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# Simcenter Nastran Basic Dynamic Analysis User's Guide

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# 1. About this Book

This guide describes and gives examples of the basic types of dynamic analysis capabilities available in Simcenter Nastran, including

- normal modes analysis
- transient response analysis
- frequency response analysis
- enforced motion

This guide only presents theoretical derivations of the mathematics used in dynamic analysis as they pertain to the proper understanding of the use of each capability.

For more information on dynamic reduction, response spectrum analysis, random response analysis, complex eigenvalue analysis, nonlinear analysis, control systems, fluid-structure coupling and the Lagrange Multiplier Method, see the *Simcenter Nastran Advanced Dynamic Analysis User's Guide*.



# 2. Fundamentals of Dynamic Analysis

## 2.1 Overview of Fundamentals of Dynamic Analysis

With static structural analysis, it is possible to describe how to use Simcenter Nastran without including a detailed discussion of the fundamental equations. However, because there are several types of dynamic analyses, each with a different mathematical form, you must have some knowledge of both the physics of dynamics and the manner in which the physics is represented to use Simcenter Nastran efficiently for dynamic analysis.

This chapter:

- Contains important information on notation and terminology used throughout the rest of the book.
- Introduces the equations of motion for a single degree-of-freedom dynamic system (see [Equations of Motion](#)).
- Illustrates the dynamic analysis process (see [Dynamic Analysis Process](#) ).
- Characterizes the types of dynamic analyses described in this guide (see [Dynamic Analysis Types](#)).

See [References and Bibliography](#) for a list of references for structural dynamic analysis.

## Dynamic Analysis Versus Static Analysis

Two basic aspects of dynamic analysis differ from static analysis:

- Dynamic loads are applied as a function of time.
- This time-varying load application induces time-varying response (displacements, velocities, accelerations, forces, and stresses). These time-varying characteristics make dynamic analysis more complicated and more realistic than static analysis.

## 2.2 Equations of Motion

The basic types of motion in a dynamic system are displacement  $u$  and the first and second derivatives of displacement with respect to time. These derivatives are velocity and acceleration, respectively, given below:

$$\dot{u} = \frac{du}{dt} = v = \text{velocity}$$

$$\ddot{u} = \frac{d^2u}{dt^2} = \frac{dv}{dt} = a = \text{acceleration}$$

Equation 2-1.

## Velocity and Acceleration

Velocity is the rate of change in the displacement with respect to time. Velocity can also be described as the slope of the displacement curve. Similarly, acceleration is the rate of change of the velocity with respect to time, or the slope of the velocity curve.

## Single Degree-of-Freedom System

The most simple representation of a dynamic system is a single degree-of-freedom (SDOF) system (see [Figure 2-1](#)). In an SDOF system, the time-varying displacement of the structure is defined by one component of motion  $u(t)$ . Velocity  $\dot{u}(t)$  and acceleration  $\ddot{u}(t)$  are derived from the displacement.

$m$	=	mass (inertia)
$b$	=	damping (energy dissipation)
$k$	=	stiffness (restoring force)
$p$	=	applied force
$u$	=	displacement of mass
$\dot{u}$	=	velocity of mass
$\ddot{u}$	=	acceleration of mass

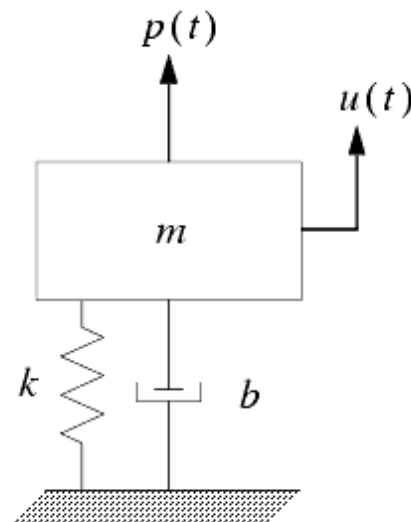


Figure 2-1. Single Degree-of-Freedom (SDOF) System

## Dynamic and Static Degrees-of-Freedom

Mass and damping are associated with the motion of a dynamic system. Degrees-of-freedom with mass or damping are often called dynamic degrees-of-freedom; degrees-of-freedom with stiffness are called



static degrees-of-freedom. It is possible (and often desirable) in models of complex systems to have fewer dynamic degrees-of-freedom than static degrees-of-freedom.

The four basic components of a dynamic system are mass, energy dissipation (damper), resistance (spring), and applied load. As the structure moves in response to an applied load, forces are induced that are a function of both the applied load and the motion in the individual components. The equilibrium equation representing the dynamic motion of the system is known as the equation of motion.

## Equation of Motion

This equation, which defines the equilibrium condition of the system at each point in time, is represented as

$$m\ddot{u}(t) + b\dot{u}(t) + ku(t) = p(t)$$

Equation 2-2.

The equation of motion accounts for the forces acting on the structure at each instant in time. Typically, these forces are separated into internal forces and external forces. Internal forces are found on the left-hand side of the equation, and external forces are specified on the right-hand side. The resulting equation is a second-order linear differential equation representing the motion of the system as a function of displacement and higher-order derivatives of the displacement.

## Inertia Force

An accelerated mass induces a force that is proportional to the mass and the acceleration. This force is called the inertia force  $m\ddot{u}(t)$ .

## Viscous Damping

The energy dissipation mechanism induces a force that is a function of a dissipation constant and the velocity. This force is known as the viscous damping force  $b\dot{u}(t)$ . The damping force transforms the kinetic energy into another form of energy, typically heat, which tends to reduce the vibration.

## Elastic Force

The final induced force in the dynamic system is due to the elastic resistance in the system and is a function of the displacement and stiffness of the system. This force is called the elastic force or occasionally the spring force  $ku(t)$ .

## Applied Load

The applied load  $p(t)$  on the right-hand side of the following equation is defined as a function of time. This load is independent of the structure to which it is applied (e.g., an earthquake is the same

earthquake whether it is applied to a house, office building, or bridge), yet its effect on different structures can be very different.

$$m\ddot{u}(t) + b\dot{u}(t) + ku(t) = p(t)$$

Equation 2-3.

For more information, see [Equation of Motion](#).

## Solution of the Equation of Motion

The solution of the equation of motion for quantities such as displacements, velocities, accelerations, and/or stresses—all as a function of time—is the objective of a dynamic analysis. The primary task for the dynamic analyst is to determine the type of analysis to be performed. The nature of the dynamic analysis in many cases governs the choice of the appropriate mathematical approach. The extent of the information required from a dynamic analysis also dictates the necessary solution approach and steps.

Dynamic analysis can be divided into two basic classifications: free vibrations and forced vibrations. Free vibration analysis is used to determine the basic dynamic characteristics of the system with the right-hand side of [Equation 2-4](#) set to zero (i.e., no applied load). If damping is neglected, the solution is known as undamped free vibration analysis.

$$m\ddot{u}(t) + b\dot{u}(t) + ku(t) = p(t)$$

Equation 2-4.

For more information, see [Equation of Motion](#).

## Free Vibration Analysis

In undamped free vibration analysis, the SDOF equation of motion reduces to

$$m\ddot{u}(t) + ku(t) = 0$$

Equation 2-5.

[Equation 2-5](#) has a solution of the form

$$u(t) = A \sin \omega_n t + B \cos \omega_n t$$

Equation 2-6.

The quantity  $u(t)$  is the solution for the displacement as a function of time  $t$ . As shown in [Equation 2-6](#), the response is cyclic in nature.

## Circular Natural Frequency

One property of the system is termed the circular natural frequency of the structure  $\omega_n$ . The subscript  $n$  indicates the “natural” for the SDOF system. In systems having more than one mass degree of freedom and more than one natural frequency, the subscript may indicate a frequency number. For an SDOF system, the circular natural frequency is given by

$$\omega_n = \sqrt{\frac{k}{m}}$$

Equation 2-7.

The circular natural frequency is specified in units of radians per unit time.

## Natural Frequency

The natural frequency  $f_n$  is defined by

$$f_n = \frac{\omega_n}{2\pi}$$

Equation 2-8.

The natural frequency is often specified in terms of cycles per unit time, commonly cycles per second (cps), which is more commonly known as Hertz (Hz). This characteristic indicates the number of sine or cosine response waves that occur in a given time period (typically one second).

The reciprocal of the natural frequency is termed the period of response  $T_n$  given by

$$T_n = \frac{1}{f_n} = \frac{2\pi}{\omega_n}$$

Equation 2-9.

The period of the response defines the length of time needed to complete one full cycle of response.

In the solution of [Equation 2-10](#),  $A$  and  $B$  are the integration constants.

$$u(t) = A \sin \omega_n t + B \cos \omega_n t$$

Equation 2-10.

For more information, see [Free Vibration Analysis](#).

These constants are determined by considering the initial conditions in the system. Since the initial displacement of the system  $u(t=0)$  and the initial velocity of the system  $\dot{u}(t=0)$  are known,  $A$  and  $B$  are evaluated by substituting their values into the solution of the equation for displacement and its first derivative (velocity), resulting in

$$B = u(t=0) \quad \text{and} \quad A = \frac{\dot{u}(t=0)}{\omega_n}$$

Equation 2-11.

These initial value constants are substituted into the solution, resulting in

$$u(t) = \frac{\dot{u}(0)}{\omega_n} \sin \omega_n t + u(0) \cos \omega_n t$$

Equation 2-12.

**Equation 2-12** is the solution for the free vibration of an undamped SDOF system as a function of its initial displacement and velocity. Graphically, the response of an undamped SDOF system is a sinusoidal wave whose position in time is determined by its initial displacement and velocity as shown in [Figure 2-2](#).

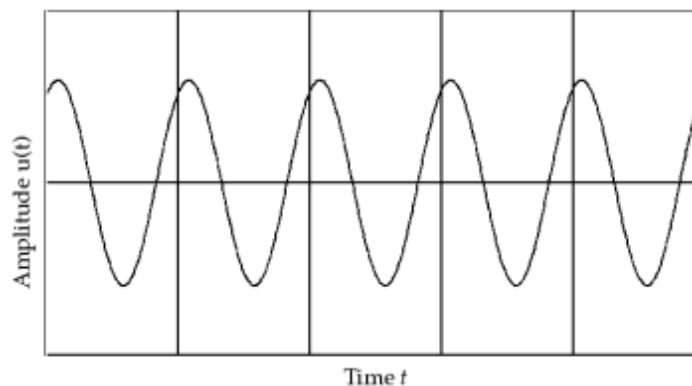


Figure 2-2. SDOF System – Undamped Free Vibrations

If damping is included, the damped free vibration problem is solved. If viscous damping is assumed, the equation of motion becomes

$$m\ddot{u}(t) + b\dot{u}(t) + ku(t) = 0$$

Equation 2-13.

## Damping Types

The solution form in this case is more involved because the amount of damping determines the form of the solution. The three possible cases for positive values of  $b$  are

- Critically damped
- Overdamped
- Underdamped

Critical damping occurs when the value of damping is equal to a term called critical damping  $b_{cr}$ . The critical damping is defined as

$$b_{cr} = 2\sqrt{km} = 2m\omega_n$$

Equation 2-14.

For the critically damped case, the solution becomes

$$u(t) = (A + Bt)e^{-bt/2m}$$

Equation 2-15.

Under this condition, the system returns to rest following an exponential decay curve with no oscillation.

A system is overdamped when  $b > b_{cr}$  and no oscillatory motion occurs as the structure returns to its undisplaced position.

## Underdamped System

The most common damping case is the underdamped case where  $b < b_{cr}$ . In this case, the solution has the form

$$u(t) = e^{-bt/2m}(A \sin \omega_d t + B \cos \omega_d t)$$

Equation 2-16.

Again,  $A$  and  $B$  are the constants of integration based on the initial conditions of the system. The new term  $\omega_d$  represents the damped circular natural frequency of the system. This term is related to the undamped circular natural frequency by the following expression:

$$\omega_d = \omega_n \sqrt{1 - \zeta^2}$$

Equation 2-17.

The term  $\zeta$  is called the damping ratio and is defined by

$$\zeta = \frac{b}{b_{cr}}$$

Equation 2-18.

The damping ratio is commonly used to specify the amount of damping as a percentage of the critical damping.

In the underdamped case, the amplitude of the vibration reduces from one cycle to the next following an exponentially decaying envelope. This behavior is shown in **Figure 2-3**. The amplitude change from one cycle to the next is a direct function of the damping. Vibration is more quickly dissipated in systems with more damping.

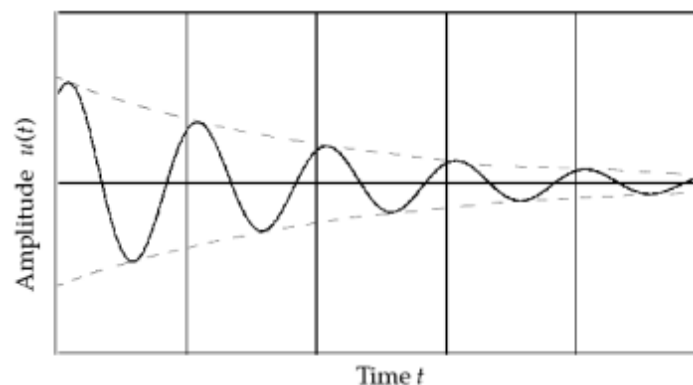


Figure 2-3. Damped Oscillation, Free Vibration

The damping discussion may indicate that all structures with damping require damped free vibration analysis. In fact, most structures have critical damping values in the 0 to 10% range, with values of 1 to 5% as the typical range. If you assume 10% critical damping, [Equation 2-19](#) indicates that the damped and undamped natural frequencies are nearly identical. This result is significant because it avoids the computation of damped natural frequencies, which can involve a considerable computational effort for most practical problems. Therefore, solutions for undamped natural frequencies are most commonly used to determine the dynamic characteristics of the system (see [Real Eigenvalue Analysis \(SOL 103\)](#)). However, this does not imply that damping is neglected in dynamic response analysis. Damping can be included in other phases of the analysis as presented later for frequency and transient response (see [Frequency Response Analysis \(SOLs 108 and 111\)](#) and [Transient Response Analysis \(SOLs 109 and 112\)](#)).

$$u(t) = A \sin \omega_n t + B \cos \omega_n t$$

Equation 2-19.

For more information, see [Free Vibration Analysis](#).

## Forced Vibration Analysis

Forced vibration analysis considers the effect of an applied load on the response of the system. Forced vibrations analyses can be damped or undamped. Since most structures exhibit damping, damped forced vibration problems are the most common analysis types.

The type of dynamic loading determines the mathematical solution approach. From a numerical viewpoint, the simplest loading is simple harmonic (sinusoidal) loading. In the undamped form, the equation of motion becomes

$$m\ddot{u}(t) + ku(t) = p \sin \omega t$$

Equation 2-20.

In this equation the circular frequency of the applied loading is denoted by  $\omega$ . This loading frequency is entirely independent of the structural natural frequency  $\omega_n$ , although similar notation is used.

This equation of motion is solved to obtain

$$u(t) = \underbrace{A \sin \omega_n t + B \cos \omega_n t}_{\text{Initial Condition Solution}} + \underbrace{\frac{p/k}{1 - \omega^2 / \omega_n^2} \sin \omega t}_{\text{Steady-State Solution}}$$

Equation 2-21.

where:

$$A = \frac{\dot{u}(t=0)}{\omega_n} - \frac{\omega p/k}{(1 - \omega^2 / \omega_n^2) \omega_n}$$

$$B = u(t=0)$$

Again,  $A$  and  $B$  are the constants of integration based on the initial conditions. The third term in **Equation 2-21** is the steady-state solution. This portion of the solution is a function of the applied loading and the ratio of the frequency of the applied loading to the natural frequency of the structure.

The numerator and denominator of the third term demonstrate the importance of the relationship of the structural characteristics to the response. The numerator  $p/k$  is the static displacement of the system. In other words, if the amplitude of the sinusoidal loading is applied as a static load, the resulting static displacement  $u$  is  $p/k$ . In addition, to obtain the steady state solution, the static displacement is scaled by the denominator.

The denominator of the steady-state solution contains the ratio between the applied loading frequency and the natural frequency of the structure.

## Dynamic Amplification Factor for No Damping

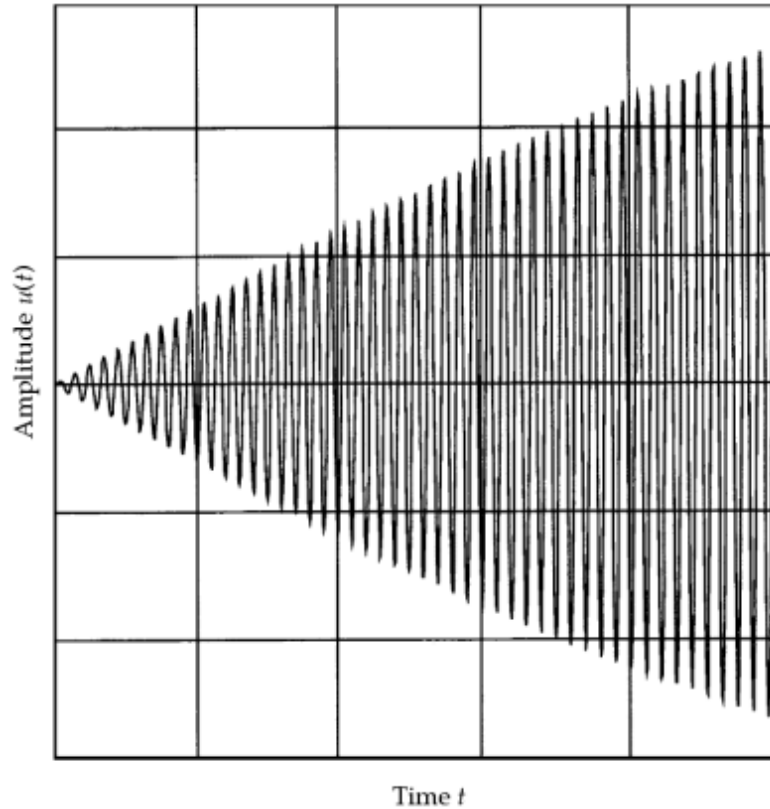
The term

$$\frac{1}{1 - \omega^2 / \omega_n^2}$$

is called the dynamic amplification (load) factor. This term scales the static response to create an amplitude for the steady state component of response. The response occurs at the same frequency as the loading and in phase with the load (i.e., the peak displacement occurs at the time of peak loading). As the applied loading frequency becomes approximately equal to the structural natural frequency, the



ratio  $\omega/\omega_n$  approaches unity and the denominator goes to zero. Numerically, this condition results in an infinite (or undefined) dynamic amplification factor. Physically, as this condition is reached, the dynamic response is strongly amplified relative to the static response. This condition is known as resonance. The resonant buildup of response is shown in **Figure 2-4**.



**Figure 2-4. Harmonic Forced Response with No Damping**

It is important to remember that resonant response is a function of the natural frequency and the loading frequency. Resonant response can damage and even destroy structures. The dynamic analyst is typically assigned the responsibility to ensure that a resonance condition is controlled or does not occur.

Solving the same basic harmonically loaded system with damping makes the numerical solution more complicated but limits resonant behavior. With damping, the equation of motion becomes

$$m\ddot{u}(t) + b\dot{u}(t) + ku(t) = p \sin \omega t$$

**Equation 2-22.**

In this case, the effect of the initial conditions decays rapidly and may be ignored in the solution. The solution for the steady-state response is

$$u(t) = p/k \frac{\sin(\omega t + \theta)}{\sqrt{(1 - \omega^2 / \omega_n^2)^2 + (2\zeta\omega / \omega_n)^2}}$$

Equation 2-23.

The numerator of the above solution contains a term that represents the phasing of the displacement response with respect to the applied loading. In the presence of damping, the peak loading and peak response do not occur at the same time. Instead, the loading and response are separated by an interval of time measured in terms of a phase angle  $\theta$  as shown below:

$$\theta = -\tan^{-1} \frac{2\zeta\omega / \omega_n}{(1 - \omega^2 / \omega_n^2)}$$

Equation 2-24.

The phase angle  $\theta$  is called the phase lead, which describes the amount that the response leads the applied force.

Note:

Some texts define  $\theta$  as the phase lag, or the amount that the response lags the applied force. To convert from phase lag to phase lead, change the sign of  $\theta$  in [Equation 2-23](#) and [Equation 2-24](#).

## Dynamic Amplification Factor with Damping

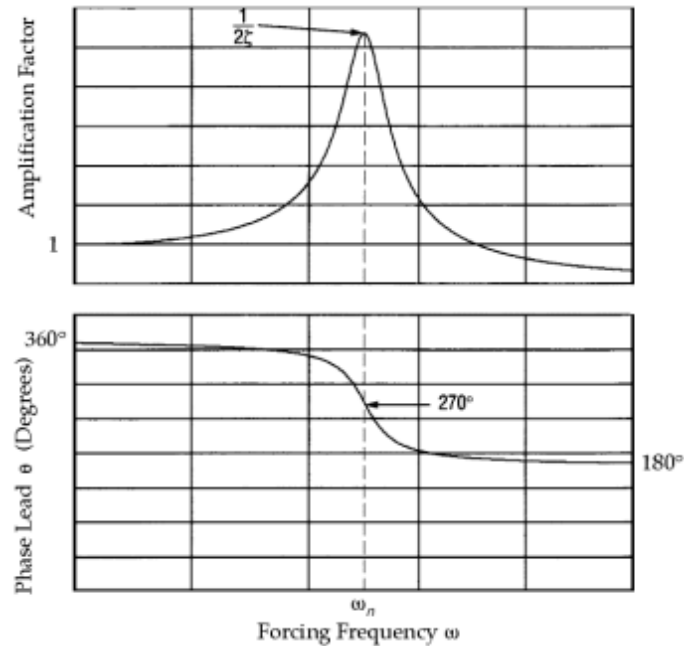
The dynamic amplification factor for the damped case is

$$\frac{1}{\sqrt{(1 - \omega^2 / \omega_n^2)^2 + (2\zeta\omega / \omega_n)^2}}$$

Equation 2-25.

The interrelationship among the natural frequency, the applied load frequency, and the phase angle can be used to identify important dynamic characteristics. If  $\omega/\omega_n$  is much less than 1, the dynamic amplification factor approaches 1 and a static solution is represented with the displacement response in phase with the loading. If  $\omega/\omega_n$  is much greater than 1, the dynamic amplification factor approaches zero, yielding very little displacement response. In this case, the structure does not respond to the loading because the loading is changing too fast for the structure to respond. In addition, any measurable displacement response will be 180 degrees out of phase with the loading (i.e., the

displacement response will have the opposite sign from the force). Finally if  $\omega/\omega_n = 1$ , resonance occurs. In this case, the magnification factor is  $1/(2\zeta)$ , and the phase angle is 270 degrees. The dynamic amplification factor and phase lead are shown in **Figure 2-5** and are plotted as functions of forcing frequency.



**Figure 2-5. Harmonic Forced Response with Damping**

In contrast to harmonic loadings, the more general forms of loading (impulses and general transient loading) require a numerical approach to solving the equations of motion. This technique, known as numerical integration, is applied to dynamic solutions either with or without damping. Numerical integration is described in **"Transient Response Analysis"**.

## 2.3 Dynamic Analysis Process

Before conducting a dynamic analysis, you should first define the goal of the analysis prior to the formulation of the finite element model. Consider the dynamic analysis process shown in **Figure 2-6**. You must evaluate the finite element model in terms of the type of dynamic loading to be applied to the structure. This dynamic load is known as the dynamic environment. The dynamic environment governs the solution approach (i.e., normal modes, transient response, frequency response, etc.). This environment also indicates the dominant behavior that must be included in the analysis (i.e., contact, large displacements, etc.). Proper assessment of the dynamic environment leads to the creation of a more refined finite element model and more meaningful results.

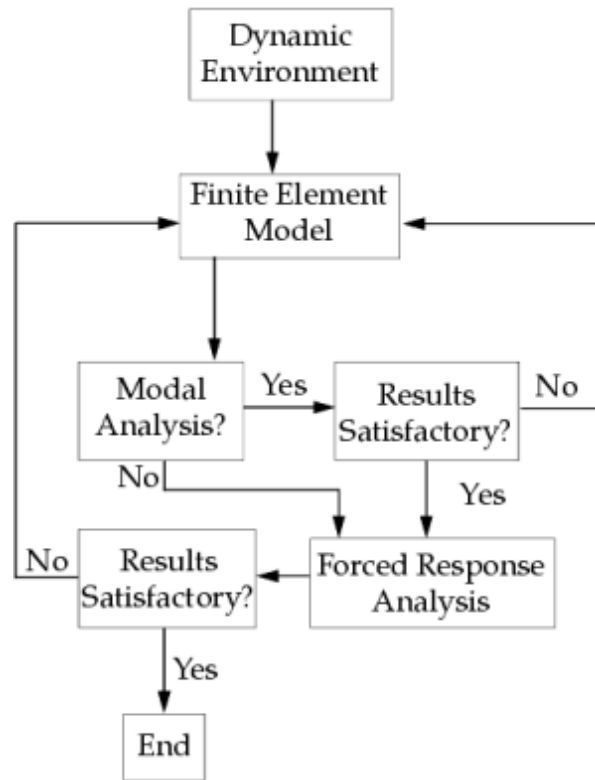


Figure 2-6. Overview of Dynamic Analysis Process

An overall system design is formulated by considering the dynamic environment. As part of the evaluation process, a finite element model is created. This model should take into account the characteristics of the system design and, just as importantly, the nature of the dynamic loading (type and frequency) and any interacting media (fluids, adjacent structures, etc.). At this point, the first step in many dynamic analyses is a modal analysis to determine the structure's natural frequencies and mode shapes (see **"Real Eigenvalue Analysis"**).

In many cases the natural frequencies and mode shapes of a structure provide enough information to make design decisions. For example, in designing the supporting structure for a rotating fan, the design requirements may require that the natural frequency of the supporting structure have a natural frequency either less than 85% or greater than 110% of the operating speed of the fan. Specific knowledge of quantities such as displacements and stresses are not required to evaluate the design.

Forced response is the next step in the dynamic evaluation process. The solution process reflects the nature of the applied dynamic loading. A structure can be subjected to a number of different dynamic loads with each dictating a particular solution approach. The results of a forced response analysis are evaluated in terms of the system design. Necessary modifications are made to the system design. These changes are then applied to the model and analysis parameters to perform another iteration on the design. The process is repeated until an acceptable design is determined, which completes the design process.

The primary steps in performing a dynamic analysis are summarized as follows:

1. Define the dynamic environment (loading).
2. Formulate the proper finite element model.
3. Select and apply the appropriate analysis approach(es) to determine the behavior of the structure.
4. Evaluate the results.

## 2.4 Dynamic Analysis Types

You can perform the following types of basic dynamic analysis with Simcenter Nastran:

- Real eigenvalue analysis (undamped free vibrations).
- Linear frequency response analysis (steady-state response of linear structures to loads that vary as a function of frequency).
- Linear transient response analysis (response of linear structures to loads that vary as a function of time).

Additionally, Simcenter Nastran allows you to perform a number of type of advanced dynamics analysis, such as shock/response spectrum analysis, random response analysis, design sensitivity, design optimization, aeroelasticity, and component mode synthesis.

### Overview of Real Eigenvalue Analysis

Real eigenvalue analysis is used to determine the basic dynamic characteristics of a structure. The results of an eigenvalue analysis indicate the frequencies and shapes at which a structure naturally tends to vibrate. Although the results of an eigenvalue analysis are not based on a specific loading, they can be used to predict the effects of applying various dynamic loads. Real eigenvalue analysis is described in [“Real Eigenvalue Analysis”](#).

### Overview of Frequency Response Analysis

Frequency response analysis is an efficient method for finding the steady-state response to sinusoidal excitation. In frequency response analysis, the loading is a sine wave for which the frequency, amplitude, and phase are specified. Frequency response analysis is limited to linear elastic structures. Frequency response analysis is described in [Frequency Response Analysis](#).

### Overview of Transient Response analysis

Transient response analysis is the most general method of computing the response to time-varying loads. The loading in a transient analysis can be of an arbitrary nature, but is explicitly defined (i.e., known) at every point in time. The time-varying (transient) loading can also include nonlinear effects that are a function of displacement or velocity. Transient response analysis is most commonly applied to

structures with linear elastic behavior. Transient response analysis is described in **“Transient Response Analysis”**.

Additional types of dynamic analysis are available with Simcenter Nastran. These types are described briefly in **“Advanced Dynamic Analysis Capabilities”** and will be described fully in the *Simcenter Nastran Advanced Dynamic Analysis User’s Guide*.

# 3. Finite Element Input Data

## 3.1 Overview of Finite Element Input Data

When you use Simcenter Nastran to perform a dynamic analysis, how you construct the model depends upon the desired results and the type of dynamic loading. For example, you need to obtain stress data, you would only need to define a finer finite element mesh if you needed to obtain system level displacement patterns. Many of the modeling considerations involved in a static analysis are similarly applied in dynamic analysis. However, with a dynamic analysis, you must also include additional input data to define the dynamic character of the structure.

In static analysis the stiffness properties are defined by element and material properties. These same properties are also required for dynamic analysis along with the addition of mass and damping. **“Mass Input”** describes mass input, and **“Damping Input”** describes damping input.

Correct specification of units is very important for dynamic analysis. Incorrect specification of units is probably more difficult to diagnose in dynamic analysis than in static analysis. Because Simcenter Nastran has no built-in set of units, you must ensure their consistency (and accuracy). **“Units in Dynamic Analysis”** describes the common variables and units for dynamic analysis.

**“Direct Matrix Input”** concludes this chapter with a discussion of direct matrix input.

## 3.2 Mass Input

Mass input is one of the major entries in a dynamic analysis. Mass can be represented in a number of ways in Simcenter Nastran. The mass matrix is automatically computed when mass density or nonstructural mass is specified for any of the standard finite elements (CBAR, CQUAD4, etc.) in Simcenter Nastran, when concentrated mass elements are entered, and/or when full or partial mass matrices are entered.

### Lumped and Coupled Mass

Lumped mass matrices contain uncoupled, translational components of mass. Coupled mass matrices contain translational components of mass with coupling between the components. Coupled mass can be more accurate than lumped mass. However, lumped mass is more efficient and is preferred for its computational speed in dynamic analysis. Note that the CBAR, CBEAM, CROD, CONMi and CMASSi mass matrices include or have the potential to include torsional/rotational components.

For most elements, the mass matrix formulation is controlled with PARAM,COUPMASS in the bulk data. When the lumped mass option is selected (default), the distributed mass of an element is divided and located at the connected grid points. When the coupled mass option is requested with PARAM,COUPMASS,1, the distributed mass is replaced by a nondiagonal mass matrix which couples the connected grid points.

Although the CONMi and CMASSi mass inputs are independent of the COUPMASS parameter, both can be defined such that they produce lumped or coupled mass matrices. See the section “Coupled Mass Matrix Terms” in Chapter 2 of the *Advanced Dynamic Analysis User's Guide* for more information.

The table below shows the mass options available for each element type.

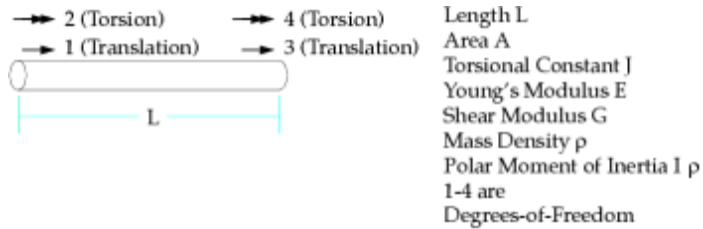
**Table 3-1. Element Mass Types**

Element Type	Lumped Mass	Coupled Mass	Controlled by PARAM,COUPMASS
CBAR	X	X	Yes
CBEAM	X	X	Yes
CBEND		X	No
CONEAX	X		No
CONMi	X	X	No
CONROD	X	X	Yes
CRAC2D	X	X	Yes
CRAC3D	X	X	Yes
CHEXA	X	X	Yes
CMASSi	X	X	No
CPENTA	X	X	Yes
CPYRAM	X	X	Yes
CQUAD4	X	X	Yes
CQUAD8	X	X	Yes
CQUADR	X	X	Yes
CQUADX4	X	X	Yes
CQUADX8	X	X	Yes
CROD	X	X	Yes
CSHEAR	X		No
CTETRA	X	X	Yes
CTRIAX3	X	X	Yes
CTRIAX6	X	X	Yes
CTRIA3	X	X	Yes
CTRIA6	X	X	Yes
CTRIAR	X	X	Yes



Element Type	Lumped Mass	Coupled Mass	Controlled by PARAM,COUPMASS
CTRIAX6	X	X	Yes
CTUBE	X	X	Yes

The Simcenter Nastran coupled mass formulation is a modified approach to the classical consistent mass formulation found in most finite element texts. The Simcenter Nastran lumped mass is identical to the classical lumped mass approach. The various formulations of mass matrices can be compared using the CROD element. Assume the following properties:



## CROD Element Stiffness Matrix

The CROD element's stiffness matrix  $[K]$  is given by:

$$[K] = \begin{bmatrix} \frac{AE}{L} & 0 & -\frac{AE}{L} & 0 \\ 0 & \frac{GJ}{L} & 0 & -\frac{GJ}{L} \\ -\frac{AE}{L} & 0 & \frac{AE}{L} & 0 \\ 0 & -\frac{GJ}{L} & 0 & \frac{GJ}{L} \end{bmatrix} \begin{matrix} 1 \\ 2 \\ 3 \\ 4 \end{matrix}$$

Equation 3-1.

The zero entries in the matrix create independent (uncoupled) translational and rotational behavior for the CROD element, although for most other elements these degrees-of-freedom are coupled.

## CROD Lumped Mass Matrix

The CROD element classical lumped mass matrix is the same as the Simcenter Nastran lumped mass matrix. This lumped mass matrix is

$$[M] = \rho AL \begin{bmatrix} \frac{1}{2} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & \frac{1}{2} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}$$

Equation 3-2.

The lumped mass matrix is formed by distributing one-half of the total rod mass to each of the translational degrees-of-freedom. These degrees-of-freedom are uncoupled and there are no torsional mass terms calculated.

The CROD element classical consistent mass matrix is

$$[M] = \rho AL \begin{bmatrix} \frac{1}{3} & 0 & \frac{1}{6} & 0 \\ 0 & \frac{I\rho}{3A} & 0 & \frac{I\rho}{6A} \\ \frac{1}{6} & 0 & \frac{1}{3} & 0 \\ 0 & \frac{I\rho}{6A} & 0 & \frac{I\rho}{3A} \end{bmatrix}$$

Equation 3-3.

This classical mass matrix is similar in form to the stiffness matrix because it has both translational and rotational masses. Translational masses may be coupled to other translational masses, and rotational masses may be coupled to other rotational masses. However, translational masses may not be coupled to rotational masses.

### CROD Coupled Mass Matrix

The CROD element Simcenter Nastran coupled mass matrix is

$$[M] = \rho AL \begin{bmatrix} \frac{5}{12} & 0 & \frac{1}{12} & 0 \\ 0 & 0 & 0 & 0 \\ \frac{1}{12} & 0 & \frac{5}{12} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}$$

Equation 3-4.

The axial terms in the CROD element coupled mass matrix represent the average of lumped mass and classical consistent mass. This average is found to yield the best results for the CROD element as described below. The mass matrix terms in the directions transverse to the element axes are lumped mass, even when the coupled mass option is selected. Note that the torsional inertia is not included in the CROD element mass matrix.

## Lumped Mass Versus Coupled Mass Example

The difference in the axial mass formulations can be demonstrated by considering a fixed-free rod modeled with a single CROD element (**Figure 3-1**). The exact quarter-wave natural frequency for the first axial mode is

$$1.5708 \sqrt{\frac{E/\rho}{l}}$$

Using the lumped mass formulation for the CROD element, the first frequency is predicted to be

$$1.414 \sqrt{\frac{E/\rho}{l}}$$

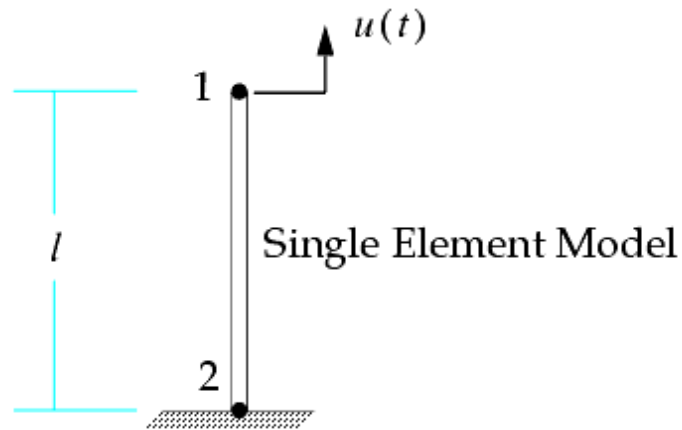
which underestimates the frequency by 10%. Using a classical consistent mass approach, the predicted frequency

$$1.732 \sqrt{\frac{E/\rho}{l}}$$

is overestimated by 10%. Using the coupled mass formulation in Simcenter Nastran, the frequency

$$1.549 \sqrt{\frac{E/\rho}{l}}$$

is underestimated by 1.4%. The purpose of this example is to demonstrate the possible effects of the different mass formulations on the results of a simple problem. Remember that not all dynamics problems have such a dramatic difference. Also, as the model's mesh becomes finer, the difference in mass formulations becomes negligible.



Theoretical Natural Frequency	$\omega_n = \frac{\pi}{2} \sqrt{\frac{E/\rho}{l}} = 1.5708 \sqrt{\frac{E/\rho}{l}}$
Simcenter Nastran Lumped Mass	$\omega_n = \sqrt{2} \sqrt{\frac{E/\rho}{l}} = 1.414 \sqrt{\frac{E/\rho}{l}}$
Classical Consistent Mass	$\omega_n = \sqrt{3} \sqrt{\frac{E/\rho}{l}} = 1.732 \sqrt{\frac{E/\rho}{l}}$
Simcenter Nastran Coupled Mass	$\omega_n = \sqrt{12/5} \sqrt{\frac{E/\rho}{l}} = 1.549 \sqrt{\frac{E/\rho}{l}}$

Figure 3-1. Comparison of Mass Formulations for a ROD

### CBAR, CBEAM Lumped Mass

The CBAR element lumped mass matrix is identical to the CROD element lumped mass matrix. The CBEAM element lumped mass matrix is identical to that of the CROD and CBAR mass matrices with the exception that torsional inertia is included.

## CBAR, CBEAM Coupled Mass

The CBAR element coupled mass matrix is identical to the classical consistent mass formulation except for two terms: (1) the mass in the axial direction is the average of the lumped and classical consistent masses, as explained for the CROD element; and (2) there is no torsional inertia. The CBEAM element coupled mass matrix is also identical to the classical consistent mass formulation except for two terms: (1) the mass in the axial direction is the lumped mass; and (2) the torsional inertia is the lumped inertia.

Another important aspect of defining mass is the units of measure associated with the mass definition. Simcenter Nastran assumes that consistent units are used in all contexts. You must be careful to specify structural dimensions, loads, material properties, and physical properties in a consistent set of units.

All mass entries should be entered in mass consistent units. Weight units may be input instead of mass units, if this is more convenient. However, you must convert the weight to mass by dividing the weight by the acceleration of gravity defined in consistent units:

$$\rho_m = (1/g)\rho_w$$

Equation 3-5.

where:

$\rho_m$	=	mass or mass density
$g$	=	acceleration of gravity
$\rho_w$	=	weight or weight density

The parameter

PARAM,WTMASS,factor

performs this conversion. The value of the factor should be entered as  $1/g$ . The default value for the factor is 1.0. Hence, the default value for WTMASS assumes that mass (and mass density) is entered, instead of weight (and weight density).

When using English units if the weight density of steel is entered as  $RHO = 0.3 \text{ lb/in}^3$ , using PARAM,WTMASS,0.002588 converts the weight density to mass density for the acceleration of gravity  $g = 386.4 \text{ in/sec}^2$ . The mass density, therefore, becomes  $7.76\text{E-}4 \text{ lb}_f\text{-sec}^2/\text{in}^4$ . If the weight density of steel is entered as  $RHO = 80000 \text{ N/m}^3$  when using metric units, then using PARAM,WTMASS,0.102 converts the weight density to mass density for the acceleration of gravity  $g = 9.8 \text{ m/sec}^2$ . The mass density, therefore, becomes  $8160 \text{ kg/m}^3$ .

PARAM,WTMASS is used once per run, and it multiplies all weight/mass input (including CMASSi, CONMi, and nonstructural mass input). Therefore, do not mix input type; use all mass (and mass density) input or all weight (or weight density) input. PARAM,WTMASS does not affect direct input matrices M2GG or M2PP (see **“Direct Matrix Input”**). PARAM,CM2 can be used to scale M2GG; there is no parameter scaling for M2PP. PARAM,CM1 is similar to PARAM,WTMASS since CM1 scales all weight/mass input (except for M2GG and M2PP), but it is active only when M2GG is also used. In other words, PARAM,CM1 is used in addition to PARAM,WTMASS if M2GG is used.

## Simcenter Nastran Mass Input

Mass is input to Simcenter Nastran via a number of different entries. The most common method to enter mass is using the RHO field on the MATi entry. This field is assumed to be defined in terms of mass density (mass/unit volume). To determine the total mass of the element, the mass density is multiplied by the element volume (determined from the geometry and physical properties). For a MAT1 entry, a mass density for steel of  $7.76E-4 \text{ lb}_f\text{-sec}^2/\text{in}^4$  is entered as follows:

1	2	3	4	5	6	7	8	9	10
\$MAT1	MID	E	G	NU	RHO	A	TREF	GE	
MAT1	2	30.0E6		0.3	7.76E-4				

Grid point masses can be entered using the CONM1, CONM2, and CMASSi entries. The CONM1 entry allows input of a fully coupled 6x6 mass matrix. You define half of the terms, and symmetry is assumed. The CONM2 entry defines mass and mass moments of inertia for a rigid body. The CMASSi entries define scalar masses.

## Nonstructural Mass

An additional way to input mass is to use nonstructural mass, which is mass not associated with the geometric cross-sectional properties of an element. Examples of nonstructural mass are insulation, roofing material, and special coating materials. Nonstructural mass is input as mass/length for line elements and mass/area for elements with two-dimensional geometry. Nonstructural mass is defined on the element property entry (PBAR, for example).

## 3.3 Damping Input

Damping is a mathematical approximation used to represent the energy dissipation observed in structures. Damping is difficult to model accurately since it is caused by many mechanisms including

- Viscous effects (dashpot, shock absorber)
- External friction (slippage in structural joints)
- Internal friction (characteristic of the material type)
- Structural nonlinearities (plasticity, gaps)

Because these effects are difficult to quantify, damping values are often computed based on the results of a dynamic test. Simple approximations are often justified because the damping values are low.

## Viscous and Structural Damping

Two types of damping are generally used for linear-elastic materials: viscous and structural. The viscous damping force is proportional to velocity, and the structural damping force is proportional to displacement. Which type to use depends on the physics of the energy dissipation mechanism(s) and is sometimes dictated by regulatory standards.

The viscous damping force  $f_v$  is proportional to velocity and is given by

$$f_v = b\dot{u}$$

Equation 3-6.

where:

$b$  = viscous damping coefficient  
 $\dot{u}$  = velocity

The structural damping force  $f_s$  is proportional to displacement and is given by

$$f_s = i \cdot G \cdot k \cdot u$$

Equation 3-7.

where:

$G$  = structural damping coefficient  
 $k$  = stiffness  
 $u$  = displacement  
 $i$  =  $\sqrt{-1}$  (phase change of 90 degrees)

For a sinusoidal displacement response of constant amplitude, the structural damping force is constant, and the viscous damping force is proportional to the forcing frequency. **Figure 3-2** depicts this and also shows that for constant amplitude sinusoidal motion the two damping forces are equal at a single frequency.

At this frequency,

$$G k = b \omega^* \text{ or } b = \frac{G k}{\omega^*}$$

Equation 3-8.

where  $\omega^*$  is the frequency at which the structural and viscous damping forces are equal for a constant amplitude of sinusoidal motion.

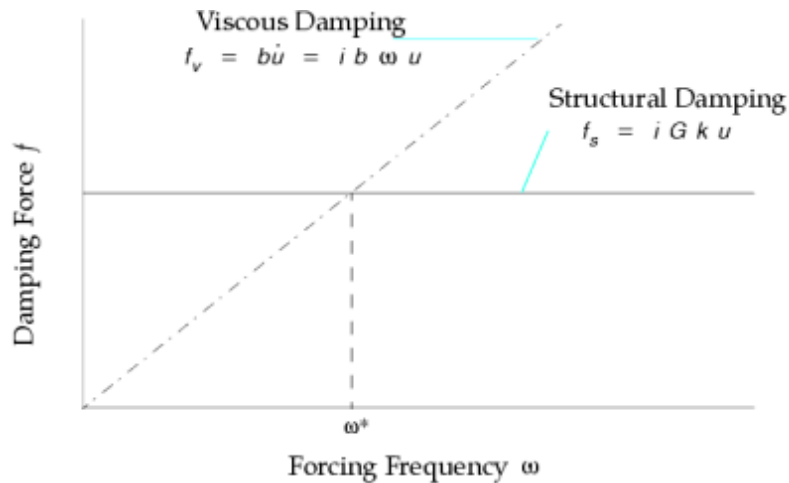


Figure 3-2. Structural Damping and Viscous Damping Forces for Constant Amplitude Sinusoidal Displacement

If the frequency  $\omega^*$  is the circular natural frequency  $\omega_n$ , Equation 3-8 becomes

$$b = \frac{G k}{\omega_n} = G \omega_n m$$

Equation 3-9.

Recall the definition of critical damping from the equation in **Damping Types**:

$$b_{cr} = 2 \sqrt{km} = 2m\omega_n$$

Equation 3-10.

Some equalities that are true at resonance ( $\omega_n$ ) for constant amplitude sinusoidal displacement are



$$\frac{b}{b_{cr}} = \zeta = \frac{G}{2}$$

Equation 3-11.

and

$$Q = \frac{1}{2\zeta} = \frac{1}{G}$$

Equation 3-12.

where  $Q$  is the quality or dynamic magnification factor, which is inversely proportional to the energy dissipated per cycle of vibration.

## The Effect of Damping

Damping is the result of many complicated mechanisms. The effect of damping on computed response depends on the type and loading duration of the dynamic analysis. Damping can often be ignored for short duration loadings, such as those resulting from a crash impulse or a shock blast, because the structure reaches its peak response before significant energy has had time to dissipate. Damping is important for long duration loadings (such as earthquakes), and is critical for loadings (such as rotating machinery) that continually add energy to the structure. The proper specification of the damping coefficients can be obtained from structural tests or from published literature that provides damping values for structures similar to yours.

As is discussed in detail in **“Frequency Response Analysis”** and **“Transient Response Analysis”**, certain solution methods allow specific forms of damping to be defined. The type of damping used in the analysis is controlled by both the solution being performed and the Simcenter Nastran data entries. In transient response analysis, for example, structural damping must be converted to equivalent viscous damping.

## Structural Damping Specification

Structural damping is specified on the MATi and PARAM,G Bulk Data entries. The GE field on the MATi entry is used to specify overall structural damping for the elements that reference this material entry. This definition is via the structural damping coefficient GE.

For example, the MAT1 entry:

1	2	3	4	5	6	7	8	9	10
\$MAT1	MID	E	G	NU	RHO	A	TREF	GE	
MAT1	2	30.0E6		0.3	7.764E-4			0.10	

specifies a structural damping coefficient of 0.1.

An alternate method for defining structural damping is through `PARAM,G,r` where `r` is the structural damping coefficient. This parameter multiplies the stiffness matrix to obtain the structural damping matrix. The default value for `PARAM,G` is 0.0. The default value causes this source of structural damping to be ignored. Two additional parameters are used in transient response analysis to convert structural damping to equivalent viscous damping: `PARAM,W3` and `PARAM,W4`.

`PARAM,G` and `GE` can both be specified in the same analysis.

## Viscous Damping Specification

Viscous damping is defined by the following elements:

CDAMP1 entry	Scalar damper between two degrees-of-freedom (DOFs) with reference to a PDAMP property entry.
CDAMP2 entry	Scalar damper between two DOFs without reference to a property entry.
CDAMP3 entry	Scalar damper between two scalar points (SPOINTS) with reference to a PDAMP property entry.
CDAMP4 entry	Scalar damper between two scalar points (SPOINTS) without reference to a property entry.
CVISC entry	Element damper between two grid points with reference to a PVISC property entry.
CBUSH entry	A generalized spring-and-damper structural element that may be nonlinear or frequency dependent. It references a PBUSH entry.

Viscous damping for modal transient response and modal frequency response is specified with the `TABDMP1` entry.

Note that `GE` and `G` by themselves are dimensionless; they are multipliers of the stiffness. The `CDAMPi` and `CVISC` entries, however, have damping units.

Damping is further described in [Frequency Response Analysis](#) and [“Transient Response Analysis”](#) as it pertains to frequency and transient response analyses.

## 3.4 Units in Dynamic Analysis

Because Simcenter Nastran doesn't assume a particular set of units, you must ensure that the units in your Simcenter Nastran model are consistent. Because there is more input in dynamic analysis than in static analysis, it is easier to make a mistake in units when performing a dynamic analysis. The most frequent source of error in dynamic analysis is incorrect specification of the units, especially for mass and damping.

**Table 3-8** shows typical dynamic analysis variables, fundamental and derived units, and common English and metric units. Note that for English units all “lb” designations are  $lb_f$ . The use of “lb” for mass (i.e.,  $lb_m$ ) is avoided.

**Table 3-8. Engineering Units for Common Variables**

Variable	Dimensions <sup>1</sup>	Common English Units	Common Metric Units
Length	L	in	m
Mass	M	lb-sec <sup>2</sup> /in	kg
Time	T	sec	sec
Area	L <sup>2</sup>	in <sup>2</sup>	m <sup>2</sup>
Volume	L <sup>3</sup>	in <sup>3</sup>	m <sup>3</sup>
Velocity	LT <sup>-1</sup>	in/sec	m/sec
Acceleration	LT <sup>-2</sup>	in/sec <sup>2</sup>	m/sec <sup>2</sup>
Rotation	–	rad	rad
Rotational Velocity	T <sup>-1</sup>	rad/sec	rad/sec
Rotational Acceleration	T <sup>-2</sup>	rad/sec <sup>2</sup>	rad/sec <sup>2</sup>
Circular Frequency	T <sup>-1</sup>	rad/sec	rad/sec
Frequency	T <sup>-1</sup>	cps; Hz	cps; Hz
Eigenvalue	T <sup>-2</sup>	rad <sup>2</sup> /sec <sup>2</sup>	rad <sup>2</sup> /sec <sup>2</sup>
Phase Angle	–	deg	deg
Force	MLT <sup>-2</sup>	lb	N
Weight	MLT <sup>-2</sup>	lb	N
Moment	ML <sup>2</sup> T <sup>-2</sup>	in-lb	N-m
Mass Density	ML <sup>-3</sup>	lb-sec <sup>2</sup> /in <sup>4</sup>	kg/m <sup>3</sup>
Young’s Modulus	ML <sup>-1</sup> T <sup>-2</sup>	lb/in <sup>2</sup>	Pa; N/m <sup>2</sup>
Poisson’s Ratio	–	–	–
Shear Modulus	ML <sup>-1</sup> T <sup>-2</sup>	lb/in <sup>2</sup>	Pa; N/m <sup>2</sup>
Area Moment of Inertia	L <sup>4</sup>	in <sup>4</sup>	m <sup>4</sup>
Torsional Constant	L <sup>4</sup>	in <sup>4</sup>	m <sup>4</sup>
Mass Moment of Inertia	ML <sup>2</sup>	in-lb-sec <sup>2</sup>	kg-m <sup>2</sup>
Stiffness	MT <sup>-2</sup>	lb/in	N/m

Variable	Dimensions <sup>1</sup>	Common English Units	Common Metric Units
Viscous Damping Coefficient	$MT^{-1}$	lb-sec/in	N-sec/m
Stress	$ML^{-1}T^{-2}$	lb/in <sup>2</sup>	Pa; N/m <sup>2</sup>
Strain	–	–	–

L denotes length, M denotes mass , T denotes time, – denotes dimensionless

### 3.5 Contact Conditions in Dynamic Solutions

Simcenter Nastran provides a contact capability for SOL 101 linear static analysis, and also in consecutive dynamic solutions SOL 103, 105, 111 and 112. When contact conditions are included in a linear statics solution, the solver will search and detect when element faces come into contact. The software then creates contact elements, thus preventing the faces from penetrating and allowing finite sliding with optional friction effects.

In a SOL 103 normal mode solution, contact stiffness result is added from the end of the converged linear statics contact solution. The contact stiffness values in the normal mode solution represent the final contact condition of the structure around the contact interface. Thus, it will appear that the resulting contact surfaces are attached during the normal mode analysis. Since the calculated normal modes include the final contact interface conditions, the response calculation (SOLs 111 and 112) which use these normal modes automatically include the same conditions.

The inputs for the normal mode solution are consistent with differential stiffness solutions which require a linear statics subcase. The difference is that the linear statics subcase should include the BCSET case control command. When defining the normal modes subcase, a STATSUB case control command must be included to reference the subcase id containing the contact definition. The contact solution in the linear statics subcase must fully converge before moving to the normal mode portion of the run.

Contact conditions can be used with the element iterative solver. However, differential stiffness conditions cannot be generated with the element iterative solver. Therefore, the default sparse solver will always be used, even when the element iterative solver is requested.

See the chapter on contact in the Simcenter Nastran User's Guide for more information.

### 3.6 Direct Matrix Input

The finite element approach simulates the structural properties with mathematical equations written in matrix format. The structural behavior is then obtained by solving these equations. Usually, all of the structural matrices are generated internally based on the information that you provide in your Simcenter Nastran model. Once you provide the grid point locations, element connectivities, element properties, and material properties, Simcenter Nastran generates the appropriate structural matrices.

## External Matrices

If structural matrices are available externally, you can input the matrices directly into Simcenter Nastran without providing all the modeling information. Normally this is not a recommended procedure since it requires additional work on your part. However, there are occasions where the availability of this feature is very useful and in some cases crucial. Some possible applications are listed below:

- Suppose you are a subcontractor on a classified project. The substructure that you are analyzing is attached to the main structure built by the main contractor. The stiffness and mass effects of this main structure are crucial to the response of your component, but geometry of the main structure is classified. The main contractor, however, can provide you with the stiffness and mass matrices of the classified structure. By reading these stiffness and mass matrices and adding them to your Simcenter Nastran model, you can account for the effect of the attached structure without compromising security.
- Perhaps you are investigating a series of design options on a component attached to an aircraft bulkhead. Your component consists of 500 DOFs and the aircraft model consists of 100,000 DOFs. The flexibility of the backup structure is somewhat important. You can certainly analyze your component by including the full aircraft model (100,500 DOFs). However, as an approximation, you can reduce the matrices for the entire aircraft down to a manageable size using dynamic reduction (see **"Advanced Dynamic Analysis Capabilities"**). These reduced mass and stiffness matrices can then be read and added to your various component models. In this case, you may be analyzing a 2000-DOF system, instead of a 100,500-DOF system.
- The same concept can be extended to a component attached to a test fixture. If the finite element model of the fixture is available, then the reduced mass and stiffness matrices of the fixture can be input. Furthermore, there are times whereby the flexibility of the test fixture at the attachment points can be measured experimentally. The experimental stiffness matrix is the inverse of the measured flexibility matrix. In this instance, this experimental stiffness matrix can be input to your model.

One way of reading these external matrices is through the use of the direct matrix input feature in Simcenter Nastran.

## Direct Matrix Input

The direct matrix input feature can be used to input stiffness, mass, viscous damping, structural damping, and load matrices attached to the grid and/or scalar points in dynamic analysis. These matrices are referenced in terms of their external grid IDs and are input via DMIG Bulk Data entries. As shown in **Table 3-9**, there are seven standard kinds of DMIG matrices available in dynamic analysis.

**Table 3-9. Types of DMIG Matrices in Dynamics**

Matrix	G Type	P Type
Stiffness	K2GG	K2PP
Mass	M2GG	M2PP

Matrix	G Type	P Type
Viscous Damping	B2GG	B2PP
Structural Damping	K42GG	–
Load	P2G	–

The symbols for g-type matrices in mathematical format are  $[K_{gg}^2]$ ,  $[M_{gg}^2]$ ,  $[B_{gg}^2]$ ,  $[K_{gg}^{4,2}]$ , and  $\{P_g^2\}$ . The four matrices K2GG, M2GG, B2GG, and K42GG must be real and symmetric. These matrices are implemented at the g-set level (see [“The Set Notation System Used in Dynamic Analysis”](#) for a description of the set notation for dynamic analysis). In other words, these terms are added to the corresponding structural matrices at the specified DOFs prior to the application of constraints (MPCs, SPCs, etc.).

The symbols for p-type matrices in standard mathematical format are  $[K_{pp}^2]$ ,  $[M_{pp}^2]$ , and  $[B_{pp}^2]$ . The p-set is a union of the g-set and extra points. These matrices need not be real or symmetric. The p-type matrices are used in applications such as control systems. Only the g-type DMIG input matrices are covered in this guide.

### DMIG Bulk Data User Interface

In the Bulk Data Section, the DMIG matrix is defined by a single DMIG header entry followed by a series of DMIG data entries. Each of these DMIG data entries contains a column of nonzero terms for the matrix.

#### Header Entry Format:

1	2	3	4	5	6	7	8	9	10
DMIG	NAME	"0"	IFO	TIN	TOUT	POLAR		NCOL	

#### Column Entry Format:

DMIG	NAME	GJ	CJ		G1	C1	A1	B1	
	G2	C2	A2	B2	-etc.-				

#### Example:

DMIG	STIF	0	6		1				
DMIG	STIF	5	3			5	3	250.	
	5	5	-125.			6	3	-150.	

Field	Contents
NAME	Name of the matrix.
IFO	Form of matrix input: 1 = Square

Field	Contents
	9 or 2 = Rectangular 6 = Symmetric (input only the upper or lower half)
TIN	Type of matrix being input: 1 = Real, single precision (one field is used per element) 2 = Real, double precision (one field per element) 3 = Complex, single precision (two fields are used per element) 4 = Complex, double precision (two fields per element)
TOUT	Type of matrix to be created: 0 = Set by precision system cell (default) 1 = Real, single precision 2 = Real, double precision 3 = Complex, single precision 4 = Complex, double precision
POLAR	Input format of Ai, Bi. (Integer = blank or 0 indicates real, imaginary format; integer > 0 indicates amplitude, phase format.)
NCOL	Number of columns in a rectangular matrix. Used only for IFO = 9.
GJ	Grid, scalar, or extra point identification number for the column index or column number for IFO = 9.
CJ	Component number for GJ for a grid point.
Gi	Grid, scalar, or extra point identification number for the row index.
Ci	Component number for Gi for a grid point.
Ai, Bi	Real and imaginary (or amplitude and phase) parts of a matrix element. If the matrix is real (TIN = 1 or 2), then Bi must be blank.

## DMIG Case Control User Interface

In order to include these matrices, the Case Control must contain the appropriate K2GG, M2GG, B2GG, or K42GG command. (Once again, only the g-type DMIG input matrices are included in this guide.)

### Examples

1. K2GG = mystiff  
The above Case Control command adds terms that are defined by the DMIG Bulk Data entries with the name "mystiff" to the g-set stiffness matrix.
2. M2GG = yourmass  
The above Case Control command adds terms that are defined by the DMIG Bulk Data entries with the name "yourmass" to the g-set mass matrix.
3. B2GG = ourdamp

The above Case Control command adds terms that are defined by the DMIG Bulk Data entries with the name "ourdamp" to the g-set viscous damping matrix.

4. K42GG = strdamp

The above Case Control command adds terms that are defined by the DMIG Bulk Data entries with the name "strdamp" to the g-set structural damping matrix.

Use of the DMIG entry for inputting mass and stiffness is illustrated in one of the examples in "**Real Eigenvalue Analysis**".



# 4. Real Eigenvalue Analysis (SOL 103)

## 4.1 Overview of Real Eigenvalue Analysis (SOL 103)

The usual first step in performing a dynamic analysis is determining the natural frequencies and mode shapes of the structure with damping neglected. These results characterize the basic dynamic behavior of the structure and are an indication of how the structure will respond to dynamic loading.

Note:

This section describes real eigenvalue analysis (SOL 103). For information on complex eigenvalue analysis (SOL 107) or modal complex eigenvalue analysis (SOL 110), see the *Simcenter Nastran Advanced Dynamics User's Guide*.

### Natural Frequencies

The natural frequencies of a structure are the frequencies at which the structure naturally tends to vibrate if it is subjected to a disturbance. For example, the strings of a piano are each tuned to vibrate at a specific frequency. Some alternate terms for the natural frequency are characteristic frequency, fundamental frequency, resonance frequency, and normal frequency.

### Mode Shapes

The deformed shape of the structure at a specific natural frequency of vibration is termed its normal mode of vibration. Some other terms used to describe the normal mode are mode shape, characteristic shape, and fundamental shape. Each mode shape is associated with a specific natural frequency.

Natural frequencies and mode shapes are functions of the structural properties and boundary conditions. A cantilever beam has a set of natural frequencies and associated mode shapes (**Figure 4-1**). If the structural properties change, the natural frequencies change, but the mode shapes may not necessarily change. For example, if the elastic modulus of the cantilever beam is changed, the natural frequencies change but the mode shapes remain the same. If the boundary conditions change, then the natural frequencies and mode shapes both change. For example, if the cantilever beam is changed so that it is pinned at both ends, the natural frequencies and mode shapes change (see **Figure 4-2**).

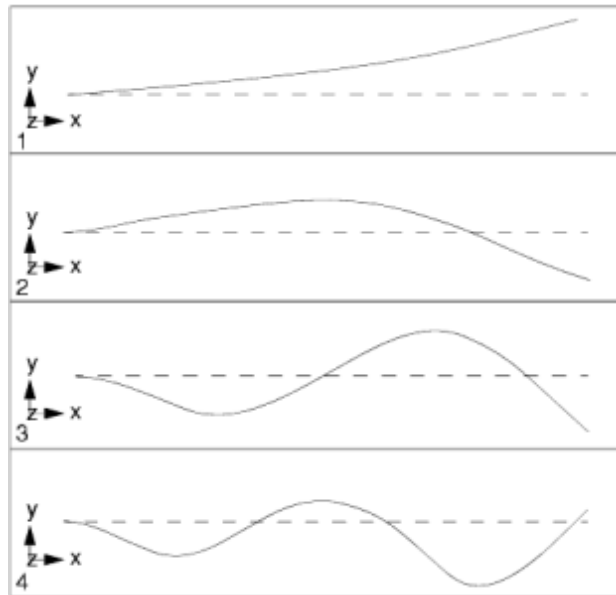


Figure 4-1. The First Four Mode Shapes of a Cantilever Beam

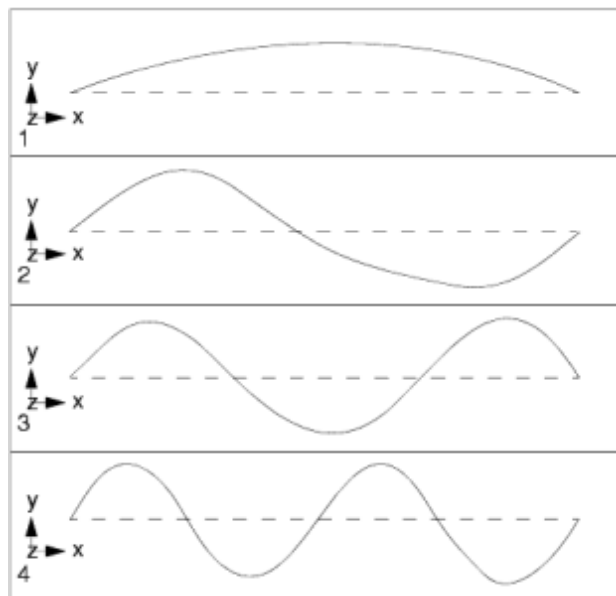


Figure 4-2. The First Four Mode Shapes of a Simply Supported Beam

Computation of the natural frequencies and mode shapes is performed by solving an eigenvalue problem as described in **“Rigid-Body Mode of a Simple Structure”**. Next, we solve for the eigenvalues (natural frequencies) and eigenvectors (mode shapes). Because damping is neglected in the analysis, the eigenvalues are real numbers. (The inclusion of damping makes the eigenvalues complex numbers; see **“Advanced Dynamic Analysis Capabilities”**.) The solution for undamped natural frequencies and mode shapes is called real eigenvalue analysis or normal modes analysis.

The remainder of this chapter describes the various eigensolution methods for computing natural frequencies and mode shapes, and it concludes with several examples.

## 4.2 Reasons to Compute Normal Modes

There are many reasons to compute the natural frequencies and mode shapes of a structure. One reason is to assess the dynamic interaction between a component and its supporting structure. For example, if a rotating machine, such as an air conditioner fan, is to be installed on the roof of a building, it is necessary to determine if the operating frequency of the rotating fan is close to one of the natural frequencies of the building. If the frequencies are close, the operation of the fan may lead to structural damage or failure.

Decisions regarding subsequent dynamic analyses (i.e., transient response, frequency response, response spectrum analysis, etc.) can be based on the results of a natural frequency analysis. The important modes can be evaluated and used to select the appropriate time or frequency step for integrating the equations of motion. Similarly, the results of the eigenvalue analysis—the natural frequencies and mode shapes—can be used in modal frequency and modal transient response analyses (see [Frequency Response Analysis](#) and [“Transient Response Analysis”](#)).

The results of the dynamic analyses are sometimes compared to the physical test results. A normal modes analysis can be used to guide the experiment. In the pretest planning stages, a normal modes analysis can be used to indicate the best location for the accelerometers. After the test, a normal modes analysis can be used as a means to correlate the test results to the analysis results.

Design changes can also be evaluated by using natural frequencies and normal modes. Does a particular design modification cause an increase in dynamic response? Normal modes analysis can often provide an indication.

In summary, there are many reasons to compute the natural frequencies and mode shapes of a structure. All of these reasons are based on the fact that real eigenvalue analysis is the basis for many types of dynamic response analyses. Therefore, an overall understanding of normal modes analysis as well as knowledge of the natural frequencies and mode shapes for your particular structure is important for all types of dynamic analysis.

## 4.3 Overview of Normal Modes Analysis

The solution of the equation of motion for natural frequencies and normal modes requires a special reduced form of the equation of motion. If there is no damping and no applied loading, the equation of motion in matrix form reduces to

$$[M]\{\ddot{u}\} + [K][u] = 0$$

Equation 4-1.

where:

$[M]$  = mass matrix  
 $[K]$  = stiffness matrix

This is the equation of motion for undamped free vibration. To solve **Equation 4-1** assume a harmonic solution of the form

$$\{u\} = \{\phi\} \sin \omega t$$

Equation 4-2.

where:

$\{\phi\}$  = the eigenvector or mode shape  
 $\omega$  = is the circular natural frequency

Aside from this harmonic form being the key to the numerical solution of the problem, this form also has a physical importance. The harmonic form of the solution means that all the degrees-of-freedom of the vibrating structure move in a synchronous manner. The structural configuration does not change its basic shape during motion; only its amplitude changes.

If differentiation of the assumed harmonic solution is performed and substituted into the equation of motion, the following is obtained:

$$-\omega^2 [M] \{\phi\} \sin \omega t + [K] \{\phi\} \sin \omega t = 0$$

Equation 4-3.

which after simplifying becomes

$$([K] - \omega^2 [M]) \{\phi\} = 0$$

Equation 4-4.

This equation is called the eigenequation, which is a set of homogeneous algebraic equations for the components of the eigenvector and forms the basis for the eigenvalue problem. An eigenvalue problem is a specific equation form that has many applications in linear matrix algebra. The basic form of an eigenvalue problem is

$$[A - \lambda I]x = 0$$

Equation 4-5.

where:

- A = square matrix
- $\lambda$  = eigenvalues
- I = identity matrix
- x = eigenvector

In structural analysis, the representations of stiffness and mass in the eigenequation result in the physical representations of natural frequencies and mode shapes. Therefore, the eigenequation is written in terms of  $K$ ,  $\omega$ , and  $M$  as shown in **Equation 4-4** with  $\omega^2 = \lambda$ .

There are two possible solution forms for **Equation 4-4**:

1. If  $\det ([K] - \omega^2 [M]) \neq 0$ , the only possible solution is

$$\{\phi\} = 0$$

**Equation 4-6.**

This is the trivial solution, which does not provide any valuable information from a physical point of view, since it represents the case of no motion. ("det" denotes the determinant of a matrix.)

2. If  $\det ([K] - \omega^2 [M]) = 0$ , then a non-trivial solution ( $\{\phi\} \neq 0$ ) is obtained for

$$([K] - \omega^2 [M])\{\phi\} = 0$$

**Equation 4-7.**

From a structural engineering point of view, the general mathematical eigenvalue problem reduces to one of solving the equation of the form

$$\det ([K] - \omega^2 [M]) = 0$$

**Equation 4-8.**

or

$$\det ([K] - \lambda[M]) = 0$$

Equation 4-9.

where  $\lambda = \omega^2$

The determinant is zero only at a set of discrete eigenvalues  $\lambda_i$  or  $\omega_i^2$ . There is an eigenvector  $\{\phi_i\}$  which satisfies Equation 4-7 and corresponds to each eigenvalue. Therefore, Equation 4-7 can be rewritten as

$$[K - \omega_i^2 M]\{\phi_i\} = 0 \quad i = 1, 2, 3 \dots$$

Equation 4-10.

Each eigenvalue and eigenvector define a free vibration mode of the structure. The  $i$ -th eigenvalue  $\lambda_i$  is related to the  $i$ -th natural frequency as follows:

$$f_i = \frac{\omega_i}{2\pi}$$

Equation 4-11.

where:

$$\begin{aligned} f_i &= \text{i-th natural frequency} \\ \omega_i &= \sqrt{\lambda_i} \end{aligned}$$

The number of eigenvalues and eigenvectors is equal to the number of degrees-of-freedom that have mass or the number of dynamic degrees-of-freedom.

There are a number of characteristics of natural frequencies and mode shapes that make them useful in various dynamic analyses. First, when a linear elastic structure is vibrating in free or forced vibration, its deflected shape at any given time is a linear combination of all of its normal modes

$$\{u\} = \sum_i (\phi_i) \xi_i$$

Equation 4-12.

where:

$\{u\}$  = vector of physical displacements

$\{\phi_i\}$  =  $i$ -th mode shape

$\xi_i$  =  $i$ -th modal displacement

Second, if  $[K]$  and  $[M]$  are symmetric and real (as is the case for all the common structural finite elements), the following mathematical properties hold:

$$\{\phi_i\}^T [M] \{\phi_j\} = 0 \quad \text{if } i \neq j$$

Equation 4-13.

$$\{\phi_j\}^T [M] \{\phi_j\} = m_j = \text{j-th generalized mass}$$

Equation 4-14.

and

$$\{\phi_i\}^T [K] \{\phi_j\} = 0 \quad \text{if } i \neq j$$

Equation 4-15.

$$\{\phi_j\}^T [K] \{\phi_j\} = k_j = \text{j-th generalized stiffness} = \omega_j^2 m_j$$

Equation 4-16.

Also, from [Equation 4-14](#) and [Equation 4-16](#) Rayleigh's equation is obtained

$$\omega_j^2 = \frac{\{\phi_j\}^T [K] \{\phi_j\}}{\{\phi_j\}^T [M] \{\phi_j\}}$$

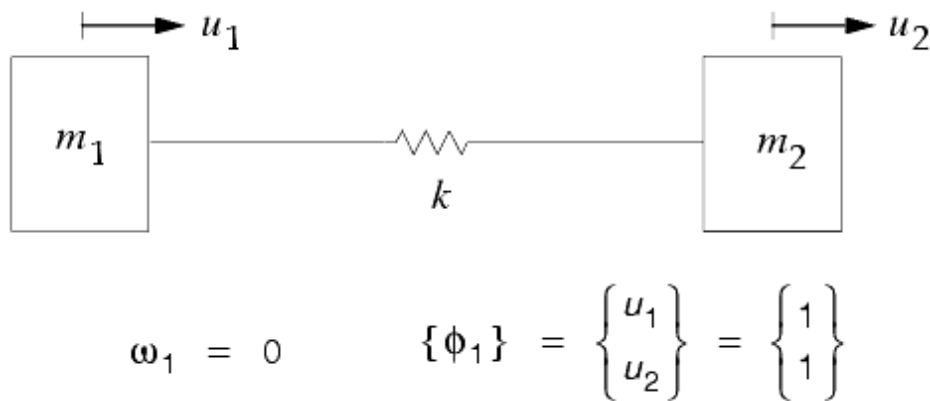
Equation 4-17.

**Equation 4-13** and **Equation 4-15** are known as the orthogonality property of normal modes, which ensures that each normal mode is distinct from all others. Physically, orthogonality of modes means that each mode shape is unique and one mode shape cannot be obtained through a linear combination of any other mode shapes.

In addition, a natural mode of the structure can be represented by using its generalized mass and generalized stiffness. This is very useful in formulating equivalent dynamic models and in component mode synthesis (see **"Advanced Dynamic Analysis Capabilities"** ).

If a structure is not totally constrained in space, it is possible for the structure to displace (move) as a rigid body or as a partial or complete mechanism. For each possible component of rigid-body motion or mechanism, there exists one natural frequency which is equal to zero. The zero-frequency modes are called rigid-body modes. Rigid-body motion of all or part of a structure represents the motion of the structure in a stress-free condition. Stress-free, rigid-body modes are useful in conducting dynamic analyses of unconstrained structures, such as aircraft and satellites. Also, rigid-body modes can be indicative of modeling errors or an inadequate constraint set.

For example, the simple unconstrained structure in **Figure 4-3** has a rigid-body mode.



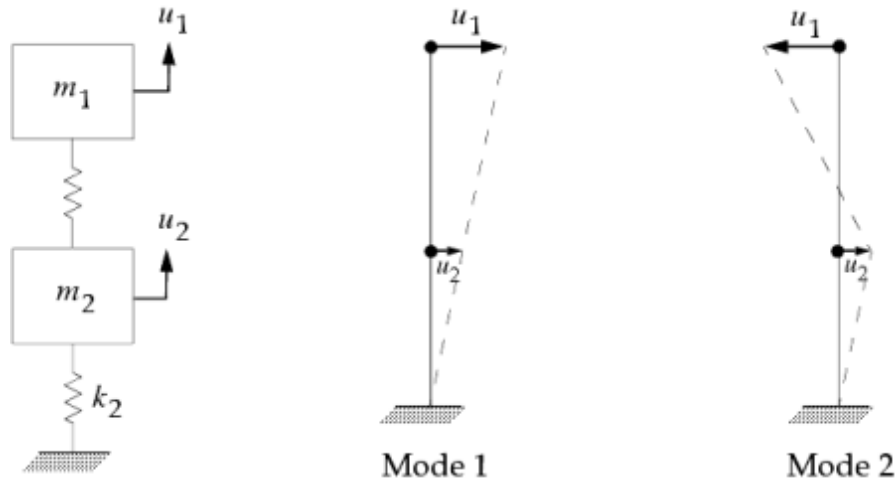
**Figure 4-3. Rigid-Body Mode of a Simple Structure**

When both masses move the same amount (as a rigid body), there is no force induced in the connecting spring. A detailed discussion of rigid-body modes is presented in **"Rigid-body Modes"**.

An important characteristic of normal modes is that the scaling or magnitude of the eigenvectors is arbitrary. Mode shapes are fundamental characteristic shapes of the structure and are therefore relative quantities. In the solution of the equation of motion, the form of the solution is represented as a shape with a time-varying amplitude. Therefore, the basic mode shape of the structure does not change while it is vibrating; only its amplitude changes.

For example, three different ways to represent the two modes of a two-DOF structure are shown in **Figure 4-4**. The graphical representation of the eigenvectors in the figure shows the modal displacements rotated by 90 degrees in order to view the deformation better.





$$\phi_1 = \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} = \begin{Bmatrix} 3 \\ 1 \end{Bmatrix} = \begin{Bmatrix} 300 \\ 100 \end{Bmatrix} = \begin{Bmatrix} -0.6 \\ -0.2 \end{Bmatrix} \{\phi_2\} = \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} = \begin{Bmatrix} -1.6 \\ 0.8 \end{Bmatrix} = \begin{Bmatrix} 160 \\ -80 \end{Bmatrix} = \begin{Bmatrix} 1.0 \\ -0.5 \end{Bmatrix}$$

Figure 4-4. Representations of Mode Shapes for a Two-DOF System

A common misconception about mode shapes is that they define the structural response. Again, mode shapes are relative quantities. They cannot be used alone to evaluate dynamic behavior. As described earlier, it is the relation between the structural loading and the natural frequencies that determines the absolute magnitude of dynamic response. The relation of a specific loading to a set of natural frequencies provides explicit scale factors that are used to determine the extent to which each particular mode is excited by the loading. After the individual modal responses to a particular loading are determined, only then can the various engineering design decisions be made with the actual (absolute) values of stress and/or displacement. Methods that use the modal results to determine forced response are termed modal methods or modal superposition methods. Modal frequency response analysis and modal transient response analysis are described in [Frequency Response Analysis](#) and ["Transient Response Analysis,"](#) respectively.

## Mode Shape Normalization

Although the scaling of normal modes is arbitrary, for practical considerations mode shapes should be scaled (i.e., normalized) by a chosen convention. In Simcenter Nastran there are four normalization choices, MASS, MAX, AF, and POINT normalization.

MASS normalization is the default method of eigenvector normalization. This method scales each eigenvector to result in a unit value of generalized mass.

$$\{\phi_j\}^T [M] \{\phi_j\} = 1.0$$

Equation 4-18.

Numerically this method results in a modal mass matrix that is an identity matrix. This normalization approach is appropriate for modal dynamic response calculations because it simplifies both computational and data storage requirements. When mass normalization is used with a model of a heavy, massive structure, the magnitude of each of the terms of the eigenvectors is very small.

In MAX normalization, each eigenvector is normalized with respect to the largest a-set component. (“Advanced Dynamic Analysis Capabilities” and “The Set Notation System Used in Dynamic Analysis” provide discussions of the a-set.) This normalization results in the largest a-set displacement value being set to a unit (1.0) value. This normalization approach can be very useful in the determination of the relative participation of an individual mode. A small generalized mass obtained using MAX normalization may indicate such things as local modes or isolated mechanisms.

In AF (amplitude-frequency) normalization, the normalized eigenvectors are given by:

$$(C / \omega \Delta_{max}) \{x\}$$

where  $C$  is an optional user-defined scale factor,  $\omega$  is the natural frequency for the mode in Hz,  $\Delta_{max}$  is the magnitude of the maximum grid point translation for the mode, and  $\{x\}$  is the un-normalized eigenvector for the mode.

The optional user-defined scale factor,  $C$ , is specified with the AFNORM parameter. The default for the AFNORM parameter is 1.0.

If the natural frequency is very small, AF normalization will revert to MAX normalization rather than risk overflow. The tolerance to determine a small frequency is 1.0e-4.

If you use the AF or MAX normalization with RDMODES for large models with output requested at relatively few degrees-of-freedom (DOF), the sparse eigenvector recovery greatly improves the RDMODES performance by computing eigenvectors only at the requested DOF. Therefore, the maximum in the AF normalization holds over the computed DOF only; changing the output DOF may change the resulting normalization. If normalization over all DOF is critical, the sparse eigenvector recovery may be deactivated with PARAM,RDSPARSE,NO, but this will significantly impact performance.

POINT normalization of eigenvectors allows you to choose a specific displacement component at which the modal displacement is set to 1 or -1. This method is not recommended because for complex structures the chosen component in the non-normalized eigenvector may have a very small value of displacement (especially in higher modes). This small value can cause larger numbers to be normalized by a small number, resulting in possible numerical roundoff errors in mode shapes.

Although mode shapes are relative quantities, a number of modal quantities can be helpful in predicting qualitative responses or in isolating troublesome modal frequencies. Since relative strains, internal loads, and stresses develop when a structure deforms in a mode shape, you may recover these quantities during a normal modes analysis. Basically, any quantity that you can recover for static analysis is also available for normal modes analysis.

It is important to remember that these output quantities are based on the relative displacements of a mode shape. The output quantities can be compared for a given mode, but not necessarily from one mode to another. However, they can still be effectively used in the analysis/design process.

Modal quantities can be used to identify problem areas by indicating the more highly stressed elements. Elements that are consistently highly stressed across many or all modes will probably be highly stressed when dynamic loads are applied.

Modal strain energy is a useful quantity in identifying candidate elements for design changes to eliminate problem frequencies. Elements with large values of strain energy in a mode indicate the location of large elastic deformation (energy). These elements are those which most directly affect the deformation in a mode. Therefore, changing the properties of these elements with large strain energy should have more effect on the natural frequencies and mode shapes than if elements with low strain energy were changed.

Structures with two or more identical eigenvalues are said to have repeated roots. Repeated roots occur for structures that have a plane of symmetry or that have multiple, identical pieces (such as appendages). The eigenvectors for the repeated roots are not unique because many sets of eigenvectors can be found that are orthogonal to each other. An eigenvector that is a linear combination of the repeated eigenvectors is also a valid eigenvector. Consequently, small changes in the model can make large changes in the eigenvectors for the repeated roots. Different computers can also find different eigenvectors for the repeated roots. Rigid-body modes (see [“Rigid-body Modes”](#)) represent a special case of repeated roots.

## 4.4 Eigenvalue Extraction Methods

Seven methods of real eigenvalue extraction are provided in Simcenter Nastran. These methods are numerical approaches to solving for natural frequencies and modes shapes. The reason for seven different numerical techniques is because no one method is the best for all problems. While most of the methods can be applied to all problems, the choice is often based on the efficiency of the solution process.

The methods of eigenvalue extraction belong to one or both of the following two groups:

- Transformation methods
- Tracking methods

In the transformation method, the eigenvalue equation is first transformed into a special form from which eigenvalues may easily be extracted. In the tracking method, the eigenvalues are extracted one at a time using an iterative procedure.

The recommended real eigenvalue extraction method in Simcenter Nastran is the Lanczos method. The Lanczos method combines the best characteristics of both the tracking and transformation methods. For most models the Lanczos method is the best method to use.

Two of the real eigenvalue extraction methods available in Simcenter Nastran are transformation methods:

- Householder method

- Modified Householder method

The real eigenvalue extraction method available in Simcenter Nastran is classified as tracking methods:

- Sturm modified inverse power method

The remainder of this section briefly describes the various methods. The theory and algorithms behind each method can be found in the *Simcenter Nastran Numerical Methods User's Guide*.

## Lanczos Method

The Lanczos method overcomes the limitations and combines the best features of the other methods. It requires that the mass matrix be positive semidefinite and the stiffness be symmetric. Like the transformation methods, it does not miss roots, but has the efficiency of the tracking methods, because it only makes the calculations necessary to find the roots requested by the user. This method computes accurate eigenvalues and eigenvectors. Unlike the other methods, its performance has been continually enhanced since its introduction giving it an advantage. The Lanczos method is the preferred method for most medium- to large-sized problems, since it has a performance advantage over other methods.

## RDMODES Method

The Recursive Domain Lanczos method (RDMODES) extends the Lanczos capability via substructuring technology for very large scale problems. It requires that the stiffness and mass matrices be symmetric positive semidefinite. It is a dimensional reduction method based on many small, local, and inexpensive eigenvalue problems solved via Lanczos method. In general, it dramatically reduces the computational cost without a significant loss of accuracy as compared to the Lanczos method, especially for modal frequency response analysis. Due to the independent nature of local eigenvalue problems, the RDMODES method can easily take advantage of parallel processing methods. See the *Parallel Processing User's Guide*.

