 Frequency, Participation Factors, and Modal Weights for Each Mode Used in the NRL Sum.

Table E-1. Modal Analysis Results With Closely Spaced Modes.

Mode No.	Frequency (Hz)	Vertical Participation Factor	Modal Weight (lb)		
			Longitudinal	Transverse	Vertical
1	56.62	-0.0100	295.82	0.01918	0.0224
2	64.68	0.0026	0.0148	275.70	0.0014
3	198.80	-0.0870	42.68	1.056	1.0609
4	227.43	-0.0187	2.178	36.164	0.0415
5	253.22	-0.0512	45.54	0.171	0.3077
6	323.32	10.5355	0.0217	0.0011	206.37
7	350.79	-11.0740	0.1375	0.0218	275.29
8	380.54	0.8610	1.237	3.6897	63.28
9	401.14	-0.2271	0.0518	74.43	5.5472
10	441.45	0.6362	0.1365	0.0012	92.06
11	676.51	0.1602	0.0020	0.5714	3.349

Table E-2. Modal Analysis Results without Closely Spaced Modes.

Mode No.	Frequency (Hz)	Vertical Participation Factor	Modal Weight (lb)		
			Longitudinal	Transverse	Vertical
1	56.616	-0.0099	295.82	0.0192	0.0220
2	64.68	0.0026	0.0148	275.6973	0.0014
3	198.804	-0.0854	42.7008	1.0570	1.022393
4	227.429	-0.0183	2.1749	36.163	0.0395
5	253.237	-0.0564	45.5134	0.1714	0.3731
6	337.352	1.7753	0.1159	0.0071	459.8850
7	379.979	0.08230	1.2811	3.5690	80.3386
8	401.124	-0.2347	0.0500	70.5675	5.8884
9	440.776	0.5239	0.1397	0.0008	92.4231
10	676.498	0.1602	0.0020	0.5714	3.3495
11	736.586	0.0705	0.10558	0.0007	0.8406

E.12.2 Modal Mass vs. Frequency and Eigenvector vs. Node Number Charts.

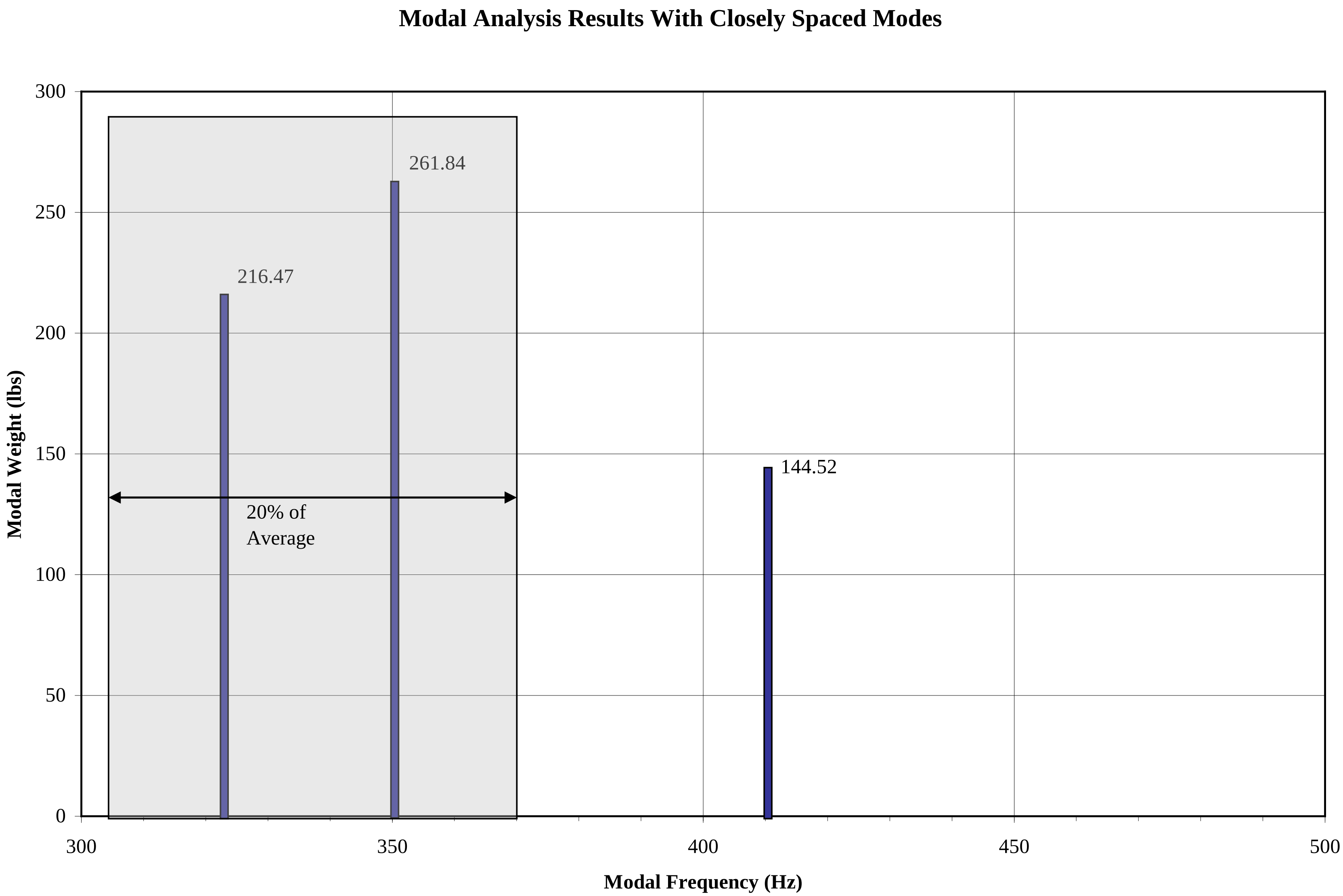


Figure E-14. Modal Analysis Results (with Closely Spaced Modes).

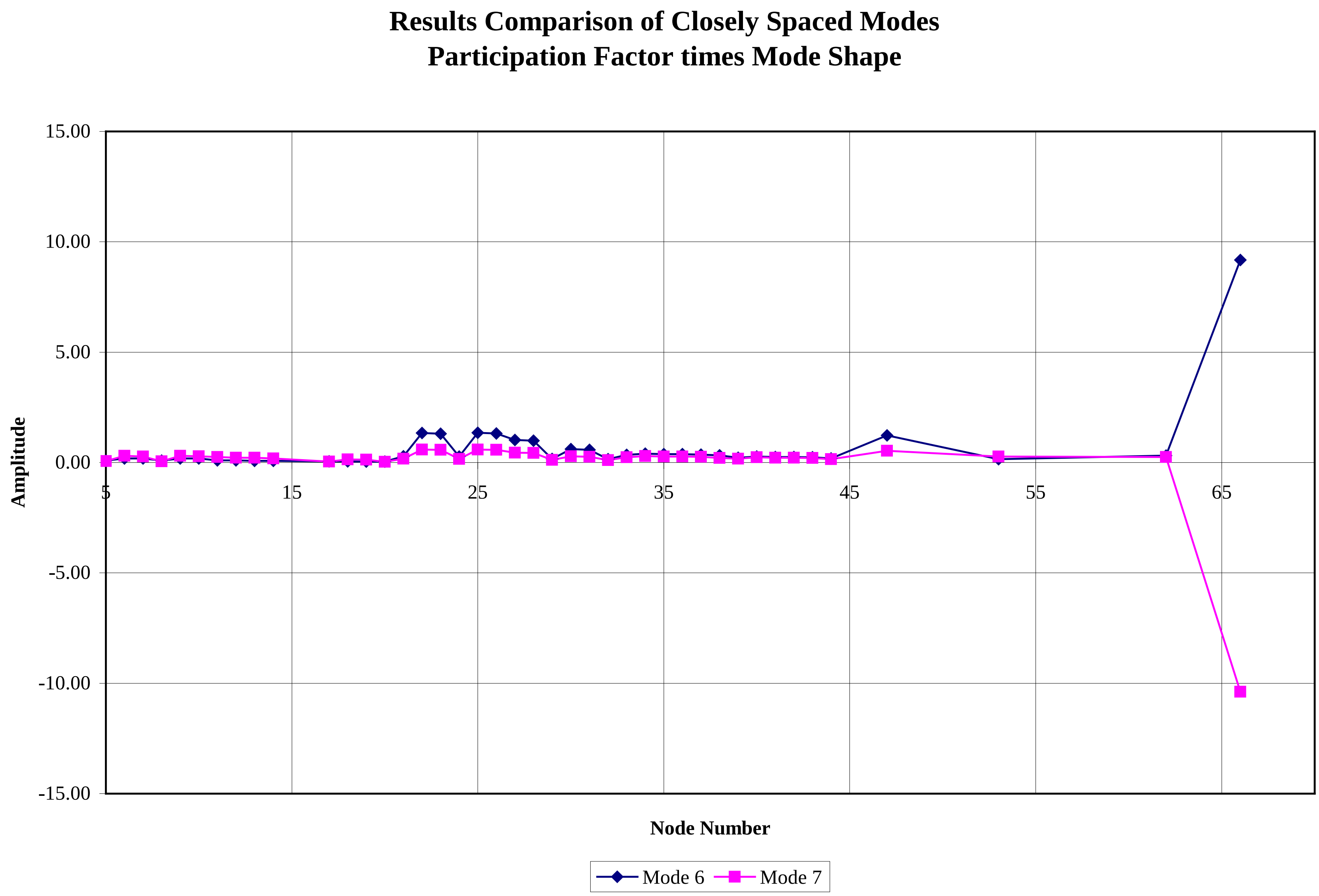


Figure E-15. Results Comparison of Closely Spaced Modes (Participation Factor Times Mode Shape).

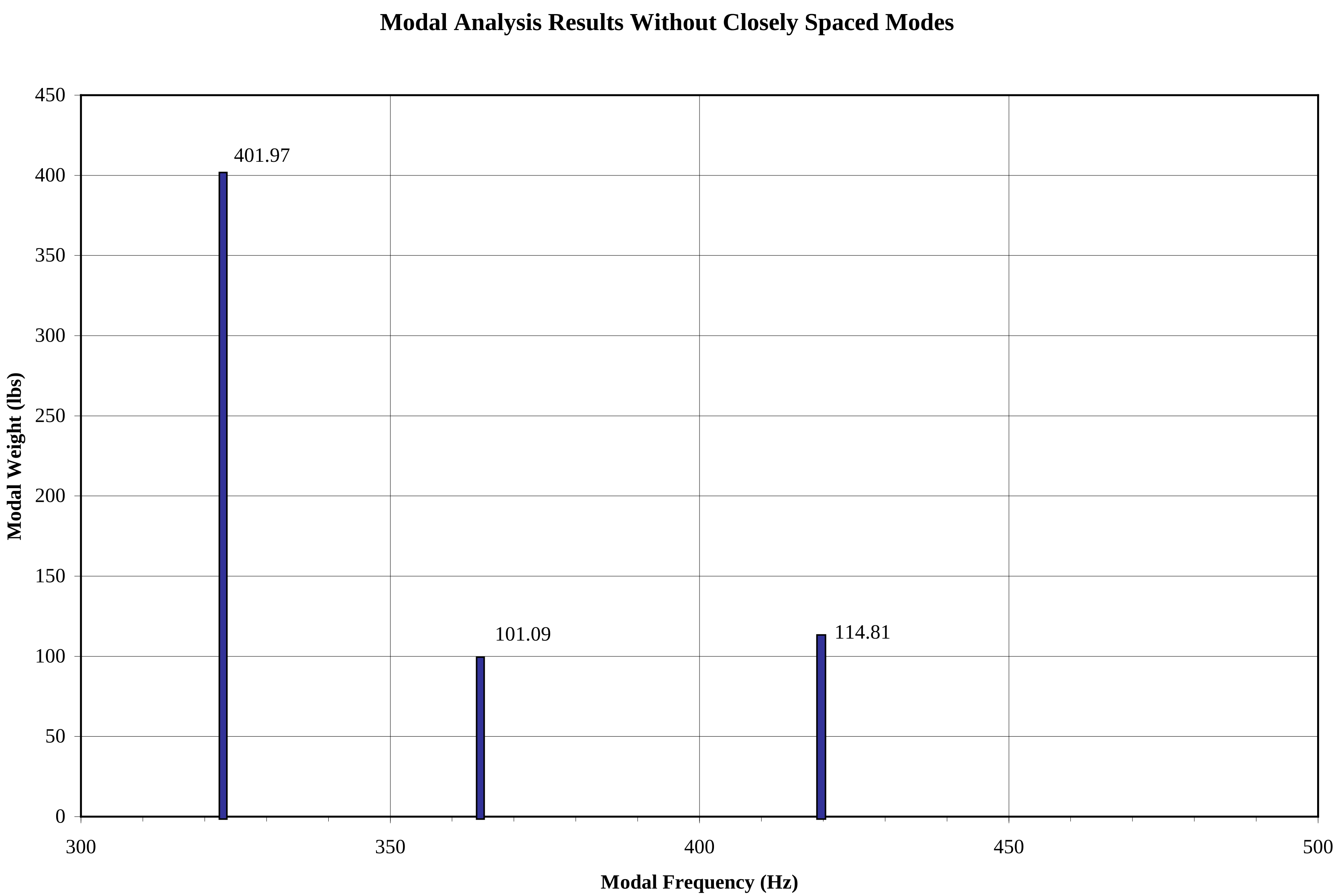


Figure E-16. Modal Analysis Results without Closely Spaced Modes.

E.12.3 Modal Output (Forces).