## Householder Methods

The Householder modal extraction method requires a positive definite mass matrix (all degrees-of-freedom must have mass). There is no restriction on the stiffness matrix except that it must be symmetric. These matrices always result in real (positive) eigenvalues. The Householder method is the most efficient method for small problems and problems with dense matrices when a large portion of the eigenvectors are needed. This method finds all of the eigenvalues and as many eigenvectors as requested. While this method does not take advantage of sparse matrices, they are efficient with the dense matrices sometimes created using dynamic reduction (see **"Advanced Dynamic Analysis Capabilities"**).

The Householder method will fail if the mass matrix is not positive definite. To minimize this problem, the auto-omit process occurs in which degrees-of-freedom with null columns are removed by the automatic application of static condensation (see **"Advanced Dynamic Analysis Capabilities"**). Applying the auto-omit process is a precaution and may not remove all possible causes of mass matrix singularity, such as a point mass offset from a grid point, but it greatly improves the reliability and convenience of the Householder method.

The Householder method uses different transformation schemes to obtain the eigenvalues. The Householder method can take advantage of parallel processing methods.

## Modified Householder Methods

The modified Householder method is similar to the standard method with the exception that the mass matrix can be singular. Although the mass matrix is not required to be nonsingular in the modified method, a singular mass matrix can produce one or more infinite eigenvalues. Due to roundoff error, these infinite eigenvalues appear in the output as very large positive or negative eigenvalues. To reduce the incidence of such meaningless results, degrees-of-freedom with null masses are eliminated by automatic static condensation as in the case of the unmodified method.

The modified methods require more computer time than the standard method.

## Automatic Householder Methods

Many times you may not know whether the mass matrix is singular. To assist you in choosing the appropriate method, automatic Householder, is available. Initially the automatic method uses the standard method. In the first step of the method, if the mass matrix is not well-conditioned for decomposition, the method shifts to the corresponding modified method. The modified method is more expensive and may introduce numerical noise due to the shift, but they resolve most of the numerical problems of the ill-conditioned mass matrix. The automatic method, therefore, uses the modified method when necessary for numerical stability but uses the standard method when the numerical stability is accurate.

## Sturm Modified Inverse Power Method

This method uses Sturm sequence logic to ensure that all modes are found. The Sturm sequence check determines the number of eigenvalues below a trial eigenvalue, then finds all of the eigenvalues below this trial eigenvalue until all modes in the designed range are computed. This process helps to ensure that modes are not missed.

The Sturm modified inverse power method is useful for models in which only the lowest few modes are needed. This method is also useful as a backup method to verify the accuracy of other methods.

## 4.5 Comparison of Methods

Since Simcenter Nastran provides a variety of real eigensolution methods, you must decide which is best for your application. The best method for a particular model depends on four factors: the size of the model (the total number of degrees-of-freedom as well as the number of dynamic degrees-of-freedom), the number of eigenvalues desired, the available real memory of your computer, and the conditioning of the mass matrix (whether there are massless degrees-of-freedom). In general, the Lanczos method is the most reliable and efficient, and is the recommended choice.

For small, dense models whose matrices fit into memory, we recommend using the automatic method (automatic Householder ). The automatic Householder runs modified methods if the mass matrix is

singular; however, it runs the unmodified methods, which is faster, if the mass matrix is not singular. The automatic Householder method runs faster on computers with vector processing and also supports parallel processing computers. Note that most real world problems are not small and dense, unless you use reductive methods such as superelements.

The Sturm modified inverse power method can be the best choice when the model is too large to fit into memory, only a few modes are needed, and a reasonable eigenvalue search range is specified. This method is also a backup method for the other methods and is used when a check of the other methods' results is needed.

For medium to large models the Lanczos method is the recommended method. In addition to its reliability and efficiency, the Lanczos method supports sparse matrix methods that substantially increase its speed and reduce disk space requirements.

When reduced accuracy is acceptable, the RDMODES method is the preferred method for large frequency range and large size problems. In many applications, the RDMODES method with sparse eigenvector recovery method can further significantly reduce the overall computation time and storage resource. See the *Parallel Processing User's Guide*.

**Table 4-6. Comparison of Eigenvalue Methods**

	Method			
	Householder	Modified Householder	Lanczos	RDMODES
Reliability	High	High	High	High
Relative Cost:				
Few Modes	Low	Medium	Medium	Medium
Many Modes	High	High	Medium	Low
Limitations	Cannot analyze singular [M]. Expensive for problems that do not fit in memory.	Expensive for many modes. Expensive for problems that do not fit in memory.	Difficulty with massless mechanisms.	Less accurate. Low performance benefit for small problems or when few modes are requested.
Best Application	Small, dense matrices that fit in memory. Use with dynamic reduction ( <a href="#">Chapter 11</a> ).	Small, dense matrices that fit in memory. Use with dynamic reduction ( <a href="#">Chapter 11</a> ).	Medium to large models.	Large frequency range and (very) large models.

## 4.6 User Interface for Real Eigenvalue Analysis

The EIGR and EIGRL Bulk Data entries define the method and select the parameters that control the eigenvalue extraction procedure. The EIGRL entry is used for the Lanczos method, and the EIGR entry is used for all of the other methods.

## User Interface for the Lanczos Method

The format and description of the EIGRL entry is provided below.

### Format:

1	2	3	4	5	6	7	8	9	10
EIGRL	SID	V1	V2	ND	MSGLVL	MAXSET	SHFSCL	NORM	
option_1=value_1, option_2=value_2, etc.									

Field	Contents				
SID	Set identification number. (Unique Integer > 0)				
V1, V2	The V1 field defines the lower frequency bound; the V2 field defines the upper frequency field. For vibration analysis: frequency range of interest. For buckling analysis: eigenvalue range of interest. (Real or blank, $-5 \times 10^{16} \leq V1 < V2 \leq 5 \times 10^{16}$ )				
ND	Number of eigenvalues and eigenvectors desired. (Integer > 0 or blank)				
MSGLVL	Diagnostic level. ( $0 \leq \text{Integer} \leq 4$ ; Default = 0)				
MAXSET	Number of vectors in block or set. ( $1 \leq \text{Integer} \leq 15$ ; Default = 7)				
SHFSCL	Estimate of the first flexible mode natural frequency. (Real or blank)				
NORM	Method for normalizing eigenvectors (Character: "MASS" or "MAX")				
	<table border="0"> <tr> <td>MASS</td> <td>Normalize to unit value of the generalized mass. Not available for buckling analysis. (Default for normal modes analysis.)</td> </tr> <tr> <td>MAX</td> <td>Normalize to unit value of the largest displacement in the analysis set. Displacements not in the analysis set may be larger than unity. (Default for buckling analysis.)</td> </tr> </table>	MASS	Normalize to unit value of the generalized mass. Not available for buckling analysis. (Default for normal modes analysis.)	MAX	Normalize to unit value of the largest displacement in the analysis set. Displacements not in the analysis set may be larger than unity. (Default for buckling analysis.)
MASS	Normalize to unit value of the generalized mass. Not available for buckling analysis. (Default for normal modes analysis.)				
MAX	Normalize to unit value of the largest displacement in the analysis set. Displacements not in the analysis set may be larger than unity. (Default for buckling analysis.)				

### Example:

EIGRL	1	0.1	3.2						
-------	---	-----	-----	--	--	--	--	--	--

Examples of the results of using explicit or default values for the V1, V2, and ND fields are shown in [Table 4-11](#). The defaults on the EIGRL entry are designed to provide the minimum number of roots in cases where the input is ambiguous. Alternatively, you can write the entry above using the new free field format. You must specify certain new parameters such as ALPH,NUMS, and Fi when using the free field format. See the Simcenter Nastran Quick Reference Guide for details.

1	2	3	4	5	6	7	8	9	10
EIGRL	1								
V1 = 0.1, V2 = 3.2									

**Table 4-11. Number and Type of Roots Found with the EIGRL Entry**

Case	V1	V2	ND	Number and Type of Roots Found
1	V1	V2	ND	Lowest ND in range or all in range, whichever is smaller
2	V1	V2		All in range
3	V1		ND	Lowest ND in range $[V1, \infty]$
4	V1			Lowest root in range $[V1, \infty]$
5			ND	Lowest ND roots in $[-\infty, \infty]$
6				Lowest root
7		V2	ND	Lowest ND or all in range $[-\infty, V2]$ , whichever is smaller
8		V2		All below V2

The MSGVLV field of the EIGRL entry is used to control the amount of diagnostic output. The value of 0 produces no diagnostic output. The values 1, 2, or 3 provide more output with the higher values providing increasingly more output. In some cases, higher diagnostic levels may help to resolve difficulties with special modeling problems.

The MAXSET field is used to control the block size. The default value of 7 is recommended for most applications. There may be special cases where a larger value may result in quicker convergence of many multiple roots or a lower value may result in more efficiency when the structure is lightly coupled. However, the default value has been chosen after reviewing the results from a wide range of problems on several different computer types with the goal of minimizing the computer time.

A common occurrence is for the block size to be reset by Simcenter Nastran during the run because there is insufficient memory for a block size of 7. Computational efficiency tends to degrade as the block size decreases. Therefore, you should examine the eigenvalue analysis summary output to determine whether Simcenter Nastran has sufficient memory to use an efficient block size. A smaller block size may be more efficient when only a few roots are requested. The minimum recommended block size is 2.

The SHFSCL field allows a user-designated shift to be used to improve performance especially when large mass techniques are used in enforced motion analysis (see **"Enforced Motion"**). Large mass techniques can cause a large gap between the rigid body (see **"Rigid-body Modes"**) and flexible frequencies, which can degrade performance of the Lanczos method or cause System Fatal Message 5299. When SHFSCL is used, its value should be set close to the expected first nonzero natural frequency.

The Lanczos method normalizes (i.e., scales) the computed eigenvectors using the MASS or MAX method. These methods are specified using the NORM field (new for Version 68). The MASS method normalizes to a unit value of the generalized mass (i.e.,  $m_j = 1.0$ ). The MAX method normalizes to a unit value of the largest component in the a-set (see **"Advanced Dynamic Analysis Capabilities"**). The default is MASS.

You can use the continuation entry to specify V1, V2, ND, MSGVLV, MAXSET, SHFSCL and NORM if you have not specified them on the parent entry. To apply the continuation entry use the following format:



'option\_i=value\_i', e.g., ND=6. Using the continuation entry is the only way to specify the three new options, ALPH, NUMS and Fi.

**NUMS**

The number of segments that a frequency range will be broken into for parallel processing. You must define a value greater than 1 to take advantage of parallel processing. You may also specify NUMS using the NUMSEG keyword on the NASTRAN statement. If you specify both, then NUMS takes precedence.

**Fi**

Directly specifies the upper frequencies of each segment, such that  $V1 < F1 < F2 < \dots < F15 < V2$ .

**ALPH**

Automatically generates the Fi values based on the following formula:

$$F_i = (V2 - V1) \cdot (1.0 - ALPH^i) / (1.0 - ALPH^{NUMS})$$

If you specify both ALPH and Fi, then Fi takes precedence over ALPH as long as they are consistent. If ALPH is multiplied by 100, it may be specified on the FRQSEQ keyword of the NASTRAN statement.

For additional information on the EIGRL entry, see the *Simcenter Nastran Numerical Methods User's Guide*.

## User Interface for the Other Methods

The data entered on the EIGR entry selects the eigenvalue method and the frequency range or number of required roots. The basic format of the Bulk Data entry is as follows:

1	2	3	4	5	6	7	8	9	10
EIGR	SID	METHOD	F1	F2	NE	ND			
	NORM	G	C						

The METHOD field selects the eigenvalue method from the following list:

<b>AHOU</b>	Automatic Householder method
<b>HOU</b>	Householder method
<b>MHOU</b>	Modified Householder method
<b>SINV</b>	Sturm modified inverse power method

The F1 field specifies the lowest frequency of interest in the eigenvalue extraction. The F2 field specifies the highest frequency of interest in the eigenvalue extraction. The units are cycles per unit time. The value for F1 must be greater than or equal to 0.0.

The ND field is used to specify the desired number of roots (for tracking methods) or eigenvectors (for transformation methods), beginning with F1.

The NORM field on the continuation entry is used to specify the method of eigenvector normalization. The choices are

<b>MASS</b>	Mass normalization (default—if used, the continuation entry is not required).
<b>MAX</b>	Normalization to a unit value of the largest component in the analysis set.
<b>POINT</b>	Normalization to a unit value at a user-specified a-set grid point G and component C.

There is an interrelationship among the F1, F2, and ND fields on the EIGR entry as defined in [Table 4-16](#).

**Table 4-16. Relationship Between the METHOD Field and Other Fields**

Field	METHOD Field	
	SINV	HOU, MHOU, or AHOU
F1, F2	Frequency range of interest. F1 must be input. If METHOD = "SINV" and ND is blank, then F2 must be input.  (Real $\geq$ 0.0)	Frequency range of interest. If ND is not blank, F1 and F2 are ignored. If ND is blank, eigenvectors are found whose natural frequencies lie in the range between F1 and F2.  (Real $\geq$ 0.0; F1 < F2)
NE	Estimate of number of roots in range. Not used by "SINV" method.  (Integer > 0)	Not used
ND	Desired number of roots. If this field is blank and METHOD = "SINV", then all roots between F1 and F2 are searched, and the limit is 600 roots.	Desired number of eigenvectors. If ND is zero, the number of eigenvectors is determined from F1 and F2. If all three are blank, then ND is automatically set to one more than the number of degrees-of-freedom listed on SUPORT entries.  (Integer $\geq$ 0; Default = 0)

The rules for METHOD = HOU, MHOU, and AHOU are identical. If any of these methods are selected, Simcenter Nastran finds all of the eigenvalues but only computes the eigenvectors specified by F1 and F2 or those specified by ND (the desired number). F1 and F2 specify the lower and upper bounds of the frequency range in which eigenvectors are computed, and ND specifies the number of eigenvectors, beginning with the lowest (or the first rigid-body mode, if present). If F1, F2, and ND entries are present, ND takes precedence.

If METHOD = SINV, the values of F1, F2, and ND determine the number of eigenvalues and eigenvectors that are computed. These entries also provide hints to help Simcenter Nastran find the eigenvalues. F1 and F2 specify the frequency range of interest within which Simcenter Nastran searches for modes. Simcenter Nastran attempts to find all of the modes in the range between F1 and F2 or the number specified by ND, whichever is less. If searching stops because ND modes are found, there is no

guarantee that they are the lowest eigenvalues. If ND modes are not found in the range of interest, SINV usually finds one mode (or possibly more) outside the range F1 and F2 before stopping the search.

The SINV method is particularly efficient when only a small number of eigenvalues and eigenvectors are to be computed. Often only the lowest mode is of interest. The following example illustrates an EIGR entry which extracts only the lowest nonzero eigenvalue.

1	2	3	4	5	6	7	8	9	10
EIGR	13	SINV	0.0	0.01		1			

It is assumed in the example above that the frequency of the lowest mode is greater than 0.01 cycles per unit time. Simcenter Nastran finds one eigenvalue outside the range F1, F2, and then stops the search. The eigenvalue found is the lowest nonzero eigenvalue (or a member of the lowest closely spaced cluster of eigenvalues in cases with close roots) provided that there are no negative eigenvalues and that the SUPORT entry has been used to specify the correct number of zero eigenvalues (see **“Rigid-body Modes”**)

The following examples demonstrate the use of the EIGR data entry.

1	2	3	4	5	6	7	8	9	10
EIGR	1	AHOU				10			

In this example, the automatic Householder method is selected, and the lowest 10 modes are requested. Since the default MASS eigenvector normalization is requested, no continuation entry is needed.

EIGR	2	AHOU		100.					
	MAX								

In this example, the same method is requested, but all the modes below 100 cycles per unit time are requested with MAX vector normalization.

EIGR	3	SINV	0.1	100.0		6			
	POINT	32	3						

In this example, the Sturm modified inverse power method is requested for the first six modes found in the range specified (0.1 to 100 Hz). The POINT normalization method is requested with each eigenvector scaled such that grid point 32 in the T3 direction has a magnitude of 1.0. Note that this degree-of-freedom must be in the a-set.

## 4.7 Input Requirements for Normal Modes Analysis

This section describes input required for the selection and control of normal modes analysis.

## Executive Control Section

You can run a normal modes analysis, as an independent solution, in SOL 103 of the Structured Solution Sequences. The Executive Control Section can also contain diagnostic DIAG16, which prints the iteration information used in the SINV method. You may also run a normal modes analysis as part of the other solution sequences (such as modal transient response, modal frequency response, design optimization, and aeroelasticity).

## Case Control Section

A METHOD Case Control command is required. The METHOD command references the set identification number of an EIGR or EIGRL bulk entry that contains the real eigenvalue extraction parameters.

When you perform a modal analysis, the Simcenter Nastran output file contains various diagnostic messages and an eigenvalue analysis summary. Optional grid and element output are available using standard Case Control output requests. Eigenvectors are printed only if a DISPLACEMENT command is included. A summary of these requests are summarized in [Table 4-21](#).

To request output of modal effective mass, modal participation factors, and modal effective mass fractions, you can include a MEFFMASS Case Control command. You can optionally specify the THRESH describer on a MEFFMASS Case Control command to limit the amount of output based on effective mass fraction.

You can also use the MEFFMASS case control command to control the amount of modal content that is calculated. To do so, specify values for the MINT1, MINT2, and MINT3 describers. The software continues to calculate modes until the total effective mass fractions in the translational X-, Y-, and Z-directions exceed the values you specify for the MINT1, MINT2, and MINT3 describers, respectively.

You cannot specify the THRESH describer in combination with the MINT1, MINT2, or MINT3 describers as it causes a fatal error. You can use the MEFFMASS case control command either to exclude modes from being output or to control the amount of modal content that is calculated, but not both.

You can specify any combination of the MINT1, MINT2, and MINT3 describers. If only one describer is specified, the calculation of additional modes terminates after the specified total effective mass fraction in the corresponding direction is exceeded. If two describers are specified, the calculation of additional modes terminates when the specified total effective mass fractions in both corresponding directions are exceeded.

To satisfy the values specified for the MINT1, MINT2, and MINT3 describers, the software makes a maximum of five iterations. After the iteration limit is reached, the software stops calculating additional modes even if all of the values specified for the MINT1, MINT2, and MINT3 describers are not satisfied. To override the iteration limit of five, use the MAXIT describer.

On the EIGR and EIGRL bulk entries, you can use combinations of the F1 (V1), F2 (V2), and ND fields to specify the frequency range for mode calculation, the number of modes to calculate, or the number of modes to calculate above a frequency limit. If any of the MINT1, MINT2, and MINT3 describers is specified, the software calculates additional modes when either of the following conditions is true:

- The upper limit to the frequency range is reached prior to satisfying the MINT1, MINT2, and MINT3 describer specification. The upper limit is increased to accommodate additional mode calculation.
- The number of modes calculated does not satisfy the MINT1, MINT2, and MINT3 describer specification.

The following table summarizes the output available in SOL 103. Only SORT1 output format is supported. In general, the SORT2 option isn't available when you're performing real eigenvalue (including buckling) analyses, complex eigenvalue analyses, or flutter analyses.

**Table 4-21. Eigenvalue Extraction Output Requests**

<b>Grid Output</b>	
DISPLACEMENT	Requests the eigenvector (mode shape) for a set of grid points.
GPFORCE	Requests the grid point force balance table to be computed for each mode for a set of grid points.
GPSTRESS	Requests grid point stresses to be computed for a set of grid points. This request must be accompanied by the ELSTRESS Case Control request and the definition of stress surfaces and/or stress volumes in the OUTPUT(POST) section of the Case Control.
SPCFORCE	Requests forces of single-point constraint to be computed for a set of grid points for each mode.
<b>Element Output</b>	
ELSTRESS (or STRESS)	Requests the computation of modal stresses for a set of elements for each mode.
ESE	Requests the computation of modal element strain energies for a set of elements for each mode.
ELFORCE (or FORCE)	Requests the computation of modal element forces for a set of elements for each mode.
STRAIN	Requests the computation of modal element strains for a set of elements.
<b>Miscellaneous</b>	
MEFFMASS	Requests the computation of modal effective mass, modal participation factors, and modal effective mass fractions.
MODES	A special request that permits selective output requests to be processed on selective modes.

## Bulk Data Section

In addition to Bulk Data entries required to define the structural model, the only other required Bulk Data entry is the eigenvalue selection entry EIGR or EIGRL. The EIGR entry is used to select the modal

extraction parameters for the inverse power, Sturm modified inverse power, Householder, modified Householder, and automatic Householder methods. The EIGRL entry is used to select the modal extraction parameters for the Lanczos method.

## 4.8 Examples

This section provides several normal-modes analysis examples showing the input and output. These examples are as follows:

Model	Number of Grid Points	Element Types	Output Requests	Analysis Method	Units
bd03two	3	CELAS2 CONM2	DISPLACEMENT SPCFORCE ELFORCE	AHOU	Metric
bd03bar1, bd03bar2	11	CBAR	DISPLACEMENT	SINV	Metric
bd03bkt	236	CQUAD4 CONM2 RBE2	DISPLACEMENT STRESS ESE MODES	Lanczos	English
bd03car	972	CQUAD4 CTRIA3 CELAS2	DISPLACEMENT ESE	Lanczos	English
bd03fix	8157	CHEXA CPENTA	DISPLACEMENT	Lanczos	English
bd03plt1, bd03plt2	81	CQUAD4	DISPLACEMENT	Lanczos	Metric
bd03dmi	7	CBAR DMIG	None	Lanczos	Metric

These examples are described in the sections that follow.

### Two-DOF Model

This example is a restrained two-DOF model with two springs and two masses as illustrated in [Figure 4-5](#).

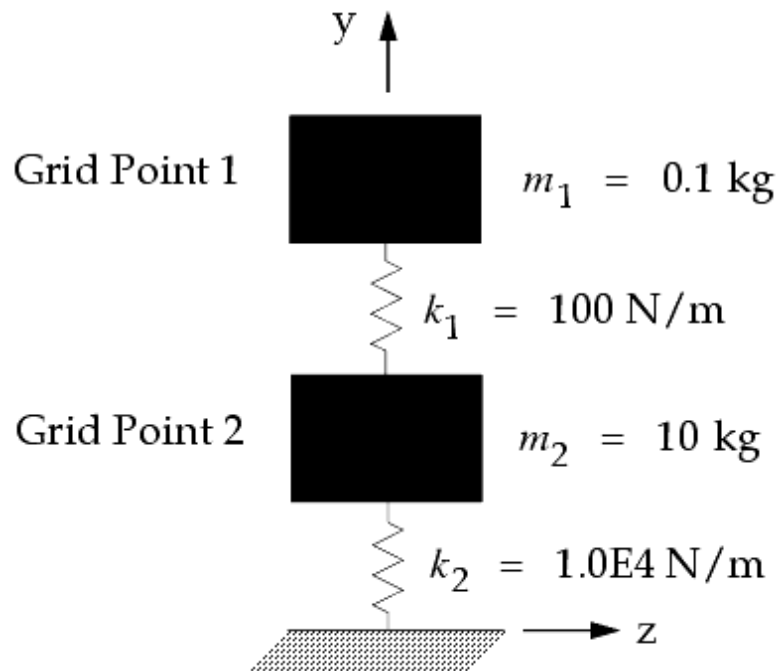


Figure 4-5. Two-DOF Model

The masses are constrained to deflect in only the y-direction. The example illustrates normal modes analysis (SOL 103) using automatic selection of the Householder or modified Householder method (METHOD = AHOU on the EIGR entry). The eigenvectors are normalized to the unit value of the largest displacement component (NORM = MAX on the EIGR entry). The input file is shown in [Figure 4-6](#).

```

$ FILE bd03two.dat
$
$ TWO DOF SYSTEM
$ CHAPTER 3, NORMAL MODES
$
TIME 5
SOL 103 $ NORMAL MODES ANALYSIS
CEND
$
TITLE = TWO DOF SYSTEM
SUBTITLE = NORMAL MODES ANALYSIS
$
$ SELECT SPC
SPC = 10
$
$ SELECT EIGR ENTRY
METHOD = 99
$
$ SELECT OUTPUT
SET 1 = 1,2
DISPLACEMENT = 1
SET 2 = 3
SPCFORCE = 2
SET 3 = 11,12
ELFORCE = 3
$

```

```

BEGIN BULK
$
$. . . . . 2 . . . . . 3 . . . . . 4 . . . . . 5 . . . . . 6 . . . . . 7 . . . . . 8 . . . . . 9 . . . . . 10 . . . . . $
$
$EIGR SID METHOD F1 F2 NE ND +EIG
$+EIG NORM G C
EIGR 99 AHOU 0. 200. +EIG1
+EIG1 MAX
$
GRID 1 0. 2. 0.
GRID 2 0. 1. 0.
GRID 3 0. 0. 0.
GRDSET 13456
CONM2 1 1 0.1
CONM2 2 2 10.0
CELAS2 11 100.0 1 2 2 2
CELAS2 12 1.0E4 2 2 3 2
SPC 10 3 2
$
ENDDATA

```

Figure 4-6. Input File for the Two-DOF Model

The printed output is shown in [Figure 4-7](#). The eigenvalue summary lists the eigenvalue  $\omega_n^2$ , circular frequency  $\omega_n$  (radians per second), natural frequency  $f_n$  (cycles per second), generalized mass (see [Equation 4-19](#)), and generalized stiffness (see [Equation 4-20](#)) for each mode.

$$\{\phi_j\}^T [M] \{\phi_j\} = m_j = \text{j-th generalized mass}$$

Equation 4-19.

$$\{\phi_j\}^T [K] \{\phi_j\} = k_j = \text{j-th generalized stiffness} = \omega^2 m_j$$

Equation 4-20.

For more information, see [Overview of Normal Modes Analysis](#).

The eigenvectors, SPC forces, and spring forces are shown for each mode.

MODE GENERALIZED NO.	EXTRACTION ORDER	EIGENVALUE GENERALIZED	R E A L E I G E N V A L U E S	
			RADIANS	CYCLES
1	1	9.048751E+02	3.008114E+01	4.787562E+00
1.904875E-01		1.723674E+02		
2	2	1.105125E+03	3.324342E+01	5.290854E+00
2.105125E-01		2.326426E+02		



```

EIGENVALUE = 9.048751E+02
CYCLES = 4.787562E+00      R E A L   E I G E N V E C T O R   N
O .
1
POINT ID.   TYPE           T1           T2           T3
R1          R2           R3
1          G           0.0          1.000000E+00  0.0          0.0
0.0        0.0
2          G           0.0          9.512492E-02  0.0          0.0
0.0        0.0
EIGENVALUE = 1.105125E+03
CYCLES = 5.290854E+00      R E A L   E I G E N V E C T O R   N
O .
2
POINT ID.   TYPE           T1           T2           T3
R1          R2           R3
1          G           0.0          1.000000E+00  0.0          0.0
0.0        0.0
2          G           0.0          -1.051249E-01  0.0          0.0
0.0        0.0
EIGENVALUE = 9.048751E+02
F O R C E S   O F   S I N G L E - P O I N T   C O N S T R
A I N T
POINT ID.   TYPE           T1           T2           T3
R1          R2           R3
3          G           0.0          -9.512491E+02  0.0          0.0
0.0        0.0
EIGENVALUE = 1.105125E+03
F O R C E S   O F   S I N G L E - P O I N T   C O N S T R
A I N T
POINT ID.   TYPE           T1           T2           T3
R1          R2           R3
3          G           0.0          1.051249E+03  0.0          0.0
0.0        0.0
EIGENVALUE = 9.048751E+02
F O R C E S   I N   S C A L A R   S P R I N G S   ( C
E L A S 2 )
ELEMENT     FORCE           ELEMENT     FORCE           ELEMENT
FORCE      ELEMENT     FORCE      ELEMENT     FORCE      ELEMENT
ID.         ID.           ID.         ID.           ID.         ID.
11         9.048751E+01  12         9.512491E+02
EIGENVALUE = 1.105125E+03
F O R C E S   I N   S C A L A R   S P R I N G S   ( C
E L A S 2 )
ELEMENT     FORCE           ELEMENT     FORCE           ELEMENT
FORCE      ELEMENT     FORCE      ELEMENT     FORCE      ELEMENT
ID.         ID.           ID.         ID.           ID.         ID.
11         1.105125E+02  12         -1.051249E+03

```

Figure 4-7. Output from the Two-DOF System

## Cantilever Beam Model

This example is a fixed-free aluminum cantilever beam with properties as shown in [Figure4-8](#).



Figure 4-8. Cantilever Beam Model

$L = 3.0\text{m}$	$r = 0.014\text{m}$	$J = 6.0\text{E-}8\text{m}^4$
$A = 6.158\text{E-}4\text{m}^2$	$I1 = I2 = 3.0\text{E-}8\text{m}^4$	$\rho_w = 2.65\text{E}4\text{N/m}^3$
$E = 7.1\text{E}10\text{N/m}^2$	$\nu = 0.33$	
Nonstructural Weight = 2.414N/m		

The  $\rho_w$  term is the weight density and must be converted to mass density  $\rho_m$  for consistency of units. PARAM,WTMASS is used to convert this weight density to mass density.

$\text{WTMASS} = 1/g = 1/9.81 = 0.102 \text{ sec}^2/\text{m}$  where  $g$  is the acceleration of gravity in  $\text{m/sec}^2$ .

Therefore,  $\rho_m = \rho_w \cdot \text{WTMASS} = 2.65\text{E}4 \cdot (0.102) = 2703 \text{ kg/m}^3$ . The nonstructural weight of 2.414 N/m is added to the beam. This nonstructural weight per length is also scaled by PARAM,WTMASS.

The example illustrates normal modes analysis (SOL 103) using the Sturm modified inverse power method (METHOD = SINV on the EIGR entry). Mass normalization (the default) is chosen for the eigenvectors. All frequencies between 0 and 50 Hz are requested. Two models are run. In the first model, manufacturing tolerances make the cross section slightly out of round, making I1 and I2 slightly different. In the second model, the cross section is perfectly round, making I1 and I2 identical.

Consider the first model. Due to the manufacturing tolerances,  $I1 = 2.9\text{E-}8\text{m}^4$  and  $I2 = 3.1\text{E-}8\text{m}^4$ . The input file is shown in [Figure 4-9](#).

```

$ FILE bd03bar1.dat
$
$ CANTILEVER BEAM MODEL
$ CHAPTER 3, NORMAL MODES
$
SOL 103 $ NORMAL MODES ANALYSIS
TIME 10
CEND
$
TITLE = CANTILEVER BEAM
SUBTITLE = NORMAL MODES
LABEL = MODEL 1 (I1 NE I2)
$

```

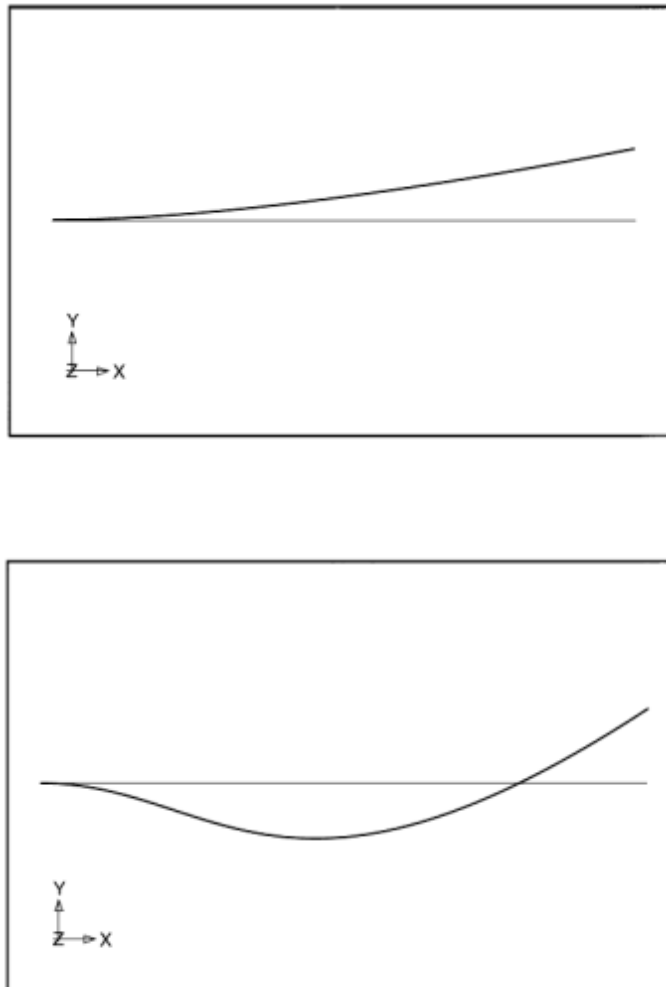
```

SPC = 1
$
$ OUTPUT REQUEST
DISPLACEMENT = ALL
$
$ SELECT EIGR ENTRY
METHOD = 10
$
BEGIN BULK
$
$. . . . .2. . . . .3. . . . .4. . . . .5. . . . .6. . . . .7. . . . .8. . . . .9. . . . .10. . . . . $
$
$EIGR SID METHOD F1 F2 NE ND +EIG
$+EIG NORM G C
EIGR 10 SINV 0. 50.
$
$ ALUMINUM PROPERTIES:
$ E = 7.1E10 N/m**2, NU = 0.33, RHO = 2.65E4 N/m**3 (W E I G H T DENSITY)
$MAT1 MID E G NU RHO
MAT1 1 7.1+10 0.33 2.65+4
$
$ CONVERT WEIGHT TO MASS: MASS = (1/G)*WEIGHT
$ G = 9.81 m/sec**2 --> WTMASS = 1/G = 0.102
PARAM WTMASS 0.102
$
$ I1 AND I2 SLIGHTLY DIFFERENT DUE TO MANUFACTURING TOLERANCE
$ ADD NONSTRUCTURAL WEIGHT OF 2.414 N/M
PBAR 1 1 6.158-4 2.9-8 3.1-8 6.-8 2.414
$
$. . . . .2. . . . .3. . . . .4. . . . .5. . . . .6. . . . .7. . . . .8. . . . .9. . . . .10. . . . . $
$
CBAR 1 1 1 2 0. 1. 0.
CBAR 2 1 2 3 0. 1. 0.
CBAR 3 1 3 4 0. 1. 0.
CBAR 4 1 4 5 0. 1. 0.
CBAR 5 1 5 6 0. 1. 0.
CBAR 6 1 6 7 0. 1. 0.
CBAR 7 1 7 8 0. 1. 0.
CBAR 8 1 8 9 0. 1. 0.
CBAR 9 1 9 10 0. 1. 0.
CBAR 10 1 10 11 0. 1. 0.
GRID 1 0.0 0. 0.
GRID 2 0.3 0. 0.
GRID 3 0.6 0. 0.
GRID 4 0.9 0. 0.
GRID 5 1.2 0. 0.
GRID 6 1.5 0. 0.
GRID 7 1.8 0. 0.
GRID 8 2.1 0. 0.
GRID 9 2.4 0. 0.
GRID 10 2.7 0. 0.
GRID 11 3.0 0. 0.
SPC1 1 123456 1
$
ENDDATA

```

Figure 4-9. Input File for the First Beam Model

The first two resulting y-direction modes are illustrated in **Figure 4-10**. Displacements in the y-direction are controlled by the I1 term. Because the structure is also free to displace in the z-direction, similar modes occur in that direction and are controlled by the I2 term.



**Figure 4-10. First Two Mode Shapes in the Y-Direction**

Printed output is shown in **Figure 4-11**. Note that modes 1 and 3 are y-direction (T2) modes and modes 2 and 4 are z-direction (T3) modes.

```

E I G E N V A L U E   A N A L Y S I S   S U M M A R Y   (STURM
INVERSE POWER)
EXTRACTED . . . . .          6      NUMBER OF EIGENVALUES
DECOMPOSITIONS . . . . .      9      NUMBER OF TRIANGULAR
ITERATIONS . . . . .         59     TOTAL NUMBER OF VECTOR
RANGE.                        REASON FOR TERMINATION: ALL EIGENVALUES FOUND IN
                                R E A L   E I G E N V A L U E S
                                RADIANS   CYCLES
MODE   EXTRACTION   EIGENVALUE

```

```

GENERALIZED          GENERALIZED
NO.
ORDER
MASS                STIFFNESS
    1                1      1.629657E+02      1.276580E+01      2.031740E+00
1.000000E+00        1.629657E+02
    2                2      1.742047E+02      1.319866E+01      2.100632E+00
1.000000E+00        1.742047E+02
    3                3      6.258656E+03      7.911166E+01      1.259101E+01
1.000000E+00        6.258656E+03
    4                4      6.690287E+03      8.179417E+01      1.301795E+01
1.000000E+00        6.690287E+03
    5                6      4.809111E+04      2.192968E+02      3.490218E+01
1.000000E+00        4.809111E+04
    6                5      5.140773E+04      2.267327E+02      3.608563E+01
1.000000E+00        5.140773E+04
EIGENVALUE = 1.629657E+02
CYCLES = 2.031740E+00
O .
POINT ID. TYPE T1 T2 T3
R1 R2 R3
    1      G    0.0    0.0    0.0
0.0    0.0
    2      G    0.0    1.391872E-02    2.259793E-10    0.0
-1.470464E-09 9.057010E-02
    3      G    0.0    5.301210E-02    8.606855E-10    0.0
-2.725000E-09 1.678406E-01
    4      G    0.0    1.133067E-01    1.839608E-09    0.0
-3.765826E-09 2.319480E-01
    5      G    0.0    1.908986E-01    3.099362E-09    0.0
-4.598456E-09 2.832321E-01
    6      G    0.0    2.820258E-01    4.578870E-09    0.0
-5.232973E-09 3.223138E-01
    7      G    0.0    3.831632E-01    6.220901E-09    0.0
-5.685052E-09 3.501587E-01
    8      G    0.0    4.911331E-01    7.973862E-09    0.0
-5.976744E-09 3.681249E-01
    9      G    0.0    6.032288E-01    9.793808E-09    0.0
-6.137029E-09 3.779972E-01
   10      G    0.0    7.173455E-01    1.164657E-08    0.0
-6.202180E-09 3.820101E-01
   11      G    0.0    8.321185E-01    1.350998E-08    0.0
-6.215970E-09 3.828595E-01
EIGENVALUE = 1.742047E+02
CYCLES = 2.100632E+00
O .
POINT ID. TYPE T1 T2 T3
R1 R2 R3
    1      G    0.0    0.0    0.0
0.0    0.0
    2      G   -1.363453E-33    2.249974E-10   -1.391872E-02    0.0
9.057010E-02 1.464532E-09
    3      G   -2.693333E-33    8.574880E-10   -5.301210E-02    0.0
1.678406E-01 2.716750E-09
    4      G   -3.956893E-33    1.834005E-09   -1.133067E-01    0.0
2.319480E-01 3.758543E-09
    5      G   -5.123021E-33    3.092054E-09   -1.908986E-01    0.0
2.832321E-01 4.594735E-09
    6      G   -6.163003E-33    4.571193E-09   -2.820258E-01    0.0
3.223138E-01 5.234397E-09

```

#### 4. Real Eigenvalue Analysis (SOL 103)

```

      7      G      -7.051232E-33      6.214489E-09      -3.831632E-01      0.0
3.501587E-01  5.692031E-09
      8      G      -7.765834E-33      7.970303E-09      -4.911331E-01      0.0
3.681249E-01  5.988579E-09
      9      G      -8.289217E-33      9.794348E-09      -6.032288E-01      0.0
3.779972E-01  6.152242E-09
     10      G      -8.608492E-33      1.165196E-08      -7.173455E-01      0.0
3.820101E-01  6.219055E-09
     11      G      -8.715798E-33      1.352052E-08      -8.321185E-01      0.0
3.828595E-01  6.233260E-09
      EIGENVALUE = 6.258656E+03
      CYCLES = 1.259101E+01
R E A L      E I G E N V E C T O R      N

```

```

O .      3
      POINT ID.      TYPE      T1      T2      T3
R1      R2      R3
      1      G      0.0      0.0      0.0      0.0
0.0      0.0
      2      G      0.0      -7.568120E-02      -9.828895E-08      0.0
5.937006E-07 -4.571418E-01
      3      G      0.0      -2.464387E-01      -3.200556E-07      0.0
8.257535E-07 -6.358200E-01
      4      G      0.0      -4.318525E-01      -5.608564E-07      0.0
7.289605E-07 -5.612913E-01
      5      G      0.0      -5.632111E-01      -7.314543E-07      0.0
3.723918E-07 -2.867391E-01
      6      G      0.0      -5.916957E-01      -7.684467E-07      0.0
-1.426249E-07  1.098159E-01
      7      G      0.0      -4.941767E-01      -6.417950E-07      0.0
-6.984853E-07  5.378201E-01
      8      G      0.0      -2.743728E-01      -3.563287E-07      0.0
-1.184615E-06  9.121327E-01
      9      G      0.0      4.170797E-02      5.417504E-08      0.0
-1.522753E-06  1.172494E+00
     10      G      0.0      4.159041E-01      5.401546E-07      0.0
-1.689206E-06  1.300660E+00
     11      G      0.0      8.124724E-01      1.055190E-06      0.0
-1.730573E-06  1.332512E+00
      EIGENVALUE = 6.690287E+03
      CYCLES = 1.301795E+01
R E A L      E I G E N V E C T O R      N

```

```

O .      4
      POINT ID.      TYPE      T1      T2      T3
R1      R2      R3
      1      G      0.0      0.0      0.0      0.0
0.0      0.0
      2      G      3.698388E-22      1.207686E-08      7.568131E-02      0.0
-4.571424E-01  1.427123E-08
      3      G      7.305710E-22      -2.529794E-08      2.464390E-01      0.0
-6.358204E-01 -3.100643E-07
      4      G      1.073314E-21      -1.866785E-07      4.318528E-01      0.0
-5.612913E-01 -7.615683E-07
      5      G      1.389628E-21      -4.615435E-07      5.632114E-01      0.0
-2.867385E-01 -1.006117E-06
      6      G      1.671725E-21      -7.409781E-07      5.916957E-01      0.0
1.098168E-01 -7.569575E-07
      7      G      1.912658E-21      -8.602044E-07      4.941765E-01      0.0
5.378208E-01  4.733732E-08
      8      G      2.106496E-21      -6.796384E-07      2.743725E-01      0.0
9.121327E-01  1.181005E-06
      9      G      2.248464E-21      -1.591279E-07      -4.170818E-02      0.0
1.172493E+00  2.239405E-06

```

```

10      G      2.335068E-21   6.217935E-07  -4.159040E-01   0.0
1.300659E+00  2.875692E-06
11      G      2.364175E-21   1.520816E-06  -8.124720E-01   0.0
1.332511E+00  3.057266E-06

```

Figure 4-11. Printed Results from the First Model

Now, consider the second model for which I1 and I2 are identical. Printed output is shown in **Figure 4-12**. Note that modes 1 and 2; 3 and 4; 5 and 6; etc., have identical frequencies; this is a case of repeated roots. Also note that the eigenvectors are not pure y- or pure z-translation (as they were in the first model); the eigenvectors are linear combinations of the y and z modes since this model has repeated roots.

```

                                E I G E N V A L U E   A N A L Y S I S   S U M M A R Y   (STURM
INVERSE POWER)
                                NUMBER OF EIGENVALUES
EXTRACTED . . . . .           8
                                NUMBER OF TRIANGULAR
DECOMPOSITIONS . . . . .     6
                                TOTAL NUMBER OF VECTOR
ITERATIONS . . . . .         68
                                REASON FOR TERMINATION: ALL EIGENVALUES FOUND IN
RANGE.
                                R E A L   E I G E N V A L U E S
                                RADIANS           CYCLES
MODE      EXTRACTION      EIGENVALUE
GENERALIZED      GENERALIZED
NO.
ORDER
MASS
      1          1          1.685851E+02          1.298403E+01          2.066473E+00
1.000000E+00    1.685851E+02
      2          3          1.685851E+02          1.298403E+01          2.066473E+00
1.000000E+00    1.685851E+02
      3          2          6.474471E+03          8.046410E+01          1.280626E+01
1.000000E+00    6.474471E+03
      4          5          6.474471E+03          8.046410E+01          1.280626E+01
1.000000E+00    6.474471E+03
      5          4          4.974941E+04          2.230458E+02          3.549883E+01
1.000000E+00    4.974941E+04
      6          6          4.974941E+04          2.230458E+02          3.549883E+01
1.000000E+00    4.974941E+04
      7          7          1.870792E+05          4.325266E+02          6.883875E+01
1.000000E+00    1.870792E+05
      8          8          1.870792E+05          4.325266E+02          6.883875E+01
1.000000E+00    1.870792E+05
      EIGENVALUE = 1.685851E+02
      CYCLES = 2.066473E+00
                                R E A L   E I G E N V E C T O R   N
O .      1
POINT ID.  TYPE      T1      T2      T3
R1      R2      R3
      1      G      0.0      0.0      0.0      0.0
0.0      0.0
      2      G      4.198658E-20  9.109908E-03  1.052335E-02  0.0
-6.847622E-02  5.927882E-02
      3      G      8.293447E-20  3.469681E-02  4.008020E-02  0.0
-1.268971E-01  1.098529E-01
      4      G      1.218317E-19  7.416008E-02  8.566642E-02  0.0
-1.753660E-01  1.518116E-01

```

#### 4. Real Eigenvalue Analysis (SOL 103)

```

      5      G      1.577188E-19      1.249444E-01      1.443303E-01      0.0
-2.141397E-01      1.853772E-01
      6      G      1.897125E-19      1.845877E-01      2.132276E-01      0.0
-2.436876E-01      2.109563E-01
      7      G      2.170278E-19      2.507827E-01      2.896933E-01      0.0
-2.647398E-01      2.291808E-01
      8      G      2.389961E-19      3.214496E-01      3.713246E-01      0.0
-2.783232E-01      2.409396E-01
      9      G      2.550813E-19      3.948168E-01      4.560753E-01      0.0
-2.857873E-01      2.474010E-01
     10      G      2.648916E-19      4.695067E-01      5.423540E-01      0.0
-2.888212E-01      2.500274E-01
     11      G      2.681884E-19      5.446261E-01      6.291288E-01      0.0
-2.894634E-01      2.505833E-01
      EIGENVALUE = 1.685851E+02
      CYCLES = 2.066473E+00
R E A L   E I G E N V E C T O R   N
O .
      2
      POINT ID.  TYPE      T1      T2      T3
R1
      R2      R3
      1      G      0.0      0.0      0.0
0.0
      2      G      1.219015E-33      -1.052334E-02      9.109882E-03      0.0
-5.927866E-02      -6.847615E-02
      3      G      2.407892E-33      -4.008016E-02      3.469673E-02      0.0
-1.098526E-01      -1.268970E-01
      4      G      3.537263E-33      -8.566635E-02      7.415994E-02      0.0
-1.518114E-01      -1.753659E-01
      5      G      4.579275E-33      -1.443302E-01      1.249442E-01      0.0
-1.853771E-01      -2.141396E-01
      6      G      5.508284E-33      -2.132276E-01      1.845875E-01      0.0
-2.109563E-01      -2.436876E-01
      7      G      6.301480E-33      -2.896932E-01      2.507825E-01      0.0
-2.291809E-01      -2.647399E-01
      8      G      6.939436E-33      -3.713246E-01      3.214495E-01      0.0
-2.409399E-01      -2.783234E-01
      9      G      7.406566E-33      -4.560753E-01      3.948168E-01      0.0
-2.474014E-01      -2.857874E-01
     10      G      7.691474E-33      -5.423541E-01      4.695069E-01      0.0
-2.500278E-01      -2.888214E-01
     11      G      7.787220E-33      -6.291289E-01      5.446264E-01      0.0
-2.505838E-01      -2.894636E-01
      EIGENVALUE = 6.474471E+03
      CYCLES = 1.280626E+01
R E A L   E I G E N V E C T O R   N
O .
      3
      POINT ID.  TYPE      T1      T2      T3
R1
      R2      R3
      1      G      0.0      0.0      0.0
0.0
      2      G      -1.606174E-26      6.893317E-02      3.123881E-02      0.0
-1.886937E-01      4.163812E-01
      3      G      -3.172614E-26      2.244652E-01      1.017221E-01      0.0
-2.624463E-01      5.791277E-01
      4      G      -4.660607E-26      3.933468E-01      1.782550E-01      0.0
-2.316833E-01      5.112443E-01
      5      G      -6.033448E-26      5.129929E-01      2.324757E-01      0.0
-1.183568E-01      2.611722E-01
      6      G      -7.257351E-26      5.389377E-01      2.442332E-01      0.0
4.532863E-02      -1.000244E-01
      7      G      -8.302282E-26      4.501138E-01      2.039804E-01      0.0
2.219953E-01      -4.898660E-01

```



```

      8      G      -9.142668E-26   2.499085E-01   1.132523E-01   0.0
3.764997E-01 -8.308035E-01
      9      G      -9.757999E-26  -3.798927E-02  -1.721591E-02   0.0
4.839685E-01 -1.067950E+00
     10      G      -1.013329E-25  -3.788206E-01  -1.716723E-01   0.0
5.368715E-01 -1.184688E+00
     11      G      -1.025940E-25  -7.400293E-01  -3.353632E-01   0.0
5.500190E-01 -1.213700E+00
      EIGENVALUE = 6.474471E+03
      CYCLES = 1.280626E+01      R E A L      E I G E N V E C T O R      N
O .
      4
      POINT ID.      TYPE      T1      T2      T3
R1      R2      R3
      1      G      0.0      0.0      0.0      0.0
0.0      0.0
      2      G      -6.181717E-19   3.123822E-02  -6.893279E-02   0.0
4.163791E-01 1.886905E-01
      3      G      -1.221121E-18   1.017205E-01  -2.244643E-01   0.0
5.791260E-01 2.624437E-01
      4      G      -1.794002E-18   1.782531E-01  -3.933455E-01   0.0
5.112445E-01 2.316835E-01
      5      G      -2.322707E-18   2.324743E-01  -5.129921E-01   0.0
2.611744E-01 1.183602E-01
      6      G      -2.794219E-18   2.442331E-01  -5.389376E-01   0.0
-1.000213E-01 -4.532376E-02
      7      G      -3.196926E-18   2.039817E-01  -4.501146E-01   0.0
-4.898637E-01 -2.219917E-01
      8      G      -3.520915E-18   1.132542E-01  -2.499097E-01   0.0
-8.308033E-01 -3.764996E-01
      9      G      -3.758207E-18  -1.721460E-02   3.798841E-02   0.0
-1.067952E+00 -4.839722E-01
     10      G      -3.902959E-18  -1.716725E-01   3.788207E-01   0.0
-1.184692E+00 -5.368777E-01
     11      G      -3.951610E-18  -3.353654E-01   7.400307E-01   0.0
-1.213704E+00 -5.500259E-01

```

**Figure 4-12. Printed Results from the Second Model**

This second model was rerun, changing the mass from lumped (the default) to coupled by adding PARAM,COUPMASS,1 to the Bulk Data. The resulting frequencies are shown below and are compared to those of the lumped mass model and the theoretical results. Note that the frequency difference is greater at higher frequencies. For most production-type models (i.e., complex three-dimensional structures), this difference is negligible.

Frequencies (Hz)		
Theory	Lumped Mass Model	Coupled Mass Model
2.076	2.066	2.076
13.010	12.806	13.010
36.428	35.499	36.437
71.384	68.838	71.451

## Bracket Model

This example is a steel bracket as shown in [Figure 4-13](#).

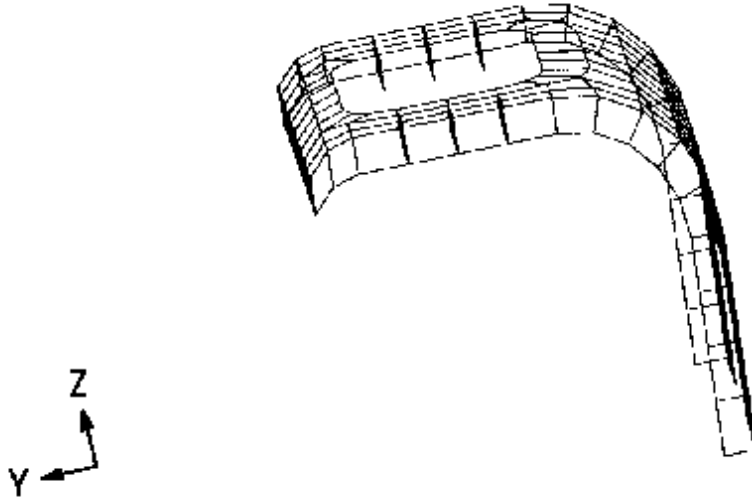


Figure 4-13. Bracket Model

A concentrated mass is suspended from the center of the hole in bracket. This mass has the following properties:

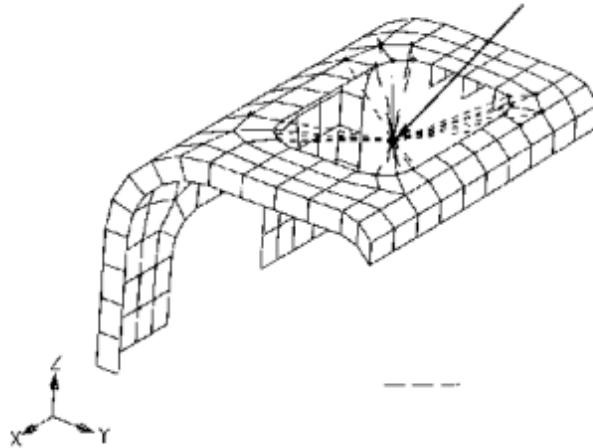
$$m = 0.0906 \text{ lb-sec}^2 / \text{in}$$

$$I_{11} = 0.35 \text{ in-lb-sec}^2$$

$$I_{22} = 0.56 \text{ in-lb-sec}^2$$

$$I_{33} = 0.07 \text{ in-lb-sec}^2$$

The concentrated mass (grid point 999) is connected to the bracket by an RBE2 element connecting 24 grid points, as shown in [Figure 4-14](#).



**Figure 4-14. Bracket Model Showing RBE2 Element (Dashed Lines)**

The bracket is clamped by constraining six degrees-of-freedom for each of 12 grid points near the base.

This example illustrates a normal modes analysis (SOL 103) using the Lanczos method (EIGRL entry). All frequencies below 100 Hz are requested. The MODES Case Control command is used to specify the number of times a subcase is repeated and therefore enables different output requests for each mode. The output requests for this problem are eigenvectors for all modes (DISPLACEMENT = ALL above the subcase level), corner stresses for the first two modes (STRESS(CORNER) = ALL and MODES = 2 in Subcase 1), and element strain energies for the third mode (ESE = ALL). An abridged version of the input file is shown in [Figure 4-15](#).

```

$ FILE bd03bkt.dat
$
$ BRACKET MODEL
$ CHAPTER 3, NORMAL MODES
$
TIME 10
SOL 103      $  NORMAL MODES ANALYSIS
CEND
$
TITLE = BRACKET MODEL
SUBTITLE = NORMAL MODES ANALYSIS
$
SPC    = 1
$
$ SELECT EIGRL
METHOD = 777
$
$ OUTPUT REQUESTS
DISPLACEMENT = ALL
SUBCASE 1
  MODES = 2  $ USE FOR FIRST TWO MODES
  STRESS(CORNER) = ALL
SUBCASE 3
  ESE = ALL
$

```

```

BEGIN BULK
$
$. . . . . 2 . . . . . 3 . . . . . 4 . . . . . 5 . . . . . 6 . . . . . 7 . . . . . 8 . . . . . 9 . . . . . 10 . . . . . $
$
$EIGRL  SID      V1      V2      ND      MSGlvl  MAXSET  SHFSCL  NORM
EIGRL   777      100.
$
$  CONCENTRATED MASS, SUPPORTED WITH AN RBE2 ELEMENT
$
GRID    999      3.0     3.0     3.8
$
$CONM2  EID      G      CID      M      X1      X2      X3      +CON1
$+CON1  I11     I21     I22     I31     I32     I33
CONM2   999     999      0.0906      +CM1
+CM1    0.35      0.56      0.07
$
RBE2    999     999     123456  126     127     91      81      80      +RB1
+RB1    90      95      129     128     96      86      85      228     +RB2
+RB2    229     199     189     190     200     195     227     226     +RB3
+RB3    194     184     185
$
$  STEEL,  M A S S  DENSITY FOR RHO
MAT1    1      3.+7    1.153+7      7.76-4
$
... basic model ...
$
ENDDATA
    
```

Figure 4-15. Abridged Input File for the Bracket Model

Figure 4-16 shows an abridged version of the resulting Simcenter Nastran output. The circular total element strain energy (ESE) for each mode of the entire sim model is  $ESE = \omega^2_i/2$  for the i-th mode when  $\varphi^T_i M \varphi_i = 1$ . The frequency of the third mode is 234.49 radians per second squared; therefore, the total strain energy of the model is  $ESE_{total} = (234.49)^2/2 = 27493$  for the third mode. The printed  $ESE_{total} = 27492$ . The eigenvectors are printed for each mode, the element corner stresses are printed for the first and second modes, and the element strain energies are printed for the third mode. (Only the headers are shown in the figure in order to save space.)

```

                                E I G E N V A L U E  A N A L Y S I S  S U M M A R Y
(LANCZOS ITERATION)
                                BLOCK SIZE USED ..... 6
                                NUMBER OF DECOMPOSITIONS ..... 2
                                NUMBER OF ROOTS FOUND ..... 3
                                NUMBER OF SOLVES REQUIRED ..... 3
                                TERMINATION MESSAGE :  REQUIRED NUMBER OF

EIGENVALUES FOUND.
                                R E A L  E I G E N V A L U E S
                                RADIANS          CYCLES
MODE      EXTRACTION      EIGENVALUE
GENERALIZED  GENERALIZED
  NO.
ORDER
MASS      STIFFNESS
      1      1      3.930304E+03      6.269214E+01      9.977763E+00
1.000000E+00      3.930304E+03
      2      2      2.878402E+04      1.696586E+02      2.700200E+01
1.000000E+00      2.878402E+04
      3      3      5.498442E+04      2.344876E+02      3.731985E+01
    
```

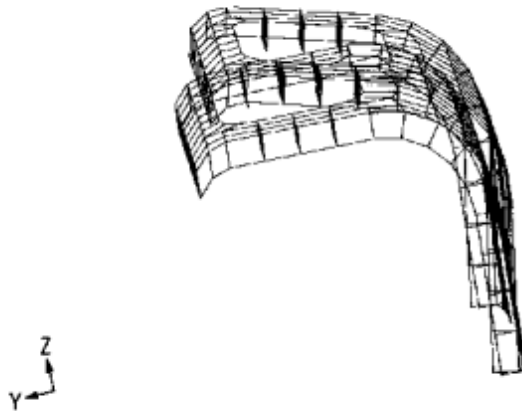
```

1.000000E+00      5.498442E+04
      EIGENVALUE = 3.930304E+03
      CYCLES = 9.977763E+00      R E A L      E I G E N V E C T O R      N
O .      1
      EIGENVALUE = 2.878402E+04
      CYCLES = 2.700200E+01      R E A L      E I G E N V E C T O R      N
O .      2
      EIGENVALUE = 5.498442E+04
      CYCLES = 3.731985E+01      R E A L      E I G E N V E C T O R      N
O .      3
      EIGENVALUE = 3.930304E+03
      S T R E S S E S      I N      Q U A D R I L A T E R A L      E L E M E N T
S ( Q U A D 4 ) OPTION = CUBIC
      EIGENVALUE = 2.878402E+04
      S T R E S S E S      I N      Q U A D R I L A T E R A L      E L E M E N T
S ( Q U A D 4 ) OPTION = CUBIC
      E L E M E N T      S T R A I N      E N E R G I E S
      * TOTAL ENERGY OF ALL ELEMENTS IN
PROBLEM      =      ELEMENT-TYPE = QUAD4
      MODE      3
      TOTAL ENERGY OF ALL ELEMENTS IN
SET      -1 = 2.749221E+04
*
      TYPE = QUAD4      SUBTOTAL      2.749221E+04
100.0000

```

**Figure 4-16. Abridged Output from the Bracket Model**

The deformed shape resulting from the first mode is illustrated in [Figure 4-17](#) and is overlaid on the undeformed shape. [Figure 4-18](#) illustrates the stress contours plotted on the deformed shape of the second mode. The element strain energy contour plot for the third mode is shown in [Figure 4-19](#).



**Figure 4-17. Deformed Shape of the First Mode**

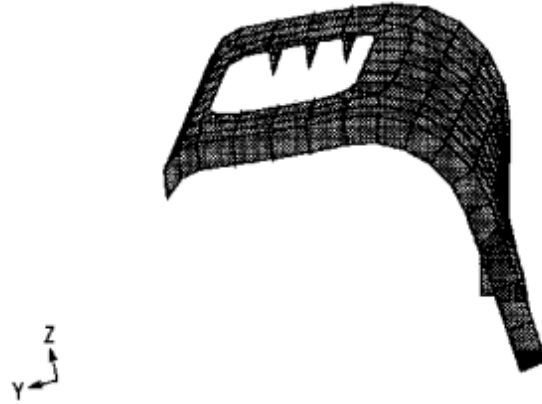


Figure 4-18. Second Mode Deformation with Element Stress Contours

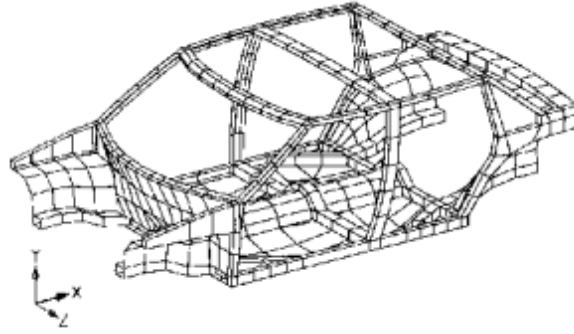


Figure 4-19. Element Strain Energy Contours for the Third Mode

## Car Frame Model

**Figure 4-20** shows a model of an aluminum car frame. The frame model is comprised of plate elements (CQUAD4 and CTRIA3), with springs (CELAS2) representing the suspension. Spring stiffnesses are input in the three translational directions; a stiffness of 500 lb/in is used in the vertical direction (T2), and stiffnesses of 1000 lb/in are used in the other translational directions (T1 and T3). When using CELASi elements to connect two grid points, it is recommended that the coordinates of the two grid points be identical in order to represent coaxial springs (noncoincident coordinates can lead to errors). The goal of the analysis is to compute resonant frequencies up to 50 Hz using the Lanczos method. Element strain energies are computed for the springs in order to help characterize the resulting modes.

**Figure 4-21** shows the input file. Modal displacements are written to the plot file and are not printed. The rigid body mass matrix is computed via the PARAM,GRDPNT,0 entry. "Include" files are used to partition the input file into several smaller files. The INCLUDE statement inserts an external file into the input file. The basic file is bd03car.dat. The springs are contained in file bd03cars.dat (**Figure 4-22**), and the rest of the input file is contained in file bd03carb.dat (not shown).



**Figure 4-20. Car Frame Model**

```

$ FILE bd03car.dat
$
$ CAR FRAME MODEL
$ CHAPTER 3, NORMAL MODES
$
$ MODEL COURTESY LAPCAD ENGINEERING
$ CHULA VISTA, CALIFORNIA
$
SOL 103 $ NORMAL MODES ANALYSIS
TIME 30
CEND
$
TITLE = CAR MODEL WITH SUSPENSION SPRINGS
SUBTITLE = MODAL ANALYSIS CASE CONTROL
ECHO = UNSORT
$
METHOD = 1
$
DISPLACEMENT(PLOT) = ALL
SET 99 = 1001,1002,1003,1011,1012,1013,
1021,1022,1023,1031,1032,1033
ESE = 99
$
BEGIN BULK
$
$. . . . .2. . . . .3. . . . .4. . . . .5. . . . .6. . . . .7. . . . .8. . . . .9. . . . .1
0. . . . . $
$
INCLUDE 'bd03cars.dat' $ Car springs
$
$EIGRL SID V1 V2 ND MSGVLV MAXSET SHFSCL NORM
EIGRL 1 -1. 50.
$
$MAT1 MID E G NU RHO

```

```

MAT1 1 1.0E7 0.33 2.59E-4
$
$ PRINT RIGID-BODY MASS
PARAM,GRDPNT,0
$
ECHOOFF
INCLUDE 'bd03carb.dat' $ Rest of Bulk Data
$
ENDDATA

```

Figure 4-21. Basic Input File for the Car Model

```

$ FILE bd03cars.dat
$
$ SPRINGS CONNECTED TO GROUND
$ GRIDS 1059,1562,1428,1895 HAVE THE SAME COORDS. AS 59,562,428,895
$
GRID 1059 152.012 32.7964 -2.90000 123456
GRID 1562 152.012 32.7964 -67.1000 123456
GRID 1428 35.6119 30.8257 -0.66667 123456
GRID 1895 35.6119 30.8257 -69.3333 123456
$
CELAS2 1001 1000. 59 1 1059 1
CELAS2 1002 500. 59 2 1059 2
CELAS2 1003 1000. 59 3 1059 3
CELAS2 1011 1000. 562 1 1562 1
CELAS2 1012 500. 562 2 1562 2
CELAS2 1013 1000. 562 3 1562 3
CELAS2 1021 1000. 428 1 1428 1
CELAS2 1022 500. 428 2 1428 2
CELAS2 1023 1000. 428 3 1428 3
CELAS2 1031 1000. 895 1 1895 1
CELAS2 1032 500. 895 2 1895 2
CELAS2 1033 1000. 895 3 1895 3

```

Figure 4-22. Input File for the Springs

**Figure 4-23** shows the grid point weight generator output. The grid point weight generator indicates that the translational mass is  $0.8027 \text{ lb-sec}^2/\text{in}$ . This type of information is useful in verifying your model. **Figure 4-24** shows some of the rest of the output. The eigenvalue analysis summary indicates that there are 12 modes below 50 Hz. The large element strain energies in the first six modes indicate that these are primarily suspension modes, comprised of the car frame acting as a rigid body supported by the flexible springs. Approximately 95% of the total strain energy is represented by spring deformation as opposed to frame deformation. Modes 7 and above show insignificant spring strain energy, indicating that these are primarily frame modes.

```

GENERATOR                                OUTPUT FROM GRID POINT WEIGHT
                                           REFERENCE POINT = 0

```



```

M O
* 8.027376E-01 0.000000E+00 0.000000E+00 0.000000E+00
-2.808118E+01 -2.179473E+01 *
* 0.000000E+00 8.027376E-01 0.000000E+00 2.808118E+01
0.000000E+00 7.610537E+01 *
* 0.000000E+00 0.000000E+00 8.027376E-01 2.179473E+01
-7.610537E+01 0.000000E+00 *
* 0.000000E+00 2.808118E+01 2.179473E+01 2.325008E+03
-2.153940E+03 2.662697E+03 *
* -2.808118E+01 0.000000E+00 -7.610537E+01 -2.153940E+03
1.020870E+04 7.626398E+02 *
* -2.179473E+01 7.610537E+01 0.000000E+00 2.662697E+03
7.626398E+02 9.437676E+03 *

S
* 1.000000E+00 0.000000E+00 0.000000E+00 *
* 0.000000E+00 1.000000E+00 0.000000E+00 *
* 0.000000E+00 0.000000E+00 1.000000E+00 *

DIRECTION
MASS AXIS SYSTEM (S)      MASS      X-C.G.
Y-C.G.      Z-C.G.
X            8.027376E-01    0.000000E+00  2.715050E
+01 -3.498177E+01
Y            8.027376E-01    9.480728E+01  0.000000E
+00 -3.498177E+01
Z            8.027376E-01    9.480728E+01  2.715050E
+01 0.000000E+00

I (S)
* 7.509408E+02 8.764180E+01 -3.962963E-01 *
* 8.764180E+01 2.011031E+03 -2.217236E-01 *
* -3.962963E-01 -2.217236E-01 1.630595E+03 *

I (Q)
* 2.017097E+03 *
* 7.448741E+02 *
* 1.630595E+03 *

Q
* 6.905332E-02 9.976128E-01 5.012719E-04 *
* -9.976128E-01 6.905355E-02 -4.555999E-04 *
* -4.891269E-04 -4.686146E-04 9.999998E-01 *

```

Figure 4-23. Output from the Grid Point Weight Generator

```

E I G E N V A L U E   A N A L Y S I S   S U M M A R Y
(LANCZOS ITERATION)
BLOCK SIZE USED ..... 7
NUMBER OF DECOMPOSITIONS ..... 3
NUMBER OF ROOTS FOUND ..... 12
NUMBER OF SOLVES REQUIRED ..... 11
TERMINATION MESSAGE : REQUIRED NUMBER OF

EIGENVALUES FOUND.

MODE      EXTRACTION      EIGENVALUE      R E A L   E I G E N V A L U E S
GENERALIZED  GENERALIZED
NO.
ORDER
MASS
1          1          2.346479E+03    4.844047E+01    7.709540E+00
1.000000E+00  2.346479E+03
2          2          2.654886E+03    5.152559E+01    8.200551E+00

```

#### 4. Real Eigenvalue Analysis (SOL 103)

1.000000E+00	2.654886E+03			
3	3	3.769821E+03	6.139887E+01	9.771934E+00
1.000000E+00	3.769821E+03			
4	4	4.633242E+03	6.806792E+01	1.083335E+01
1.000000E+00	4.633242E+03			
5	5	5.078395E+03	7.126286E+01	1.134184E+01
1.000000E+00	5.078395E+03			
6	6	8.485758E+03	9.211817E+01	1.466106E+01
1.000000E+00	8.485758E+03			
7	7	2.805541E+04	1.674975E+02	2.665805E+01
1.000000E+00	2.805541E+04			
8	8	5.350976E+04	2.313218E+02	3.681600E+01
1.000000E+00	5.350976E+04			
9	9	5.940912E+04	2.437399E+02	3.879240E+01
1.000000E+00	5.940912E+04			
10	10	8.476198E+04	2.911391E+02	4.633622E+01
1.000000E+00	8.476198E+04			
11	11	9.134271E+04	3.022296E+02	4.810133E+01
1.000000E+00	9.134271E+04			
12	12	9.726959E+04	3.118807E+02	4.963736E+01
1.000000E+00	9.726959E+04			

E L E M E N T S T R A I N E N E R G I E S

\* TOTAL ENERGY OF ALL ELEMENTS IN

PROBLEM = ELEMENT-TYPE = ELAS2

MODE 1 TOTAL ENERGY OF ALL ELEMENTS IN

SET 99 = 1.104569E+03

\* OF TOTAL STRAIN-ENERGY-DENSITY

	ELEMENT-ID	STRAIN-ENERGY	PERCENT
23.3116	1002	2.735009E+02	
	1003		
4.059090E-02	1012	2.685884E+02	
22.8929	1013		
3.240471E-02	1021		
8.017746E-02	1022	2.833448E+02	
24.1506	1023		
3.963123E-01	1031		
7.903841E-02	1032	2.781467E+02	
23.7076	1033		
3.573737E-01			
	TYPE = ELAS2	SUBTOTAL	1.104569E+03
94.1469			

E L E M E N T S T R A I N E N E R G I E S

\* TOTAL ENERGY OF ALL ELEMENTS IN

PROBLEM = ELEMENT-TYPE = ELAS2

MODE 7 TOTAL ENERGY OF ALL ELEMENTS IN

SET 99 = 3.172818E+03

\* OF TOTAL STRAIN-ENERGY-DENSITY

	ELEMENT-ID	STRAIN-ENERGY	PERCENT
--	------------	---------------	---------

OF TOTAL	STRAIN-ENERGY-DENSITY		
		1001	1.278503E
+02	.9114		
		1002	5.884620E+02
4.1950			
		1003	8.020268E
+01	.5717		
		1011	1.278420E
+02	.9114		
		1012	5.883779E+02
4.1944			
		1013	8.022697E
+01	.5719		
		1021	7.525866E
+01	.5365		
		1022	4.885996E+02
3.4831			
		1023	2.255234E+02
1.6077			
		1031	7.528390E
+01	.5367		
		1032	4.896509E+02
3.4906			
		1033	2.255398E+02
1.6078			
	TYPE = ELAS2	SUBTOTAL	3.172818E+03
22.6182			

**Figure 4-24. Abridged Output from the Car Model**

Mode shapes for modes 7, 8, 9, and 10 are shown in **Figure 4-25**. Mode 7 is an overall twisting mode; mode 8 is a "roof collapse" mode; mode 9 is a local (front) roof mode; and mode 10 is a local rear mode. Plots such as these, in conjunction with element strain energies, help to illustrate each of the mode shapes.

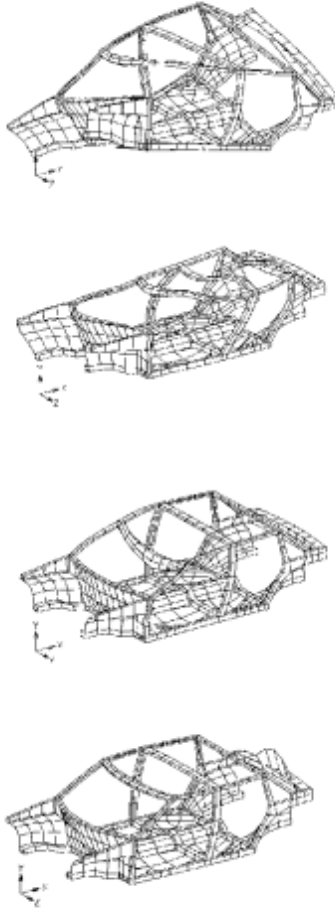


Figure 4-25. Mode Shapes for Modes 7, 8, 9, and 10

## Test Fixture Model

This example is an aluminum test fixture, which is shown in [Figure 4-26](#). The model is comprised of 8157 grid points, 5070 CHEXA elements, and 122 CPENTA elements. The primary plates are 1 inch thick, and the gusset plates are 0.5 inch thick. The base of the fixture is constrained to have no vertical ( $y$ ) motion, and the bolt holes at the base are constrained to also have no horizontal ( $x$  and  $z$ ) motion.

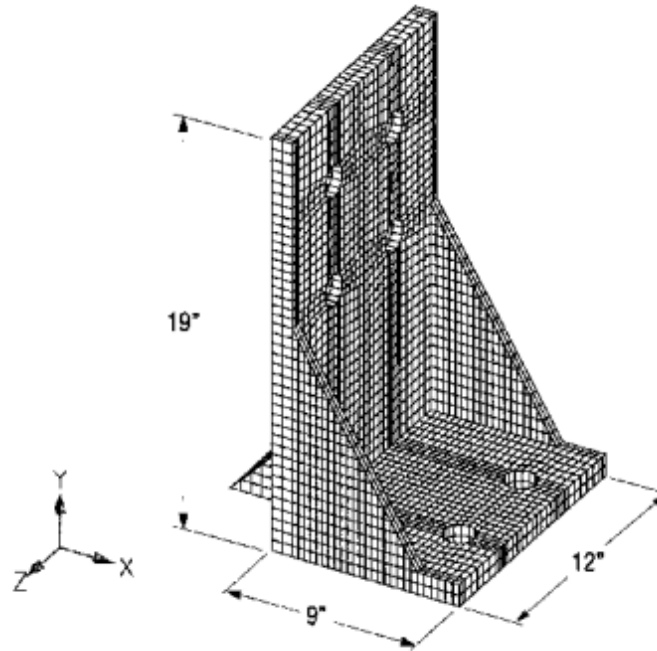


Figure 4-26. Test Fixture Model

A portion of the input file is shown in [Figure 4-27](#). The Lanczos method is used to compute the modes. The first six modes are requested (ND is 6, with V1 and V2 blank). The GRDSET Bulk Data entry removes the rotational DOFs (456) from the analysis since the solid elements have no rotational stiffness. The Bulk Data is in free format.

```

$ FILE bd03fix.dat
$
$ TEST FIXTURE
$ CHAPTER 3, NORMAL MODES
$
TIME 240
SOL 103
CEND
ECHO = NONE
DISPLACEMENT (PLOT) = ALL
SPC = 1
$
METHOD = 1
$
BEGIN BULK
$
$GRDSET, ,CP, , , , CD, PS
GRDSET , , , , , , , 456
$
$EIGRL, SID, V1, V2, ND, MSGLVL, MAXSET, SHFACL, NORM
EIGRL , 1, , , , 6

```

```

$
$MAT1, MID, E, G, NU, RHO
MAT1 , 3, 1.00E7, , 0.334, 2.5383-4
$
... basic model ...
$
ENDDATA

```

Figure 4-27. Abridged Input File for Test Fixture Model

Figure 4-28 shows the first four mode shapes. The first mode is a bending mode, the second and third modes are twist modes, and the fourth mode is a bending mode.

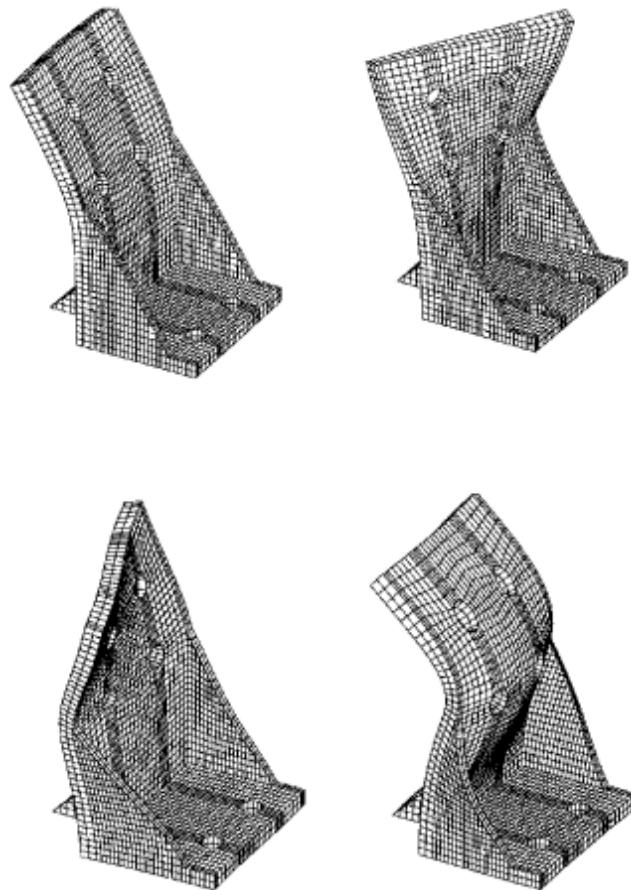


Figure 4-28. Test Fixture Mode Shapes

## Quarter Plate Model

This example is a quarter model of a simply supported flat plate, shown in Figure 4-29. This example illustrates the use of multiple boundary conditions (new in Version 68) for modeling symmetric structures. In this case the plate is doubly symmetric.

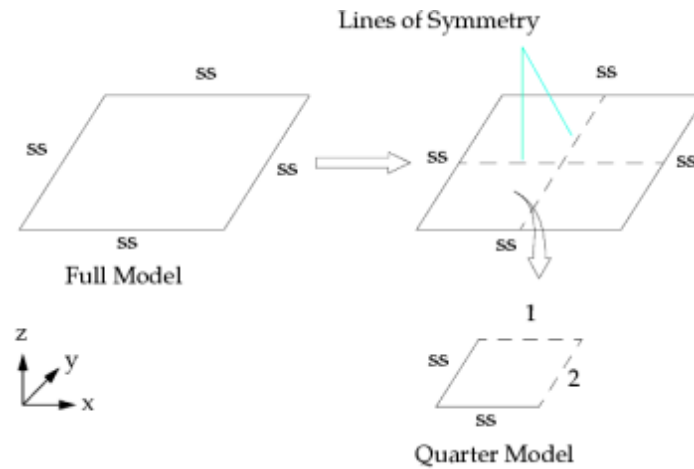


Figure 4-29. Derivation of Quarter Plate Model

A portion of the input file is shown in [Figure 4-30](#). Four subcases are used—one for each of the following sets of boundary conditions for the quarter model:

- Symmetric-antisymmetric
- Antisymmetric-symmetric
- Symmetric-symmetric
- Antisymmetric-antisymmetric

The BC Case Control command identifies multiple boundary conditions. The SPCADD Bulk Data entry defines a union of SPC sets.

```

$ FILE bd03plt1.dat
$
$ QUARTER PLATE MODEL
$ CHAPTER 3, NORMAL MODES
$
SOL 103 $ NORMAL MODES ANALYSIS
TIME 10
CEND
TITLE = SIMPLY SUPPORTED PLATE USING SYMMETRY
SUBTITLE = NORMAL MODES CASE CONTROL
LABEL = QUARTER PLATE MODEL
$
DISPLACEMENT = ALL
$
SUBCASE 1
LABEL = SYM-ASYM
BC = 1
METHOD = 1

```

```

SPC = 101
SUBCASE 2
LABEL = ASYM-SYM
BC = 2
SPC = 102
METHOD = 1
SUBCASE 3
LABEL = SYM-SYM
BC = 3
SPC = 103
METHOD = 1
SUBCASE 4
LABEL = ASYM-ASYM
BC = 4
SPC = 104
METHOD = 1
BEGIN BULK
$
$.....2.....3.....4.....5.....6.....7.....8.....9.....1
0.....
$
$ SYM-ASYM
SPCADD 101 11 1 4
$ ASYM-SYM
SPCADD 102 11 2 3
$ SYM-SYM
SPCADD 103 11 1 3
$ ASYM-ASYM
SPCADD 104 11 2 4
$
$EIGRL SID V1 V2
EIGRL 1 -0.1 100.
$
SPC 1 1 246 0.00
SPC 1 2 246 0.00
... etc. ...
SPC 11 80 12356 0.00
SPC 11 81 123456 0.00
$
... basic model ...
$
ENDDATA

```

**Figure 4-30. Input File (Abridged) for the Quarter Plate Model**

**Figure 4-31** shows the quarter plate mode shapes and the corresponding mode shapes for a full model of the same structure. Note that the quarter plate modes match the full plate modes.



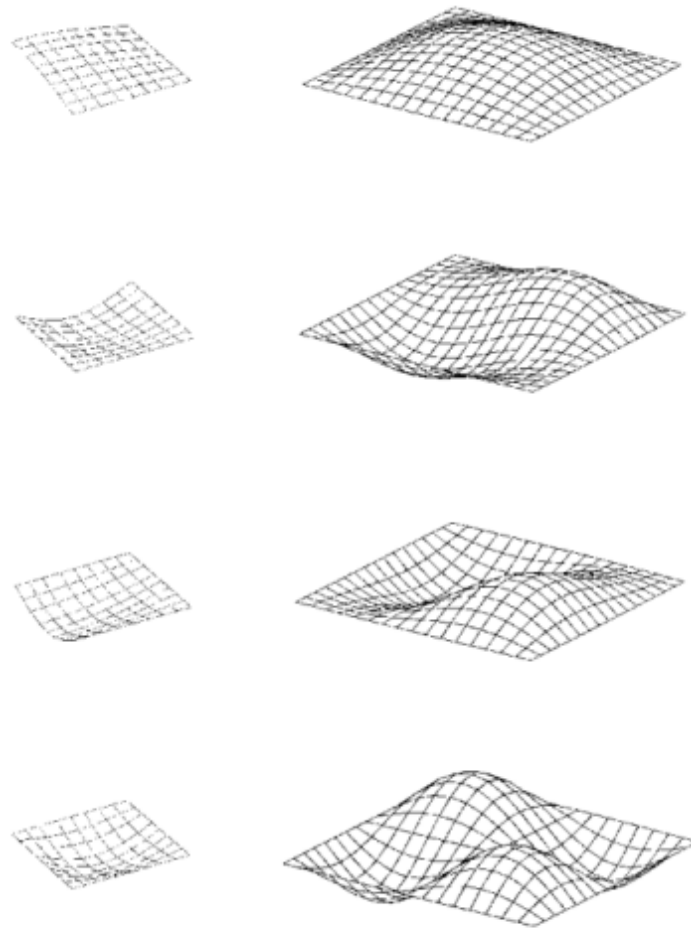


Figure 4-31. Mode Shapes for the Quarter Model (Left) and Full Model (Right)

## DMIG Example

This example illustrates the use of a DMIG entry to input external mass and stiffness. The cantilever beam model shown in [Figure 4-32](#) is used for this purpose. The model consists of CBAR elements 1 through 4. Element 5 is a model from another subcontractor that is input via DMIG entries. The model contains two DOFs (R2 and T3) per grid point.

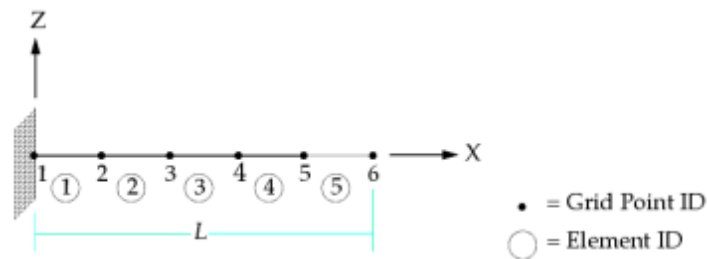


Figure 4-32. Planar Cantilever Beam

The stiffness and mass matrices from the contractor for element 5 are as follows:

$$K_5 = \begin{matrix} & \begin{matrix} 5,3 & 5,5 & 6,3 & 6,5 \end{matrix} \\ \begin{matrix} 5,3 \\ 5,5 \\ 6,3 \\ 6,5 \end{matrix} & \left( \begin{array}{cccc} 500038.8 & & & \\ -250019.4 & 166679.6 & & \\ -500038.8 & 250019.4 & 500038.8 & \\ -250019.4 & 83339.8 & 250019.4 & 166679.6 \end{array} \right) \end{matrix}$$

Equation 4-21.

$$M_5 = \begin{matrix} & \begin{matrix} 5,3 & 5,5 & 6,3 & 6,5 \end{matrix} \\ \begin{matrix} 5,3 \\ 5,5 \\ 6,3 \\ 6,5 \end{matrix} & \left( \begin{array}{cccc} 3.5829 & 0. & 0. & 0. \\ 0. & 0. & 0. & 0. \\ 0. & 0. & 3.5829 & 0. \\ 0. & 0. & 0. & 0. \end{array} \right) \end{matrix}$$

Equation 4-22.

Since the matrices are symmetric, only the lower or upper triangular portion of the matrices need to be provided via the DMIG entries.

The corresponding input file is shown in [Figure 4-33](#).

```

$ FILE bd03dmi.dat
$
$ DMIG EXAMPLE
$ CHAPTER 3, NORMAL MODES
$
SOL 103 $ NORMAL MODES ANALYSIS
TIME 10
CEND
TITLE = DMIG TO READ STIFFNESS AND MASS FOR ELEM 5
SUBTITLE = PLANAR PROBLEM
$
SPC = 10
$
$ SPECIFY K2GG AND M2GG
K2GG = EXSTIF
M2GG = EXMASS
$
METHOD = 10
$
BEGIN BULK

```

```

$
$. . . . . 2 . . . . . 3 . . . . . 4 . . . . . 5 . . . . . 6 . . . . . 7 . . . . . 8 . . . . . 9 . . . . . 10 . . . . . $
$
$EIGRL SID V1 V2 ND
EIGRL 10 2
$
CBAR 1 1 1 2 10
CBAR 2 1 2 3 10
CBAR 3 1 3 4 10
CBAR 4 1 4 5 10
$
$ HEADER ENTRY FOR STIFFNESS
DMIG EXSTIF 0 6 1
$
DMIG EXSTIF 5 3 5 3 500039. +000001
++0000015 5 -250019. 6 3 -500039. +000002
++0000026 5 -250019. +000003
$
DMIG EXSTIF 5 5 5 5 166680. +000004
++0000046 3 250019. 6 5 83340. +000005
$
DMIG EXSTIF 6 3 6 3 500039. +000006
++0000066 5 250019. +000007
$
DMIG EXSTIF 6 5 6 5 166680.
$
$ HEADER ENTRY FOR MASS
DMIG EXMASS 0 6 1
$
$ DATA ENTRIES FOR MASS
$
DMIG EXMASS 5 3 5 3 3.5829
DMIG EXMASS 6 3 6 3 3.5829
$
GRID 1 0. 0. 0. 1246
GRID 2 1. 0. 0. 1246
GRID 3 2. 0. 0. 1246
GRID 4 3. 0. 0. 1246
GRID 5 4. 0. 0. 1246
GRID 6 5. 0. 0. 1246
GRID 10 0. 0. 10. 123456
MAT1 1 7.1+10 .33 2700.
PBAR 1 1 2.654-3 5.869-7
SPC1 10 123456 1
$
ENDDATA

```

**Figure 4-33. Input File for the DMIG Example**

In this example, EXSTIF is chosen as the name of the input stiffness matrix. Therefore, in order to bring in this stiffness matrix and add it to the global stiffness matrix, the Case Control command K2GG = EXSTIF is required. Note that these stiffness terms are additions to any existing terms in the global stiffness matrix at that location and not replacements of the stiffness terms at that location. In the Bulk Data Section, five DMIG entries are required—one for the header entry and four for the data column entries since there are four non-null columns in the above matrix.

For the header entry, the same name EXSTIF must be used to match the name called out in the Case Control Section. The third field is "0", which must be the value used for the header entry. The fourth field (IFO) is set to "6" to denote a symmetric matrix input. The fifth field (TIN) is set to "1" to denote that the matrix is real, single precision.

The terms in the matrix are referenced in terms of their external grid IDs when using the DMIG entries. Physically, each term in a particular column of the stiffness matrix ( $K_{ij}$ ) represents the induced reactive load in the  $i$ -th degree-of-freedom due to a unit displacement in the  $j$ -th direction with all other displacement degrees-of-freedom held to zero. Since the matrix is symmetric, only the lower triangular portion of the matrix is input.

The first DMIG data column entry defines the first column of the above matrix. Field 2 of this DMIG entry must have the same name EXSTIF as referenced by the Case Control K2GG = EXSTIF command. Fields 3 and 4 of this entry identify this column in terms of its external grid ID and corresponding degree-of-freedom, respectively. In this case, it is grid point 5, degree-of-freedom 3 (z-translation at grid point 5). Once this column is defined, follow the format description as described in the section for column data entry format, and then you can input the four terms in this column row-by-row. These four terms are defined by sets of three fields. They are the external grid ID number, corresponding degree-of-freedom, and the actual matrix term, respectively. The first term of column one is defined by external grid ID 5, degree-of-freedom 3 (z-translation at grid point 5) with a stiffness value of 500039. The second term of column one is defined by external grid ID 5, degree-of-freedom 5 (y-rotation at grid point 5) with a stiffness value of -250019. The third term of column one is defined by external grid ID 6, degree-of-freedom 3 (z-translation at grid point 6) with a stiffness value of -500039. The fourth term of column one is defined by external grid ID 6, degree-of-freedom 5 (y-rotation at grid point 6) with a stiffness value of -250019.

The next DMIG entry defines the second column of the above matrix. Fields 3 and 4 of this entry identify this column in terms of its external grid ID and corresponding degree-of-freedom, respectively. In this case, it is grid point 5, degree-of-freedom 5 (y-rotation at grid point 5). The rest of the procedure is similar to that of column one with the exception that only three terms need to be input due to symmetry.

The next two DMIG entries defines columns three and four of the stiffness matrix, respectively. Note that due to symmetry, one less row needs to be defined for each additional column.

The mass matrix is input in a similar manner as the stiffness matrix with the following exceptions:

- The command M2GG = EXMASS instead of K2GG = EXSTIF is used in the Case Control Section. In this case, EXMASS is the name of the mass matrix referenced in field two of the DMIG Bulk Data entries.
- The matrix defined in the DMIG entries is expressed in the mass matrix terms rather than in stiffness matrix terms.
- Since there are only two non-null columns for the mass matrix, only two DMIG data entries are required instead of the four entries needed for the stiffness matrix.

Mass matrices input using DMIG are not scaled by PARAM,WTMASS.

In this example the small-field input format is used, and the maximum number of characters that can be input are eight (including sign and decimal point). Greater input precision can be achieved by using the large-field format and by changing the TIN field to 2 for the DMIG entries.

The first two computed natural frequencies for this example are 1.676 Hz and 10.066 Hz.

## 4.9 Residual Vectors for Modal Response Solutions

Typically, only a subset of all the normal modes of a structure are computed for use in a modal response solution (SOL 111 or 112). Although the computed modes may represent most of the dynamic content of the structure, the omission of higher frequency modes can lead to errors in the resulting response prediction because higher frequency modes may contribute statically to the response. Residual vectors can optionally be used to improve the accuracy of the response prediction by accounting for the missing static flexibility associated with higher frequency modes.

Residual vectors are displacement shapes found from a static analysis that have been normalized. If a residual vector is not orthogonal to the normal mode shapes and the other residual vectors, it is discarded. The residual vectors that remain are used to augment the normal mode shapes in the eigenvector matrix. The augmented eigenvalue matrix is then used to transform the system matrices in physical space to a reduced representation in modal space. The reduced representation of the system in modal space is then used in the response calculation.

To tailor a residual vector request to your needs, you can use the RESVEC case control command. The residual vectors obtained by specifying the various describers of the RESVEC case control command are summarized as follows:

- If you specify the APPLDID describer, residual vectors are requested for the DOF associated with the applied loads. There are two input scenarios to determine the applied loads for computing residual vectors.
  - The load set IDs selected with the EXCITEID field on all RLOAD1, RLOAD2, TLOAD1, TLOAD2, ACSRCE, and SELOAD entries in the bulk data are processed. No DLOAD case control is required in this case. If a LOADSET case control command exists, the load set IDs selected with the EXCITEID field on RLOAD1, RLOAD2, TLOAD1, TLOAD2, ACSRCE, and SELOAD entries are ignored.
  - If a LOADSET case control command exists, and it selects an LSEQ bulk entry, the load set ID selected in the LID field in the LSEQ bulk entry is processed.
- If you specify the RVDOF describer, residual vectors are requested for DOF included in RVDOF and RVDOF1 bulk entries. RVDOF and RVDOF1 are bulk entries that you can use to select DOF in both the residual structure (a-set) and superelements (o-set).  
 In comparison, when requesting residual vectors using the PARAM,RESVEC,YES entry, the USET,U6 bulk entry can be used to select residual structure DOF (a-set), and the SEUSET,U6 bulk entry can be used to select superelement DOF (o-set). The RVDOF and RVDOF1 bulk entries have the advantage that they can select DOF in both the residual structure (a-set) and in superelements (o-set).  
 In addition, the unit loads applied to the interior points of a superelement due to RVDOF and RVDOF1 bulk entries are passed downstream to the residual for the purpose of residual vector processing by all

superelements in its downstream path. This produces more accurate results as compared to the results produced when USETi,U6 or SEUSETi,U6 bulk entries are used for residual vector processing. When USETi,U6 or SEUSETi,U6 bulk entries are used, unit loads on a superelement are not passed downstream for residual vector processing by the downstream superelements.

- If you specify the RVEL describer, residual vectors are requested for DOF used to define CBUSH, CBUSH1D, CDAMPi, CELASi, and CVISC elements that are referenced by RVEL bulk entries even if the referenced CBUSH, CBUSH1D, CDAMPi, CELASi, and CVISC elements have one end grounded.
- If you specify the INRLOD describer, residual vectors are requested for DOF associated with inertia relief loads (unit acceleration of mass).
- If you specify the DAMPLOD describer, residual vectors are requested for DOF used to define viscous damping on CBUSHi, CDAMPi, and CVISC elements. If the DAMPLOD and RVEL describers are both specified, residual vectors are requested for DOF used to define viscous damping on CBUSHi, CDAMPi, and CVISC elements, and DOF listed on the RVEL bulk entries.
- If you specify the DYNRSP describer, the dynamic effect of residual vector modes are included in the forced response solution.

The right hand side of the RESVEC command includes the SYSTEM / NOSYSTEM, COMPONENT / NOCOMPONENT, BOTH or YES, and NO describers.

- You can use the SYSTEM describer to request residual vectors for residual structure DOF (a-set) only.
- You can use the COMPONENT describer to request residual vectors for superelement DOF (o-set) only.
- You can use the BOTH or YES describers to request residual vectors for both residual structure DOF (a-set) and for superelement DOF (o-set).
- You can use the NO describer to request that no residual vectors be computed.
- A unique RESVEC case control definition can be defined in different superelement subcases. You can use the NOSYSTEM and NOCOMPONENT describers in a superelement subcase to turn off a setting which may have been defined globally.

In the absence of a RESVEC case control command, the residual vectors calculated by default depend on solution type.

- For SOLs 103, 106 (PARAM,NMLOOP,0), 110, 115, 153 (PARAM,NMLOOP,0), and 187, the residual vectors calculated are equivalent to specifying RESVEC = COMPONENT.
- For SOLs 111, 112, 118, 146, and 200 (ANALYSIS = MODES, MCEIG, MTRAN, or MFREQ), the residual vectors calculated are equivalent to specifying RESVEC = BOTH.

To disable residual vectors being calculated by default, add PARAM,RESVEC,NO and PARAM,RESVINER,NO to *installation\_path/conf/nast9.rcf*.

You can optionally use the RESVEC parameter to make many of the same residual vector requests. **Table 4-24** summarizes the various residual vector requests you can make using either the RESVEC case control command or the RESVEC parameter. If the RESVEC case control command is present, the parameters RESVEC and RESVNER are ignored.

**Table 4-24. Residual Vector Request Options**

RESVEC Case Control Command Describer	Equivalent Parameter Input	Result
APPLOD	PARAM,RESVEC,YES	The APPLOD describer and PARAM,RESVEC,YES request residual vectors for DOF associated with applied loads.  PARAM,RESVEC,YES also requests residual vectors for all DOF included in USET,U6 and SEUSET,U6 bulk entries. Although, the software ignores USET,U6 and SEUSET,U6 bulk entries when the RESVEC case control command is defined.
RVDOF		Residual vectors are requested for all DOF included in RVDOF and RVDOF1 bulk entries.
INRLOD	PARAM,RESVNER,YES	Residual vectors are requested for DOF associated with inertia relief loads (unit acceleration of mass).
RVEL		Residual vectors are requested for DOF used to define CBUSH, CBUSH1D, CDAMPi, CELASi, and CVISC elements that are referenced by RVEL bulk entries.
SYSTEM		Residual vectors are requested for residual structure DOF (a-set) only.
COMPONENT		Residual vectors are requested for superelement DOF (o-set) only.
BOTH or YES		Residual vectors are requested for both residual structure DOF (a-set) and for superelement DOF (o-set).

**Table 4-25** contains other parameters and RESVEC describers that optionally can be used when requesting residual vectors. The RESVEC case control command or PARAM,RESVEC,YES must be specified in order to use any of the options listed in **Table 4-25**.

**Table 4-25. Other Parameters and Describers Related to Residual Vectors**

Parameter Input	Result
DYNRSP / NODYNRSP describers	DYNRSP includes the dynamic effects of residual vector modes in the response calculation. NODYNRSP excludes the dynamic effects of residual vector modes in the response calculation.
PARAM,RESVSE	The default PARAM,RESVSE,NO will skip the printing of the strain energy of the static shapes (labeled "EXTERNAL WORK"). You can use PARAM,RESVSE,YES to print the strain energy.
PARAM,RESVSLI	The default PARAM,RESVSLI,YES will test for linearly independent shapes. You can use PARAM,RESVSLI,NO to skip the tests.
PARAM,RESVSO	The default PARAM,RESVSO,YES will clean up the static vectors to insure orthogonality to modes. If the stiffness matrix is well conditioned this is unnecessary. You can use PARAM,RESVSO,NO to skip the reorthogonalization.
PARAM,RESMETH	Method used for eigensolution in reduced basis. PARAM,RESMETH,LAN is an alternate method, somewhat more robust but slower for this class of reduced basis problem. Default = AHOU.
PARAM,RESVPGF	This parameter is used to filter out small terms in the residual vectors. The default is 1.E-6, meaning that terms in the residual vectors whose absolute values are smaller than 1.E-6 will get filtered out before they are used. To eliminate filtering of residual vectors, you can set RESVPGF to 0.0.
PARAM,RESVRAT	Filter value for discarding trial vectors with little independent content. You can decrease to discard more vectors. Default = 1.e8.

For additional information, see Residual Flexibility in the SubDMAP RESVEC in the *User's Guide*, and the RESVEC case control command in the *Quick Reference Guide*.

## 4.10 Differential Stiffness

To account for differential stiffness (stress stiffening) in a dynamic analysis, two subcases are required. The first subcase is a static subcase that contains the loads that give rise to stress-stiffening. The second subcase is the dynamic subcase. The dynamic subcase must contain  $STATSUB = n$ , where  $n$  is the static subcase identification number. If the structure is unconstrained or partially constrained, it is not necessary for you to fully constrain the structure in the static subcase. The unconstrained or partially constrained model will run without singularity in the static subcase, and the differential stiffness will be calculated and passed to the dynamic subcase automatically if you include PARAM,AUTOSPRT,YES (the default) in your input file. This functionality is available for SOLs 103, 107, 108, 109, 110, 111, and 112. To disable this functionality, include PARAM,AUTOSPRT,NO in your input file.

For modal solutions with differential stiffness, there will be some grounding of the rotational modes. For unconstrained structures, a modal solution with differential stiffness will produce translational rigid



body modes, but will not produce rotational modes with zero eigenvalues. This behavior is a known characteristic of differential (stress) stiffness matrices.



# 5. Rigid Body Modes

## 5.1 Overview of Rigid Body Modes

A structure or a portion of a structure can displace without developing internal loads or stresses if it is not sufficiently tied to ground (constrained). These stress-free displacements are categorized as rigid-body modes or mechanism modes.

### Rigid-body Modes

Rigid-body modes occur in unconstrained structures, such as satellites and aircraft in flight. For a general, unconstrained 3-D structure without mechanisms, there are six rigid-body modes often described as T1, T2, T3, R1, R2, and R3, or combinations thereof. Rigid-body modes can also be approximated for certain kinds of dynamic or modal tests in which the test specimen is supported by very flexible supports, such as bungee cords or inflatable bags. In this case the test specimen itself does not distort for the lowest mode(s) but instead displaces as a rigid body with all of the deformation occurring in the flexible support. Rigid-body modes can improperly occur if a structure that should be constrained is not fully constrained (for example, in a building model for which the boundary conditions (SPCs) were forgotten).

### Mechanism Modes

A mechanism mode occurs when a portion of the structure can displace as a rigid body, which can occur when there is an internal hinge in the structure. An example of a mechanism is a ball and socket joint or a rudder in an airplane. A mechanism mode can also occur when two parts of a structure are improperly joined. A common modeling error resulting in a mechanism is when a bar is cantilevered from a solid element; the bar has rotational stiffness and the solid has no rotational stiffness, resulting in a pinned connection when the two are joined.

The presence of rigid-body and/or mechanism modes is indicated by zero frequency eigenvalues. Due to computer roundoff, the zero frequency eigenvalues are numerical zeroes on the order of 1.0E-4 Hz or less for typical structures. The same unconstrained model may give different values of the rigid-body frequencies when run on different computer types.

Rigid-body modes generated by Simcenter Nastran are orthogonal with respect to each other and with respect to the flexible (or elastic) modes. Each rigid-body mode has the following property:

$$\{\phi_{rig}\}^T [M] \{\phi_{rig}\} > 0$$
$$\{\phi_{rig}\}^T [K] \{\phi_{rig}\} = 0$$

Equation 5-1.

where  $\{\phi_{rig}\}$  denotes the rigid-body mode shapes,  $[M]$  denotes the mass matrix, and  $[K]$  denotes the stiffness matrix. The rigid-body modes contain no strain energy.

**Figure 5-1** depicts the rigid-body modes for a two-dimensional model of a straight beam. Because the two-dimensional model has three DOFs per grid point (x-translation, y-translation, and z-rotation), there are three rigid-body modes. The figure shows classical rigid-body modes, whereby one mode is purely x-translation, another is purely y-translation, and another is purely z-rotation about the center of the beam. Because rigid-body modes are a special case of repeated roots, any linear combination of these displacement shapes also comprises a valid set of rigid-body modes.

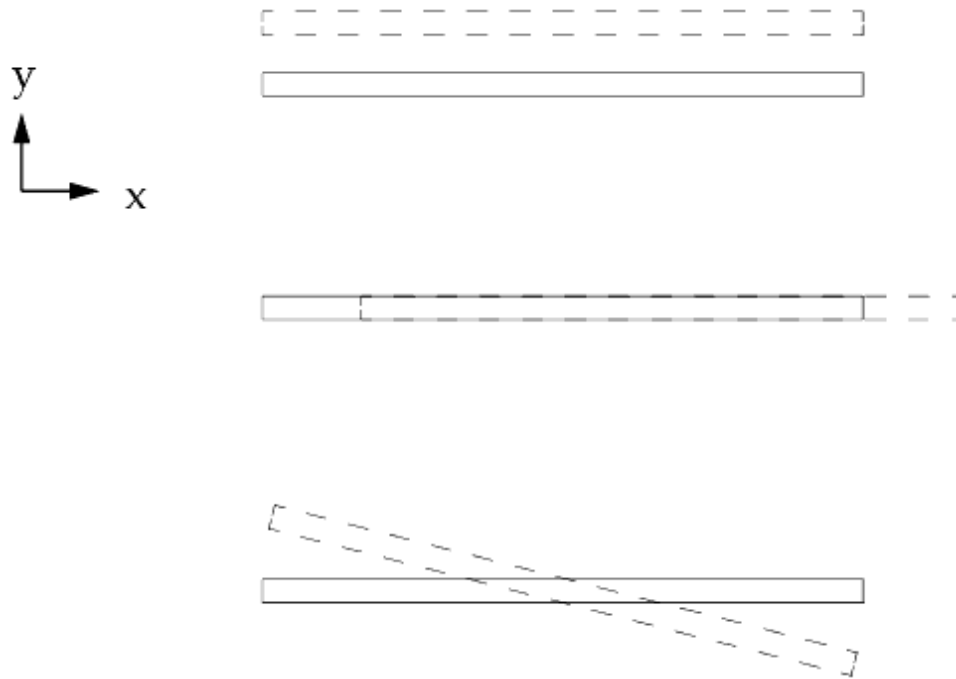


Figure 5-1. Rigid-Body Modes of a 2-D Beam

## 5.2 SUPORT Entry

Rigid-body modes are computed in Simcenter Nastran without requiring special user intervention, although the use of a SUPORT Bulk Data entry makes the mode shapes look cleaner. The SUPORT entry also makes the rigid-body mode shapes repeatable when the mass or stiffness of the model changes and the model is reanalyzed. The SUPORT (note spelling!) entry does not constrain the model; it simply provides a frame of reference for the rigid-body shape calculations by defining the r-set components of motion. The SUPORT entry is not required for any of the dynamic analysis methods except for response spectrum analysis (see **“Advanced Dynamic Analysis Capabilities”**).

If the SUPORT is used, one DOF should be specified for each rigid-body mode or mechanism. The form of the SUPORT entry is shown below:

1	2	3	4	5	6	7	8	9	10
SUPPORT	ID1	C1	ID2	C2	ID3	C3	ID4	C4	

Field	Contents
IDi	Grid or scalar point identification number.
Ci	Component numbers. (0 or blank for scalar points and any unique combination of the integers 1 through 6 for grid points).

## Treatment of SUPORT by Eigenvalue Analysis Methods

The eigenvalue extraction methods treat the SUPORT entry differently as described below.

### EIGR Methods

Each of the eigenvalue extraction methods selected on the EIGR Bulk Data entry ( AHOU, HOU, MHOu, and SINv) treats the SUPORT in the same manner. Eigenvalues are first computed using the information on the EIGR Bulk Data entry. The frequencies of the first  $N_r$  modes (where  $N_r$  is the number of SUPORT DOFs) are replaced with a value of 0.0 Hz. The first  $N_r$  eigenvectors are replaced by modes that are calculated by moving each SUPORT DOF a unit distance and then mass orthogonalizing them with respect to the other modes. The fact that the eigenvectors are calculated via kinematics is the reason that the SUPORT entry produces cleaner rigid-body modes (as opposed to the rigid-body modes computed without the use of the SUPORT entry).

Note that Simcenter Nastran has no built-in checks to warn if any of the  $N_r$  eigenvalues are not rigid-body modes. This replacement can hide potential modeling problems. The results from UIM 3035 should be carefully checked (see **“Examples”**). The strain energy term for each SUPORT DOF should be zero.

A poor choice of DOFs on the SUPORT entry can adversely affect the computation of the flexible modes for the SINv method. Flexible modes computed with the other methods ( AHOU, HOU, and MHOu) are not adversely affected by a poor choice of SUPORT DOFs. Again, the results of UIM 3035 should be carefully checked to ensure that a proper choice of SUPORT DOFs is made.

### Lanczos Method

In the Lanczos method, the SUPORT entry attempts to give 0.0 Hz modes. The Lanczos routine compares the values of the calculated modes (without the SUPORT entry) and determines if the calculated frequencies for the  $N_r$  modes are near 0.0 Hz. If the computed frequencies are near 0.0 Hz, the computed values are replaced with 0.0 Hz. If a computed frequency is not near 0.0 Hz, then it is retained. (Note that this may cause problems in response spectrum analysis where 0.0 Hz is required for the SUPORT frequencies.) The Lanczos-computed, rigid-body eigenvectors are used, i.e., the rigid-body eigenvectors computed by the SUPORT entry are not used.

In the above paragraph, “near” means that the eigenvalues are less than 0.01 times the shift scale (the SHFSCl field on the EIGRL entry). All computed eigenvalues less than this threshold are candidate rigid-body modes. There is some logic to determine the relationship of these candidate rigid-body modes to the number of DOFs on the SUPORT entry. Suppose that there are three eigenvalues less than the threshold of 0.01 times the shift scale. If your SUPORT entry defines two DOFs, then the first two frequencies become 0.0 Hz, and the third becomes something that is nonzero but small (on the order of 1.0E-6, for example). On the other hand, if your SUPORT entry defines four DOFs, then only the first

three are treated as rigid-body modes, and the fourth is not replaced. Furthermore, the use of a SUPORT entry forces an extra decomposition, which increases the computer run time.

Because the SUPORT entry is not used to compute the rigid-body eigenvectors, there is probably little to be gained by using the SUPORT entry with the Lanczos method unless response spectrum analysis is being performed.

## Theoretical Considerations for Using SUPORT

Degrees-of-freedom defined on the SUPORT entry are placed in the r-set. When an r-set is present, static rigid-body vectors are calculated in Simcenter Nastran by first partitioning the a-set into the r- and l-sets

$$\{u_a\} = \begin{Bmatrix} u_l \\ u_r \end{Bmatrix}$$

Equation 5-2.

Introducing this partitioning in the stiffness matrix results in

$$\begin{bmatrix} K_{ll} & K_{lr} \\ K_{rl} & K_{rr} \end{bmatrix} \begin{Bmatrix} u_l \\ u_r \end{Bmatrix} = \begin{Bmatrix} 0 \\ P_r \end{Bmatrix}$$

Equation 5-3.

for the rigid-body modes defined in the r-set.

There is no load on l-set DOFs. The load  $\{P_r\}$  on the r-set is not needed in subsequent equations. Then solve for  $u_l$  in terms of  $u_r$

$$\{u_l\} = [D]\{u_r\}$$

Equation 5-4.

where:

$$[D] = -[K_{ll}]^{-1}[K_{lr}]$$

The matrix  $[D]$  is used to construct a set of rigid-body vectors  $[\psi_{rig}]$

$$[\psi_{rig}] = \begin{bmatrix} D \\ I_r \end{bmatrix}$$

Equation 5-5.

where  $[\psi_{rig}]$  represents the motion of the a-set for a unit motion of each SUPORT DOF with all other SUPORT DOFs constrained and  $[I_r]$  is an  $r \cdot r$  identity matrix.

The rigid-body vectors can be used to create a rigid-body mass matrix  $[M_r]$

$$[M_r] = [\psi_{rig}]^T [M_{aa}] [\psi_{rig}] = \begin{bmatrix} D \\ I_r \end{bmatrix}^T [M_{aa}] \begin{bmatrix} D \\ I_r \end{bmatrix}$$

Equation 5-6.

To improve the quality of the rigid-body mode shapes, orthogonalization is applied to create a diagonal mass matrix  $[M_o]$  by

$$[M_o] = [\phi_{ro}]^T [M_r] = [\phi_{ro}]$$

Equation 5-7.

where:

$[\phi_{ro}]$  is a transformation matrix.

This transformation matrix is used to construct the final set of rigid-body mode shape vectors by

$$[\phi_{rig}] = [\psi_{rig}] [\phi_{ro}] = \begin{bmatrix} D \phi_{ro} \\ \phi_{ro} \end{bmatrix}$$

Equation 5-8.

such that

$$[\phi_{rig}]^T [K_{aa}] [\phi_{rig}] \approx 0$$

Equation 5-9.

$$[\phi_{rig}]^T [M_{aa}] [\phi_{rig}] = [M_o]$$

Equation 5-10.

where:

$$[M_o]$$

is a diagonal matrix.

Care must be taken when selecting SUPORT DOFs. Each SUPORT DOF must be able to displace independently without developing internal stresses. In other words, the SUPORT DOFs must be statically determinate. The SUPORT is used only to facilitate the calculation of rigid-body vectors. If you do not specify the r-set DOFs, the rigid-body modes are calculated directly by the method selected for the flexible frequency modes. If an insufficient number of r-set DOFs are specified, the calculation of rigid-body modes is unreliable.

As a modeling aid, Simcenter Nastran calculates equivalent internal strain energy (work) for each rigid-body vector as follows:

$$[X] = \begin{bmatrix} D^T & I_r \end{bmatrix} \begin{bmatrix} K_{ll} & K_{lr} \\ K_{rl} & K_{rr} \end{bmatrix} \begin{bmatrix} D \\ I_r \end{bmatrix}$$

Equation 5-11.

which can be simplified as

$$[X] = [D]^T [K_{ll}] [D] + [K_{rr}]$$

Equation 5-12.



When r-set DOFs exist, the printed strain energies are the diagonal elements of  $[X]$  divided by 2 and should be approximately zero.

Note that  $[X]$  is the transformation of the stiffness matrix  $[K_{\alpha\alpha}]$  to r-set coordinates, which by definition of rigid-body (i.e., zero frequency) vector properties should be null. If this is not the case, the equilibrium may be violated by the r-set choice or other modeling errors may exist. The matrix  $[X]$  is also called the rigid-body check matrix.

Simcenter Nastran also calculates a rigid-body error ratio

$$\varepsilon = \frac{\| [K_{rr}] + [K_{lr}]^T [D] \|}{\| K_{rr} \|}$$

Equation 5-13.

where:  $\| \|$  = Euclidean norm of the matrix =  $\sqrt{\sum_i \sum_j x_{ij}^2}$

One value of  $\varepsilon$  is calculated using **Equation 5-13** based on all SUPORT DOFs. Therefore, in UIM 3035 the same  $\varepsilon$  is printed for every supported DOF.

The rigid-body error ratio and the strain energy should be zero if a set of statically determinate SUPORT DOFs is chosen. Roundoff error may lead to computational zero values for these quantities. ("Computational zero" is a small number ( $10^{-5}$ , for example) that normally is 0.0 except for numerical roundoff.) The rigid-body error ratio and strain energy may be significantly nonzero for any of the following reasons:

- Roundoff error accumulation.
- The  $u_r$  set is overdetermined, leading to redundant supports. The condition gives high strain energy.
- The  $u_r$  set is underspecified, leading to a singular reduced-stiffness matrix and a MAXRATIO error. This condition gives a high rigid-body error ratio.
- The multipoint constraints are statically indeterminate. This condition gives high strain energy and a high rigid-body error ratio.
- There are too many single-point constraints. This condition gives high strain energy and a high rigid-body error ratio.
- $[K_{rr}]$  is null. This condition gives a unit value for the rigid-body error but low strain energy (see **"Advanced Dynamic Analysis Capabilities"** ).

## Modeling Considerations for using SUPORT

When using a SUPORT you must select a set of DOFs that is capable of constraining all the rigid-body modes. Another way to state this requirement is that the r-set must be able to constrain the structure in a statically determinate manner. There are usually many choices of DOFs that satisfy this requirement. Two choices that work for simple three-dimensional structures are:

- Six DOFs on one grid point when all its degrees-of-freedom have stiffness.
- Three translation DOFs normal to one plane, two translation DOFs normal to an orthogonal plane, and then one translational DOF normal to the last orthogonal plane. Such a system can be used, for instance, on a model composed entirely of solid elements that have no inherent stiffness for grid point rotation. See [Figure 5-2](#).

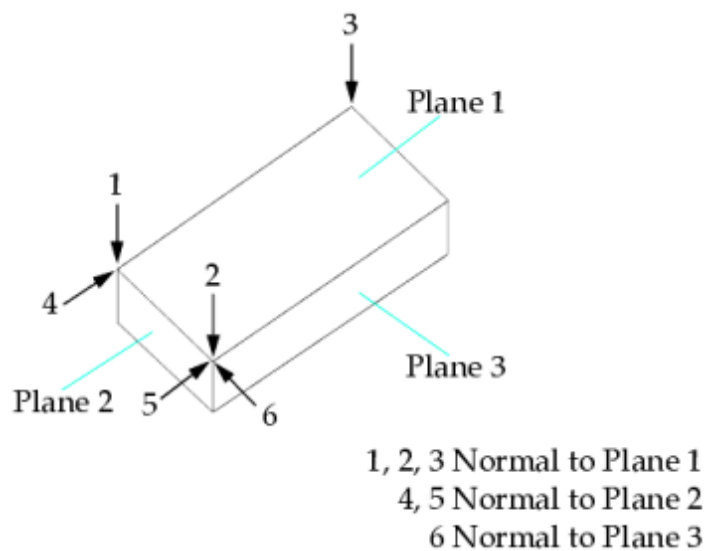


Figure 5-2. Statically Determinate r-set

There are special cases where a model need not have six rigid-body modes. A planar model has only three rigid-body modes, while an airplane with a free rudder has seven, for example. If you use the SUPORT, it is your responsibility to determine all the modes of rigid-body motion, then provide r-set DOFs that define these rigid-body modes. Another special case is the application of enforced motion by the large mass technique (see [“Enforced Motion”](#)). If the input points describe redundant load paths, diagnostics are produced that indicate overconstraint. For this case, these diagnostics may be safely ignored.

Poorly-constrained rigid-body modes result from either constraining DOFs with relatively little stiffness or from constraining a set of DOFs that are almost linearly dependent on one another. An example of the former is a model of a very thin cylindrical shell. The degrees-of-freedom normal to the shell and their associated bending degrees-of-freedom may all be too soft to avoid numerical conditioning problems. A modeling cure for this condition is to connect many grid points to a new reference grid point with an RBE3 element and then to place the reference grid point in the r-set. The RBE3 element does not affect the flexible modes when applied in this manner.

An example of a structure whose r-set shows poor linear independence is a slightly curved bar modeled using coordinate systems that follow the curve of the bar such that the x-axis is always tangent to the bar. The x DOFs at each end of the bar describe linearly independent DOFs in a mathematical sense. However, numerical truncation produces poor conditioning if the angle between the ends is less than a few degrees. This condition is detected by the automatic diagnostics discussed earlier. This problem can be corrected (or better yet, avoided) by making a careful sketch of all r-set DOFs, including their locations in space and the orientation of their global coordinates. Then apply the three-plane test described earlier.

Using a physical analogy, a good r-set can be chosen by finding one grid point that sustains all possible loadings well if it is tied to ground in an actual hardware test. If there is no such grid point, the ties to ground should be spread over enough grid points to sustain the loads without damaging the structure. An RBE3 element used for this purpose can provide good rigid-body modes without affecting the flexible modes.

## 5.3 Examples

This section provides several rigid-body modes examples showing input and output. These examples are as follows:

**Table 5-3. Unconstrained Beam Model Summary**

Model	Analysis Method	SUPPORT/ No SUPPORT	Redundancy of SUPPORT
bd04bar1	Lanczos	No SUPPORT	–
bd04bar2	Lanczos	SUPPORT	Statically determinate
bd04bar3	Lanczos	SUPPORT	Underdetermined
bd04bar4	Lanczos	SUPPORT	Overdetermined
bd04bar5	SINV	No SUPPORT	–
bd04bar6	SINV	SUPPORT	Statically determinate
bd04bar7	SINV	SUPPORT	Underdetermined
bd04bar8	SINV	SUPPORT	Overdetermined
bd04bkt	Lanczos	No SUPPORT	–

These examples are described in the sections that follow.

### Unconstrained Beam Model

The constraints (SPCs) on the example cantilever beam model from **“Cantilever Beam Model”** are removed to create an unconstrained structure as shown in **Figure 5-3**. A GRDSET entry is added with the z-translation, x-rotation, and y-rotation directions constrained to make the problem two-dimensional. Therefore, there are three DOFs per grid point (x-translation, y-translation, and z-rotation) and three rigid-body modes.



Figure 5-3. Unconstrained Beam Model

Modes are computed using two methods (Lanczos and SINV), with and without a SUPORT entry. The SUPORT entry is used in three ways:

- Statically determinate (grid point 1, components 1, 2, and 6)
- Underdetermined (grid point 1, components 1 and 2)
- Overdetermined (grid point 1, components 1, 2, and 6, plus grid point 11, component 1)

Figure 5-4 shows a portion of the input file for the statically determinate SUPORT and the Lanczos method.

```

$ FILE bd04bar2.dat
$
$ CANTILEVER BEAM MODEL
$ CHAPTER 4, RIGID-BODY MODES
$
SOL 103
TIME 10
CEND
TITLE = CANTILEVER BEAM
SUBTITLE = NORMAL MODES
LABEL = USE SUPORT, STATICALLY DETERMINATE
$
$ OUTPUT REQUEST
DISPLACEMENT = ALL
$
$ SELECT EIGRL ENTRY
METHOD = 10
$
BEGIN BULK
$
$. . . . .2. . . . .3. . . . .4. . . . .5. . . . .6. . . . .7. . . . .8. . . . .9. . . . .10. . . . $
$
$ STATICALLY DETERMINATE SUPORT
$SUPORT G C
SUPORT 1 126
$
$ MAKE 2D MODEL
GRDSET 345
$
$EIGRL SID V1 V2
EIGRL 10 -0.1 50.
$

```

```
... basic model ...
$
ENDDATA
```

**Figure 5-4. Input File for Cantilever Beam Model**

**Table 5-4** lists the computed frequencies. The overdetermined run for the Lanczos method works well; the same run for the SINV method gives an extra zero frequency mode that does not really exist. The overdetermined runs have redundant SUPORTs in the x-direction; therefore, two rigid-body modes are computed in this direction when using the SINV method. In all cases the flexible (greater than 0) frequencies are correct.

**Table 5-4. Frequencies for the Unconstrained Beam Models**

Mode	Lanczos Method Frequencies (Hz)				SINV Method Frequencies (Hz)			
	No SUPORT	Under Det.	Stat. Det. SUPORT	Over- Det. SUPORT	No SUPORT	Under Det.	Stat. Det. SUPORT	Over Det. SUPORT
1	2.15E-7	0.0	0.0	0.0	6.88E-7	0.0	0.0	0.0
2	4.02E-7	0.0	0.0	0.0	6.79E-6	0.0	0.0	0.0
3	8.07E-6	3.07E-5	0.0	0.0	7.12E-6	1.45E-6	0.0	0.0
4	12.82	12.82	12.82	12.82	12.82	12.82	12.82	0.0
5	34.62	34.62	34.62	34.62	34.62	34.62	34.62	12.82
6					66.60	66.60	66.60	34.62
7								66.60

**Figure 5-5** shows the output for the rigid-body modes computed for the SINV method when using no SUPORT. The rigid-body frequencies are denoted by computational zeroes on the order of  $10^{-5}$  Hz or less. Note that the magnitude may be different when the same problem is run on a different computer type.

**Figure 5-6** shows the output for the rigid-body modes computed for the SINV method when using the statically determinate SUPORT. The accuracy of the statically determinate SUPORT DOFs is verified by the computational zeroes for epsilon and the strain energy printed in UIM 3035. Note that the three rigid-body modes have frequencies of 0.0 Hz. The SUPORT entry provides cleaner mode shapes than those shown in **Figure 5-5** as illustrated by the purely x-translation, y-translation, and z-rotation eigenvectors shown in **Figure 5-6**.

MODE GENERALIZED NO. ORDER MASS	EXTRACTION GENERALIZED STIFFNESS	EIGENVALUE GENERALIZED	R E A L E I G E N V A L U E S	
			RADIANS	CYCLES
1	1	1.866121E-11	4.319862E-06	6.875275E-07
1.000000E+00	1.866121E-11			
2	2	1.819082E-09	4.265070E-05	6.788069E-06
1.000000E+00	1.819082E-09			
3	3	2.000299E-09	4.472470E-05	7.118156E-06
1.000000E+00	2.000299E-09			
4	4	6.483918E+03	8.052278E+01	1.281560E+01

## 5. Rigid Body Modes

```

1.000000E+00      6.483918E+03
      5          5          4.732272E+04          2.175379E+02          3.462222E+01
1.000000E+00      4.732272E+04
      6          6          1.751285E+05          4.184836E+02          6.660372E+01
1.000000E+00      1.751285E+05
      EIGENVALUE = 1.866121E-11
      CYCLES = 6.875275E-07          R E A L    E I G E N V E C T O R    N
O .
      1
POINT ID.  TYPE          T1          T2          T3
R1
      R2          R3
      1          G          4.273806E-34 -2.470432E-01  0.0          0.0
0.0
      -1.067134E-01
      2          G          4.273806E-34 -2.790573E-01  0.0          0.0
0.0
      -1.067134E-01
      3          G          4.273806E-34 -3.110713E-01  0.0          0.0
0.0
      -1.067134E-01
      4          G          4.273806E-34 -3.430853E-01  0.0          0.0
0.0
      -1.067134E-01
      5          G          4.273806E-34 -3.750993E-01  0.0          0.0
0.0
      -1.067134E-01
      6          G          4.273806E-34 -4.071134E-01  0.0          0.0
0.0
      -1.067134E-01
      7          G          4.273806E-34 -4.391274E-01  0.0          0.0
0.0
      -1.067134E-01
      8          G          4.273806E-34 -4.711414E-01  0.0          0.0
0.0
      -1.067134E-01
      9          G          4.273806E-34 -5.031555E-01  0.0          0.0
0.0
      -1.067134E-01
      10         G          4.273806E-34 -5.351695E-01  0.0          0.0
0.0
      -1.067134E-01
      11         G          4.273806E-34 -5.671835E-01  0.0          0.0
0.0
      -1.067134E-01
      EIGENVALUE = 1.819082E-09
      CYCLES = 6.788069E-06          R E A L    E I G E N V E C T O R    N
O .
      2
POINT ID.  TYPE          T1          T2          T3
R1
      R2          R3
      1          G          -2.953214E-01 -5.597336E-01  0.0          0.0
0.0
      3.291537E-01
      2          G          -2.953214E-01 -4.609875E-01  0.0          0.0
0.0
      3.291537E-01
      3          G          -2.953214E-01 -3.622414E-01  0.0          0.0
0.0
      3.291537E-01
      4          G          -2.953214E-01 -2.634953E-01  0.0          0.0
0.0
      3.291537E-01
      5          G          -2.953214E-01 -1.647492E-01  0.0          0.0
0.0
      3.291537E-01
      6          G          -2.953214E-01 -6.600305E-02  0.0          0.0
0.0
      3.291537E-01
      7          G          -2.953214E-01  3.274305E-02  0.0          0.0
0.0
      3.291537E-01
      8          G          -2.953214E-01  1.314892E-01  0.0          0.0
0.0
      3.291538E-01
      9          G          -2.953214E-01  2.302354E-01  0.0          0.0
0.0
      3.291538E-01
      10         G          -2.953214E-01  3.289815E-01  0.0          0.0
0.0
      3.291538E-01
      11         G          -2.953214E-01  4.277276E-01  0.0          0.0
0.0
      3.291538E-01
      EIGENVALUE = 2.000299E-09

```

```

CYCLES = 7.118156E-06      R E A L   E I G E N V E C T O R   N
O .
3
POINT ID.  TYPE      T1      T2      T3
R1         R2         R3
1          G      -2.953611E-01  5.596586E-01  0.0      0.0
0.0        -3.291096E-01
2          G      -2.953611E-01  4.609257E-01  0.0      0.0
0.0        -3.291096E-01
3          G      -2.953611E-01  3.621928E-01  0.0      0.0
0.0        -3.291096E-01
4          G      -2.953611E-01  2.634599E-01  0.0      0.0
0.0        -3.291096E-01
5          G      -2.953611E-01  1.647270E-01  0.0      0.0
0.0        -3.291096E-01
6          G      -2.953611E-01  6.599414E-02  0.0      0.0
0.0        -3.291095E-01
7          G      -2.953611E-01 -3.273870E-02  0.0      0.0
0.0        -3.291095E-01
8          G      -2.953611E-01 -1.314715E-01  0.0      0.0
0.0        -3.291095E-01
9          G      -2.953611E-01 -2.302044E-01  0.0      0.0
0.0        -3.291095E-01
10         G      -2.953611E-01 -3.289373E-01  0.0      0.0
0.0        -3.291095E-01
11         G      -2.953611E-01 -4.276701E-01  0.0      0.0
0.0        -3.291095E-01
    
```

Figure 5-5. Unconstrained Beam Modes Without SUPORT (SINV Method)

```

*** USER INFORMATION MESSAGE 3035 FOR DATA BLOCK   KLR
SUPPORT PT.NO.      EPSILON      STRAIN      ENERGY      EPSILONS LARGER
THAN 0.001 ARE FLAGGED WITH ASTERISKS
1          7.7496606E-17      5.5879354E-09
2          7.7496606E-17      0.0000000E+00
3          7.7496606E-17      7.1622708E-11
                                R E A L   E I G E N V A L U E S
                                RADIANS      CYCLES
MODE      EXTRACTION      EIGENVALUE
GENERALIZED      GENERALIZED
ORDER
MASS      STIFFNESS
1          1          0.0      0.0      0.0
1.000000E+00      0.0
2          2          0.0      0.0      0.0
1.000000E+00      0.0
3          3          0.0      0.0      0.0
1.000000E+00      0.0
4          4          6.483918E+03      8.052278E+01      1.281560E+01
1.000000E+00      6.483918E+03
5          5          4.732272E+04      2.175379E+02      3.462222E+01
1.000000E+00      4.732272E+04
6          6          1.751285E+05      4.184836E+02      6.660372E+01
1.000000E+00      1.751285E+05
7          7          4.616299E+05      6.794335E+02      1.081352E+02
1.000000E+00      4.616299E+05
EIGENVALUE = 0.000000E+00
CYCLES = 0.000000E+00
                                R E A L   E I G E N V E C T O R   N
O .
1
POINT ID.  TYPE      T1      T2      T3
    
```

## 5. Rigid Body Modes

```

R1          R2          R3
0.0         1          G          4.176756E-01  0.0          0.0          0.0
0.0         2          G          4.176756E-01  0.0          0.0          0.0
0.0         3          G          4.176756E-01  0.0          0.0          0.0
0.0         4          G          4.176756E-01  0.0          0.0          0.0
0.0         5          G          4.176756E-01  0.0          0.0          0.0
0.0         6          G          4.176756E-01  0.0          0.0          0.0
0.0         7          G          4.176756E-01  0.0          0.0          0.0
0.0         8          G          4.176756E-01  0.0          0.0          0.0
0.0         9          G          4.176756E-01  0.0          0.0          0.0
0.0        10          G          4.176756E-01  0.0          0.0          0.0
0.0        11          G          4.176756E-01  0.0          0.0          0.0
0.0         0.0

```

EIGENVALUE = 0.000000E+00

CYCLES = 0.000000E+00

R E A L E I G E N V E C T O R N

```

O .
2
POINT ID.  TYPE          T1          T2          T3
R1         R2          R3
0.0         1          G          0.0          4.176756E-01  0.0          0.0
0.0         2          G          0.0          4.176756E-01  0.0          0.0
0.0         -3.593777E-15
0.0         3          G          0.0          4.176756E-01  0.0          0.0
0.0         -6.723840E-15
0.0         4          G          0.0          4.176756E-01  0.0          0.0
0.0         -9.575675E-15
0.0         5          G          0.0          4.176756E-01  0.0          0.0
0.0         -1.220145E-14
0.0         6          G          0.0          4.176756E-01  0.0          0.0
0.0         -1.439249E-14
0.0         7          G          0.0          4.176756E-01  0.0          0.0
0.0         -1.596912E-14
0.0         8          G          0.0          4.176756E-01  0.0          0.0
0.0         -1.698929E-14
0.0         9          G          0.0          4.176756E-01  0.0          0.0
0.0         -1.745155E-14
0.0        10          G          0.0          4.176756E-01  0.0          0.0
0.0         -1.770949E-14
0.0        11          G          0.0          4.176756E-01  0.0          0.0
0.0         -1.758923E-14

```

EIGENVALUE = 0.000000E+00

CYCLES = 0.000000E+00

R E A L E I G E N V E C T O R N

```

O .
3
POINT ID.  TYPE          T1          T2          T3
R1         R2          R3
0.0         1          G          0.0          -7.163078E-01  0.0          0.0
0.0         4.775385E-01
0.0         2          G          0.0          -5.730462E-01  0.0          0.0
0.0         4.775385E-01
0.0         3          G          0.0          -4.297847E-01  0.0          0.0

```



0.0	4.775385E-01					
	4	G	0.0	-2.865231E-01	0.0	0.0
0.0	4.775385E-01					
	5	G	0.0	-1.432615E-01	0.0	0.0
0.0	4.775385E-01					
	6	G	0.0	1.508516E-14	0.0	0.0
0.0	4.775385E-01					
	7	G	0.0	1.432615E-01	0.0	0.0
0.0	4.775385E-01					
	8	G	0.0	2.865231E-01	0.0	0.0
0.0	4.775385E-01					
	9	G	0.0	4.297847E-01	0.0	0.0
0.0	4.775385E-01					
	10	G	0.0	5.730463E-01	0.0	0.0
0.0	4.775385E-01					
	11	G	0.0	7.163078E-01	0.0	0.0
0.0	4.775385E-01					

**Figure 5-6. Unconstrained Beam Modes With Statically Determinate SUPORT (SINV Method)**

**Table 5-7** shows the epsilon and strain energy printed in UIM 3035 for the three SUPORT cases (statically determinate, overdetermined, and underdetermined).

It can be seen from this example that aside from clean rigid-body vectors there is no advantage to using a SUPORT entry to compute rigid-body modes.

Statically determinate SUPORT:

```
*** USER INFORMATION MESSAGE 3035 FOR DATA BLOCK KLR
SUPPORT PT.NO.      EPSILON      STRAIN  ENERGY  EPSILONS LARGER THAN
0.001 ARE FLAGGED WITH ASTERISKS
      1      7.7496606E-17      5.5879354E-09
      2      7.7496606E-17      0.0000000E+00
      3      7.7496606E-17      7.1622708E-11
```

Overdetermined SUPORT:

```
*** USER INFORMATION MESSAGE 3035 FOR DATA BLOCK KLR
SUPPORT PT.NO.      EPSILON      STRAIN  ENERGY  EPSILONS LARGER THAN
0.001 ARE FLAGGED WITH ASTERISKS
      1      1.9913979E-01      7.2869660E+06      ****
      2      1.9913979E-01      0.0000000E+00      ****
      3      1.9913979E-01      7.1850081E-11      ****
      4      1.9913979E-01      7.2869660E+06      ****
```

Underdetermined SUPORT:

```
*** USER INFORMATION MESSAGE 4158---STATISTICS FOR SYMMETRIC DECOMPOSITION OF DATA BLOCK
KLL FOLLOW
MAXIMUM RATIO OF MATRIX DIAGONAL TO FACTOR DIAGONAL = 9.1E+13 AT ROW
NUMBER 31
*** USER WARNING MESSAGE 4698. STATISTICS FOR DECOMPOSITION OF MATRIX KLL .
THE FOLLOWING DEGREES OF FREEDOM HAVE FACTOR DIAGONAL RATIOS GREATER THAN
1.00000E+05 OR HAVE NEGATIVE TERMS ON THE FACTOR DIAGONAL.
```

GRID POINT ID	DEGREE OF FREEDOM	MATRIX/FACTOR DIAGONAL RATIO	MATRIX DIAGONAL
11	R3	9.49483E+13	2.84000E+04

```
*** USER INFORMATION MESSAGE 3035 FOR DATA BLOCK KLR
SUPPORT PT.NO.      EPSILON      STRAIN  ENERGY  EPSILONS LARGER
THAN 0.001 ARE FLAGGED WITH ASTERISKS
```

1	7.6288287E-17	5.5879354E-09
2	7.6288287E-17	-7.2759576E-12

Figure 5-7. UIM 3035 Results

## Unconstrained Bracket Example

The constraints (SPCs) on the example bracket model from “**Real Eigenvalue Analysis**” are removed to create an unconstrained model (see “**Examples**” for a description of the model). **Figure 5-8** shows the bracket model. The model is a three-dimensional model and therefore produces six rigid-body modes. The Simcenter Nastran results are shown in **Figure 5-9** for the first eight modes. The Lanczos method is used.

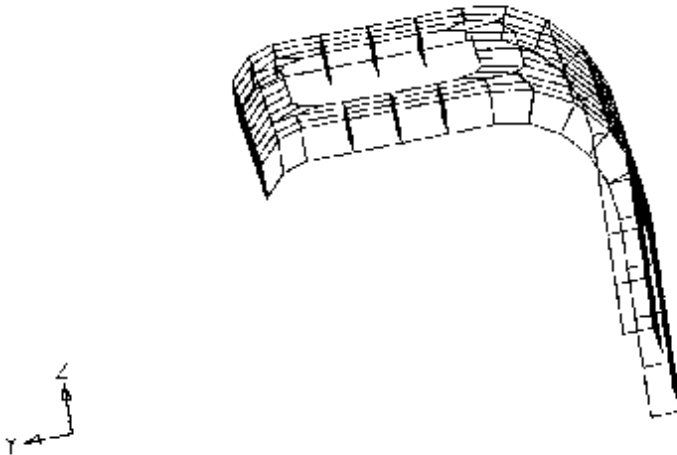


Figure 5-8. Unconstrained Bracket Model

MODE GENERALIZED NO.	EXTRACTION ORDER	EIGENVALUE GENERALIZED	R E A L   E I G E N V A L U E S	
			RADIANS	CYCLES
MASS	STIFFNESS			
1	1	-1.690642E-07	4.111741E-04	6.544039E-05
1.000000E+00	2	-1.690642E-07		
2	2	-9.807991E-09	9.903530E-05	1.576196E-05
1.000000E+00		-9.807991E-09		
3	3	-5.515176E-09	7.426423E-05	1.181952E-05
1.000000E+00		-5.515176E-09		
4	4	-3.390596E-09	5.822883E-05	9.267405E-06
1.000000E+00		-3.390596E-09		
5	5	1.266017E-08	1.125174E-04	1.790770E-05
1.000000E+00		1.266017E-08		
6	6	1.726585E-08	1.313996E-04	2.091289E-05
1.000000E+00		1.726585E-08		
7	7	2.649932E+06	1.627861E+03	2.590821E+02
1.000000E+00		2.649932E+06		

8	8	4.279463E+06	2.068686E+03	3.292416E+02
1.000000E+00	4.279463E+06			

**Figure 5-9. Unconstrained Bracket Frequencies**

The six rigid-body modes have computational zero frequencies on the order of  $10^{-5}$  Hz. Note that the magnitudes of the rigid-body modes may be different when the same problem is run on a different computer type. Also note that the output is sorted by the value of the eigenvalue in ascending order.



# 6. Frequency Response Analysis (SOLs 108 and 111)

## 6.1 Overview of Frequency Response Analysis (SOLs 108 and 111)

Frequency response analysis is used to compute the structural response to a steady-state oscillatory excitation. Examples of oscillatory excitation include rotating machinery, unbalanced tires, and helicopter blades. In frequency response analysis the excitation is explicitly defined in the frequency domain. Excitations can be in the form of applied forces and enforced motions (displacements, velocities, or accelerations).

In Simcenter Nastran, use:

- SOL 108 to perform a direct frequency response analysis. The direct frequency response method solves the coupled equations of motion in terms of forcing frequency.
- SOL 111 to perform a modal frequency response analysis. The modal frequency response method utilizes the mode shapes of the structure to:
  - Reduce the problem size because not all of the modes are typically calculated or retained.
  - Uncouple the equations of motion when either no damping or only modal damping is used.
  - Compute the response at a particular forcing frequency, which is a summation of the individual modal responses.

### Phase Shift

Oscillatory loading is sinusoidal in nature. In its simplest case, this loading is defined as having an amplitude at a specific frequency. The steady-state oscillatory response occurs at the same frequency as the loading. The response may be shifted in time due to damping in the system. The shift in response is called a phase shift because the peak loading and peak response no longer occur at the same time. An example of phase shift is shown in **Figure 6-1**.

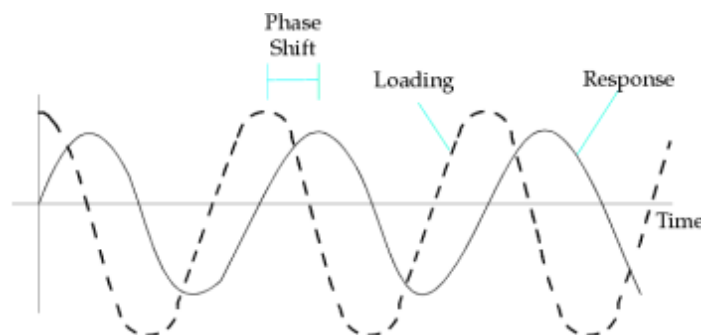


Figure 6-1. Phase Shift

## Complex Numbers

The important results obtained from a frequency response analysis usually include the displacements, velocities, and accelerations of grid points as well as the forces and stresses of elements. The computed responses are complex numbers defined as magnitude and phase (with respect to the applied force) or as real and imaginary components, which are vector components of the response in the real/imaginary plane. These quantities are graphically presented in **Figure 6-2**.

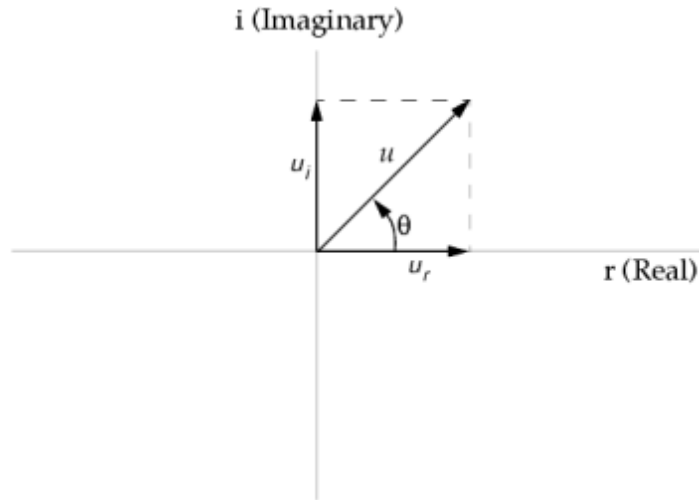


Figure 6-2. Complex Plane

where:

$$\begin{aligned}
 u &= \text{Magnitude} = \sqrt{u_r^2 + u_i^2} \\
 \theta &= \text{Phase angle} = \tan^{-1}(u_i/u_r) \\
 u_r &= \text{Real component} = u \cos \theta \\
 u_i &= \text{Imaginary component} = u \sin \theta
 \end{aligned}$$

## 6.2 Direct Frequency Response Analysis (SOL 108)

In direct frequency response analysis, structural response is computed at discrete excitation frequencies by solving a set of coupled matrix equations using complex algebra. The equation of motion for a damped system subjected to a harmonic excitation is:

$$[M]\{\ddot{x}(t)\} + [B]\{\dot{x}(t)\} + [K]\{x(t)\} = \{P(\omega)\}e^{i\omega t}$$

Equation 6-1.

The load in **Equation 6-1** is introduced as a complex vector, which is convenient for the mathematical solution of the problem. From a physical point of view, the load can be real or imaginary, or both. The same interpretation is used for response quantities.

For harmonic motion (which is the basis of a frequency response analysis), assume a harmonic solution of the form:

$$\{x\} = \{u(\omega)\} e^{i\omega t}$$

Equation 6-2.

where  $\{u(\omega)\}$  is a complex displacement vector. Taking the first and second time derivatives of **Equation 6-2** yields:

$$\begin{aligned}\{\dot{x}\} &= i\omega\{u(\omega)\} e^{i\omega t} \\ \{\ddot{x}\} &= -\omega^2\{u(\omega)\} e^{i\omega t}\end{aligned}$$

Equation 6-3.

Substituting **Equation 6-2** and **Equation 6-3** into **Equation 6-1** yields:

$$-\omega^2 [M]\{u(\omega)\} e^{i\omega t} + i\omega [B]\{u(\omega)\} e^{i\omega t} + [K]\{u(\omega)\} e^{i\omega t} = \{P(\omega)\} e^{i\omega t}$$

Equation 6-4.

After dividing by  $e^{i\omega t}$ , **Equation 6-4** simplifies to:

$$[-\omega^2 M + i\omega B + K]\{u(\omega)\} = \{P(\omega)\}$$

Equation 6-5.

The frequency response is obtained by solving **Equation 6-5** for  $\{u(\omega)\}$  at various forcing frequencies,  $\omega$ .

## Damping in Direct Frequency Response Analysis

Damping simulates the energy dissipation characteristics of a structure. Damping in direct frequency response is represented by the viscous damping matrix  $[B]$  and additions to the stiffness matrix  $[K]$ .

The viscous damping matrix is comprised of several matrices:

$$[B] = [B^1] + [B^2]$$

Equation 6-6.

where:

- $[B^1]$  = Viscous damping from elements (CVISC, CDAMPi) and B2GG
- $[B^2]$  = Viscous damping from B2PP direct input matrix and transfer functions

PARAM,G and values from GE fields on material, property, and element entries form the complex stiffness matrix:

$$[K] = (1 + iG)[K] + i \sum G_E [K_E]$$

Equation 6-7.

where:

- $[K]$  = Global stiffness matrix
- $G$  = Overall structural damping coefficient (PARAM,G)
- $[K_E]$  = Element stiffness matrix
- $G_E$  = Element structural damping coefficient (GE on material, property, or element entries)

Because frequency response analysis allows complex numbers, it is not necessary to convert structural damping to viscous damping. Both structural and viscous damping can be used in the same analysis. Complex stiffness is automatically computed and added into the equation of motion if either PARAM,G or GE on material, property, or element entries are specified.

### 6.3 Modal Frequency Response Analysis (SOL 111)

Modal frequency response analysis is an alternate method to compute frequency response. This method uses the mode shapes of the structure to uncouple the equations of motion (when no damping or only modal damping is used) and, depending on the number of modes computed and retained, reduce the problem size. Both of these factors tend to make modal frequency response analysis computationally more efficient than direct frequency response analysis. Because modal frequency response analysis uses the mode shapes of a structure, modal frequency response analysis is a natural extension of normal modes analysis.

To understand modal frequency response analysis, begin by ignoring viscous and structural damping in the following equation.



$$[M]\{\ddot{x}(t)\} + [B]\{\dot{x}(t)\} + [K]\{x(t)\} = \{P(\omega)\}e^{i\omega t}$$

Equation 6-8.

Thus, **Equation 6-8** simplifies to:

$$[M]\{\ddot{x}(t)\} + [K]\{x(t)\} = \{P(\omega)\}e^{i\omega t}$$

Equation 6-9.

Once again, assume a harmonic solution of the form:

$$\{x\} = \{u(\omega)\}e^{i\omega t}$$

Equation 6-10.

where  $\{u(\omega)\}$  is a complex displacement vector.

In **Equation 6-9**, ignore the forcing term and substitute **Equation 6-10** and the second time derivative of **Equation 6-10**. Solving the resulting equation yields the natural frequencies and mode shapes for the system.

Now use the mode shapes,  $[\phi]$ , to transform the response from physical coordinates,  $\{u(\omega)\}$ , to modal coordinates,  $\{\xi(\omega)\}$ :

$$\{x\} = [\phi]\{\xi(\omega)\}e^{i\omega t}$$

Equation 6-11.

**Equation 6-11** represents an equality if all modes are calculated and used. When a subset of all modes are calculated or used, **Equation 6-11** represents an approximation.

Substituting **Equation 6-11** and the second time derivative of **Equation 6-11** into **Equation 6-9** and simplifying yields the equation of motion in terms of the modal coordinates:

$$-\omega^2[M][\phi]\{\xi(\omega)\} + [K][\phi]\{\xi(\omega)\} = \{P(\omega)\}$$

Equation 6-12.

The equations forming **Equation 6-12** are coupled. To uncouple the equations, premultiply **Equation 6-12** by the transpose of the eigenvector matrix,  $[\phi]^T$ :

$$-\omega^2 [\phi]^T [M] [\phi] \{\xi(\omega)\} + [\phi]^T [K] [\phi] \{\xi(\omega)\} = [\phi]^T \{P(\omega)\}$$

**Equation 6-13.**

Doing so, diagonalizes the mass and stiffness matrices. The transformed mass and stiffness matrices and forcing vector are referred to as:

$$\begin{aligned} [\phi]^T [M] [\phi] &= \text{Modal (generalized) mass matrix} \\ [\phi]^T [K] [\phi] &= \text{Modal (generalized) stiffness matrix} \\ [\phi]^T \{P\} &= \text{Modal force vector} \end{aligned}$$

In the uncoupled form, the equation of motion can be written as a set of single degree-of-freedom systems:

$$-\omega^2 m_i \xi_i(\omega) + k_i \xi_i(\omega) = p_i(\omega)$$

**Equation 6-14.**

where:

$$\begin{aligned} m_i &= \text{Modal mass for the } i^{\text{th}} \text{ mode} \\ k_i &= \text{Modal stiffness for the } i^{\text{th}} \text{ mode} \\ p_i &= \text{Modal force for the } i^{\text{th}} \text{ mode} \end{aligned}$$

From **Equation 6-14**, the individual modal responses,  $\xi_i(\omega)$ , are calculated for a specific forcing frequency,  $\omega$ . The physical response at the specified forcing frequency is then recovered from:

$$\{x\} = [\phi] \{\xi(\omega)\} e^{i\omega t}$$

**Equation 6-15.**

These responses are in complex form (magnitude/phase or real/imaginary) and are used to recover additional output quantities requested in the case control section of the input file.

## Damping in Modal Frequency Response Analysis

If viscous damping is present, the orthogonality property of the modes does not, in general, diagonalize the generalized viscous damping matrix:

$$[\phi]^T [B] [\phi] \neq \text{diagonal}$$

Equation 6-16.

Similarly, if structural damping is present, the orthogonality property does not, in general, diagonalize the generalized stiffness matrix:

$$[\phi]^T [(1 + iG)[K] + i \sum G_E [K_E]] [\phi] \neq \text{diagonal}$$

Equation 6-17.

Consequently, in the presence of viscous damping or structural damping, the equations of motion are typically coupled. For such cases, the modal frequency response method:

1. Solves the undamped system for the natural frequencies and corresponding mode shapes.
2. Premultiplies the equation of motion of the damped system by the transpose of the matrix of mode shapes,  $[\phi]^T$ , to yield:

$$[-\omega^2 [\phi]^T [M] [\phi] + i\omega [\phi]^T [B] [\phi] + [\phi]^T [K] [\phi]] \{\xi(\omega)\} = [\phi]^T \{P(\omega)\}$$

Equation 6-18.

where **Equation 6-18** is expressed in modal coordinates and is generally coupled.

3. Solves **Equation 6-18** with a direct frequency response solution.

Because only a subset of all the modes are typically required for an accurate representation of the frequency response, the size of **Equation 6-18** is typically much smaller than the size of **Equation 6-19** in a direct frequency response solution. As a result, modal frequency response analysis typically requires less computational effort to produce an accurate solution than direct frequency response analysis.

$$[-\omega^2 M + i\omega B + K] \{u(\omega)\} = \{P(\omega)\}$$

Equation 6-19.

For more information, see [Direct Frequency Response Analysis \(SOL 108\)](#).

Another way to include damping in a modal frequency response analysis is to use modal damping. Modal damping is either viscous or structural damping that is applied to each mode separately so that in the absence of other sources of damping the equations of motion remain uncoupled. If other sources of damping are present, modal damping is added into the equations of motion along with the other damping.

You request modal damping by including a SDAMPING case control command in the input file. You can use the KDAMP parameter to specify whether the modal damping is applied as modal viscous damping or modal structural damping. Modal viscous damping is the default.

When other sources of damping are absent and modal damping is applied as modal viscous damping, the equations of motion have the following form:

$$-\omega^2 m_i \xi_i(\omega) + i\omega b_i \xi_i(\omega) + k_i \xi_i(\omega) = p_i(\omega)$$

Equation 6-20.

where  $b_i$  is the viscous damping at the natural frequency of the  $i^{\text{th}}$  mode. Each of the modal responses is computed from:

$$\xi_i(\omega) = \frac{p_i(\omega)}{-m_i \omega^2 + i b_i \omega + k_i}$$

Equation 6-21.

The SDAMPING case control command should reference the TID of a TABDMP1 bulk entry. On the TABDMP1 entry, you define frequency/damping data. The software uses the frequency/damping data to calculate the damping value to apply at each modal frequency.

The TYPE field on the TABDMP1 entry is used to specify whether the damping values on the TABDMP1 entry represent structural (hysteretic) damping, damping factor (fraction of critical damping), or quality factor (amplification factor). Structural damping is the default. To find the damping value at a specific modal frequency, the software linearly interpolates or linearly extrapolates the values provided on the TABDMP1 entry.

For example, suppose damping factor versus frequency data is entered on a TABDMP1 entry as follows:

1	2	3	4	5	6	7	8	9	10
\$TABDMP1	TID	TYPE							
\$	f1	g1	f2	g2	f3	g3	etc.		

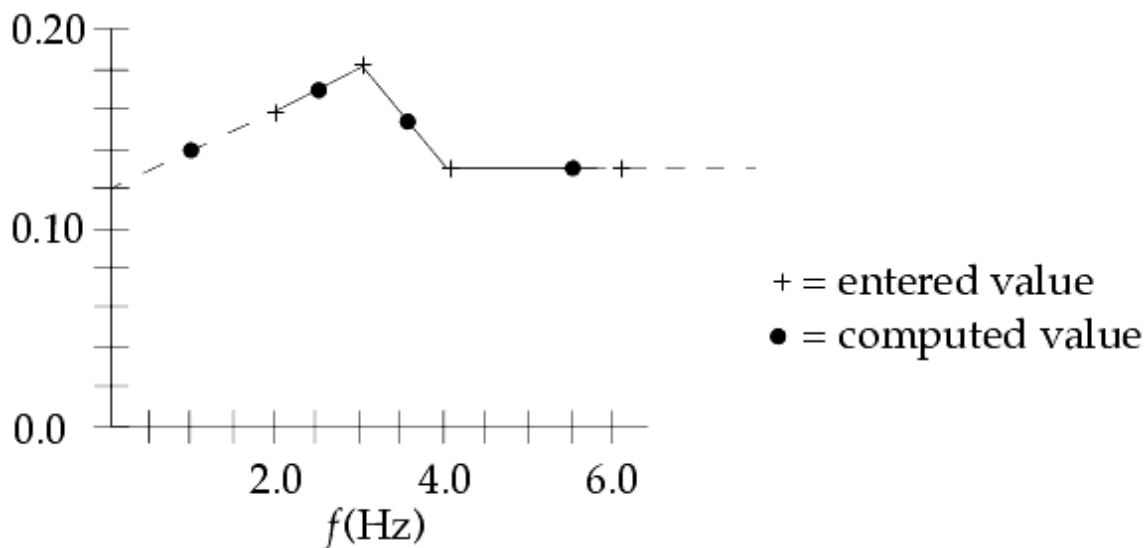
1	2	3	4	5	6	7	8	9	10
TABDMP1	10	CRIT							
	2.0	0.16	3.0	0.18	4.0	0.13	6.0	0.13	
	ENDT								

Also suppose the structure being analyzed has modes at 1.0, 2.5, 3.6, and 5.5 Hz. The software will interpolate and extrapolate the TABDMP1 data to produce estimates of damping factor at the modal frequencies as indicated in **Table 6-7** and **Figure 6-3**.

**Table 6-7. Results from Interpolating/Extrapolating TABDMP1 Data**

Entered		Computed	
$f$	$\zeta$	$f$	$\zeta$
2.0	0.16	1.0	0.14 <sup>(1)</sup>
3.0	0.18	2.5	0.17
4.0	0.13	3.6	0.15
6.0	0.13	5.5	0.13

<sup>(1)</sup> The value is extrapolated from data at 2.0 Hz and 3.0 Hz.



**Figure 6-3. Interpolating/Extrapolating TABDMP1 Data**

By default, the damping values specified on a TABDMP1 entry are used to calculate modal viscous damping values  $b_i$  in **Equation 6-20**. However, if you include PARAM,KDAMP,-1 in the bulk data section, the software will use the TABDMP1 damping values to calculate modal structural damping and the uncoupled equations of motion assume the following form:

$$-\omega^2 m_i \xi_i(\omega) + (1 + iG_i)k_i \xi_i(\omega) = p_i(\omega)$$

where  $G_i$  is the structural damping at the natural frequency of the  $i^{\text{th}}$  mode. Each of the modal responses is computed from:

$$\xi_i(\omega) = \frac{p_i(\omega)}{-m_i \omega^2 + (1 + iG_i)k_i}$$

**Table 6-8** lists the equations the software uses to convert the interpolated or extrapolated damping values from the TABDMP1 entry to modal viscous damping,  $b_i$ , or modal structural damping,  $G_i$ .

**Table 6-8. Conversion of Interpolated/Extrapolated Damping Values to Modal Damping Values**

Type of damping in TABDMP1 entry	Modal viscous damping PARAM,KDAMP,1 (Default)	Modal structural damping PARAM,KDAMP,-1
Structural damping <sup>(1)</sup>	$b_i = g(f_i) b_{cr} / 2$	$G_i = g(f_i)$
Damping factor <sup>(2)</sup>	$b_i = g(f_i) b_{cr}$	$G_i = 2 g(f_i)$
Quality <sup>(3)</sup>	$b_i = b_{cr} / (2 g(f_i))$	$G_i = 1 / g(f_i)$

where:  
 $b_i$  = Modal viscous damping for the  $i^{\text{th}}$  mode  
 $G_i$  = Modal structural damping for the  $i^{\text{th}}$  mode  
 $f_i$  = Undamped natural frequency of the  $i^{\text{th}}$  mode  
 $g(f_i)$  = Interpolated or extrapolated damping value at  $f_i$  from damping/frequency data on TABDMP1 entry  
 $b_{cr} = 2 m_i \omega_i$  = Critical damping for the  $i^{\text{th}}$  mode with  $\omega_i = 2\pi f_i$   
 $m_i$  = Modal mass for the  $i^{\text{th}}$  mode  
<sup>(1)</sup> Also referred to as hysteretic damping  
<sup>(2)</sup> Also referred to as fraction of critical damping  
<sup>(3)</sup> Also referred to as amplification factor

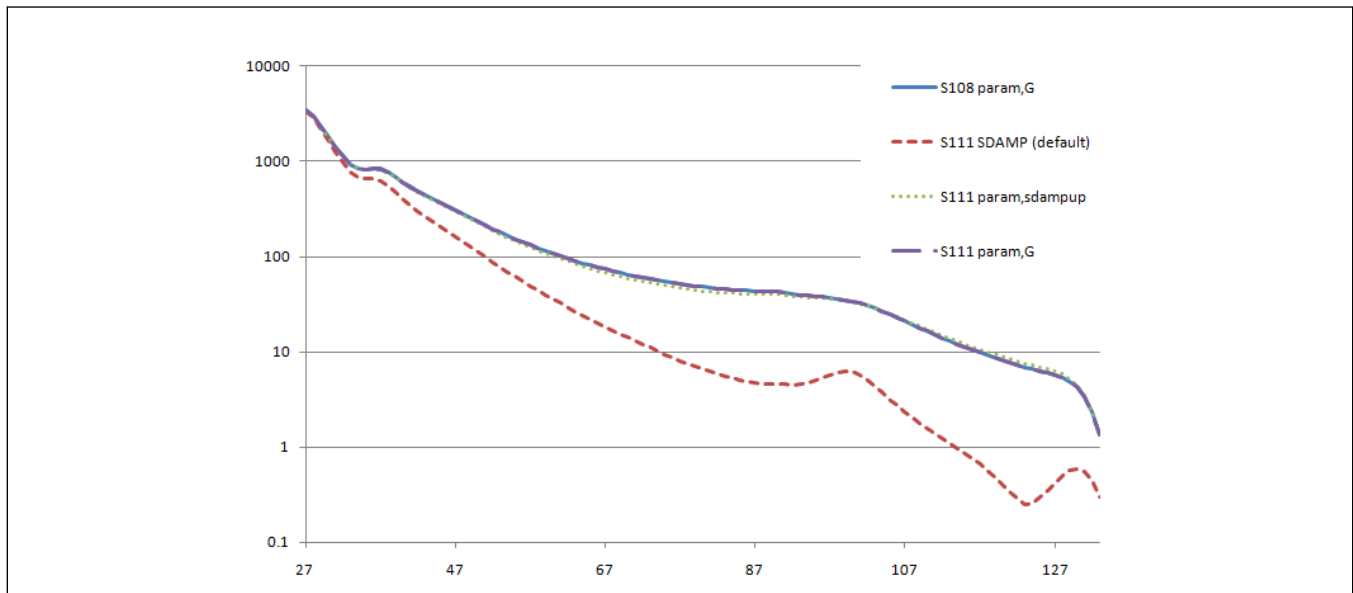
By default, Simcenter Nastran computes modal damping only once from the original modal solution. This can lead to inaccurate results when the model includes elements with frequency-dependent stiffness properties. For example, frequency-dependent stiffness properties can be defined for certain elements using the PBUSHT and PELAST property entries.

You can improve the accuracy of the results for such a model by specifying PARAM,SDAMPUP,YES in the bulk data section of the input file. When PARAM,SDAMPUP,YES is specified, the software updates the frequency-dependent modal stiffness and recomputes the modal frequency and eigenvector while remaining in modal coordinates (h-set). The updated frequency is used to compute the modal damping for each frequency response calculation. The penalty for improving the accuracy of the results is that the

runtime may increase, particularly if the number of modes is large. You can specify PARAM,SDAMPUP,YES when either modal viscous damping or modal structural damping are used.

Repeated eigenvalues and rigid body modes are not supported when specifying the SDAMPUP parameter. Rigid body modes may be removed either by adjusting F1 on the EIGRL bulk entry or by setting PARAM,LFREQ. If repeated eigenvalues are present, the SDAMPUP parameter is ignored.

**Figure 6-4** depicts the accuracy improvement that may occur when you specify the SDAMPUP parameter. The results from four different combinations of solution type and damping specification are presented. The model producing these results contains approximately 3000 degree-of-freedom and one scalar spring element with frequency-dependent stiffness. The frequency response up to 133 Hz is computed using modes with frequencies up to 350 Hz.



**Figure 6-4. Frequency Response Predicted Using Various Combinations of Solutions and Damping**

The four solution and damping combinations include:

- Direct frequency response with structural damping specified using PARAM,G. (The “S108 param,G” curve in **Figure 6-4.**)
- Modal frequency response with modal damping specified using the SDAMPING case control command only. (The “S111 SDAMP (default)” curve in **Figure 6-4.**)
- Modal frequency response with modal damping specified using the SDAMPING case control command and PARAM,SDAMPUP,YES. (The “S111 param,sdampup” curve in **Figure 6-4.**)
- Modal frequency response with structural damping specified using PARAM,G. (The “S111 param,G” curve in **Figure 6-4.**) This solution is a direct solution to the reduced modal representation.

The results from all the solution type and damping specification combinations are essentially identical except for those from the modal solution where PARAM,SDAMPUP,YES is not specified. Once PARAM,SDAMPUP,YES is specified, the modal damping solution is consistent with the other three solutions. By adding the PARAM,SDAMPUP,YES specification to the modal solution, the runtime increases from 0:19 to 0:36. By comparison, the runtime for the direct frequency response solution is 0:58.

## Mode Truncation in Modal Frequency Response Analysis

It is possible that not all of the computed modes are required in the frequency response solution. At an absolute minimum, you should include all the modes whose frequencies lie within the range of forcing frequencies. For example, if the forcing frequency range lies between 200 and 2000 Hz, all modes whose frequencies are in this range should be retained in the modal frequency response analysis. For better accuracy, include all modes up to at least two to three times the highest forcing frequency in the modal frequency response analysis. Residual vectors can also be requested to improve the response by adding the missing static flexibility associated with these higher modes. See [Residual Vectors for Modal Response Solutions](#) for more information.

By default, Simcenter Nastran uses all of the computed modes in a modal frequency response analysis. However, you can control the modes used in the modal frequency response analysis by:

- Specifying a frequency range on the eigenvalue entry (EIGR or EIGRL).
- Using the LFREQ, HFREQ, and LMODES parameters.

PARAM,LFREQ gives the lower limit on the frequency range of retained modes.

PARAM,HFREQ gives the upper limit on the frequency range of retained modes.

PARAM,LMODES gives the number of the lowest modes to be retained.

- Including a MODSEL case control command to directly select or exclude individual mode numbers from the modal frequency response calculation. The MODSEL case control command can be used for fluid modes as well as structural modes, and can be used in a restart run. The restart capability allows you to select modes from a SOL 103 normal modes analysis to use in a consecutive SOL 111 modal frequency response analysis.

## Dynamic Data Recovery in Modal Frequency Response Analysis

In modal frequency response analysis, two options are available for recovering displacements and stresses: the mode displacement method and the matrix method. Both methods give the same answers, although with differences in the computational effort required.

The mode displacement method computes the total physical displacements for each excitation frequency from the modal displacements, and then computes element stresses from the total physical displacements. The number of operations is proportional to the number of excitation frequencies.



The matrix method computes displacements per mode and element stresses per mode, and then computes physical displacements and element stresses as the summation of modal displacements and element stresses. Computational effort is proportional to the number of modes.

Because the number of modes is usually much less than the number of excitation frequencies, the matrix method is usually more efficient and is the default. The mode displacement method can be selected by including PARAM,DDRMM,-1 in the bulk data section. The mode displacement method is required when “frequency-frozen” structural plots are requested (see **“Plotted Output”**).

## 6.4 Modal Versus Direct Frequency Response

Some general guidelines can be used when selecting modal frequency response analysis versus direct frequency response analysis. These guidelines are summarized in **Table 6-9**.

**Table 6-9. Recommended use of Modal Versus Direct Frequency Response**

	Modal	Direct
Small Model		X
Large Model	X	
Few Excitation Frequencies		X
Many Excitation Frequencies	X	
High Frequency Excitation		X
Nonmodal Damping		X
Higher Accuracy		X

In general, larger models may be solved more efficiently with modal frequency response because the numerical solution is typically the solution to a smaller system of uncoupled equations. The modal method is particularly advantageous if the natural frequencies and mode shapes are computed during a previous stage of the analysis. In that case, you simply perform a restart (see **“Restarts In Dynamic Analysis”**). Using the modal approach is very efficient, even for very large numbers of excitation frequencies. On the other hand, the major portion of the effort in a modal frequency response analysis is the calculation of the modes. For large systems with a large number of modes, this operation can be as computationally intensive as a direct solution. This is especially true for high-frequency excitation. To capture high frequency response in a modal solution, less accurate, high-frequency modes must be computed. For small models with a few excitation frequencies, the direct method may be the most efficient because it solves the equations without first computing the modes. The direct method is typically more accurate than the modal method because of mode truncation.

## 6.5 Frequency-Dependent Elements and Materials

The following elements and materials support frequency dependent properties for SOLs 108 and 111:

- CBUSH element referencing the PBUSH and PBUSHT property entries.