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SAMPLE PROBLEM WITH CLOSELY SPACE MODES

VERTICAL (Z) DIRECTED SHOCK SUBCASE 3

M O D A L   E F F E C T I V E   W E I G H T

MODE	CYCLES	PARTICIPATION	MODAL		CUMULATIVE	
NUMBER		FACTOR	WEIGHT	PERCENT	WEIGHT	PERCENT
7	3.502345E+02	8.236175E-01	2.618415E+02	34.5754	2.618415E+02	34.5754
6	3.230249E+02	-7.488650E-01	2.164683E+02	28.5840	4.783098E+02	63.1594
8	4.103466E+02	-6.118832E-01	1.445188E+02	19.0833	6.228286E+02	82.2427

MASS AVAILABLE = 100.0000 PERCENT
MASS USED = 82.2427 PERCENT

M O D A L   R E A C T I O N

MODE	CYCLES	PARTICIPATION	RESPONSE		INPUT	
NUMBER		FACTOR	ACCELERATION	REACTION	ACCELERATION	SOURCE
7	3.502345E+02	8.236175E-01	2.145652E+04	1.767196E+04	6.749108E+01	ACCELERATION
6	3.230249E+02	-7.488650E-01	1.963039E+04	-1.470052E+04	6.791069E+01	ACCELERATION
8	4.103466E+02	-6.118832E-01	1.619978E+04	-9.912372E+03	6.858880E+01	ACCELERATION

T O T A L   B A S E   R E A C T I O N

COMPONENT	MAXIMUM MODE	REACTION	SRSS	NRL SUM
1	8	1.484462E+03	5.024374E+02	1.986899E+03
2	8	1.562534E+02	3.451065E+01	1.907641E+02
3	7	1.767196E+04	1.773021E+04	3.540217E+04
4	7	2.350570E+05	2.368720E+05	4.719291E+05
5	7	1.671783E+05	1.653679E+05	3.325462E+05
6	8	2.293471E+04	7.490249E+03	3.042496E+04

P E A K   M O D A L   R E S P O N S E

MODE NUMBER	DISPLACEMENT	VELOCITY	ACCELERATION
7	4.430796E-03	9.750356E+00	2.145652E+04
6	4.765378E-03	9.671931E+00	1.963039E+04
8	2.436956E-03	6.283163E+00	1.619978E+04

SAMPLE PROBLEM WITHOUT CLOSELY SPACED MODES

VERTICAL (Z) DIRECTED SHOCK SUBCASE 3

M O D A L   E F F E C T I V E   W E I G H T

MODE NUMBER	CYCLES	PARTICIPATION FACTOR	MODAL WEIGHT	MODAL PERCENT	CUMULATIVE WEIGHT	CUMULATIVE PERCENT
6	3.226493E+02	-1.020475E+00	4.019684E+02	53.1490	4.019684E+02	53.1490
9	4.197478E+02	5.453701E-01	1.148074E+02	15.1800	5.167759E+02	68.3290
7	3.644335E+02	5.117531E-01	1.010900E+02	13.3663	6.178659E+02	81.6953

MASS AVAILABLE = 100.0000 PERCENT
MASS USED = 81.6953 PERCENT

M O D A L   R E A C T I O N

MODE NUMBER	CYCLES	PARTICIPATION FACTOR	RESPONSE ACCELERATION	RESPONSE REACTION	INPUT ACCELERATION	INPUT SOURCE
6	3.226493E+02	-1.020475E+00	2.608928E+04	-2.662346E+04	6.623270E+01	ACCELERATION
9	4.197478E+02	5.453701E-01	1.449875E+04	7.907187E+03	6.887348E+01	ACCELERATION
7	3.644335E+02	5.117531E-01	1.363119E+04	6.975804E+03	6.900585E+01	ACCELERATION

T O T A L   B A S E   R E A C T I O N

COMPONENT	MAXIMUM MODE	REACTION	SRSS	NRL SUM
1	7	7.951962E+02	4.572364E+02	1.252433E+03
2	7	1.948406E+03	1.233133E+02	2.071719E+03
3	6	2.662346E+04	1.054445E+04	3.716791E+04
4	6	3.636893E+05	1.081825E+05	4.718718E+05
5	6	2.777648E+05	8.160968E+04	3.593745E+05
6	7	1.284791E+04	8.305496E+03	2.115341E+04

P E A K   M O D A L   R E S P O N S E

MODE NUMBER	DISPLACEMENT	VELOCITY	ACCELERATION
6	6.348059E-03	1.286920E+01	2.608928E+04
9	2.084462E-03	5.497463E+00	1.449875E+04
7	2.599785E-03	5.952996E+00	1.363119E+04

MODE = 6 EIGENVALUE = 4.109804E+06 CYCLES = 3.226493E+02 SUBCASE 1

F O R C E S   I N   B A R   E L E M E N T S

ELEMENT	DISTANCE	BENDING MOMENT		SHEAR FORCE		AXIAL	TORQUE
ID		PLANE 1	PLANE 2	PLANE 1	PLANE 2	FORCE	
45	0.0000	1.696027E+03	1.801981E+05	1.215204E+04	2.842751E+04	4.446199E+04	5.119256E+03
	1.0000	-1.076723E+05	-7.564946E+04	1.215204E+04	2.842751E+04	4.446199E+04	5.119256E+03
46	0.0000	-3.506782E+04	-7.660692E+04	1.408713E+02	-2.317263E+03	3.156554E+04	-9.790710E+01
	1.0000	-3.654697E+04	-5.227566E+04	1.408713E+02	-2.317263E+03	3.156554E+04	-9.790710E+01
47	0.0000	-1.085322E+05	-5.127796E+04	-1.294982E+04	-3.039524E+04	4.407849E+04	-5.812191E+03
	1.0000	1.541271E+03	2.070816E+05	-1.294982E+04	-3.039524E+04	4.407849E+04	-5.812191E+03
48	0.0000	-1.939118E+03	2.427565E+05	-7.533949E+03	1.361603E+04	-1.358294E+06	-8.273768E+02
	1.0000	6.209945E+04	1.270203E+05	-7.533949E+03	1.361603E+04	-1.358294E+06	-8.273768E+02
49	0.0000	-6.008499E+05	1.302093E+05	-1.073967E+05	2.374557E+04	-1.198354E+06	9.253647E+02
	1.0000	1.547085E+06	-3.447022E+05	-1.073967E+05	2.374557E+04	-1.198354E+06	9.253647E+02
50	0.0000	-1.532025E+06	-1.646913E+05	-9.887659E+04	-1.020599E+04	-4.377154E+05	-2.366612E+02
	1.0000	5.443834E+05	4.963443E+04	-9.887659E+04	-1.020599E+04	-4.377154E+05	-2.366612E+02
51	0.0000	3.000499E+05	-1.113390E+05	1.679578E+04	-2.285322E+04	-1.318012E+06	1.039116E+03
	1.0000	1.572857E+05	8.291341E+04	1.679578E+04	-2.285322E+04	-1.318012E+06	1.039116E+03
52	0.0000	1.611607E+05	-6.892713E+05	2.776940E+04	-1.188881E+05	-1.144268E+06	-5.672099E+02
	1.0000	-3.942272E+05	1.688491E+06	2.776940E+04	-1.188881E+05	-1.144268E+06	-5.672099E+02
53	0.0000	-1.522926E+05	-1.538658E+06	-8.693604E+03	-9.772908E+04	-3.513384E+05	4.293611E+02
	1.0000	3.027313E+04	5.136523E+05	-8.693604E+03	-9.772908E+04	-3.513384E+05	4.293611E+02

E.12.4 Modal Output (Stresses).

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SAMPLE PROBLEM WITHOUT CLOSELY SPACED MODES

VERTICAL (Z) DIRECTED SHOCK SUBCASE 3

S T R E S S E S I N B A R E L E M E N T S

ELEMENT ID	DISTANCE	SX-C	SX-D	SX-E	SX-F	AXIAL	SX-MAX	SX-MIN
1	0.0000	4.118015E+00	1.089405E+00	0.000000E+00	0.000000E+00	3.361204E-05	4.081385E+00	1.206752E+00
	0.0000	4.184938E+00	1.107110E+00	0.000000E+00	0.000000E+00	3.361204E-05	1.216258E+00	4.151247E+00
2	0.0000	1.917036E+04	4.378643E+03	0.000000E+00	0.000000E+00	1.280785E+03	7.882850E+03	1.567764E+04
	0.0000	1.355433E+04	3.095902E+03	0.000000E+00	0.000000E+00	1.280785E+03	2.884694E+03	1.377262E+04
3	0.0000	1.355433E+04	3.095902E+03	0.000000E+00	0.000000E+00	0.000000E+00	3.895406E+03	1.322828E+04
	0.0000	1.844511E+03	4.212992E+02	0.000000E+00	0.000000E+00	0.000000E+00	1.844509E+03	4.213119E+02
4	0.0000	1.844511E+03	4.212992E+02	0.000000E+00	0.000000E+00	1.321744E+00	1.844141E+03	4.217043E+02
	0.0000	5.962625E+02	1.361905E+02	0.000000E+00	0.000000E+00	1.321744E+00	4.872963E+02	2.604096E+02
5	0.0000	5.961935E+02	1.361747E+02	0.000000E+00	0.000000E+00	1.888206E-01	4.883970E+02	2.590143E+02
	0.0000	9.451726E+03	2.158839E+03	0.000000E+00	0.000000E+00	1.888206E-01	2.158960E+03	9.451605E+03
6	0.0000	9.451726E+03	2.158839E+03	0.000000E+00	0.000000E+00	1.330882E+03	2.490339E+03	1.029295E+04
	0.0000	1.901466E+04	4.343081E+03	0.000000E+00	0.000000E+00	1.330882E+03	8.349142E+03	1.511012E+04
7	0.0000	1.479694E+04	3.379724E+03	0.000000E+00	0.000000E+00	7.230719E+02	6.049407E+03	1.263094E+04
	0.0000	1.641549E+04	3.749412E+03	0.000000E+00	0.000000E+00	7.230719E+02	9.073304E+03	1.134425E+04
8	0.0000	1.945784E+03	4.444305E+02	0.000000E+00	0.000000E+00	0.000000E+00	1.665232E+03	8.417502E+02
	0.0000	1.479694E+04	3.379724E+03	0.000000E+00	0.000000E+00	0.000000E+00	6.548279E+03	1.240293E+04
9	0.0000	6.126377E+02	1.399307E+02	0.000000E+00	0.000000E+00	1.322256E+00	4.915144E+02	2.812439E+02
	0.0000	1.945784E+03	4.444305E+02	0.000000E+00	0.000000E+00	1.322256E+00	1.665954E+03	8.405815E+02
10	0.0000	8.779296E+03	2.005252E+03	0.000000E+00	0.000000E+00	1.888937E-01	3.640663E+03	7.754313E+03
	0.0000	6.127050E+02	1.399461E+02	0.000000E+00	0.000000E+00	1.888937E-01	4.904415E+02	2.826509E+02
11	0.0000	1.570461E+04	3.587041E+03	0.000000E+00	0.000000E+00	7.419199E+02	9.146291E+03	1.019945E+04
	0.0000	8.779296E+03	2.005252E+03	0.000000E+00	0.000000E+00	7.419199E+02	3.215137E+03	7.901840E+03
12	0.0000	2.098075E+03	7.984985E+02	0.000000E+00	0.000000E+00	2.214076E+00	2.047095E+03	9.092782E+02
	0.0000	2.092408E+03	7.963417E+02	0.000000E+00	0.000000E+00	2.214076E+00	9.069667E+02	2.040315E+03
13	0.0000	1.510943E+03	4.726555E+02	0.000000E+00	0.000000E+00	2.107620E+00	6.257975E+02	1.453554E+03
	0.0000	1.476240E+03	4.617997E+02	0.000000E+00	0.000000E+00	2.107620E+00	1.411601E+03	6.239277E+02
14	0.0000	4.043820E+03	5.378617E+03	3.505223E+03	0.000000E+00	1.046621E+04	1.047514E+04	1.073472E+04
	0.0000	6.193451E+03	3.840364E+03	1.123537E+04	0.000000E+00	1.046621E+04	1.810458E+04	1.107352E+04

VERTICAL (Z) DIRECTED SHOCK SUBCASE 3

S T R E S S E S I N B A R E L E M E N T S

ELEMENT ID	DISTANCE	SX-C	SX-D	SX-E	SX-F	AXIAL	SX-MAX	SX-MIN
15	0.0000	3.027650E+03	4.187568E+03	6.796524E+03	0.000000E+00	9.699770E+03	1.104158E+04	1.215325E+04
	0.0000	7.898875E+03	2.436800E+03	1.022140E+04	0.000000E+00	9.699770E+03	7.655664E+03	1.365584E+04
16	0.0000	3.530045E+03	7.974337E+03	1.319775E+04	0.000000E+00	6.547973E+03	1.377774E+04	1.217647E+04
	0.0000	3.603027E+03	4.779574E+03	1.017559E+04	0.000000E+00	6.547973E+03	9.226533E+03	1.041319E+04
17	0.0000	5.667869E+02	4.699608E+03	4.205936E+03	0.000000E+00	8.319081E+00	4.200954E+03	4.702975E+03
	0.0000	5.955047E+02	4.725011E+03	4.234411E+03	0.000000E+00	8.319081E+00	4.721900E+03	4.239336E+03
18	0.0000	2.424285E+04	5.727634E+04	2.063188E+04	0.000000E+00	5.539015E+02	2.383624E+04	5.770993E+04
	0.0000	2.020847E+04	4.608812E+04	1.556929E+04	0.000000E+00	5.539015E+02	4.568874E+04	2.059612E+04
19	0.0000	1.948018E+04	4.667246E+04	1.719336E+04	0.000000E+00	3.736411E+02	4.636741E+04	1.978307E+04
	0.0000	1.877235E+04	4.502952E+04	1.661321E+04	0.000000E+00	3.736411E+02	4.470381E+04	1.909773E+04
20	0.0000	1.946577E+04	4.449427E+04	1.513398E+04	0.000000E+00	8.866461E+02	4.405196E+04	1.989685E+04
	0.0000	2.422140E+04	5.748427E+04	2.086286E+04	0.000000E+00	8.866461E+02	2.375877E+04	5.802173E+04
21	0.0000	1.648265E+04	1.656534E+04	4.151919E+04	0.000000E+00	2.510799E+03	3.635248E+04	2.178238E+04
	0.0000	1.101605E+04	1.824312E+04	3.448559E+04	0.000000E+00	2.510799E+03	2.477482E+04	2.888884E+04
22	0.0000	1.172963E+04	1.603372E+04	3.365743E+04	0.000000E+00	9.131072E+02	2.124393E+04	2.912418E+04
	0.0000	6.154704E+03	8.188742E+03	1.735723E+04	0.000000E+00	9.131072E+02	1.308575E+04	1.308949E+04
23	0.0000	5.032522E+03	1.159647E+04	1.866130E+04	0.000000E+00	9.416783E+02	1.583896E+04	1.527693E+04
	0.0000	1.338828E+04	1.346489E+04	3.373403E+04	0.000000E+00	9.416783E+02	2.598034E+04	2.178739E+04
24	0.0000	9.398006E+02	4.640097E+02	1.886644E+03	0.000000E+00	3.911356E+02	6.264012E+02	2.241308E+03
	0.0000	8.743939E+03	2.283219E+04	9.638090E+03	0.000000E+00	3.911356E+02	2.246347E+04	1.002773E+04
25	0.0000	8.743939E+03	2.283219E+04	9.638091E+03	0.000000E+00	0.000000E+00	2.283219E+04	9.649211E+03
	0.0000	3.110497E+03	5.454633E+03	9.559397E+02	0.000000E+00	0.000000E+00	5.472214E+03	3.110497E+03
26	0.0000	3.110497E+03	5.454633E+03	9.559396E+02	0.000000E+00	3.850118E+02	5.813846E+03	2.757649E+03
	0.0000	8.198176E+02	7.614095E+02	2.002272E+03	0.000000E+00	3.850118E+02	2.383308E+03	4.405814E+02
27	0.0000	7.836970E+02	8.226190E+02	2.007562E+03	0.000000E+00	4.766322E+02	2.395405E+03	8.356415E+02
	0.0000	2.039972E+03	7.353321E+03	4.469431E+03	0.000000E+00	4.766322E+02	5.984338E+03	6.033313E+03
28	0.0000	2.039972E+03	7.353321E+03	4.469431E+03	0.000000E+00	4.766322E+02	5.984338E+03	6.033313E+03
	0.0000	3.370589E+03	1.154729E+04	6.465810E+03	0.000000E+00	4.766322E+02	8.853020E+03	9.284296E+03

SAMPLE PROBLEM WITHOUT CLOSELY SPACED MODES

VERTICAL (Z) DIRECTED SHOCK SUBCASE 3

S T R E S S E S I N B A R E L E M E N T S

ELEMENT ID	DISTANCE	SX-C	SX-D	SX-E	SX-F	AXIAL	SX-MAX	SX-MIN
29	0.0000	3.370589E+03	1.154729E+04	6.465810E+03	0.000000E+00	0.000000E+00	8.469474E+03	9.664185E+03
	0.0000	1.844935E+03	1.951521E+03	3.282524E+03	0.000000E+00	0.000000E+00	2.858855E+03	3.356038E+03
30	0.0000	1.844935E+03	1.951521E+03	3.282524E+03	0.000000E+00	3.187445E+02	2.586864E+03	3.645501E+03
	0.0000	1.514473E+03	1.519861E+03	3.808435E+03	0.000000E+00	3.187445E+02	1.528794E+03	3.990882E+03
31	0.0000	2.701671E+03	4.757367E+03	7.242859E+03	0.000000E+00	8.653567E+03	9.442029E+03	9.991329E+03
	0.0000	3.654107E+03	5.795760E+03	3.088123E+03	0.000000E+00	8.653567E+03	1.055456E+04	8.416767E+03
32	0.0000	4.225714E+03	9.120777E+03	4.691101E+03	0.000000E+00	8.076248E+03	9.313692E+03	1.368822E+04
	0.0000	7.986735E+03	1.024723E+04	3.249225E+03	0.000000E+00	8.076248E+03	6.327569E+03	1.346562E+04
33	0.0000	3.609755E+03	1.312338E+04	7.659280E+03	0.000000E+00	5.066478E+03	1.279759E+04	1.088432E+04
	0.0000	3.523339E+03	9.655802E+03	4.366390E+03	0.000000E+00	5.066478E+03	8.261559E+03	8.028643E+03
34	0.0000	9.321414E+02	4.326622E+03	5.472385E+03	0.000000E+00	9.406982E+00	4.328604E+03	5.468023E+03
	0.0000	9.374677E+02	4.418512E+03	5.586721E+03	0.000000E+00	9.406982E+00	5.590814E+03	4.416720E+03
35	0.0000	8.768209E+03	1.253097E+04	2.572825E+04	0.000000E+00	4.355361E+02	1.265963E+04	2.564628E+04
	0.0000	6.550320E+03	1.164443E+04	2.145497E+04	0.000000E+00	4.355361E+02	2.156681E+04	1.150681E+04
36	0.0000	7.436548E+03	9.417621E+03	2.067207E+04	0.000000E+00	5.645905E+02	2.095578E+04	9.147101E+03
	0.0000	6.961629E+03	8.951138E+03	1.943584E+04	0.000000E+00	5.645905E+02	1.970288E+04	8.664642E+03
37	0.0000	6.122294E+03	1.088595E+04	2.012866E+04	0.000000E+00	5.252679E+02	2.024663E+04	1.076799E+04
	0.0000	8.251195E+03	1.179016E+04	2.421828E+04	0.000000E+00	5.252679E+02	1.194767E+04	2.415619E+04
38	0.0000	8.109274E+02	1.272647E+03	2.496896E+03	0.000000E+00	3.881016E+02	2.109042E+03	1.660712E+03
	0.0000	9.705578E+03	2.354672E+04	8.853144E+03	0.000000E+00	3.881016E+02	2.317494E+04	1.008755E+04
39	0.0000	9.705578E+03	2.354672E+04	8.853144E+03	0.000000E+00	0.000000E+00	2.354672E+04	9.714790E+03
	0.0000	1.682752E+03	6.716747E+03	4.221441E+03	0.000000E+00	0.000000E+00	6.716747E+03	4.221441E+03
40	0.0000	1.682752E+03	6.716747E+03	4.221441E+03	0.000000E+00	3.955731E+02	7.085465E+03	3.843165E+03
	0.0000	7.449572E+02	7.850286E+02	1.912381E+03	0.000000E+00	3.955731E+02	1.187853E+03	1.517777E+03
41	0.0000	2.761876E+03	3.011433E+03	7.184979E+03	0.000000E+00	4.061931E+02	3.572802E+03	6.739146E+03
	0.0000	5.304461E+03	1.080320E+04	2.940216E+03	0.000000E+00	4.061931E+02	7.028999E+03	9.475938E+03
42	0.0000	5.304461E+03	1.080320E+04	2.940216E+03	0.000000E+00	4.061931E+02	7.028999E+03	9.475938E+03
	0.0000	7.101849E+03	1.860166E+04	7.868687E+03	0.000000E+00	4.061931E+02	1.034286E+04	1.627302E+04

VERTICAL (Z) DIRECTED SHOCK SUBCASE 3

S T R E S S E S   I N   B A R   E L E M E N T S

ELEMENT ID	DISTANCE	SX-C	SX-D	SX-E	SX-F	AXIAL	SX-MAX	SX-MIN
43	0.0000	7.101849E+03	1.860166E+04	7.868687E+03	0.000000E+00	0.000000E+00	1.001600E+04	1.659958E+04
	0.0000	9.916300E+02	5.960103E+03	5.379214E+03	0.000000E+00	0.000000E+00	5.599930E+03	5.863573E+03
44	0.0000	9.916300E+02	5.960103E+03	5.379214E+03	0.000000E+00	4.941808E+02	5.193129E+03	6.303174E+03
	0.0000	1.072974E+03	9.780746E+02	2.598633E+03	0.000000E+00	4.941808E+02	2.013298E+03	1.687953E+03
45	0.0000	7.102593E+03	7.187905E+03	1.794068E+04	0.000000E+00	1.195305E+03	1.640549E+04	8.766213E+03
	0.0000	2.807082E+03	1.322499E+04	1.618063E+04	0.000000E+00	1.195305E+03	1.498999E+04	1.457468E+04
46	0.0000	4.970923E+03	6.481195E+03	1.380907E+04	0.000000E+00	3.829393E+02	8.014425E+03	1.244199E+04
	0.0000	2.167524E+03	6.826570E+03	9.898588E+03	0.000000E+00	3.829393E+02	8.178559E+03	8.625162E+03
47	0.0000	2.006147E+03	8.528019E+03	1.029315E+04	0.000000E+00	3.986624E+02	9.364082E+03	9.600062E+03
	0.0000	5.871881E+03	5.957164E+03	1.484674E+04	0.000000E+00	3.986624E+02	1.249310E+04	8.349336E+03
48	0.0000	3.345956E+03	9.457748E+03	8.729570E+03	0.000000E+00	1.442048E+04	1.779636E+04	1.837875E+04
	0.0000	1.000249E+04	2.632302E+04	1.246772E+04	0.000000E+00	1.442048E+04	3.641576E+04	1.927740E+04
49	0.0000	6.551358E+03	1.706167E+04	7.645128E+03	0.000000E+00	1.203340E+04	1.102538E+04	2.043161E+04
	0.0000	9.576200E+03	3.319446E+04	1.879437E+04	0.000000E+00	1.203340E+04	1.234381E+04	3.990262E+04
50	0.0000	1.227790E+04	2.659519E+04	8.122473E+03	0.000000E+00	6.881007E+03	2.652987E+04	1.508586E+04
	0.0000	4.927890E+03	1.116454E+04	3.773734E+03	0.000000E+00	6.881007E+03	7.909710E+03	1.289902E+04
51	0.0000	4.422255E+03	6.717017E+03	8.589457E+03	0.000000E+00	1.233745E+04	1.417873E+04	1.656522E+04
	0.0000	4.794189E+03	7.147607E+03	1.255451E+04	0.000000E+00	1.233745E+04	2.038028E+04	1.573723E+04
52	0.0000	8.635657E+03	9.880715E+03	2.256991E+04	0.000000E+00	1.050577E+04	1.204416E+04	2.574497E+04
	0.0000	1.022092E+04	1.975128E+04	3.490996E+04	0.000000E+00	1.050577E+04	1.275498E+04	4.200862E+04
53	0.0000	1.242176E+04	8.308047E+03	2.706996E+04	0.000000E+00	5.370495E+03	2.653244E+04	1.457988E+04
	0.0000	4.798709E+03	3.953770E+03	1.116139E+04	0.000000E+00	5.370495E+03	7.302486E+03	1.093253E+04

MAXIMUM BAR ELEMENT TOTAL STRESS = 4.658087E+04 AT ELEMENT 19
MINIMUM BAR ELEMENT TOTAL STRESS = 4.187817E+00 AT ELEMENT 1

APPENDIX F
EXAMPLE APPLICATION OF THE ALLOWABLE STRESS CRITERIA

F.1 INTRODUCTION.

Allowable stress criteria used to evaluate the adequacy of structures under shock loading are detailed in Chapter 6. Examples for their application are provided in Tables F-1 and F-2.

In general, all stresses exceeding allowable stress criteria of Chapter 6 must be documented in the DDAM report. Many of these overstresses may be found acceptable per the proper evaluation of the stress and choice of the acceptance criteria using this appendix.

The first concept which must be understood is that shock is a one-time event and that peak stresses caused by stress concentrations, which are a concern for fatigue, are not a concern for shock. When evaluating membrane stresses, any peak stresses at locations of stress concentrations as defined in Table F-1 have no stress limitation and may be accepted as long as the membrane stress in the overall cross-section and through the penetration meet the membrane stress criteria.

The next concept which must be understood is that a membrane stress in a complex finite element analysis may occur due to a membrane mode or a bending mode. The classic example of this would be an I-beam constructed from plate elements. The flanges of the I-beam will have membrane stresses in the upper and lower flanges. The membrane stress may be caused by a pure tensile mode or it may be caused by a bending mode of the I-beam. In the case of a pure tensile mode of the beam, the proper allowable stress would be membrane stress allowables. In the case of a bending mode of the beam, a case could be made to use membrane plus bending allowables to assess the membrane stress in the flanges. A conservative approach would be to evaluate the stresses in the flanges for membrane allowables and only if the membrane allowables are exceeded would the next step be taken to try and evaluate which modes the membrane plus bending allowables might be considered.

When evaluating whether a stress is "local", the stress in the thickness of interest must be varying through that thickness otherwise the stress should be considered "general".

Table F-1. Example Applications of the Allowable Stress Criteria.

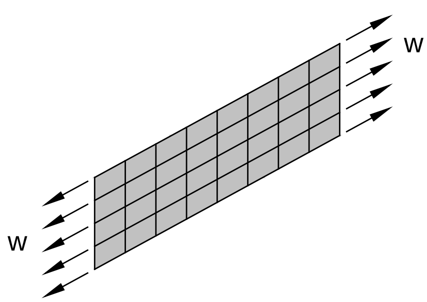
Note: S_a = Allowable Design Stress Stress Region is Shaded	
Description of Load Types and Stress Regions	**Remarks**
1 	General membrane

Table F-1. Example Applications of the Allowable Stress Criteria – Continued.