- CELAS1 and CELAS3 elements referencing the PELAS and PELAST property entries.
- CDAMP1 and CDAMP3 elements referencing the PDAMP and PDAMPT property entries.
- You can use the MAT1 and MAT1F entries to define Young's modulus (E), shear modulus (G), Poisson's ratio (NU), mass density (RHO), and structural damping (GE) as frequency dependent. The MAT1F entry is supported by the following elements:
  - Solid elements CTETRA, CHEXA, CPENTA, and CPYRAM using the PSOLID property.
  - Shell elements CTRIA3, CQUAD4, CTRIA6, CQUAD8, CTRIAR, and CQUADR using the PSHELL property.
  - Bar element CBAR with the PBAR property.
  - Beam element CBEAM with the PBEAM property.

## 6.6 Frequency-Dependent Excitation Definition

An important aspect of a frequency response analysis is the definition of the loading function. In a frequency response analysis, the load is defined as a function of frequency. Loads are defined in the same manner regardless of whether the direct or modal method is used.

The following bulk entries are typically used when frequency-dependent dynamic loads are defined:

Bulk Entry	Description	
RLOAD1	Defines frequency-dependent dynamic load by real and imaginary parts.	
RLOAD2	Defines frequency-dependent dynamic load by magnitude and phase.	
RLOADEX	Defines frequency-dependent dynamic load from an external HDF5 file.	
DELAY	Defines time delay of dynamic load.	
DPHASE	Defines phase lead of dynamic load.	
TABLEDi	Defines frequency dependence of dynamic load via tabular data.	
DAREA	Defines spatial distribution of dynamic load.	
FORCEi		
MOMENTi		
PLOAD		
PLOAD1		
SELOAD		
SPCD		
DLOAD		Combines dynamic loads.

To fully specify a dynamic load, you must:

- Define the location and direction that the load is applied to the structure. Because this characteristic locates the loading in space, it is called the spatial distribution of the dynamic loading.

- Define the frequency variation in the load. This characteristic of a dynamic load differentiates it from a static load. The frequency variation is called the temporal distribution of the load.

Simcenter Nastran makes it possible for you to apply many complicated and temporally similar loadings with a minimum of input by using Table IDs and Set IDs. You can also easily combine simple loadings to create complicated load distributions that vary in position as well as frequency.

The remainder of this section describes the bulk entries for frequency-dependent excitation. For more complete bulk entry descriptions, see the *Simcenter Nastran Quick Reference Guide*.

## Frequency-Dependent Loads – RLOAD1 Entry

The RLOAD1 entry defines a dynamic loading of the form:

$$\{P(f)\} = \{A[C(f) + iD(f)]e^{i\{\theta - 2\pi f\tau\}}\}$$

$C(f)$  and  $D(f)$  are defined in tabular format on TABLEDi bulk entries if frequency-dependent, or directly on the RLOAD1 entry as real input if constant. For tabular input, you need not explicitly define the dynamic load at every excitation frequency. Only the values at frequencies that describe the character of the loading are required. Simcenter Nastran linearly interpolates for intermediate values.

Define time delay,  $\tau$ , and phase angle,  $\theta$ , for each degree-of-freedom referenced through the EXCITEID field. If the values vary across degree-of-freedom, reference DELAY and DPHASE bulk entries to define time delay,  $\tau$ , and phase angle,  $\theta$ , respectively. If the values for time delay,  $\tau$ , and/or phase angle,  $\theta$ , are uniform across all degree-of-freedom, enter the value(s) directly on the RLOAD1 entry as real input.

1	2	3	4	5	6	7	8	9	10
RLOAD1	SID	EXCITEID	DELAY	DPHASE	TC	TD	TYPE		

Field	Contents
SID	Load set identification number.
EXCITEID	Identification number of a DAREA, FORCEi, MOMENTi, PLOAD, PLOAD1, SELOAD, or SPCD entry that lists each degree-of-freedom to apply the excitation and the corresponding scale factor, A, for the excitation.
DELAY	Time delay, $\tau$ .
DPHASE	Phase angle, $\theta$ .
TC	$C(f)$ definition.
TD	$D(f)$ definition.
TYPE	Defines the type of the dynamic excitation.

Note that  $f$  is the frequency in cycles per unit time.

## Frequency-Dependent Loads – RLOAD2 Entry

The RLOAD2 entry is a variation of the RLOAD1 entry used for defining a frequency-dependent dynamic load. Whereas the RLOAD1 entry defines the real and imaginary parts of the complex load, the RLOAD2 entry defines the magnitude and phase.

The RLOAD2 entry defines a dynamic loading of the form:

$$\{P(f)\} = \{AB(f)e^{i\{\phi(f) + \theta - 2\pi f\tau\}}\}$$

Equation 6-22.

The RLOAD2 definition may be related to the RLOAD1 definition by:

$$\underbrace{C(f) + iD(f)}_{\text{RLOAD1 Definition}} = \underbrace{B(f)e^{i\phi(f)}}_{\text{RLOAD2 Definition}}$$

Equation 6-23.

$B(f)$  and  $\phi(f)$  are defined in tabular format on TABLEDi bulk entries if frequency-dependent, or directly on the RLOAD2 entry as real input if constant. For tabular input, you need not explicitly define the dynamic load at every excitation frequency. Only the values at frequencies that describe the character of the loading are required. Simcenter Nastran interpolates for intermediate values.

Define time delay,  $\tau$ , and phase angle,  $\theta$ , for each degree-of-freedom referenced through the EXCITEID field. If the values vary across degree-of-freedom, reference DELAY and DPHASE bulk entries to define time delay,  $\tau$ , and phase angle,  $\theta$ , respectively. If the values for time delay,  $\tau$ , and/or phase angle,  $\theta$ , are uniform across all degree-of-freedom, enter the value(s) directly on the RLOAD2 entry as real input.

1	2	3	4	5	6	7	8	9	10
RLOAD2	SID	EXCITEID	DELAY	DPHASE	TB	TP	TYPE		

Field	Contents
SID	Load set identification number.
EXCITEID	Identification number of a DAREA, FORCEi, MOMENTi, PLOAD, PLOAD1, SELOAD, or SPCD entry that lists each degree-of-freedom to apply the excitation and the corresponding scale factor, A, for the excitation.
DELAY	Time delay, $\tau$ .
DPHASE	Phase angle, $\theta$ .

Field	Contents
TB	$B(f)$ definition.
TP	$\varphi(f)$ definition.
TYPE	Defines the type of the dynamic excitation.

Note that  $f$  is the frequency in cycles per unit time.

## Time Delay – DELAY Entry

The DELAY entry defines the time delay,  $\tau$ , for specified degree-of-freedom.

1	2	3	4	5	6	7	8	9	10
DELAY	SID	P1	C1	T1	P2	C2	T2		

Field	Contents
SID	Identification number.
Pi	Grid, extra, or scalar point ID.
Ci	Component number.
Ti	Time delay, $\tau$ .

Any number of DELAY entries may be used; all those with the same SID are combined.

## Phase Lead – DPHASE Entry

The DPHASE entry defines the phase lead,  $\theta$ , for specified degree-of-freedom.

1	2	3	4	5	6	7	8	9	10
DPHASE	SID	P1	C1	TH1	P2	C2	TH2		

Field	Contents
SID	Identification number.
Pi	Grid, extra, or scalar point ID.
Ci	Component number.
THi	Phase angle, $\theta$ , in degrees.

Any number of DPHASE entries may be used; all those with the same SID are combined.

## Dynamic Load Tabular Function – TABLEDi Entries

TABLEDi entries ( $i = 1$  through 4) define a tabular function for use in generating frequency-dependent dynamic loads. The form of each TABLEDi entry varies slightly, depending on the value of  $i$ , as does the algorithm for  $y(x)$ . The values of  $x$  need not be evenly spaced

For TABLED1, TABLED2, and TABLED3 entries, the software uses linear interpolation to look up the value of  $y$  when  $x$  is inside the range of the tabular data. For TABLED1 entries, you can optionally specify that the software perform the interpolation in Cartesian, semi-log, or log-log space.

For values of  $x$  that lie outside the range of the tabular data, you can specify how the software looks up values from the following options:

- Linearly extrapolate the two starting or two ending data points to obtain the value for  $y$ , as shown in [Figure 6-5](#).
- Use the value at the starting or ending data point as the value for  $y$ .
- Specify the value for  $y$ .

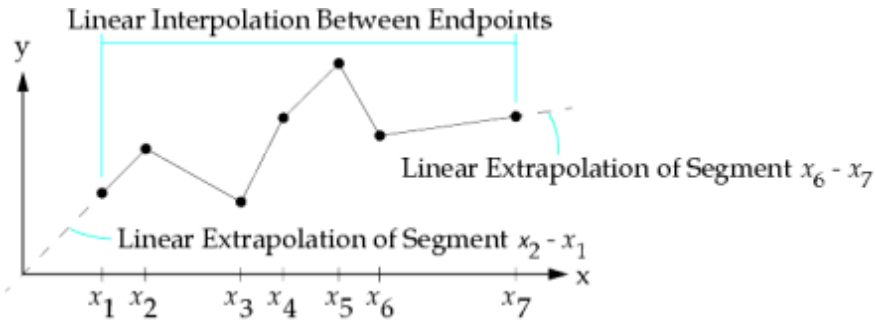


Figure 6-5. Interpolation and Extrapolation for TABLED1, TABLED2, and TABLED3 Entries

The TABLED1 entry has the following format:

1	2	3	4	5	6	7	8	9	10
TABLED1	TID	XAXIS	YAXIS	EXTRAP	LOEXT	HIEXT			
	x1	y1	x2	y2	x3	y3	-etc.-	"ENDT"	

Field	Contents
TID	Table identification number.
XAXIS, YAXIS	Specifies whether the interpolation is linear in Cartesian, semi-log, or log-log space.

For interpolation in Cartesian space, specify XAXIS = LINEAR and YAXIS = LINEAR.

Field	Contents
	For interpolation in semi-log space, specify XAXIS = LINEAR and YAXIS = LOG, or XAXIS = LOG and YAXIS = LINEAR.
	For interpolation in log-log space, specify XAXIS = LOG and YAXIS = LOG.
EXTRAP	Extrapolation option.  If EXTRAP = 0, obtain table lookup values for $x < x_{\min}$ and $x > x_{\max}$ by applying the interpolation formula to the two starting and two ending data points.  If EXTRAP = 1, use the value of the table field at $x_{\min}$ for the table lookup when $x < x_{\min}$ and use the value of the table field at $x_{\max}$ for the table lookup when $x > x_{\max}$ .  If EXTRAP = 2, use LOEXT for the table lookup when $x < x_{\min}$ and use HIEXT for the table lookup when $x > x_{\max}$ .
LOEXT	For EXTRAP = 2, table lookup value when $x < x_{\min}$ .
HIEXT	For EXTRAP = 2, table lookup value when $x > x_{\max}$ .
xi, yi	Tabular values.
ENDT	Ends the table input.

The TABLED1 entry uses the algorithm:

$$y = y_T(x)$$

Equation 6-24.

The linear interpolation and extrapolation algorithms are as follows:

XAXIS	YAXIS	$y_T(x)$
LINEAR	LINEAR	$\frac{x_j - x}{x_j - x_i} y_i + \frac{x - x_i}{x_j - x_i} y_j$
LOG	LINEAR	$\frac{\ln(x_j/x)}{\ln(x_j/x_i)} y_i + \frac{\ln(x/x_i)}{\ln(x_j/x_i)} y_j$

XAXIS	YAXIS	$y_T(x)$
LINEAR	LOG	$\exp \left[ \frac{x_j - x}{x_j - x_i} \ln y_i + \frac{x - x_i}{x_j - x_i} \ln y_j \right]$
LOG	LOG	$\exp \left[ \frac{\ln(x_j/x)}{\ln(x_j/x_i)} \ln y_i + \frac{\ln(x/x_i)}{\ln(x_j/x_i)} \ln y_j \right]$

The TABLED2 entry has the following format:

1	2	3	4	5	6	7	8	9	10
TABLED2	TID	X1	EXTRAP	LOEXT	HIEXT				
	x1	y1	x2	y2	x3	y3	-etc.-	"ENDT"	

Field	Contents
TID	Table identification number.
X1	Table parameter.
EXTRAP	Extrapolation option.  If EXTRAP = 0, obtain table lookup values for $(x - X1) < x_{\min}$ and $(x - X1) > x_{\max}$ by linearly extrapolating the two starting and two ending data points.  If EXTRAP = 1, use the value of the table field at $x_{\min}$ for the table lookup when $(x - X1) < x_{\min}$ and use the value of the table field at $x_{\max}$ for the table lookup when $(x - X1) > x_{\max}$ .  If EXTRAP = 2, use LOEXT for the table lookup when $(x - X1) < x_{\min}$ and use HIEXT for the table lookup when $(x - X1) > x_{\max}$ .
LOEXT	For EXTRAP = 2, table lookup value when $(x - X1) < x_{\min}$ .
HIEXT	For EXTRAP = 2, table lookup value when $(x - X1) > x_{\max}$ .
$x_i, y_i$	Tabular values.
ENDT	Ends the table input.

The TABLED2 entry uses the algorithm:

$$y = y_T(x - X1)$$

Equation 6-25.



The TABLED3 entry has the following format:

1	2	3	4	5	6	7	8	9	10
TABLED3	TID	X1	X2	EXTRAP	LOEXT	HIEXT			
	x1	y1	x2	y2	x3	y3	-etc.-	"ENDT"	

Field	Contents
TID	Table identification number.
X1, X2	Table parameters.
EXTRAP	Extrapolation option.  If EXTRAP = 0, obtain table lookup values for $(x - X1) / X2 < x_{\min}$ and $(x - X1) / X2 > x_{\max}$ by linearly extrapolating the two starting and two ending data points.  If EXTRAP = 1, use the value of the table field at $x_{\min}$ for the table lookup when $(x - X1) / X2 < x_{\min}$ and use the value of the table field at $x_{\max}$ for the table lookup when $(x - X1) / X2 > x_{\max}$ .  If EXTRAP = 2, use LOEXT for the table lookup when $(x - X1) / X2 < x_{\min}$ and use HIEXT for the table lookup when $(x - X1) / X2 > x_{\max}$ .
LOEXT	For EXTRAP = 2, table lookup value when $(x - X1) / X2 < x_{\min}$ .
HIEXT	For EXTRAP = 2, table lookup value when $(x - X1) / X2 > x_{\max}$ .
xi, yi	Tabular values.
ENDT	Ends the table input.

The TABLED3 entry uses the algorithm:

$$y = y_T \left( \frac{x - X1}{X2} \right)$$

Equation 6-26.

The TABLED4 entry has the following format:

1	2	3	4	5	6	7	8	9	10
TABLED4	TID	X1	X2	X3	X4				
	A0	A1	A2	A3	A4	A5	-etc.-	"ENDT"	

Field	Contents
TID	Table identification number.
Xi	Table parameters. ( $X2 \neq 0.0$ ; $X3 < X4$ )
Ai	Coefficients.
ENDT	Ends the table input.

The TABLED4 entry uses the algorithm:

$$y = \sum_{i=0}^N Ai \left( \frac{x - X1}{X2} \right)^i$$

Equation 6-27.

N is the degree of the power series. When  $x < X3$ ,  $X3$  is used for  $x$ ; when  $x > X4$ ,  $X4$  is used for  $x$ . This condition has the effect of placing bounds on the table; there is no extrapolation outside of the table boundaries.

## Spatial Distribution of Loading – DAREA Entry

The EXCITEID field on an RLOADi bulk entry can select DAREA, FORCEi, MOMENTi, PLOAD, PLOAD1, SELOAD, and SPCD bulk entries. These entries are used to define the degree-of-freedom where the dynamic load is applied and the scale factor that is applied to the loading.

The DAREA entry provides a simple, direct means to define the spatial distribution for a dynamic loading.

1	2	3	4	5	6	7	8	9	10
DAREA	SID	P1	C1	A1	P2	C2	A2		

Field	Contents
SID	Identification number.
Pi	Grid, extra, or scalar point ID.
Ci	Component number.
Ai	Scale factor.

Any number of DAREA entries may be used; all those with the same SID are combined.

## Dynamic Load Set Combination – DLOAD

The SID of all RLOAD1, RLOAD2, and RLOADEX bulk entries must be unique. The DLOAD bulk entry allows you to use multiple RLOAD1, RLOAD2, and RLOADEX entries. The DLOAD entry constructs the total applied load from a linear combination of the individual dynamic loads as follows:

$$\{P\} = S \sum_i S_i \{P_i\}$$

Equation 6-28.

where the index  $i$  refers to the  $i^{\text{th}}$  dynamic load and not the  $i^{\text{th}}$  mode. Specifically:

- $P_i$  = Dynamic load  $i$  (either an RLOAD1, RLOAD2, or RLOADEX entry)
- $S_i$  = Scale factor for dynamic load  $i$
- $S$  = Overall scale factor
- $P$  = Total applied load

The DLOAD bulk entry has the following format:

1	2	3	4	5	6	7	8	9	10
DLOAD	SID	S	S1	L1	S2	L2	S3	L3	
	S4	L4	- etc. -						

Field	Contents
SID	Load Set ID.
S	Overall scale factor.
Si	Individual scale factors.
Li	Identification numbers of RLOAD1, RLOAD2, and RLOADEX entries.

### DLOAD Example

Suppose the following command is in the case control section of the input file:

DLOAD = 33

Also suppose the following entries are in the bulk data section of the input file:

1	2	3	4	5	6	7	8	9	10
\$DLOAD	SID	S	S1	L1	S2	L2			
DLOAD	33	1.0	1.0	35	1.0	36			
\$RLOAD1	SID	EXCITEID	DELAY	DPHASE	TC	TD	TYPE		
RLOAD1	35	29	0.2	5.0	40	0.0	0		
RLOAD1	36	29	31	32	4.0	41	0		
\$DAREA	SID	GRID	COMP	SCALE					
DAREA	29	30	1	5.2					
\$DELAY	SID	GRID	COMP	LAG					
DELAY	31	30	1	0.2					
\$DPHASE	SID	GRID	COMP	ANGLE					
DPHASE	32	30	1	5.0					
\$TABLED1	TID	XAXIS	YAXIS						
\$	x1	y1	x2	y2	x3	y3	x4	y4	
TABLED1	40	LINEAR	LINEAR						
	0.0	4.0	2.0	8.0	6.0	8.0	ENDT		
TABLED1	41	LINEAR	LINEAR						
	0.0	0.5	0.6	0.4	0.8	0.7	ENDT		

In the example:

- The DLOAD case control command selects the loading referenced by the DLOAD bulk entry having SID = 33 as the dynamic loading for the analysis.
- The DLOAD bulk entry combines the dynamic loads defined by two RLOAD1 entries having SIDs of 35 and 36. Neither dynamic load is scaled using the DLOAD entry.
- Both RLOAD1 entries reference the same DAREA entry. Thus, both dynamic loads are applied to the same degree-of-freedom. In this example, it is a single degree-of-freedom, Component 1 of Grid 30. Both dynamic loads are scaled 5.2 times by the DAREA entry.
- Because the dynamic loads are applied at only one degree-of-freedom, the time delay and phase angle can be defined directly on the RLOAD1 entries. This is the case for the RLOAD1 entry having SID = 35. However, for demonstration purposes, the RLOAD1 entry having SID = 36 references DELAY and DPHASE bulk entries. Both approaches define a delay of 0.2 and a phase angle of 5.0 for the corresponding dynamic load.
- $C(f)$  for the RLOAD1 entry having SID = 35 is defined by the TABLED1 entry having TID = 40. (See [Figure 6-6](#).)  $D(f)$  for this same RLOAD1 entry is defined as zero.
- $C(f)$  for the RLOAD1 entry having SID = 36 is a constant value of 4.0.  $D(f)$  for this same RLOAD entry is defined by the TABLED1 entry having TID = 41.

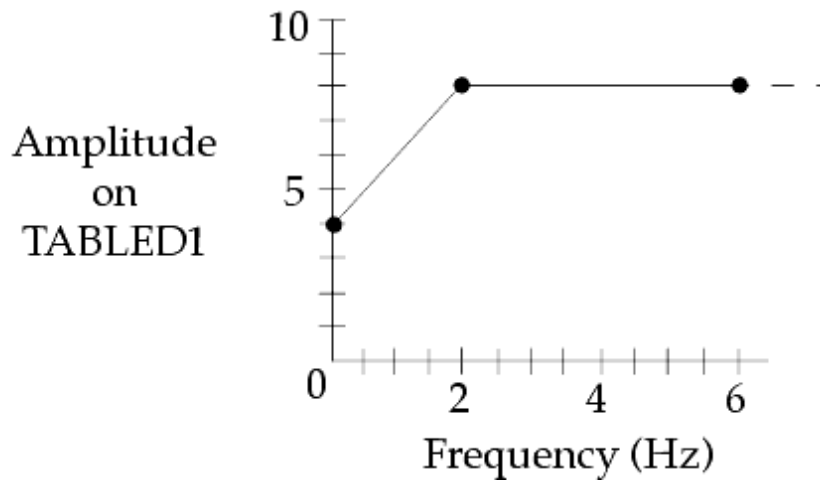


Figure 6-6. Amplitude vs. Frequency for TABLED1 with TID = 40

## LSEQ Method

The recommended method to define dynamic loads in a frequency response analysis is to use the DLOAD construction outlined in the previous example. However, a legacy method based on the use of the LSEQ bulk entry is still supported.

The LSEQ entry has the following format:

1	2	3	4	5	6	7	8	9	10
LSEQ	SID	EXCITEID	LID	TID					

- The SID field is referenced by a LOADSET case control command.
- The LID field selects static loads like FORCE<sub>i</sub>, PLOAD<sub>i</sub>, and GRAV entries that have a load set identification number equal to LID. The degree-of-freedom referenced in the static loads define the spatial distribution of the dynamic loading.
- The EXCITEID field selects the RLOAD<sub>i</sub> entries that have the same EXCITEID. These RLOAD<sub>i</sub> entries constitute the temporal distribution of the dynamic loading.

The LSEQ entry links the temporal and spatial distributions so that the appropriate excitation is applied at the correct degree-of-freedom. In effect, it is as if the EXCITEID fields of the LSEQ and RLOAD<sub>i</sub> entries referenced the same DAREA entry, even though a DAREA entry is not present in the input file. **Figure 6-7** depicts the relationships between the various inputs.

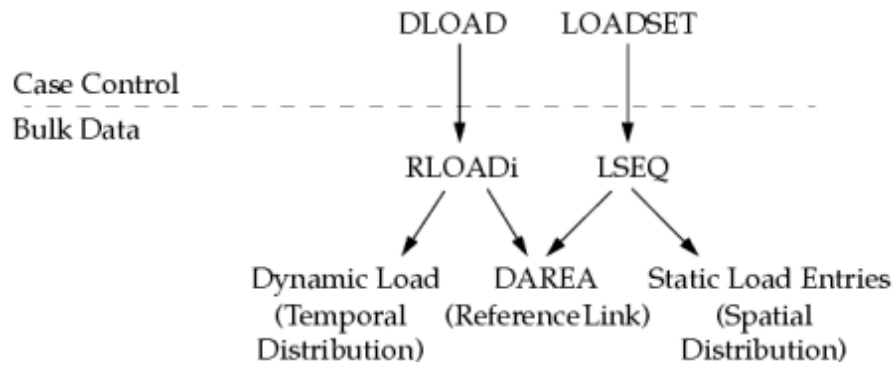


Figure 6-7. Relationship of Dynamic and Static Load Entries

## LSEQ Example

Suppose the following commands are in the case control section of the input file:

LOADSET = 27

DLOAD = 25

Also suppose the following entries are in the bulk data section of the input file:

1	2	3	4	5	6	7	8	9	10
\$LSEQ	SID	EXCITEID	LID						
LSEQ	27	28	26						
\$RLOAD1	SID	EXCITEID	DELAY	DPHASE	TC	TD			
RLOAD1	25	28	0.0	10.0	29				
\$FORCE	SID	GRID	CID	F	N1	N2	N3		
FORCE	26	425		2.5	1.0				
\$PLOAD	SID	PRES	GRID1	GRID2	GRID3	GRID4			
PLOAD	26	50.0	63	64	88	91			
\$TABLED1	TID	XAXIS	YAXIS						
\$	x1	y1	x2	y2	x3	y3	x4	y4	
TABLED1	29	LINEAR	LINEAR						
	0.0	0.5	0.6	0.4	0.8	0.7	ENDT		

In the example:

- The LOADSET request in case control selects the LSEQ entry having SID = 27.
- The LSEQ entry references the static loads having SID = 26. These loads include the FORCE and PLOAD entries. The FORCE and PLOAD entries provide the spatial distribution of the dynamic loading.
- The DLOAD request in case control selects the RLOAD1 entry having SID = 25.

- The RLOAD1 entry references a TABLED1 entry having TID = 29. This TABLED1 entry defines  $C(f)$  for the RLOAD1 entry. Because the TD field on the RLOAD1 entry is undefined,  $D(f)$  defaults to zero.
- The EXCITEID fields of the LSEQ and RLOAD1 entries are both 28, thereby linking the temporal and spatial distributions of the dynamic loading. Thus, the dynamic load defined by the RLOAD1 entry is:
  - Scaled by 2.5 and applied as a force to Component 1 of Grid 425.
  - Scaled by 50.0 and applied as a pressure to the quadrilateral element face defined by Grids 63, 64, 88, and 91.

## 6.7 Solution Frequencies

A major consideration when you conduct a frequency response analysis is selecting the frequency at which the solution is to be performed. There are six bulk entries that you can use to select the solution frequencies. It is important to remember that each specified frequency results in an independent solution at the specified excitation frequency.

To select the loading frequencies, use the FREQ, FREQ1, FREQ2, FREQ3, FREQ4 and FREQ5 bulk entries.

Bulk Entry	Description
FREQ	Defines discrete excitation frequencies.
FREQ1	Defines a starting frequency $F_{start}$ , a frequency increment $\Delta f$ , and the number of frequency increments.
FREQ2	Defines a starting frequency $F_{start}$ , and ending frequency $F_{end}$ , and the number of logarithmic intervals.
FREQ3 <sup>(1)</sup>	Defines the number of excitation frequencies used between modal pairs in a given range.
FREQ4 <sup>(1)</sup>	Defines excitation frequencies using a spread about each normal mode within a range.
FREQ5 <sup>(1)</sup>	Defines excitation frequencies as all frequencies in a given range as a defined fraction of the normal modes.
<sup>(1)</sup> Used for modal solution only.	

The FREQUENCY case control command selects FREQ<sub>i</sub> bulk entries. All FREQ<sub>i</sub> entries with the same selected Set ID are applied in the analysis; therefore, you can use any combination of FREQ, FREQ1, FREQ2, FREQ3, FREQ4 and FREQ5 entries.

The examples that follow show the formats of the FREQ<sub>i</sub> entries. The six sets of excitation frequencies shown in the examples will be combined in a single analysis if the Set IDs are identical.

### FREQ

This FREQ entry specifies ten unequally spaced frequencies.

1	2	3	4	5	6	7	8	9	10
\$FREQ	SID	F1	F2	F3	F4	F5	F6	F7	
\$	F8	F9	F10	-etc.-					
FREQ	3	2.98	3.05	17.9	21.3	25.6	28.8	31.2	
	29.2	22.4	19.3						

Field	Contents
SID	Identification number.
Fi	Frequency (cycles per unit time).

## FREQ1

This FREQ1 entry specifies 14 frequencies between 2.9 and 9.4 cycles per unit time in increments of 0.5 cycles per unit time.

\$FREQ1	SID	$F_{start}$	$\Delta f$	NDF					
FREQ1	6	2.9	0.5	13					

Field	Contents
SID	Identification number.
$F_{start}$	Starting frequency in set (cycles per unit time).
$\Delta f$	Frequency increment (cycles per unit time).
NDF	Number of frequency increments.

## FREQ2

This FREQ2 entry specifies six logarithmic frequency intervals between 1.0 and 8.0 cycles per unit time, resulting in frequencies at 1.0, 1.4142, 2.0, 2.8284, 4.0, 5.6569, and 8.0 cycles per unit time.

1	2	3	4	5	6	7	8	9	10
\$FREQ2	SID	$F_{start}$	$F_{end}$	NF					
FREQ2	9	1.0	8.0	6					

Field	Contents
SID	Identification number.
$F_{start}$	Starting frequency (cycles per unit time).
$F_{end}$	Ending frequency (cycles per unit time).
NF	Number of logarithmic intervals.



## FREQ3

This FREQ3 entry specifies 10 frequencies between each set of modes within the range 20 and 2000 cycles per unit time, plus ten frequencies between 20 cycles per unit time and the lowest mode in the range, plus 10 frequencies between the highest mode in the range and 2000 cycles per unit time.

1	2	3	4	5	6	7	8	9	10
\$FREQ3	SID	F1	F2	TYPE	NEF	CLUSTER			
FREQ3	6	20.0	2000.0	LINEAR	10	2.0			

Field	Contents
SID	Identification number.
F1	Lower bound of modal frequency range in cycles per unit time.
F2	Upper bound of modal frequency range in cycles per unit time.
TYPE	Specifies linear or logarithmic interpolation between frequencies.
NEF	Number of excitation frequencies within each subrange including the end points.
CLUSTER	A CLUSTER value greater than 1 provides closer spacing of excitation frequencies near the modal frequencies, where greater resolution is needed.

## FREQ4

This FREQ4 entry specifies 21 equally spaced frequencies across a frequency band of  $0.7 \cdot f_N$  to  $1.3 \cdot f_N$  for each natural frequency between 20 and 2000 cycles per unit time.

1	2	3	4	5	6	7	8	9	10
\$FREQ4	SID	F1	F2	FSPD	NFM				
FREQ4	6	20.0	2000.0	0.30	21				

Field	Contents
SID	Identification number.
F1	Lower bound of modal frequency range in cycles per unit time.
F2	Upper bound of modal frequency range in cycles per unit time.
FSPD	Frequency spread, +/- the fractional amount specified for each mode which occurs in the frequency range F1 to F2.
NFM	Number of evenly spaced frequencies per "spread" mode.

## FREQ5

This FREQ5 entry specifies excitation frequencies that are 0.6, 0.8, 0.9, 0.95, 1.0, 1.05, 1.1, and 1.2 times the natural frequencies for all natural frequencies, but use only the computed frequencies that fall within the range 20 and 2000 cycles per unit time.

1	2	3	4	5	6	7	8	9	10
\$FREQ5	SID	F1	F2	FR1	FR2	FR3	FR4	FR5	
\$	FR6	FR7	-etc.-						
FREQ5	6	20.0	2000.0	1.0	0.6	0.8	0.9	0.95	
	1.05	1.1	1.2						

Field	Contents
SID	Identification number.
F1	Lower bound of modal frequency range in cycles per unit time.
F2	Upper bound of modal frequency range in cycles per unit time.
FRi	Fractions of the natural frequencies in the range F1 to F2.

## 6.8 Frequency Response Considerations

Exciting an undamped or viscous damped system at 0.0 Hz using direct frequency response analysis gives the same results as a static analysis. Depending on the number of retained modes, a modal frequency response analysis provides nearly the same result. Therefore, if the maximum excitation frequency is much less than the lowest resonant frequency of the system, a static analysis is probably sufficient.

Undamped or very lightly damped structures exhibit large dynamic responses for excitation frequencies near resonant frequencies. A small change in the model (or running it on another computer) may result in large changes in such responses.

Use a fine enough frequency step size ( $\Delta f$ ) to adequately predict peak response. Use at least five points across the half-power bandwidth (which is approximately  $2\xi f_n$  for a single degree-of-freedom system) as shown in [Figure 6-8](#).

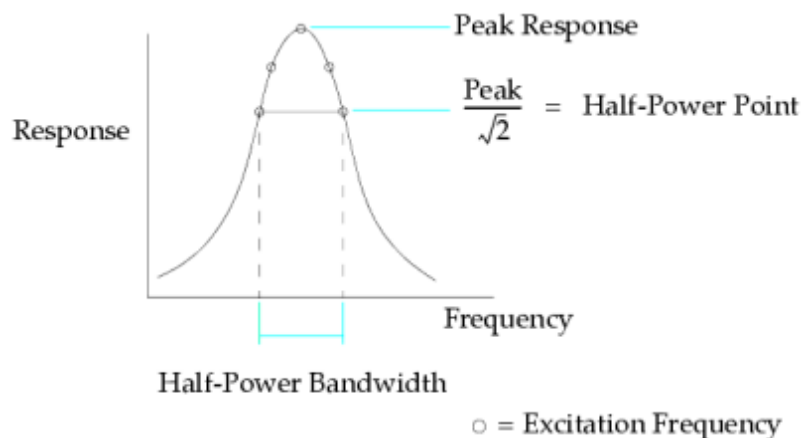


Figure 6-8. Half-Power Bandwidth

For maximum efficiency, an uneven frequency step size should be used. Smaller frequency spacing should be used in regions near resonant frequencies, and larger frequency step sizes should be used in regions away from resonant frequencies.

## 6.9 Frequency Response Solution Control and Output

The following tables summarize the data entries that can be used to control a frequency response analysis. Certain data entries are required, some data entries are optional, and others are user selectable.

In the executive control section of the Simcenter Nastran input file, you must select a solution using the SOL n statement where n is an integer value chosen from [Table 6-47](#).

**Table 6-47. Frequency Response Solutions in Simcenter Nastran**

Solution Type	Structured Solution Sequences
Direct	108
Modal	111

In the case control section of the Simcenter Nastran input file, you must select the solution parameters associated with the current analysis like frequencies, loads, and boundary conditions, and also the output quantities required from the analysis. The case control commands directly related to frequency response analysis are listed in [Table 6-48](#). They can be combined in the standard fashion with the more generic commands like SPC and MPC, among others.

**Table 6-48. Case Control Commands for Frequency Response Solution Control**

Case Control Command	Direct or Modal	Description	Required/Optional
DLOAD	Both	Select the dynamic load set from bulk data	Required
FREQUENCY	Both	Select FREQi entries from bulk data	Required
METHOD	Modal	Select the eigenvalue extraction parameters	Required
LOADSET	Both	Select the LSEQ set from bulk data	Optional
SDAMPING	Modal	Select TABDMP1 entry from bulk data	Optional
OFREQUENCY	Both	Select the frequencies for output (default = all)	Optional

A number of bulk entries are unique to frequency response analysis. These entries can be combined with other generic entries in the bulk data section. Bulk entries directly related to frequency response analysis are summarized in [Table 6-49](#).

**Table 6-49. Bulk Entries for Frequency Response Analysis**

Bulk Entry	Direct or Modal	Description	Required/Optional
FREQ, FREQi	Both	Excitation frequencies	Required
RLOAD1, RLOAD2, RLOADEX	Both	Dynamic loading	Required
EIGR or EIGRL	Modal	Eigenvalue analysis parameters	Required
DLOAD	Both	Dynamic load combination	Optional <sup>(1)</sup>
LSEQ	Both	Dynamic loading from static loads	Optional
TABLEDi	Both	Frequency-dependent tables for RLOADi	Optional <sup>(2)</sup>
DAREA	Both	Load component and scale factor	Optional
DELAY	Both	Time delay on dynamic load	Optional
DPHASE	Both	Phase angle on dynamic load	Optional
TABDMP1	Modal	Modal damping table	Optional
<sup>(1)</sup> Required when multiple RLOADi entries are used.			
<sup>(2)</sup> Required to define frequency-dependent dynamic loading functions.			

The types of results available from a frequency response analysis are similar to those for a typical static analysis except that the results are a complex function of the applied loading frequency. Additional quantities are also available, which is characteristic of dynamics problems. The output quantities are summarized in [Table 6-50](#) and [Table 6-51](#).

**Table 6-50. Grid Output from a Frequency Response Analysis**

Case Control Command	Description
ACCELERATION	Grid point acceleration results for a set of grid points.
DISPLACEMENT	Grid point displacement results for a set of grid points.
GRDCON	Contribution of grids on structural panels to the acoustic response results
OLOAD	Requests the applied load table to be output for a set of grid points.

Case Control Command	Description
SACCELERATION	Requests the solution set acceleration output: d-set in direct solutions and modal variables in modal solutions.
SDISPLACEMENT	Requests the solution set displacement output: d-set in direct solutions and modal variables in modal solutions.
SVECTOR	Requests the real eigenvector output for the a-set in modal solutions.
SVELOCITY	Requests the solution set velocity output: d-set in direct solutions and modal variables in modal solutions.
SPCFORCES	Requests the forces of a single-point constraint for a set of grid points.
VELOCITY	Grid point velocity results for a set of grid points.

Frequency response output is in real/imaginary format (the default) or magnitude/phase format (the phase angle is in degrees). Frequency response output is also in SORT1 or SORT2 format. In SORT1 format, the results are listed by frequency; in SORT2 format, the results are listed by grid point or element number. SORT1 is the default for direct frequency response analysis (SOL 108), and SORT2 is the default for modal frequency response analysis (SOL 111). PARAM,CURVPLOT,1 and PARAM,DDRMM,-1 are necessary to obtain SORT1 output in SOL 111. These output formats are specified with the case control commands. The command:

```
DISPLACEMENT (PHASE, SORT2) = ALL
```

prints displacements in magnitude/phase and SORT2 formats. The output formats are illustrated in the first example in the [“Examples of Frequency Response Analysis”](#) section.

**Table 6-51. Element and Modal Output from a Frequency Response Analysis**

Case Control Command	Description
ELFORCE (or FORCE)	Element force results for a set of elements
ERP	Equivalent radiated power results of structural panels and shell elements
MODCON	Contribution of modes to structural or acoustic response results
PANCON	Contributions of structural panels to acoustic response results
STRAIN	Element strain results for a set of elements
ELSTRESS (or STRESS)	Element stress results for a set of elements

### von Mises Stress and Strain

von Mises stress and strain are computed by default in SOL 108 or SOL 111 when stress and strain results are requested. The von Mises stress and strain is calculated using the Charron method. For information on this method, see Charron, Donato, and Fontaine, *Exact Calculation of Minimum Margin*

of Safety for Frequency Response Analysis Stress Results Using Yielding or Failure Theories, MSC 1993 World Users' Conference Proceedings.

**Table 6-52. Elements Supported for von Mises Stress and Strain**

Element Name	Item Code	Description
CHEXA	67	Linear format
CHEXA	269	Composite – center only
CPENTA	68	Linear format
CPENTA	270	Composite – center only
CPLSTN3	271	Linear format – plane strain – center only
CPLSTN4	272	Linear format – plane strain – center and corners
CPLSTN6	273	Linear format – plane strain – center and corners
CPLSTN8	274	Linear format – plane strain – center and corners
CPLSTS3	275	Linear format – plane stress – center only
CPLSTS4	276	Linear format – plane stress – center and corners
CPLSTS6	277	Linear format – plane stress – center and corners
CPLSTS8	278	Linear format – plane stress – center and corners
CPYRAM	255	Linear format
CQUAD4	33	Linear format – center only
CQUAD4	95	Composite – center only
CQUAD4	144	Linear format – center and corners
CQUAD8	64	Linear format – center and corners
CQUAD8	96	Composite – center only
CQUADR	82	Linear format – center and corners
CQUADR	228	Linear format – center only
CQUADR	232	Composite – center only
CQUADX4	243	Linear format – grid and Gauss – center and corners
CQUADX8	245	Linear format – grid and Gauss – center and corners
CTETRA	39	Linear format
CTRAX3	242	Linear format – grid and Gauss – center and corners
CTRAX6	244	Linear format – grid and Gauss – center and corners
CTRIA3	74	Linear format – center only
CTRIA3	97	Composite – center only
CTRIA6	75	Linear format – center and corners

Element Name	Item Code	Description
CTRIA6	98	Composite – center only
CTRIAR	70	Linear format – center and corners
CTRIAR	227	Linear format – center only
CTRIAR	233	Composite – center only

To disable the computation of von Mises stress and strain, you can specify the system cell as follows:

```
NASTRAN SYSTEM(579)=1 or NASTRAN FREQVM=1
```

When NASTRAN SYSTEM(579)=0 (default), stress and strain results are written to the OESVM/OSTRVM output data blocks. If NASTRAN SYSTEM(579)=1, stress and strain results are written to the OES/OSTR output data blocks.

## Frequency Response Laminate Output

The PCOMP or PCOMPG property entries can be used to model a laminate with the shell elements CTRIA3, CTRIA6, CTRIAR, CQUAD4, CQUAD8, and CQUADR. The PCOMPS property entry can be used to model a laminate with the solid elements CHEXA and CPENTA. The following output is supported when a shell or solid laminate is included in the frequency response solutions 108 and SOL 111 including random response. See the remarks on the PCOMP, PCOMPG, and PCOMPS bulk entries in the *Quick Reference Guide* for information on these output requests. Also see the “Laminates” section of the *Simcenter Nastran User’s Guide*.

- Stress and strain output for the individual lamina in either SORT1 or SORT2 format.
- Stress resultants, which are requested with the FORCE case control command.
- Interlaminar shear stress in the bonding material.
- The failure indices and strength ratios. Because frequency response analysis results are complex, the software calculates the failure indices and strength ratios at discrete phase angles over a full 360 degree range. The worst case value is output. By default, the calculation is performed in one degree increments. However, you can optionally specify that the software use a different angular increment by using the parameter SWPANGLE.
- If you are doing a random response analysis, you can use the PSDF, ATOC, CRMS, RMS, or RALL descriptors on the STRESS or STRAIN case control commands to request random output for an element which is defined as a laminate. The requested random output is computed and output for each ply. Elements which are defined as a laminate cannot be selected on the RCROSS bulk entry.
- Complex stresses and strains are not computed for QUADR and TRIAR elements that are used to model laminates when SYSTEM(370)=1.

## Modal, Panel, and Grid Contribution Output

In frequency response analysis, you can use the MODCON, PANCON, and GRDCON case control commands to produce output that relates the response at specified locations at each excitation frequency to structural or fluid modes, structural panels, or grids in structural panels.

- MODCON is applicable to modal solutions only and can be used in the following types of frequency response analysis problems:
  - Structural  
Use MODCON to examine how each structural mode contributes to the structural response.
  - Acoustic  
Use MODCON to examine how each fluid mode contributes to the acoustic response.
  - Vibro-acoustic  
Use MODCON to examine how each structural mode contributes to the acoustic response, how each fluid mode contributes to the structural response, or how with the PANELMC describer, each structural mode contributes to the acoustic response on a structural panel-by-structural panel basis.

You can also use MODCON in structural transient response analysis to examine how each structural mode contributes to the structural response at a specific location at each time step.

- PANCON is applicable to direct and modal solutions and can be used in vibro-acoustic frequency response analysis only.  
Use PANCON to examine how the motion of structural panels contribute to the acoustic response.
- GRDCON is applicable to direct and modal solutions and can be used in vibro-acoustic frequency response analysis only.  
Use GRDCON to examine how the motion of grids in structural panels contribute to the acoustic response.

For more information on using the PANCON and GRDCON case control commands in a vibro-acoustic frequency response analysis, see the *Simcenter Nastran Acoustics User's Guide*.

### MODCON case control command

For SORT1 output, at each excitation frequency or time (as specified by `FREQ` or `TSTEP`) and response (as specified by `SETMC` grid and `DOF` entries) combination, the software ranks the contribution that each mode makes to the response based on the results of the following equation:

$$MAG_i(j) = \left| \frac{X_i(j)}{X(j)} \right|$$

where:



$MAG_i(j)$  = magnitude of contribution at excitation frequency or time  $j$  from mode  $i$  (real)

$X_i(j)$  = modal contribution to response at excitation frequency or time  $j$  from mode  $i$  (real or complex)

$X(j)$  = total response at excitation frequency or time  $j$  (real or complex)

You can use the TOPS describer to specify the number of top contributing structural modes that the software outputs for each excitation frequency or time and response combination. In the output, the modes are listed from the highest contributor to the lowest contributor.

For example, if you specify TOPS = 10, for each excitation frequency or time and response combination, the software outputs the 10 modes that contribute the most. If you do not specify the TOPS describer, by default, the software outputs the 5 modes that contribute the most.

Similarly, you can use the TOPF describer to specify the number of top contributing fluid modes that the software outputs for each excitation frequency or time and response combination.

You can use the ABS, NORM, or BOTH describers to specify the format of the output.

For example, specify NORM to output normalized modal contributions and specify ABS to output non-normalized modal contributions. Specify BOTH to output modal contributions in both formats.

For SORT1 output, the software calculates the normalized modal contributions as follows:

$$NMC_i(j) = \frac{X_i(j)}{X(j)}$$

where  $NMC_i(j)$  is the normalized modal contribution to the response at excitation frequency or time  $j$  from mode  $i$ .

Example SORT1 input:

```

...
$ CASE CONTROL
SETMC 99 = DISP/899 (T1) , 899 (T2) , 899 (T3) ,
           899 (R1) , 899 (R2) , 899 (R3) ,
           SPCF/2732 (T1) , 2732 (T2) , 2732 (T3) ,
           2627 (R1) , 2627 (R2) , 2627 (R3) ,
           STRE/1483 (3) , 1483 (4) , 1483 (5) ,
           1483 (6) , 1483 (7) , 1483 (8) ,
           1499 (6) , 1609 (6) , 1563 (3) ,
           STRA/1483 (3) , 1483 (4) , 1483 (5) ,
           1483 (6) , 1483 (7) , 1483 (8) ,
           1499 (6) , 1609 (6) , 1563 (3) ,
           ELFO/1483 (2) , 1483 (3) , 1483 (4) ,
           1483 (10) , 1483 (11) , 1483 (12) ,

```

```

1563(2)
MODCON(SORT1,PRINT,ABS) = 99
FREQ = 1
...
BEGIN BULK
FREQ1,1,0.0,400.0,4
...
ENDDATA

```

Example SORT1 output:

```

...
                                COMPLEX ABSOLUTE MODAL CONTRIBUTIONS
FREQUENCY = 0.000000E+00      DISPLACEMENT, DOF = 899X
CONTRIBUTIONS FROM STRUCTURAL MODES
      MODE      MODAL FREQUENCY      REAL      IMAGINARY
TOTAL
  9      1.717074E+03      -7.823264E+00      -1.199629E+01
  7      1.242137E+03      -1.778529E+01      -2.727217E+01
  8      1.546779E+03      4.683915E+00      7.182372E+00
 11      5.682238E+03      3.681103E+00      5.644648E+00
 13      2.443404E+04      1.371542E+00      2.103139E+00
 13      2.443404E+04      3.012281E-01      4.619068E-01
...

```

```

...
                                COMPLEX ABSOLUTE MODAL CONTRIBUTIONS
FREQUENCY = 4.000000E+02      DISPLACEMENT, DOF = 899X
CONTRIBUTIONS FROM STRUCTURAL MODES
      MODE      MODAL FREQUENCY      REAL      IMAGINARY
TOTAL
  9      1.717074E+03      1.945155E+01      4.264571E+01
  7      1.242137E+03      4.609176E+01      1.005117E+02
  8      1.546779E+03      -1.293350E+01      -2.787089E+01
 11      5.682238E+03      -9.694567E+00      -2.107223E+01
 13      2.443404E+04      -3.326621E+00      -7.390866E+00
 13      2.443404E+04      -7.236785E-01      -1.617209E+00
...

```

For SORT2 output, at each modal frequency and response (as specified by SETMC grid and DOF entries) combination, the software ranks the contribution that the mode makes at each excitation frequency or time (as specified by FREQ or TSTEP) to the response based on the results of the following equation:

$$MAG_i = \frac{\sqrt{\frac{1}{N} \sum_{j=1}^N X_i(j) * X_i^*(j)}}{\sqrt{\frac{1}{N} \sum_{j=1}^N X(j) * X^*(j)}}$$

where:

$MAG_i(j)$  = magnitude of contribution at excitation frequency or time  $j$  from mode  $i$  (real)

$X_i(j)$  = modal contribution to response at excitation frequency or time  $j$  from mode  $i$  (real or complex)

$X_i^*(j)$  = complex conjugate of  $X_i(j)$

$X(j)$  = total response at excitation frequency or time  $j$  (real or complex)

$X^*(j)$  = complex conjugate of  $X(j)$

$N$  = total number of excitation frequency increments or time steps

You can use the TOPS describer to specify the number of excitation frequencies or times where the structural mode contributes most to the response for the software to output for each modal frequency and response combination. In the output, the excitation frequencies or times are listed from the highest contributor to the lowest contributor.

For example, if you specify TOPS = 10, for each structural mode and response combination, the software outputs the contribution of the structural mode at the 10 excitation frequencies or times where the contribution of the structural mode is greatest. If you do not specify the TOPS describer, by default, the software outputs the contribution of the structural mode at the 5 excitation frequencies or times where the contribution of the structural mode is greatest.

Similarly, you can use the TOPF describer to specify the number of excitation frequencies or times where the fluid mode contributes most to the response for the software to output for each modal frequency and response combination.

You can use the ABS, NORM, or BOTH describers to specify the format of the output.

For example, specify NORM to output normalized modal contributions and specify ABS to output non-normalized modal contributions. Specify BOTH to output modal contributions in both formats.

For SORT2 output, the software calculates the normalized modal contributions as follows:

$$NMC_i(j) = \frac{X_i(j)}{\sqrt{\frac{1}{N} \sum_{j=1}^N X(j) * X^*(j)}}$$

where  $NMC_i(j)$  is the normalized modal contribution to the response at excitation frequency or time  $j$  from mode  $i$ .

Example SORT2 input:

```

...
$ CASE CONTROL
SETMC 99 = DISP/899 (T1), 899 (T2), 899 (T3),
          899 (R1), 899 (R2), 899 (R3),
          SPCF/2732 (T1), 2732 (T2), 2732 (T3),
          2627 (R1), 2627 (R2), 2627 (R3),
          STRE/1483 (3), 1483 (4), 1483 (5),
          1483 (6), 1483 (7), 1483 (8),
          1499 (6), 1609 (6), 1563 (3),
          STRA/1483 (3), 1483 (4), 1483 (5),
          1483 (6), 1483 (7), 1483 (8),
          1499 (6), 1609 (6), 1563 (3),
          ELFO/1483 (2), 1483 (3), 1483 (4),
          1483 (10), 1483 (11), 1483 (12),
          1563 (2)
MODCON (SORT2, PRINT, ABS) = 99
FREQ = 1
...
BEGIN BULK
FREQ1, 1, 0.0, 400.0, 4
...
ENDDATA

```

Example SORT2 output:

```

...
          COMPLEX ABSOLUTE MODAL CONTRIBUTIONS
TOTAL          DISPLACEMENT, DOF = 899X
CONTRIBUTIONS FROM STRUCTURAL MODES
      FREQUENCY          REAL          IMAGINARY

      0.000000E+00      -7.823264E+00      -1.199629E+01
      4.000000E+02       1.945155E+01       4.264571E+01
      8.000000E+02      -6.615797E+01      -1.462096E+02
      1.200000E+03      -1.317159E+03      -4.915609E+02
      1.600000E+03      -3.859322E+03      -7.242293E+03
...
          COMPLEX ABSOLUTE MODAL CONTRIBUTIONS
MODE 9,  MODAL FREQUENCY = 1.717074E+03  DISPLACEMENT, DOF = 899X
CONTRIBUTIONS FROM STRUCTURAL MODES
      FREQUENCY          REAL          IMAGINARY

      0.000000E+00      -1.778529E+01      -2.727217E+01
      4.000000E+02       4.609176E+01       1.005117E+02
      8.000000E+02      -1.954668E+02      -4.110245E+02
      1.200000E+03       5.426884E+02       1.055978E+03
      1.600000E+03      -4.028579E+03      -4.822341E+03
...

```

When you use the MODCON case control command in a vibro-acoustic analysis, and you specify the PANELMC describer, you can examine how each structural mode contributes to the acoustic response on a structural panel-by-structural panel basis. The ranking of modal contributions and the normalization of the modal contributions are the same as when the PANELMC describer is not specified.

Example SORT1/PANELMC input:

```

...
$ CASE CONTROL
SETMC 99 = PRES/198(T1)
MODCON(SORT1,PRINT,ABS,TOPS=3,PANELMC=ALL) = 99
FREQ = 1
...
BEGIN BULK
FREQ1,1,0.0,400.0,4
...
ENDDATA
    
```

Example SORT1/PANELMC output:

```

...
                                COMPLEX ABSOLUTE MODAL CONTRIBUTIONS
FREQUENCY = 4.000000E+02      PRESSURE, GRID =      198
CONTRIBUTIONS FROM STRUCTURAL MODES
    MODE                MODAL FREQUENCY                REAL                IMAGINARY

TOTAL
  7                4.401321E+01                -7.950658E-09                -1.763784E-08
  9                5.033400E+01                5.730663E-09                -2.514131E-09
 18                9.068459E+01                8.421640E-11                1.270300E-09

                                CONTRIBUTIONS FROM PANELS
MODE                7                MODAL FREQUENCY = 4.401321E+01
PANEL                REAL                IMAGINARY

TOTAL MC                -7.950658E-09                -1.763784E-08
TOP                -2.304793E-09                -4.016536E-09
BOTTOM                -2.304793E-09                -4.016536E-09
LEFT                -2.304793E-09                -4.016536E-09
RIGHT                -2.304793E-09                -4.016536E-09

MODE                9                MODAL FREQUENCY = 5.033400E+01
PANEL                REAL                IMAGINARY

TOTAL MC                5.730663E-09                -2.514131E-09
TOP                9.568939E-10                -5.644531E-10
BOTTOM                9.568939E-10                -5.644531E-10
    
```

LEFT		9.568939E-10	-5.644531E-10
RIGHT		9.568939E-10	-5.644531E-10
MODE	18	MODAL FREQUENCY = 9.068459E+01	
PANEL		REAL	IMAGINARY
TOTAL MC		8.421640E-11	1.270300E-09
TOP		-1.359490E-10	3.881714E-10
BOTTOM		-1.359490E-10	3.881714E-10
LEFT		-1.359490E-10	3.881714E-10
RIGHT		-1.359490E-10	3.881714E-10
...			

## Equivalent Radiated Power Output

You can use the ERP case control command to help identify the areas of a structure that are the primary sources of the noise due to structural vibration. You can request equivalent radiated power output for panels that are constructed from the following:

- CQUAD4, CQUAD8, CQUADR, CTRIA3, CTRIA6, and CTRIAR shell elements
- PLOTEL3, PLOTEL4, PLOTEL6, and PLOTEL8 visualization elements that surface coat exterior surfaces of solid elements.

**Note:**

Because visualization elements do not contribute mass, damping, or stiffness to the model, they inherit their motion directly from the underlying solid elements.

To request that equivalent radiated power be calculated and output, include the ERP case control command in your input file.

ERP output is supported in the frequency response solutions 108 and 111. You can request ERP output for models that contain only structural elements, or models that include both structural and fluid elements.

Equivalent radiated power is a measure of the normal velocity distribution over the face of an element. The equivalent radiated power attributable to the  $i^{\text{th}}$  element at frequency  $\omega_k$  is given by:

$$ERPE_i(\omega_k) = \frac{1}{2} c \int_{S_i} v_n(x, \omega_k) v_n^*(x, \omega_k) ds$$

where:

$c$  is the scaling coefficient specified with the ERPCOEFF describer, which is usually taken to be the product of density and speed of sound in the fluid medium.

$v_n(x, \omega_k)$  is the normal velocity as a function of position,  $x$ , at frequency  $\omega_k$ . Frequencies  $\omega_k$  are specified by the SOLUTION describer.

$v_n^*(x, \omega_k)$  is the complex conjugate of  $v_n(x, \omega_k)$ .

$S_i$  is the surface area of the  $i^{\text{th}}$  element.

The equivalent radiated power density for the  $i^{\text{th}}$  element at frequency  $\omega_k$  is given by:

$$ERPED_i(\omega_k) = \frac{ERPE_i(\omega_k)}{S_i}$$

The equivalent radiated power for the  $j^{\text{th}}$  panel at frequency  $\omega_k$  is the sum of the equivalent radiated power for each element comprising the  $j^{\text{th}}$  panel at frequency  $\omega_k$  and is given by:

$$ERPP_j(\omega_k) = \sum_i ERPE_i(\omega_k)$$

The equivalent radiated power density for the  $j^{\text{th}}$  panel at frequency  $\omega_k$  and is given by:

$$ERPD_j(\omega_k) = \frac{ERPP_j(\omega_k)}{A_j}$$

where  $A_j$  is the surface area of the  $j^{\text{th}}$  panel.

When the SORT1 (default) describer is specified, the values in the "FRACTION" column represent the fraction of the total equivalent radiated power for all selected panels that is attributable to the  $j^{\text{th}}$  panel at frequency  $\omega_k$ . These values are computed as follows:

$$\overline{ERPP}_j(\omega_k) = \frac{ERPP_j(\omega_k)}{\sum_{m=1}^M ERPP_m(\omega_k)}$$

where  $M$  is the total number of selected panels.

The output is then grouped according to frequency and sorted by panel in descending order of equivalent radiated power.

For a given frequency, the value for equivalent radiated power density in the "TOTAL" row is the ratio of the total equivalent radiated power for all the selected panels at that frequency to the total area of those panels.

When the SORT2 descriptor is specified, for an individual panel, the values in the "FRACTION" column represent the fraction of the total equivalent radiated power for all frequencies that is attributable to the  $j^{\text{th}}$  panel at frequency  $\omega_k$ . These values are computed as follows:

$$\overline{ERPP}_j(\omega_k) = \frac{ERPP_j(\omega_k)}{\sum_{n=1}^N ERPP_j(\omega_n)}$$

where  $N$  is the total number of frequencies specified by the SOLUTION descriptor.

In the "PANEL=TOTAL" section of the SORT2 results, the values in the "FRACTION" column represent the fraction of the total equivalent radiated power for all panels at all frequencies that is attributable to all the panels at frequency  $\omega_k$ . These values are computed as follows:

$$\overline{ERPPT}(\omega_k) = \frac{\sum_{m=1}^M ERPP_m(\omega_k)}{\sum_{n=1}^N \left( \sum_{m=1}^M ERPP_m(\omega_n) \right)}$$

The following SORT1 and SORT2 output are from a model where the ERP case control command specifies output for three panels and two forcing frequencies.

### SORT1 Output

FREQUENCY = 3.000000E+03

EQUIVALENT RADIATED POWER

PANEL NAME	ERP	FRACTION	NORMAL AREA	ERP DENSITY
PANEL3	1.491040E+11	4.901360E-01	3.000000E+04	4.970134E+06
PANEL1	1.392054E+11	4.575974E-01	1.250000E+04	1.113644E+07
PANEL2	1.590001E+10	5.226666E-02	1.750000E+04	9.085721E+05
TOTAL	3.042095E+11	1.000000E+00	6.000000E+04	5.070158E+06

FREQUENCY = 4.000000E+03

EQUIVALENT RADIATED POWER

PANEL NAME	ERP	FRACTION	NORMAL AREA	ERP DENSITY
PANEL3	1.874405E+10	4.868036E-01	3.000000E+04	6.248018E+05
PANEL1	1.761857E+10	4.575737E-01	1.250000E+04	1.409486E+06
PANEL2	2.141717E+09	5.562273E-02	1.750000E+04	1.223838E+05
TOTAL	3.850434E+10	1.000000E+00	6.000000E+04	6.417390E+05



For an individual panel at each frequency, the values are calculated as follows:

- For the FRACTION column, the values are the ERP values for the panels normalized with respect to the total ERP for all the panels listed in the output. For example, the fraction of the total ERP at a frequency of 3000 for Panel 3 is  $1.491040 \times 10^{11} / 3.042095 \times 10^{11} = 4.901360 \times 10^{-1}$ .
- For the ERP DENSITY column, the values are the ERP value for the panel normalized with respect to the normal area value for the panel. For example, the ERP density at a frequency of 3000 for Panel 3 is  $1.491040 \times 10^{11} / 3.000000 \times 10^4 = 4.970134 \times 10^6$ .

In the TOTAL row at each frequency, the values are calculated as follows:

- For the ERP, FRACTION, and NORMAL AREA columns, the values are the sums of the values in the corresponding column. For example, the total ERP at a frequency of 3000 is  $1.491040 \times 10^{11} + 1.392054 \times 10^{11} + 1.590001 \times 10^{10} = 3.042095 \times 10^{11}$ .
- For the ERP DENSITY column, the value is the total ERP value normalized with respect to the total normal area value. For example, the ERP density at a frequency of 3000 is  $3.042095 \times 10^{11} / 6.000000 \times 10^4 = 5.070158 \times 10^6$ .

## **SORT2 Output**

```
PANEL=PANEL3 (AREA=3.000000E+04 ERP. MAXIMUM=1.491040E+11)
      EQUIVALENT RADIATED POWER
      FREQUENCY      ERP      FRACTION      ERP DENSITY
      3.000000E+03  1.491040E+11  8.883273E-01  4.970134E+06
      4.000000E+03  1.874405E+10  1.116727E-01  6.248018E+05
```

```
PANEL=PANEL1 (AREA=1.250000E+04 ERP. MAXIMUM=1.392054E+11)
      EQUIVALENT RADIATED POWER
      FREQUENCY      ERP      FRACTION      ERP DENSITY
      3.000000E+03  1.392054E+11  8.876539E-01  1.113644E+07
      4.000000E+03  1.761857E+10  1.123461E-01  1.409486E+06
```

```
PANEL=PANEL2 (AREA=1.750000E+04 ERP. MAXIMUM=1.590001E+10)
      EQUIVALENT RADIATED POWER
      FREQUENCY      ERP      FRACTION      ERP DENSITY
      3.000000E+03  1.590001E+10  8.812909E-01  9.085721E+05
      4.000000E+03  2.141717E+09  1.187091E-01  1.223838E+05
```

```
PANEL=TOTAL (AREA=6.000000E+04 ERP. MAXIMUM=1.590001E+10)
      EQUIVALENT RADIATED POWER
      FREQUENCY      ERP      FRACTION      ERP DENSITY
      3.000000E+03  3.042095E+11  8.876487E-01  5.070158E+06
      4.000000E+03  3.850434E+10  1.123513E-01  6.417390E+05
```

In a section for an individual panel such as the PANEL = PANEL1 section, the values are calculated as follows:

- For the FRACTION column, the values are the ERP values at a given frequency normalized with respect to the sum of the ERP values for the panel at all the frequencies listed in the output. For example, the fraction value for Panel 3 for a frequency of 3000 is  $1.491040 \times 10^{11} / (1.491040 \times 10^{11} + 1.874405 \times 10^{10}) = 8.883273 \times 10^{-1}$ .
- For the ERP DENSITY column, the values are the ERP values normalized with respect to the corresponding normal area values. For example, the ERP density at a frequency of 3000 for Panel 3 is  $1.491040 \times 10^{11} / 3.000000 \times 10^4 = 4.970134 \times 10^6$ .

In the "PANEL = TOTAL" section, the values are calculated as follows:

- For the ERP column, the values are the sums of the ERP values for each of the panels listed in the output at the corresponding frequencies. For example, the total ERP at a frequency of 3000 is  $1.491040 \times 10^{11} + 1.392054 \times 10^{11} + 1.590001 \times 10^{10} = 3.042095 \times 10^{11}$ .
- For the ERP DENSITY column, the values are the total ERP values normalized with respect to the total normal area. For example, the ERP density at a frequency of 3000 is  $3.042095 \times 10^{11} / 6.000000 \times 10^4 = 5.070158 \times 10^6$ .
- For the FRACTION column, the values are the total ERP values normalized with respect to the sum of the total ERP for each frequency listed in the output. For example, the fraction of the total ERP at a frequency of 3000 is  $3.042095 \times 10^{11} / (3.042095 \times 10^{11} + 3.850434 \times 10^{10}) = 8.876487 \times 10^{-1}$ .

## Limiting Output to Peak Responses

In a SOL 108 or SOL 111 structural, acoustic, or vibro-acoustic frequency response analysis, you can limit the output to frequencies where peak responses occur. This capability is called *peakout*.

To obtain these frequencies, you must specify the following:

- The subset of DOF that you want the software to filter to identify the peak responses.
- The filtering criteria.

Without this capability, two solves are required for you to perform an equivalent analysis. The first solve calculates the frequency responses. From the frequency responses, you manually identify the frequencies where peak responses occur. You then edit the input file so that the second solve occurs at those frequencies only.

Because peakout requires only one solve and no input file editing, it can save you time and reduce the chances for error.

When you use the peakout capability, the software does the following:

1. Calculates the frequency response for all the DOF identified by the output request where the PEAKOUT describer is specified.
2. Filters the frequency responses for each DOF listed on the PEAKOUT bulk entries. Each frequency response that satisfies the filtering criteria is a peak response. You specify the filtering criteria for each DOF on the PEAKOUT bulk entries.
3. For all the DOF identified by the output request where SOLUTION = PEAKOUT is specified, outputs frequency response results at only the frequencies where the peak responses occur.

When multiple output requests with SOLUTION = PEAKOUT specified are present, the software outputs the frequency response results for each output request at the frequencies of the peak responses. For an output request that does not have SOLUTION = PEAKOUT specified, the software outputs the frequency response results for that output request at all the frequencies if SOLUTION = ALL, or at a subset of the frequencies if SOLUTION = *setf*.

The filtering criteria that you can use to identify the peak responses, in the order that the software applies them, are as follows:

1. Frequency range of interest — The response at frequencies outside the range of interest is excluded from the peak response search.
2. Response cutoff — The response within the frequency range of interest, but below the response cutoff, is excluded from the peak response search.
3. Response proximity — Responses within the frequency range of interest and above the response cutoff, but in close proximity to other responses with higher magnitude, are excluded from the peak response search.
4. Requested number of peaks — Of the responses that remain after the software applies the first three criteria, the requested number of responses with the highest magnitude responses are the peak responses. The other responses are not considered peak responses unless they satisfy the outlier recovery criterion.  
If the number of responses that remain after applying the first three criteria is less than the requested number of peaks, only the responses that satisfy the first three criteria are peak responses.
5. Outlier recovery — The outlier recovery criterion identifies as peak responses those responses that were eliminated because of the requested number of peaks criterion, but are at much different frequencies than the peak responses that the software identified after it applied the first four criteria. Thus, it is possible to have more peak responses than the number you requested. For a detailed explanation of how the software applies this criterion, see the remarks for the FAR field in the PEAKOUT bulk entry.

## Implementing the peakout capability

To use the peakout capability, at a minimum, add the following to the input file for a frequency response analysis: a PEAKOUT case control command, an output request that supports the PEAKOUT describer, and a PEAKOUT bulk entry.

- **PEAKOUT case control command**  
Include PEAKOUT case control commands to trigger the software to generate a set of frequencies at which peak responses occur for the DOF that you specify. You can place a single PEAKOUT case control command above the subcases and in each subcase.  
When you place a PEAKOUT case control command above the subcases, it is applied to all the subcases. However, when you place a PEAKOUT case control command above the subcases and inside subcases, the PEAKOUT case control commands inside the subcases take precedence.  
A single PEAKOUT case control command can reference multiple PEAKOUT bulk entries that have the same identification number. When this occurs, the set of frequencies at which peak responses occur for a given DOF is the union of the peak responses identified by each PEAKOUT bulk entry separately for the DOF.  
For example, suppose that a PEAKOUT case control command references two PEAKOUT bulk entries. If 100 Hz, 145 Hz, 176 Hz, and 232 Hz satisfy the criteria for one PEAKOUT bulk entry, and 85 Hz, 133 Hz, 176 Hz, and 210 Hz satisfy the criteria for the other PEAKOUT bulk entry, the set of frequencies at which peak responses occur is 85 Hz, 100 Hz, 133 Hz, 145 Hz, 176 Hz, 210 Hz, and 232 Hz.
- **Output requests**  
On the output requests, specify the new SOLUTION = PEAKOUT describer. The output requests that support the peakout capability are as follows:
  - For structural frequency response analysis: DISPLACEMENT, VELOCITY, ACCELERATION, GPFORCE, MODCON (SOL 111 only).
  - For acoustic frequency response analysis: PRESSURE, MODCON (SOL 111 only)
  - For vibro-acoustic frequency response analysis: DISPLACEMENT, VELOCITY, ACCELERATION, GPFORCE, PRESSURE, MODCON (SOL 111 only), PANCON, GRDCON

**Note:**

The types of responses on the PEAKOUT bulk entries specify the types of frequency response results that the software uses when it identifies the peak responses. The output request specifies the type of frequency response results to output at the frequencies for the peak responses.

- **PEAKOUT bulk entries**  
Use one or more PEAKOUT bulk entries to specify the following:
  - The DOF for which the software retrieves frequency responses that it filters to identify the peak responses.

Use the GIDi and CIDi fields to specify these DOF.

- The type of frequency response results to filter for each DOF.

Use the TYPE field to specify the response type. Valid types are displacement, acceleration, velocity, and pressure results. You can include multiple response types on a single PEAKOUT bulk entry. You can also include the same DOF after multiple response types on a single PEAKOUT bulk entry. When you do so, the software creates the set of frequencies at which peak responses occur for the DOF from the union of the sets of frequencies at which peak responses occur for the DOF for each response type.

- The response cutoff for each DOF.

Use the CUTOFFi field to specify the response cutoff for each DOF. Use real input to specify a single cutoff value that the software applies across the frequency range of interest. Use integer input to reference a TABLED1, TABLED2, TABLED3, or TABLED4 bulk entry that defines the cutoff as a tabular function of frequency.

- The frequency range of interest.

Use the LFREQ and HFREQ fields to specify the frequency range of interest.

- The response proximity frequency difference.

Use the NEAR field to specify the minimum allowable frequency difference between adjacent peak responses.

- The requested number of peaks.

Use the NPEAK field to specify the desired number of peak responses.

- The outlier recovery criterion.

Use the FAR field to specify frequency bands about each response that is retained after the NPEAK criterion is applied. If a response is excluded based on the NPEAK criterion and it lies outside these frequency bands, it is included in the peak responses.

In the frequency response input file, on the FREQi bulk entry, define enough solution frequencies to ensure that you do not miss any peak responses. If you include an OFREQ case control command, the results for all frequencies specified by the OFREQ command that satisfy the PEAKOUT filtering criteria are output.

## 6.10 Examples of Frequency Response Analysis

This section provides several examples showing the input and output. These examples are:

Model	Frequency Response Bulk Entries	Output
bd05two	EIGRL, FREQ1, TABDMP1, RLOAD1, DAREA, TABLED1	X-Y plots (linear), printed results (SORT1, SORT2)
bd05bar	EIGRL, FREQ1, TABDMP1, DLOAD, RLOAD2, DAREA, DPHASE, TABLED1	X-Y plots (log)
bd05bkt	EIGRL, FREQ1, TABDMP1, RLOAD1, LSEQ, TABLED1, PLOAD4	X-Y plot (log)

## Two-DOF Model

Consider the two-DOF system shown in **Figure 6-9**. Modal frequency response (SOL 111) is run with a 20 N load applied to the primary mass (Grid 2) across a frequency range of 2 to 10 Hz with an excitation frequency increment of 0.05 Hz. Uniform modal damping of 5% critical damping is used. A portion of the input file is shown below.

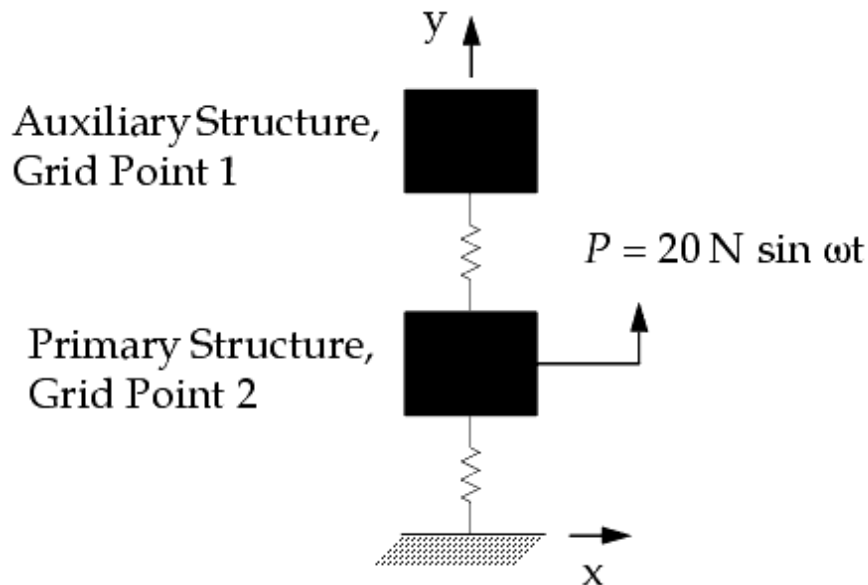


Figure 6-9. Two-DOF Model

```

$ FILE bd05two.dat
$
$ TWO-DOF SYSTEM
$ CHAPTER 5, FREQUENCY RESPONSE
$
TIME 5
SOL 111      $ MODAL FREQUENCY RESPONSE
CEND
TITLE = TWO-DOF SYSTEM
SUBTITLE = MODAL FREQUENCY RESPONSE
LABEL = 20 N FORCE APPLIED TO PRIMARY MASS
$
$ SPECIFY SPC

```

```


SPC = 996
$
$ SPECIFY MODAL EXTRACTION
METHOD = 10
$
$ SPECIFY DYNAMIC INPUT
DLOAD = 999
FREQ = 888
SDAMPING = 777
$
$ SELECT OUTPUT
SET 11 = 1,2
DISPLACEMENT(PHASE,PLOT) = 11
$
$ XYPLOTS
$
... X-Y plot commands ...
$
BEGIN BULK
$
$. . . . . 2 . . . . . 3 . . . . . 4 . . . . . 5 . . . . . 6 . . . . . 7 . . . . . 8 . . . . . 9 . . . . . 10 . . . . . $
$
$ ENTRIES FOR FREQUENCY RESPONSE
$
$ LOAD DEFINITION
$RLOAD1 SID      DAREA          TC
RLOAD1  999      997            901
$DAREA  SID      P1      C1      A1
DAREA   997      2       2       20.0
$TABLED1 TID                                +TABL1
$+TABL1 X1      Y1      X2      Y3      ETC.
TABLED1 901                                +TAB901
+TAB901 0.0      1.0      10.0      1.0      ENDT
$
$ ALTERNATE LOAD DEFINITION USING DLOAD
$DLOAD  SID      S      S1      RLOAD1
$DLOAD  999      1.0      1.0      998
$RLOAD1 SID      DAREA          TC
$RLOAD1 998      997            901
$
$ FREQUENCY RANGE 2-10 HZ
$FREQ1  SID      F1      DF      NDF
FREQ1   888      2.      0.05    160
$
$ MODAL DAMPING OF 5% CRITICAL
$TABDMP1 TID      TYPE                                +TABD1
$+TABD1 F1      G1      F2      G2      ETC.
TABDMP1 777      CRIT
+TABD7  0.      0.05    100.    0.05    ENDT
$
$ MODAL EXTRACTION
$EIGRL  SID      V1      V2      ND      MSGGLVL
EIGRL   10      -0.1    20.    0      0
$
... basic model ...
$
ENDDATA

```

Figure 6-10. Input File (Abridged) for the Two-DOF Example

**Table 6-54** shows the relationship between the case control commands and the bulk entries. The RLOAD1 entry references the DAREA and TABLED1 entries. Because there is only a single RLOAD1 entry, the DLOAD bulk entry is not required, but is included for demonstration purposes.

**Table 6-54. Relationship Between the Case Control Commands and Bulk Entries for the Two-DOF Model**

Case Control	Bulk Entries
METHOD	EIGRL
FREQUENCY	FREQ1
SDAMPING	TABDMP1
DLOAD	RLOAD1 

The RLOAD1 entry describes a sinusoidal load in the form:

$$\{P(f)\} = \{A[C(f) + iD(f)]e^{i\{\theta - 2\pi f\tau\}}\}$$

**Equation 6-29.**

where:

- A = 20.0 (entered on the DAREA entry)
- C = 1.0 for all frequencies entered on the TABLED1 entry
- D = 0.0 (field 7 of the RLOAD1 entry is blank)
- $\theta$  = 0.0 (field 5 of the RLOAD1 entry is blank)
- $\tau$  = 0.0 (field 4 of the RLOAD1 entry is blank)

Output can be printed in either real/imaginary or magnitude/phase format and in either SORT1 or SORT2 format. These formats are illustrated in **Figure 6-11**, **Figure 6-12**, and **Figure 6-13** showing a portion of their printed output.

```

POINT-ID =      1
                                C O M P L E X   D I S P L A C E M E N T   V E C T
O R
                                (REAL/IMAGINARY)
      FREQUENCY  TYPE      T1      T2      T3
R1    2.000000E+00  G      0.0      2.813052E-03  0.0      0.0
0.0    0.0          0.0      -2.107985E-04  0.0      0.0
0.0    0.0          0.0      0.0          0.0      0.0

```



```

    2.050000E+00    G    0.0          2.866642E-03    0.0          0.0
0.0                0.0          0.0          -2.229164E-04    0.0          0.0
0.0                0.0          0.0          2.923141E-03    0.0          0.0
    2.100000E+00    G    0.0          2.923141E-03    0.0          0.0
0.0                0.0          0.0          -2.358382E-04    0.0          0.0
0.0                0.0          0.0          2.982732E-03    0.0          0.0
    2.150000E+00    G    0.0          2.982732E-03    0.0          0.0
0.0                0.0          0.0          -2.496362E-04    0.0          0.0
0.0                0.0          0.0          3.045609E-03    0.0          0.0
    2.200000E+00    G    0.0          3.045609E-03    0.0          0.0
0.0                0.0          0.0          -2.643908E-04    0.0          0.0
0.0                0.0          0.0          0.0             0.0          0.0
    POINT-ID =      2
O R
                                C O M P L E X   D I S P L A C E M E N T   V E C T
                                (REAL/IMAGINARY)
    FREQUENCY    TYPE      T1      T2      T3
R1              R2              R3
    2.000000E+00    G    0.0          2.374954E-03    0.0          0.0
0.0              0.0          0.0          -1.129933E-04    0.0          0.0
0.0              0.0          0.0          2.397706E-03    0.0          0.0
    2.050000E+00    G    0.0          2.397706E-03    0.0          0.0
0.0              0.0          0.0          -1.180853E-04    0.0          0.0
0.0              0.0          0.0          2.421475E-03    0.0          0.0
    2.100000E+00    G    0.0          2.421475E-03    0.0          0.0
0.0              0.0          0.0          -1.234173E-04    0.0          0.0
0.0              0.0          0.0          2.446311E-03    0.0          0.0
    2.150000E+00    G    0.0          2.446311E-03    0.0          0.0
0.0              0.0          0.0          -1.290072E-04    0.0          0.0
0.0              0.0          0.0          2.472262E-03    0.0          0.0
    2.200000E+00    G    0.0          2.472262E-03    0.0          0.0
0.0              0.0          0.0          -1.348744E-04    0.0          0.0
0.0              0.0          0.0          0.0             0.0          0.0

```

**Figure 6-11. Real/Imaginary Output in SORT2 Format**

```

    POINT-ID =      1
O R
                                C O M P L E X   D I S P L A C E M E N T   V E C T
                                (MAGNITUDE/PHASE)
    FREQUENCY    TYPE      T1      T2      T3
R1              R2              R3
    2.000000E+00    G    0.0          2.820939E-03    0.0          0.0
0.0              0.0          0.0          355.7145          0.0          0.0
0.0              0.0          0.0          2.875296E-03    0.0          0.0
    2.050000E+00    G    0.0          2.875296E-03    0.0          0.0
0.0              0.0          0.0          355.5535          0.0          0.0
0.0              0.0          0.0          0.0             0.0          0.0

```

## 6. Frequency Response Analysis (SOLs 108 and 111)

```

2.100000E+00    G    0.0          2.932640E-03    0.0          0.0
0.0            0.0          0.0          355.3874          0.0          0.0
0.0            0.0          0.0          2.993161E-03    0.0          0.0
2.150000E+00    G    0.0          2.993161E-03    0.0          0.0
0.0            0.0          0.0          355.2159          0.0          0.0
0.0            0.0          0.0          3.057064E-03    0.0          0.0
2.200000E+00    G    0.0          3.057064E-03    0.0          0.0
0.0            0.0          0.0          355.0386          0.0          0.0
0.0            0.0          0.0          0.0              0.0          0.0
POINT-ID =      2
O R
C O M P L E X    D I S P L A C E M E N T    V E C T
(MAGNITUDE/PHASE)
FREQUENCY    TYPE    T1    T2    T3
R1    R2    R3
2.000000E+00    G    0.0    2.377640E-03    0.0    0.0
0.0            0.0    0.0    357.2761          0.0    0.0
0.0            0.0    0.0    2.400612E-03    0.0    0.0
2.050000E+00    G    0.0    2.400612E-03    0.0    0.0
0.0            0.0    0.0    357.1805          0.0    0.0
0.0            0.0    0.0    2.424619E-03    0.0    0.0
2.100000E+00    G    0.0    2.424619E-03    0.0    0.0
0.0            0.0    0.0    357.0823          0.0    0.0
0.0            0.0    0.0    2.449710E-03    0.0    0.0
2.150000E+00    G    0.0    2.449710E-03    0.0    0.0
0.0            0.0    0.0    356.9813          0.0    0.0
0.0            0.0    0.0    2.475939E-03    0.0    0.0
2.200000E+00    G    0.0    2.475939E-03    0.0    0.0
0.0            0.0    0.0    356.8773          0.0    0.0
0.0            0.0    0.0    0.0              0.0    0.0

```

**Figure 6-12. Magnitude/Phase Output in SORT2 Format**

```

FREQUENCY = 2.000000E+00
O R
C O M P L E X    D I S P L A C E M E N T    V E C T
(REAL/IMAGINARY)
POINT ID.    TYPE    T1    T2    T3
R1    R2    R3
1    G    0.0    2.813051E-03    0.0    0.0
0.0            0.0    -2.107985E-04    0.0    0.0
0.0            0.0    2.374954E-03    0.0    0.0
2    G    0.0    2.374954E-03    0.0    0.0
0.0            0.0    -1.129933E-04    0.0    0.0
0.0            0.00
FREQUENCY = 2.050000E+00
O R
C O M P L E X    D I S P L A C E M E N T    V E C T

```

```

                                (REAL/IMAGINARY)
R1   POINT ID.  TYPE          T1          T2          T3
      1      G      0.0      2.866640E-03  0.0      0.0
0.0      0.0      0.0      -2.229163E-04  0.0      0.0
0.0      0.0      0.0      0.0      0.0      0.0
      2      G      0.0      2.397706E-03  0.0      0.0
0.0      0.0      0.0      -1.180853E-04  0.0      0.0
0.0      0.00
FREQUENCY = 2.100000E+00
                                C O M P L E X   D I S P L A C E M E N T   V E C T
O R
                                (REAL/IMAGINARY)
R1   POINT ID.  TYPE          T1          T2          T3
      1      G      0.0      2.923141E-03  0.0      0.0
0.0      0.0      0.0      -2.358381E-04  0.0      0.0
0.0      0.0      0.0      0.0      0.0      0.0
      2      G      0.0      2.421475E-03  0.0      0.0
0.0      0.0      0.0      -1.234173E-04  0.0      0.0
0.0      0.00
FREQUENCY = 2.150000E+00
                                C O M P L E X   D I S P L A C E M E N T   V E C T
O R
                                (REAL/IMAGINARY)
R1   POINT ID.  TYPE          T1          T2          T3
      1      G      0.0      2.982731E-03  0.0      0.0
0.0      0.0      0.0      -2.496362E-04  0.0      0.0
0.0      0.0      0.0      0.0      0.0      0.0
      2      G      0.0      2.446311E-03  0.0      0.0
0.0      0.0      0.0      -1.290072E-04  0.0      0.0
0.0      0.00
FREQUENCY = 2.200000E+00
                                C O M P L E X   D I S P L A C E M E N T   V E C T
O R
                                (REAL/IMAGINARY)
R1   POINT ID.  TYPE          T1          T2          T3
      1      G      0.0      3.045608E-03  0.0      0.0
0.0      0.0      0.0      -2.643907E-04  0.0      0.0
0.0      0.0      0.0      0.0      0.0      0.0
      2      G      0.0      2.472263E-03  0.0      0.0
0.0      0.0      0.0      -1.348744E-04  0.0      0.0
0.0      0.0

```

**Figure 6-13. Real/Imaginary Output in SORT1 Format**

**Figure 6-14** shows the plots of the resulting displacement magnitudes for Grids 1 and 2. Note that the response for Grid 1 is nearly an order of magnitude larger than that of Grid 2. This large difference in response magnitudes is characteristic of dynamic absorbers (also called tuned mass dampers), in which

an auxiliary structure (i.e., the small mass and stiffness) is attached to the primary structure in order to decrease the dynamic response of the primary structure. If this same model is rerun without the auxiliary structure, the response of the primary structure (Grid 2) at 5.03 Hz is twice what it is with the auxiliary structure attached, as shown in **Figure 6-15**.