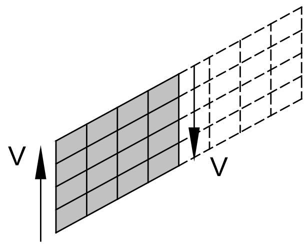
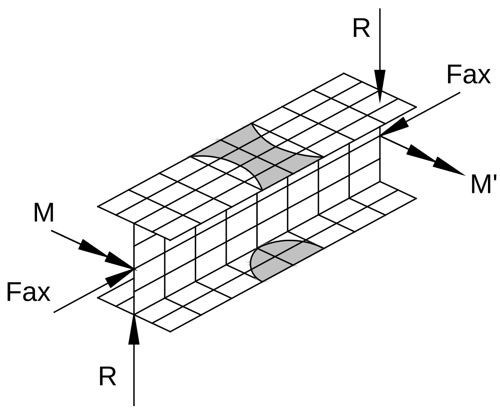
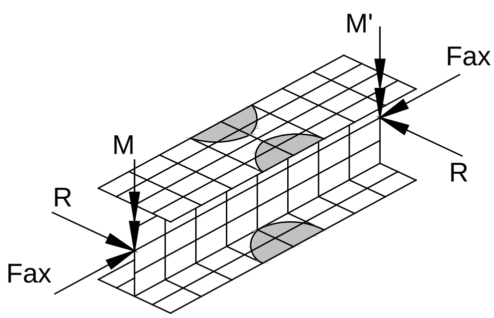
Description of Load Types and Stress Regions	Remarks
2 	General membrane For structural cross-sections, stress in segments typically thought of as shear load carrying members (i.e., segments parallel to the direction of the shear load) must remain less than $S_{a.}$
3 	General membrane plus bending Bending stress is varying across the cross-section. Stresses evaluated at the outermost fibers must remain less than $S_{a.}$ The web of the cross-section is the shear load carrying segment.
4 	General membrane plus bending Bending stress is varying across the cross-section. Stresses evaluated at the outermost fibers must remain less than S_a. The flanges of the cross-section are the shear load carrying segments.

Table F-1. Example Applications of the Allowable Stress Criteria – Continued.

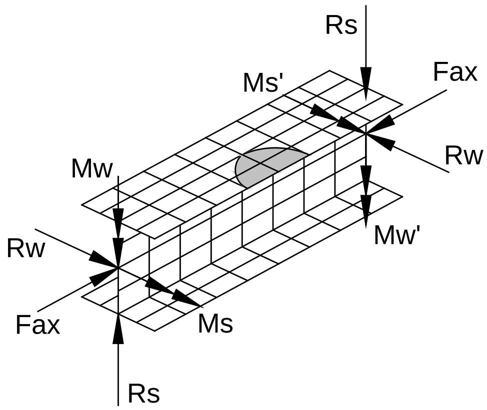
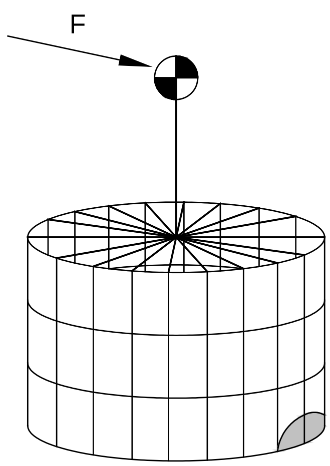
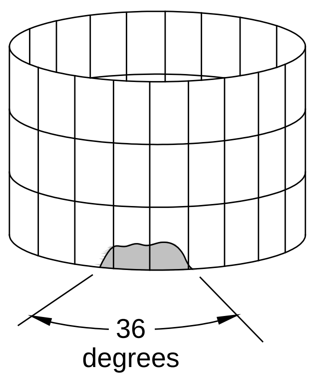
Description of Load Types and Stress Regions	Remarks
5 	General membrane plus bending Bending stress is varying across the cross-section. Stresses evaluated at the outermost fibers must remain less than S_a. The web and flanges of the cross-section are the shear load carrying segments.
6 	General membrane plus bending The bending stress distribution is through the entire cross-section. The stress is in the section must remain less than S_a.
7 	Local membrane plus bending The local bending stress distribution is varying through the plate thickness due to a structural discontinuity. The stress in any 36-degree section (10 percent of effective area) must remain less than S_a.

Table F-1. Example Applications of the Allowable Stress Criteria – Continued.

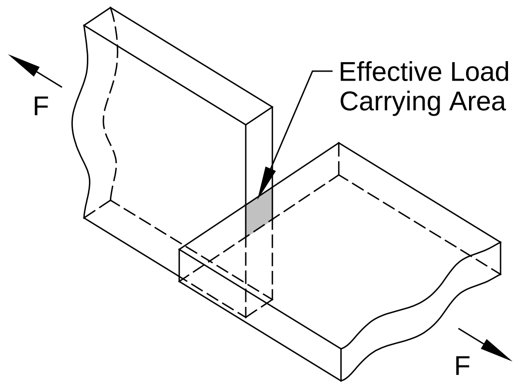
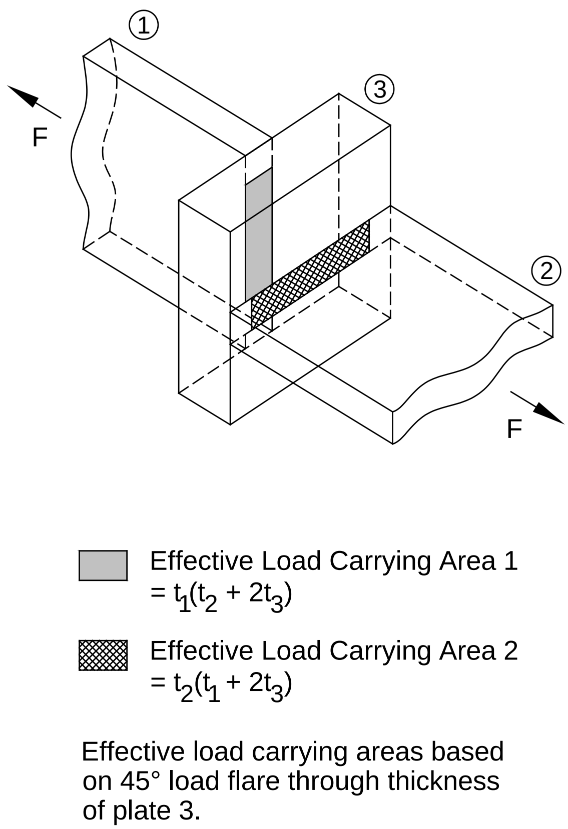
Description of Load Types and Stress Regions	Remarks
8 	General membrane Stress levels in the effective load carrying area must remain less than S_a.
9 	General membrane Stress levels in the effective load carrying area must remain less than S_a.

Table F-1. Example Applications of the Allowable Stress Criteria – Continued.

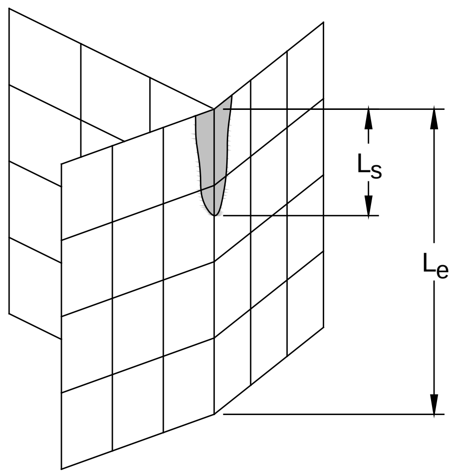
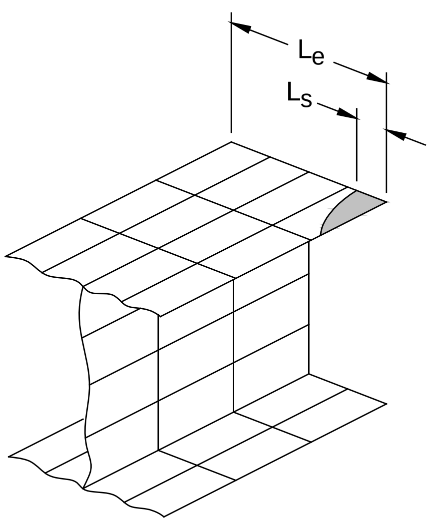
Description of Load Types and Stress Regions	Remarks
10  L_e = Total length of intersection L_s = Length of intersection over which stress exceeds general allowable	Structural discontinuity-web intersection Local membrane plus bending No greater than 10 percent of the length of the line formed by the intersection of the plates may experience stress greater than the general allowables. $\frac{L_s}{L_e} < 0.10$ (10%)
11  L_e = Total length of intersection L_s = Length of intersection over which stresses exceed general allowables	Local membrane plus bending The local bending stress distribution is varying through the flange thickness due to a structural discontinuity. No greater than 10 percent of the length of the boundary of the flange may experience stress greater than the general allowable. $\frac{L_s}{L_e} < 0.10$ (10%)

Table F-1. Example Applications of the Allowable Stress Criteria – Continued.

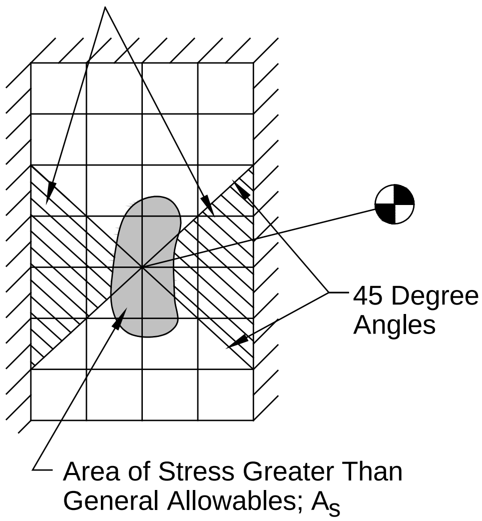
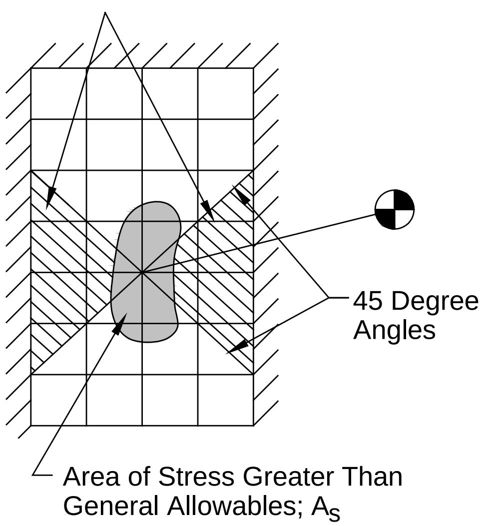
Description of Load Types and Stress Regions	Remarks
12 Effective load carrying area of the section, 45-degree load flares, A_e  	Local membrane plus bending The area of stress A_s in which general allowables are exceeded must not exceed 10 percent of the effective load carrying area of the section A_e. $\frac{A_s}{A_e} < 0.10$ (10%) In order for a load path to be effective, at least one of the two angles created by the flare boundaries and the edge of the plate must be 45 degrees. Average shear stresses from tear-out and punch through calculations are limited to general membrane allowables.
13 Effective load carrying area of the section, 45-Degree Load Flares, A_e  	Local membrane plus bending In cases where local allowables are exceeded, the area of stress, A_s, greater than the local allowables must not exceed 5 percent of the effective load carrying area of the section, A_e. $\frac{A_s}{A_e} < 0.05$ (5%) In order for a load path to be effective, at least one of the two angles created by the flare boundaries and the edge of the plate must be 45 degrees. Average shear stresses from tear-out and punch through calculations are limited to general membrane allowables

Table F-1. Example Applications of the Allowable Stress Criteria – Continued.

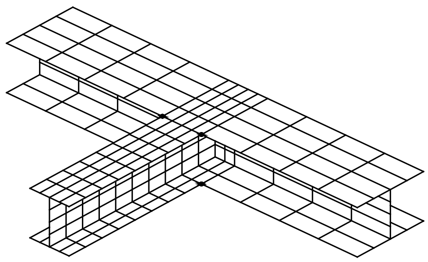
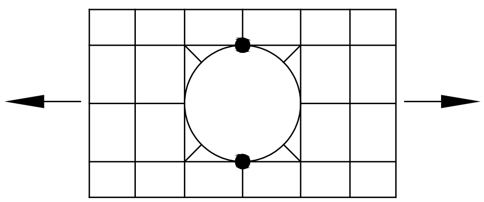
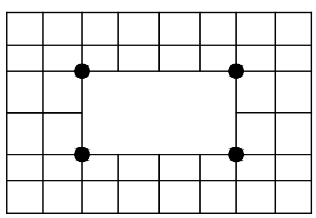
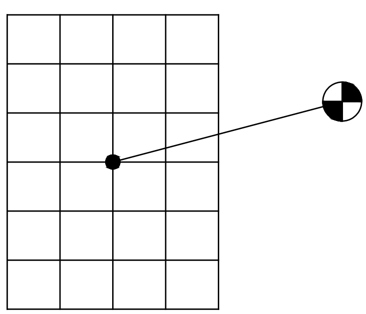
Description of Load Types and Stress Regions	Remarks
14  Flange intersection   Cutouts & penetrations  Points of load introduction	Stress concentrations Concentrated stresses are ordinarily computed for determining fatigue adequacy of a structure. Because adequacy for fatigue is not a requirement of shock induced loads, nodal stresses occurring at points of stress concentration (i.e., corners, cutouts, points of load introduction) are unlimited. General stress requirements for the gross section must still be satisfied. In the case of local load introduction, shear, tear-out, and punch-through requirements must still be satisfied.

Table F-2. Special Applications of the Allowable Stress Criteria.

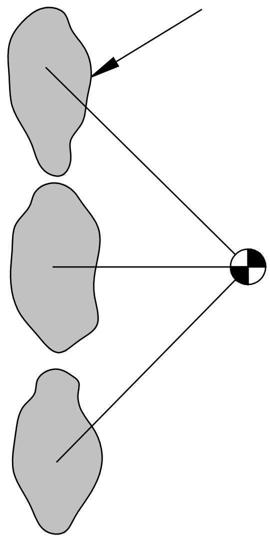
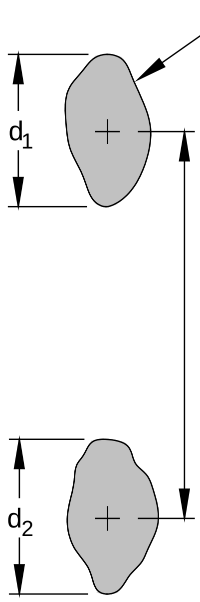
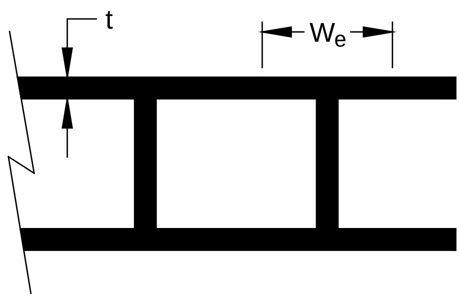
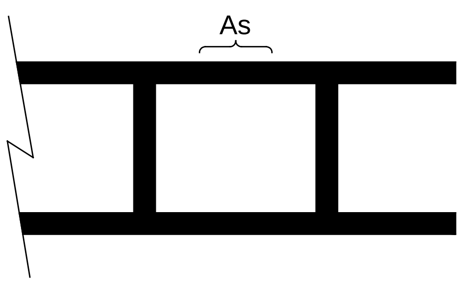
Description of Load Types and Stress Regions	Remarks
1 Boundary on which stress exceeds general allowables 	Adjacent local stress regions Membrane plus bending Discrete regions of local stress resulting from concentrated loads shall not overlap.
2 Boundary on which stress exceeds general allowables 	Adjacent local stress regions The centers of adjacent stressed regions classified as local cannot be closer than 2.5 times the average dimension of each locally stressed area unless the sum of the areas is less than 10 percent of the effective load carrying area. The length of each locally stressed area (d1, d2) shall be taken as the distance over which the stresses exceed the general stress limits along a line of action between the centers of each pair of adjacent locally stressed areas. Not applicable to concentrated loads

Table F-2. Special Applications of the Allowable Stress Criteria – Continued.

Description of Load Types and Stress Regions	Remarks
3  W_e = Effective width of flange σ_y = Allowable yield stress of flange E = Modulus of elasticity t = Plate thickness	Stiffened and sandwiched plate structures Calculate the effective flange width of the continuous plate from T9070-AN-DPC-040/100-4 $W_e = 2t\sqrt{\frac{E}{\sigma_y}}$ Use as an aid in determining effective area for local stress evaluations
4 	Stiffened and sandwiched plate structures The region of stress As located between the effective flange widths is evaluated as a typical plate.

APPENDIX G
BOLT AND BOLT JOINT DESIGN

G.1 INTRODUCTION.

Material for this appendix has been adapted from http://www.boltscience.com/pages/basics1.htm with permission from Bolt Science Limited. The complexity of the simple nut and bolt is frequently underestimated. A properly preloaded bolt does not perform like a loose bolt. A properly preloaded bolted joint can sustain millions of load cycles without problems; a joint consisting of non-preloaded bolts will frequently fail within a few cycles. The reason for this is the way a bolted joint carries an external load. A properly preloaded bolt sustains only a small portion of any externally applied load. No matter how well designed and manufactured the bolt itself is, it cannot alone make the joint more reliable. Bolt selection based upon an understanding of the mechanics of how a threaded fastener sustains loading and the influence tightening procedures can have on a bolted joint is also needed. It is not widely understood how a bolted joint carries a direct load. A properly preloaded bolt can survive in an application that a non-preloaded, or loose bolt, would fail in a matter of seconds. When a load is applied to a joint containing a preloaded bolt, it does not sustain the full effect of the load but usually only a small part of it. A properly preloaded bolt that is installed in accordance with the ship spec and applicable Navy installation processes and requirements is necessary to develop the full potential strength of the bolted joint.

Figure G-1 shows a bolt and nut securing a bracket to a support plate.

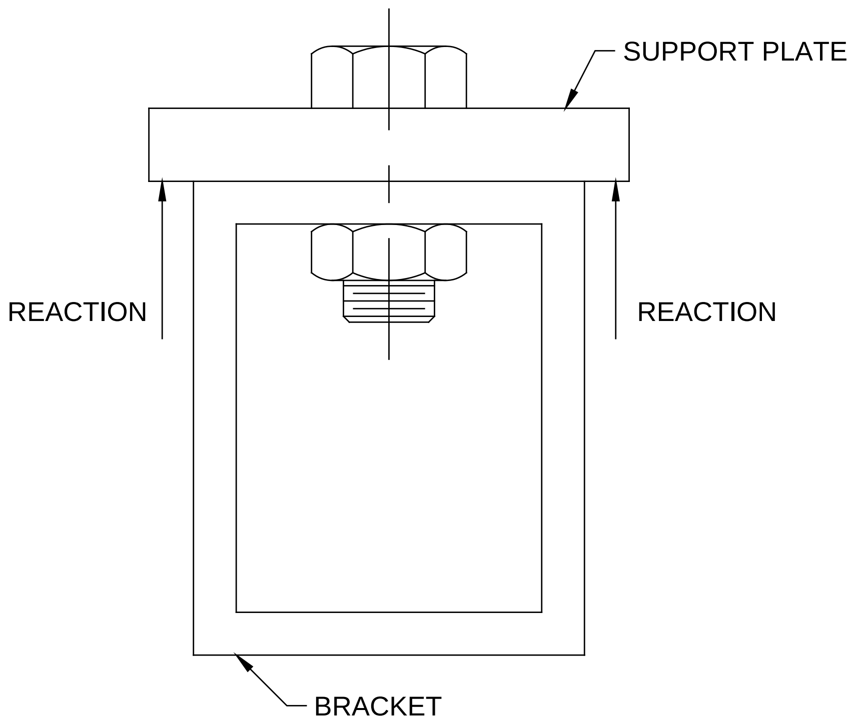


Figure G-1. Bolt Securing a Bracket to a Support Plate.

With no preload on the bolt, if a weight of 1 pound is added to the bracket, as shown in Figure G-2, then the force in the bolt shank will increase by 1 pound. However, if preload is now applied and the weight applied, the force in the bolt shank will not increase by 1 pound, but by only a fraction of this amount. An understanding of why the bolt does not sustain the full effect of the applied load is fundamental to the subject.

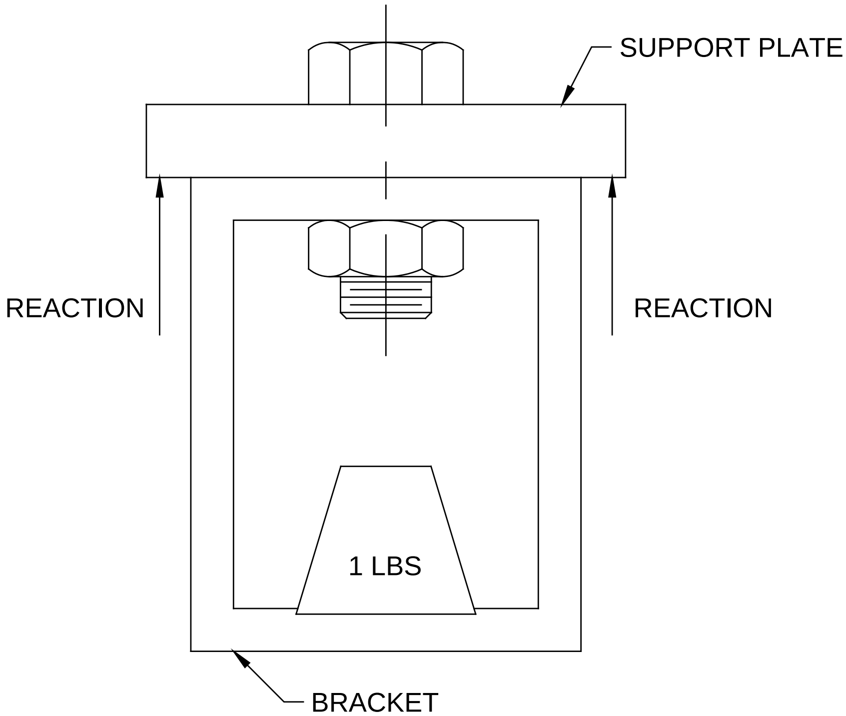


Figure G-2. Bolt Securing a Bracket to a Support Plate with 1-Pound Load.

G.2 APPLYING AN EXTERNAL FORCE TO A BOLTED JOINT.

A model can often be helpful in understanding why the bolt does not sustain the full effect of the applied load. With the special fastener shown in Figure G-3, the bolt is free to move within its casing; a compression spring is included within the casing so that when the bolt is pulled down the spring will compress. A scale on the side of the casing indicates the force present in the spring and hence the force present in the shank of the bolt.

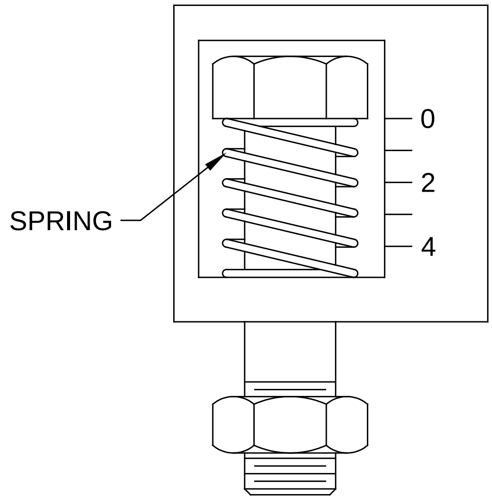


Figure G-3. Model of a Bolt.

The bolt is now inserted through a hole in a support plate and a bracket attached to the special fastener by securing a nut to the threaded shank. If the nut is rotated so that the head of the bolt is pulled down, the spring will be compressed. If the nut is rotated so that two units of force are indicated on the casing, the compressive force acting on the spring will be two and the tensile force in the bolt shank will also be two. This illustrates a preloaded bolt without any working load applied and is shown in Figure G-4.

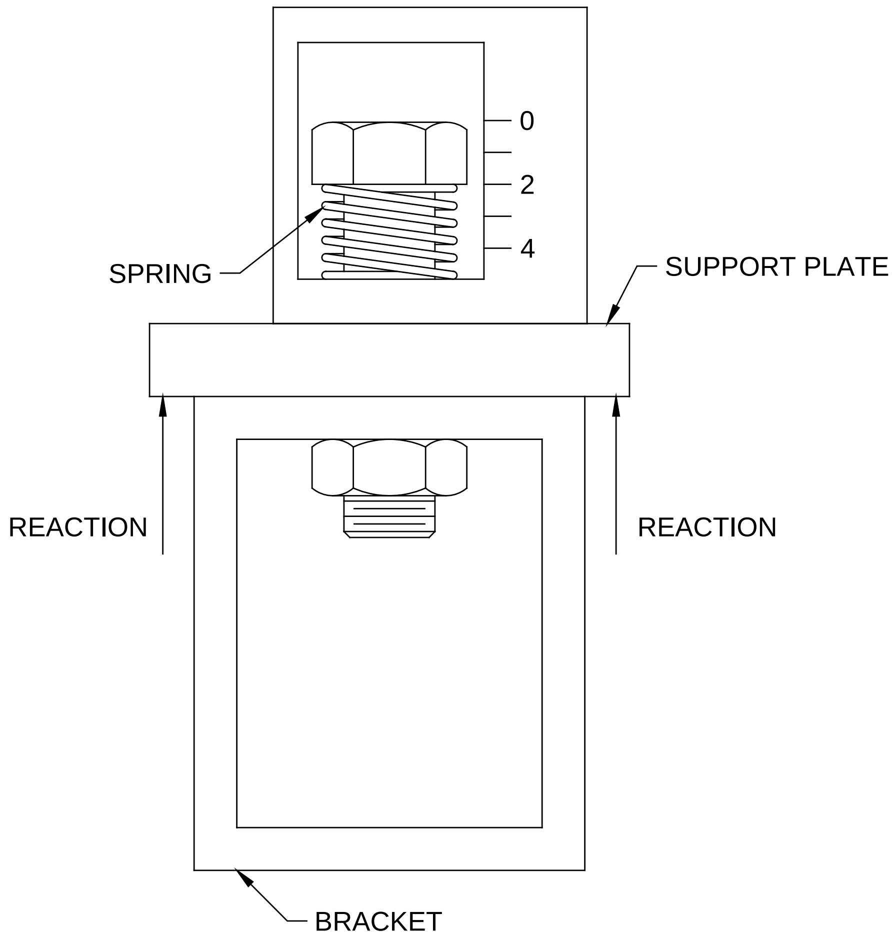


Figure G-4. Model of a Preloaded Bolted Joint.

If a weight of one force unit is now added to the bracket, the initial reaction is to think that the load in the bolt must increase, otherwise what happens to the additional force? Surprisingly, it will maintain its existing value of two units of force. It will not 'feel' any of the additional force. To visualize why this is so, imagine what would happen if the load in the bolt did increase. The bolt head would compress the spring more and a gap would occur between the bracket and the plate. If such a gap was to form then it would mean that there would be two units of force acting upwards, due to the spring, and one unit of force acting downwards from the applied weight. Clearly this force imbalance would not occur. What does happen is that the effect of the applied load decreases the clamping force that exists between the plate and the bracket. With no load applied, the clamping force is two units; with the load applied, the clamping force decreases to one unit of force. The bolt does not actually 'feel' any of the applied force until the force from the applied load exceeds the bolts clamping force.