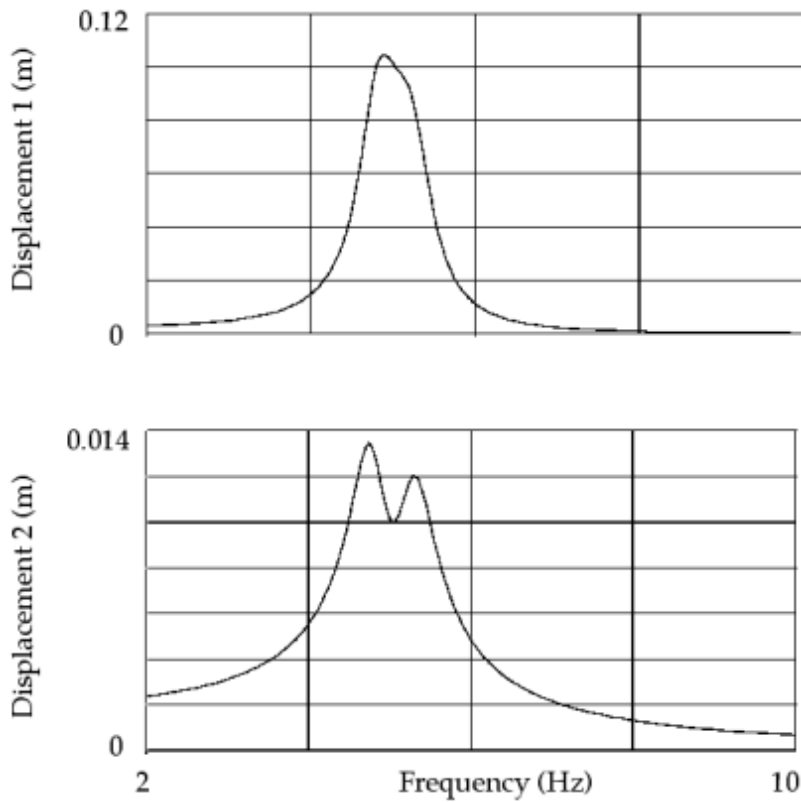


Figure 6-14. Displacement Response Magnitudes With the Auxiliary Structure

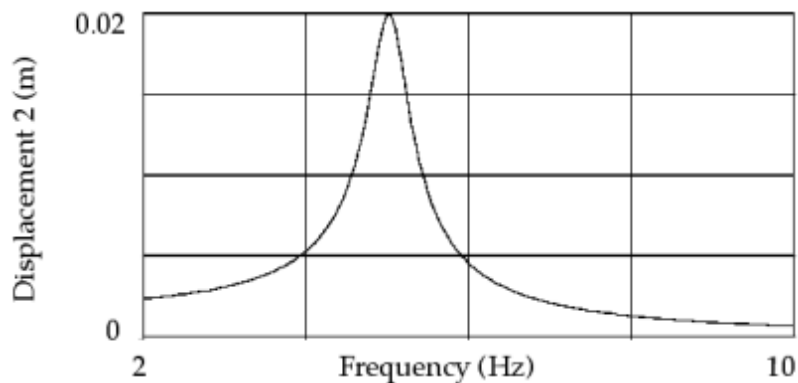


Figure 6-15. Displacement Response Magnitude Without the Auxiliary Structure

## Cantilever Beam Model

Consider the cantilever beam shown in **Figure 6-16**. This model is a planar model of the cantilever beam introduced in **"Real Eigenvalue Analysis"** with unrestrained DOFs in the T2 and R3 directions. Two

loads are applied: one at Grid 6 and the other at Grid 11. The loads have the frequency variation shown in **Figure 6-17**. The loads in the figure are indicated with a heavy line in order to emphasize their values. The load at Grid 6 has a 45 degree phase lead, and the load at Grid 11 is scaled to be twice that of the load at Grid 6. Modal frequency response is run across a frequency range of 0 to 20 Hz. Modal damping is used with 2% critical damping between 0 and 10 Hz and 5% critical damping above 10 Hz. Modes to 500 Hz are computed using the Lanczos method.

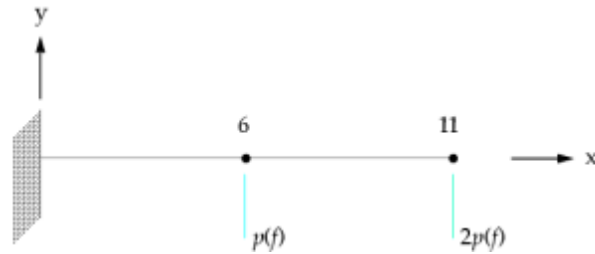


Figure 6-16. Cantilever Beam Model with Applied Loads

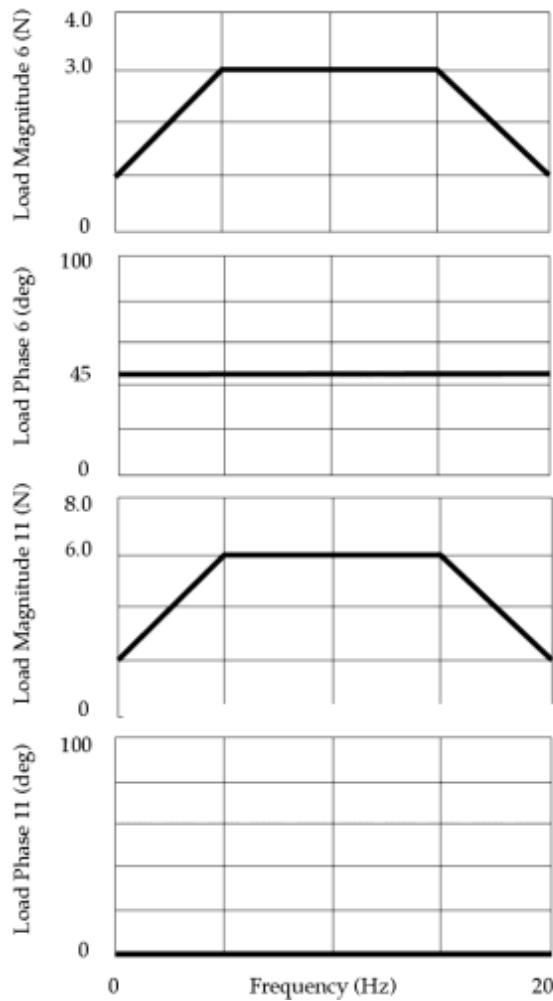


Figure 6-17. Applied Loads

The abridged input file is shown below. The output quantities, as defined in the case control section, are the applied loads (OLOAD) for Grids 6 and 11, physical displacements (DISPLACEMENT) for Grids 6 and 11, solution set displacements (SDISPLACEMENT) for Modes 1 and 2, and element forces (ELFORCE) for Element 6. These output quantities are plotted rather than printed.

```

$ FILE bd05bar.dat
$
$ CANTILEVER BEAM MODEL
$ CHAPTER 5, FREQUENCY RESPONSE
$
SOL 111      $ MODAL FREQUENCY RESPONSE
TIME 10
CEND
TITLE = CANTILEVER BEAM
SUBTITLE = MODAL FREQUENCY RESPONSE
$
SPC = 21
$
DLOAD = 22
FREQ = 27
SDAMPING = 20
$
METHOD = 10
$
SET 15 = 6,11
OLOAD(PHASE,PLOT) = 15
$
$ PHYSICAL OUTPUT REQUEST
SET 11 = 6,11
DISPLACEMENT(PHASE,PLOT) = 11
$
$ MODAL SOLUTION SET OUTPUT
SET 12 = 1,2
SDISP(PHASE,PLOT) = 12
$
$ ELEMENT FORCE OUTPUT
SET 13 = 6
ELFORCE(PHASE,PLOT) = 13
$
$ XYPLOTS
$
... X-Y plot commands ...
$
BEGIN BULK
$
$. . . . .2. . . . .3. . . . .4. . . . .5. . . . .6. . . . .7. . . . .8. . . . .9. . . . .10. . . . . $
$
EIGRL  10      -0.1    500.      0
$
FREQ1  27      0.0     0.05    400
$
TABDMP1 20      CRIT
+TABD1  0.0     0.02    10.0    0.02    10.01   0.05    25.0    0.05    +TABD1
+TABD2  ENDT
$
$ DYNAMIC LOADING
$DLOAD  SID      S      S1      L1      S2      L2
DLOAD  22      1.0    1.0    231    1.0    232
$RLOAD2 SID      DAREA  DELAY  DPHASE TB      TP

```

```

RLOAD2  231      241                261      25
RLOAD2  232      242                25
$DAREA  SID      P1      C1      A1
DAREA   241      6       2       1.0
DAREA   242      11      2       2.0
$DPHASE SID      P1      C1      TH1
DPHASE  261      6       2       45.
$TABLED1 TID                                +TABL1
$+TABL1  X1      Y1      X2      Y2      ETC.
TABLED1  25
+TABL1   0.      1.      5.0     3.      15.0   3.0   20.0   1.      +TABL1
+TABL2   25.0    1.      ENDT
$
... basic model ...
$
ENDDATA

```

**Figure 6-18. Input File (Abridged) for the Beam Example**

**Table 6-56** shows the relationship between the case control commands and the bulk entries. The DLOAD bulk entry references two RLOAD2 entries, each of which references a separate DAREA entry and a common TABLED1 entry. The RLOAD2 entry for Grid 6 also references a DPHASE entry that defines the 45 degree phase lead.

**Table 6-56. Relationship Between Case Control Commands and Bulk Entries for the Beam Model**

Case Control	Bulk Entries
METHOD	EIGRL
FREQUENCY	FREQ1
SDAMPING	TABDMP1
DLOAD	<p>DLOAD</p> <pre> graph TD     DLOAD --&gt; RLOAD2_231[RLOAD2 231]     DLOAD --&gt; RLOAD2_232[RLOAD2 232]     RLOAD2_231 --&gt; DAREA_231[DAREA]     RLOAD2_231 --&gt; DPHASE[DPHASE]     RLOAD2_231 --&gt; TABLED1[TABLED1]     RLOAD2_232 --&gt; DAREA_232[DAREA] </pre>

The RLOAD2 entry describes a sinusoidal load in the form:

$$\{P(f)\} = \{AB(f)e^{i\{\phi(f) + \theta - 2\pi f\tau\}}\}$$

Equation 6-30.

where:

- A = 1.0 for Grid 6 and 2.0 for Grid 11 (entered on the DAREA entry)  
 B = function defined on the TABLED1 entry  
 $\phi$  = 0.0 (field 7 of the RLOAD2 entry is blank)  
 $\theta$  = phase lead of 45 degrees for Grid 6 (entered on the DPHASE entry)  
 $\tau$  = 0.0 (field 4 of the RLOAD2 entry is blank)

Logarithmic plots of the output are shown in the following figures. **Figure 6-19** shows the magnitude of the displacements for Grids 6 and 11. **Figure 6-20** shows the magnitude of the modal displacements for Modes 1 and 2. **Figure 6-21** shows the magnitude of the bending moment at end A in plane 1 for Element 6. Logarithmic plots are especially useful for displaying frequency response results since there can be several orders of magnitude between the maximum and minimum response values.

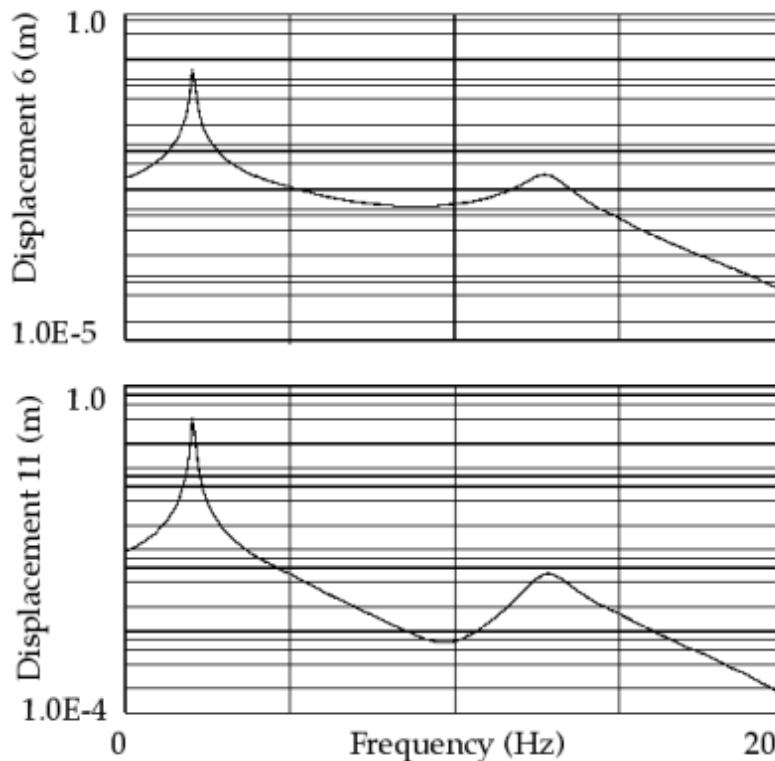


Figure 6-19. Displacement Magnitude (Log)



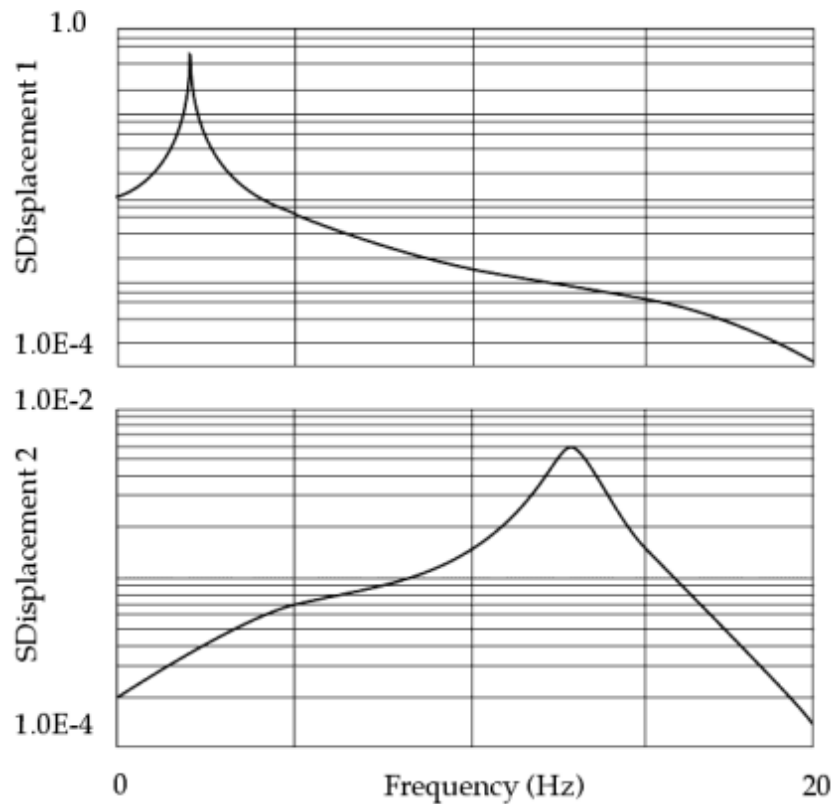


Figure 6-20. Modal Displacement Magnitude (Log)

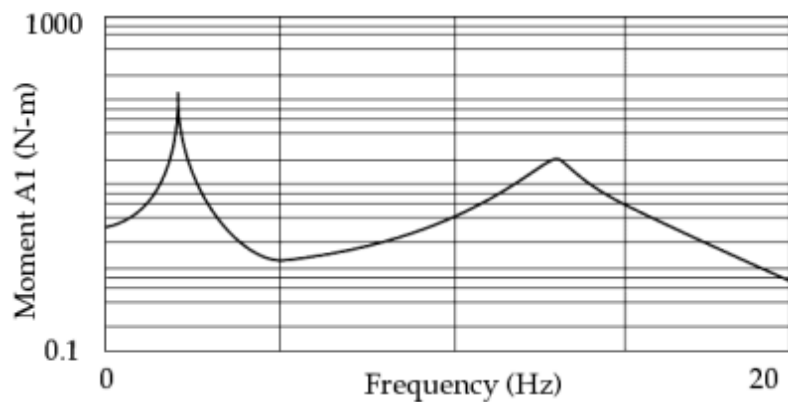


Figure 6-21. Bending Moment Magnitude at End A, Plane 1 (Log)

## Bracket Model

Consider the bracket model shown in [Figure 6-22](#). An oscillating pressure load of 3 psi is applied to the elements on the top face in the z-direction. The model is constrained at its base. Modal frequency response is run from 0 to 100 Hz with a frequency step size of 0.2 Hz. Eigenvalues are computed up to 1000 Hz using the Lanczos method. Modal damping is applied as 2% critical damping for all modes.

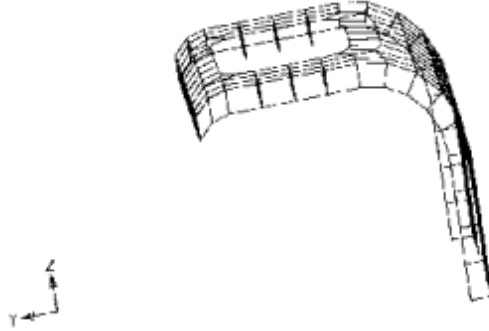


Figure 6-22. Bracket Model

Figure 6-23 shows the abridged input file. The LSEQ entry is used to apply the pressure loads (PLOAD4 entries). The LSEQ and RLOAD1 entries reference a common DAREA ID (999) and that there is no explicit DAREA entry. Table 6-58 shows the relationship between the case control commands and the bulk entries.

```

$ FILE bd05bkt.dat
$
$ BRACKET MODEL
$ CHAPTER 5, FREQUENCY RESPONSE
$
SOL 111      $ MODAL FREQUENCY RESPONSE
TIME 100
CEND
TITLE = BRACKET MODEL
SUBTITLE = MODAL FREQUENCY RESPONSE ANALYSIS
$
SPC = 1
$
METHOD = 777
DLOAD = 2
LOADSET = 3
SDAMPING = 4
FREQUENCY = 5
$
$ OUTPUT REQUEST
SET 123 = 999
DISPLACEMENT (PHASE, PLOT)=123
$
$ XYPLOTS
$
... X-Y plot commands ...
$
BEGIN BULK
$
$. . . . .2. . . . .3. . . . .4. . . . .5. . . . .6. . . . .7. . . . .8. . . . .9. . . . .10. . . . .
$
$ NORMAL MODES TO 1000 HZ
$EIGRL SID V1 V2
EIGRL 777 -0.1 1000.
$
$ EXCITATION FREQUENCY DEFINITION 0 TO 100 HZ
$FREQ1 SID F1 DF NDF

```

```

FREQ1  5      0.0    0.2    500
$
$ MODAL DAMPING OF 2% CRITICAL FOR ALL MODES
$TABDMP1 TID      TYPE                                +TABD1
$+TABD1 F1      G1      F2      G2      ETC.
TABDMP1 4        CRIT                                +TABD1
+TABD1  0.0     0.02   1000.0  0.02   ENDT
$
$ LOAD DEFINITION
$
$RLOAD1 SID      DAREA  DELAY  DPHASE TC      TD
RLOAD1  2        999
$
$LSEQ  SID      DAREA  LID      TID
LSEQ   3        999    1
$
$TABLED1 TID                                +TABL1
$+TABL1 X1      Y1      X2      Y2      ETC.
TABLED1 22                                +TABL1
+TABL1  0.0     1.0    1000.0  1.0    ENDT
$
$ PRESURE LOAD OF 3 PSI PER ELEMENT
$PLOAD4 SID      EID      P1
PLOAD4  1        171     -3.
PLOAD4  1        172     -3.
PLOAD4  1        160     -3.
etc.
$
... basic model ...
$
ENDDATA

```

Figure 6-23. Abridged Input File for the Bracket Model

Table 6-58. Relationship Between Case Control Commands and Bulk Entries for the Bracket Model

Case Control	Bulk Entries
METHOD	EIGRL
FREQUENCY	FREQ1
SDAMPING	TABDMP1
LOADSET	 LSEQ → PLOAD4 LSEQ → (DAREA ID)
DLOAD	 RLOAD1 → (DAREA ID) RLOAD1 → TABLED1

Figure 6-24 shows a logarithmic plot of the z-displacement magnitude of Grid 999, which is the concentrated mass at the center of the cutout.

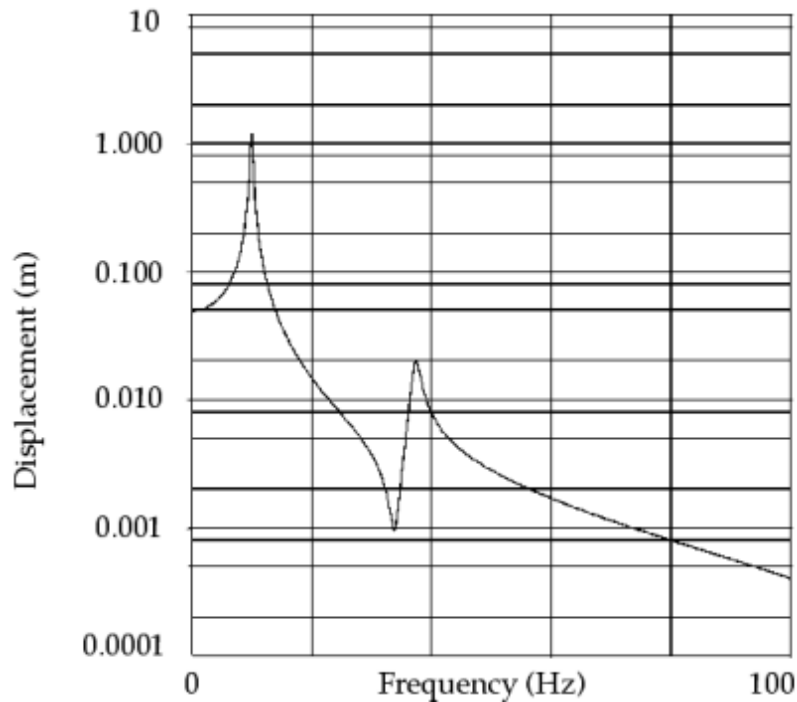


Figure 6-24. Displacement Magnitude (Log)

## 6.11 Modal Frequency Response Solution Methods

Several solution methods are available for modal frequency response analysis (SOL 111). By specifying a solution method that is well suited to your problem, you can improve system performance. To specify the solution method, specify system cell 462 as follows:

- Specifying `SYSTEM(462) = 0` (the default option) selects the original FRRD1 method. This method is a good choice when the number of modes is small or the model is undamped.
- Specifying `SYSTEM(462) = 1` selects the in-core direct FRRD1 method introduced in NX Nastran 9. This method is suitable for any damped model, and it scales well in SMP. This method is particularly beneficial if a large amount of RAM, and a large number of cores is available. Some solution times have been reduced almost 50% with this method.
- Specifying `SYSTEM(462) = 1` when `ITER = YES` is also specified selects the in-core iterative FRRD1 method introduced in NX Nastran 6. This method is a good choice when the number of modes is large and the model is damped.
- Specifying `SYSTEM(462) = 2` selects the FRRDRU method introduced in NX Nastran 7, and then enhanced in NX Nastran 8. This method is a good choice for models that contain significant structural damping, but relatively few viscous dampers. This method is recommended when the number of modes is large (more than 3000), and the desired frequency range is also large (more than 20

frequencies). This method cannot handle multiple eigenvalues (either repeated roots or rigid body modes). If multiple eigenvalues occur, the method defaults to the original FRRD1 method.

- Specifying `SYSTEM(462) = 4` selects the enhanced FRRDRU method introduced in NX Nastran 8.5. This method is a good choice for models that contain significant structural damping, but relatively few viscous dampers. This method is recommended when the number of modes is large (more than 3000), and the desired frequency range is also large (more than 20 frequencies). This method allows for multiple eigenvalues. If the model contains modal viscous damping, this method automatically converts it to modal structural damping by setting `KDAMP = -1`.

Both FRRDRU methods are available in an optimization solution (SOL 200) that includes a modal frequency response subcase (ANALYSIS = MFREQ).

Both FRRDRU methods can be used in combination with RDMODES to obtain the best performance on large models.

### Performance Example

A modal frequency response analysis (SOL 111) is performed on a model of an automobile body. The model consists of:

- 1,096,000 grid points.
- 1,080,000 elements.
- 5,912,000 degree-of-freedom.

2,500 modes are calculated and used to compute 400 frequency responses. Shared memory parallel (SMP) (Parallel = 8) is used as the parallel option. The results of the analysis are as follows:

Method	Original FRRD1 Method (default)	Original FRRDRU Method (nastran system(462)=2)
Elapsed time (min:sec)	75:08	20:18
I/O (GB)	270.6	111.6
CPU (sec) <sup>(1)</sup>	32980	3287
<sup>(1)</sup> Because SMP is used, times are approximate.		

## 6.12 Direct-Input, Frequency-Dependent Components

In a SOL 108 or 111 frequency response analysis, you can define frequency-dependent components by direct input of tabular dynamic stiffness versus frequency or dynamic flexibility versus frequency data. With this capability, you can use empirically obtained dynamic stiffness or dynamic flexibility data to represent a component of an assembly.

The dynamic stiffness represents the cumulative effect of inertia, damping, and stiffness within the component.

For example, the dynamic stiffness for an FE model is given as:

$$[K_{dynamic}] = -\omega^2 [M] + i\omega [B] + (1 + iG)[K] + i \sum G_E [K_E]$$

where:

$\omega$	Frequency in radians per unit time
$[M]$	Global mass matrix
$[B]$	Global viscous damping matrix
$G$	Overall structural damping coefficient (PARAM,G)
$[K]$	Global stiffness matrix
$G_E$	Structural damping coefficient of the elements (GE on material, property, or element entries)
$[K_E]$	Stiffness matrix of the elements

The dynamic flexibility matrix is the inverse of the dynamic stiffness matrix.

The dynamic stiffness is expressed in terms of force per unit displacement (moment per unit angular displacement). However, you can also enter dynamic stiffness data as force per unit velocity or force per unit acceleration (moment per unit angular velocity or moment per unit angular acceleration).

The dynamic flexibility is expressed in terms of displacement per unit force (angular displacement per unit moment). However, you can also enter dynamic flexibility data as velocity per unit force or acceleration per unit force (angular velocity per unit moment or angular acceleration per unit moment).

The values you enter for dynamic stiffness or dynamic flexibility must align with the physical coordinates and directions of the attachment grid points.

If you enter dynamic stiffness data in terms of velocity or acceleration, the software multiplies velocity data by  $i\omega$  and acceleration data by  $-\omega^2$  to convert the data to force (moment) per unit displacement (angular displacement).

If you enter dynamic flexibility data in terms of velocity or acceleration, the software divides velocity data by  $i\omega$  and acceleration data by  $-\omega^2$  to convert the data to force (moment) per unit displacement (angular displacement). The software then inverts the dynamic flexibility matrix to obtain the dynamic stiffness matrix.

Because the unit basis of the dynamic stiffness or flexibility data may differ from the unit basis of the FE model, you can define displacement and force scaling factors. The software uses the scaling factors to convert the dynamic stiffness or flexibility data to the units of the FE model.

### Direct frequency response analysis

For a SOL 108 direct frequency response analysis, the software:

1. Adds the dynamic stiffness of the component to the dynamic stiffness of the FE model.
2. Performs the direct frequency response analysis of the combined system.

### Modal frequency response analysis

For a SOL 111 modal frequency response analysis, the software:

1. Performs a real eigenvalue solution on the FE portion of the model only. During this solve, the software ignores the frequency-dependent components and automatically calculates residual vectors for the interface DOF of the attachment grid points.
2. Uses the results of the real eigenvalue solution to transform the dynamic stiffness or flexibility for the component from physical coordinates to modal coordinates.
3. Adds the transformed dynamic stiffness of the component to the modal dynamic stiffness of the FE model.
4. Performs the modal frequency response analysis of the combined system.

### User interface requirements

You use FRFSTIF or FRFFLEX bulk entries to define the dynamic stiffness or dynamic flexibility for the frequency-dependent component. The dynamic stiffness or dynamic flexibility matrix you define should represent the dynamic stiffness or dynamic flexibility expressed at the interface DOF of the attachment grid points. The FRFSTIF and FRFFLEX bulk entries reference TABLED6 bulk entries, which contain the tabular data of dynamic stiffness or dynamic flexibility versus frequency in cycles per unit time. You use the new FRFIN case control command to trigger the addition of the dynamic stiffness for the direct-input, frequency-dependent component to the dynamic stiffness of the FE model. When using the FRFFLEX bulk entry, you can use the FLEXINV parameter to specify the method that the software uses to invert the flexibility data into stiffness data.

## 6.13 Output Transformation Matrices

During results recovery in a SOL 108 or 111 frequency response analysis, the software can calculate results for grid and scalar points that are not part of the FE model.

In a structural frequency response analysis, the results for these points can be linear combinations of:

- Displacement results from the FE solve.
- Time derivatives of the displacement results from the FE solve.
- Forces that the software calculates from an FRFSTIF or FRFFLEX bulk entry and the displacement results from the FE solve.

For an acoustic frequency response analysis, the results for these points can be linear combinations of:

- Pressure results from the FE solve.

To get these additional results, you use tabular data to define a frequency-dependent output transformation matrix (OTM) that the software uses to combine the results of the FE solve. The OTM relationship can be expressed mathematically as:

$$\{y(\omega)\} = [OTM(\omega)]\{x(\omega)\}$$

where:

$\{x(\omega)\}$	N x 1 input vector of values that are derived from the results of the FE solve
$[OTM(\omega)]$	M x N output transformation matrix
$\{y(\omega)\}$	M x 1 output vector that the software calculates during results recovery

Typically, the tabular data that you use to define the OTM is obtained through physical testing.

For a SOL 111 modal frequency response analysis, the software performs the OTM calculations after it transforms the results of the FE solve to physical coordinates.

As an example of how you might use this capability, consider performing a frequency response analysis on a model that includes a direct-input, frequency-dependent component that is defined by an FRFSTIF or FRFFLEX bulk entry. When you perform such an analysis, the only results that the software calculates for the component during the FE solve are the responses at the DOF that connect the component to the FE model.

To obtain results for other points on the component, you can use the FRFOTM bulk entry to define an OTM that relates the results at DOF in the FE model to results at the other points on the component. During results recovery, the software uses the OTM and the results of the FE solve to calculate results for these points. You can then assign a physical meaning to the results and use them in post-processing.

### OTM in structural frequency response analysis

In a structural frequency response analysis, you can specify displacements, velocities, and accelerations for any DOF in the FE model to populate  $\{x(\omega)\}$ . If you are using the OTM capability in conjunction with a direct-input, frequency-dependent component that is defined by an FRFSTIF or FRFFLEX bulk entry, you



can also populate  $\{x(\omega)\}$  with loads for DOF that connect the direct-input, frequency-dependent component to the FE model.

The software calculates velocities and accelerations from the displacements. Because the motion is harmonic, the velocity and acceleration are related to the displacement as follows:

- For velocity:

$$\dot{u} = i\omega u$$

- For acceleration:

$$\ddot{u} = -\omega^2 u$$

where:

$u$	Displacement result from the FE solve
$\omega$	Frequency in radians per unit time

Because the unit of length for the OTM entries may differ from the unit of length used in the FE model, you can define a length scaling factor (LSCALE field on the FRFOTM bulk entry) that the software uses to convert the displacement results from the FE model unit of length to the OTM unit of length.

For example, suppose unit of length in the FE model is mm and the unit of length in the OTM is m. For this case, you enter 1000.0 in the LSCALE field of the FRFOTM bulk entry.

When  $\{x(\omega)\}$  includes load entries, the software calculates the loads from the FRFSTIF matrix (or the inverse of the FRFFLEX matrix) and the displacements as follows:

$$\{f(\omega)\} = [FRFSTIF(\omega)]\{u(\omega)\}$$

**Note:**

The inverse of an FRFFLEX matrix is equivalent to an FRFSTIF matrix. Thus, the remainder of this topic refers to the FRFSTIF matrix only.

Because the unit of length for the FRFSTIF matrix may differ from the unit of length used in the FE model, you can define a length scaling factor (LSCALE field on the FRFSTIF bulk entry) that the software uses to convert the displacement results from the FE model unit of length to the FRFSTIF unit of length.

For example, suppose unit of length in the FE model is mm and the unit of length in the FRFSTIFF is inches. For this case, you enter 25.4 in the LSCALE field of the FRFSTIF bulk entry.

In addition, the unit of force obtained from:

$$\{f(\omega)\} = [FRFSTIF(\omega)]\{u(\omega)\}$$

may not be the same as the unit of force used in the OTM calculation. Thus, you can define a force scaling factor (FSCALE field on the FRFOTM bulk entry) that the software uses to convert the force units of  $\{f(\omega)\}$  to the force units of the OTM.

For example, suppose that the OTM needs the unit of force for  $\{f(\omega)\}$  to be N and the unit of force in the FRFSTIFF is lbf. For this case, you enter 4.448 in the FSCALE field of the FRFOTM bulk entry.

**Note:**

When a model includes a direct-input, frequency-dependent component that is defined by an FRFSTIF bulk entry, during the FE solve, the software uses the value in the FSCALE field of the FRFSTIF bulk entry to convert the force units of the FRFSTIF entries to the force unit of the FE model.

## OTM in acoustic frequency response analysis

In an acoustic frequency response analysis, you can specify pressure for any DOF in the FE model to populate  $\{x(\omega)\}$ .

Because the unit of pressure for the OTM entries may differ from the unit of pressure used in the FE model, you can define a pressure scaling factor (PSCALE field on the FRFOTM bulk entry) that the software uses to convert the pressure results from the FE model unit of pressure to the OTM unit of pressure.

For example, suppose the unit of pressure in the FE model is Pa and the unit of pressure in the OTM is psi. For this case, you enter 6895.0 in the PSCALE field of the FRFOTM bulk entry.

## Interpreting OTM results

By default, the software does not assign physical meaning to the values in  $\{y(\omega)\}$ .

For example, if the values in the OTM represent stiffness, and the values in  $\{x(\omega)\}$  represent displacement, you might assume that the software interprets the result of the OTM calculation to be force, but it does not.

You can use the FRFOMAP bulk entry to assign a physical meaning to the values in  $\{y(\omega)\}$ .

- In a structural frequency response analysis, you can designate values in  $\{y(\omega)\}$  to be displacements, velocities, accelerations, or forces. If an entry in  $\{y(\omega)\}$  is associated with a rotational DOF, the software automatically uses the rotational analog. That is, angular displacement for displacement, moment for force, and so on.

- In an acoustic frequency response analysis, you can designate values in  $\{y(\omega)\}$  to be pressures only.

### FRFOTM bulk entry

To define an OTM, use the FRFOTM bulk entry as follows:

- Reference TABLED6 bulk entries that contain the tabular data that defines how each entry in the OTM varies as a function of frequency in cycles per unit time.  
The TIDij fields on the FRFOTM bulk entry reference the TABLED6 bulk entries.
- Specify the DOF and type combinations that comprise  $\{x(\omega)\}$ .  
The IGRIDj and ICOMPj fields on the FRFOTM bulk entry specify the DOF. The TYPEj field on the FRFOTM bulk entry specifies whether to use the displacement, velocity, acceleration, or load for the DOF.
- Specify the DOF with which the values in  $\{y(\omega)\}$  are associated.  
The OGRIDi and OCOMPi fields on the FRFOTM bulk entry specify these DOF.
- Define the scaling factors that the software uses to convert the units for the displacement, force, pressure, and acoustic source strength from those used in the OTM to those used in the FE model as follows:

$$L_{FEMODEL} = LSCALE \cdot L_{OTM}$$

$$F_{FEMODEL} = FSCALE \cdot F_{OTM}$$

$$P_{FEMODEL} = PSCALE \cdot P_{OTM}$$

$$Q_{FEMODEL} = QSCALE \cdot Q_{OTM}$$

### FRFOMAP bulk entry

To assign physical meaning to values in  $\{y(\omega)\}$ , use the FRFOMAP bulk entry as follows:

- Specify the physical meaning in the YTYPEi field.
- For each YTYPEi, specify all the YGRIDj and YCOMPj combinations to which to assign that physical meaning type.

### FRFIN case control command

To trigger the software to use the FRFOTM bulk entry during results recovery, include the FRFIN case control command in your input file.

### Example problem using an OTM

Suppose you perform a structural frequency response analysis that includes a direct-input, frequency-dependent component defined by an FRFSTIF (or FRFFLEX) bulk entry and you want the software to calculate the following results:

$$y_1 = 2.3u_1 + 3.0\dot{u}_1 - 4.5u_2 + 1.6f_5$$

$$y_2 = 3.7u_1 + 6.0\dot{u}_2 + 8.3\ddot{u}_4$$

$$y_3 = 4.1f_2 + 6.1u_3 - 2.8\ddot{u}_5$$

where:

$u_1, u_2, \dots, u_5$  Displacements calculated during the FE solve

$f_2, f_5$  Loads calculated during results recovery from the FRFSTIF (or FRFFLEX) matrix and displacements calculated during the FE solve

The following table maps each displacement to a grid point and displacement component combination.

Displacement	Grid point	Component
$u_1$	100	3
$u_2^{(1)}$	100	2
$u_3$	108	1
$u_4$	105	3
$u_5^{(1)}$	112	1

<sup>(1)</sup> DOF that connect the direct-entry, frequency-dependent component to the FE model

The following table maps each result to a grid point and displacement component combination.

Result	Grid point	Component
$y_1$	104	2
$y_2$	121	3
$y_3$	108	2

**Note:**

Any grid point and displacement component combination that is used as an input DOF on an FRFOTM bulk entry cannot be used as an output DOF on any FRFOTM bulk entry in the input file. In addition, the output from one OTM cannot be the input to another OTM.

Assuming the units of the FE model match the units of the OTM matrix, the following table outlines how you specify the FRFOTM bulk entries. In the actual coding of the FRFOTM bulk entries, the cells that contain real values like 2.3, 3.7, and so on, actually contain integer identification numbers of TABLED6 bulk entries that contain the numerical data.

FRFOTM	10	0							
FRFOTM	10	100	3	DISP					
	104	2	2.3	121	3	3.7			
FRFOTM	10	100	3	VELO					
	104	2	3.0						
FRFOTM	10	100	2	DISP					
	104	2	-4.5						
FRFOTM	10	100	2	VELO					
	121	3	6.0						
FRFOTM	10	100	2	LOAD					
	108	2	4.1						
FRFOTM	10	108	2	DISP					
	108	2	6.3						
FRFOTM	10	105	3	ACCE					
	121	3	8.3						
FRFOTM	10	112	1	ACCE					
	108	2	-2.8						
FRFOTM	10	112	1	LOAD					
	104	2	1.6						

Suppose that you want to define  $y_1$  as a displacement,  $y_2$  as an acceleration, and  $y_3$  as a load. To do so, you can define an FRFOMAP bulk entry as follows:

FRFOMAP	10								
	DISP	104	2						
	ACCE	121	3						
	LOAD	108	2						

## Visualizing OTM results

You can use line, shell, and solid visualization elements to model geometry without adding physical effects such as mass, stiffness, and damping to the FE model.

The visualization elements are as follows:

Visualization Element	Description
PLOTEL	Line visualization element with two grid points
PLOTEL3	Triangular visualization element with three grid points
PLOTEL4	Quadrilateral visualization element with four grid points
PLOTEL6	Triangular visualization element with six grid points
PLOTEL8	Quadrilateral visualization element with eight grid points
PLOTHEX	Six-sided hexahedron visualization element with eight to twenty grid points
PLOTPEN	Five-sided pentahedron visualization element with six to fifteen grid points
PLOTPYR	Five-sided pyramid visualization element with five to thirteen grid points
PLOTTET	Four-sided tetrahedron visualization element with four to ten grid points

Typically, you use visualization elements in substructuring applications where the underlying FE mesh of the substructure is unavailable. For such situations, you can create a mesh of visualization elements from the grid points associated with the substructure. Because the mesh is a geometric representation of the substructure, you can display the analysis results across the entire model in post-processing.

For example, suppose you want to perform a frequency response analysis on an airframe and you want to:

- Use frequency response test results to define the dynamic stiffness of the wings.
- Use FE to model the remainder of the airframe.
- Visualize the analysis results over the entire airframe.

You can use a direct-input, frequency-dependent component to define the dynamic stiffness matrix of the wings from the test results. To do so, include an FRFSTIF bulk entry for each wing. Each FRFSTIF bulk entry defines the dynamic stiffness matrix of a wing as seen by the DOF that connect the wing to the fuselage.

You can then define an output transformation matrix that relates the response at other points on the wing where test data is collected to the DOF that connect the wing to the fuselage. To do so, define grid

points at these locations on the wings and include an FRFOTM bulk entry for each wing. Each FRFOTM bulk entry relates the response at these grid points to the response of the DOF that connect the wing to the fuselage.

During the frequency response analysis solve, the software computes the response at the DOF that connect the wings to the fuselage. During results recovery, the software uses the response at these DOF and the FRFOTM bulk entries to calculate the response at the other grid points on the wing. To assign physical meaning to the calculated response at these grid points, you can include an FRFOMAP bulk entry for each FRFOTM bulk entry.

To view the response of the entire airframe, the post-processor requires a geometric representation of the wing. You can use the new visualization elements and the grid points on the wings to create meshes of the wings over which the post-processor can display results.





# 7. Transient Response Analysis (SOLs 109 and 112)

## 7.1 Overview of Transient Response Analysis (SOLs 109 and 112)

Transient response analysis is the most general method for computing forced dynamic response. The purpose of a transient response analysis is to compute the behavior of a structure subjected to time-varying excitation. The transient excitation is explicitly defined in the time domain. All of the loads applied to the structure are known at each instant in time. Loads can be in the form of applied forces and enforced motions (see “**Enforced Motion**”). The results obtained from a transient response analysis are typically displacements, velocities, and accelerations of grid points, and forces and stresses in elements.

In Simcenter Nastran, use:

- SOL 109 to perform direct transient response analysis.
- SOL 112 to perform modal transient response analysis.

The direct transient response method performs a numerical integration on the complete coupled equations of motion. The modal transient response method utilizes the mode shapes of the structure to:

- Reduce the problem size because not all of the modes are typically calculated or retained.
- Uncouple the equations of motion when either no damping or only modal damping is used.

The physical solution is then obtained through the summation of the individual modal responses.

The choice of method depends upon the structure and the nature of the loading. The solutions to direct and modal transient response analysis are always real responses. Thus, complex terms in the equation of motion like those resulting from structural damping are not allowed.

## 7.2 Direct Transient Response Analysis (SOL 109)

In direct transient response analysis, the structural response is computed by solving the equation of motion using direct numerical integration. The equation of motion is a set of coupled second-order differential equations that when expressed in matrix form is given by:

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

Equation 7-1.

where  $[M]$ ,  $[B]$ , and  $[K]$  are the mass, viscous damping, and stiffness matrices, respectively.

In direct transient response analysis, the equation of motion is approximated by a central finite difference representation. With the central finite difference representation, the displacement is solved for at discrete times, typically with a fixed integration time step  $\Delta t$ , and the velocity and acceleration at the  $n^{\text{th}}$  time step are given by:

$$\begin{aligned}\{\dot{u}_n\} &= \frac{1}{2\Delta t}\{u_{n+1} - u_{n-1}\} \\ \{\ddot{u}_n\} &= \frac{1}{\Delta t^2}\{u_{n+1} - 2u_n + u_{n-1}\}\end{aligned}$$

Equation 7-2.

Averaging the applied force over three adjacent time points, the equation of motion can be rewritten as:

$$\begin{aligned}\left[\frac{M}{\Delta t^2}\right](u_{n+1} - 2u_n + u_{n-1}) + \left[\frac{B}{2\Delta t}\right](u_{n+1} - u_{n-1}) \\ + \left[\frac{K}{3}\right](u_{n+1} + u_n + u_{n-1}) = \frac{1}{3}(P_{n+1} + P_n + P_{n-1})\end{aligned}$$

Equation 7-3.

Collecting terms, the equation of motion can be rewritten as:

$$[A_1]\{u_{n+1}\} = [A_2] + [A_3]\{u_n\} + [A_4]\{u_{n-1}\}$$

Equation 7-4.

where:

$$\begin{aligned}[A_1] &= \left[\frac{M}{\Delta t^2} + \frac{B}{2\Delta t} + \frac{K}{3}\right] \\ [A_2] &= \frac{1}{3}\{P_{n+1} + P_n + P_{n-1}\}\end{aligned}$$

$$[A_3] = \left[ \frac{2M}{\Delta t^2} - \frac{K}{3} \right]$$

$$[A_4] = \left[ -\frac{M}{\Delta t^2} + \frac{B}{2\Delta t} - \frac{K}{3} \right]$$

Matrix  $[A_1]$  is termed the dynamic matrix, and  $[A_2]$  is the applied force (averaged over three adjacent time points). This approach is similar to the classical Newmark-Beta direct integration method except that  $\{P(t)\}$  is averaged over three time points and  $[K]$  is modified such that the dynamic equation of motion reduces to a static solution  $[K]\{u_n\} = \{P_n\}$  if no  $[M]$  or  $[B]$  exists.

The transient solution is obtained by decomposing  $[A_1]$  and applying it to the right-hand side of the above equation. In this form, the solution behaves like a succession of static solutions with each time step performing a forward-backward substitution (FBS) on a new load vector. The transient nature of the solution is carried through by modifying the applied force matrix  $[A_2]$  with the  $[A_3]$  and  $[A_4]$  terms.

In its simplest form, the  $[M]$ ,  $[B]$ , and  $[K]$  matrices are assumed to be constant throughout the analysis and do not change with time. Special solution methods are available in Simcenter Nastran for variations in these matrices (see the *Simcenter Nastran Advanced Dynamic Analysis User's Guide*).

A significant benefit presents itself if  $\Delta t$  remains constant during the analysis. If you specify a constant  $\Delta t$ , the  $[A_1]$  matrix needs to be decomposed only once. Each progressive step in the analysis is only an FBS of a new load vector. However, if  $\Delta t$  changes,  $[A_1]$  must be redecomposed. Having to redecompose  $[A_1]$  repeatedly can be computationally intensive, especially for large problems.

In many cases it is not necessary to obtain output at each time step. You can reduce the amount of output in a direct transient response analysis by specifying an output time interval that is greater than the solution time step. For example, you can specify a time step of 0.001 second and request output at every fifth time step or every 0.005 second.

## Damping in Direct Transient Response Analysis

Because transient response analysis does not allow for complex stiffness, structural damping must be converted to viscous damping. In direct transient response analysis, structural damping is converted to viscous damping and then added to the other viscous damping sources as follows:

$$[B] = [B^1] + [B^2] + \frac{G}{W_3}[K] + \frac{1}{W_4} \sum G_E [K_E]$$

Equation 7-5.

where:

- [B] = Viscous damping matrix
- [B<sup>1</sup>] = Viscous damping from elements (CBUSH, CDAMPi, CVISC) and B2GG
- [B<sup>2</sup>] = Viscous damping from B2PP direct input matrix plus transfer functions
- G = Overall structural damping coefficient (PARAM,G)
- W<sub>3</sub> = Frequency in radians per unit time (PARAM,W3) for the conversion of overall structural damping into viscous damping
- [K] = Global stiffness matrix
- G<sub>E</sub> = Element structural damping coefficient (GE field on material, property, and element entries)
- W<sub>4</sub> = Frequency in radians per unit time (PARAM,W4) for conversion of element structural damping into viscous damping
- [K<sub>E</sub>] = Element stiffness matrix

Two parameters are used in the conversion of structural damping to viscous damping. PARAM,W3,r, where r is the circular frequency to use in the conversion, converts the overall structural damping that arises from PARAM,G specification. The default value for the W3 parameter is 0.0, which causes the structural damping from PARAM,G specification to be ignored in the transient response analysis.

PARAM,W4,r, where r is the circular frequency to use in the conversion, converts element structural damping that arises from specifying values in the GE field of material, property, or element entries. The default value for the W4 parameter is 0.0, which causes the element structural damping to be ignored in the transient response analysis. The units for PARAM,W3 and PARAM,W4 are radians per unit time.

To best select values for W3 and W4, you should consider the modal content of the structure and the harmonic content of the excitation. Typically, the dominant frequency at which the damping is active is used to specify a value for W3 and W4. Often, the first natural frequency is chosen, but isolated individual element damping can occur at different frequencies, which can be handled by using a different value for W4.

A thorough approach to selecting values for W3 and W4 might be to first perform a Fourier transform of the excitation to obtain its harmonic content. Then using the harmonic content of the excitation, perform a frequency response analysis of the structure. Of course, the frequency response analysis allows for structural damping. From the frequency response analysis, use the dominant frequency as the value for W3 and W4.

To assist in selecting values for the W3 and W4 parameters, it is helpful to understand the basis of the structural-to-viscous damping conversion in [Equation 7-5](#).

Begin by considering the equation of motion when only viscous damping is present:

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

Equation 7-6.

Suppose the system consists of only a single degree-of-freedom. Then the equation of motion simplifies to:

$$m\ddot{u}(t) + b\dot{u}(t) + k u(t) = P(t)$$

Equation 7-7.

where the damping force resulting from the viscous damping,  $f_v(t)$ , is given by:

$$f_v(t) = b\dot{u}(t)$$

Equation 7-8.

Now consider the equation of motion when only structural damping is present:

$$[M]\{\ddot{u}(t)\} + (1 + iG)[K]\{u(t)\} = \{P(t)\}$$

Equation 7-9.

where  $G$  is the structural damping coefficient. Again, suppose the system consists of only a single degree-of-freedom. The equation of motion simplifies to:

$$m\ddot{u}(t) + (1 + iG)k u(t) = P(t)$$

Equation 7-10.

Comparing [Equation 7-7](#) and [Equation 7-10](#), it is apparent that the damping force resulting from structural damping is given by:

$$f_s(t) = iGk u(t)$$

Equation 7-11.

where  $f_s(t)$  is the damping force.

Now examine the steady-state response of both single degree-of-freedom systems to a harmonic excitation. Using phasor notation, the forcing function can be written:

$$P(t) = P_o e^{i\omega t}$$

Equation 7-12.

where  $P_o$  is a real constant representing the amplitude of the excitation and  $\omega$  is the circular frequency of the excitation.

The corresponding steady-state response in phasor notation is given by:

$$u(t) = \bar{A} e^{i\omega t}$$

**Equation 7-13.**

where  $\bar{A}$  is a vector representing the amplitude and phase of the response.

Taking the time derivative of **Equation 7-13** yields:

$$\dot{u}(t) = i\omega \bar{A} e^{i\omega t} = i\omega u(t)$$

**Equation 7-14.**

Substituting **Equation 7-14** into **Equation 7-8** yields:

$$f_v(t) = i b \omega u(t)$$

**Equation 7-15.**

In **Figure 7-1**, structural damping force and viscous damping force are plotted versus excitation circular frequency. The viscous damping force is a linear function of the excitation circular frequency and the structural damping force is independent of excitation circular frequency. At an excitation circular frequency of  $\omega_c$  the damping forces are equal, and, it can be shown, the resulting steady-state responses of the single degree-of-freedom systems are equal as well.

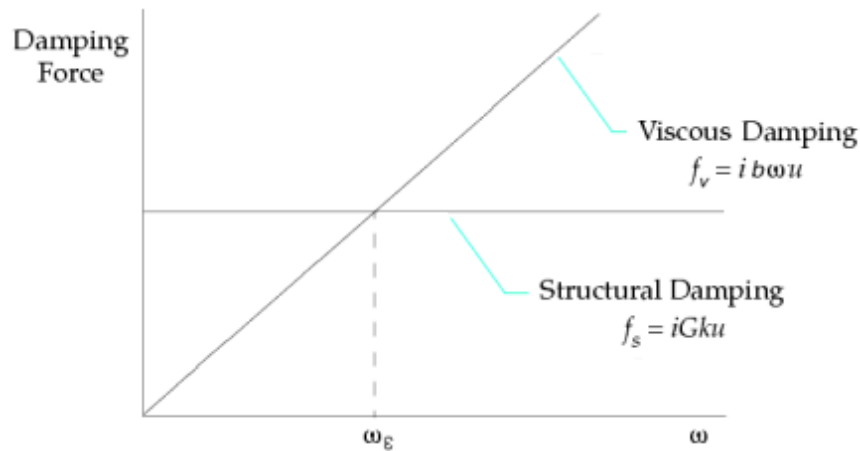


Figure 7-1. Structural damping force and viscous damping force versus excitation frequency for steady-state response to harmonic excitation

Consequently, a single degree-of-freedom system with only structural damping,  $G$ , that is subjected to a harmonic excitation at circular frequency,  $\omega_\epsilon$ , will have a steady-state response identical to a single degree-of-freedom system with only viscous damping,  $b$ , that is subjected to the same harmonic excitation if:

$$Gk = b \omega_\epsilon$$

Equation 7-16.

After rearranging [Equation 7-16](#) to:

$$b = \frac{Gk}{\omega_\epsilon}$$

Equation 7-17.

the basis of [Equation 7-5](#) becomes apparent. It is simply an extension of [Equation 7-17](#) to systems having numerous degree-of-freedom. However, unlike the single degree-of-freedom systems where converting the structural damping to viscous damping using [Equation 7-17](#) guarantees an identical steady-state response, the conversion of structural damping to viscous damping using [Equation 7-5](#) in general will only approximate the actual transient response of the system. How well the predicted transient response matches the actual transient response depends on characteristics of the system and loading, and how appropriately the values for the W3 and W4 parameters are selected.

## Initial Conditions in Direct Transient Response Analysis

For direct transient response analysis, initial conditions are defined in physical space. To begin solving the following equation, values for displacements  $\{u_0\}$  and  $\{\dot{u}_{-1}\}$ , and loads  $\{P_0\}$  and  $\{P_{-1}\}$  are required.

$$[A_1]\{u_{n+1}\} = [A_2] + [A_3]\{u_n\} + [A_4]\{u_{n-1}\}$$

Equation 7-18.

For more information, see [Direct Transient Response Analysis \(SOL 109\)](#).

Once the initial displacements,  $\{u_0\}$ , and initial velocities,  $\{\dot{u}_0\}$ , are known, Simcenter Nastran calculates the other required values from:

$$\{u_{-1}\} = \{u_0\} - \{\dot{u}_0\}\Delta t$$

Equation 7-19.

$$\{P_0\} = [K]\{u_0\} + [B]\{\dot{u}_0\}$$

Equation 7-20.

$$\{P_{-1}\} = [K]\{u_{-1}\} + [B]\{\dot{u}_0\}$$

Equation 7-21.

The software provides you with a number of options for defining the initial displacements,  $\{u_0\}$ , and initial velocities,  $\{\dot{u}_0\}$ , for a direct transient response analysis.

- To specify the initial displacements and velocities directly in physical space, include IC(PHYSICAL) in the case control section that references TIC bulk entries. On the TIC entries, define the initial displacements and velocities. For this case, the initial displacement and velocity for any degree-of-freedom that is unspecified is taken to be zero.
- To use the displacements from a static subcase as the initial displacements, include IC(STATSUB) in the case control section. For this case, TIC entries are not used and the initial velocities are taken to be zero. To account for differential stiffness from the static subcase, include IC(STATSUB,DIFFK) in the case control section.
- To use the static deflections resulting from the dynamic loading evaluated at  $t = 0$ , include IC(TZERO) in the case control section. When you do so, the software computes the static deflections resulting from the dynamic loading at  $t = 0$  and uses it for the initial displacements. For this case, TIC entries are not used and the initial velocities are taken to be zero. To account for differential stiffness from the static deflections, include IC(TZERO,DIFFK) in the case control section.

Regardless of how the initial conditions are specified, the initial acceleration for all points in the structure is taken to be zero (constant initial velocity).



For more information on initial conditions, see the IC case control command in the *Simcenter Nastran Quick Reference Guide*.

## 7.3 Modal Transient Response Analysis (SOL 112)

Modal transient response analysis is another method for computing the transient response of a structure. This method uses the mode shapes of the structure to reduce the size, uncouple the equations of motion (when no damping or only modal damping is used), and make the numerical integration more efficient (when the equations of motion are uncoupled). Because the mode shapes are typically computed as part of the characterization of the structure, modal transient response analysis is a natural extension of a normal modes analysis.

As a first step in the formulation, transform the variables from physical coordinates,  $\{u(t)\}$ , to modal coordinates,  $\{\xi(t)\}$ , by assuming:

$$\{u(t)\} = [\phi]\{\xi(t)\}$$

Equation 7-22.

The mode shapes  $[\phi]$  are used to transform the problem in terms of the behavior of the modes as opposed to the behavior of the grid points. **Equation 7-22** represents an equality if all modes are calculated and used; however, because all modes are rarely calculated or used, the equation typically represents an approximation.

By temporarily ignoring damping, **Equation 7-23**

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

Equation 7-23.

simplifies to

$$[M]\{\ddot{u}(t)\} + [K]\{u(t)\} = \{P(t)\}$$

Equation 7-24.

For more information, see **Damping in Direct Transient Response Analysis**.

Substituting **Equation 7-22** into **Equation 7-24** yields:

$$[M][\phi]\{\ddot{\xi}(t)\} + [K][\phi]\{\xi(t)\} = \{P(t)\}$$

Equation 7-25.

At this point, the equation of motion is generally coupled. To uncouple the equation of motion, premultiply by  $[\phi]^T$  to obtain:

$$[\phi]^T [M][\phi]\{\ddot{\xi}\} + [\phi]^T [K][\phi]\{\xi\} = [\phi]^T \{P(t)\}$$

Equation 7-26.

where:

$$[\phi]^T [M][\phi] = \text{Modal (generalized) mass matrix}$$

$$[\phi]^T [K][\phi] = \text{Modal (generalized) stiffness matrix}$$

$$[\phi]^T \{P\} = \text{Modal force vector}$$

The modal mass and stiffness matrices are diagonal matrices. Because these matrices do not have off-diagonal terms, the equations of motion are uncoupled and can be written as a set of single degree-of-freedom systems:

$$m_i \ddot{\xi}_i(t) + k_i \xi_i(t) = p_i(t)$$

Equation 7-27.

where:

$$m_i = \text{Modal mass for the } i^{\text{th}} \text{ mode}$$

$$k_i = \text{Modal stiffness for the } i^{\text{th}} \text{ mode}$$

$$p_i = \text{Modal force for the } i^{\text{th}} \text{ mode}$$

Once the individual modal responses,  $\xi_i(t)$ , are computed, physical responses are recovered as the summation of the modal responses using:

$$\{u(t)\} = [\phi]\{\xi(t)\}$$

Equation 7-28.

Because numerical integration is applied to the relatively small number of uncoupled equations, there is not as large a computational penalty for varying  $\Delta t$  as there is in direct transient response analysis. However, a constant  $\Delta t$  is still recommended.

Another efficiency option in the modal transient solution is that the output time interval may be greater than the solution time interval. In many cases, it is not necessary to sample output at each solution time. For example, if the solution is performed every 0.001 second, obtaining output every fifth time step or every 0.005 second may be perfectly acceptable.

## Viscous Damping in Modal Transient Response Analysis

If the viscous damping matrix,  $[B]$ , exists, the orthogonality property of the modes does not, in general, diagonalize the generalized damping matrix. That is:

$$[\phi]^T [B] [\phi] \neq \text{diagonal}$$

Equation 7-29.

Consequently, the equations of motion are coupled in modal space. To solve the coupled equations, the numerical integration method described in the **Direct Transient Response Analysis (SOL 109)** section is applied to the modal system as follows:

$$[A_1] \{\xi_{n+1}\} = [A_2] + [A_3] \{\xi_n\} + [A_4] \{\xi_{n-1}\}$$

Equation 7-30.

where:

$$\begin{aligned} [A_1] &= [\phi]^T \left[ \frac{M}{\Delta t^2} + \frac{B}{2\Delta t} + \frac{K}{3} \right] [\phi] \\ [A_2] &= \frac{1}{3} [\phi]^T \{P_{n+1} + P_n + P_{n-1}\} \\ [A_3] &= [\phi]^T \left[ \frac{2M}{\Delta t^2} - \frac{K}{3} \right] [\phi] \\ [A_4] &= [\phi]^T \left[ -\frac{M}{\Delta t^2} + \frac{B}{2\Delta t} - \frac{K}{3} \right] [\phi] \end{aligned}$$

Because the number of modes computed is typically much less than the number of physical variables, direct integration of the modal equations has computational benefits when compared to a direct transient response analysis.

Another way to include damping in a modal transient response analysis is to use modal damping. Modal damping is either viscous or structural damping that is applied to each mode separately so that in the

absence of other sources of damping the equations of motion remain uncoupled. If other sources of damping are present, modal damping is added into the equations of motion along with the other damping.

You request modal damping by including a SDAMPING case control command in the input file. You can use the KDAMP parameter to specify whether the modal damping is applied as modal viscous damping or modal structural damping. Modal viscous damping is the default. Because modal transient response analysis allows only real responses, using KDAMP to specify modal structural damping in a modal transient response analysis causes a fatal error to occur.

You can include structural damping in a modal transient analysis by specifying the GE field on material, property, and element entries, and/or PARAM,G. If you do so, you must also specify some combination of the W3, W4, and/or WMODAL parameters to convert the structural damping to viscous damping. For more information on converting structural damping to viscous damping, see the [Structural Damping in Modal Transient Response Analysis](#) section.

The SDAMPING case control command should reference the TID of a TABDMP1 bulk entry. On the TABDMP1 entry, you define frequency/damping data. The software uses the frequency/damping data to calculate the damping value to apply at each modal frequency.

The TYPE field on the TABDMP1 entry is used to specify whether the damping values on the TABDMP1 entry represent structural (hysteretic) damping, damping factor (fraction of critical damping), or quality factor (amplification factor). Structural damping is the default. To find the damping value at a specific modal frequency, the software linearly interpolates or linearly extrapolates the values provided on the TABDMP1 entry.

For example, suppose damping factor versus frequency data is entered on a TABDMP1 entry as follows:

1	2	3	4	5	6	7	8	9	10
\$TABDMP1	TID	TYPE							
\$	f1	g1	f2	g2	f3	g3	etc.		
TABDMP1	10	CRIT							
	2.0	0.16	3.0	0.18	4.0	0.13	6.0	0.13	
	ENDT								

Also suppose the structure being analyzed has modes at 1.0, 2.5, 3.6, and 5.5 Hz. The software will interpolate and extrapolate the TABDMP1 data to produce estimates of damping factor at the modal frequencies as indicated in [Table 7-7](#) and [Figure 7-2](#).

Table 7-7. Results from Interpolating/Extrapolating TABDMP1 Data

Entered		Computed	
$f$	$\zeta$	$f$	$\zeta$
2.0	0.16	1.0	0.14 <sup>(1)</sup>
3.0	0.18	2.5	0.17
4.0	0.13	3.6	0.15
6.0	0.13	5.5	0.13

<sup>(1)</sup> The value is extrapolated from data at 2.0 Hz and 3.0 Hz.

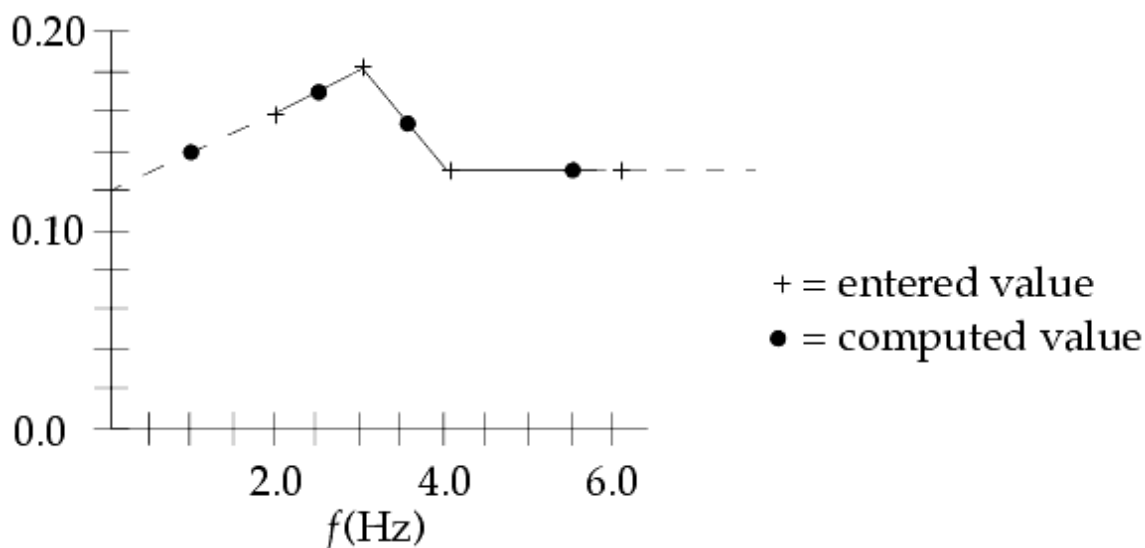


Figure 7-2. Interpolating/Extrapolating TABDMP1 Data

**Table 7-8** lists the equations the software uses to convert the interpolated or extrapolated damping values from the TABDMP1 entry to modal viscous damping,  $b_i$ , or modal structural damping,  $G_i$ .

Table 7-8. Conversion of Interpolated/Extrapolated Damping Values to Modal Damping Values

Type of damping in TABDMP1 entry	Modal viscous damping PARAM,KDAMP,1 (Default)	Modal structural damping PARAM,KDAMP,-1
Structural damping <sup>(1)</sup>	$b_i = g(f_i) b_{cr} / 2$	Specifying PARAM,KDAMP,-1 causes a fatal error to occur when running SOL 112.
Damping factor <sup>(2)</sup>	$b_i = g(f_i) b_{cr}$	
Quality <sup>(3)</sup>	$b_i = b_{cr} / (2 g(f_i))$	

where:  
 $b_i$  = Modal viscous damping for the  $i^{\text{th}}$  mode  
 $f_i$  = Undamped natural frequency of the  $i^{\text{th}}$  mode

Type of damping in TABDMP1 entry	Modal viscous damping PARAM,KDAMP,1 (Default)	Modal structural damping PARAM,KDAMP,-1
$g(f_i)$ = Interpolated or extrapolated damping value at $f_i$ from damping/frequency data on TABDMP1 entry $b_{cr} = 2 m_i \omega_i$ = Critical damping for the $i^{\text{th}}$ mode with $\omega_i = 2\pi f_i$ $m_i$ = Modal mass for the $i^{\text{th}}$ mode (1) Also referred to as hysteretic damping (2) Also referred to as fraction of critical damping (3) Also referred to as amplification factor		

When all other sources of damping are absent and modal viscous damping is applied, the equations of motion in modal space are uncoupled and have the following form:

$$m_i \ddot{\xi}_i(t) + b_i \dot{\xi}_i(t) + k_i \xi_i(t) = p_i(t)$$

Equation 7-31.

where  $b_i$  is the modal viscous damping at the natural frequency of the  $i^{\text{th}}$  mode.

If the equations of motion in modal space are uncoupled, they can be integrated efficiently using Duhamel's integral. For each mode, the solution to [Equation 7-31](#) is:

$$\begin{aligned} \xi(t) = & e^{-bt/2m} \left( \xi_0 \cos \omega_d t + \frac{\dot{\xi}_0 + \xi_0 b / (2m)}{\omega_d} \sin \omega_d t \right) \\ & + e^{-bt/2m} \frac{1}{m\omega_d} \int_0^t e^{b\tau/2m} p(\tau) \sin \omega_d(t - \tau) d\tau \end{aligned}$$

Equation 7-32.

where:

$\xi_0$	=	Initial modal displacement
$\dot{\xi}_0$	=	Initial modal velocity
$\omega_d$	=	Damped natural frequency = $\omega_n (1 - \zeta^2)^{1/2}$
$\omega_n$	=	Undamped natural frequency
$\zeta$	=	Damping factor = $b / (2m\omega_n)$

If sources of damping other than modal viscous damping are present, the equations of motion in modal space will typically be coupled. When the equations of motion in modal space are coupled, the software performs a direct solution on the equations of motion in modal space. Although you still realize a computational benefit from the modal problem size being typically reduced, you do incur a computational penalty from having to use a direct solution. Thus, you should consider using only modal viscous damping when you use modal transient response analysis. If you must include other sources of damping in the transient response analysis, you should consider using direct transient response analysis.

## Structural Damping in Modal Transient Response Analysis

Transient response analysis allows for only real responses. If you specify modal structural damping for a modal transient response analysis, a fatal error results. However, you can include structural damping in a modal transient response analysis by specifying the GE field on material, property, and element entries, and/or PARAM,G. When you do so, the structural damping must be converted to viscous damping. You can choose between two methods to convert structural damping to viscous damping. In the first method, you specify either one or two conversion factors using the W3 and W4 parameters. Typically, these conversion factors are selected to account for the dominant frequency at which the damping is active. With these parameters specified, the structural damping is converted to viscous damping using:

$$\frac{G}{W3}[K_{dd}^1] + \frac{1}{W4}[K_{dd}^4]$$

where:

$[K_{dd}^1]$  = Assembled stiffness matrix.

$[K_{dd}^4]$  = Assembled elemental damping matrix from specifying the GE field on material, property, and element entries.

$G$  = Overall structural damping coefficient from specifying PARAM,G.

$W3$  = User-defined conversion frequency for overall structural damping. (See the W3 parameter.)

$W4$  = User-defined conversion frequency for elemental structural damping. (See the W4 parameter.)

Then, the modal viscous damping matrix is calculated using:

$$[B_{qq}] = [\Phi_{dq}^T] \left( [B_{dd}^1] + [B_{dd}^2] + \frac{G}{W3}[K_{dd}^1] + \frac{1}{W4}[K_{dd}^4] \right) [\Phi_{dq}]$$

where:

$[B_{qq}]$  = Modal viscous damping matrix.

$[\Phi_{dq}]$  = Mode shape matrix.

$[B_{dd}^1]$  = Assembled viscous damping matrix.

$[B_{dd}^2]$  = Viscous damping matrix from DMIG input.

$\frac{G}{W3}[K_{dd}^1] + \frac{1}{W4}[K_{dd}^4]$  = Converted structural damping matrix.

The second method uses the WMODAL parameter to convert structural damping to viscous damping. It can be used either independently or in conjunction with the first method. When you use it independently, you do not need to specify any conversion frequency factors to use in the conversion. The software automatically uses the solved modal frequencies in the conversion. As a result, the conversion is valid over a much wider range of frequencies. With the WMODAL parameter specified, the structural damping is converted to viscous damping using:

$$\begin{bmatrix} 1/\omega_1 & 0 & \dots & 0 \\ 0 & 1/\omega_2 & \dots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \dots & 1/\omega_n \end{bmatrix} [\Phi_{dq}^T] (G[K_{dd}^1] + [K_{dd}^4]) [\Phi_{dq}] \begin{bmatrix} 1/\omega_1 & 0 & \dots & 0 \\ 0 & 1/\omega_2 & \dots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \dots & 1/\omega_n \end{bmatrix}$$

where  $\omega_i$  are the modal frequencies in rad/sec.

For rigid body modes, the software partitions the diagonal entries,  $1/\omega_i$ , out of the calculation. The FZERO parameter is used to identify rigid body modes. (See the FZERO parameter.)

Because the WMODAL parameter can be used in conjunction with the W3 and W4 parameters, you now have three options for converting structural damping to viscous damping in a modal transient response analysis.

- If you specify one or both of the W3 and W4 parameters, and either omit the WMODAL parameter from the input file or specify WMODAL = NO, the structural-to-viscous damping conversion is calculated using:

$$\frac{G}{W3}[K_{dd}^1] + \frac{1}{W4}[K_{dd}^4]$$

Therefore, the modal viscous damping matrix is calculated using:

$$[B_{qq}] = [\Phi_{dq}^T] \left( [B_{dd}^1] + [B_{dd}^2] + \frac{G}{W3}[K_{dd}^1] + \frac{1}{W4}[K_{dd}^4] \right) [\Phi_{dq}]$$

- If you specify WMODAL = YES, and either omit the W3 and W4 parameters from the input file or accept their default value of 0.0, the structural-to-viscous damping conversion is calculated using:



$$\begin{bmatrix} 1/\omega_1 & 0 & \dots & 0 \\ 0 & 1/\omega_2 & \dots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \dots & 1/\omega_q \end{bmatrix} [\Phi_{dq}^T] (G[K_{dd}^1] + [K_{dd}^4]) [\Phi_{dq}] \begin{bmatrix} 1/\omega_1 & 0 & \dots & 0 \\ 0 & 1/\omega_2 & \dots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \dots & 1/\omega_q \end{bmatrix}$$

Therefore, the modal viscous damping matrix is calculated using:

$$[B_{yy}] = [\Phi_{dq}^T] ([B_{dd}^1] + [B_{dd}^2]) [\Phi_{dq}] + \begin{bmatrix} 1/\omega_1 & 0 & \dots & 0 \\ 0 & 1/\omega_2 & \dots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \dots & 1/\omega_q \end{bmatrix} [\Phi_{dq}^T] (G[K_{dd}^1] + [K_{dd}^4]) [\Phi_{dq}] \begin{bmatrix} 1/\omega_1 & 0 & \dots & 0 \\ 0 & 1/\omega_2 & \dots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \dots & 1/\omega_q \end{bmatrix}$$

- If you specify WMODAL = YES and specify one or both of the W3 and W4 parameters, the contributions from both structural-to-viscous damping conversion methods are summed and the modal viscous damping matrix is calculated using:

$$[B_{yy}] = [\Phi_{dq}^T] ([B_{dd}^1] + [B_{dd}^2] + \frac{G}{W3}[K_{dd}^1] + \frac{1}{W4}[K_{dd}^4]) [\Phi_{dq}] + \begin{bmatrix} 1/\omega_1 & 0 & \dots & 0 \\ 0 & 1/\omega_2 & \dots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \dots & 1/\omega_q \end{bmatrix} [\Phi_{dq}^T] (G[K_{dd}^1] + [K_{dd}^4]) [\Phi_{dq}] \begin{bmatrix} 1/\omega_1 & 0 & \dots & 0 \\ 0 & 1/\omega_2 & \dots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \dots & 1/\omega_q \end{bmatrix}$$

An example of a complete input file containing the WMODAL parameter is *wmod\_rbm.dat*. It can be found in *install\_dir/scnas/nast/tpl*.

## Initial Conditions in Modal Transient Response Analysis

For modal transient response analysis, initial conditions can be defined in either physical space or modal space. If the initial conditions are defined in physical space, the software automatically converts them to modal space during the solve. The options to define initial conditions for modal transient response analysis are similar to those for defining initial conditions for direct transient response analysis.

- To specify the initial displacements and velocities directly in physical space, include IC(PHYSICAL) in the case control section that references TIC bulk entries. On the TIC entries, define the initial displacements and velocities. For this case, the initial displacement and velocity for any degree-of-freedom that is unspecified is taken to be zero.
- To use the displacements from a static subcase as the initial displacements, include IC(STATSUB) in the case control section. For this case, TIC entries are not used and the initial velocities are taken to be

zero. To account for differential stiffness from the static subcase, include IC(STATSUB,DIFFK) in the case control section.

- To use the static deflections resulting from the dynamic loading evaluated at  $t = 0$ , include IC(TZERO) in the case control section. When you do so, the software computes the static deflections resulting from the dynamic loading at  $t = 0$  and uses it for the initial displacements. For this case, TIC entries are not used and the initial velocities are taken to be zero. To account for differential stiffness from the static deflections, include IC(TZERO,DIFFK) in the case control section.

With modal transient response analysis, you have an additional option.

- To specify the initial displacements and/or velocities directly in modal space, include IC(MODAL) in the case control section that references TIC bulk entries. On the TIC entries, define the initial modal displacements. For this case, the initial modal displacement and/or velocities for any degree-of-freedom that is unspecified is taken to be zero.

When you directly specify the initial modal displacements,  $\{\xi_0\}$ , and initial modal velocities,  $\{\dot{\xi}_0\}$ , the software calculates the other required values to begin solving **Equation 7-33**:

$$[A_1]\{\xi_{n+1}\} = [A_2] + [A_3]\{\xi_n\} + [A_4]\{\xi_{n-1}\}$$

Equation 7-33.

from:

$$\{\xi_{-1}\} = \{\xi_0\} - \{\dot{\xi}_0\}\Delta t$$

Equation 7-34.

$$\{P_0\} = [K][\phi]\{\xi_0\} + [B][\phi]\{\dot{\xi}_0\}$$

Equation 7-35.

$$\{P_{-1}\} = [K][\phi]\{\xi_{-1}\} + [B][\phi]\{\dot{\xi}_0\}$$

Equation 7-36.

For more information, see **Viscous Damping in Modal Transient Response Analysis**.

Regardless of how the initial conditions are specified, the initial acceleration for all points in the structure is taken to be zero (constant initial velocity).

For more information on initial conditions, see the IC case control command in the *Simcenter Nastran Quick Reference Guide*.

## Mode Truncation in Modal Transient Response Analysis

It is possible that not all of the computed modes are required to accurately predict the transient response of a system. Often, only the lowest few modes are necessary to use in a modal transient response analysis. It is quite common to evaluate the frequency content of transient loads and determine a frequency above which no modes are noticeably excited. This frequency is called the cutoff frequency. The act of specifically not using all of the modes of a system in the response solution is termed mode truncation. Mode truncation assumes that an accurate solution can be obtained using a reduced set of modes.

By default, Simcenter Nastran uses all of the computed modes in a modal transient response analysis. However, you can control the modes used in the modal transient response analysis by:

- Specifying a frequency range on the eigenvalue entry (EIGR or EIGRL).
- Using the LFREQ, HFREQ, and LMODES parameters.

PARAM,LFREQ gives the lower limit on the frequency range of retained modes.

PARAM,HFREQ gives the upper limit on the frequency range of retained modes.

PARAM,LMODES gives the number of the lowest modes to be retained.

- Including a MODSEL case control command to directly select or exclude individual mode numbers from the modal transient response calculation. The MODSEL case control command can be used for fluid modes, as well as, structural modes, and can be used in a restart run. The restart capability allows you to select modes from a SOL 103 normal modes analysis to use in a consecutive SOL 112 modal transient response analysis.

It is very important to remember that truncating modes in a particular frequency range may truncate a significant portion of the behavior in that frequency range. Typically, high-frequency modes are truncated because they are more costly to compute. So, truncating high-frequency modes truncates high frequency response. In most cases, high-frequency mode truncation is not of concern. You should evaluate the truncation in terms of the loading frequency and the important characteristic frequencies of the structure. Residual vectors can also be requested to improve the response by adding the missing static flexibility associated with these higher modes. See [Residual Vectors for Modal Response Solutions](#) for more information.

## Dynamic Data Recovery in Modal Transient Response Analysis

In modal transient response analysis, two options are available for recovering displacements and stresses: the mode displacement method and the matrix method. Both methods give the same answers, although with differences in the computational effort required.

The mode displacement method computes the total physical displacements for each time step from the modal displacements and then computes element stresses from the total physical displacements. The number of operations is proportional to the number of time steps.

The matrix method computes displacements per mode and element stresses per mode, and then computes physical displacements and element stresses as the summation of modal displacements and element stresses. The number of operations is proportional to the number of modes.

Because the number of modes is usually much less than the number of time steps, the matrix method is usually more efficient and is the default. The mode displacement method can be selected by including PARAM,DDRMM,-1 in the bulk data section. The mode displacement method is required when “time frozen” deformed structure plots are requested (see **“Plotted Output”** ).

The mode acceleration method (**“Advanced Dynamic Analysis Capabilities”**) is another form of data recovery for modal transient response analysis. This method can provide better accuracy since detailed local stresses and forces are subject to mode truncation and may not be as accurate as the results computed with the direct method.

## 7.4 Modal Versus Direct Transient Response

Some general guidelines can be used in selecting modal transient response analysis versus direct transient response analysis. These guidelines are summarized in **Table 7-10**.

**Table 7-10. Modal Versus Direct Transient Response**

	Modal	Direct
Small Model		X
Large Model	X	
Few Time Steps		X
Many Time Steps	X	
High Frequency Excitation		X
Normal Damping		X
Higher Accuracy		X

In general, larger models may be solved more efficiently using modal transient response analysis because the size of the problem is typically reduced in modal space. In addition, when no damping or only modal damping is used in a modal transient response analysis, the equations of motion are uncoupled and **Equation 7-37** can be very efficiently used to calculate the response for even long duration transients. These advantages of modal transient response analysis are especially true when the modes are already available from a previous modal solution.

$$\xi(t) = e^{-bt/2m} \left( \xi_0 \cos \omega_d t + \frac{\dot{\xi}_0 + \xi_0 b / (2m)}{\omega_d} \sin \omega_d t \right) + e^{-bt/2m} \frac{1}{m\omega_d} \int_0^t e^{b\tau/2m} p(\tau) \sin \omega_d (t - \tau) d\tau$$

Equation 7-37.

For more information, see [Viscous Damping in Modal Transient Response Analysis](#).

Because the major portion of the effort in a modal transient response analysis is to calculate the modes, for large systems that require the calculation of a large number of modes, a modal transient response analysis can be just as computationally intensive as a direct transient response analysis. This is especially true for high-frequency excitation. To capture high frequency response in a modal transient response analysis, less accurate high-frequency modes must be computed. For small models with only a few time steps, the direct method may be the most efficient because it solves the equations without first computing the modes. The direct method is more accurate than the modal method because the direct method is not concerned with mode truncation.

## 7.5 Transient Excitation Definition

An important aspect of a transient response analysis is the definition of the loading function. In a transient response analysis, the load must be defined as a function of time. Loads are defined in the same manner whether the direct or modal method is used.

The following bulk entries are typically used to define time-dependent loads:

Bulk Entry	Description
TLOAD1	Defines time-dependent dynamic load.
TLOAD2	Defines time-dependent dynamic load (alternate form).
DELAY	Defines time delay of dynamic load.
TABLEDi	Defines frequency dependence of dynamic load via tabular data.
DAREA	Defines spatial distribution of dynamic load.
FORCEi	
MOMENTi	
PLOAD	
PLOAD1	
SELOAD	

Bulk Entry	Description
SPCD	
DLOAD	Combines dynamic loads.

To fully specify a dynamic load, you must:

- Define the location and direction that the load is applied to the structure. Because this characteristic locates the loading in space, it is called the spatial distribution of the dynamic loading.
- Define the time variation in the load. This characteristic of a dynamic load differentiates it from a static load. The time variation is called the temporal distribution of the load.

Simcenter Nastran makes it possible for you to apply many complicated and temporally similar loadings with a minimum of input by using Table IDs and Set IDs. You can also easily combine simple loadings to create complicated load distributions that vary in position as well as time.

The remainder of this section describes the bulk entries for transient excitation. For more complete bulk entry descriptions, see the *Simcenter Nastran Quick Reference Guide*.

## Time-Dependent Loads – TLOAD1 Entry

The TLOAD1 entry is the most general form in which to define a time-dependent load. It defines a dynamic loading of the form:

$$\{P(t)\} = \{A \cdot F(t - \tau)\}$$

Equation 7-38.

$F(t)$  is defined in tabular format on TABLEDi bulk entries if time-dependent, or directly on the TLOAD1 entry as real input if constant. For tabular input, you need not explicitly define the load at every instant in time for which the transient solution is evaluated. Only those values which describe the character of the loading are required. Simcenter Nastran linearly interpolates for intermediate values.