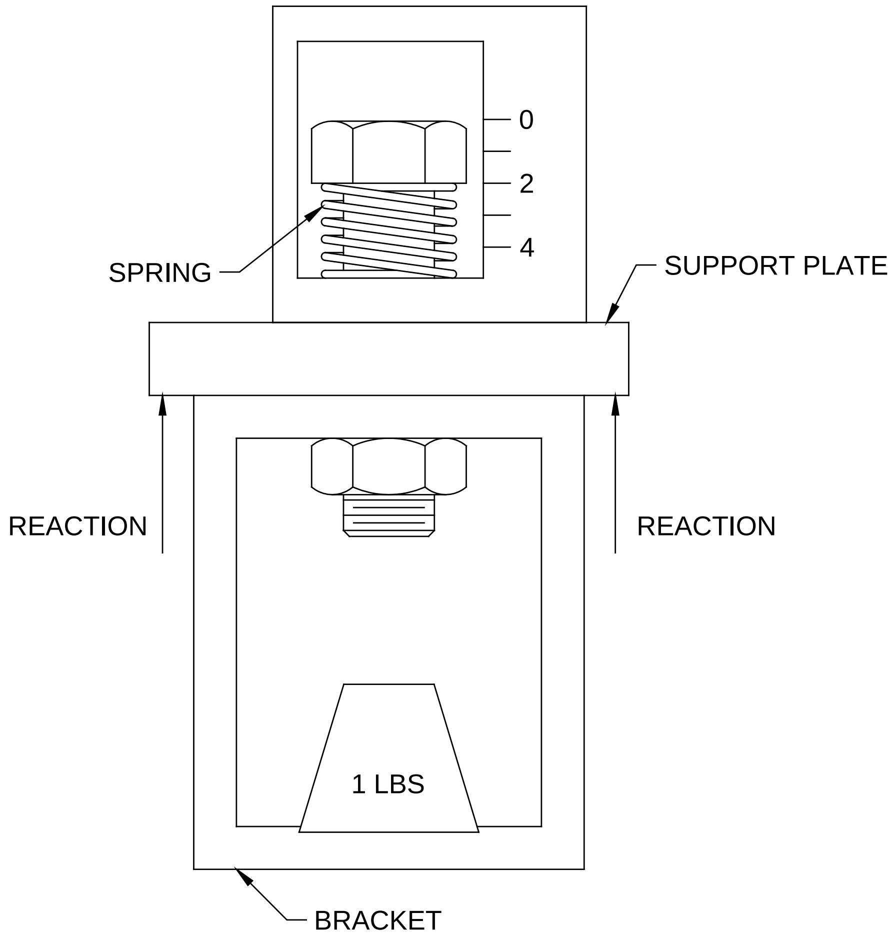


Figure G-5. Preloaded Bolt with Load.

Practical fasteners differ from that shown in Figure G-5 in that elongation of the fastener and compression of the clamped parts occurs upon tightening. This compression results in the bolt sustaining a proportion of the applied load. As the applied force reduces the clamping force existing within the joint an additional strain is felt by the bolt which increases the force it sustains. The amount of the additional force the bolt sustains is smaller than the applied force to the joint. The actual amount of force the bolt sustains depends upon the ratio of stiffnesses of the bolt to the joint material. The best way to understand and visualize how the force sustained by the bolt depends upon the joint stiffness is by the use of joint diagrams.

G.3 JOINT DIAGRAMS.

Joint diagrams have been developed to help visualize the loading within bolted connections. A joint diagram is a means of displaying the load deflection characteristics of the bolt and the material that it clamps. Joint diagrams can be used to assist in visualizing how a bolted joint sustains an external force and explain why the bolt does not sustain the whole of this force.

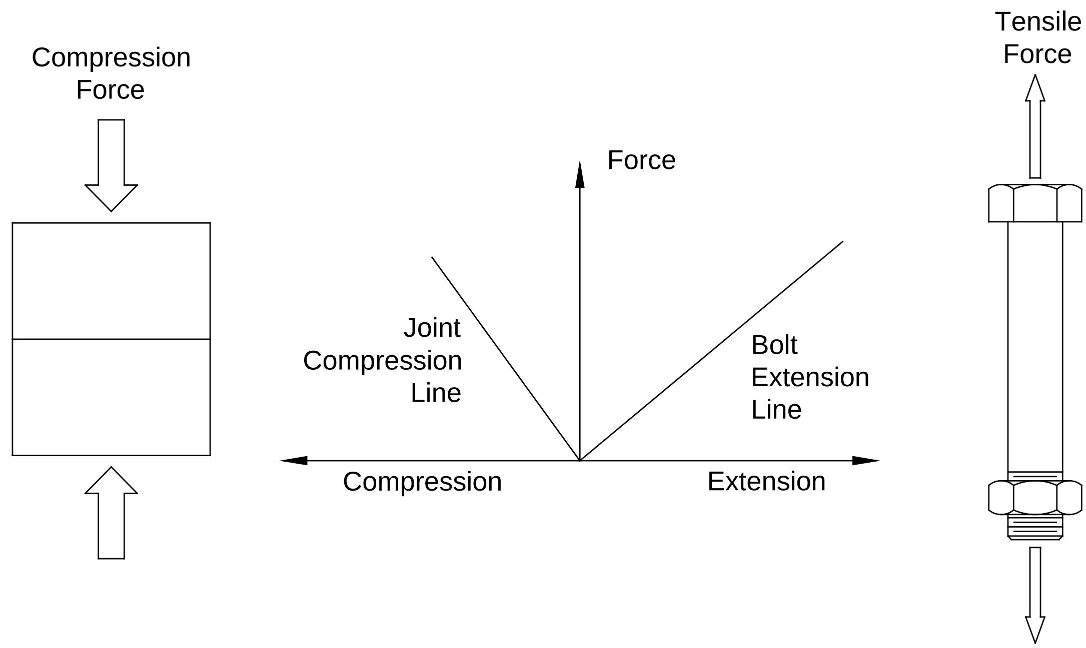


Figure G-6. Bolt and Joint Displacement Diagrams.

The diagram shown in Figure G-6 presents the way the basic joint diagram is constructed. As a nut is rotated on a bolt's screw thread against a joint, the bolt is extended. Internal forces within the bolt resist this extension, a tension force or bolt preload is generated. The reaction to this force is a clamping force that is caused by the joint being compressed. The force-displacement diagram presented in Figure G-6 shows the bolt extension and the joint compression. The slope of the lines represents the relative stiffness of the bolt and joint. The clamped joint normally is stiffer than the bolt. The basic joint diagram, shown in Figure G-7, is developed by moving the compression line of the joint to the right. A triangle is formed because the clamped force tending to compress the joint is equal to the bolt preload. Positive extension is to the right and is sustained by the bolt; negative extension (compression) is to the left and is sustained by the joint material. Before any external load is applied, the bolt preload force is equal to the joint compression force and the bolt extension is equal to the joint compression.

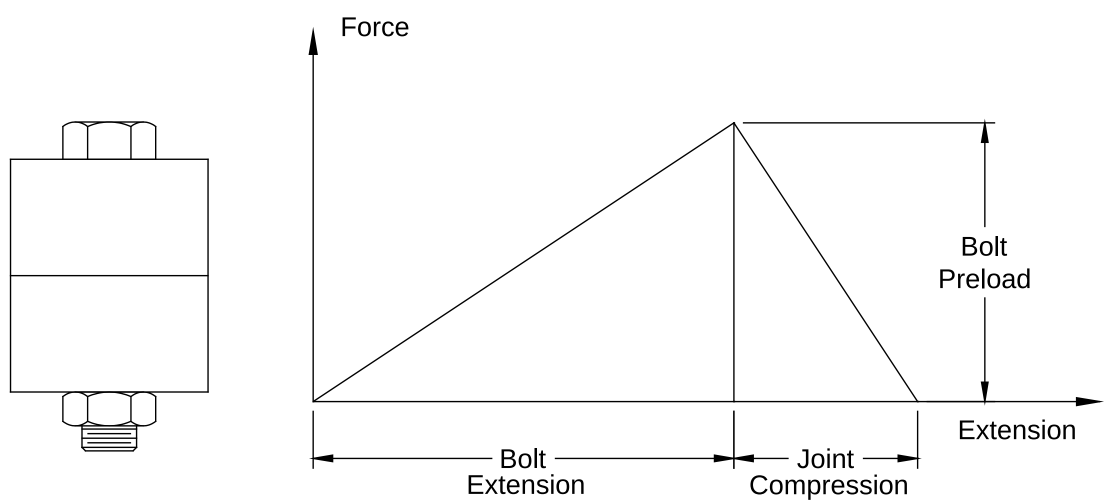


Figure G-7. Joint Diagram.

G.4 JOINT DIAGRAMS WITH EXTERNAL FORCES APPLIED.

When an external tensile force is applied to the joint it has the effect of reducing some of the clamping force caused by the bolt's preload and applying an additional force to the bolt itself. This is illustrated in the joint diagram shown in Figure G-8.

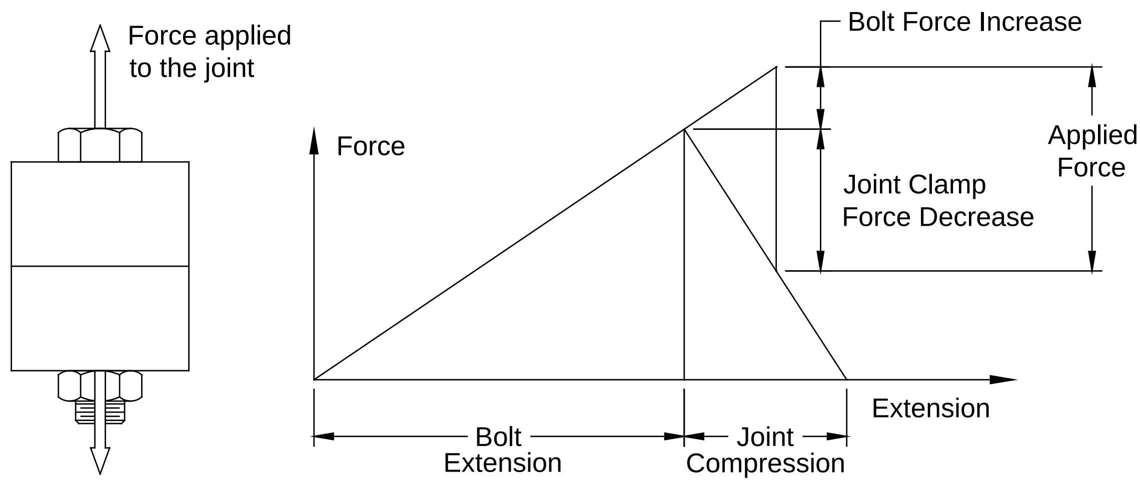


Figure G-8. Joint Diagram with Applied Force.

The external force acts through the joint material and then subsequently into the bolt. At first glance it may seem a bit strange to place the applied force in the position shown in the diagram. However, it should be realized that the load on the bolt cannot be applied without first decreasing the clamping force acting on the joint. As can be observed from the diagram in Figure G-8, the actual increase in the bolt force is dependent upon the relative stiffness of the bolt to the joint. Therefore, the tensile force in the bolt is equal to:

$$F_b = F_p + \left(\frac{K_b}{K_b + K_j}\right) F_a$$

Where:

$F_b = Tensile\ force\ in\ the\ bolt$

$F_p = Preload\ in\ the\ bolt$

$F_a = Applied\ Force\ to\ the\ joint$

$K_b = Stiffness\ of\ the\ bolt$

$K_j = Stiffness\ of\ the\ joint$

As an illustration of the importance of the relative stiffness of the bolt to the joint, Figure G-9 is a joint diagram for a 'hard' joint (a low stiffness bolt with a high stiffness joint). In this case, because of the steep stiffness slope of the joint, the bolt will only sustain a small proportion of the applied force.

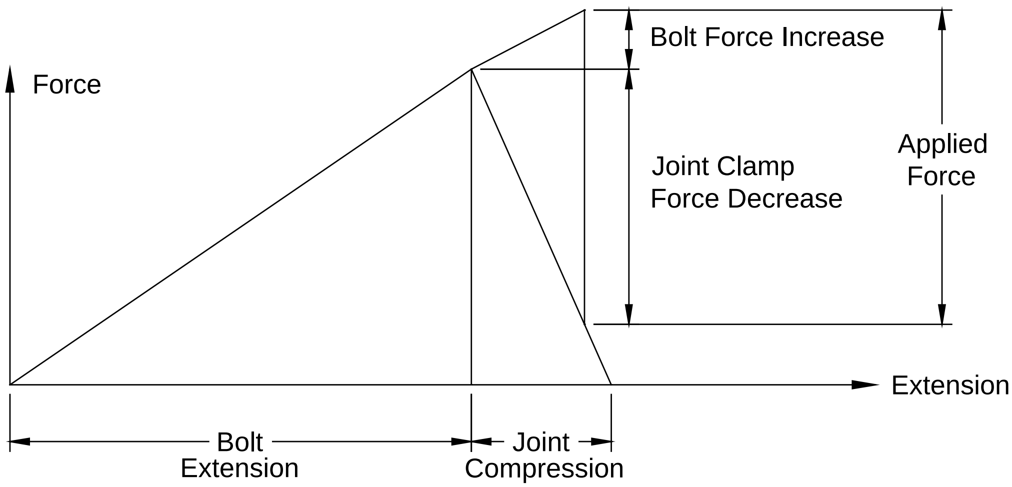


Figure G-9. Joint Diagram for a Hard Joint.

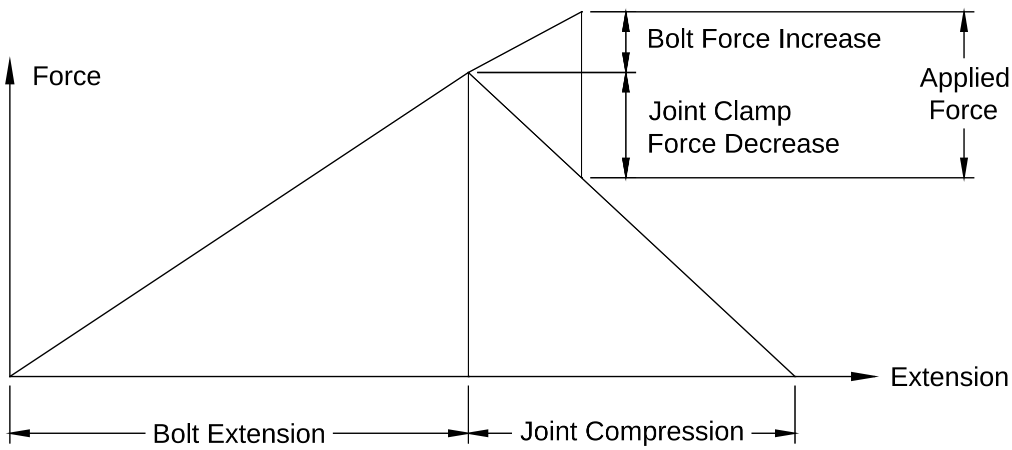


Figure G-10. Joint Diagram for a Soft Joint.

With a 'soft' joint, shown in Figure G-10 (a high stiffness bolt with a low stiffness joint), in this case, because of the shallow stiffness slope of the joint, the bolt will sustain the majority of the applied force. The bolt stiffness in both Figures G-9 and G-10 are the same. Study of these diagrams provides understanding of why high performance bolts have shanks that have been reduced to a diameter below that of the outside diameter of the thread. By reducing the shank diameter in this manner, the stiffness of the fastener is reduced so that it will not sustain as much of any applied force than it would otherwise do.

As the external force increases, the clamping force in the joint will continue to decrease to a point where the flanges of the joint will separate. At this point, all of the load will be carried by the bolt and the joint no longer assists in carrying the external load. In addition, at this point bending forces and torsional forces may be introduced into the joint causing the bolt to fail quickly after flange separation. Therefore, in design it is never recommended to allow a bolted joint to separate under the applied load.

For equipment and foundations designed for shock, a bolt preload based on 67-percent of the bolt material yield stress is required. Using this value in the above equation and the DDAM allowable bolt stress of yield:

$$A_tF_y = 0.67A_tF_y + \left(\frac{K_b}{K_b + K_j}\right)A_tF_y$$

Canceling A_tF_y,

$$1.0 = 0.67 + \left(\frac{K_b}{K_b + K_j}\right)1.0$$

$$0.33 = \left(\frac{K_b}{K_b + K_j}\right)$$

Therefore, if the ratio of the bolt stiffness to the sum of the bolt and joint stiffnesses remains under 0.3 then the bolt may be designed using the entire shock force and neglect the preload. In most cases, joints designed with steel bolts and steel joint components will meet these criteria. If joints include dissimilar materials, exaggerated shim, or washer configurations then these joint effects must be considered in a more detailed analysis of the bolted joint in the DDAM analysis.

Example Bolt Calculation. This section discusses a sample DDAM bolt calculation.

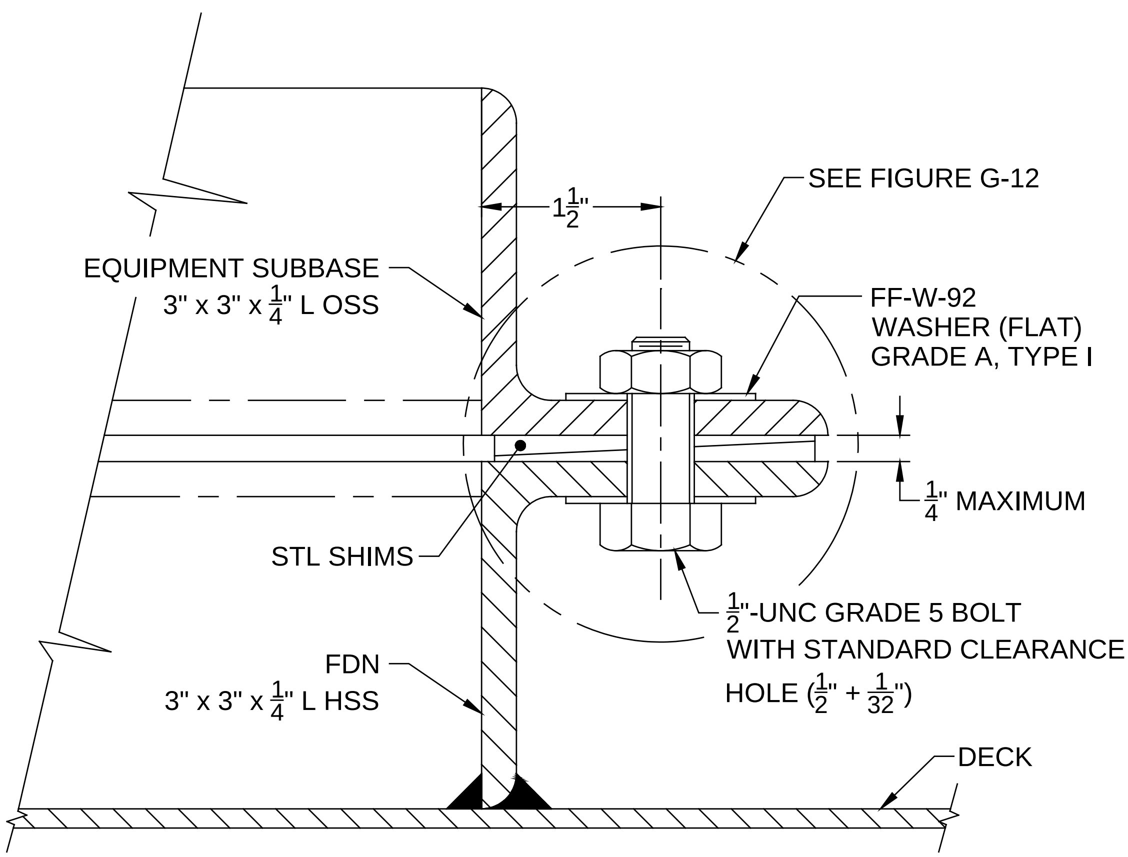


Figure G-11. Typical Bolt Connection.

Using Shigley's fulstrum approximation for the stiffness of the joint, the stiffness of each layer of the joint will be:

$$K_i = \frac{\pi E_i d tan\alpha}{\ln\left\{\frac{(2t_i tan\alpha + D_i - d)(D_i + d)}{(2t_i tan\alpha + D_i + d)(D_i - d)}\right\}}$$

The total stiffness of the joint will be:

$$\frac{1}{K_t} = \sum_{i}^{n} \frac{1}{K_{\ i}}$$

Where:

$K_t = Total\ stiffness$

$K_i = Stiffness\ of\ individual\ joint\ layer$

$\pi = Pi\ (3.1416)$

$E_i = Modulus\ of\ Elasticity\ of\ the\ joint\ layer\ (29{,}600{,}000\ psi)$

$\alpha = Assumed\ stress\ angle, normally\ taken\ as\ 30\ degrees$

$d = Bolt\ hole\ diameter$

$D_i = Outer\ Diameter\ of\ fulstrum\ at\ beginning\ of\ layer$

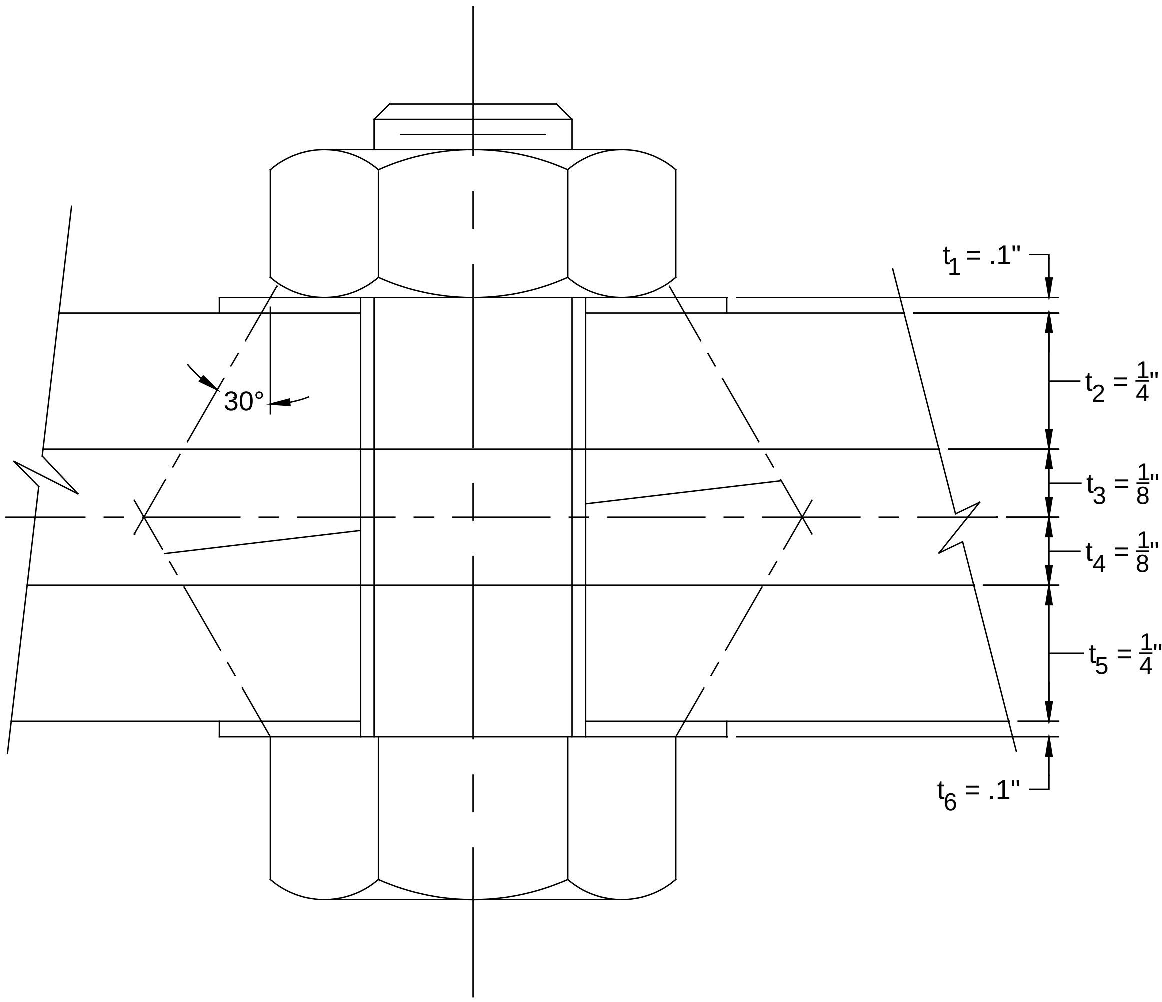


Figure G-12. Joint Configuration.

This example is symmetric, therefore K1 = K6, K2 = K5, and K3 = K4.

Calculation of K1 and K6

Because of the variable nature of the stiffness and sizes of washers, the initial D of the fulstrum is assumed to begin at the head of the bolt, which is normally assumed to be 1.5 times the diameter of the bolt. In this example, the diameter of the head of the bolt would be 1.5 by 0.5 inches or 0.75 inches. Therefore,

$$K_{1,6} = \frac{3.1416(29{,}600{,}000)(0.53125)(0.5774)}{\ln\left\{\frac{(2(0.1)(0.5774) + 0.75 - 0.53125)(0.75 + 0.53125)}{(2(0.1)(0.5774) + 0.75 + 0.53125)(0.75 - 0.53125)}\right\}}$$

$$K_{1,6} = \frac{28{,}524{,}518.5}{\ln(1.402)} = 84{,}417{,}042.0$$

Calculation of K2 and K5

The initial D of the fulstrum is 0.75 + 0.1(tan 30) = 0.80774. Therefore,

$$K_{1,6} = \frac{3.1416(29{,}600{,}000)(0.53125)(0.5774)}{\ln\left\{\frac{(2(0.25)(0.5774) + 0.80774 - 0.53125)(0.80774 + 0.53125)}{(2(0.25)(0.5774) + 0.80774 + 0.53125)(0.80774 - 0.53125)}\right\}}$$

$$K_{1,6} = \frac{28{,}524{,}518.5}{\ln(1.6816)} = 54{,}886{,}508.6$$

Calculation of K3 and K4

The initial D of the fulstrum is 0.80774 + 0.25(tan 30) = 0.95209. Therefore,

$$K_{1,6} = \frac{3.1416(29{,}600{,}000)(0.53125)(0.5774)}{\ln\left\{\frac{(2(0.125)(0.5774) + 0.95209 - 0.53125)(0.95209 + 0.53125)}{(2(0.125)(0.5774) + 0.95209 + 0.53125)(0.95209 - 0.53125)}\right\}}$$

$$K_{1,6} = \frac{28{,}524{,}518.5}{\ln(1.224)} = 141{,}126{,}650.0$$

Therefore K_t of the joint is:

$$\frac{1}{K_t} = \frac{1}{(84{,}417{,}042.0)} + \frac{1}{(84{,}417{,}042.0)} + \frac{1}{(54{,}866{,}508.6)} + \frac{1}{(54{,}866{,}508.6)} + \frac{1}{(141{,}126{,}650.0)} + \frac{1}{(141{,}126{,}650.0)}$$

Total stiffness of the joint is 13,456,111.1 lb/in.