Define time delay,  $\tau$ , for each degree-of-freedom referenced through the EXCITEID field. If the value for time delay,  $\tau$ , varies across degree-of-freedom, reference DELAY bulk entries to define the time delay,  $\tau$ . If the value for time delay,  $\tau$ , is uniform across all degree-of-freedom, enter the value directly on the TLOAD1 entry as real input.

1	2	3	4	5	6	7	8	9	10
TLOAD1	SID	EXCITEID	DELAY	TYPE	TID				

Field	Contents
SID	Load set identification number.
EXCITEID	Identification number of a DAREA, FORCEi, MOMENTi, PLOAD, PLOAD1, SELOAD, or SPCD entry that lists each degree-of-freedom to apply the excitation and the corresponding scale factor, A, for the excitation.
DELAY	Time delay, $\tau$ .
TYPE	Defines the type of the dynamic excitation.
TID	$F(t)$ definition.

## Time-Dependent Loads – TLOAD2 Entry

The TLOAD2 entry is a general analytical form with which to define a time-dependent load. The TLOAD2 entry defines a dynamic loading of the form:

$$\{P(t)\} = \begin{cases} 0 & , t < (T1 + \tau) \text{ or } t > (T2 + \tau) \\ A\tilde{t}^B e^{C\tilde{t}} \cos(2\pi F\tilde{t} + P) & , (T1 + \tau) \leq t \leq (T2 + \tau) \end{cases}$$

Equation 7-39.

where:

$$\tilde{t} = t - (T1 - \tau)$$

With TLOAD2 entries, the value of the load at a particular instant in time is determined by evaluating the analytic function at the specific time. Only the parameters of the load need to be defined. All of these parameters are defined directly on the TLOAD2 entry except the time delay,  $\tau$ .

Define time delay,  $\tau$ , for each degree-of-freedom referenced through the EXCITEID field. If the value for time delay,  $\tau$ , varies across degree-of-freedom, reference DELAY bulk entries to define the time delay,  $\tau$ . If the value for time delay,  $\tau$ , is uniform across all degree-of-freedom, enter the value directly on the TLOAD1 entry as real input.

1	2	3	4	5	6	7	8	9	10
TLOAD2	SID	EXCITEID	DELAY	TYPE	T1	T2	F	P	
	C	B							

Field	Contents
SID	Load set identification number.
EXCITEID	Identification number of a DAREA, FORCEi, MOMENTi, PLOAD, PLOAD1, SELOAD, or SPCD entry that lists each degree-of-freedom to apply the excitation and the corresponding scale factor, A, for the excitation.
DELAY	Time delay, $\tau$ .
TYPE	Defines the type of the dynamic excitation.
T1, T2	Time constants.
F	Frequency in cycles per unit time.
P	Phase angle in degree.
C	Exponential coefficient.
B	Growth coefficient.

### Time Delay – DELAY Entry

The DELAY entry defines the time delay,  $\tau$ , for specified degree-of-freedom.

1	2	3	4	5	6	7	8	9	10
DELAY	SID	P1	C1	T1	P2	C2	T2		

Field	Contents
SID	Identification number.
Pi	Grid, extra, or scalar point ID.
Ci	Component number.
Ti	Time delay, $\tau$ .

Any number of DELAY entries may be used; all those with the same SID are combined.

### Dynamic Load Tabular Function – TABLEDi Entries

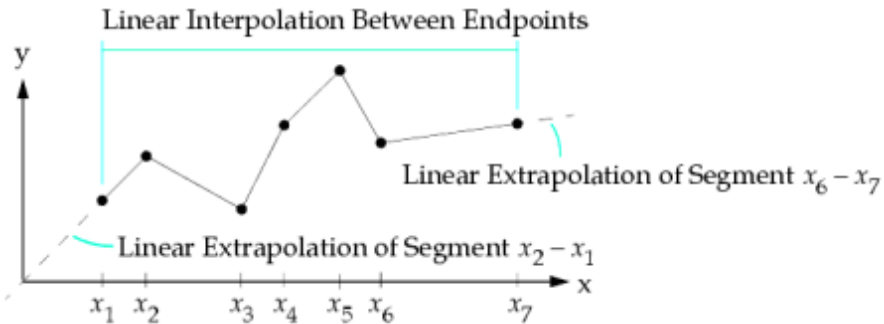
TABLEDi entries ( $i = 1$  through 4) each define a tabular function for use in generating time-dependent dynamic loads. The form of each TABLEDi entry varies slightly, depending on the value of  $i$ , as does the algorithm for  $y(x)$ . The  $x$  values need not be evenly spaced.

For TABLED1, TABLED2, and TABLED3 entries, the software uses linear interpolation to look up the value of  $y$  when  $x$  is inside the range of the tabular data. For TABLED1 entries, you can optionally specify that the software perform the interpolation in Cartesian, semi-log, or log-log space.

For values of  $x$  that lie outside the range of the tabular data, you can specify how the software looks up values from the following options:



- Linearly extrapolate the two starting or two ending data points to obtain the value for y, as shown in **Figure 7-3**.
- Use the value at the starting or ending data point as the value for y.
- Specify the value for y.



**Figure 7-3. Interpolation and Extrapolation for TABLED1, TABLED2, and TABLED3 Entries**

The TABLED1 entry has the following format:

1	2	3	4	5	6	7	8	9	10
TABLED1	TID	XAXIS	YAXIS	EXTRAP	LOEXT	HIEXT			
	x1	y1	x2	y2	x3	y3	-etc.-	"ENDT"	

Field	Contents
TID	Table identification number.
XAXIS, YAXIS	Specifies whether the interpolation is linear in Cartesian, semi-log, or log-log space.  For interpolation in Cartesian space, specify XAXIS = LINEAR and YAXIS = LINEAR.  For interpolation in semi-log space, specify XAXIS = LINEAR and YAXIS = LOG, or XAXIS = LOG and YAXIS = LINEAR.  For interpolation in log-log space, specify XAXIS = LOG and YAXIS = LOG.
EXTRAP	Extrapolation option.  If EXTRAP = 0, obtain table lookup values for $x < x_{\min}$ and $x > x_{\max}$ by applying the interpolation formula to the two starting and two ending data points.  If EXTRAP = 1, use the value of the table field at $x_{\min}$ for the table lookup when $x < x_{\min}$ and use the value of the table field at $x_{\max}$ for the table lookup when $x > x_{\max}$ .

Field	Contents
	If EXTRAP = 2, use LOEXT for the table lookup when $x < x_{\min}$ and use HIEXT for the table lookup when $x > x_{\max}$ .
LOEXT	For EXTRAP = 2, table lookup value when $x < x_{\min}$ .
HIEXT	For EXTRAP = 2, table lookup value when $x > x_{\max}$ .
xi, yi	Tabular values.
ENDT	Ends the table input.

The TABLED1 entry uses the algorithm:

$$y = y_T(x)$$

Equation 7-40.

The linear interpolation and extrapolation algorithms are as follows:

XAXIS	YAXIS	$y_T(x)$
LINEAR	LINEAR	$\frac{x_j - x}{x_j - x_i} y_i + \frac{x - x_i}{x_j - x_i} y_j$
LOG	LINEAR	$\frac{\ln(x_j/x)}{\ln(x_j/x_i)} y_i + \frac{\ln(x/x_i)}{\ln(x_j/x_i)} y_j$
LINEAR	LOG	$\exp \left[ \frac{x_j - x}{x_j - x_i} \ln y_i + \frac{x - x_i}{x_j - x_i} \ln y_j \right]$
LOG	LOG	$\exp \left[ \frac{\ln(x_j/x)}{\ln(x_j/x_i)} \ln y_i + \frac{\ln(x/x_i)}{\ln(x_j/x_i)} \ln y_j \right]$

The TABLED2 entry has the following format:

1	2	3	4	5	6	7	8	9	10
TABLED2	TID	X1	EXTRAP	LOEXT	HIEXT				
	x1	y1	x2	y2	x3	y3	-etc.-	"ENDT"	

Field	Contents
TID	Table identification number.
X1	Table parameter.
EXTRAP	Extrapolation option.  If EXTRAP = 0, obtain table lookup values for $(x - X1) < x_{min}$ and $(x - X1) > x_{max}$ by linearly extrapolating the two starting and two ending data points.  If EXTRAP = 1, use the value of the table field at $x_{min}$ for the table lookup when $(x - X1) < x_{min}$ and use the value of the table field at $x_{max}$ for the table lookup when $(x - X1) > x_{max}$ .  If EXTRAP = 2, use LOEXT for the table lookup when $(x - X1) < x_{min}$ and use HIEXT for the table lookup when $(x - X1) > x_{max}$ .
LOEXT	For EXTRAP = 2, table lookup value when $(x - X1) < x_{min}$ .
HIEXT	For EXTRAP = 2, table lookup value when $(x - X1) > x_{max}$ .
$x_i, y_i$	Tabular values.
ENDT	Ends the table input.

The TABLED2 entry uses the algorithm:

$$y = y_T(x - X1)$$

Equation 7-41.

The TABLED3 entry has the following format:

1	2	3	4	5	6	7	8	9	10
TABLED3	TID	X1	X2	EXTRAP	LOEXT	HIEXT			
	x1	y1	x2	y2	x3	y3	-etc.-	"ENDT"	

Field	Contents
TID	Table identification number.
X1, X2	Table parameters.
EXTRAP	Extrapolation option.  If EXTRAP = 0, obtain table lookup values for $(x - X1) / X2 < x_{min}$ and $(x - X1) / X2 > x_{max}$ by linearly extrapolating the two starting and two ending data points.

Field	Contents
	If EXTRAP = 1, use the value of the table field at $x_{\min}$ for the table lookup when $(x - X1) / X2 < x_{\min}$ and use the value of the table field at $x_{\max}$ for the table lookup when $(x - X1) / X2 > x_{\max}$ .
	If EXTRAP = 2, use LOEXT for the table lookup when $(x - X1) / X2 < x_{\min}$ and use HIEXT for the table lookup when $(x - X1) / X2 > x_{\max}$ .
LOEXT	For EXTRAP = 2, table lookup value when $(x - X1) / X2 < x_{\min}$ .
HIEXT	For EXTRAP = 2, table lookup value when $(x - X1) / X2 > x_{\max}$ .
xi, yi	Tabular values.
ENDT	Ends the table input.

The TABLED3 entry uses the algorithm:

$$y = y_T \left( \frac{x - X1}{X2} \right)$$

Equation 7-42.

The TABLED4 entry has the following format:

1	2	3	4	5	6	7	8	9	10
TABLED4	TID	X1	X2	X3	X4				
	A0	A1	A2	A3	A4	A5	-etc.-	"ENDT"	

Field	Contents
TID	Table identification number.
Xi	Table parameters. ( $X2 \neq 0.0$ ; $X3 < X4$ )
Ai	Coefficients.
ENDT	Ends the table input.

The TABLED4 entry uses the algorithm:

$$y = \sum_{i=0}^N A_i \left( \frac{x - X1}{X2} \right)^i$$

Equation 7-43.

N is the degree of the power series. When  $x < X3$ ,  $X3$  is used for  $x$ ; when  $x > X4$ ,  $X4$  is used for  $x$ . This condition has the effect of placing bounds on the table; there is no extrapolation outside of the table boundaries.

## Spatial Distribution of Loading – DAREA Entry

The EXCITEID field on a TLOAD<sub>i</sub> entry can select DAREA, FORCE<sub>i</sub>, MOMENT<sub>i</sub>, PLOAD, PLOAD1, SELOAD, and SPCD bulk entries. These entries are used to define the degree-of-freedom where the dynamic load is applied and the scale factor that is applied to the loading.

The DAREA entry provides a simple, direct means to define the spatial distribution for a dynamic loading.

1	2	3	4	5	6	7	8	9	10
DAREA	SID	P1	C1	A1	P2	C2	A2		

Field	Contents
SID	Identification number.
P <sub>i</sub>	Grid, extra, or scalar point ID.
C <sub>i</sub>	Component number.
A <sub>i</sub>	Scale factor.

Any number of DAREA entries may be used; all those with the same SID are combined.

## Dynamic Load Set Combination – DLOAD

The SID of all TLOAD1 and TLOAD2 bulk entries must be unique. The DLOAD bulk entry allows you to use multiple TLOAD1 and TLOAD2 entries. The DLOAD entry constructs the total applied load from a linear combination of the individual dynamic loads as follows:

$$\{P\} = S \sum_i S_i \{P_i\}$$

Equation 7-44.

where the index  $i$  refers to the  $i^{\text{th}}$  dynamic load and not the  $i^{\text{th}}$  mode. Specifically:

$P_i$	=	Dynamic load $i$ (either an TLOAD1 or TLOAD2 entry)
$S_i$	=	Scale factor for dynamic load $i$
$S$	=	Overall scale factor
$P$	=	Total applied load

The DLOAD bulk entry has the following format:

1	2	3	4	5	6	7	8	9	10
DLOAD	SID	S	S1	L1	S2	L2	S3	L3	
	S4	L4	-etc.-						

Field	Contents
SID	Load set ID.
S	Overall scale factor.
Si	Individual scale factors.
Li	Identification numbers of TLOAD1 and TLOAD2 entries.

## DLOAD Example

Suppose the following command is in the case control section of the input file:

DLOAD = 33

Also suppose the following entries are in the bulk data section of the input file:

1	2	3	4	5	6	7	8	9	10
\$DLOAD	SID	S	S1	L1	S2	L2			
DLOAD	33	1.0	1.0	35	1.0	36			
\$TLOAD1	SID	EXCITEID	DELAY	TYPE	TID				
TLOAD1	35	29	0.2	LOAD	40				
TLOAD1	36	29	31	LOAD	41				
\$DAREA	SID	GRID	COMP	SCALE					
DAREA	29	30	1	4.0					
\$DELAY	SID	GRID	COMP	LAG					
DELAY	31	30	1	0.2					
\$TABLED1	TID	XAXIS	YAXIS						
\$	x1	y1	x2	y2	x3	y3	x4	y4	
TABLED1	40	LINEAR	LINEAR						
	0.0	0.0	0.3	1.0	2.0	1.0	ENDT		
TABLED1	41	LINEAR	LINEAR						
	0.0	0.0	0.5	0.5	1.0	2.0	ENDT		

In the example:

- The DLOAD case control command selects the loading referenced by the DLOAD bulk entry having SID = 33 as the dynamic loading for the analysis.

- The DLOAD bulk entry combines the dynamic loads defined by two TLOAD1 entries having SIDs of 35 and 36. Neither dynamic load is scaled using the DLOAD entry.
- Both TLOAD1 entries reference the same DAREA entry. Thus, both dynamic loads are applied to the same degree-of-freedom. In this example, it is a single degree-of-freedom, Component 1 of Grid 30. Both dynamic loads are scaled 4.0 times by the DAREA entry.
- Because the dynamic loads are applied at only one degree-of-freedom, the time delay can be defined directly on the TLOAD1 entries. This is the case for the TLOAD1 entry having SID = 35. However, for demonstration purposes, the TLOAD1 entry having SID = 36 references a DELAY bulk entry. Both approaches define a delay of 0.2 for the corresponding dynamic load.
- $F(t)$  for the TLOAD1 entry having SID = 35 is defined by the TABLED1 entry having TID = 40. (See [Figure 7-4.](#))
- $F(t)$  for the TLOAD1 entry having SID = 36 is defined by the TABLED1 entry having TID = 41.

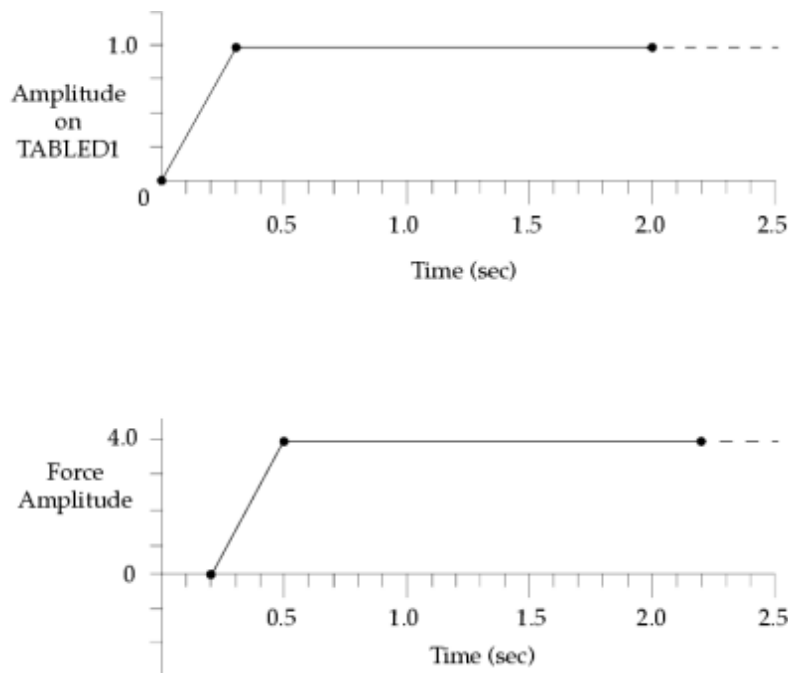


Figure 7-4. Time History for TABLED1 with TID = 40. TABLED1 Data (Top) and Scaled Applied Load (Bottom)

## LSEQ Method

The recommended method to define dynamic loads in a transient response analysis is to use the DLOAD construction outlined in the previous example. However, a legacy method based on the use of the LSEQ bulk entry is still supported.

The LSEQ entry has the following format:

1	2	3	4	5	6	7	8	9	10
LSEQ	SID	EXCITEID	LID	TID					

- The SID field is referenced by a LOADSET case control command.
- The LID field selects static loads like FORCEi, PLOADi, and GRAV entries that have a load set identification number equal to LID. The degree-of-freedom referenced in the static loads define the spatial distribution of the dynamic loading.
- The EXCITEID field selects the TLOADi entries that have the same EXCITEID. These TLOADi entries constitute the temporal distribution of the dynamic loading.

The LSEQ entry links the temporal and spatial distributions so that the appropriate excitation is applied at the correct degree-of-freedom. In effect, it is as if the EXCITEID fields of the LSEQ and TLOADi entries referenced the same DAREA entry, even though a DAREA entry is not present in the input file. **Figure 7-5** depicts the relationships between the various inputs.

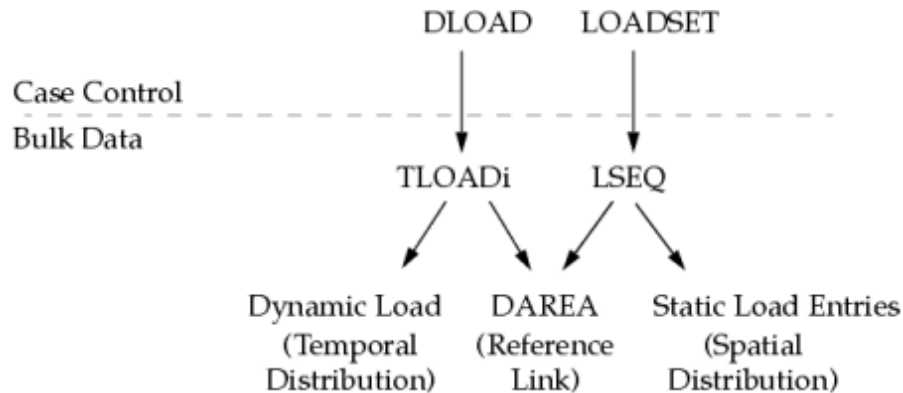


Figure 7-5. Relationship of Dynamic and Static Load Entries

## LSEQ Example

Suppose the following commands are in the case control section of the input file:

LOADSET = 27

DLOAD = 25

Also suppose the following entries are in the bulk data section of the input file:

1	2	3	4	5	6	7	8	9	10
\$LSEQ	SID	EXCITEID	LID						
LSEQ	27	28	26						
\$TLOAD1	SID	EXCITEID	DELAY	TYPE	TID				
TLOAD1	25	28	0.0	LOAD	29				



1	2	3	4	5	6	7	8	9	10
\$FORCE	SID	GRID	CID	F	N1	N2	N3		
FORCE	26	425		2.5	1.0				
\$PLOAD	SID	PRES	GRID1	GRID2	GRID3	GRID4			
PLOAD	26	50.0	63	64	88	91			
\$TABLED1	TID	XAXIS	YAXIS						
\$	x1	y1	x2	y2	x3	y3	x4	y4	
TABLED1	29	LINEAR	LINEAR						
	0.0	0.0	0.3	1.0	2.0	1.0	ENDT		

In the example:

- The LOADSET request in case control selects the LSEQ entry having SID = 27.
- The LSEQ entry references the static loads having SID = 26. These loads include the FORCE and PLOAD entries. The FORCE and PLOAD entries provide the spatial distribution of the dynamic loading.
- The DLOAD request in case control selects the TLOAD1 entry having SID = 25.
- The TLOAD1 entry references a TABLED1 entry having TID = 29. This TABLED1 entry defines  $F(t)$  for the TLOAD1 entry.
- The EXCITEID fields of the LSEQ and TLOAD1 entries are both 28, thereby linking the temporal and spatial distributions of the dynamic loading. Thus, the dynamic load defined by the TLOAD1 entry is:
  - Scaled by 2.5 and applied as a force to Component 1 of Grid 425.
  - Scaled by 50.0 and applied as a pressure to the quadrilateral element face defined by Grids 63, 64, 88, and 91.

## 7.6 Integration Time Step

The TSTEP bulk entry is used to select the integration time step for direct and modal transient response analysis. This entry also controls the duration of the solution and which time steps are to be output. The TSTEP entry is selected by the Set ID referenced on the TSTEP case control command.

The integration time step must be small enough to represent accurately the variation in the loading. The integration time step must also be small enough to represent the maximum frequency of interest. The maximum frequency of interest is often called the cutoff frequency. It is recommended to use at least ten solution time steps per period of response for the cutoff frequency. For a given integration time step, integration errors increase with increasing natural frequency because there is an upper limit to the frequency that can be represented by a given time step. Also, integration errors accumulate with total time.

In both direct and modal transient analysis, the cost of integration is directly proportional to the number of time steps. For example, doubling the load duration doubles the integration effort.

In specifying the duration of the analysis on the TSTEP entry, it is important to use an adequate length of time to properly capture long period (low frequency) response. In many cases, the peak dynamic response does not occur at the peak value of load nor necessarily during the duration of the loading function. As a rule, always solve for at least one cycle of response for the lowest frequency mode after the peak excitation.

You may change  $\Delta t$  during a run but doing so causes the dynamic matrix to be redecomposed, which can be costly in direct transient response analysis.

The TSTEP entry has the following format:

1	2	3	4	5	6	7	8	9	10
TSTEP	SID	N1	$\Delta t_1$	NO1					
		N2	$\Delta t_2$	NO2					
		-etc.-							

Field	Contents
SID	Set identification number.
Ni	Number of time steps of value $\Delta t_i$ .
$\Delta t_i$	Integration time step.
NOi	Output every NOi-th time step.

## 7.7 Transient Excitation Considerations

When applying transient loads, you should consider the following:

- The averaging of applied loads (see [Equation 7-45](#)) in the integration smooths the force and decreases the apparent frequency content. Very sharp spikes in a loading function induce a high-frequency transient response. If the high-frequency transient response is of primary importance in an analysis, a very small integration time step should be used.

$$\left[ \frac{M}{\Delta t^2} \right] (u_{n+1} - 2u_n + u_{n-1}) + \left[ \frac{B}{2\Delta t} \right] (u_{n+1} - u_{n-1}) + \left[ \frac{K}{3} \right] (u_{n+1} + u_n + u_{n-1}) = \frac{1}{3} (P_{n+1} + P_n + P_{n-1})$$

Equation 7-45.

For more information, see [Direct Transient Response Analysis \(SOL 109\)](#).

- The numerical integration of discontinuous forcing functions may cause different results for the same analysis run on different computers because of slight numerical differences on different computer types. If the analysis calls for loadings with sharp impulses, it is best to smooth the impulse over at least one integration time increment.
- The loading function must accurately describe the spatial and temporal distribution of the dynamic load. Simplifying assumptions should not change the character of the load in magnitude, location, or frequency content.

## 7.8 Transient Response Solution Control and Output

The following tables summarize the data entries that can be used to control a transient response analysis. Certain data entries are required, some data entries are optional, while others are user selectable.

In the executive control section of the Simcenter Nastran input file, you must select a solution using the SOL n statement where n is an integer value chosen from [Table 7-36](#).

**Table 7-36. Transient Response Solutions in Simcenter Nastran**

Solution Type	Structured Solution Sequences
Direct	109
Modal	112

In the case control section of the Simcenter Nastran input file, you must select the solution parameters associated with the current analysis like time steps, loads, and boundary conditions, and also the output quantities required from the analysis. The case control commands directly related to transient response analysis are listed in [Table 7-37](#). They can be combined in the standard fashion with the more generic commands like SPC and MPC, among others.

**Table 7-37. Transient Response Case Control Commands**

Case Control Command	Direct or Modal	Description	Required/Optional
DLOAD	Both	Select the dynamic load set from bulk data	Required <sup>(1)</sup>
TSTEP	Both	Select the TSTEP entry from the bulk data	Required
METHOD	Modal	Select the eigenvalue extraction parameters	Required

Case Control Command	Direct or Modal	Description	Required/Optional
LOADSET	Both	Select the LSEQ set from bulk data	Optional
SDAMPING	Modal	Select TABDMP1 entry from bulk data	Optional
IC	Both	Select TIC entries for initial conditions from the bulk data	Optional
OTIME	Both	Select the times for output (default = all)	Optional
<sup>(1)</sup> Not required when using initial conditions.			

A number of bulk entries are unique to transient response analysis. They can be combined with other generic entries in the bulk data section. Bulk entries directly related to transient analysis are summarized in **Table 7-38**.

**Table 7-38. Bulk Entries for Transient Response Analysis**

Bulk Entry	Direct or Modal	Description	Required/Optional
TSTEP	Both	Integration time step and solution control	Required
TLOADi	Both	Dynamic loading	Required <sup>(1)</sup>
EIGR or EIGRL	Modal	Eigenvalue analysis parameters	Required
DLOAD	Both	Dynamic load combination	Optional <sup>(2)</sup>
LSEQ	Both	Dynamic loading from static loads	Optional
TABLEDi	Both	Time-dependent tables for TLOADi	Optional <sup>(3)</sup>
DAREA	Both	Load component and scale factor	Optional
DELAY	Both	Time delay on dynamic load	Optional
TIC	Both	Initial conditions on grid, scalar, and extra points	Optional
TABDMP1	Modal	Modal damping table	Optional
<sup>(1)</sup> Optional in an unforced transient analysis.			
<sup>(2)</sup> Required when multiple TLOADi entries are used.			
<sup>(3)</sup> Required to define time-dependent dynamic loading functions.			

The types of results available from a transient response analysis are similar to those for a static analysis except that the results vary with time. Additional quantities are also available, which is characteristic of dynamics problems. The output quantities are summarized in [Table 7-39](#) and [Table 7-40](#).

**Table 7-39. Grid Point Output from a Transient Response Analysis**

Case Control Command	Description
ACCELERATION	Grid point acceleration time history for a set of grid points
DISPLACEMENT	Grid point displacement time history for a set of grid points
GPSTRESS	Grid point stress time history (requires SURFACE/VOLUME definition in the OUTPUT(POST) section of the case control)
OLOAD	Requests applied load table to be output for a set of grid points
SACCELERATION	Requests solution set acceleration output: d-set in direct solutions and modal variables in modal solutions
SDISPLACEMENT	Requests solution set displacement output: d-set in direct solutions and modal variables in modal solutions
SVECTOR	Requests real eigenvector output for the a-set in modal solutions
SVELOCITY	Requests solution set velocity output: d-set in direct solutions and modal variables in modal solutions
SPCFORCES	Requests forces of single-point constraint for a set of grid points
VELOCITY	Grid point velocity time history for a set of grid points

**Table 7-40. Element and Modal Output from a Transient Response Analysis**

Case Control Command	Description
ELSTRESS (or STRESS)	Element stress time history for a set of elements
ELFORCE (or FORCE)	Element force time history for a set of elements
<b>MODCON</b>	Modal contribution output using the MODCON case control command is used to determine which of the modes has the most influence on the response.
STRAIN	Element strain time history for a set of elements

**Addition output information**

- Because the results may be output for many time steps, the volume of output can be very large. Prudent selection of the output quantities is recommended.
- In transient response analysis, you can specify output in either the SORT1 or SORT2 formats. However, if you specify the SORT2 format, the results are automatically output in both the SORT1 and SORT2 formats.

- The PCOMP or PCOMPG property entries can be used to model a laminate with the shell elements CTRIA3, CTRIA6, TRIAR, CQUAD4, CQUAD8, and CQUADR. The PCOMPS property entry can be used to model a laminate with the solid elements CHEXA and CPENTA. The following output is supported when a shell or solid laminate is included in the transient response solutions 109, 112, and 129. Note that the ply results are not computed in solutions 129 when PARAM,LGDISP,1 is defined. See the remarks on the PCOMP, PCOMPG, and PCOMPS bulk entries in the *Quick Reference Guide* for information on these output requests. Also see the “Laminates” section of the *Simcenter Nastran User’s Guide*.
  - Stress and strain output for the individual lamina in either SORT1 or SORT2 format.
  - Stress resultants, which are requested with the FORCE case control command.
  - Interlaminar shear stress in the bonding material.
  - The failure indices and strength ratios.

Complex stresses and strains are not computed for QUADR and TRIAR elements that are used to model laminates when SYSTEM(370)=1.

## 7.9 Examples of Transient Response Analysis

This section provides several examples showing the input and output. These examples are:

Model	Transient Response Bulk Entries	Output
bd06two	TSTEP, TIC	X-Y plots
bd06bar	EIGRL, TSTEP, TABDMP1, DLOAD, TLOAD2, DAREA, DELAY	X-Y plots
bd06bkt	EIGRL, TSTEP, TABDMP1, TLOAD1, LSEQ, TABLED1, PLOAD4	X-Y plot

### Two-DOF Model

Consider the two-DOF system shown in [Figure 7-6](#). Direct transient response (SOL 109) is run with an initial displacement of 0.1 meter at Grid 2. The analysis is run for a duration of 10 seconds with  $\Delta t = 0.01$  second. Assume damping is negligible. A portion of the input file is shown below.

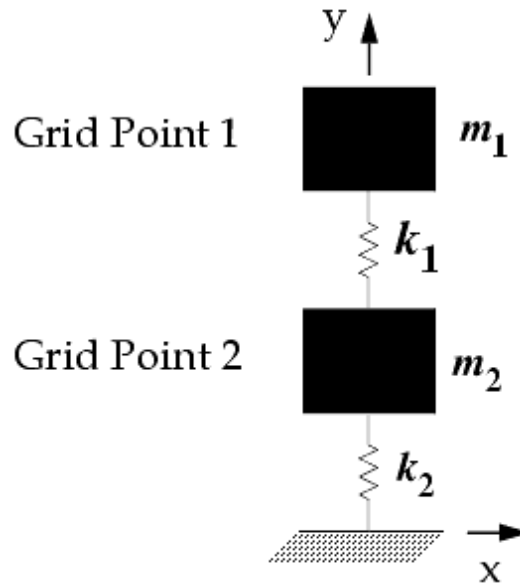


Figure 7-6. Two-DOF Model

```

$ FILE bd06two.dat
$
$ TWO-DOF SYSTEM
$ CHAPTER 6, TRANSIENT RESPONSE
$
TIME 5
SOL 109 $ DIRECT TRANSIENT RESPONSE
CEND
TITLE = TWO-DOF SYSTEM
SUBTITLE = DIRECT FREQUENCY RESPONSE
LABEL = INITIAL DISPL. AT GRID 2
$
$ SPECIFY SPC
SPC = 996
$
$ SPECIFY DYNAMIC INPUT
TSTEP = 888
IC = 777
$
$ SELECT OUTPUT
SET 11 = 1,2
DISPLACEMENT(PLOT) = 11
$
$ XYPLOTS
$
... X-Y plot commands ...
$
BEGIN BULK
$
$. . . . .2. . . . .3. . . . .4. . . . .5. . . . .6. . . . .7. . . . .8. . . . .9. . . . .10. . . . $
$
$ ENTRIES FOR TRANSIENT RESPONSE
$
$ INITIAL CONDITION
$TIC SID G C U0 V0

```

```

TIC      777      2      2      0.1
$
$  TIME STEP
$TSTEP  SID      N1      DT1      NO1
TSTEP   888      1000   0.01    1
$
... basic model ...
$
ENDDATA

```

**Figure 7-7. Input File (Abridged) for the Two-DOF Example**

**Table 7-42** shows the relationship between the case control commands and the bulk entries. This example represents the simplest form of dynamic response input. The only required entries are those that define the time step and the initial conditions. Any unspecified initial conditions are assumed to be zero.

**Table 7-42. Relationship Between Case Control Commands and Bulk Entries for the Two-DOF Model**

Case Control	Bulk
TSTEP	TSTEP
IC	TIC

**Figure 7-8** shows the plots of the resulting displacements for Grids 1 and 2. There are two frequencies of response: a higher frequency of about 5 Hz, and a lower frequency of about 0.25 Hz. The energy (and hence response) appears to be transferred repetitively between Grids 1 and 2 as represented by the lower frequency response. This energy transfer is called beating. Beating occurs when there are closely-spaced modes (in this case, 4.79 Hz and 5.29 Hz) in which energy transfer can readily occur. The response is comprised of two frequencies as given below:

$$f_{higher} = \frac{f1 + f2}{2}$$

$$f_{lower} = \frac{f2 - f1}{2}$$

**Equation 7-46.**

where:

- f1 = lower of the closely-spaced mode frequencies  
f2 = higher of the closely-spaced mode frequencies

In this example,  $f_{higher}$  is 5.04 Hz and  $f_{lower}$  is 0.25 Hz. The lower frequency is called the beat frequency and is the frequency at which energy transfer occurs.



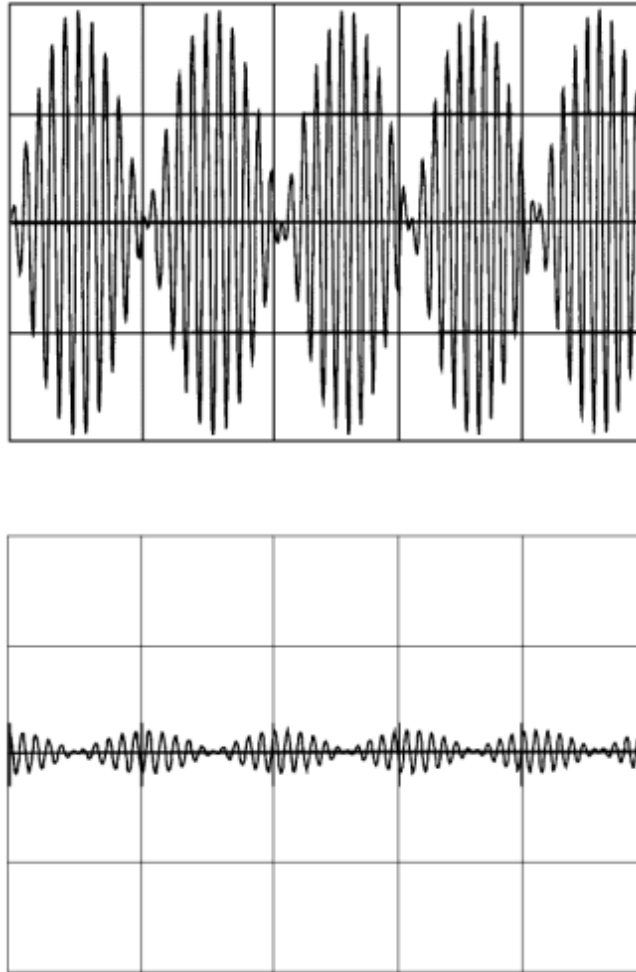


Figure 7-8. Displacements of Grids 1 and 2

## Cantilever Beam Model

Consider the cantilever beam shown below. Modal transient response (SOL 112) is run with loads applied to Grids 6 and 11 as shown in [Figure 7-10](#). The analysis is run for a duration of 2 seconds with  $\Delta t = 0.001$  second. Modal damping of 5% critical damping is used for all modes. Modes up to 3000 Hz are computed using the Lanczos method. [Figure 7-11](#) shows a portion of the input file.

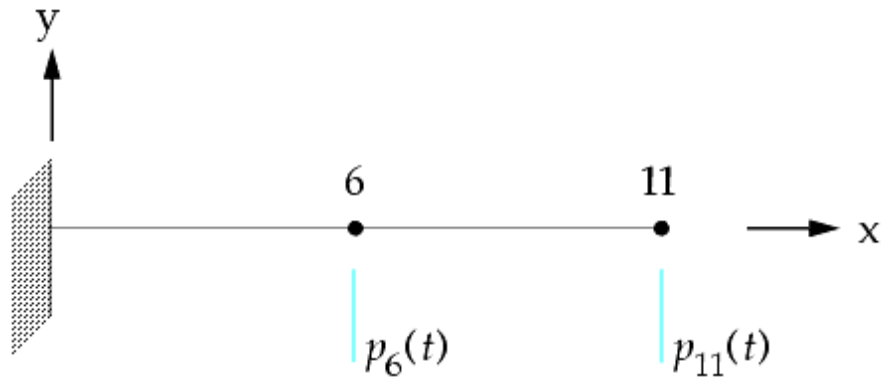


Figure 7-9. Cantilever Beam Model with Applied Loads

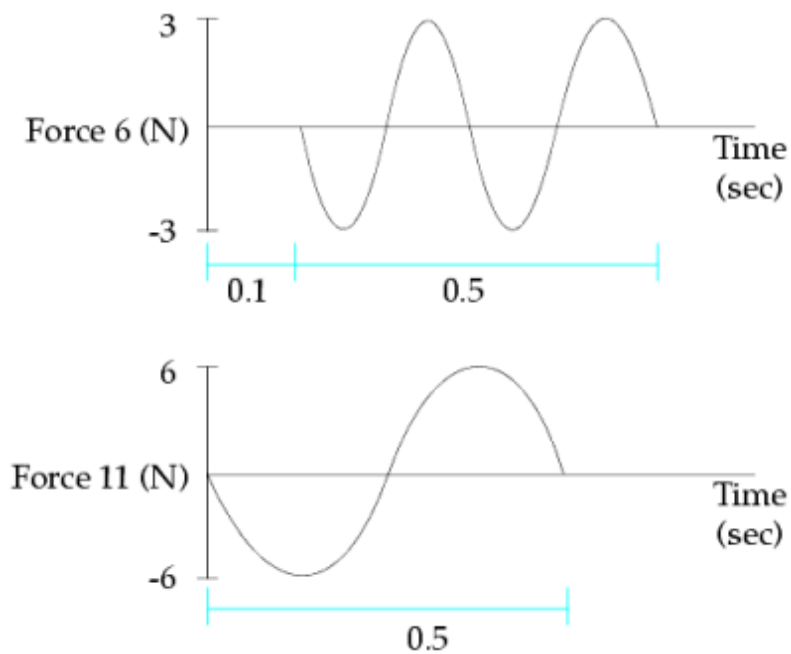


Figure 7-10. Applied Loads for the Cantilever Beam Model

```

$ FILE bd06bar.dat
$
$ CANTILEVER BEAM MODEL
$ CHAPTER 6, TRANSIENT RESPONSE
$
SOL 112      $ MODAL TRANSIENT RESPONSE
TIME 10
CEND
TITLE = CANTILEVER BEAM
SUBTITLE = MODAL TRANSIENT RESPONSE
$
SPC = 21
DLOAD = 22
TSTEP = 27
SDAMPING = 25
$

```

```

METHOD = 10
$
$ PHYSICAL OUTPUT REQUEST
SET 11 = 6,11
DISPLACEMENT(PLOT) = 11
ACCELERATION(PLOT) = 11
$
$ MODAL SOLUTION SET OUTPUT
SET 12 = 1,2
SDISP(PLOT) = 12
$
$ ELEMENT FORCE OUTPUT
SET 13 = 6
ELFORCE(PLOT) = 13
$
$ APPLIED LOAD OUTPUT
SET 15 = 6,11
OLOAD(PLOT) = 15
$
$ XYPLOTS
$
... X-Y plot commands ...
$
BEGIN BULK
$
$. . . . . 2 . . . . . 3 . . . . . 4 . . . . . 5 . . . . . 6 . . . . . 7 . . . . . 8 . . . . . 9 . . . . . 10 . . . . . $
$
$EIGRL  SID      V1      V2      MSGLVL
EIGRL   10      -0.1    3000.    0
$
$TSTEP  SID      N1      DT1     NO1
TSTEP   27      2000    0.001    1
$
$ MODAL DAMPING OF 5% IN ALL MODES
$TABDMP1 TID     TYPE
$+TABD   F1      G1      F2      G2      ETC.
TABDMP1  25     CRIT
+TABD    0.     0.05   1000.   0.05   ENDT
$
$ DYNAMIC LOADING
$DLOAD  SID      S      S1      L1      S2      L2
DLOAD   22      1.0    1.0    231    1.0    232
$TLOAD2 SID     DAREA  DELAY  TYPE    T1      T2      F      P      +TL1
$+TL1   C      B
TLOAD2  231    241          0      0.0    0.5    2.0    90.
TLOAD2  232    242    262    0      0.0    0.5    4.0    90.
$DAREA  SID     P1     C1     A1
DAREA   241    11     2      6.0
DAREA   242    6      2      3.0
$DELAY  SID     P1     C1     T1
DELAY   262    6      2      0.1
$
... basic model ...
$
ENDDATA

```

Figure 7-11. Input File (Abridged) for the Cantilever Beam Model

**Table 7-44** shows the relationship between the case control commands and the bulk entries. The DLOAD bulk entry references two TLOAD2 entries, each of which references separate DAREA entries. A TLOAD2 entry also references a DELAY entry to define the time delay for the load at Grid 6.

**Table 7-44. Relationship Between Case Control Commands and Bulk Entries for the Cantilever Beam Model**

Case Control	Bulk Entries
METHOD	EIGRL
TSTEP	TSTEP
SDAMPING	TABDMP1
DLOAD	DLOAD <ul style="list-style-type: none"> <li>→ TLOAD2 231               <ul style="list-style-type: none"> <li>→ DAREA</li> </ul> </li> <li>→ TLOAD2 232               <ul style="list-style-type: none"> <li>→ DAREA</li> <li>→ DELAY</li> </ul> </li> </ul>

Plotted output is shown in the following figures. **Figure 7-12** shows the applied loads at Grids 6 and 11. **Figure 7-13** shows the plots of the displacements for Grids 6 and 11. **Figure 7-14** shows the accelerations for Grids 6 and 11. **Figure 7-15** shows the bending moment at end A in plane 1 for Element 6. **Figure 7-16** shows the modal displacements for Modes 1 and 2.

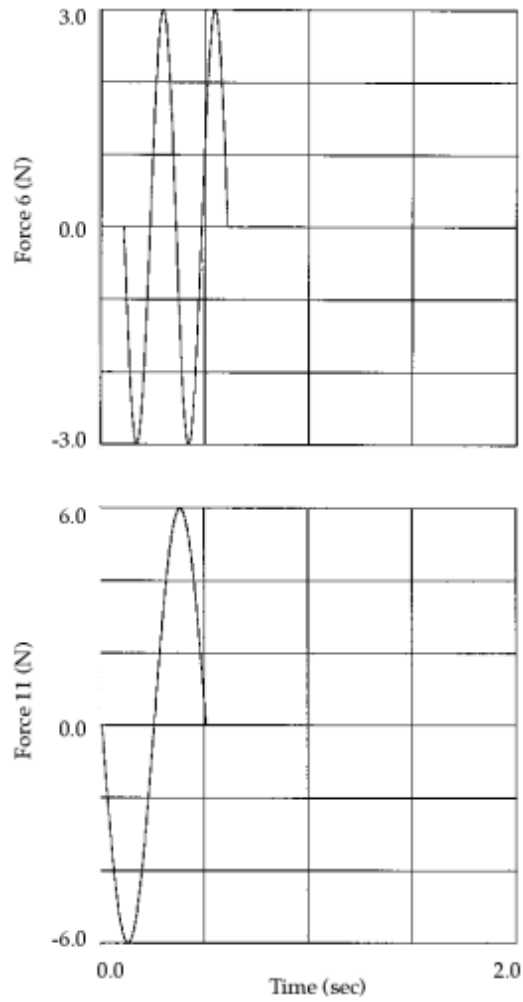


Figure 7-12. Applied Loads at Grids 6 and 11

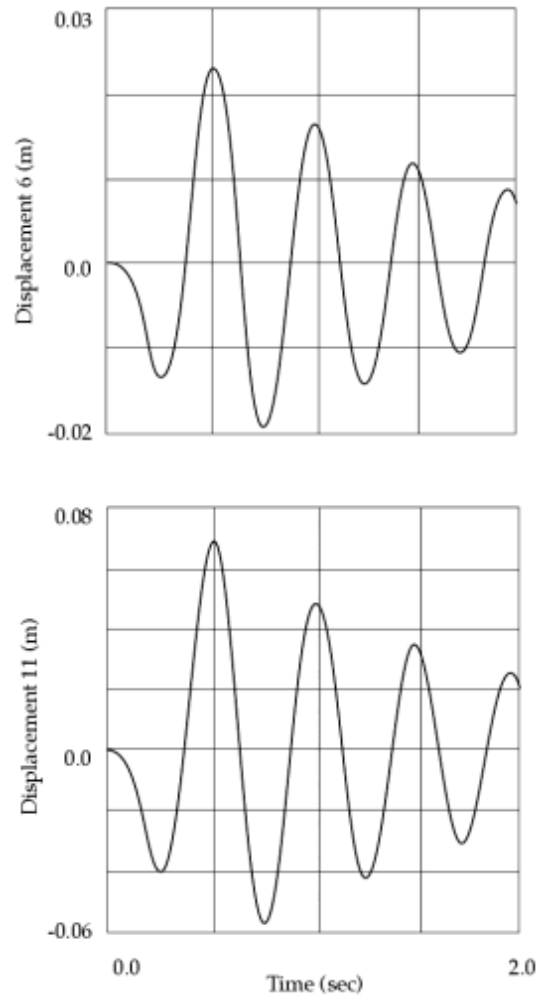


Figure 7-13. Displacements at Grids 6 and 11

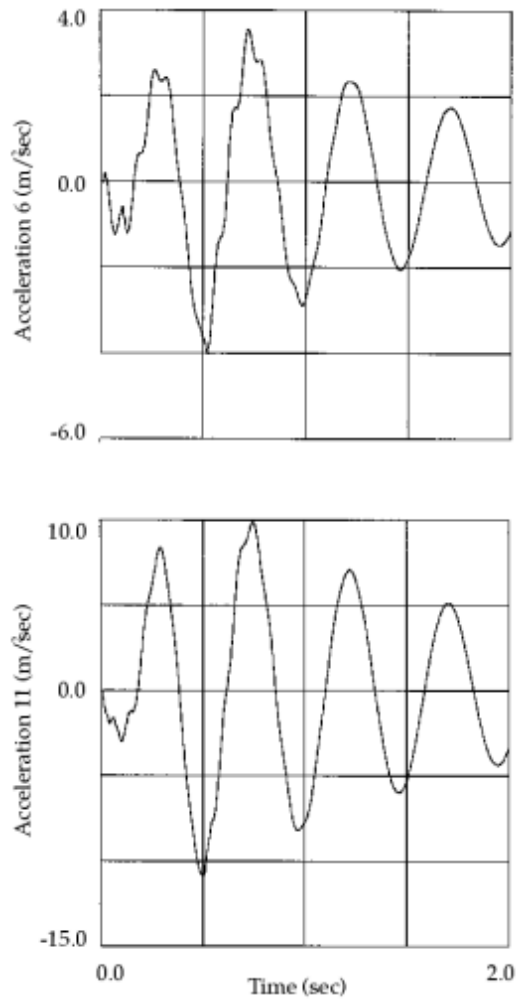


Figure 7-14. Accelerations at Grids 6 and 11

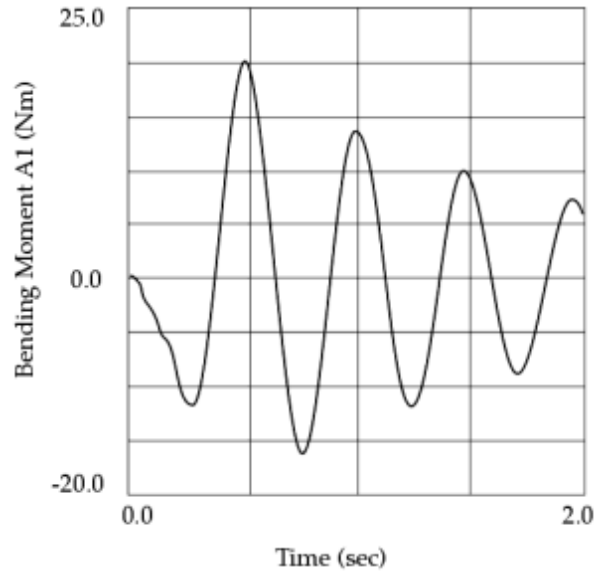


Figure 7-15. Bending Moment A1 for Element 6

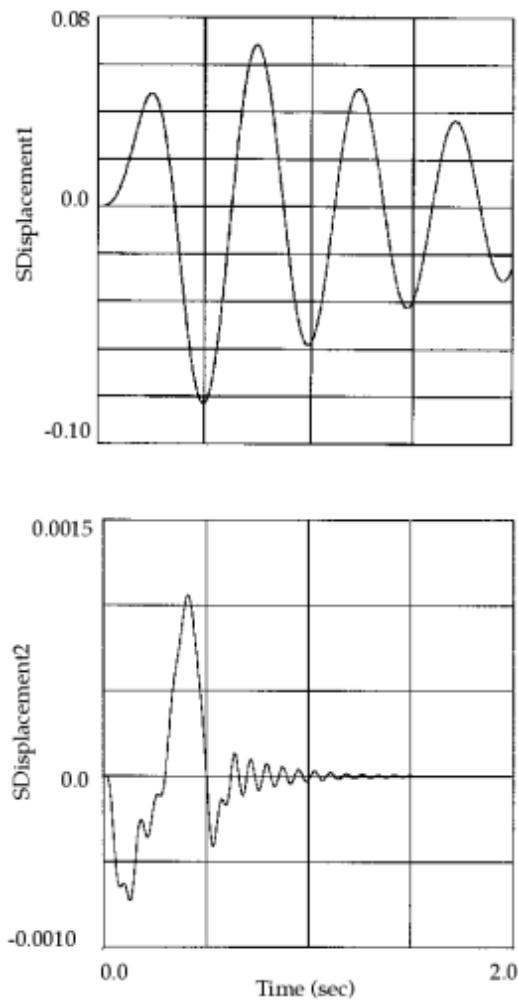


Figure 7-16. Modal Displacements for Modes 1 and 2



## Bracket Model

Consider the bracket model shown in **Figure 7-17**. A pressure load of 3 psi is applied to the elements on the top face in the z-direction with the time history shown in **Figure 7-18**. The modal transient analysis is run for 4 seconds with a time step size of 0.005 second. Modal damping of 2% critical damping is used for all modes. Modes up to 3000 Hz are computed with the Lanczos method. The model is constrained near the base.

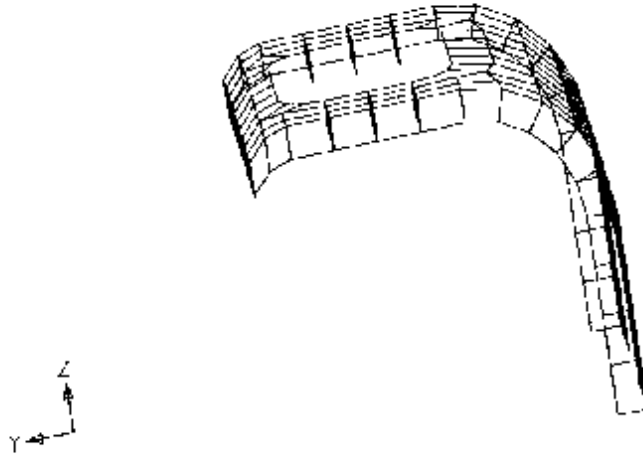


Figure 7-17. Bracket Model

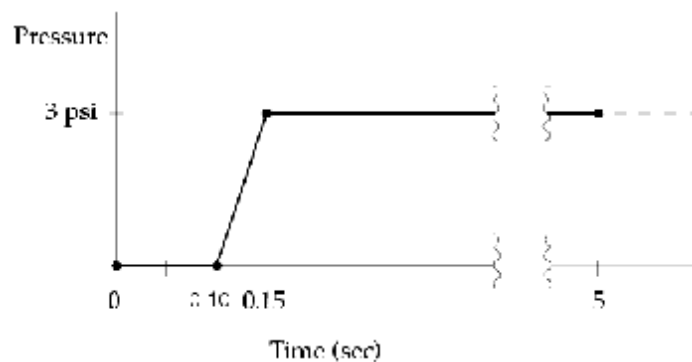


Figure 7-18. Time Variation for Applied Load

**Figure 7-19** shows the abridged input file. The LSEQ entry is used to apply the pressure loads (PLOAD4 entries). The LSEQ and TLOAD1 entries reference a common DAREA ID (999) and that there is no explicit DAREA entry. **Table 7-45** shows the relationship between the case control commands and the bulk entries.

```

$ FILE bd06bkt.dat
$
$ BRACKET MODEL
$ CHAPTER 6, TRANSIENT RESPONSE
$
SOL 112 $ MODAL TRANSIENT RESPONSE
TIME 100
CEND
TITLE = BRACKET MODEL
SUBTITLE = MODAL TRANSIENT RESPONSE ANALYSIS
$
SPC = 1
$
METHOD = 777
$
DLOAD = 2
LOADSET = 3
SDAMPING = 4
TSTEP = 5
$
$ OUTPUT REQUEST
SET 123 = 999
DISPLACEMENT(PLOT)=123
$
$ XYPLOTS
$
... X-Y plot commands ...
$
BEGIN BULK
$
$. . . . . 2 . . . . . 3 . . . . . 4 . . . . . 5 . . . . . 6 . . . . . 7 . . . . . 8 . . . . . 9 . . . . . 10 . . . . .
$
$ NORMAL MODES TO 3000 HZ
$EIGRL SID V1 V2
EIGRL 777 -0.1 3000.
$
$ 4 SECONDS OF RESPONSE
$TSTEP SID N1 DT1 NO1
TSTEP 5 800 0.005 1
$
$ MODAL DAMPING OF 2% CRITICAL
$TABDMP1 TID TYPE +TABD1
$+TABD1 F1 G1 F2 G2 ETC.
TABDMP1 4 CRIT +TABD1
+TABD1 0.0 0.02 3000.0 0.02 ENDT
$
$ LOAD DEFINITION
$
$TLOAD1 SID DAREA DELAY TYPE TID
TLOAD1 2 999
$
$LSEQ SID DAREA LID TID
LSEQ 3 999 1
$
$ TIME HISTORY
$TABLED1 TID +TABL1
$+TABL1 X1 Y1 X2 Y2 ETC.
TABLED1 22 +TABL1
+TABL1 0.0 0.0 0.1 0.0 0.15 1.0 5.0 1.0 +TABL2
+TABL2 ENDT

```

```

$
$ PRESSURE LOAD OF 3 PSI PER ELEMENT
$PLOAD4 SID      EID      P1
PLOAD4  1        171      -3.
PLOAD4  1        172      -3.
PLOAD4  1        160      -3.
etc.
$
... basic model ...
$
ENDDATA

```

Figure 7-19. Input File (Abridged) for the Bracket Model

Table 7-45. Relationship Between Case Control Commands and Bulk Entries for the Bracket Model



Case Control	Bulk Data
METHOD	EIGRL
FREQUENCY	FREQ1
SDAMPING	TABDMP1
LOADSET	
DLOAD	

Figure 7-20 shows a plot of the z-displacement of Grid 999, which is the concentrated mass at the center of the cutout.

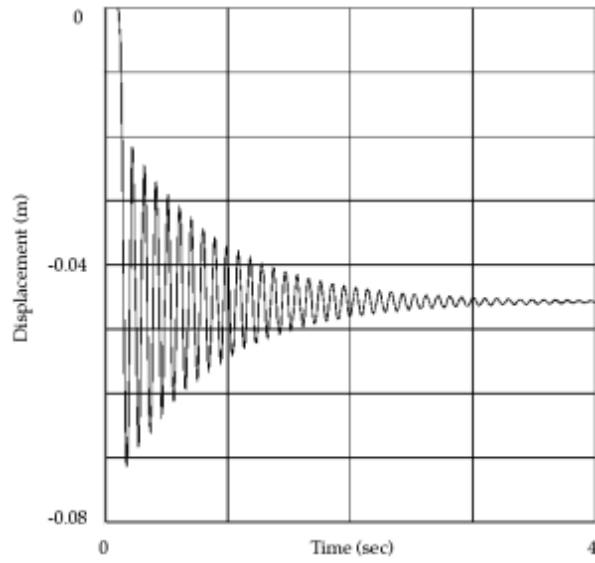


Figure 7-20. Displacement Time History for Grid 999

# 8. Enforced Motion

## 8.1 Overview of Enforced Motion

Enforced motion specifies the displacement, velocity, and/or acceleration at a set of grid points for frequency or transient response. Enforced motion is used when base motion is specified instead of, or in conjunction, with applied loads. A common application is an earthquake excitation applied to a building. In this example there are no applied loads. Instead the base of the building undergoes an enforced displacement or acceleration time history.

Simcenter Nastran supports several enforced motion options.

- **SPC/SPCD methods**  
The SPC/SPCD methods of enforced motion allow you to directly specify displacements, velocities or accelerations via SPC/SPC1/SPCD data. There are two SPC/SPCD method choices: the constraint mode method which uses both normal mode shapes and constraint mode shapes to calculate the response, and the absolute method which only uses the normal mode shapes.
- **SPCF method**  
The SPCF method of enforced motion allows you to apply a large number of frequency-dependent enforced motions on structural grid points. For the enforced motions, you use displacement results from an OUTPUT2 (.op2) file that was generated from a previous frequency response analysis. The same two SPC/SPCD methods apply to the SPCF method.  
The displacement coordinate systems of the grids in the referenced OP2 file and grids in your current model must be identical, and the subcases in the referenced OP2 file and subcases in the current solution must match.
- **Large Mass method**  
With the large mass method, you connect a mass which is much larger than the mass of the structure to a degree-of-freedom, then apply a force to the degree-of-freedom. The structure receives the applied force as an enforced acceleration, velocity or displacement.

## 8.2 The SPC/SPCD Methods

The SPC/SPCD methods of enforced motion allow you to directly specify displacements, velocities or accelerations via SPC/SPC1/SPCD data. The SPC/SPCD methods are supported in direct and modal frequency analysis (Solutions 108 and 111), direct and modal transient analysis (Solutions 109 and 112), and in design optimization (Solution 200).

There are two SPC/SPCD method choices: the constraint mode method and the absolute method. The system cell 422 (ENFMOTN) is used to select the method as described below.

- The constraint mode method is selected with the system cell ENFMOTN=0 (default).

The constraint mode method uses both normal mode shapes and constraint mode shapes to calculate the response. Residual vectors can be used with the SPC/SPCD constraint mode method but are not critical for accuracy.

- The absolute method is selected with the system cell ENFMOTN=1 (does not include modal viscous damping coupling) or ENFMOTN=2 (includes modal viscous damping coupling). The absolute method uses only normal mode shapes to calculate the response. Residual vectors are generally needed for accurate stress, strain, and force response when using the absolute method. When modal viscous damping is included in the model, it is important to use the ENFMOTN=2 option in order to include the damping coupling term.

The constraint mode method and the absolute method give almost identical results as long as the absolute method includes residual vectors, and ENFMOTN=2 is used when modal viscous damping exists. Modal energy output requests using the MODALE case control command are not supported by the constraint mode method. The absolute or large mass methods must be used if this is requested.

Issues that can arise when using the constraint mode method include the following:

- The constraint mode method is sensitive to very high acceleration inputs. For example, an enforced displacement that has a large change in value over a small time step will produce large acceleration inputs. This can produce a “bump” in the response calculations around the time that this “discontinuity” occurs. Thus, it is recommended that such large acceleration inputs be avoided, especially when they occur over a single time step. If such a “discontinuity” is required, it is recommended that a very small time step be used to reduce the magnitude of the acceleration and lesson the “bump” that occurs near the “discontinuity”.
- The constraint mode method relies on stiffness being present at the point where the enforced motion is applied. If stiffness is not present, load is not transmitted.

Note:

A fatal error occurs if the constraint mode method is used with the NONLINEAR case control command.

The constraint mode method can be computationally expensive in large models when the number of SPCDs is greater than 200. When the model contains large numbers of SPCDs, as a best practice, use the absolute method. To use the absolute method, specify SYSTEM(422)=1 or 2. However, this does not apply when PARAM,SPCSTR,YES is specified, because, for this case, structural analysis is not performed.

### 8.3 The SPC/SPCD Methods in Frequency Response Analysis

The fundamental assumption in frequency response analysis is that transient effects are negligible, and the time-dependent nature of the loads can be expressed purely in terms of harmonic forcing functions. The forced response of the structure to these harmonic loads thus occurs at the same frequency, and in proportion to the magnitude of the applied loads.

If instead of an harmonic force, you applied an enforced motion, the effect would be similar. That is, a response with proportional forces of constraint, at the same frequency of the enforcing motion. Any one of the enforced displacement, velocity, or acceleration must uniquely determine the other two (because they differ only by multiples of frequency), with resultant forces of constraint derived from a solution of the governing equations.

To illustrate this, an applied harmonic forcing function of the form:

$$P(t) = P(\omega) \cdot e^{i\omega t}$$

results in the in-phase displacement

$$u(t) = U(\omega) \cdot e^{i\omega t}$$

with the corresponding velocity and acceleration

$$\dot{u}(t) = i\omega U(\omega) \cdot e^{i\omega t}$$

and

$$\ddot{u}(t) = -\omega^2 U(\omega) \cdot e^{i\omega t}$$

Here they are written after multipoint constraint partitioning operations have been performed leaving just the free (f-set) and constrained (s-set) degrees-of-freedom

$$\left( -\omega^2 \begin{bmatrix} M_{ff} & M_{fs} \\ M_{sf} & M_{ss} \end{bmatrix} + i\omega \begin{bmatrix} B_{ff} & B_{fs} \\ B_{sf} & B_{ss} \end{bmatrix} + \begin{bmatrix} K_{ff} & K_{fs} \\ K_{sf} & K_{ss} \end{bmatrix} + i \begin{bmatrix} \tilde{K}_{ff} & \tilde{K}_{fs} \\ \tilde{K}_{sf} & \tilde{K}_{ss} \end{bmatrix} \right) \begin{Bmatrix} U_f \\ U_s \end{Bmatrix} = \begin{Bmatrix} P_f \\ P_s + q_s \end{Bmatrix}$$

Equation 8-1.

The set nomenclature is simplified with  $f$  representing the free physical degrees of freedom (DOF), and  $s$  representing the SPCD enforced motion DOF, thus

$u_f$  = absolute displacements of the f-set DOF

$u_s$  = applied enforced motion of the s-set DOF

$M$ ,  $B$ ,  $K$ , and  $\tilde{K}$  are the mass, viscous damping, stiffness, and structural damping matrices. The vector  $P$  is the applied force load. The structural damping matrix,  $\tilde{K}$ , consists of the uniform and elemental structural damping components so that

$$\tilde{K} = GK + \sum G_E K_E$$

where  $G$  is the uniform structural damping coefficient,  $G_E$  are elemental structural coefficients, and  $K_E$  are elemental stiffness matrices.

### Frequency Response with the Absolute Method

If the constraints specify zero motion ( $U_s = \{0\}$ ), the solution for the free degrees-of-freedom may be obtained directly from the upper part of **Equation 8-1** as:

$$(-\omega^2 M_{ff} + i\omega B_{ff} + K_{ff} + i\tilde{K}_{ff})U_f = P_f$$

Equation 8-2.

and the corresponding constraint forces from the lower part of **Equation 8-1** as:

$$q_s = -P_s + (-\omega^2 M_{sf} + i\omega B_{sf} + K_{sf} + i\tilde{K}_{sf})U_f$$

Equation 8-3.

If enforced displacements, velocities, or accelerations are applied,  $U_s \neq \{0\}$  and the free degrees-of-freedom, again from **Equation 8-1** are:

$$(-\omega^2 M_{ff} + i\omega B_{ff} + K_{ff} + i\tilde{K}_{ff})U_f = P_f - (-\omega^2 M_{fs} + i\omega B_{fs} + K_{fs} + i\tilde{K}_{fs})U_s$$

Equation 8-4.

With corresponding constraint forces:

$$q_s = -P_s + (-\omega^2 M_{sf} + i\omega B_{sf} + K_{sf} + i\tilde{K}_{sf})U_f + (-\omega^2 M_{ss} + i\omega B_{ss} + K_{ss} + i\tilde{K}_{ss})U_s$$

Equation 8-5.

Comparing **Equation 8-4** and **Equation 8-5** with **Equation 8-2** and **Equation 8-3** shows that the enforced motion modifies the forces applied to the f-set degrees-of-freedom. Note also that, were enforced velocities to be applied instead, the resultant applied displacement would differ by a factor of  $(1/i\omega)$ , and by  $(1/\omega^2)$  in the case of an applied acceleration.

### Frequency Response with the Constraint Mode Method



The formulation for the constraint mode method is obtained by separating the absolute displacement into a static enforced motion component,  $-K_{ff}^{-1} K_{fs} u_s$ , and a dynamic relative displacement component,  $v_f$ , as

$$u_f = v_f + G_{fs} u_s$$

where

$$G_{fs} = -K_{ff}^{-1} K_{fs}$$

are the constraint modes. Thus, the constraint mode method relies on stiffness being present at the point where the enforced motion is applied. If stiffness is not present, load is not transmitted.

Substituting into the equation of motion, a new equation of motion in terms of the dynamic displacement component is obtained

$$(-\omega^2 M_{ff} + i\omega B_{ff} + K_{ff} + i\tilde{K}_{ff})v_f = P_f + [\omega^2 (M_{fs} + M_{ff}G_{fs}) - i\omega (B_{fs} + B_{ff}G_{fs}) - i(\tilde{K}_{fs} + \tilde{K}_{ff}G_{fs})]u_s$$

The above equation is solved for  $v_f$  and then the absolute displacement is obtained by adding the static enforced motion displacement.

Since the constraint mode method uses static mode shapes, it is not as critical that residual vectors be computed as for the absolute method. But it is still possible to use residual vectors with the constraint mode method. In that case, residual vectors are computed for inertial, damping, and structural damping loads. Thus, the residual vectors for the constraint mode method in a frequency response solution are computed from the following terms

$$\begin{aligned} &-\omega^2 (M_{fs} + M_{ff}G_{fs})u_s \\ &i\omega (B_{fs} + B_{ff}G_{fs})u_s \\ &i(\tilde{K}_{fs} + \tilde{K}_{ff}G_{fs})u_s \end{aligned}$$

## 8.4 The SPC/SPCD Methods in Transient Response Analysis

Because they are written in terms of displacement and its higher-order time derivatives, the solution of the transient dynamic equations of motion must be performed in a stepwise integral fashion for every time step,  $t$ . As with frequency response, the implications for constrained motion are that any enforced displacement, velocity, or acceleration uniquely defines the other two quantities for that degree-of-freedom, with the necessary forces of constraint determined by solution of the governing equations of motion at the particular time step(s) of interest. Unlike frequency response, however, higher-order displacement derivatives for the enforced degrees-of-freedom must be determined by finite difference,

with lower-order quantities (for example, displacements and velocities for an applied acceleration) determined by numerical integration.

### Transient Response with the Absolute Method

After multipoint constraint partitioning, the transient dynamic equations of motion, expressed in terms of the free (f-set) and constrained (s-set) degrees-of-freedom are:

$$\begin{bmatrix} M_{ff} & M_{fs} \\ M_{sf} & M_{ss} \end{bmatrix} \begin{Bmatrix} \ddot{u}_f \\ \ddot{u}_s \end{Bmatrix} + \begin{bmatrix} B_{ff} & B_{fs} \\ B_{sf} & B_{ss} \end{bmatrix} \begin{Bmatrix} \dot{u}_f \\ \dot{u}_s \end{Bmatrix} + \begin{bmatrix} K_{ff} & K_{fs} \\ K_{sf} & K_{ss} \end{bmatrix} \begin{Bmatrix} u_f \\ u_s \end{Bmatrix} = \begin{Bmatrix} P_f(t) \\ P_s(t) + q_s(t) \end{Bmatrix}$$

Equation 8-6.

In the case of zero constrained motion, the solution for the free degrees-of-freedom are available directly from **Equation 8-6** as:

$$M_{ff} \ddot{u}_f + B_{ff} \dot{u}_f + K_{ff} u_f = P_f(t)$$

Equation 8-7.

with corresponding forces of constraint from:

$$q_s(t) = -P_s(t) + (M_{sf} \ddot{u}_f + B_{sf} \dot{u}_f + K_{sf} u_f)$$

Equation 8-8.

In the case of enforced motion, the solution for the free degrees-of-freedom is again from **Equation 8-6**:

$$M_{ff} \ddot{u}_f + B_{ff} \dot{u}_f + K_{ff} u_f = P_f(t) - (M_{sf} \ddot{u}_s + B_{sf} \dot{u}_s + K_{fs} u_s)$$

Equation 8-9.

with constraint forces,

$$q_s(t) = -P_s(t) + [M_{sf} \quad M_{ss}] \begin{Bmatrix} \ddot{u}_f \\ \ddot{u}_s \end{Bmatrix} + [B_{sf} \quad B_{ss}] \begin{Bmatrix} \dot{u}_f \\ \dot{u}_s \end{Bmatrix} + [K_{sf} \quad K_{ss}] \begin{Bmatrix} u_f \\ u_s \end{Bmatrix}$$

Equation 8-10.

As with frequency response, note that the effect of enforced motion is to modify the loads on the f-set, and the s-set forces of constraint.

### Transient Response with the Constraint Mode Method

The absolute SPC/SPCD method has the following equation of motion:

$$M_{ff} \ddot{u}_f + B_{ff} \dot{u}_f + K_{ff} u_f = P_f(t) - (M_{sf} \ddot{u}_s + B_{sf} \dot{u}_s + K_{fs} u_s)$$

As with frequency response, the formulation for the constraint mode method in transient response is obtained by separating the absolute displacement into a static enforced motion component,

$-K_{ff}^{-1} K_{fs} u_s$ , and a dynamic relative displacement component,  $v_f$ , as

$$u_f = v_f + G_{fs} u_s$$

where

$$G_{fs} = -K_{ff}^{-1} K_{fs}$$

are the constraint modes.

Substituting into the absolute equation of motion, a new equation of motion in terms of the dynamic displacement component is obtained

$$M_{ff} \ddot{v}_f + B_{ff} \dot{v}_f + K_{ff} v_f = P_f - (M_{fs} + M_{ff} G_{fs}) \ddot{u}_s - (B_{fs} + B_{ff} G_{fs}) \dot{u}_s$$

The residual vectors for the constraint mode method in a transient response solution are computed from the following terms

$$\begin{aligned} & (M_{fs} + M_{ff} G_{fs}) \ddot{u}_s \\ & (B_{fs} + B_{ff} G_{fs}) \dot{u}_s \end{aligned}$$

## 8.5 SPC/SPCD Method Inputs and Procedure

The TYPE field (field 5) on the TLOAD1 and TLOAD2 entries specifies if an enforced displacement, velocity, or acceleration are applied for a transient response solution. Similarly, the TYPE field (field 8) on the RLOAD1 and RLOAD2 entries specifies if an enforced displacement, velocity, or acceleration are applied for a frequency response solution.

Value	Excitation Function
0, or LOAD	Applied load (Default)
1, or DISP	Enforced displacement using large mass or SPC/SPCD data
2, or VELO	Enforced velocity using large mass or SPC/SPCD data
3, or ACCE	Enforced acceleration using large mass or SPC/SPCD data

The character fields may be shortened to as little as a single character on input, if desired.

In order to allow maximum input flexibility and to support existing input data files which had used the large mass method, the decision regarding whether to use the older, large mass method, or the newer direct enforced motion will be made based on a search for SPC/SPC1/SPCD data as follows: If the TYPE field on the TLOAD1/2 entries indicates an enforced motion, the EXCITEID field will first be assumed to point to SPC-type data. If not present, the software will then assume the excitation is enforced motion using large masses and will then look for DAREA and various static loading data, just as in the case of applied loads. RLOAD1 and RLOAD2 entries will only look for SPC-type data in the case of enforced motion. For all entries (TLOAD1/2, RLOAD1/2), if the TYPE field indicates an applied load, the program will search only for static loading data.

With the SPC/SPCD methods, enforced motion data is supplied via the SPC/SPC1/SPCD bulk entries. (Note that components specified in SPCD data must be also be referenced on SPC or SPC1 entries.) Enforced motion could be defined solely with SPC data (without the use of SPCD's), but such usage is discouraged in favor of SPC/SPC1 data in connection with SPCD's. In this way, SPCD entries are selected via the EXCITEID field in the dynamic load data, while the SPC/SPC1 sets are used to identify the constrained displacement set in case control. With SPC data alone, enforced motion is applied solely via case control. The entries referenced by the EXCITEID field on the TLOAD1/2 or RLOAD1/2 entries do not have to exist. This method of specifying enforced motion is less elegant and somewhat more difficult to interpret.

### Procedure for Using the SPC/SPCD Enforced Motion

In summary, the procedure for applying enforced motion in a dynamic analysis:

1. Specify the appropriate type of enforced motion via the TYPE field on the RLOAD1/RLOAD2 (field 8) or TLOAD1/TLOAD2 (field 5) bulk entries.
2. Define the desired enforced motion using the SPCD Bulk Data entry. The set IDs of these SPCD data must match the IDs appearing in the EXCITEID fields of the corresponding dynamic load data in step 1.
3. Ensure that the components referenced in the SPCD bulk entry above are also specified in the SPC1 bulk entry and this SPC1 is also selected in the case control.
4. Define system cell 422 (ENFMOTN) to the appropriate SPC/SPCD method. The default (ENFMOTN=0) is the constraint mode method.
5. Ensure PARAM,RESVEC is set to YES in the bulk data section if the absolute method is used.

The following simple examples illustrate these steps for a frequency and transient response analysis, respectively.

### SPC/SPCD Enforced Motion Input Example in Frequency Response Analysis

```

.
.
.
SOL 108
CEND
TITLE = SPECIFICATION OF ENFORCED DISPLACEMENT
$ THE FOLLOWING REQUEST SELECTS SPC1 ENTRY 1000
SPC = 1000
$ THE FOLLOWING REQUEST SELECTS RLOAD1 ENTRY 100
DLOAD = 100
.
BEGIN BULK
$ THE FOLLOWING SPC1 ENTRY IS SELECTED
$ VIA CASE CONTROL
SPC1,1000,1,70,80,90
$ THE TYPE FIELD IN THE FOLLOWING RLOAD1 ENTRY
$ SPECIFIES ENFORCED DISPLACEMENT. ACCORDINGLY,
$ EXCITEID ID 200 HERE REFERS TO SPC/SPCD DATA.
$RLOAD1,SID,EXCITEID,DELAY,DPHASE,TC,TD,TYPE
RLOAD1,100,200, , , 300, ,DISP
$ THE FOLLOWING SPCD ENTRY IS SELECTED BY THE
$ RLOAD1 DATA ABOVE
$ NOTE THAT GRID POINT 80 - COMPONENT 1 SPECIFIED
$ HERE ALSO APPEARS IN THE SELECTED SPC1 DATA.
$ NOTE ALSO THAT THE VALUE OF 1.1 SPECIFIED HERE
$ REPRESENTS DISPLACEMENT
$SPCD,SID,G,C,VALUE
SPCD,200,80,1,1.1
.
.
.

```

### SPC/SPCD Enforced Motion Input Example in Transient Response Analysis

```

.
.
.
SOL 109
CEND
TITLE = SPECIFICATION OF ENFORCED ACCELERATION
$ THE FOLLOWING REQUEST SELECTS SPC1 ENTRY 1000
SPC = 1000
$ THE FOLLOWING REQUEST SELECTS TLOAD1 ENTRY 100

```

```

DLOAD = 100
.
.
.
BEGIN BULK
$ THE FOLLOWING SPC1 ENTRY IS SELECTED
$ VIA CASE CONTROL
SPC1,1000,3,75,85
$ THE TYPE FIELD IN THE FOLLOWING TLOAD1 ENTRY
$ SPECIFIES ENFORCED ACCELERATION. ACCORDINGLY,
$ EXCITIED ID 300 HERE REFERS TO SPC/SPCD DATA.
$TLOAD1,SID,EXCITIED,DELAY,TYPE,TID
TLOAD1,100,300, ,ACCE, 500
$ THE FOLLOWING SPCD ENTRY IS SELECTED BY THE
$ TLOAD1 DATA ABOVE.
$ NOTE THAT GRID POINT 85 - COMPONENT 3 SPECIFIED
$ HERE ALSO APPEARS IN THE SELECTED SPC1 DATA.
$ NOTE ALSO THAT THE VALUE OF 2.5 SPECIFIED HERE
$ REPRESENTS ACCELERATION
SPCD,300,85,3,2.5
.
.
.

```

## 8.6 SPCF Procedure

In summary, the procedure for applying SPCF enforced motion in a dynamic analysis:

1. Use the ASSIGN file management statement to select the OUTPUT2 file that contains the displacement results from a previous frequency response analysis.
2. Use the SPCF bulk entry to specify the following:
  - Identification number of the SPCF bulk entry.
  - Identification number of a SET1 bulk entry that contains a list of structural grid points at which the loads are to be extracted and applied to for all DOFs of grids.

**Note:**

When SET1ID is blank, the software will distribute the displacement results to all structural grids contained in the OUTPUT2 file and apply enforced motions.

- Subcase number of displacement results in OUTPUT2 file.

Note:

When OUTPUT2 file only contains one subcase, you can leave the SUBC field blank.

- Logical unit number that was assigned by the ASSIGN statement to the OUTPUT2 file.
3. Use the RLOAD1 bulk entry to reference the SPCF bulk entry.

For more information, see the SPCF bulk entry and RLOAD1 bulk entry.

## 8.7 The Large Mass Method in Direct Transient and Direct Frequency Response

If a very large mass  $m_o$ , which is several orders of magnitude larger than the mass of the entire structure, is connected to a degree-of-freedom and a dynamic load  $p$  is applied to the same degree-of-freedom, then the acceleration of the degree-of-freedom, to a close approximation, is as follows:

$$\ddot{u} = \frac{1}{m_o} p$$

Equation 8-11.

In other words, the load that produces a desired acceleration  $\ddot{u}$  is approximately

$$p = m_o \ddot{u}$$

Equation 8-12.

The accuracy of this approximation increases as  $m_o$  is made larger in comparison to the mass of the structure. The only limit for the size of  $m_o$  is numeric overflow in the computer. Generally, the value of  $m_o$  should be approximately  $10^6$  times the mass of the entire structure for an enforced translational degree-of-freedom and  $10^6$  times the mass moment of inertia of the entire structure for a rotational DOF. The factor  $10^6$  is a safe limit that should produce approximately six digits of numerical accuracy.

The large mass method is implemented in direct transient and frequency response analysis by placing large masses  $m_o$  on all enforced degrees-of-freedom and supplying applied dynamic loads specified by **Equation 8-12**; that is, the function  $\ddot{u}$  is input on entries normally used for the input of loads, and the scale factor  $m_o$  can be input on DAREA or DLOAD Bulk Data entries, whichever is more convenient. CMASSi or CONMi entries should be used to input the large masses.

**8-12** is not directly helpful if enforced displacement or enforced velocity is specified rather than enforced acceleration. However, **8-12** can be made serviceable in frequency response analysis by noting that

$$\ddot{u} = i\omega\dot{u} = -\omega^2 u$$

Equation 8-13.

so that

$$p = m_o \ddot{u} = i\omega m_o \dot{u} = -\omega^2 m_o u$$

Equation 8-14.

The added factor ( $i\omega$  or  $-\omega^2$ ) can be carried by the function tabulated on the TABLEDi entry used to specify the frequency dependence of the dynamic load.

In the case of transient analysis, provision is made on the TLOAD1 and TLOAD2 entries for you to indicate whether an enforced displacement, velocity, or acceleration is supplied (TYPE = 1, 2, or 3). Simcenter Nastran then automatically differentiates a specified velocity once or a specified displacement twice to obtain an acceleration. The remaining required user actions are the same as for enforced acceleration.

In summary, the user actions for direct frequency and direct transient response are

- Remove any constraints from the enforced degrees-of-freedom.
- Apply large masses  $m_o$  with CMASSi or CONMi Bulk Data entries to the DOFs where the motion is enforced. The magnitude of  $m_o$  should be approximately  $10^6$  times the entire mass of the structure (or approximately  $10^6$  times the entire mass moment of inertia of the structure if the component of enforced motion is a rotation).
- In the case of direct frequency response, apply a dynamic load computed according to [Equation 8-14](#) to each enforced degree-of-freedom.
- In the case of direct transient response,
  - Indicate in field 5 of the TLOAD1 and TLOAD2 entries whether the enforced motion is a displacement, velocity, or acceleration.
  - Apply a dynamic load to each enforced degree-of-freedom equal to  $m_o u$ ,  $m_o \dot{u}$ , or  $m_o \ddot{u}$ , depending on whether the enforced motion is a displacement, velocity, or acceleration.

Be careful when using PARAM,WTMASS. The WTMASS parameter multiplies the large mass value, which changes the effective enforced acceleration to



$$\ddot{u} = \frac{p}{m_o \cdot WTMASS}$$

Equation 8-15.

Enforced velocity and enforced displacement are changed likewise.

You may well ask whether a stiff spring may be used instead of a large mass. In that case the applied load is

$$p = k_o u$$

Equation 8-16.

where  $k_o$  is the stiffness of the stiff spring and  $u$  is the enforced displacement. The large stiffness method certainly works, but the large mass method is preferred because it is easier to estimate a good value for the large mass than to estimate a good value for the stiff spring. In addition and more importantly, the large mass method is far superior when modal methods are used. If very stiff springs are used for modal analysis rather than very large masses, the vibration modes corresponding to the very stiff springs have very high frequencies and in all likelihood, are not included among the modes used in the response analysis. This is the main reason that large masses should be used instead of stiff springs.

The stiff spring method is advantageous in the case of enforced displacement because it avoids the roundoff error that occurs while differentiating the displacement to obtain acceleration in the large mass method. The stiff spring method also avoids the problem of rigid-body drift when applying enforced motion on statically determinate support points. (Rigid-body drift means that the displacement increases continuously with time, which is often caused by the accumulation of small numerical errors when integrating the equations of motion.)

## 8.8 The Large Mass Method in Modal Transient and Modal Frequency Response

The steps described in the previous section must also be followed when a modal method of response analysis is used. Also, if the enforced degrees of freedom are not sufficient to suppress all rigid-body motions, which may be the case for an airplane in flight, additional DOFs that describe the remaining rigid-body motions can also be entered on the SUPORT entry. Use of the SUPORT entry is discussed in **“Rigid-body Modes.”** Note that the use of the SUPORT entry is optional.

The rigid-body mode(s) can be removed from consideration either by not computing them or by using PARAM,LFREQ,r where  $r$  is a small positive number (0.001 Hz, for example). If this is done, the displacements, velocities, and accelerations obtained are relative to the overall motion of the structure and are not absolute response quantities. Stresses and element forces are the same as when the rigid-

body modes are included because the rigid-body modes do not contribute to them. Rigid-body modes can be discarded to remove rigid-body drift.

Rigid-body modes occur when the structure is unconstrained and large masses are applied at the DOFs which, if constrained, result in a statically determinate structure. Redundant constrained DOFs, which result in a statically indeterminate structure, present a different situation when the constraints are removed and large masses are applied at those redundant DOFs. In that case, very low-frequency modes occur but they are not all rigid-body modes; some are modes that represent the motion of one large mass relative to the others. These very low-frequency, nonrigid-body modes do contribute to element forces and stresses, and they must be retained in the solution. In some cases, their frequencies are not necessarily small; they may be only an order or two in magnitude less than the frequency of the first flexible mode. If `PARAM,LFREQ,r` is used to remove the true rigid-body modes, then `r` must be set below the frequency of the first relative motion mode.

Consider the two-dimensional clamped bar in [Figure 8-1](#). Assume that each end of the bar is subjected to the same enforced acceleration time history in the y-direction. One way to model the bar is to use two large masses (one at each end), which are unconstrained in the y-direction. This model provides two very low-frequency modes: one that is a rigid-body mode and one that is not. The second mode contributes to element forces and stresses, and removing its contribution leads to an error because with two such large masses, one mass can drift over time relative to the other mass. A better way to model the case of identical inputs at multiple locations is to use one large mass connected to the end points by an RBE2 element. This model provides only one rigid-body mode, which can be safely discarded if only the answers relative to the structure are desired.

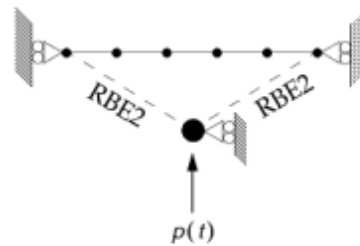
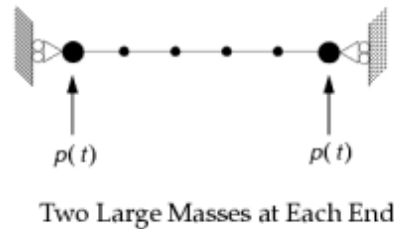
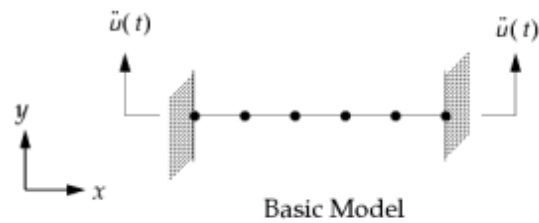


Figure 8-1. Clamped-Clamped Bar Undergoing Enforced Acceleration

## 8.9 User Interface for the Large Mass Method

There is no special user interface for the large mass method other than to specify a large mass at excitation DOFs and to specify the large force. For transient response the type of enforced motion (displacement, velocity, or acceleration) must be specified on the TLOADi entries. The remainder of the input is identical to that of frequency response ("**Frequency Response Analysis**") or transient response ("**Transient Response Analysis**") analysis.

The force applied at a point is the product of terms from the DLOAD, DAREA, and TABLEDi entries. The scaling of the large force is arbitrary; it can be on any one or more of these entries. These entries follow:

## DLOAD Format

1	2	3	4	5	6	7	8	9	10
DLOAD	SID	S	S1	L1	S2	L2	-etc.-		

## DAREA Format

DAREA	SID	P1	C1	A1	P2	C2	A2		
-------	-----	----	----	----	----	----	----	--	--

## TABLED1 Format

TABLED1	TID	XAXIS	YAXIS	EXTRAP	LOEXT	HIEXT			
	x1	y1	x2	y2	x3	y3	-etc.-	ENDT	

## TABLED2 Format

TABLED2	TID	X1	EXTRAP	LOEXT	HIEXT				
	x1	y1	x2	y2	x3	y3	-etc.-		

## TABLED3 Format

TABLED3	TID	X1	X2	EXTRAP	LOEXT	HIEXT			
	x1	y1	x2	y2	x3	y3	-etc.-		

## TABLED4 Format

TABLED4	TID	X1	X2	X3	X4				
	A0	A1	A2	A3	A4	A5	-etc.-		

The TABLED4 entry defines a power series and is convenient in frequency response for enforced constant velocity or displacement.

## Frequency Response

If a DLOAD entry is used to scale RLOAD1 input, the applied force magnitude in terms of Simcenter Nastran input is

$$P_i(f) = S \cdot \sum_i (S_i \cdot A_i [C_i(f) + i \cdot D_i(f)])$$

Equation 8-17.

where  $S$  and  $S_i$  are input on the DLOAD Bulk Data entry,  $A_i$  is input on the DAREA entry, and  $C_i(f)$  and  $D_i(f)$  are input on the TABLEDi entries. Note that the “ $i$ ” non-subscript term in the expression  $i \cdot D_i(f)$  is  $\sqrt{-1}$ .

If a DLOAD entry is used to scale RLOAD2 input, the applied force magnitude in terms of Simcenter Nastran input is

$$P_i(f) = S \cdot \sum_i (S_i \cdot A_i \cdot B_i(f))$$

Equation 8-18.

where  $S$  and  $S_i$  are input on the DLOAD Bulk Data entry,  $A_i$  is input on the DAREA entry, and  $B_i(f)$  is input on the TABLEDi entry.

Specification of the large force value depends upon whether acceleration, velocity, or displacement is enforced.

## Enforced Acceleration

Enforced acceleration is the easiest to apply since the required force is directly proportional to the desired acceleration times the large mass:

$$p(\omega) = m_o \ddot{u}(\omega)$$

Equation 8-19.

## Enforced Velocity

Enforced velocity requires a conversion factor

$$p(\omega) = i\omega m_o \dot{u} = i(2\pi f)m_o \dot{u}(\omega)$$

Equation 8-20.

For constant velocity  $\dot{u}(\omega)$ , it may be easiest to use the RLOAD1 and TABLED4 entries because the imaginary term  $i \cdot D_i(f)$  of the following equation and the frequency-dependent term  $2\pi f$  can be specified directly.

$$P_i(f) = S \cdot \sum_i (S_i \cdot A_i [C_i(f) + i \cdot D_i(f)])$$

Equation 8-21.

For more information, see [Frequency Response](#).

## Enforced Displacement

Enforced displacement also requires a conversion factor

$$p(\omega) = -\omega^2 m_o u = -(2\pi f)^2 m_o u(\omega)$$

Equation 8-22.

For constant displacement  $u(\omega)$ , it may be easiest to use the TABLED4 entry because the frequency-dependent term  $-(2\pi f)^2$  can be specified directly.

## Transient Response

For transient response, the type of enforced motion (displacement, velocity, or acceleration) is specified with the TYPE field (field 5) on the TLOAD1 and TLOAD2 Bulk Data entries.

### TLOAD1 Format

1	2	3	4	5	6	7	8	9	10
TLOAD1	SID	DAREA	DELAY	TYPE	TID				

### TLOAD2 Format

TLOAD2	SID	DAREA	DELAY	TYPE	T1	T2	F	P	
	C	B							

TYPE 0 (or blank) = applied force (default) TYPE 1 = enforced displacement TYPE 2 = enforced velocity  
TYPE 3 = enforced acceleration

Simcenter Nastran converts enforced displacements and velocities into accelerations by differentiating once for velocity and twice for displacement. Note that for enforced acceleration, you can specify either force (TYPE = 0 or blank) or acceleration (TYPE = 3); they are the same for the large mass method.

You still need to use the large mass when specifying any type of enforced motion in transient response analysis.

## 8.10 Large Mass Method Examples

The large mass examples are:

Model	Analysis Type	Enforced Motion
bd07two	Frequency Response	Constant Acceleration
bd07bar1	Transient Response	Ramp, Acceleration
bd07bar2	Transient Response	Ramp, Displacement
bd07bar3	Transient Response	Ramp, Displacement, Discard Rigid-Body Mode

These examples are described in the following sections.

### Two-DOF Model

Consider the two-DOF model first introduced in **“Real Eigenvalue Analysis”** and shown below in **Figure 8-2**. For this example, apply a constant magnitude base acceleration of  $1.0\text{m/sec}^2$  over the frequency range of 2 to 10 Hz and run modal frequency response with 5% critical damping in all modes. The acceleration input is applied to the large mass (grid point 3). The input file for this model is shown in **Figure 8-3**.

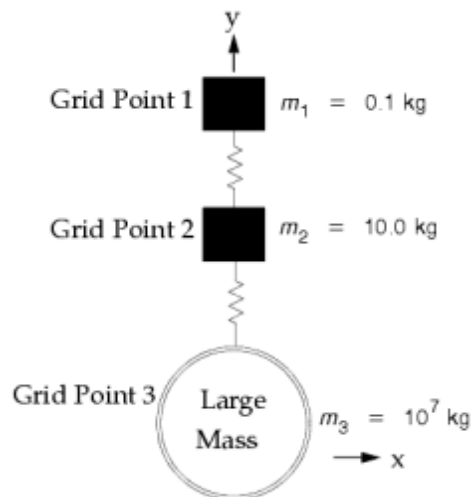


Figure 8-2. Two-DOF Model with Large Mass

```

$ FILE bd07two.dat
$
$ TWO-DOF SYSTEM
$ CHAPTER 7, ENFORCED MOTION
$
TIME 5
SOL 111

```

```

CEND
TITLE = TWO-DOF SYSTEM
SUBTITLE = MODAL FREQUENCY RESPONSE
LABEL = ENFORCED CONSTANT ACCELERATION MAGNITUDE
$
$ SPECIFY MODAL EXTRACTION
METHOD = 10
$
$ SPECIFY DYNAMIC INPUT
DLOAD = 999
FREQ = 888
SDAMPING = 777
$
$ SELECT OUTPUT
DISPLACEMENT (PHASE,PLOT) = ALL
ACCELERATION (PHASE,PLOT) = ALL
$
$ XYPLOTS
$
... X-Y plot commands ...
$
BEGIN BULK
$
$. . . . . 2 . . . . . 3 . . . . . 4 . . . . . 5 . . . . . 6 . . . . . 7 . . . . . 8 . . . . . 9 . . . . . 10 . . . . . $
$
$ LARGE MASS AT BASE GRID POINT
CONM2  999      3              1.0E7
$
$ LOAD DEFINITION (INCLUDES SCALE FACTORS FOR ENFORCED ACCELERATION)
$DLOAD  SID      S          S1      RLOAD1
DLOAD   999      1.0E7     1.0      998
$RLOAD1 SID      DAREA          TC
RLOAD1  998      997              901
$DAREA  SID      P1          C1      A1
DAREA   997      3          2        1.0
$TABLED4 TID     X1          X2      X3      X4              +TAB4
TABLED4  901     0.          1.        0.      100.             +TAB901
$+TAB4   A0      A1          A2      A3      A4          A5
+TAB901  1.0     ENDT
$
$ MODAL EXTRACTION
$EIGRL  SID      V1          V2      ND      MSGMLVL
EIGRL   10      -1.          30.      0        0
$
$ FREQUENCY RANGE 2-10 HZ
$FREQ1  SID      F1          DF      NDF
FREQ1   888     2.          0.05   160
$
$ MODAL DAMPING OF 5% CRITICAL
$TABDMP1 TID     TYPE
$+TAB1   F1      G1          F2      G2      ETC
TABDMP1  777     CRIT
+TABD7   0.      0.05       100.    0.05   ENDT              +TABD7
$
... basic model ...
$
ENDDATA

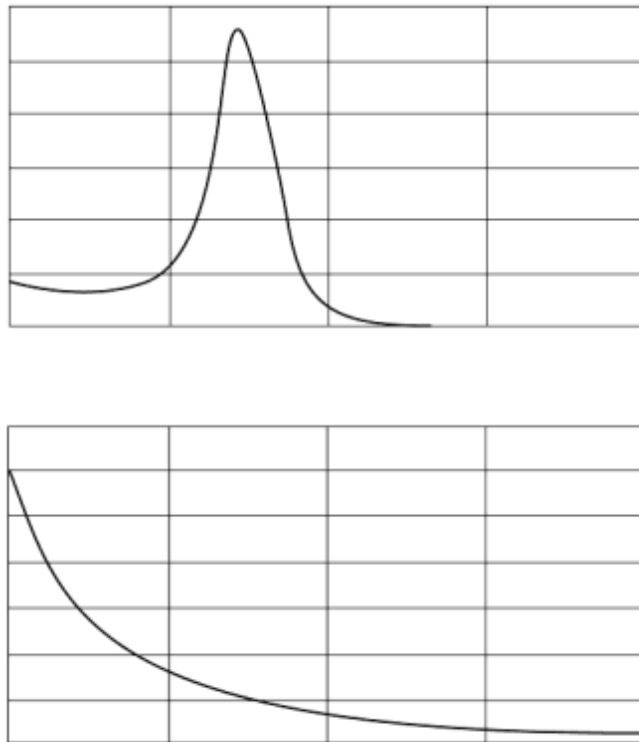
```

Figure 8-3. Input File for Enforced Constant Acceleration



The large mass value is chosen as 1.0E7 kilograms and is input via the CONM2 entry. The scale factor for the load (1.0E7) is input on the DLOAD Bulk Data entry. The factor of 1.0E7 is approximately six orders of magnitude greater than the overall structural mass (10.1 kg). The TABLED4 entry defines the constant acceleration input. (One of the other TABLEDi entries can also be used, but the TABLED4 entry is chosen to show how to use it for enforced constant velocity and displacement later in this example.)

**Figure 8-4** shows the X-Y plots resulting from the input point (grid point 3) and an output point (grid point 1). The plots show acceleration and displacement magnitudes. Note that the acceleration input is not precisely 1.0m/sec<sup>2</sup>; there is a very slight variation between 0.9999 and 1.0000 due to the large mass approximation.



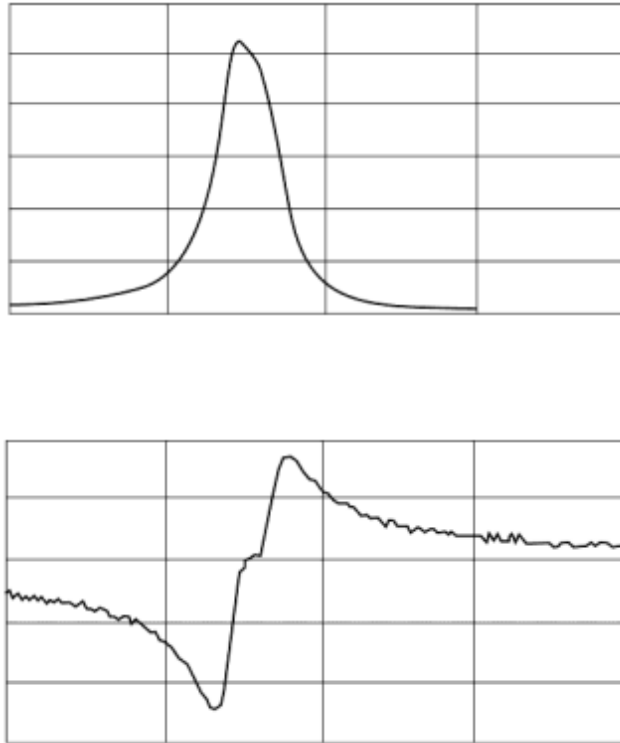


Figure 8-4. Displacements and Accelerations for the Two-DOF Model

This model was analyzed with several values of large mass. **Table 8-10** shows the results. Note that the model with the  $10^6$  mass ratio is the model discussed earlier. Peak frequency response results are compared for each model, and the natural frequencies are compared to those of the constrained model in **"Real Eigenvalue Analysis"**. The table shows that a mass ratio of  $10^6$  is a good value to use for this model.

Table 8-10. Models with Different Large Mass Ratios

Ratio of Large Mass to Structure	Natural Frequencies <sup>1</sup> (Hz)	Response Peaks (m/sec <sup>2</sup> )			
		$\ddot{x}_{1_{max}}$	$\ddot{x}_{2_{max}}$	$\ddot{x}_{3_{max}}$	$\ddot{x}_{3_{min}}$
$10^2$	$10^{-9}$	52.0552	6.5531	1.0335	0.9524
	4.8011				
	5.3025				
$10^4$	$10^{-10}$	52.2823	6.7641	1.0003	0.9995
	4.7877				
	5.2910				
$10^6$	0.0	52.2836	6.7661	1.0000	0.9999

Ratio of Large Mass to Structure	Natural Frequencies <sup>1</sup> (Hz)	Response Peaks (m/sec <sup>2</sup> )			
		$\ddot{x}_{1_{max}}$	$\ddot{x}_{2_{max}}$	$\ddot{x}_{3_{max}}$	$\ddot{x}_{3_{min}}$
	4.7876 5.2909				
10 <sup>8</sup>	0.0 4.7876 5.2909	52.2836	6.7662	1.0000	1.0000

<sup>1</sup>Resonant frequencies for the constrained model are 4.7876 and 5.2909 Hz.

This model can also be changed to apply constant velocity or constant displacement at its base. **Figure 8-5** is an abridged input file for the model, showing the Bulk Data entries required for enforced constant acceleration, enforced constant velocity, and enforced constant displacement. Note that only one of these is usually applied to any model, but all three are shown here for comparison purposes.

```

$.....2.....3.....4.....5.....6.....7.....8.....9.....10.....$
$
$ ENTRIES FOR ENFORCED MOTION
$
$ LARGE MASS AT BASE GRID POINT
CONM2 999 3 1.0E7
$
$ LOAD DEFINITION
$DLOAD SID S S1 RLOAD1
DLOAD 999 1.0E7 1.0 998
$DAREA SID P1 C1 A1
DAREA 997 3 2 1.0
$
$.....2.....3.....4.....5.....6.....7.....8.....9.....10.....$
$
$ UNIQUE ENTRIES FOR ENFORCED CONSTANT ACCELERATION MAGNITUDE
$
$RLOAD1 SID DAREA TC
RLOAD1 998 997 901
$TABLED4 TID X1 X2 X3 X4 +TAB4
TABLED4 901 0. 1. 0. 100. +TAB901
$+TAB4 A0 A1 A2 A3 A4 A5
+TAB901 1.0 ENDT
$
$.....2.....3.....4.....5.....6.....7.....8.....9.....10.....$
$
$ UNIQUE ENTRIES FOR ENFORCED CONSTANT VELOCITY MAGNITUDE
$
$RLOAD1 SID DAREA TD
RLOAD1 998 997 902
$TABLED4 TID X1 X2 X3 X4 +TAB4
TABLED4 902 0. 1. 0. 100. +TAB902
$+TAB4 A0 A1 A2 A3 A4 A5
+TAB902 0.0 6.283185 ENDT
$

```

```

$.....2.....3.....4.....5.....6.....7.....8.....9.....10.....$ $
$
$ UNIQUE ENTRIES FOR ENFORCED CONSTANT DISPLACEMENT MAGNITUDE
$
$RLOAD1 SID DAREA TC
RLOAD1 998 997 903
$TABLED4 TID X1 X2 X3 X4 +TAB4
TABLED4 903 0. 1. 0. 100. +TAB903
$+TAB4 A0 A1 A2 A3 A4 A5
+TAB903 0.0 0.0 -39.4784 ENDT

```

**Figure 8-5. Bulk Data Entries for Enforced Constant Motion**

Each input utilizes the TABLED4 entry. The TABLED4 entry uses the algorithm

$$Y = \sum_{i=0}^N A_i \left( \frac{x - X1}{X2} \right)^i$$

**Equation 8-23.**

where  $x$  is input to the table,  $Y$  is returned, and  $N$  is the degree of the power series. When  $x < X3$ ,  $X3$  is used for  $x$ ; when  $x > X4$ ,  $X4$  is used for  $x$ . This condition has the effect of placing bounds on the TABLED4 entry; note that there is no extrapolation outside of the table boundaries. There are  $N+1$  entries to this table.

Constant acceleration is the easiest to apply since the force is proportional to the mass for all frequencies. The power series for this case becomes

$$A0 + A1 \left( \frac{x - X1}{X2} \right)$$

**Equation 8-24.**

where:

A0	=	1.0
X1	=	0.0
X2	=	1.0

Therefore, these terms define a constant (1.0 in this case).

Constant velocity involves a scale factor that is directly proportional to circular frequency ( $2\pi f$ ). The power series for this case becomes

$$A0 + A1\left(\frac{x - X1}{X2}\right)$$

Equation 8-25.

where:

A0	=	0.0
A1	=	$2\pi = 6.283185$
X1	=	0.0
X2	=	1.0

Note that a phase change of 90 degrees is also required; this change is input using the TD field (field 7) of the RLOAD1 entry.

Constant displacement involves a scale factor that is proportional to the circular frequency squared  $(2\pi f)^2$  with a sign change. The power series for this case becomes

$$A0 + A1\left(\frac{x - X1}{X2}\right) + A2\left(\frac{x - X1}{X2}\right)^2$$

Equation 8-26.

where:

A0	=	0.0
A1	=	0.0
A2	=	$-(2\pi)^2 = -39.4784$
X1	=	0.0
X2	=	1.0

**Table 8-14** summarizes the coefficients for the power series.

Table 8-14. Coefficients for the Power Series

Type of Excitation	A0	A1	A2
Enforced $u$	0.0	0.0	$-(2\pi)^2$
Enforced $\dot{u}$	0.0	$2\pi$	–
Enforced $\ddot{u}$	1.0	–	–

## Cantilever Beam Model

Consider the cantilever beam first introduced in [Frequency Response Analysis](#) and shown in [Figure 8-6](#). In this case the planar model is analyzed for bending; therefore, only three DOFs per grid point are considered: T1 (x-translation), T2 (y-translation), and R3 (z-rotation). An acceleration ramp function in the y-direction is enforced at the base (grid point 1) by applying a large mass and a force. T1 and R3 are constrained at grid point 1 since the enforced motion is in only the T2-direction. Modal transient response analysis (SOL 112) is run with 5% critical damping used for all modes. Modes up to 3000 Hz are computed with the Lanczos method. [Figure 8-7](#) shows the idealized ramp function and the Simcenter Nastran implementation. The excitation is not cut off abruptly; instead it is cut off over two time steps. A time step of 0.001 second is used, and the analysis is run for 1.0 second. [Figure 8-8](#) shows the abridged input file.

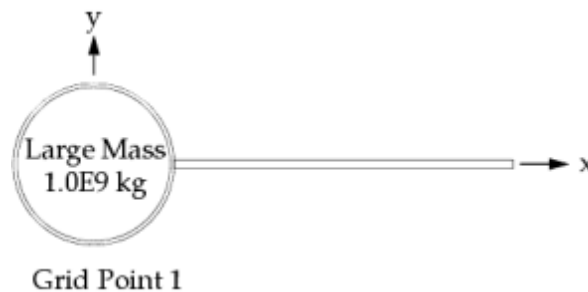
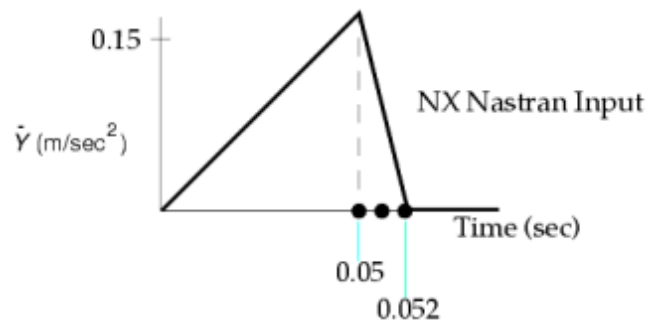
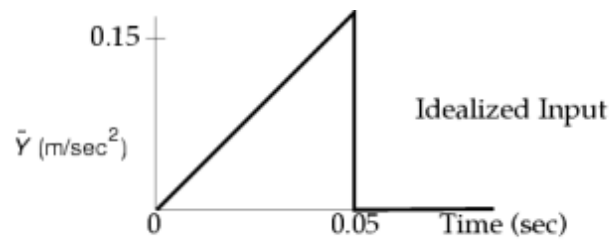


Figure 8-6. Beam Model with Large Mass



**Figure 8-7. Idealized Ramp Function Versus Simcenter Nastran Ramp Function**

```

$ FILE bd07bar1.dat
$
$ CANTILEVER BEAM MODEL
$ CHAPTER 7, ENFORCED MOTION
$
SOL 112
TIME 10
CEND
TITLE = CANTILEVER BEAM
SUBTITLE = MODAL TRANSIENT RESPONSE
LABEL = ENFORCED ACCELERATION
$
SPC = 21
DLOAD = 22
TSTEP = 27
SDAMPING = 25
$
METHOD = 10
$
$ PHYSICAL OUTPUT REQUEST
SET 11 = 1,11
DISPLACEMENT(PLOT) = 11
ACCELERATION(PLOT) = 11
$
$ XYPLOTS
$
... X-Y plot commands ...
$

```

```

BEGIN BULK
$
$. . . . . 2 . . . . . 3 . . . . . 4 . . . . . 5 . . . . . 6 . . . . . 7 . . . . . 8 . . . . . 9 . . . . . 10 . . . . . $
$
$ LARGE MASS OF 1.0E9
$CONM2 EID G CID M
CONM2 15 1 1.0E9
$
$ CONSTRAIN MASS IN 1,6 DIRECTIONS
SPC 21 1 16
$
$ DYNAMIC LOADING
$DLOAD SID S S1 L1
DLOAD 22 1.0E9 0.102 23
$TLOAD1 SID DAREA DELAY TYPE TID
TLOAD1 23 24 0 25
$DAREA SID P1 C1 A1
DAREA 24 1 2 0.15
$TABLED1 TID +TABL1
$+TABL1 X1 Y1 X2 Y2 ETC.
TABLED1 25 +TABL1
+TABL1 0.0 0.0 0.05 1.0 0.052 0.0 0.1 0.0 +TABL2
+TABL2 ENDT
$
$ CONVERT WEIGHT TO MASS: MASS = (1/G)*WEIGHT
$ G = 9.81 m/sec**2 --> WTMASS = 1/G = 0.102
PARAM WTMASS 0.102
$
$EIGRL SID V1 V2 MSGVLV
EIGRL 10 -1. 3000. 0
$
$TSTEP SID N1 DT1 NO1
TSTEP 27 1000 0.001 1
$
$ MODAL DAMPING OF 5% IN ALL MODES
$TABDMP1 TID TYPE +TABD
$+TABD F1 G1 F2 G2 ETC.
TABDMP1 25 CRIT +TABD
+TABD 0. 0.05 1000. 0.05 ENDT
$
... basic model ...
$
ENDDATA

```

**Figure 8-8. Abridged Input File for Enforced Acceleration**

A large mass of 1.0E9 kg is placed at grid point 1. This grid point is constrained in the T1- and R3- directions but is free in the T2-direction. The load is scaled to give a peak input acceleration of 0.15m/sec<sup>2</sup>. This scaling is performed by applying a scale factor of 1.0E9 in the S field (field 3) of the DLOAD entry, a scale factor of 0.102 in the S1 field (field 4) of the DLOAD entry, and a factor of 0.15 in the A1 field (field 5) of the DAREA entry. The applied load is scaled by 0.102 because the large mass is also scaled by 0.102 due to the PARAM,WTMASS entry (see [Equation 8-27](#)). The time variation is specified with the TABLED1 entry. The TLOAD1 entry specifies the type of loading (field 5) as 0 (applied force); this gives the same answers if the type is specified as 3 (enforced acceleration).

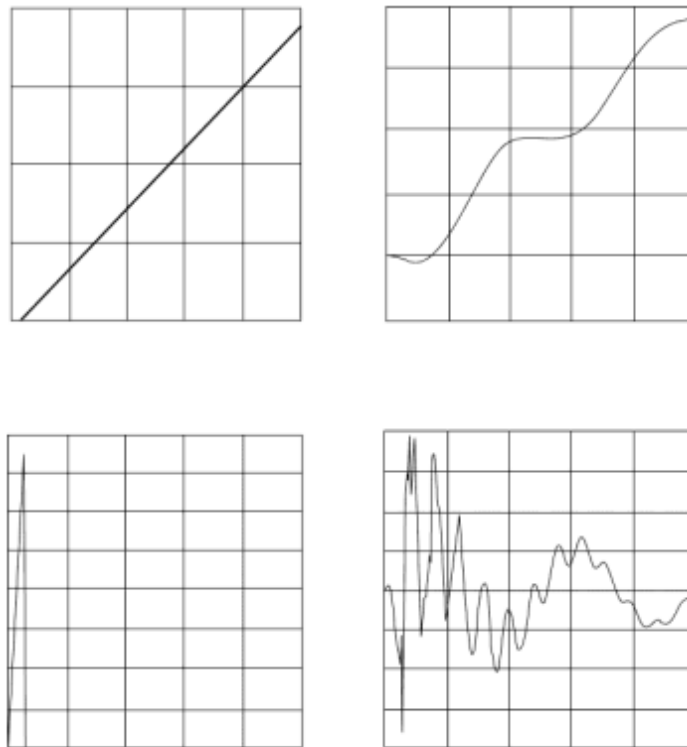


$$\ddot{u} = \frac{p}{m_o \cdot WTMASS}$$

Equation 8-27.

For more information, see [The Large Mass Method in Direct Transient and Direct Frequency Response](#).

**Figure 8-9** shows the displacement and acceleration response at grid points 1 (base) and 11 (tip). Note that at the end of the acceleration pulse the base has a constant velocity, and therefore, a linearly increasing displacement.



**Figure 8-9. Response for Enforced Acceleration**

Next, consider the same model with a 0.015 meter displacement imposed instead of an acceleration. The same ramp time history function is used (with a peak enforced displacement of 0.015 meter) so that the only change to the input file is to change the excitation type from 0 (applied force) to 1 (enforced displacement on field 5 of the TLOAD1 entry) and the amplitude in the DAREA entry from 0.15 to 0.015. **Figure 8-10** shows the idealized input displacement time history. **Figure 8-11** shows the displacement and acceleration response at grid points 1 and 11.

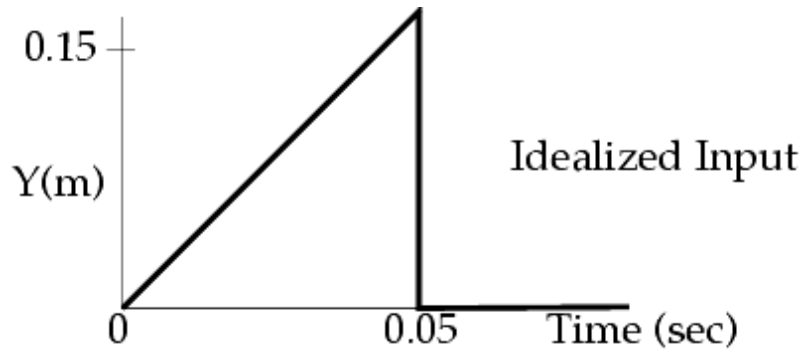


Figure 8-10. Response for Enforced Displacement (With the Rigid-Body Mode)

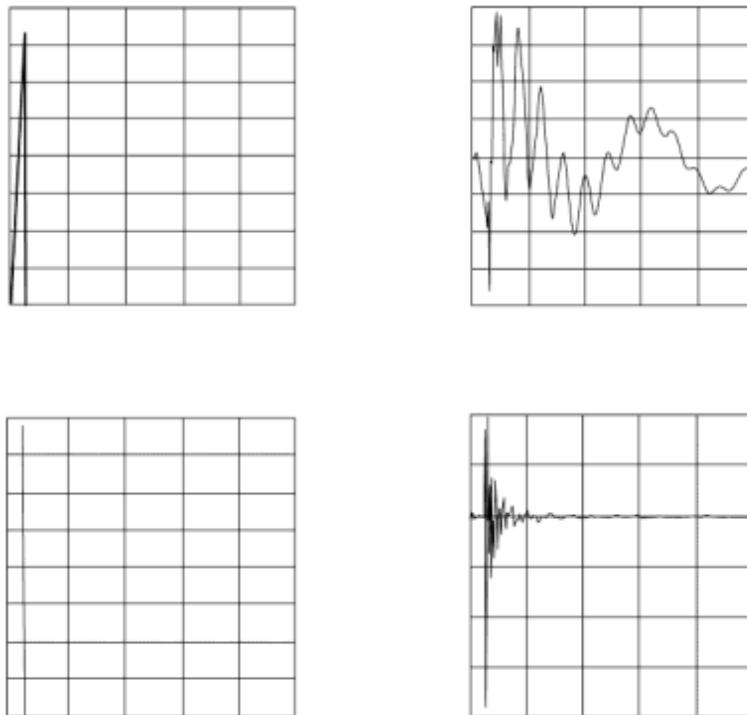
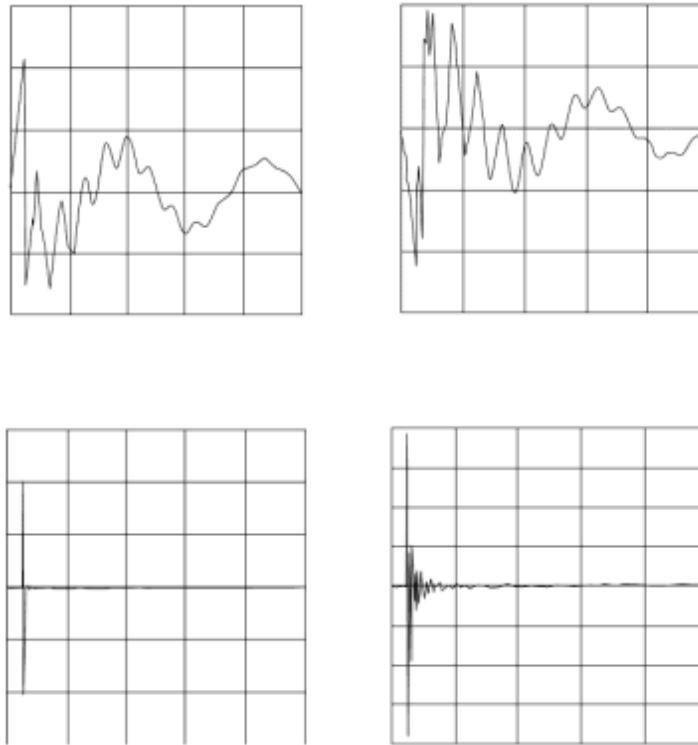


Figure 8-11. Response for Enforced Displacement (Without the Rigid-Body Mode)

Now, consider a change to the enforced displacement run. In this case, remove the rigid-body mode's contribution either by not computing the rigid-body mode (by setting V1 to a small positive value, such as 0.01 Hz) or by neglecting the rigid-body mode in the transient response (by setting PARAM,LFREQ to a small positive number, such as 0.01 Hz). **Figure 8-12** shows the resulting displacement and acceleration responses at grid points 1 and 11. Note that the responses are relative to the structure and are not absolute. The relative displacement of grid point 1 should be zero, and it is very close to zero (i.e.,  $10^{-10}$ ) as a result of the sufficiently large mass.



**Figure 8-12. Response for Enforced Displacement (Without the Rigid-Body Mode)**



# 9. Restarts in Dynamic Analysis

## 9.1 Overview of Restarts in Dynamic Analysis

A restart is a logical way of continuing from a previous run without having to start from the beginning. The savings can be in terms of both time and money. Some forms of restarts are used practically every day perhaps without you realizing it. An example of a restart can be as simple as reading a book. Normally, you probably do not finish reading a book in one continuous stretch. You may read a hundred pages today and another fifty pages tomorrow, and so on. Each time that you continue from where you left off previously is a restart. It is much more time consuming and impractical to start from page one every time that you pick up the book.

This analogy can be applied to Simcenter Nastran. In the case of a static analysis, the most expensive and time consuming part of the run is the decomposition of the stiffness matrix. This fact is especially true for large models. Now suppose after completing the original run, you want to obtain additional output (e.g., stresses, displacements, etc.) or add more load cases. You can always start from the beginning and redo the whole problem, or you can perform a restart at a fraction of the time and cost. In the case of additional output requests and additional load conditions, the decomposition of the stiffness matrix, which was performed in the previous run, is not redone if restart is used.

In dynamic analysis, the calculation of normal modes is, in general, the most expensive operation. Therefore, a common application of restart is the performance of a transient or frequency response analysis by restarting from the normal modes calculation, which was saved in the database from a previous run. This restart process avoids the recalculation of the normal modes.

The cost of restarting is measured in the disk space required to store data blocks for subsequent use. Judging whether to not save data blocks and simply rerun the analysis or to save data blocks for restarting is determined by several factors, such as the amount of available disk space and your computer's solution speed. Note that the database can be copied to tape which then provides you with more free disk space. When you are ready to perform a restart, this database can then be copied from the tape back to your disk.

## 9.2 Automatic Restarts

Simcenter Nastran's Executive System effectively uses modern database technology. Among its many features, the Executive System implements the NASTRAN Data Definition Language (NDDL). With NDDL, Simcenter Nastran is able to use automatic restart logic with the superelement solution sequences. For more information on the NDDL, refer to the *Simcenter Nastran DMAP Programmer's Guide*.

In order to maintain upward compatibility, we retained the old solution sequences and added a new series of superelement solution sequences. These solution sequences are also known as the Structured Solution Sequences (solution sequences numbers greater than 100). The automatic restart logic is available only for these solution sequences. Restart is no longer available for the rigid format solution sequences. Use these new Structured Solution Sequences rather than the old ones since all the new features are automatically implemented in these new solution sequences. Improvements made to the automatic restart logic for Version 67 and later, have made it even more efficient and robust. In this

chapter, we will address the restart logic for the Structured Solution Sequences. Note that you are not required to use superelements in order to use the superelement solution sequences. If you are familiar with the rigid format solutions, converting to the new Structured Solution Sequences only requires that you replace the solution command. For example, a normal modes run only requires replacing the SOL 3 command with the SOL 103 command in the Executive Control Section.

## 9.3 Structure of the Input File

Before presenting details on how restarts work, the following simple flow diagram of the Simcenter Nastran input file structure may be beneficial:

**Table 9-1. Structure of the Simcenter Nastran Input File**

<b>NASTRAN Statement</b>	Optional
<b>File Management Statements</b>	Optional
<b>Executive Control Statements</b>	Required Section
CEND	Required Delimiter
<b>Case Control Commands</b>	Required Section
BEGIN BULK	Required Delimiter
<b>Bulk Data Entries</b>	Required Section
ENDDATA	Required Delimiter

This order must be followed as shown for all Simcenter Nastran input files. For details regarding the statements, commands, and entries, see the *Simcenter Nastran Quick Reference Guide*.

## 9.4 User Interface

There are two types of runs—cold start and restart runs—which are described below.

### Cold Start Run

The first run, which is called the cold start run, is identical to what you usually do for submitting a Simcenter Nastran job with the exception that the database must be saved. In addition to your normal output files (e.g., F06 file), four database files are created as a result of this run, assuming that the default database initialization values are used. The convention for the filenames is machine dependent; therefore, you should refer to the machine-dependent documentation for your specific computer for the exact syntax. For a Simcenter Nastran input file named "run1.dat", the following sample submittal command can be used:

```
nastran run1
```

where “nastran” is the name of the shell script for executing Simcenter Nastran. Note that the above submittal command assumes that you have not modified the “scr=no” option as your default nastran submittal option. The “scr=no” option saves the database at the end of the run, which is required if you intend to perform restarts. The following four database files are created as a result of the above command.

```
run1.DBALL
run1.MASTER
run1.USROBJ
run1.USRSOU
```

The last two files are not needed and can be deleted if you do not want to store your own DMAP, which is usually the case.

## Restart Run

### NASTRAN Statement Section

This section is normally the same as your cold start run. The BUFFSIZE must not be changed in a restart run.

### File Management Section (FMS)

This section tells Simcenter Nastran that you are performing a restart run. The RESTART statement is required in any restart run. The general format for the RESTART statement is as follows:

```
RESTART VERSION=a,b
```

where “a” is the version from which you restart (default value for “a” is LAST) and “b” indicates whether version “a” is to be kept (KEEP) or deleted (NOKEEP) at the end of the run. The default value for “b” is NOKEEP. Due to the default values, the following two restart statements are identical:

```
RESTART VERSION=LAST,NOKEEP
RESTART
```

Whenever a restart is performed, a new version number is automatically created in the database. For each restart, the current version number is incremented by one regardless of whether the job ran successfully or not. There are two exceptions to this rule and they are discussed later on. You also need to tell Simcenter Nastran which database you want to attach to your current run. There are several ways to accomplish this; one way is to use the ASSIGN statement. For example, if you are restarting from the database created by run1.dat, the following FMS statements can be used in your current run:

```
RESTART
ASSIGN MASTER='run1.MASTER'
```

Filenames are case sensitive on some machine types. They should be entered exactly as they were created in the cold start run and enclosed with single quotes as shown above.

An alternate way is to use the DBS statement on the submittal line instead of the ASSIGN statement. Assuming the current run is called run2.dat, then the equivalent submittal statement is as follows:

```
nastran run2 dbs=run1
```

The ASSIGN statement is not needed in this case as long as the run1 database files are contained in the submittal directory. You should refer to the machine-dependent documentation for your particular computer for specific details.

## Executive Control Section

This section is the same as your normal run with the exception of perhaps the "SOL x" command. For example, if you are performing a normal modes analysis in run1.dat, then the "SOL x" command in run1.dat should reference "SOL 103". In run2.dat, if you are performing a modal transient restart from run1.dat, then the "SOL x" statement in this case should then reference "SOL 112". DIAGs can be turned on or off.

## Case Control Section

The automatic restart logic compares the modifications made to the Case Control and Bulk Data Sections in determining which operations need to be processed upon restart. Therefore, you must be very careful with the changes that you make in your restart run. Adhering to the following rules will avoid unnecessary reprocessing of previously completed operations.

- You must include all "solution-type" related Case Control commands, which are unchanged as compared to the cold start run, in your restart run. In other words, do not make unnecessary LOAD, SPC, MPC, or METHOD command changes or remove them from the Case Control Section unless these are actual changes. This process is clarified later with the example problems.
- Output requests can be modified. A typical example can be a request of the eigenvector printout which was not requested in the cold start run.

## Bulk Data Section

As mentioned in the previous section, the automatic restart logic compares the changes made in the Bulk Data Section and determines the path that it follows. A copy of the Bulk Data is stored for each version. The restart run must not contain any Bulk Data entry that was included in the previous runs and saved in the database. The Bulk Data Section in the current restart run should contain only new entries, changed entries, and/or the deletion of old entries from the database. This philosophy is slightly different than the one used in the Case Control Section.

**Table 9-2** shows an example where a normal modes run is performed in the cold start run. The eigenvalue table and the grid point weight generator output are requested in this run. For the restart run, the eigenvector output is desired; furthermore, reprinting the grid point weight generator output is



not desired. One way to accomplish this is to delete the existing "PARAM,GRDPNT,0" entry from the database.

**Table 9-2. Listing of the Cold Start and Restart Input Files**

Cold Start Run	Restart Run
\$	\$
\$ FILENAME - run1.dat	\$ FILENAME - run2.dat
\$	\$
\$ COMMAND	\$ COMMAND
\$	\$
\$ nastran run1	\$ nastran run2
\$	\$
SOL 103	RESTART
TIME 5	ASSIGN MASTER='run1.MASTER'
CEND	SOL 103
TITLE = COLDSTART RUN	TIME 5
METHOD = 10	CEND
SPC = 1	TITLE = RESTART RUN
BEGIN BULK	METHOD = 10
\$	SPC = 1
GRID,1,,0.,0.,0.	\$
GRID,2,,10.,0.,0.	\$ REQUEST FOR EIGENVECTOR
CROD,1,10,1,2	\$ PRINTOUT
PROD,10,1,1.0	\$
MAT1,1,1.+7,,.32	DISP = ALL
EIGRL,10,,,1	BEGIN BULK
CONM2,100,2,,10.	\$
PARAM,WTMASS,.00259	\$ SKIP GRID POINT WEIGHT
\$	\$ GENERATOR OUTPUT
\$ REQUEST FOR GRID POINT	\$
\$ WEIGHT GENERATOR	/,7
\$ OUTPUT	\$
\$	ENDDATA
PARAM,GRDPNT,0	
\$	
SPC1,1,123456,1	
SPC1,1,23456,2	
\$	
ENDDATA	

The format for the delete entry is "/,K1,K2" where K1 and K2 are the sorted Bulk Data sequence number of the first and last entries in the sequence to be removed, respectively. In the event that K2 = K1, the following two entries are identical:

```

/,K1,K2
/,K1

```

The values of K1 and K2 can be obtained in the F06 file of your cold start run as long as the Bulk Data is echoed with the "ECHO = SORT" Case Control command, which is the default option. From **Figure 9-1**, the sorted Bulk Data count for "PARAM,GRDPNT,0" is "7"; therefore, the delete entry in this case can be either "/,7,7" or "/,7". No additional Bulk Data entry is required for this restart run since there are no other changes involved for this model. Note that the same SPC and METHOD commands are used in both the cold start and restart runs since neither the actual boundary condition nor the desired eigenvalue calculation has changed.

CARD	S O R T E D   B U L K   D A T A   E C H O												
COUNT	. 1	.. 2	.. 3	.. 4	.. 5	.. 6	.. 7	.. 8	.. 9	..	..	..	..
10 .													
1- CONM2	100	2											
10. 2- CROD	1	10	1										
2 3- EIGRL	10												
1 4- GRID	1		0.	0.									
0. 5- GRID	2		10.	0.									
0. 6- MAT1	1												
1.+7 .32		.097											
7- PARAM	GRDPNT												
0 8- PARAM													
WTMASS .00259													
9- PROD	10	1											
1.0 10- SPC1	1	23456											
2 11- SPC1	1	123456											
1 ENDDATA													
TOTAL COUNT=		12											

**Figure 9-1. Echo of the Sorted Bulk Data Input for the Cold Start Run**

You can include as many of these delete entries as necessary. However, if the case requires many changes, it is probably more convenient to delete the entire old Bulk Data Section from the database and replace it with the fully revised Bulk Data Section, including the new and modified entries. This operation can be accomplished by including the full new Bulk Data plus the following entry in the Bulk Data Section of the restart run:

```

/1,i

```

where  $i$  is any positive integer that is greater than or equal to the number of Bulk Data entries from your cold start run.

For conventional dynamic analysis (i.e., non-superelement), restarts involving model changes (e.g., changing the thickness of a plate) are not very efficient. Therefore, the savings is probably minimal, if any. However, in the case of additional output requests or a restart from a modes run to a response run, the savings can be substantial. This type of restart is covered extensively in **“Examples”**. For superelement analysis, even restarts involving model changes can be beneficial as long as these changes are localized.

## 9.5 Determining the Version for a Restart

Not all versions in the database are restartable. For each run, a message is printed near the top of the F04 file indicating the version number of the current job. In general, if a job completes without any fatal messages, then that particular version is restartable. It is a good idea to keep a brief log of all the restartable versions since this is probably the most reliable source of information regarding whether a version is restartable. If a restart job failed, e.g., due to Bulk Data error, then this newly created version is not restartable, which is indicated by the following error message at the bottom of the F06 file.

```
*** USER WARNING MESSAGE 784 (XCLNUP) VERSION = yyy PROJECT = "zzz"
OF THIS DATA BASE IS NOT VALID FOR RESTART PURPOSES.
USER ACTION: SUBSEQUENT RESTARTS SHOULD REFERENCE VERSION = xxx
OR A PRIOR VALID VERSION
```

If the version is not restartable, you must restart from a previous valid version. The “xxx” and “yyy” above denote version numbers. The “zzz” denotes a project description provided by you. This project description is alphanumeric and can contain up to 40 characters (the default is blank). The project description is often not used and is an optional statement.

If, for some reason, the records for the old runs are no longer available, then the DBDIR FMS statement can be used to query the database contents to find out which versions are being stored in the database. The following simple setup is all that is required for this purpose.

```
ASSIGN MASTER=' ddddd.MASTER'
DBDIR VERSION=*, PROJECT=*
ENDJOB
```

where ddddd.MASTER is the name of the database being used.

The Executive Control, Case Control, or Bulk Data Section is not required in this case. Furthermore, a new version is not created in this case. Near the top of the F06 output, a PROJECT VERSION DIRECTORY TABLE is printed listing all the versions in the database. A “\*\*\*” next to a version number indicates that this particular version was deleted from the database. This deletion may be due to the NOKEEP option or the use of the DBCLEAN command when performing a restart run. A version number with a “\*\*\*” next to it is not restartable. A version number without a “\*\*\*” next to it is restartable if the run that created the version did not terminate with UWM 784. For more advanced users, this DBDIR FMS statement can also

be used to check the database directory for the existence of data blocks, e.g., UG. If the job fails very early in the run (e.g., error in the FMS section), then a new version may not be created.

It is always good practice to back up the database on a regular basis. If the system aborts the run (e.g., disk space exhausted or time quota exceeded on a system level), then there is a chance that the database is corrupted and will not allow restarts. Another good practice to ensure that only good models are retained in the database is to perform the following:

- Use `RESTART VERSION = a,KEEP` where "a" is a version number.
- If a version contains errors or is no longer of interest, then use the FMS statement `DBCLEAN` to remove obsolete or incorrect versions of the model from the database. Removing these versions allows the Executive System to reuse some of this space for new versions.

## 9.6 Examples

The examples perform a typical series of runs starting from a normal modes run and restarting into transient and frequency response analyses. **Table 9-3** summarizes this series of nine runs along with a brief description. Listings of the nine runs are also included (figures **Input File for Normal Modes Run** through **Input File to Print the Database Dictionary**).

**Table 9-3. Typical Series of Restart Runs**

Run Sequence Number	Name of Input File	Solution Sequence Number	Description of Runs	Version Created	Version Deleted
1	bd08bar1.dat	103	Perform a normal modes cold start analysis and save the database.	1	None
2	bd08bar2.dat	103	Restart from run number 1 and request eigenvector output.	2	None
3	bd08bar3.dat	103	The first two modes of the structure are very close to one of the forcing frequencies. The structure is modified in order to stay away from resonance. This run restarts from run number 2 to delete the old PBAR entry and replace it with the modified PBAR	3	2

Run Sequence Number	Name of Input File	Solution Sequence Number	Description of Runs	Version Created	Version Deleted
			entry. The modes are recalculated, and the eigenvector output is requested in this run.		
4	bd08bar4.dat	N/A	This run deletes the data blocks in the database previously occupied by Version 1. This space can then be reused by future restarts. Note that the statement DBCLEAN does not reduce the database size. It merely removes some of its contents so that this space can be reused. A new version is not created as a result of this run. This is an optional run especially if disk space is of no concern to you.	None	1
5	bd08bar5.dat	112	This is a transient restart run from the modes saved in Version 3. The applied load is a unit step function. The modes calculated in run number 3 are also saved at the end of this run. Since the calculation of the modes is the most expensive operation in a dynamic analysis, it is probably a good idea to save Version 3 once you have confidence in the results. This way you can always restart from this version. Partial output is shown at the top of the figure	4	None

Run Sequence Number	Name of Input File	Solution Sequence Number	Description of Runs	Version Created	Version Deleted
			<b>Partial Output from Transient Analysis with Unit Step Function Input.</b> A 1% critical damping value is applied to the structure.		
6	bd08bar6.dat	112	This is another transient restart run using the solution from run number 5. The purpose of this run is to request additional output. Partial output is shown at the bottom of the figure <b>Partial Output from Transient Analysis with Unit Step Function Input.</b> Note that the maximum displacement at grid point 11 is the same as in run number 5 as expected since you are asking for data recovery on the same point due to the same loading condition.	5	4
7	bd08bar7.dat	112	This is another restart run from Version 6 with a different load condition (triangular pulse). Partial output is shown in the figure <b>Partial Output from Transient Analysis with a Triangular Pulse.</b> In this case, you can just as easily restart from Version 3.	6	5
8	bd08bar8.dat	111	This is a frequency response restart run. Note that this can also be restarted from Version 3 since it	7	6

Run Sequence Number	Name of Input File	Solution Sequence Number	Description of Runs	Version Created	Version Deleted
			was saved in the database. Partial output is shown in the figure <b>Partial Output from Frequency Response Analysis</b> . A 2% critical damping value is applied to the structure.		
9	bd08bar9.dat	N/A	This is a database directory printout run. As shown in the figure <b>Partial Output from a Database Directory Run</b> , there are seven versions in the database. Run numbers 4 and 9 did not create any new versions. Only Versions 3 and 7 are restartable. This is an optional run.	None	None

## Remark

If the results for run number 1 are not going to be used for any future purposes, then you may consider making run number 3 as a cold start run instead of a restart run. Model changes do not save you much time, if any, in a non-superelement analysis. By making run 3 a cold start run, you reduce the total amount of disk space required. In this case, run number 4 is not necessary since you are starting with a new database. However, if you want to keep both physical models in the database, then run number 3 should be a restart run as shown in this example. An application of this can be a parametric study of two different configurations. This type of restart allows you to make efficient data recovery or response analysis from two different physical models. However, this type of restart is not used often in a non-superelement analysis since, in general, it is not very efficient. However, in a superelement analysis (see **"Advanced Dynamic Analysis Capabilities"**), this type of restart can still be very efficient since the changes can be localized to a small region.

```

$ FILE - bd08bar1.dat
$
$ NORMAL MODES RUN
$
ID CANT BEAM
SOL 103
TIME 10
CEND
TITLE = CANTILEVER BEAM - NORMAL MODES - COLD START RUN
SPC = 1

```

```

METHOD = 10
$
BEGIN BULK
$
CBAR 1 1 1 2 0. 1. 0.
CBAR 2 1 2 3 0. 1. 0.
CBAR 3 1 3 4 0. 1. 0.
CBAR 4 1 4 5 0. 1. 0.
CBAR 5 1 5 6 0. 1. 0.
CBAR 6 1 6 7 0. 1. 0.
CBAR 7 1 7 8 0. 1. 0.
CBAR 8 1 8 9 0. 1. 0.
CBAR 9 1 9 10 0. 1. 0.
CBAR 10 1 10 11 0. 1. 0.
EIGRL 10 -0.1 50.
GRID 1 0.0 0. 0.
GRID 2 0.3 0. 0.
GRID 3 0.6 0. 0.
GRID 4 0.9 0. 0.
GRID 5 1.2 0. 0.
GRID 6 1.5 0. 0.
GRID 7 1.8 0. 0.
GRID 8 2.1 0. 0.
GRID 9 2.4 0. 0.
GRID 10 2.7 0. 0.
GRID 11 3.0 0. 0.
MAT1 1 7.1+10 .33 2.65+4
PARAM AUTOSPC YES
PARAM WTMASS .102
PBAR 1 1 6.158-4 3.-8 3.-8 6.-8 2.414
SPC1 1 123456 1
$
ENDDATA

```

Figure 9-2. Input File for Normal Modes Run

```

$ FILE - bd08bar2.dat
$
$ NORMAL MODES RUN
$ REQUEST EIGENVECTOR PRINTOUTS FROM PREVIOUS RUN
$
RESTART VERSION=1,KEEP
ASSIGN MASTER='bd08bar1.MASTER'
$
ID CANT BEAM
SOL 103
TIME 10
CEND
TITLE = EIGENVECTORS DATA RECOVERY RESTART RUN
SPC = 1
METHOD = 10
DISP = ALL $ PRINT EIGENVECTORS
$
BEGIN BULK

```



```
$
ENDDATA
```

**Figure 9-3. Input File for Requesting Eigenvectors**

```
$ FILE - bd08bar3.dat
$
$ NORMAL MODES RUN
$ MODIFY PBAR
$
RESTART
ASSIGN MASTER='bd08bar1.MASTER'
$
ID CANT BEAM
SOL 103
TIME 10
CEND
TITLE = CANTILEVER BEAM - NORMAL MODES - RESTART RUN
SPC = 1
METHOD = 10
DISP = ALL
$
BEGIN BULK
$
$ DELETE OLD PBAR ENTRY, LINE 26 OF SORTED BULK DATA COUNT
$
/,26
$
$ ADD NEW PBAR ENTRY
$
PBAR,1,1,6.158-4,2.9-8,3.1-8,6.-8,2.414
$
ENDDATA
```

**Figure 9-4. Input File for Modifying a Bar Element**

```
$ FILE - bd08bar4.dat
$
assign master='bd08bar1.MASTER'
dbclean version=1
endjob
```

**Figure 9-5. Input File for Cleaning a Database**

```
$ FILE - bd08bar5.dat
$
$ THIS IS A TRANSIENT RESTART RUN FROM THE MODES
$ CALCULATED BY THE RUN "bd08bar3.dat"
$
RESTART VERSION=3,KEEP
```

```

ASSIGN MASTER='bd08bar1.MASTER'
ID CANT BEAM
SOL 112
TIME 10
CEND
TITLE      =  TRANSIENT RESTART - UNIT STEP FUNCTION INPUT
SUBTITLE   =  REQUEST DISPLACEMENT TIME HISTORY AT GRID POINT 11
SPC = 1
METHOD = 10
SET 1 = 11
DISP = 1
SUBCASE 1
  SDAMP    = 100
  TSTEP    = 100
  DLOAD    = 100
$
BEGIN BULK
$
$  ADDITIONAL ENTRIES FOR DYNAMIC LOADS
$  FOR UNIT STEP FUNCTION
$
$      SID      DAREA  DELAY  TYPE  TID
TLOAD1  100      101
$
DAREA,101,11,3,1.0
$
TABLED1,102,,,,,,,,+TBL1
+TBL1,0.0,0.0,.001,1.0,10.0,1.0,ENDT
$
$  TRANSIENT TIME STEPS
$
$      SID      N(1)   DT(1)  NO(1)
TSTEP    100      600    .001    5
$
$  MODAL DAMPING TABLE
$
TABDMP1,100,CRIT,,,,,,,,+TDAMP
+TDAMP,0.,.01,200.,.01,ENDT
$
ENDDATA

```

Figure 9-6. Input File for Transient Response

```

$  FILE - bd08bar6.dat
$
$  THIS IS ANOTHER TRANSIENT RESTART RUN.  THE PURPOSE
$  OF THIS RUN IS TO REQUEST ADDITIONAL OUTPUT.
$
RESTART
ASSIGN MASTER='bd08bar1.MASTER'
ID CANT BEAM
SOL 112
TIME 10
CEND
TITLE      =  T R A N S I E N T      R E S T A R T
SUBTITLE   =  U N I T      S T E P      F U N C T I O N      I N P U T

```

```

SPC = 1
METHOD = 10
SET 1 = 11
SET 2 = 10
ACCE = 2
SUBCASE 1
    SDAMP = 100
    TSTEP = 100
    DLOAD = 100
$
$
$ PLOT RESULTS
$
$...X-Y plot commands ...
$
BEGIN BULK
$
$
ENDDATA

```

**Figure 9-7. Input File for an Additional Output Request**

```

$ FILE - bd08bar7.dat
$
$ THIS IS ANOTHER TRANSIENT RESTART RUN USING
$ A DIFFERENT LOAD CONDITION. NOTE THAT SINCE
$ THERE ARE NO MODEL CHANGES, THE SAME MODES
$ WERE USED FROM THE DATABASE FOR THE RESPONSE
$ CALCULATIONS.
$
RESTART
ASSIGN MASTER='bd08bar1.MASTER'
ID CANT BEAM
SOL 112
TIME 10
CEND
$
$ NOTE THAT TITLE CHANGES HAVE NO EFFECT
$ ON SOLUTION PROCESS, THEY ONLY CHANGE THE
$ PRINTOUT TITLE
$
TITLE = T R A N S I E N T R E S T A R T
SUBTITLE = TRIANGLE PULSE - 1.0 AT T=0 AND 0.0 AFTER .2 SEC
SPC = 1
METHOD = 10
SET 1 = 11
DISP = 1
SUBCASE 1
    SDAMP = 100

```

```

TSTEP    = 100
DLOAD    = 300
$
$ PLOT RESULTS
$
$...X-Y plot commands ...
$
BEGIN BULK
$
$          SID      DAREA  DELAY   TYPE     TID
TLOAD1    300      301      .20    10.0    302
$
DAREA, 301, 11, 3, 1.0
$
TABLED1, 302, , , , , , , , +TBL3
+TBL3, 0.0, 0.0, .001, 1.0, .20, 0.0, 10.0, 0.0, +TBL4
+TBL4, ENDT
$
ENDDATA

```

**Figure 9-8. Input File for an Additional Transient Load**

```

$ FILE - bd08bar8.dat
$
$ THIS IS RESTART RUN TO PERFORM FREQUENCY RESPONSE
$
RESTART
ASSIGN MASTER='bd08bar1.MASTER'
ID CANT BEAM
SOL 111
TIME 10
CEND
TITLE = CANTILEVER BEAM - FREQUENCY RESPONSE RESTART
SPC = 1
METHOD = 10
SET 1 = 11
DISP(PHASE) = 1
SUBCASE 1
$
$ A TWO-PERCENT CRITICAL DAMPING IS
$ APPLIED TO THIS RUN AS OPPOSED TO
$ ONE-PERCENT CRITICAL DAMPING IN
$ THE TRANSIENT ANALYSIS
$
SDAMP     = 1000
DLOAD     = 1000
FREQ      = 1000
$
$ PLOT RESULTS

```

```

$
$...X-Y plot commands ...
$
BEGIN BULK
$
$   ADDITIONAL ENTRIES FOR FREQUENCY RESPONSE
$
$       SID       DAREA   M       N       TC       TD
$
RLOAD1   1000     1001             1002
$
DAREA,1001,11,3,0.1
$
TABLED1,1002
,0.,1.,200.,1.,ENDT
$
$   FORCING FREQUENCIES
$
$   RESONANT FREQUENCIES
$
FREQ,1000,2.03174,2.100632,12.59101,13.01795
FREQ,1000,34.90217,36.08563
$
$   SPREAD THROUGHOUT
$   FREQUENCY RANGE OF INTEREST
$   WITH BIAS BETWEEN
$   HALF-POWER POINTS
$
FREQ,1000,1.437,1.556,1.675,1.794,1.913
FREQ,1000,2.046,2.059,2.073,2.087
FREQ,1000,2.224,2.347,2.47,2.593,2.716
FREQ,1000,8.903,9.641,10.378,11.116,11.853
FREQ,1000,12.676,12.762,12.847,12.933
FREQ,1000,13.781,14.543,15.306,16.068,16.831
FREQ,1000,24.680,26.724,28.769,30.813,32.858
FREQ,1000,35.139,35.376,35.612,35.849
FREQ,1000,41.189,46.292,51.395,56.499,61.602
$
FREQ1,1000,0.,.5,200
$
$   DAMPING
$
TABDMP1,1000,CRIT,,,,,,,,+DAMP
+DAMP,0.,.02,200.,.02,ENDT
$
ENDDATA

```

**Figure 9-9. Input File for Frequency Response Analysis**

```

$ FILE - bd08bar9.dat
$
assign master='bd08bar1.MASTER'
dbdir
endjob
    
```

Figure 9-10. Input File to Print the Database Dictionary

```

***** PARTIAL OUTPUT FROM bd08bar5.f06 ***** POINT-ID = 11
DISPLACEMENT VECTOR
TIME TYPE T1 T2 T3
R1 R2 R3
.0
G .0 .0 .0 .0 .0 .0
5.000000E-03 G 9.024590E-22 1.614475E-21 1.923969E-05 .0
-3.001072E-05 3.518218E-20
1.000000E-02 G 3.367912E-21 8.536304E-21 7.739825E-05 .0
-1.118816E-04 1.271978E-19
.
.
2.300003E-01 G 3.941339E-19 8.273585E-18 7.832402E-03 .0
-3.655237E-03 5.515358E-18
2.350003E-01 G 3.956722E-19 8.583470E-18 7.862186E-03 .0
-3.686721E-03 5.464303E-18
2.400003E-01 G 3.957035E-19 9.029934E-18 7.880123E-03 .0
-3.709195E-03 5.756178E-18
2.450003E-01 G 3.944803E-19 9.498899E-18 7.886210E-03 .0
-3.726098E-03 6.049827E-18
2.500003E-01 G 3.927639E-19 9.878397E-18 7.884440E-03 .0
-3.754001E-03 6.040225E-18
2.550003E-01 G 3.909298E-19 1.020556E-17 7.873885E-03 .0
-3.796297E-03 5.837433E-18
2.600002E-01 G 3.885339E-19 1.063947E-17 7.844862E-03 .0
-3.831216E-03 5.871013E-18
2.650001E-01 G 3.848278E-19 1.128651E-17 7.785291E-03 .0
-3.828911E-03 6.419444E-18
2.700001E-01 G 3.796167E-19 1.207232E-17 7.690622E-03 .0
-3.778875E-03 7.287524E-18
.
.
5.949959E-01 G 1.986670E-19 -1.304284E-17 4.027668E-03 .0
-2.010369E-03 -4.506177E-18
5.999959E-01 G 2.106813E-19 -1.285776E-17 4.244778E-03 .0
-2.079488E-03 -4.219749E-18
    
```

```

-----
***** PARTIAL OUTPUT FROM bd08bar6.f06 *****
XY - O U T P U T S U M M A R Y ( R E S P O N S E )
XMIN-FRAME/ XMAX-FRAME/ YMIN-FRAME/
SUBCASE CURVE FRAME
X FOR YMAX-FRAME/ X FOR
ID TYPE NO. CURVE ID. ALL DATA ALL DATA ALL DATA
YMIN ALL DATA YMAX
1 DISP 1 11( 5) 0.000000E+00 5.999959E-01 0.000000E+00
0.000000E+00 7.886210E-03 2.450003E-01
0.000000E+00 7.886210E-03 2.450003E-01
1 ACCE 2 10( 5) 0.000000E+00 5.999959E-01 -8.511892E-01
    
```

```

2.650001E-01  9.618800E-01  5.000000E-03
                0.000000E+00  5.999959E-01  -8.511892E-01
2.650001E-01  9.618800E-01  5.000000E-03
    
```

**Figure 9-11. Partial Output from Transient Analysis with Unit Step Function Input**

```

***** PARTIAL OUTPUT FROM bd08bar7.f06 *****
POINT-ID = 11
                D I S P L A C E M E N T   V E C T O R
                T2           T3
R1      TIME      TYPE      T1
R2
R3
.0
G      .0          .0          .0          .0          .0          .0
5.000000E-03  G      8.975580E-22  1.606407E-21  1.913647E-05  .0
-2.984776E-05  3.499062E-20
1.000000E-02  G      3.317352E-21  8.430443E-21  7.628838E-05  .0
-1.102005E-04  1.252464E-19
.
1.650002E-01  G      2.077055E-19  2.232167E-18  4.099007E-03  .0
-1.833254E-03  1.845365E-18
1.700002E-01  G      2.104969E-19  2.275425E-18  4.180374E-03  .0
-1.908738E-03  1.502202E-18
1.750002E-01  G      2.131340E-19  2.401124E-18  4.255632E-03  .0
-2.000815E-03  1.338691E-18
1.800002E-01  G      2.146708E-19  2.724639E-18  4.308840E-03  .0
-2.072395E-03  1.631772E-18
1.850002E-01  G      2.144943E-19  3.204978E-18  4.328179E-03  .0
-2.097317E-03  2.240218E-18
1.900002E-01  G      2.129652E-19  3.684523E-18  4.313660E-03  .0
-2.081293E-03  2.749993E-18
1.950002E-01  G      2.109444E-19  4.053196E-18  4.272623E-03  .0
-2.048310E-03  2.916553E-18
.
5.949959E-01  G      1.315062E-19  -1.196586E-17  2.607416E-03  .0
-1.203858E-03  -4.331415E-18
5.999959E-01  G      1.410901E-19  -1.111419E-17  2.775622E-03  .0
-1.249621E-03  -3.777846E-18
X Y - O U T P U T S U M M A R Y ( R E S P O N S E )
SUBCASE CURVE FRAME XMIN-FRAME/ XMAX-FRAME/ YMIN-FRAME/
X FOR YMAX-FRAME/ X FOR
ID TYPE NO. CURVE ID. ALL DATA ALL DATA ALL DATA
YMIN ALL DATA YMAX
1 DISP 1 11( 5) 0.000000E+00 6.000000E-01 -4.039956E-03
4.199981E-01 4.328179E-03 1.850002E-01
0.000000E+00 5.999959E-01 -4.039956E-03
4.199981E-01 4.328179E-03 1.850002E-01
    
```

**Figure 9-12. Partial Output from Transient Analysis with a Triangular Pulse**

```

***** PARTIAL OUTPUT FROM bd08bar8.f06 *****
POINT-ID = 11
                C O M P L E X   D I S P L A C E M E N T   V E C T
                (MAGNITUDE/PHASE)
                T2           T3
R1      FREQUENCY  TYPE      T1
R2
R3
    
```

```

        .0          G      2.016639E-20  4.838324E-19  4.085192E-04  .0
2.024146E-04  3.208819E-19
        .0          .0          .0          .0
180.0000      .0
  5.000000E-01  G      2.136733E-20  5.467976E-19  4.323836E-04  .0
2.134119E-04  3.549737E-19
        359.4255      359.0918      359.4344      .0
179.4676      359.2066
        .
        2.073000E+00  G      4.228285E-19  1.831657E-16  8.402211E-03  .0
3.874130E-03  8.302084E-17
        303.5255      169.4034      303.5720      .0
123.7533      170.2924
  2.087000E+00  G      4.789151E-19  1.747242E-16  9.514135E-03  .0
4.382166E-03  7.877973E-17
        288.0493      145.7774      288.0963      .0
108.2796      146.6783
  2.100632E+00  G      5.003335E-19  1.561973E-16  9.936970E-03  .0
4.572148E-03  7.006467E-17
        270.0177      122.0074      270.0652      .0
90.2504      122.9200
  2.224000E+00  G      1.560747E-19  1.943312E-17  3.091950E-03  .0
1.408677E-03  8.308275E-18
        199.3224      32.8670      199.3742      .0
19.5776      33.8900
  2.347000E+00  G      7.916779E-20  5.971262E-18  1.564160E-03  .0
7.050781E-04  2.427647E-18
        190.2228      19.2229      190.2792      .0
10.5024      20.3631
        .
        9.950000E+01  G      .0          .0          5.272466E-07  .0
8.710860E-07  1.024487E-21
        .0          .0
180.4391      .0          .6610      180.7005
  1.000000E+02  G      .0          .0          5.216914E-07  .0
8.615909E-07  1.013485E-21
        .0          .0
180.4361      .0          .6566      180.6966
        .
        X Y - O U T P U T S U M M A R Y ( R E S P O N S E )
SUBCASE CURVE FRAME XMIN-FRAME/ XMAX-FRAME/ YMIN-FRAME/
X FOR YMAX-FRAME/ X FOR
ID TYPE NO. CURVE ID. ALL DATA ALL DATA ALL DATA
YMIN ALL DATA YMAX
  1 DISP 1 11( 5,--) 1.000000E-01 1.000000E+02 5.216914E-07
1.000000E+02 9.936970E-03 2.100632E+00
0.000000E+00 1.000000E+02 5.216914E-07
1.000000E+02 9.936970E-03 2.100632E+00
  1 DISP 1 11(--, 11) 1.000000E-01 1.000000E+02 1.804361E+02
1.000000E+02 3.594344E+02 5.000000E-01
0.000000E+00 1.000000E+02 0.000000E+00
0.000000E+00 3.594344E+02 5.000000E-01
    
```

Figure 9-13. Partial Output from Frequency Response Analysis



```

***** PARTIAL OUTPUT FROM bd08bar9.f06 *****
P R O J E C T   V E R S I O N   D I R E C T O R Y   P R I N T
  PROJECT_ID                ASSIGNED INT. VALUE   VERSION_ID
  CREATION TIME              -----
  -----
    " B L A N K "                1             **           1
9/ 2/93   13:22.20
                                     **           2
9/ 2/93   13:23. 0
                                     **           3
9/ 2/93   13:23.25
                                     **           4
9/ 2/93   13:23.59
                                     **           5
9/ 2/93   13:24.28
                                     **           6
9/ 2/93   13:24.57
                                     **           7
9/ 2/93   13:25.26

```

**Figure 9-14. Partial Output from a Database Directory Run**



# 10. Plotted Output

## 10.1 Overview of Plotted Output

Plotted output is important in verifying your model and understanding its results. Plots show information in a format that is much easier to interpret than printed output. Plots are especially important for dynamic analysis because the analysis can produce voluminous output. For example, consider a transient response analysis for which there are 1000 output time steps, 100 grid points of interest, and 10 elements of interest. Printed output is too large to interpret efficiently and effectively, and it does not easily show the time variation. Plotted output overcomes these problems.

There are two kinds of plotted output: structure plots and X-Y plots. Structure plots can depict the entire structure or a portion of it. Structure plots are useful for verifying proper geometry and connectivity. They also can be used to show the deformed shape or stress contours at a specified time or frequency. X-Y plots, on the other hand, show how a single-response quantity, such as a grid point displacement or element stress, varies across a portion or all of the time or frequency range.

There are numerous commercial and in-house plotting programs that interface to Simcenter Nastran for structure and/or X-Y plotting.

The commercial programs are similar because they operate interactively. The Simcenter Nastran plot capabilities, on the other hand, are performed as a batch operation, which means that you predefine your plots when you make your Simcenter Nastran run.

This chapter briefly describes the kinds of plots available in Simcenter Nastran. Detailed information is located in the *Simcenter Nastran Reference Manual*.

## 10.2 Structure Plotting

Structure plotting is performed to verify the model's geometry and element connectivity prior to performing a dynamic analysis.

After (or during) the analysis, structure plotting is performed to view deformed shapes and contours. For dynamic response, deformed shape and contour plots can be made for normal modes analysis (for which there is a plot, or set of plots, per mode), frequency response analysis (for which there is a plot, or set of plots, per output frequency), and transient response analysis (for which there is a plot, or set of plots, per output time).

Structure plot commands are described in the *Simcenter Nastran User's Guide*. In the Simcenter Nastran input file, structure plotting commands are listed in the OUTPUT(PLOT) section, which immediately precedes the Bulk Data Section. The structure plotting commands define the set of elements to be plotted (SET), the viewing axes (AXES), the viewing angles (VIEW) as well as the plot type and parameters (PLOT). Optionally, the scale of the plotted deformation (MAXIMUM DEFORMATION) can be specified; if not specified, the plotted deformation is scaled such that the maximum deformation is 5% of the maximum dimension of the structure.

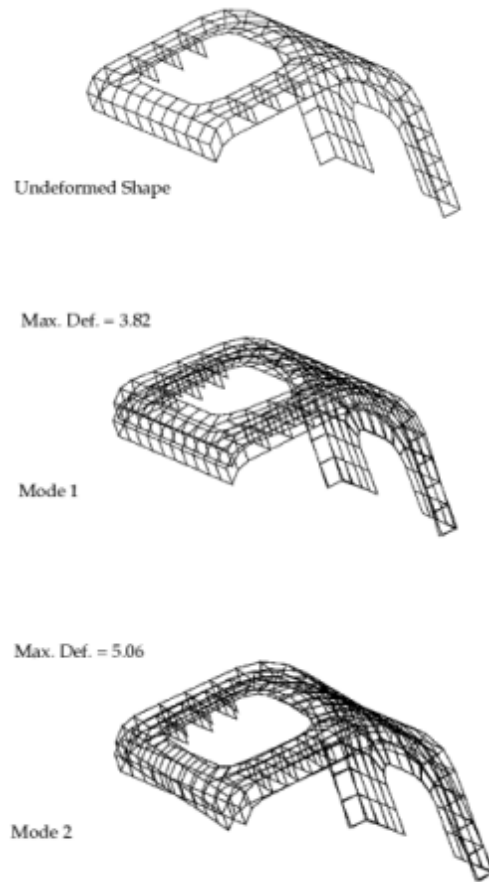
**Figure 10-1** shows the structure plotting commands applied to a normal modes analysis of the bracket model. **Figure 10-2** shows the resulting structure plots. The first plot shows the undeformed shape, and the next two plots show the undeformed shape overlaid on the deformed shapes for modes 1 and 2. The default is chosen such that the maximum plotted deformation is 5% of the maximum dimension of the bracket; the actual maximum deformation is printed at the top of the plot. The plots shown in this chapter were converted to PostScript format for printing on a PostScript-compatible printer.

```

$ PLOT COMMANDS FOR BRACKET NORMAL MODES ANALYSIS
... Executive, Case Control ...
$
$ OUTPUT REQUESTS
DISPLACEMENT (PLOT) = ALL
$
$ STRUCTURE PLOTS
OUTPUT (PLOT)
CSCALE = 3.0
SET 333 = ALL
AXES MX,MY,Z
VIEW 20.,20.,20.
FIND SCALE, ORIGIN 5, SET 333
$ PLOT UNDEFORMED SHAPE
PLOT SET 333, ORIGIN 5
$ PLOT DEFORMED, UNDEFORMED SHAPES
PLOT MODAL DEFORMATION 0,1
PLOT MODAL DEFORMATION 0,2
$
BEGIN BULK
$
... Bulk Data ...
$
ENDDATA

```

**Figure 10-1. Normal Modes Structure Plot Commands for the Bracket Model**



**Figure 10-2. Normal Modes Structure Plots for the Bracket Model**

**Figure 10-3** shows the structure plotting commands applied to a modal frequency response analysis of the cantilever beam model. The displacements are computed in magnitude/phase form. PARAM,DDRMM,-1 and PARAM,CURVPLOT,1 are required in the Bulk Data to create structure plots at specified frequencies. (Note that PARAM,DDRMM,-1 generally increases the amount of computer time and is not recommended unless otherwise required.) **Figure 10-4** shows the resulting plots. The first plot shows the undeformed shape, and the next plot shows the magnitude of response at 2.05 Hz, which is overlaid on the undeformed shape. The default is chosen such that the maximum plotted deformation is 5% of the length of the bar; the actual maximum deformation is printed at the top of the plot.

```

$ PLOT COMMANDS FOR BAR MODAL FREQUENCY RESPONSE
$ --- MAGNITUDE, PHASE ---
... Executive, Case Control ...
$
$ OUTPUT REQUESTS
$ MAGNITUDE, PHASE
DISPLACEMENT (PHASE, PLOT) = ALL
$
$ STRUCTURE PLOTS
OUTPUT (PLOT)

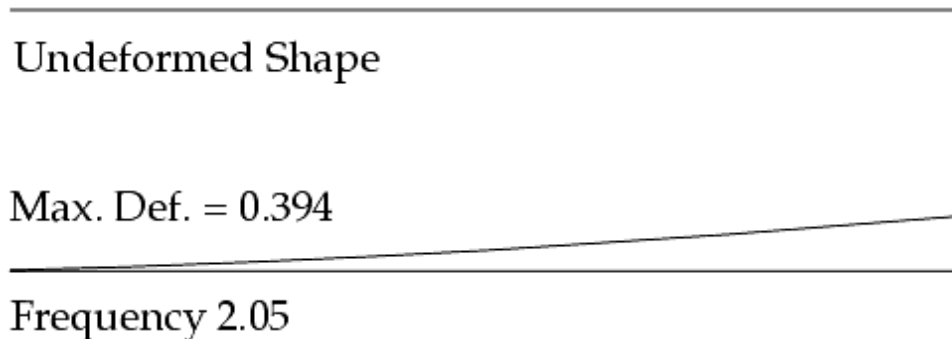
```

```

$ DEFINE ELEMENTS IN PLOT SET
SET 333 = ALL
$ PLOT AXES: R=-Z, S=X, T=Y
AXES MZ,X,Y
VIEW 0.,0.,0.
FIND SCALE, ORIGIN 5, SET 333
$ PLOT UNDEFORMED SHAPE
PLOT SET 333, ORIGIN 5
$ PLOT DEFORMED SHAPE AT 2.05 HZ
PLOT FREQUENCY DEFORMATION 0 RANGE 2.05,2.051 MAGNITUDE SET 333
$
BEGIN BULK
$
$ REQUIRED FOR "FREQUENCY FROZEN" STRUCTURE PLOTS
PARAM, DDRMM, -1
PARAM, CURVPLOT, 1
$
... rest of Bulk Data ...
$
ENDDATA

```

**Figure 10-3. Frequency Response Structure Plot Commands for the Bar Model – Magnitude/Phase**



**Figure 10-4. Frequency Response Structure Plots for the Bar Model – Magnitude/Phase**

The same plots are regenerated except that now the displacements are in real/imaginary format (the default). **Figure 10-5** shows the plot commands. Note that the imaginary component is selected by PHASE LAG -90. **Figure 10-6** shows the resulting plots. The default is chosen such that the maximum plotted deformation is 5% of the length of the bar; the actual maximum deformation is printed at the top of the plot.

```

$ PLOT COMMANDS FOR BAR MODAL FREQUENCY RESPONSE
$ --- REAL, IMAGINARY ---
... Executive, Case Control ...
$
$ OUTPUT REQUESTS
$ REAL, IMAGINARY
DISPLACEMENT (PLOT) = ALL

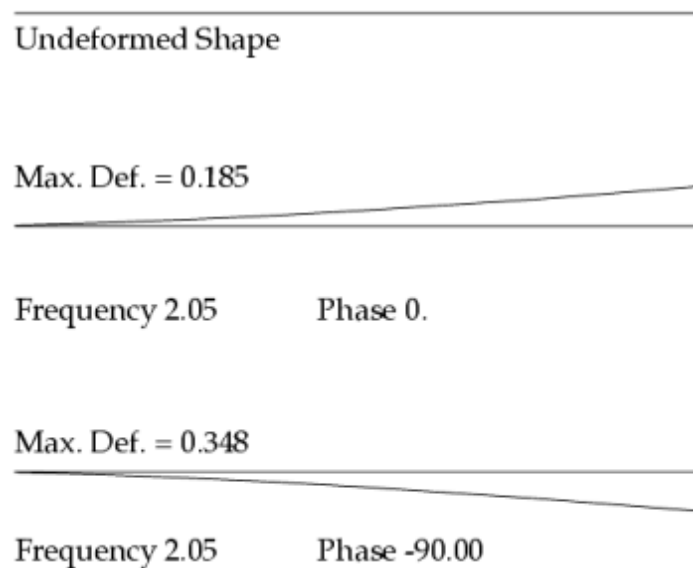
```

```

$
$ STRUCTURE PLOTS
OUTPUT(PLOT)
$ DEFINE ELEMENTS IN PLOT SET
CSCALE = 1.8
SET 333 = ALL
$ PLOT AXES: R=-Z, S=X, T=Y
AXES MZ,X,Y
VIEW 0.,0.,0.
FIND SCALE, ORIGIN 5, SET 333
$ PLOT UNDEFORMED SHAPE
PLOT SET 333, ORIGIN 5
$ PLOT DEFORMED SHAPE--REAL
PLOT FREQ DEFORM 0 RANGE 2.05,2.051 SET 333
$ PLOT DEFORMED SHAPE--IMAGINARY
PLOT FREQ DEFORM 0 RANGE 2.05,2.051 PHASE LAG -90. SET 333
$
BEGIN BULK
$
$ REQUIRED FOR "FREQUENCY FROZEN" STRUCTURE PLOTS
PARAM,DDRMM,-1
PARAM,CURVPLOT,1
$
... rest of Bulk Data ...
$
ENDDATA

```

**Figure 10-5. Frequency Response Structure Plot Commands for the Bar Model - Real/Imaginary**



**Figure 10-6. Frequency Response Structure Plots for the Bar Model – Real/Imaginary**

**Figure 10-7** shows the structure plotting commands applied to a modal transient response analysis of the cantilever beam model. `PARAM,DDRMM,-1` is required in the Bulk Data in order to create structure plots at various times and/or frequencies. (Note that `PARAM,DDRMM,-1` generally increases the amount of computer time and is not recommended unless otherwise required.) **Figure 10-8** shows the resulting plots. The plots show the displacements at the following times: 0.25, 0.50, and 0.75 seconds, which are overlaid on the undeformed shape. The default is chosen such that the maximum plotted deformation is 5% of the length of the bar; the actual maximum deformation is printed at the top of the plot. For this case it may be better to specify a maximum deformation so that the plots will show relative amplitudes.

```

$ PLOT COMMANDS FOR BAR MODAL TRANSIENT RESPONSE
... Executive, Case Control ...
$
$ OUTPUT REQUESTS
DISPLACEMENT(PLOT) = ALL
$
$ STRUCTURE PLOTS
OUTPUT(PLOT)
$ DEFINE ELEMENTS IN PLOT SET
SET 333 = ALL
$ PLOT AXES: R=-Z, S=X, T=Y
AXES MZ,X,Y
VIEW O.,O.,O.
FIND SCALE, ORIGIN 5, SET 333
$ PLOT DEFORMED SHAPE AT TIMES 0.25,0.5,0.75
PLOT TRANSIENT DEFORMATION 0 RANGE 0.25,0.251 SET 333
PLOT TRANSIENT DEFORMATION 0 RANGE 0.50,0.501 SET 333
PLOT TRANSIENT DEFORMATION 0 RANGE 0.75,0.751 SET 333
$
BEGIN BULK
$
$ REQUIRED FOR "TIME FROZEN" STRUCTURE PLOTS
PARAM,DDRMM,-1
$
... rest of Bulk Data ...
$
ENDDATA

```

**Figure 10-7. Transient Response Structure Plot Commands for the Bar Model**



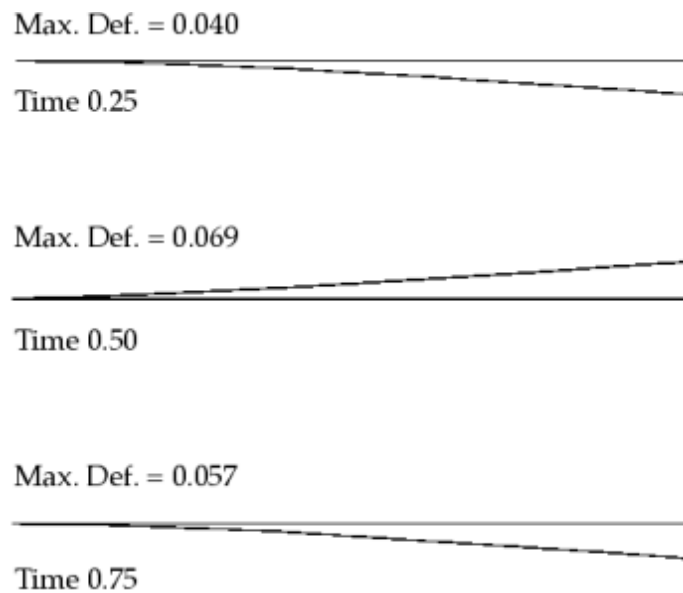


Figure 10-8. Transient Response Structure Plots for the Bar Model

Although structure plotting is best performed in an interactive environment outside of Simcenter Nastran, the batch structure plotting capability in Simcenter Nastran is nevertheless a useful tool for model verification and results processing. The batch plotting capability can save time and effort when many plots are required for a model that is run repeatedly.