The stiffness of the bolt will be the axial stiffness of the bolt within the grip length.

$$K_b = \frac{A_t E}{L} = \frac{0.196(29{,}600{,}000)}{0.95} = 6{,}106{,}947$$

The ratio of the bolt stiffness to the sum of the bolt and joint stiffness is:

$$\frac{6{,}106{,}947}{6{,}106{,}947 + 13{,}456{,}111.1} = 0.31$$

The bolt to joint stiffness ratio is only approximately 0.3, therefore the bolt may be designed to carry the full shock load at the yield strength of the bolt material.

APPENDIX H
LIST OF ABBREVIATIONS, ACRONYMS, SYMBOLS, AND DEFINITIONS

H.1 INTRODUCTION.

This appendix provides a list of the acronyms, symbols, and definitions used throughout this manual.

H.2 LIST OF ABBREVIATIONS, ACRONYMS, AND SYMBOLS.

ABBREVIATION/ACRONYM/ SYMBOL	TITLE
A_o	Acceleration limited shock design value (equivalent acceleration in g)
A	Area, in^2
a	Mode number
ASM	Algebraic summation method
c	Distance from the neutral axis of a beam cross-section, inches
[C]	Damping matrix
c_{ij}	Damping matrix element
CP	Controllable pitch
D	Shock design value, g
D_{va},D_{aa},D_{fa}	Specified spectrum design values (for mode a) along coordinate axes, g
D_a	Shock design value for mode a
[D]	Diagonal matrix of $[L][D][L]^T$, decomposition of [K]
d	Displacement, inches
DPC	Design Practices and Criteria
DDAM	Dynamic design analysis method
DDS	Design Data Sheet
E	Modulus of elasticity, lb/in^2
f	Shape factor
F	Force
{F(t)}	Force vector as a function of time
f_n	Frequency, Hertz
G	Shear modulus of elasticity
GRP	Glass reinforced plastic
g	Acceleration due to gravity, 386 in/sec^2
g_{ij}	Intermediate variable of [K] decomposition

ABBREVIATION/ACRONYM/ SYMBOL	TITLE
HVAC	Heating ventilation and air conditioning
Hz	Hertz
I	Moment of inertia, in^4
i	Mode counter
J	Polar moment of inertia, in^4
j	Mode counter
K	Spring constant (stiffness), lb/in
$\left[\overline{K}\right]$	Generalized stiffness matrix
[K]	Stiffness matrix
k_{ij}	Stiffness matrix element
ksi	Kips per square inch
L	Length, inches
[L]	Lower triangular matrix of $[L][D][L]^T$, decomposition of [K]
l_{ij}	Element of [L] matrix
$\left[\overline{M}\right]$	Generalized mass matrix
[M]	Mass matrix
m_{ij}	Mass matrix element
M_c	Modal force per given mode
M_c^t	Member force at time t
MDR	Multi-directional response
N	Total number of degrees of freedom in math model
n	Mass number counter
N^*	Master degree of freedom
NOM	Reduced number of modes
NDOF	Number of degrees of freedom
NRL	Naval Research Laboratory
POAM	Plan of action and milestones
P(t)	External force as a function of time
{Q}	Load vector
{q}	Decomposed load vector

ABBREVIATION/ACRONYM/ SYMBOL	TITLE
R_i	Response at point i
{r}	Influence coefficient vector
RPM	Revolutions per minute
T	Torque, in-lb
t	Time, seconds
t_{max}	Maximum time (seconds)
t_{inc}	Incremental time step
u	Displacement
V_o	Velocity limited shock design value (pseudo-velocity), in/sec
V_c	Shear force on plane c
VV&A	Verification, validation, and accreditation
W	Weight, kips
X,Y,Z	Coordinate directions
$\overline{X}$	Displacements, inches
$\overline{X_k}$	Intermediate eigenvector for iteration k
X_k	Normalized eigenvector for iteration k
$\dot{X}$	Notation for velocity
$\ddot{X}$	Notation for acceleration
Z	Section modulus, in^3
Δ	Flexibility matrix
δ_{ij}	Flexibility matrix element
θ	Angle between directions of attack and vertical axis
ω_n	Natural frequency, radians
ξ	Fraction of critical damping
Σ	Summation of
σ	Direct or bending stress
τ	Shear stress
Φ	Angle between direction of attack and a transverse plane through the ship

ABBREVIATION/ACRONYM/ SYMBOL	TITLE
$\left[\Phi\right]$	Mode shape matrix
$\left\{\Phi\right\}_a$	Mode shape for the ath mode
Φ_{ia}	Mode shape for the Ith degree of freedom in mode a
λ	Eigenvalue
μ	Shift parameter for vector iteration
η	Shifted eigenvalues $(\lambda - \mu)$
ρ	Eigenvalue at iteration k
π	3.14159
$[\ ]$	Matrix notation
$\{\ \}$	Vector notation
$\{\ \}^T$	Transpose of a vector
$[\ ]^{-1}$	Inverse of a matrix
*	Notation for reduced set of characteristics

H.3 LIST OF DEFINITIONS.

TERM	DEFINITION
Accreditation	The official certification that a model or simulation and its associated data are acceptable for use for a specific purpose.
Algebraic Summation Method (ASM)	ASM is a time domain method of assessing the shock response of a system from the results of a modal analysis of the system. In the DDAM, ASM is an alternate method of summing responses across the modes. This method combines the responses algebraically, thereby considering the effect of modal phase relationships on the total response and is used to evaluate the effects of closely spaced modes in the DDAM. (See 3.6.7 for detailed discussions.)
Anisotropic	Material characteristic denoting that mechanical or physical properties vary with direction.
Closely Spaced Modes	Response modes of a DDAM calculation whose frequencies are nearly equal (within ±10 percent of a common mean frequency). (Also see “split modes” and “uncombined modes”, and detailed discussion in 3.6.)
Closely Spaced Modes (CSM) Method	The Closely Spaced Modes Method is an analysis procedure which provides a technique for combining responses from two closely spaced modes. Once this combination is determined, it is used in the NRL sum of responses in place of those two modes.
Composite Material	Composites are considered to be combinations of materials differing in composition or form on a macro-scale. The constituents retain their identities in the composite. That is, they do not dissolve or otherwise merge completely into each other although they act in concert. Normally, the components can be physically identified and exhibit an interface between one another. Composite materials are typically anisotropic, examples being wood and fiberglass.
Contracting Activity	The activity having authority and responsibility to initiate purchase orders, agreements, or documents for acquisition of services, materials, or items.
Cut-Off Frequency	As used in DDAM, the cut-off frequency is the upper bound of the frequencies of interest. The cut-off frequency reflects the level of refinement in a mathematical model used to represent a system.
Ductility	The ability of a material to deform plastically before fracturing.
Dynamic Degrees of Freedom	The number of displacement components which must be considered in order to represent the effects of all significant inertia forces on a structure.
Fixed Base	The primary ship structure such as decks, bulkheads, longitudinals, and transverse frame members. For dynamic analysis purposes, a fixed base may be considered to act as a rigid, stationary (relative to the item mounted upon it) boundary in the direction of shock motion through which the shock motion is transmitted to the mounted equipment or structure.
Fixed Base Frequencies	The natural frequencies of a structure or system assuming that the mounting base of the analyzed structure or system is infinitely rigid in the direction of shock motion.
Mathematical Model	A mass-elastic system which is devised to possess a computed shock response simulating that of an actual physical system. All modeled structural elements are assumed to possess linear elastic properties.
Modal Effective Weight (Mass)	A weight (mass) that can be determined from normal mode theory which, when used in a single degree of freedom model with a similarly determined spring, results in a natural frequency which is identical to that of a given mode in a multi-degree of freedom system. The modal effective weight (mass) is also that portion of the item weight (mass) which is effectively accelerated in a given mode.

TERM	DEFINITION
Modal Mass	The force accelerating a given mass in a given mode of system shock response.
Mode Shape	The relative amplitudes of displacement of the system masses in a normal mode of vibration.
Multi-Directional Response (MDR) Analysis	Shock analysis which evaluates system responses (translation and rotation) in the direction of shock input as well as other directions of response.
Navy Standard Details	A collection of NAVSEA-developed documents that specify the structural details and design criteria for a class of Navy ships based on engineering design procedures developed through research and experience from Navy shipbuilding. These structural details and design criteria are found in the Naval Combatant Design Specification, Design Data Sheets, Military Handbooks and Standards, Design Practices and Criteria Manuals, Standard and Type Drawings, General Specifications for Overhaul of Surface Ships, and similar documents.
Node	In a finite element model, a node represents an interface joint between two separate finite elements of the model. A finite element node can include inertial properties (lumped mass) or function only as a structural connection between elements. Also, a node is a point on a structure which does not deflect during vibration in a given mode. An anti-node is a point on a structure where deflection is maximum during vibration in a given mode.
Normal Mode	A natural vibrating configuration of a linear mass-elastic system.
NRL Summation Method	The primary method, within DDAM, of determining the shock response of a system from the results of a modal analysis of the system. This method combines the responses across the modes by adding the absolute value of the largest response to the square-root of the sum of the squares of the other responses. This method takes a statistical approach to modal phasing.
Participation Factor	A value which is computed for each mode of shock response considered and indicates the relative importance of the system mode of shock response. Higher participation factors, regardless of sign, are associated with the more important system modes of shock response.
Quasi-Fixed Base	A modeling technique that eliminates certain mathematical anomalies inherent in the DDAM and permits evaluation of relative displacements between two items mounted to the same fixed base. The Quasi-Fixed Base is a fictitious mass/spring arrangement that is inserted between a mathematical model (or models) and the conventional fixed base (ship structure). The Quasi-Fixed Base mass in the equipment mathematical model(s) is to be no larger than 1 percent of the total model mass. The Quasi-Fixed Base mass is connected to the actual fixed base by a very stiff spring which is selected to assure that the frequency of the lowest dominant mode of the system is not changed by more than 10 percent.
Resilient Mount	An isolation devise that acts to reduce the unwanted effects of shock, noise, or vibration disturbances on a mechanical system. The term “Resilient Mount” is a generic term which includes shock, noise, and vibration mounts.
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 T9070-AJ-DPC-010/(C) 072-1 (formerly DDS 072-1). The minimum shock design value to be used in any mode shall be 6 g.

TERM	DEFINITION
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 that are 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.
Technical Authority	The Technical Authority has the authority, responsibility, and accountability to establish, monitor, and approve technical standards, tools, and processes in conformance to higher authority policy, requirements, architectures, and standards. NOTE: The Technical Authority is not authorized to approve contract deliverable data, except as authorized by the Contracting Activity.
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.
Validation	The process of determining the degree to which a model or simulation and its associated data are an accurate representation of the real world from the perspective of the intended uses of the model.
Verification	The process of determining that a model or simulation implementation and its associated data accurately represent the developer’s conceptual description and specifications.
Wetted Surface Mounted	Wetted surface mounted items are those located such that the item is fluid immersed and mounted below the waterline and (1) external to the shell, or (2) internally within normally filled integral tankage adjacent to shell.

TMDER INSTRUCTIONS

Ships, training activities, supply points, depots, Naval Shipyards, and Supervisors of Shipbuilding are requested to arrange for the maximum practical use and evaluation of NAVSEA and SPAWAR TMs. All errors, omissions, discrepancies, and suggestions for improvement to NAVSEA and SPAWAR TMs shall be submitted as a TMDER. All feedback comments shall be thoroughly investigated and originators will be advised of action resulting there from.

The NAVSEA/SPAWAR TMDER form, NAVSEA 4160/1, is included at the back of the TM.

The following methods are available for generation and submission of TMDERs against unclassified TMs:

For those with a Technical Data Management Information System (TDMIS) account, the most expedient and preferred method of TMDER generation and submission is via the TDMIS website at https://mercury.tdmis.navy.mil/.

For those without a TDMIS account, generate and submit a TMDER via the Naval Systems Data Support Activity (NSDSA) website at https://mercury.tdmis.navy.mil/def_external/pubsearch.cfm. (TDMIS accounts may be requested at the NSDSA website at https://nsdsa.nmci.navy.mil/ by submitting a Customer Service Request [CSR].)

When internet access is not available, submit a TMDER via hardcopy to:

COMMANDER NAVAL SURFACE WARFARE CENTER
NAVAL SYSTEMS DATA SUPPORT ACTIVITY
4363 MISSILE WAY
ATTN: CODE 310 BLDG 1389 TMDERS
PORT HUENEME, CA 93043-4307

Additional copies of the TMDER form may also be downloaded from the NSDSA website at https://nsdsa.nmci.navy.mil/ by clicking on the blue tab labeled “Reference Docs/Forms”.

TMDERs against classified/restricted TMs (includes all NOFORN) must be submitted using the hardcopy method cited above.

Urgent priority TM deficiencies shall be reported by naval message with transmission to Port Hueneme Division, Naval Surface Warfare Center (Code 310), Port Hueneme, CA. Local message handling procedures shall be used. The message shall identify each TM deficiency by Technical Manual Identification Number (TMIN) and title. This method shall be used in those instances where a TM deficiency constitutes an urgent problem (i.e., involves a condition, which if not corrected, could result in injury to personnel, damage to the equipment, or jeopardy to the safety or success of the mission).

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Ref: NAVSEAINST 4160.3 NAVSEA S0005-AA-PRO-010/TMMP

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