### 10.3 X-Y Plotting

X-Y plots are used to display frequency and transient response results where the x-axis is frequency or time and the y-axis is any output quantity. Unlike structure plotting, which is often performed in an interactive environment, X-Y plotting is ideal for the Simcenter Nastran batch environment due to the large volume of data.

X-Y plot commands are contained in the OUTPUT(XYPLOT) section that immediately precedes the Bulk Data Section. You define the titles (XTITLE and YTITLE) and plots (XYPLOT). You can specify the plots to be generated in log format (XLOG and YLOG), and you can specify different line styles. You can also specify that the plots are to be made in pairs (with a top and a bottom plot), which is particularly useful for frequency response results when you want to display magnitude/phase or real/imaginary pairs. Details about X-Y plotting and its commands are located in the *Simcenter Nastran User's Guide*.

Element force and stress component numbers are also described in the manual. The X-Y plot commands use numbers to identify a single component of grid point and element data. Note that frequency response requires the use of complex force and stress components.

Once a good set of X-Y plot commands is established, it is wise to use this set repeatedly. The examples that follow provide a good starting point.

**Figure 10-9** shows X-Y plot commands for a modal frequency response analysis of the cantilever beam model. Plots are made in pairs in magnitude/phase format. The “t” as the second letter in XTGRID,

YTGRID, YTLOG, and YTTITLE corresponds to the top plot of each pair; the letter “b” corresponds to the bottom plot (XBGRID, YBGRID, etc.) Plots are made for the applied loads (OLOAD) at grid points 6 and 11, displacements (DISP) of grid points 6 and 11, bending moment (ELFORCE) at end A in plane 1 for element 6, and modal displacements (SDISP) for modes 1 and 2. For CBAR elements, force component 2 is the bending moment component at end A in plane 1 for real or magnitude output, and force component 10 is the similar component for imaginary or phase output. **Figure 10-10** shows the resulting plots. The plots in this chapter were converted to PostScript format for printing on a PostScript-compatible printer.

The X-Y plotter makes reasonable choices of upper and lower bounds for the axes for both the x- and y-axes on most plots. If it does not, the bounds can be fixed with the XMIN, XMAX, YMIN, YMAX, and their variations for half-frame curves. Instances where setting bounds explicitly results in better plots include the following situations:

- If you expect a variable to be constant or vary only slightly but want to plot it to confirm that it indeed does not vary, set the YMIN and YMAX to include the expected value, but separate them by at least 10 percent of their average value.  
For this case the automatic bound selection chooses bounds very close to each other to make the data fill up the plot. This selection causes the bounds to be nearly equal and magnifies the scale of the plot orders of magnitude larger than other plots made of varying functions. The response appears to be erratic when, in fact, it is smooth within engineering criteria. The extreme cases occur when the function varies only in its last digit. Then the function appears to be oscillating between the upper and lower limits or it can even cause a fatal error due to numerical overflow when it attempts to divide numbers by the difference YMAX-YMIN.
- The automatic bound selector tends to round up the bounds to integer multiples of 100. When plotting phase angles, bounds that cause grid lines at 90 degrees are more readable. For example, for a plot that traverses the range of 0 to 360 degrees, the usual selected bounds are 0 to 400 degrees. If you prefer to have grid lines drawn at integer multiples of 90 degrees, set YMIN to 0.0 and YMAX at 360.0 degrees.
- When plotting log plots, any bounds you input may be rounded up or down to a value that the plotter considers more reasonable. In general, changing bounds on log plots sometimes requires experimentation before a reasonable set can be found.
- Producing good quality plots is an interactive process whether the plot is produced by an interactive or batch plotter. The restart feature discussed in **“Restarts In Dynamic Analysis”** can reduce the computer costs for this iteration since restarts performed to change only plot requests are made efficiently.

```
$ X-Y PLOT COMMANDS FOR BAR MODAL FREQUENCY RESPONSE
$ --- MAGNITUDE, PHASE ---
...Executive, Case Control ...
$
$ APPLIED LOAD OUTPUT
SET 15 = 6,11
OLOAD(PHASE,PLOT) = 15
$
```

```

$ PHYSICAL OUTPUT REQUEST
SET 11 = 6,11
DISPLACEMENT(PHASE,PLOT) = 11
$
$ MODAL SOLUTION SET OUTPUT
SET 12 = 1,2
SDISP(PHASE,PLOT) = 12
$
$ ELEMENT FORCE OUTPUT
SET 13 = 6
ELFORCE(PHASE,PLOT) = 13
$
OUTPUT(XYPLOT)
XTGRID = YES
YTGRID = YES
XBGRID = YES
YBGRID = YES
$
$ PLOT RESULTS
XTITLE = FREQUENCY
$
YTLOG = YES
YTTITLE = DISPL. MAG. 6
YBTITLE = DISPL. PHASE 6
XYPLOT DISP /6(T2RM,T2IP)
YTTITLE = DISPL. MAG. 11
YBTITLE = DISPL. PHASE 11
XYPLOT DISP /11(T2RM,T2IP)
$
YTTITLE = SDISP. MAG. MODE 1
YBTITLE = SDISP. PHASE MODE 1
$ XYPLOT SDISP /mode(T1)
XYPLOT SDISP /1(T1RM,T1IP)
YTTITLE = SDISP. MAG. MODE 2
YBTITLE = SDISP. PHASE MODE 2
XYPLOT SDISP /2(T1RM,T1IP)
$
YTTITLE = BEND. MOMENT A1 EL. 6 MAG.
YBTITLE = BEND. MOMENT A1 EL. 6 PHASE
XYPLOT ELFORCE /6(2,10)
$
YTLOG = NO
YBMAX = 90.0
YBMIN = 0.0
CURVELINESYMBOL = -2
YTMAX = 4.0
YTMIN = 0.0
YTTITLE = LOAD MAG. 6
YBTITLE = LOAD PHASE 6

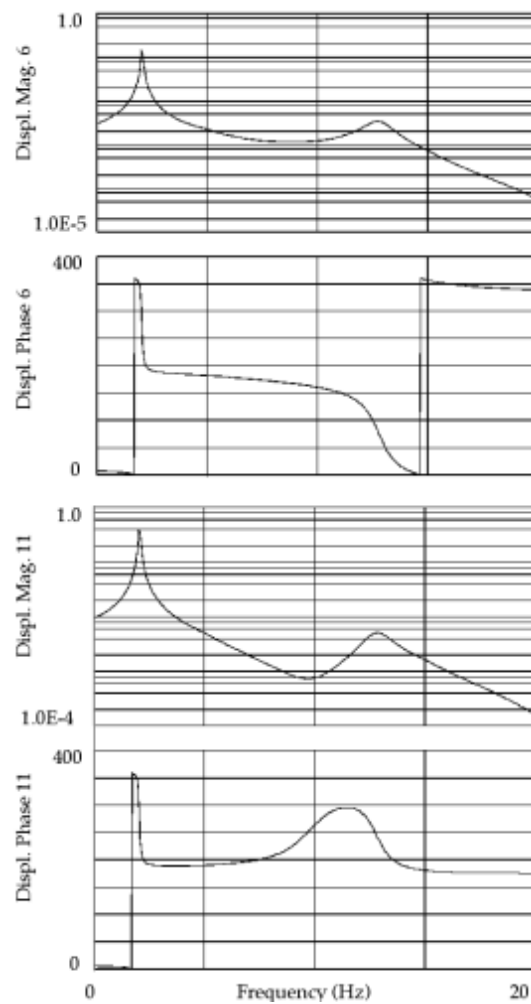
```

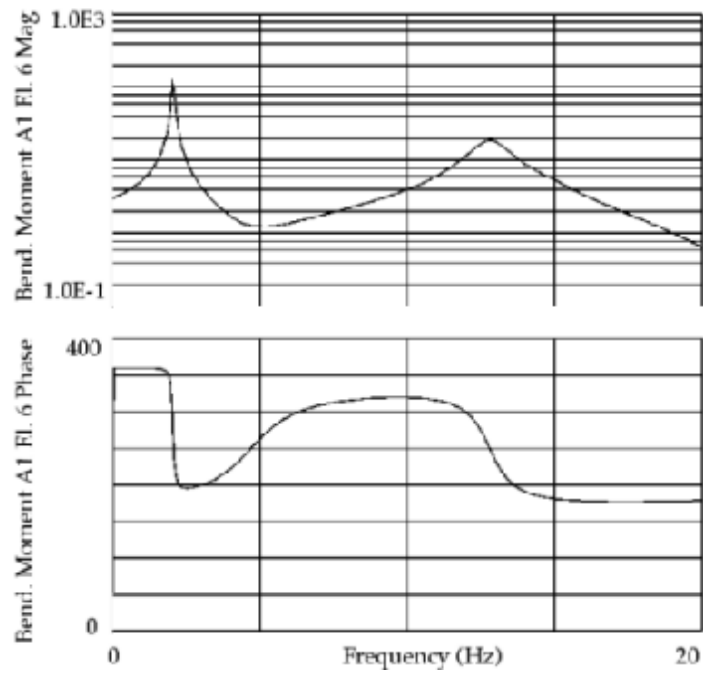
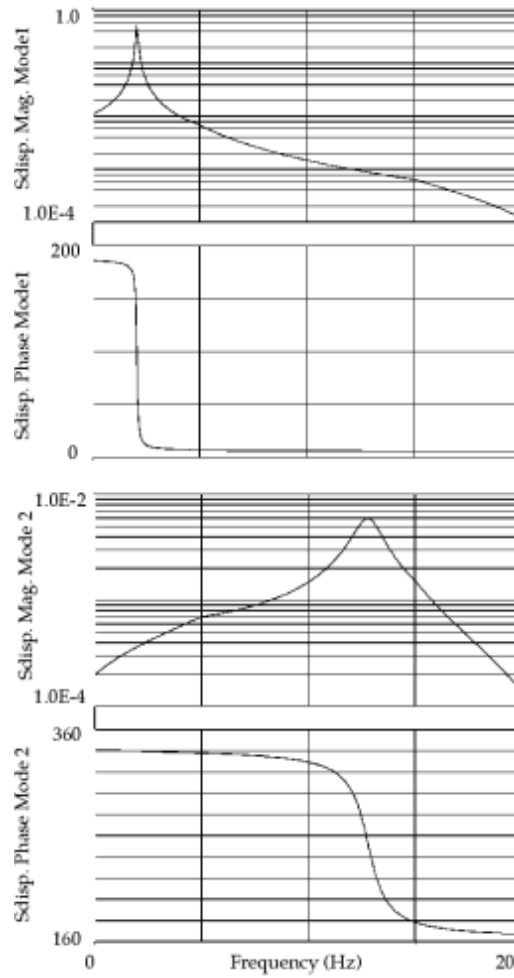
```

XYPLOT OLOAD /6 (T2RM, T2IP)
YTMAX = 8.0
YTMIN = 0.0
YTTITLE = LOAD MAG. 11
YBTITLE = LOAD PHASE 11
XYPLOT OLOAD /11 (T2RM, T2IP)
$
BEGIN BULK
$
... Bulk Data ...
$
ENDDATA

```

Figure 10-9. X-Y Plot Commands for the Bar Frequency Response Analysis





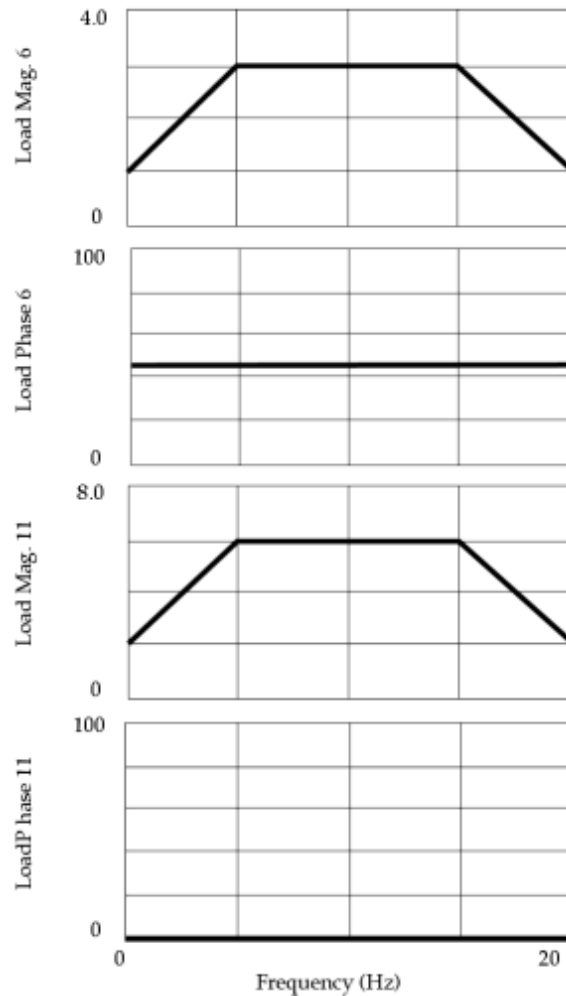


Figure 10-10. X-Y Plots for the Bar Frequency Response Analysis

Figure 10-11 shows X-Y plot commands for a modal transient response analysis of the cantilever beam model. Plots are made for the applied loads (OLOAD) at grid points 6 and 11, displacements (DISP) of grid points 6 and 11, accelerations (ACCE) for grid points 6 and 11, bending moment (ELFORCE) at end A in plane 1 for element 6, and modal displacements (SDISP) for modes 1 and 2. Figures 10-12 through 10-16 show the resulting plots.

```

$ X-Y PLOT COMMANDS FOR BAR MODAL FREQUENCY RESPONSE
$ --- MAGNITUDE, PHASE ---
...Executive, Case Control ...
$
$ APPLIED LOAD OUTPUT
SET 15 = 6,11
OLOAD(PHASE,PLOT) = 15
$
$ PHYSICAL OUTPUT REQUEST
SET 11 = 6,11
DISPLACEMENT(PHASE,PLOT) = 11
$
$ MODAL SOLUTION SET OUTPUT
SET 12 = 1,2

```

```

SDISP(PHASE,PLOT) = 12
$
$ ELEMENT FORCE OUTPUT
SET 13 = 6
ELFORCE(PHASE,PLOT) = 13
$
OUTPUT(XYPLOT)
XTGRID = YES
YTGRID = YES
XBGRID = YES
YBGRID = YES
$
$ PLOT RESULTS
XTITLE = FREQUENCY
$
YTLOG = YES
YTTITLE = DISPL. MAG. 6
YBTITLE = DISPL. PHASE 6
XYPLOT DISP /6(T2RM,T2IP)
YTTITLE = DISPL. MAG. 11
YBTITLE = DISPL. PHASE 11
XYPLOT DISP /11(T2RM,T2IP)
$
YTTITLE = SDISP. MAG. MODE 1
YBTITLE = SDISP. PHASE MODE 1
$ XYPLOT SDISP /mode(T1)
XYPLOT SDISP /1(T1RM,T1IP)
YTTITLE = SDISP. MAG. MODE 2
YBTITLE = SDISP. PHASE MODE 2
XYPLOT SDISP /2(T1RM,T1IP)
$
YTTITLE = BEND. MOMENT A1 EL. 6 MAG.
YBTITLE = BEND. MOMENT A1 EL. 6 PHASE
XYPLOT ELFORCE /6(2,10)
$
YTLOG = NO
YBMAX = 90.0
YBMIN = 0.0
CURVELINESYMBOL = -2
YTMAX = 4.0
YTMIN = 0.0
YTTITLE = LOAD MAG. 6
YBTITLE = LOAD PHASE 6
XYPLOT OLOAD /6(T2RM,T2IP)
YTMAX = 8.0
YTMIN = 0.0
YTTITLE = LOAD MAG. 11
YBTITLE = LOAD PHASE 11
XYPLOT OLOAD /11(T2RM,T2IP)
$
BEGIN BULK
$
... Bulk Data ...
$
ENDDATA

```

**Figure 10-11. X-Y Plot Commands for the Bar Transient Response Analysis**

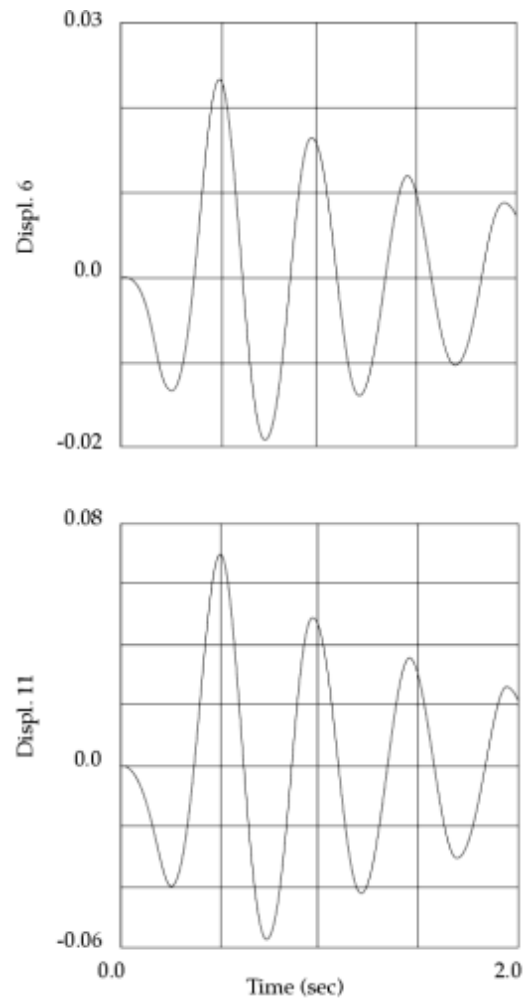


Figure 10-12. X-Y Plots for the Bar Transient Response Analysis



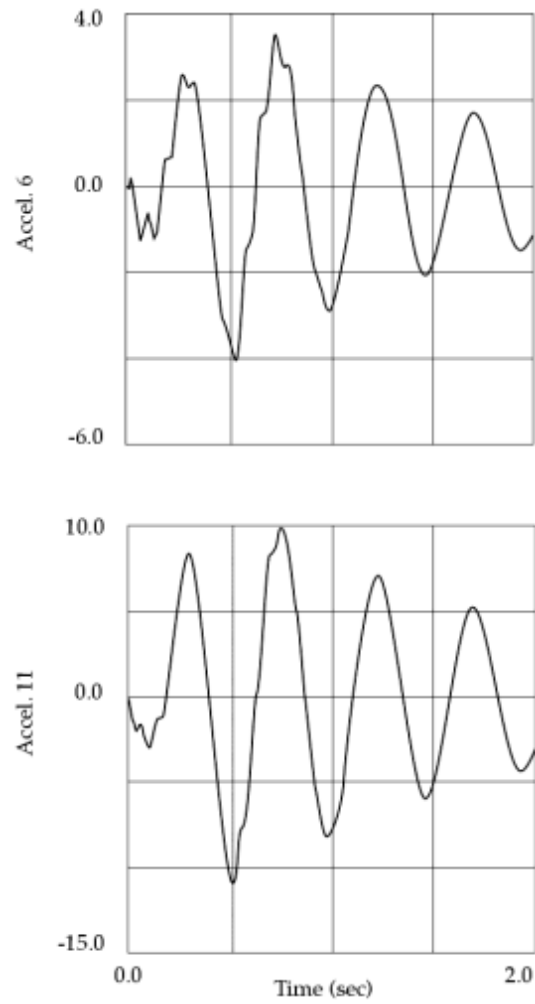


Figure 10-13. X-Y Plots for the Bar Transient Response Analysis

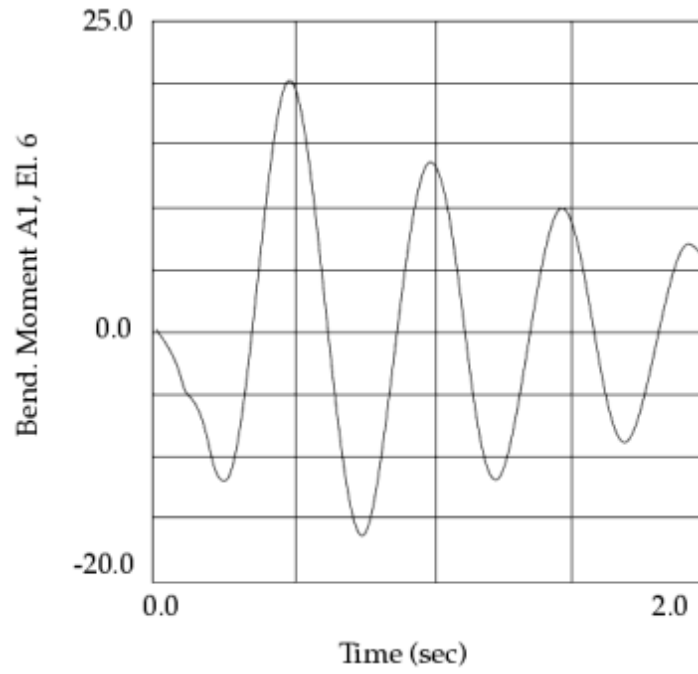


Figure 10-14. X-Y Plots for the Bar Transient Response Analysis

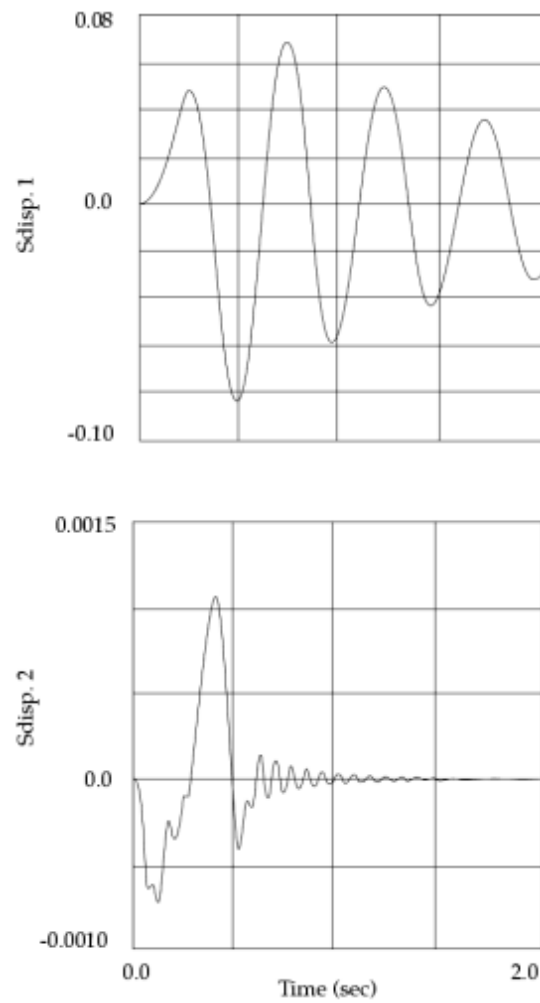


Figure 10-15. X-Y Plots for the Bar Transient Response Analysis

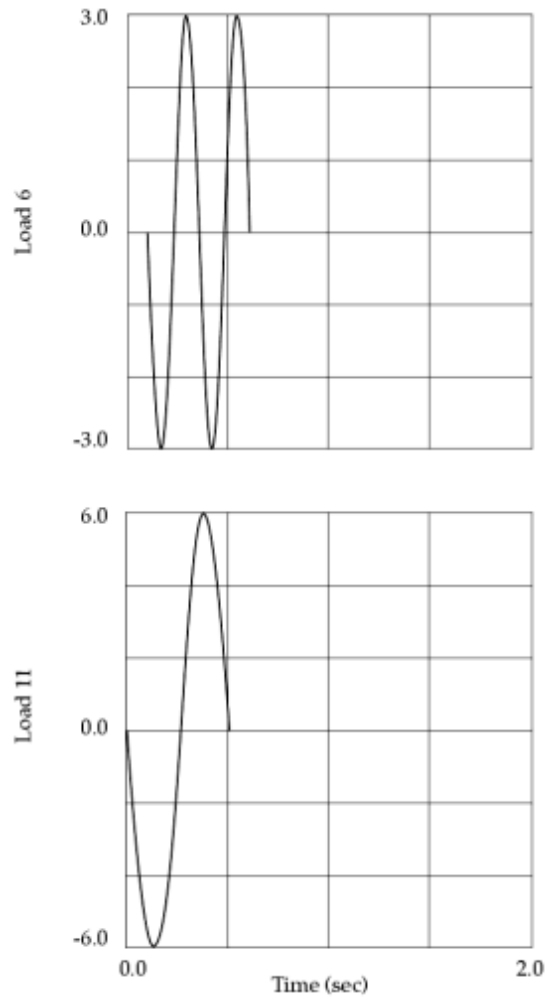


Figure 10-16. X-Y Plots for the Bar Transient Response Analysis

# 11. Guidelines for Effective Dynamic Analysis

## 11.1 Overview of Guidelines for Effective Dynamic Analysis

Dynamic analysis is more complicated than static analysis because of more input (mass, damping, and time- and frequency-varying loads) and more output (time- and frequency-varying results). Results from static analysis are usually easier to interpret, and there are numerous textbook solutions for static analysis that make it relatively easy to verify certain static analyses. Nevertheless, the guidelines in this chapter help you to perform dynamic analysis in a manner that will give you the same level of confidence in the dynamic results that you would have with static results.

This chapter covers the following topics:

- Overall analysis strategy
- Units
- Mass
- Damping
- Boundary conditions
- Loads
- Meshing
- Eigenvalue analysis
- Frequency response analysis
- Transient response analysis
- Results interpretation
- Computer resource estimation

## 11.2 Overall Analysis Strategy

Part of any analysis strategy, whether it be for dynamic analysis or static analysis, is to gain confidence with the modeling procedures first. The best way to accomplish this is to run small, simple models, preferably models that have textbook solutions. The references (see [“References and Bibliography”](#))

provide numerous textbook solutions. Start with a simple model first and then gradually add complexity, verifying the results at each stage. Follow the steps outlined below and in **Figure 11-1**.

Once you have confidence in a small model and are ready to analyze your actual model, again do the analysis in steps. The following is a suggested order for performing dynamic analysis on any structure:

1. Create the initial model only; do not apply any loads. Verify the model's connectivity, element and material properties, and boundary conditions. Make sure that mass is specified for this model.
  
2. Perform a static analysis (SOL 101) first in order to verify proper load paths and overall model integrity. (Note that you have to constrain the structure for static analysis even if you were not planning to do so for dynamic analysis.) For a three-dimensional model, you should run three load cases, each with a 1g gravity load applied in a different direction. Compute displacements and SPC forces, and verify the results. Check for unusually large grid point displacements and unreasonable SPC forces. Using a graphical postprocessor can aid you at this step.  
 Next, apply static loads that have the same spatial distribution that your subsequent dynamic loads have. Verify the results for reasonableness. Do not go to dynamic analysis until you are satisfied with the results from your static analysis.  
 It is recommended at this stage that the model contain PARAM,GRDPNT,n (where n is a reference grid point or 0, the origin of the basic coordinate system). Verify the results from the grid point weight generator in order to ensure that the model's rigid-body mass and inertia look reasonable. This step, in conjunction with the static analysis results, helps to ensure that the proper mass units are specified.
  
3. Perform an eigenvalue analysis (SOL 103) next. Compute only a few modes first, verify their frequencies, and view their mode shapes for reasonableness. If your graphical postprocessor can animate the mode shapes, do so because that helps you to visualize them. Things to check at this step are local mode shapes, in which one or a few grid points are moving a very large amount relative to the rest of the model (this can indicate poor stiffness modeling in that region), and unwanted rigid-body modes (which can arise due to improper specification of the boundary conditions or a mechanism).  
 Once you are satisfied with these results, perform the full eigenvalue analysis (for as many modes as you need).
  
4. If you have frequency-dependent loads, perform frequency response analysis (SOL 108 or SOL 111) using the dynamic load spatial distribution. If your structure is constrained, then apply the dynamic load at only one frequency, which should be 0.0 Hz. Compare the 0.0 Hz displacement results to the static analysis displacement results. The results should be the same if direct frequency response (without structural damping) is used. If the results are not equal, then there is probably an error in the specification of the dynamic load, and you should check the LSEQ and DAREA entries. If modal frequency response (without structural damping) is used, then the 0.0 Hz results should be close to the static results; the difference is due to modal truncation.  
 Next apply the load across the entire frequency range of interest. If you are running modal frequency response, then make sure that you have enough modes to ensure accurate results for even the highest forcing frequency. Also be sure to have a small enough  $\Delta f$  in order to accurately capture the peak response. Verify these results for reasonableness (it may be easier to look at magnitude and phase results instead of real and imaginary results, which are the default values).

If your ultimate goal is a transient response analysis for which damping is to be neglected, then the frequency response analysis can also omit damping. However, if damping is to be included, then use the correct damping in your frequency response analysis. The proper specification of damping can be verified by looking at the half-power bandwidth.

Plots are important at this stage to assist in results interpretation. X-Y plots are necessary in order to see the variation in response versus frequency. Deformed structure plots at a frequency near a resonant frequency can also help to interpret the results. If structure plots are made, look at the imaginary component because the single degree-of-freedom (SDOF) displacement response at resonance is purely imaginary when damping is present (this response does not occur in practice because the response is usually due to several modes).

5. If you have time-dependent loads, perform transient response (SOL 109 or SOL 112) analysis. If your structure is constrained, apply the load "very quickly" (over one or two time steps) as a step function and look at the displacement results. The duration of the analysis needs to be as long as the period of the lowest frequency mode. For an SDOF system, a quickly applied load results in a peak displacement response that is twice the response resulting from the same load applied statically. This peak response does not occur in your actual model because of multiple modes and damping, but the results should be close. If your structure is unconstrained, the displacements will grow with time (unless the rigid-body modes are excluded in a modal transient response analysis). However, the stresses should be roughly twice those from the static analysis. In any event, examine the results via X-Y plots to ensure reasonableness. Once you are satisfied, apply the correct time variation to the load and compute the results. Again, use X-Y plots to verify the accuracy of the results.
6. Finally, perform any other dynamic analyses, such as response spectrum analysis, random response, nonlinear transient response, or dynamic response optimization. The confidence gained by using the previous steps first helps to ensure that you have an accurate model at this stage.

These and other guidelines are described further in the remainder of this chapter.

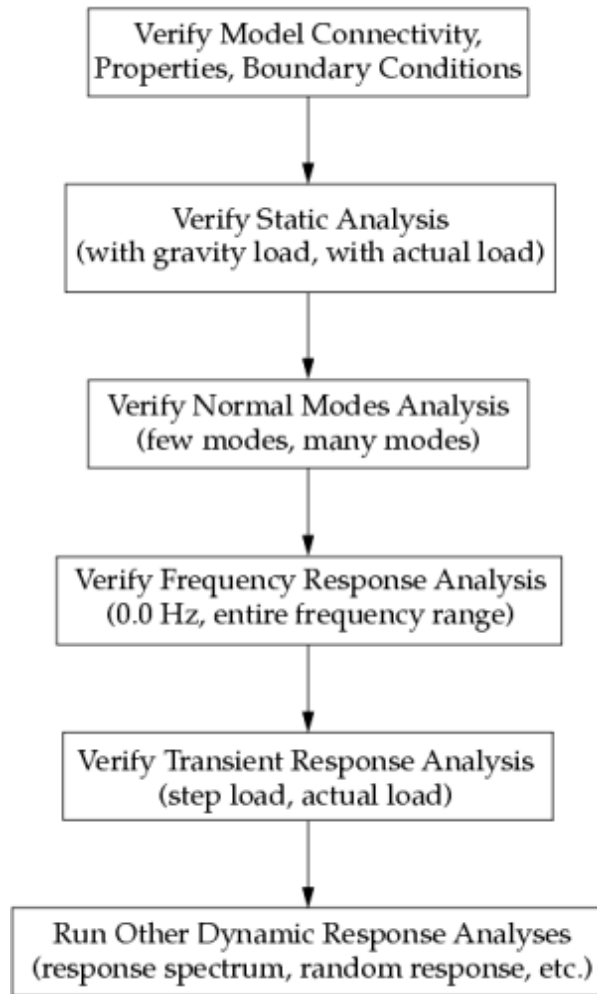


Figure 11-1. Simplified Flow Chart of the Overall Analysis Strategy

## 11.3 Units

Mistakes in units and boundary conditions (**Boundary Conditions**) are among the most common mistakes made when performing finite element analysis. Simcenter Nastran does not assume a particular set of units, but it does assume that they are consistent.

It is up to you to ensure that the units that you use are both consistent and correct. The table **Engineering Units for Common Variables** lists consistent units for common variables.

There are several ways to verify units. For mass, you can print the results from the grid point weight generator and verify that the mass is correct. For stiffness, you can apply a simple load and verify that the resulting static displacements seem reasonable. For both, you can verify that the natural frequencies are reasonable.

These checks assume that you have enough knowledge about your structure to know when the results are reasonable and when a mistake has been made. In other words, running small models and/or



proceeding through dynamic analysis via the steps outlined in the previous section are necessary in order to be confident that the correct units are specified.

## 11.4 Mass

Mistakes with mass primarily involve mistakes in mass units as described in the previous section. A common mistake is to mix mass and weight units. Using PARAM,WTMASS does not solve this problem because it scales all mass and weight input (except certain types of direct input matrices) and still leaves the mixture of units. Also, the use of PARAM,WTMASS can have unwanted effects as well because it also scales the large mass used in enforced motion, thereby scaling the value of the enforced motion input. In order to reduce the chances for error, it is recommended that only mass units be used everywhere. Therefore, avoid the use of weight input wherever possible as well as the use of PARAM,WTMASS.

One way to verify mass input is to apply gravity in several directions and print the SPC forces. This verification ensures that the mass is correctly applied and that the units are correct.

## 11.5 Damping

Proper specification of damping is probably the most difficult modeling input to verify because its verification can only be done via dynamic response analysis. In addition, its units are not necessarily familiar because damping is normally not a modeling input that you use frequently, unless dynamic response analysis is all you run. Also, there are several ways to specify damping in Simcenter Nastran, which increase the chances of making a mistake. Finally, even though the damping units are correct and the damping is input correctly in Simcenter Nastran, it is difficult to know that the damping specification itself is correct from a physical standpoint. While there are relatively easy checks for correctness in mass and stiffness input, including comparison to static and modal test data, there are no such easy checks for damping input.

The easiest way to specify damping is to use modal damping, which is often specified as the percentage of critical damping. The TABDMP1 Bulk Data entry is used to specify modal damping. The following are several potential mistakes associated with using modal damping.

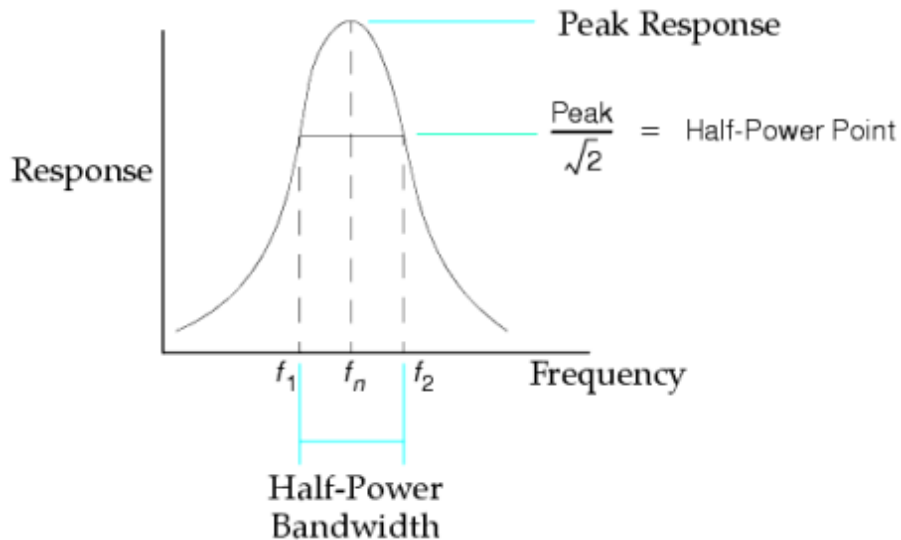
- Forgetting to select the TABDMP1 entry using the SDAMPING Case Control command.
- Forgetting to specify the damping TYPE (field 3). The default damping type is structural damping, which is twice the critical damping percentage.
- Not making the table cover an adequate range of frequencies. Like almost all of the Simcenter Nastran tables, the TABDMP1 entry extrapolates beyond the endpoints of the table by using the first two or the last two entered values. As a rule, you should try to provide enough input points so that the table lookup point is always an interpolated value.

Modal damping can be used only in modal frequency response and modal transient response. Other forms of damping have to be used for the direct methods of response. For frequency response analysis, GE (field 9 of the MAT1 entry) and PARAM,G define structural damping. These variables are also used to specify structural damping for transient response analysis but are not activated unless PARAM,W3 and PARAM,W4 are set to nonzero values. A common mistake is to forget to set these values.

Note that damping is additive, that is, the damping matrix is comprised of all of the input damping sources. For example, damping due to CDAMPI elements adds to damping due to PARAM,G and GE. Mixing of damping types can increase the chances for error, and you are cautioned against the mixing of damping types unless it is necessary.

In many cases damping is not an important consideration. For example, a structure's peak response due to an impulsive load is relatively unaffected by damping since the peak response occurs during the first cycle of response. Damping in a long duration transient excitation, such as an earthquake, can make a difference in the peak response on the order of 10 to 20% or so, but this difference is small when compared to the other modeling uncertainties. Therefore, it is often conservative to ignore damping in a transient response analysis.

For frequency response analysis, however, the value of damping is critical to the value of the computed response at resonance since the dynamic magnification factor is inversely proportional to the damping value. One way to verify the accuracy of the modal damping ratio input is to run the modal frequency response across the half-power bandwidth of the modal frequencies of interest as shown in **Figure 11-2**.



**Figure 11-2. Half-Power Bandwidth**

For lightly damped structures ( $\zeta < 0.1$ ), the approximate relationship between the critical damping ratio, the half-power bandwidth ( $f_2 - f_1$ ), and the resonant frequency  $f_n$  is

$$\zeta \approx \frac{f_2 - f_1}{2f_n}$$

**Equation 11-1.**

Another approximate way to verify damping is to run transient response analysis and look at the successive peak values of the free vibration response as shown in **Figure 11-3**. In direct transient

response analysis, this plot can be generated by using a displacement initial condition; in modal transient response analysis, it can be generated by applying an impulsive force that ramps up and down quickly with respect to the dominant period of response. The logarithmic decrement  $\Delta$  is the natural log of the amplitude ratio of two successive cycles of free vibration response given by

$$\Delta = \ln \frac{x_{(n-1)}}{x_n}$$

Equation 11-2.

The logarithmic decrement provides an approximate damping relationship for lightly damped structures as given by

$$\zeta \approx \frac{\Delta}{2\pi}$$

Equation 11-3.

In transient response analysis, remember to use PARAM,W3 or PARAM,W4 to include structural damping if GE or PARAM,G is used.

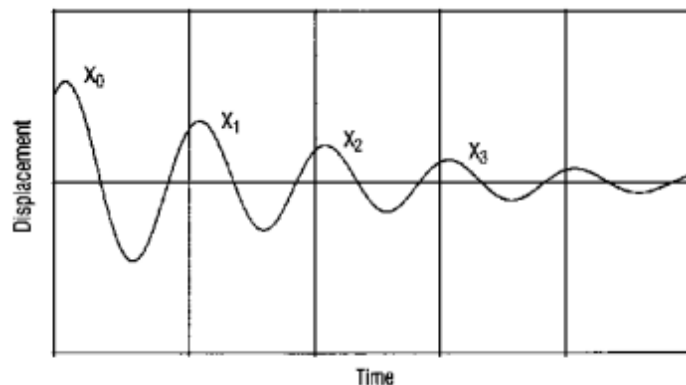


Figure 11-3. Damped Free Vibration Response

Both the half-power bandwidth method and the logarithmic decrement method assume an SDOF response. These approximations are less accurate when there are multiple modes of response; however, they are useful for verifying that the damping input is within a factor of two or three of the desired damping.

## 11.6 Boundary Conditions

The proper specification of boundary conditions is just as important for dynamic analysis as it is for static analysis. The improper specification of the boundary conditions leads to incorrect answers. One such improper specification of boundary conditions is forgetting to fully constrain the structure. Unlike static analysis, for which an under-constrained model does not run, an under-constrained model does run in dynamic analysis. You should perform an eigenvalue analysis first and verify that there are no unwanted rigid-body modes.

The large mass used for enforced motion simulates a constrained condition as well as adds a rigid-body mode. The value of the large mass is important for obtaining accurate answers. The large mass value must be large enough to properly simulate the constrained condition, but it must not be so large as to create numerical difficulties when solving the equations. A range of  $10^3$  to  $10^6$  times the overall structural mass is adequate to meet both conditions. One way to verify that a proper value is chosen is to run a normal modes analysis with the enforced DOF constrained via SPCs. Then, run a normal modes analysis with your choice of the large mass(es) and compare the frequencies of the flexible modes. If the frequencies compare favorably (i.e., to within four or five significant digits), then the large mass value is accurate. If the frequencies do not compare, then increase the value of the large mass(es). A common mistake is to use too low a value (or omit it entirely) for the rotational components. Using the grid point weight generator is very important to obtain the overall structural mass and inertias so that you can specify a good value for the large mass(es).

## 11.7 Loads

Because of their time- or frequency-varying nature, it is more complicated to apply dynamic loads than it is to apply static loads. Therefore, it is important to verify that the dynamic loads are correctly specified and that there are no discontinuous loads.

The best way to verify proper dynamic load specification is to plot the loads as a function of time or frequency. Another way to verify proper dynamic load specification is to print the loads as a function of time or frequency. Use the OLOAD Case Control command to plot or print the loads.

## 11.8 Meshing

An adequate element mesh is required to create an accurate model. For static analysis the mesh density is primarily controlled by the load paths; the element mesh must be fine enough so that there is a smooth transition of stress from one element to another in the region of interest.

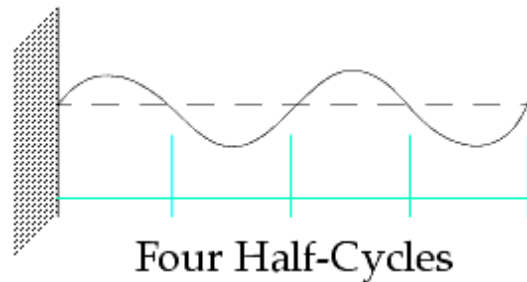
Load paths are also important for dynamic analysis, but there is an additional consideration: the mesh must be fine enough to accurately represent the deformed shape of the modes of interest. If many modes are to be considered in the analysis, then the model must be fine enough to accurately represent the mode shapes of the highest modes of interest. **Table 11-1** shows the frequencies resulting from several cantilever beam models; each column represents the same structure but with a different number of elements. Note that the frequencies are closer to the theoretical results as the model fineness is increased. In addition, note that the error is greatest in the higher modes. This table shows the computed frequencies compared to theory; it does not show the computed mode shapes compared to

theory. However, the comparison for mode shapes shows even more error than is shown for the frequencies.

A general rule is to use at least five to ten grid points per half-cycle of response amplitude. **Figure 11-4** shows the theoretical response shape for the fifth mode. Note that there are four half-cycles in the mode shape, which means that 20 to 40 grid points at a minimum are recommended to accurately represent that mode shape. This modeling guideline is also reflected in **Table 11-1**, which shows that the 40-element model is much more accurate than the 10-element model.

**Table 11-1. Frequencies for a Cantilever Beam Model (Lumped Mass)**

Mode	Frequencies (Hz)					
	Theory	10-Element Model	40-Element Model	Theory	70-Element Model	100-Element Model
1	2.08	2.07	2.08	2.08	2.08	2.08
2	13.24	12.81	13.00	13.24	13.01	13.01
3	36.47	35.50	36.37	36.47	36.41	36.42
4	71.52	68.84	71.22	71.52	71.34	71.36
5	118.21	112.53	117.65	118.21	117.90	117.95



**Figure 11-4. Fifth Mode Shape of a Cantilever Beam**

Another way to verify the accuracy of the mesh density is to apply static loads that give a deformed shape the same as the mode of interest and perform stress discontinuity calculations. This process can be laborious and is not recommended as a general checkout procedure.

## 11.9 Eigenvalue Analysis

In addition to the meshing guidelines described earlier, the other primary factor in eigenvalue analysis is the proper selection of the eigenvalue solution method. The Lanczos method is the best overall method and is the recommended method to use. The automatic Householder method is useful for small, dense matrices. The SINV method is useful when only a few modes are required. The other methods should be regarded as backup methods.

Carefully examine the computed frequencies and mode shapes. Viewing only one or the other is usually not enough to verify accuracy of your model. Modes with 0.0 Hz (or computational zero) frequencies

indicate rigid-body or mechanism modes. If these modes are unintended, then there is a mistake in boundary conditions or connectivity. The existence and cause of unintended zero-frequency modes can also be ascertained from the mode shapes. In addition, mode shape plots are useful for assessing local modes, in which a group of one or a few grid points displaces and the rest of the structure does not. Local modes may also be unintended and are often the result of incorrect connectivity or element properties.

## 11.10 Frequency Response Analysis

Several factors are important for computing accurate frequency response results. These factors include the number of retained modes (for modal frequency response analysis), the frequency increment  $\Delta f$ , and damping. These guidelines are only approximate but are nevertheless useful. Running a normal modes analysis first helps to compute accurate frequency response results.

### Number of Retained Modes

Use enough modes to cover the range of excitation. For example, if the model is to be excited from 1 to 100 Hz, you must use all of the modes with frequencies up to at least 100 Hz. This is only a minimum requirement, however. A better guideline is to use enough modes to cover two to three times the range of excitation in order to provide accurate answers at the high end of the frequency range. For example, when excitation is applied to 100 Hz, modes with frequencies up to 200 to 300 Hz should all be used.

### Size of the Frequency Increment

The size of the frequency increment  $\Delta f$  must be small enough to ensure that the magnitude of the peak response is accurately computed. To ensure this, you need to choose a frequency increment small enough so that there are at least five to ten increments within the half-power bandwidth frequencies (illustrated in the figure [Half-Power Bandwidth](#)).

The frequency increment is defined by  $\Delta f$  on the `FREQ1` Bulk Data entry. Note that `FREQ` and `FREQ2` entries can be used in conjunction with `FREQ1` to define more solutions in the areas of resonance (the frequencies of these solutions should have been determined by a prior normal modes analysis). A nonuniform  $\Delta f$  imposes no cost increase relative to a uniform  $\Delta f$ .

### Relationship of Damping to the Frequency Increment

The response at resonance is inversely proportional to the amount of damping, but the half-power bandwidth is directly proportional to the amount of damping. For lightly-damped structures ( $\zeta < 0.1$ ), an approximate relationship between the half-power bandwidth ( $f_2 - f_1$ ), resonant frequency  $f_n$ , and critical damping ratio  $\zeta$  is given by [Equation 11-4](#).

$$\zeta \approx \frac{f_2 - f_1}{2f_n}$$

Equation 11-4.

For more information, see [Damping](#).

This equation can be rewritten to define the appropriate value of  $\Delta f$  :

$$f_2 - f_1 = (m - 1)\Delta f$$

Equation 11-5.

$$\zeta \approx \frac{f_2 - f_1}{2f_n} = \frac{(m - 1)\Delta f}{2f_n}$$

Equation 11-6.

$$\Delta f = \frac{2\zeta f_n}{(m - 1)}$$

Equation 11-7.

where  $m$  is the number of frequency points within the half-power bandwidth. For example, with 1% critical damping and six points used to define the half-power bandwidth, the maximum frequency increment is  $0.0004 \cdot f_n$ . The frequency increment is smaller for lighter damped structures.

Another good check is looking at the X-Y plots. If the response curves are not smooth, there is a good chance that  $\Delta f$  is too large.

## Verification of the Applied Load

The applied load can be verified by exciting your model at 0.0 Hz and comparing the results to a static solution with the same spatial load distribution. The 0.0 Hz results should match the static results if direct frequency response analysis is used without structural damping. If modal frequency response analysis is used without structural damping, those results should be close to the static results; any difference is due to mode truncation. If the 0.0 Hz results do not match the static results, check the LSEQ and DAREA entries. Also, use OLOAD to print the applied force in order to compare the loads.

## 11.11 Transient Response Analysis

Several factors are important in computing accurate transient response. These factors include: the number of retained modes (for modal transient response), the integration time step  $\Delta t$ , the time duration of the computed response, and damping. The guidelines are only approximate but are nevertheless useful. Running a normal modes analysis first helps to compute transient response.

### Number of Retained Modes

In modal transient response analysis, a larger number of modes produces a more accurate response (although at the expense of increased run times). The number of modes must be large enough to cover the frequency range of interest. (The term "range of interest" means the range of frequencies whose response is to be computed as well as the range of frequencies for which the loading is applied.) As a general rule, there should be enough modes to cover a range up to about two times the highest frequency of interest. For example, if response needs to be computed to 100 Hz, then enough modes should be used in the transient response to cover up to at least 200 Hz. As another general rule, you should use enough modes to cover a range up to two to ten times the dominant frequency of response.

A sufficient number of modes must be retained to cover the time and spatial distribution of loading. For example, if the applied transient load has a spatial distribution resembling the fifth mode shape, then that mode should be included in the transient response.

### Size of the Integration Time Step

The value of the integration time step  $\Delta t$ , denoted by DT on the TSTEP Bulk Data entry, is important for the accurate integration of the equations of motion. Rough guidelines for the selection of  $\Delta t$  are as follows:

- $\Delta t$  must be small enough to accurately capture the magnitude of the peak response, which means that at least ten time steps per cycle of the highest mode be used. For example, if the highest frequency of interest is 100 Hz, then  $\Delta t$  should be 0.001 second or smaller.
- $\Delta t$  must be small enough to accurately represent the frequency content of the applied loading. If the applied loading has a frequency content of 1000 Hz, then  $\Delta t$  must be 0.001 second or less (preferably much less in order to represent the applied loading with more than one point per cycle).

The integration time step can be changed in a transient response analysis, but it is not recommended. Much of the cost of direct transient response occurs with the decomposition of the dynamic matrix, which occurs only once if there is a constant  $\Delta t$ . A new decomposition is performed every time  $\Delta t$  changes, thereby increasing the cost of the analysis if the time increment is changed. Therefore, the recommendation is to determine the smallest  $\Delta t$  required based on the loading and frequency range and then use that  $\Delta t$  throughout the entire transient response analysis.



## Duration of the Computed Response

The length of the time duration is important so that the lowest flexible (e.g., non-rigid body) mode oscillates through at least one cycle. For example, if the lowest flexible mode has a frequency of 0.2 Hz, then the time duration of the computed response should be at least 5.0second. A better guideline is to make the duration the longer of the following: twice the period of the lowest flexible mode or one period of the lowest flexible mode after the load has been removed or reached a constant value. The time duration is set on the TSTEP entry by multiplying (the integration time step) by the number of time steps (N).

## Value of Damping

The proper selection of the damping value is relatively unimportant for analyses that are of very short duration, such as a crash impulse or a shock blast. The specification of the damping value is most important for long duration loadings (such as earthquakes) and is critical for loadings (such as sine dwells) that continually add energy into the system.

## Verification of the Applied Load

The applied load can be verified by applying the load suddenly (over one or two time increments) and comparing the results to a static solution with the same spatial load distribution. The transient results should have a peak value of twice the static results. If the transient results are not twice the static results, check the LSEQ and DAREA entries. Another way to verify the applied load is to inspect it visually via the X-Y plots.

## 11.12 Results Interpretation and Verification

Because of the time- and frequency-varying nature of dynamic analysis, results can be more difficult to interpret than for static analysis. The key to proper results interpretation is plotting. Structure plotting is necessary for a proper understanding of the deformed shapes of the modes, and X-Y plotting is necessary for a proper understanding of the frequency and transient response.

Comparing results to hand calculations, to known results from similar models, or to test data is also very useful. Do not accept any analysis results without first performing some type of verification.

Consider the cantilever beam model shown in **Figure 11-5**. This is a planar model of the cantilever beam used in the examples in **Chapter 3** through **Chapter 7**. **Figure 11-5** shows the loads applied to the beam.

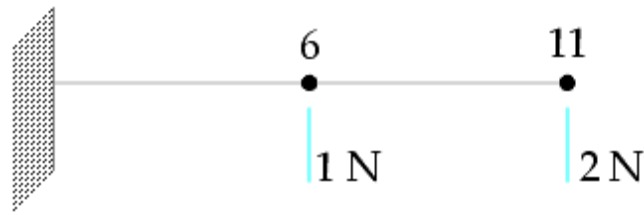


Figure 11-5. Cantilever Beam Model with Static Loads

A static analysis (SOL 101) is run first. Then, modal frequency response (SOL 111) is run from 0.0 to 20.0 Hz with a damping ratio of 2% critical damping used for all modes. Modes are computed up to 500 Hz using the Lanczos method. Finally, modal transient response (SOL112) is run with the time variation shown in **Figure 11-6**. Modes are computed up to 3000Hz (using the Lanczos method), and a damping ratio of 2% critical damping is used for all modes.

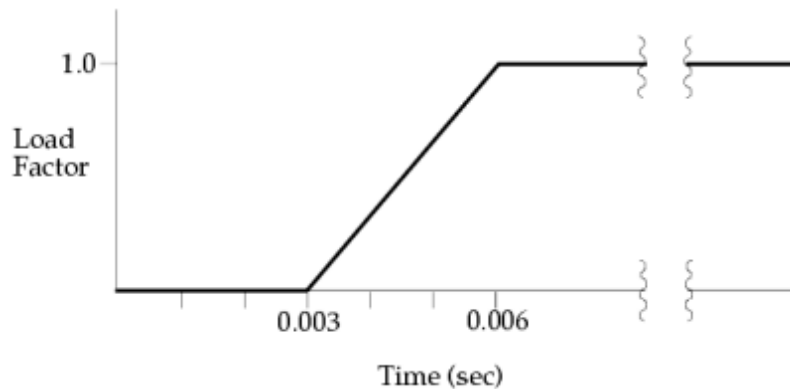


Figure 11-6. Time Variation of Transient Loads

**Figure 11-2** shows the y-displacements for grid points 6 and 11. As expected, the frequency response results at 0.0 Hz are the same as the static analysis results. The frequency response results at 2.05 Hz (very near the first natural frequency) are approximately 25 times the static analysis results. The factor of 25 is the dynamic amplification factor at resonance for a damping ratio of 2% critical damping.

$$\text{Amplification Factor at Resonance} = \frac{1}{2\zeta} = \frac{1}{2(.02)} = 25$$

The transient response results are approximately twice the static analysis results. The factor of two is the amplification of response for a transient load applied suddenly (see **"Transient Response Analysis"**).

Table 11-2. Comparison of Results for the Cantilever Beam Model

	Static Analysis Results	Frequency Response Analysis Results		Transient Response Peak Displacement
		0.0 Hz	2.05 Hz	
Displacement 6 Y	3.17E-3	3.17E-3	7.61E-2	6.29E-3
Displacement 11 Y	9.77E-3	9.77E-3	2.25E-1	1.87E-2

## 11.13 Computer Resource Requirements

The efficiency of a dynamic analysis can be measured based on computer resource requirements, such as computer runtime, disk storage, and memory. These requirements increase as the problem size increases, but they can also vary greatly depending on other factors.

In general, a dynamic analysis uses more computer resources than a static analysis. In normal modes analysis, the requirements vary depending on the density of the model, the eigenvalue extraction method used, the number of modes requested, and the type of decomposition (symmetric or sparse).

If your model has already been built, you can use the ESTIMATE utility to estimate resource requirements. See:

- “ESTIMATE” in the *Simcenter Nastran Installation and Operations Guide*

If you haven't yet created your model but still want to estimate resource needs, then read the following.

In the past, we have established benchmark runs to provide guidelines on performance for normal modes analyses. We used a cylindrical plate model scaled to various sizes and solved for ten modes using different solution methods, principally the Sparse Lanczos method. Testing has shown that this model provides a very good guideline for most industry models.

Memory requirements increase with problem size. Empirical formulas were developed for these benchmark problems to estimate the memory requirement given the number of degrees-of-freedom (DOF) contained in the model.

For the equation

$$\text{Log(mem)} = (.7934) \cdot \text{Log(DOF)} - 2.3671$$

Equation 11-8.

- mem            The memory required in megabytes.  
 DOF            The number of degrees-of-freedom contained in the model.

Note that these memory requirements are for no spill conditions. The symmetric solver can run with less memory, but because of spill conditions, the performance is degraded in both computer runtime and disk space usage.

Empirical formulas were developed for these benchmark problems to estimate the disk space requirement given the number of degrees-of-freedom (DOF) contained in the model.

For the equation

$$\text{Log(space)} = (1.0815) \cdot \text{Log(DOF)} - 2.5411$$

**Equation 11-9.**

space      The disk space usage in megabytes.

DOF        Represents the number of degrees-of-freedom contained in the model.

These values were obtained from a 32-bit computer and therefore should be doubled for a 64-bit computer. If you wish to calculate more than ten modes, then multiply the space calculated above by  $1.18^{(\#modes/10 - 1)}$ . For example, for 30 modes you would multiply the space value calculated above by  $1.18^2$ .

# 12. Advanced Dynamic Analysis Capabilities

## 12.1 Overview of Advanced Dynamic Analysis Capabilities

The previous chapters describe the most common types of dynamic analysis: normal modes analysis, frequency response analysis, and transient response analysis. However, Simcenter Nastran contains numerous additional types of dynamic analyses; many of these types are documented in the *Simcenter Nastran Advanced Dynamic Analysis User's Guide*.

The advanced dynamic analysis capabilities include

- Dynamic reduction
- Complex eigenanalysis
- Response spectrum analysis
- Random response analysis
- Mode acceleration method
- Fluid-structure interaction
- Nonlinear transient response analysis
- Superelement analysis
- Design sensitivity and optimization
- Control systems
- Aeroelastic analysis
- DMAP

These capabilities are described briefly in the remainder of this chapter.

## 12.2 Dynamic Reduction

Dynamic reduction is an optional numerical approach that can be used in Simcenter Nastran to reduce a dynamic model to one with fewer degrees-of-freedom. Typically, the intent of dynamic reduction is to

increase the efficiency of a dynamic solution by working with smaller matrices while maintaining the dynamic characteristics of the system.

Dynamic reduction is used for a number of reasons. One possible reason may be that a particular model may be too large from a computer resource standpoint (computer runtime and/or disk space) to be solved without using reduction. A related issue is that the model may have more detail than required. Many times dynamic analyses are performed on models that were initially created for detailed static stress analyses, which typically require refined meshes to accurately predict stresses. When these static models are used in a dynamic analysis, the detailed meshes result in significantly more detail than is reasonably required to predict the natural frequencies and mode shapes accurately.

Static condensation (also called Guyan reduction) is the available method in Simcenter Nastran for the dynamic reduction of the eigenequation prior to modal extraction.

Once the natural frequencies and mode shapes are calculated for the reduced model, these modes can be used in the transient or frequency response solution process.

## Static Condensation

In the static condensation process you select a set of dynamic DOFs called the a-set; these are the retained DOFs that form the analysis set. The complementary set is called the o-set and is the set of DOFs that are omitted from the dynamic analysis through a reduction process. The process distributes the o-set mass, stiffness, and damping to the a-set DOFs by using a transformation that is based on a partition of the stiffness matrix (hence the term static condensation). This reduction process is exact only for static analysis, which leads to approximations in the dynamic response. The a-set DOFs are defined by the ASET or ASET1 Bulk Data entries, and the o-set DOFs are defined by the OMIT or OMIT1 Bulk Data entries.

It is emphasized that dynamic reduction is an optional technique and is best left to the dynamic analysis specialist. Whereas dynamic reduction was required in the days of small computer memory and disk space, now it is no longer required due to increased computer resources and better eigenanalysis methods (in particular, the Lanczos method).

## 12.3 Complex Eigenvalue Analysis (SOL 107)

Complex eigenvalue analysis is necessary when the matrices contain unsymmetrical terms, damping effects, or complex numbers. It is used for the analysis of aeroelastic flutter, acoustics, and rotating bodies, among others. For more information regarding complex eigenvalue analysis, see "Complex Eigensolutions" in the *Simcenter Nastran Advanced Dynamic Analysis User's Guide*.

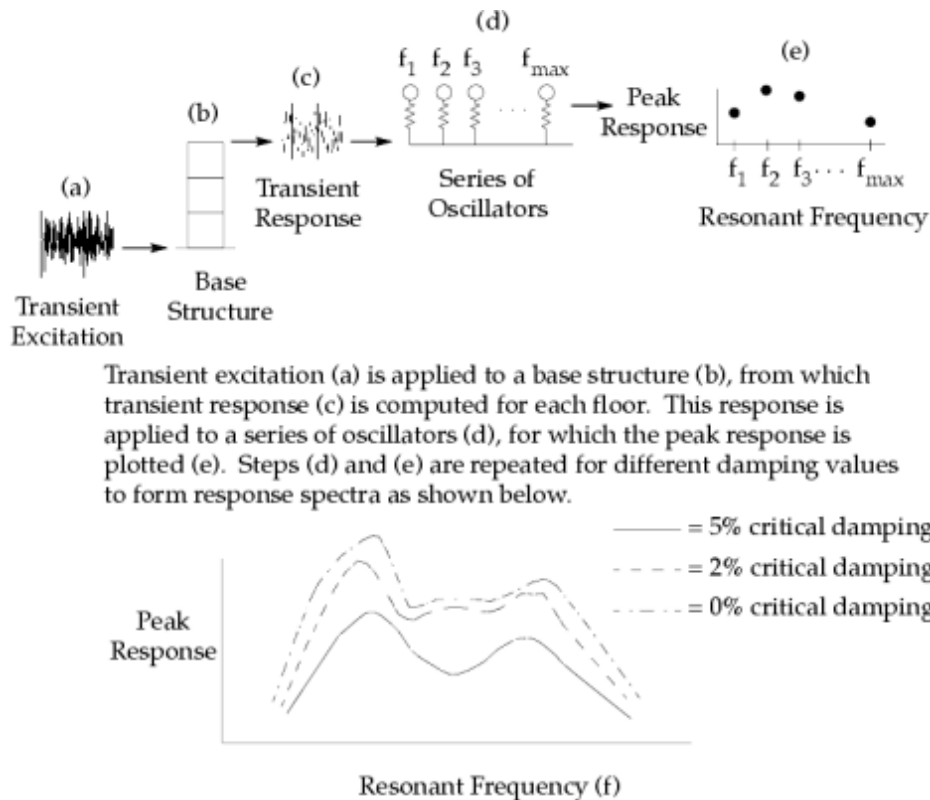
## 12.4 Response Spectrum Analysis

Response spectrum analysis is an approximate method of computing the peak response of a transient excitation applied to a simple structure or component. This method is used in civil engineering to predict the peak response of a component on a building that is subjected to earthquake excitation; it is also used in aerospace engineering to predict the peak response of equipment in a spacecraft that is

subjected to an impulsive load due to stage separation. Because it is an approximate method, response spectrum analysis is often used as a design tool. Response spectrum analysis is also called shock spectrum analysis.

There are two parts to response spectrum analysis: (1) generation of the spectrum and (2) use of the spectrum for dynamic response such as stress analysis. Both are available in Simcenter Nastran.

**Figure 12-1** depicts the generation of a response spectrum.



**Figure 12-1. Response Spectrum Generation**

Note that the peak response for one oscillator does not necessarily occur at the same time as the peak response for another oscillator. Also note that there is no phase information since only the magnitude of peak response is computed. It is assumed in this process that each oscillator mass is very small relative to the base structural mass so that the oscillator does not influence the dynamic behavior of the base structure.

Once a spectrum is computed, it can be used for the dynamic response analysis of a Simcenter Nastran model of the component. For example, the spectrum generated for a floor in a building that is subjected to an earthquake can then be applied to a complex model of a piece of equipment attached to that floor. The peak response of each mode of the equipment model is obtained from the spectrum, and these peak modal responses are combined to create the overall response.

Because the peak responses do not all occur at the same time and only the magnitude of peak responses are computed, various methods are used to combine the peak responses into the overall response. The combination methods implemented in Simcenter Nastran are SRSS (square root of the sum of the squares), ABS (absolute values), and NRL (U.S. Navy shock design modal summation). The typical response quantities computed are grid point displacements and element stresses.

## 12.5 Random Response Analysis

Random vibration is vibration that can be described only in a statistical sense. The instantaneous magnitude is not known at any given time; rather, the magnitude is expressed in terms of its statistical properties (such as mean value, standard deviation, and probability of exceeding a certain value).

Examples of random vibration include earthquake ground motion, ocean wave heights and frequencies, wind pressure fluctuations on aircraft and tall buildings, and acoustic excitation due to rocket and jet engine noise. These random excitations are usually described in terms of a power spectral density (PSD) function.

When you perform a random response analysis, Simcenter Nastran uses the results of a SOL 108 or 111 frequency response analysis and the PSD function to compute the random response.

- The results of the frequency response analysis generate a transfer function that relates the output response to the input excitation.
- The PSD function defines the input excitation. The PSD function can be in the form of auto- or cross-spectral densities.

Random response output consists of the following:

- Response PSD function and auto-correlation (ATOC) functions.
- Number of zero crossings with positive slope per unit time.
- Root-mean-square (RMS) and cumulative-root-mean-square (CRMS) values of the response.
- Zero-mean crossing for the RMS von Mises stress.

For more information on random analysis, see the "Random Analysis with Coupled Excitations" section of the *Advanced Dynamic Analysis User's Guide*.

## 12.6 Mode Acceleration Method

The mode acceleration method is an alternate form of data recovery for modal frequency response and modal transient response. The mode acceleration method accounts for the higher truncated modes to give more accurate answers than either the mode displacement or matrix methods (see **Modal Frequency Response Analysis** and **"Modal Transient Response Analysis"**), particularly if the number of retained modes is small compared to the number of physical degrees-of-freedom.



The higher modes respond in a quasi-static manner to lower frequency excitation. Therefore, the inertia and damping forces contain little contribution from these higher modes. In the mode acceleration method the inertia and damping forces are first computed from the modal solution. These forces are then combined with the applied forces and are used to obtain more accurate displacements by performing a static analysis. The mode acceleration method provides more accurate answers but with increased cost.

Because residual vectors also account for the static portion of the truncated modes, it is recommended that you do not use residual vectors when using the mode acceleration method.

The mode acceleration method is selected by specifying `PARAM,DDRMM,-1` and `PARAM,MODACC,0`.

## 12.7 Fluid structure interaction

Methods are available in Simcenter Nastran to model fluid effects. These methods are listed and described below.

### Virtual fluid mass

Small motions of incompressible fluids may be coupled directly to the structure with this option. Fluids may be coupled to the interior and exterior surfaces (with infinite fluid boundaries). There is no explicit fluid model; only the wetted structural elements (ELIST) have to be defined. Although free surfaces are allowed, no gravity effects are included directly. Because the fluid is represented by a coupled mass matrix attached directly to the structural points, this capability is allowed in all dynamic solution sequences. This capability may be used to model a wide variety of fluid-structure interaction problems. Some examples are fuel tanks, nuclear fluid containers, drilling platforms, and underwater devices.

### Acoustics and coupled acoustics

You can use Simcenter Nastran to perform an interior or an exterior acoustic analysis. You can model both fluid and structural meshes for a coupled vibro-acoustic analysis, or you can define a fluid only mesh to perform a pure acoustic analysis. The Simcenter Nastran acoustic formulation is linear, and assumes no flow of the fluid medium.

You model the 3-D fluid with CHEXA, CTETRA, CPENTA, or CPYRAM solid elements which reference a fluid property and material definition. Each fluid grid point includes a pressure degree-of-freedom. If your model includes both fluid and structure meshes, you can use the ACMODL bulk entry to request Simcenter Nastran to automatically couple the fluid and structure. You can also define acoustic absorber elements on the structural surfaces to represent a soundproofing material.

To represent a region exterior to the fluid and structure medium, you can use the Automatically Matched Layer (AML) and a microphone mesh. The AML represents the non-reflective acoustic boundary condition at the free fluid boundary.

Acoustic and coupled acoustic analyses can be performed using both the direct and modal dynamic solution sequences.

For more information, see the *Acoustics User's Guide*.

## 12.8 Nonlinear Transient Response Analysis (SOL 129)

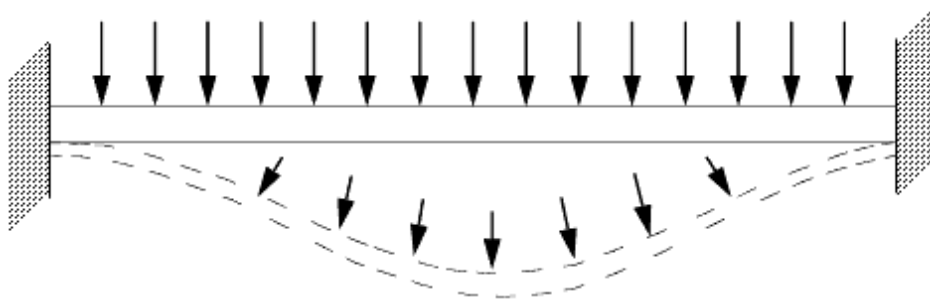
The analysis techniques described thus far are applicable for linear-elastic analysis for which the mass, stiffness, and damping matrices are constant over time and constant for all values of applied force. However, there are many cases for which the matrices are not constant, and these cases must be solved with nonlinear analysis techniques.

Nonlinear analysis requires iterative solution methods, thereby making it far more computationally intensive than a corresponding linear analysis. Nonlinear transient response analysis is available in Simcenter Nastran in SOL 129. Nonlinear problems are classified into three broad categories: geometric nonlinearity, material nonlinearity, and contact.

### Geometric Nonlinearity

Geometrically nonlinear problems involve large displacements; “large” means that the displacements invalidate the small displacement assumptions inherent in the equations of linear analysis. For example, consider a thin plate subject to an out-of-plane load. If the deflection of the plate’s midplane is approximately equal to the thickness of the plate, then the displacement is considered large, and a linear analysis is not applicable.

Another aspect of geometric nonlinear analysis involves follower forces. Consider a clamped plate subject to a pressure load. As shown in **Figure 12-2**, the load has followed the plate to its deformed position. Capturing this behavior requires the iterative update techniques of nonlinear analysis.



**Figure 12-2. Follower Forces on a Cantilever Beam**

### Material Nonlinearity

Material nonlinear analysis can be used to analyze problems where the stress-strain relationship of the material is nonlinear. In addition, moderately large strain values can be analyzed. Examples of material nonlinearities include metal plasticity, materials (such as soils and concrete), and rubbery materials (where the stress-strain relationship is nonlinear elastic). Various yield criteria, such as von Mises or Tresca (for metals) and Mohr-Coulomb or Drucker-Prager (for frictional materials, such as soils or concrete), can be selected. Three hardening rules are available in Simcenter Nastran: isotropic hardening, kinematic hardening, or combined isotropic and kinematic hardening. With such generality,

most plastic material behavior with or without the Bauschinger effect can be modeled. In addition, gaps can be used to model the effects due to structural separation.

## Contact

Contact occurs when two or more structures (or portions of structures) collide. In SOL 129, contact can be modeled as point-to-point contact (CGAP) or as contact along a line (BLSEG).

## Nonlinear-Elastic Transient Response Analysis

There are numerous structures that contain nonlinear elastic elements. These elements possess nonlinear force-deflection relationships, yet they are elastic since they load and unload along the same force-deflection curve. Examples of the nonlinear elastic force-deflection curves are shown in [Figure 12-3](#). These types of elements are common in many mechanical and structural systems including piping systems supported by nonlinear springs, chains (stiffness in tension only), or constant force devices; base-mounted equipment and structure supported by nonlinear shock isolation systems; structures with gaps (piping systems and buildings); and soil or concrete structures (which only exhibit stiffness when in compression). Other systems exhibit nonlinearities that are proportional to the velocity, such as seat belts and piping supports.

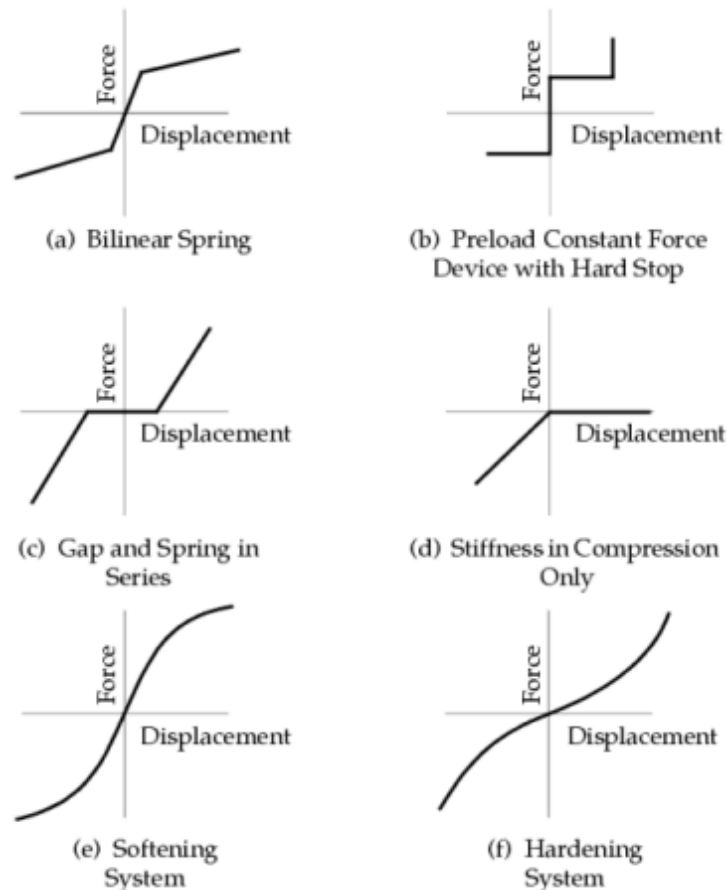


Figure 12-3. Examples of Nonlinear Elastic Systems

There are several methods in Simcenter Nastran for solving nonlinear elastic problems. A general nonlinear solution scheme can be used, in which the material properties are specified as nonlinear elastic (NLELAST on the MATS1 entry). Nonlinear element stiffness matrices are generated, and equilibrium iterations and convergence checks are required at each load step. The new CBUSH element, introduced in Version 69 adds some capabilities in this area, also. In addition, gap elements (CGAP) can be used in conjunction with elastic elements to produce systems with piece-wise linear force-deflection curves. This process also requires the additional computations of nonlinear element stiffness matrix generation, equilibrium iteration, and convergence testing. Nonlinear analyses requiring these additional computations can be substantially more costly than an equivalent linear analysis even if the nonlinearities are relatively few in number.

An efficient technique called the pseudoforce method exists in Simcenter Nastran, in which the localized nonlinearities are treated so that they require no additional computer time when compared to a purely linear analysis. In this method, which is available only for transient response solutions (SOLs 109 and 112, for example), deviations from linearity are treated as additional applied loads. The dynamic equations of motion are written as

$$[M]\{\ddot{u}(t)\} + [B]\{\dot{u}(t)\} + [K]\{u(t)\} = \{P(t)\} + \{N(t)\}$$

Equation 12-1.

where  $[M]$ ,  $[B]$ , and  $[K]$  denote the system mass, damping, and stiffness matrices, respectively. The vectors  $\{P(t)\}$  and  $\{u(t)\}$  denote applied nodal loads and system displacements, respectively, as functions of time. The vector  $\{N(t)\}$  denotes the nonlinear forces, which are added to the right-hand side of **Equation 12-1** (and hence are treated as additional applied loads).

The nonlinear forces are evaluated at the end of one time step for use in the successive time step. The equations of motion therefore become the following:

$$[M]\{\ddot{u}(t)\} + [C]\{\dot{u}(t)\} + [K]\{u(t)\} = \{P(t)\} + \{N(t - \Delta t)\}$$

Equation 12-2.

Note that the nonlinear force lags the true solution by a time step, which may require using small integration time steps (i.e., smaller than those required for a purely linear analysis). **Equation 12-2** can be solved in physical or modal coordinates (the nonlinearity itself must be expressed in terms of physical coordinates).

A nonlinear force can be used in conjunction with a linear elastic element to produce the desired force-deflection curve as illustrated in **Figure 12-4**. The nonlinear dynamic force is formulated using a NOLINi entry and a TABLEDi entry, which contains a force-versus-deflection table describing the nonlinear force. For desired force-deflection curves more complicated than the bilinear stiffness shown in the figure, the nonlinear force is made correspondingly more complex.

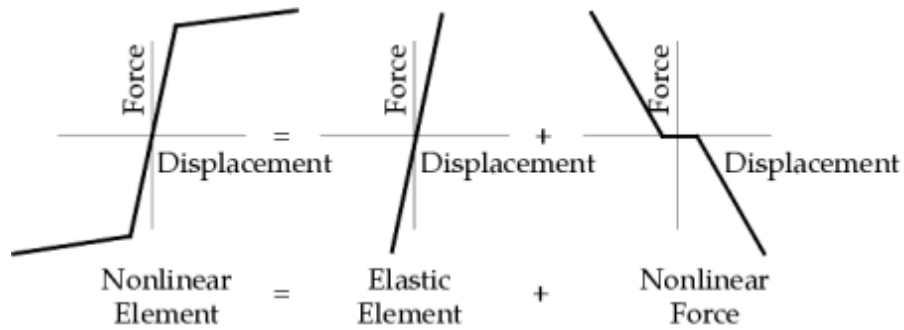


Figure 12-4. Formulation of a Nonlinear Element

## Nonlinear Normal Modes Analysis

There are times when normal modes need to be calculated for a nonlinear structure. For inflatable, rotating, or tension structures, the stiffness—and therefore, the modes—depends upon the applied load and deformation. A nonlinear analysis can be restarted into SOL 103 to compute the normal modes of the deformed structure.

### 12.9 Superelement Analysis

Superelement analysis is a form of substructuring wherein a model is divided into pieces with each piece represented by a reduced stiffness, mass, and damping matrix. Each superelement is processed individually, and the solutions are then combined to solve the entire model. The final analysis (in which all of the individual superelement solutions are combined) involves much smaller matrices than would be required if the entire model were solved in a one step process. An example of a superelement model is shown in [Figure 12-5](#).

Superelement analysis has the advantage of reducing computer resource requirements, especially if changes are made to only one portion (superelement) of the model; in this case, only the affected superelement needs to be reanalyzed and the final analysis repeated.

Superelement analysis presents procedural advantages as well, particularly when multiple engineering contractors are involved in an analysis. Imagine a model of a rocket and payload: one contractor models the booster, another contractor models the engines, and another contractor models the satellite. Each contractor can reduce his model to its boundary degrees-of-freedom, which is suitable for superelement analysis. The systems integrator then combines these reduced models into one model for a liftoff analysis. Superelement analysis has the advantage that matrices can be passed from one organization to another without revealing proprietary modeling details or concern about whether the same superelement interior grid point and element numbers are used by every participant.

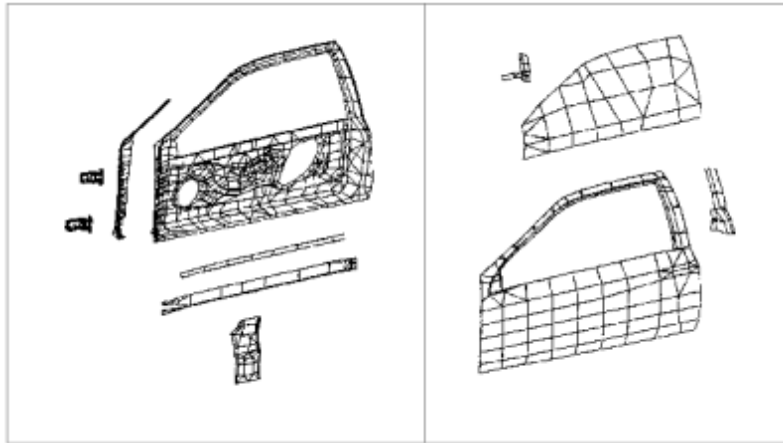


Figure 12-5. Superelements Used to Model a Car Door

## 12.10 Design Optimization and Sensitivity (SOL 200)

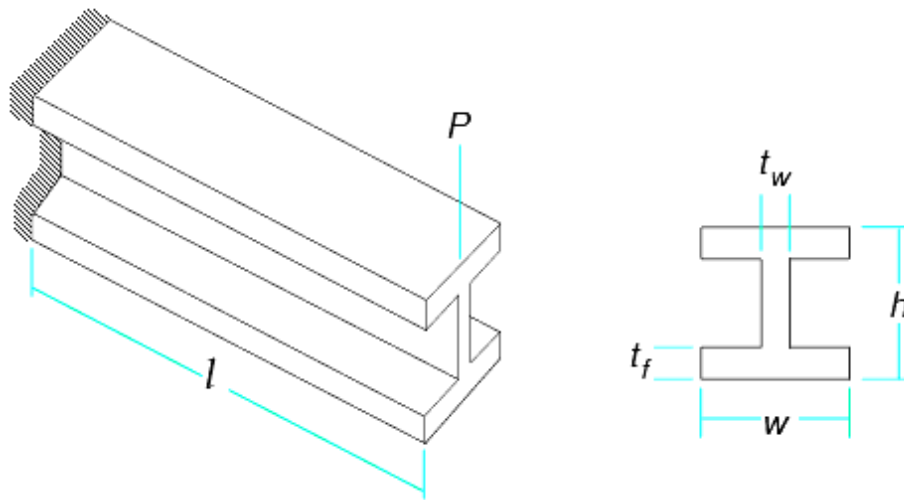
Design optimization is used to produce a design that possesses some optimal characteristics, such as minimum weight, maximum first natural frequency, or minimum noise levels. Design optimization is available in Simcenter Nastran in SOL 200, in which a structure can be optimized considering simultaneous static, normal modes, buckling, transient response, frequency response, and aeroelastic analyses.

Possible analysis response types include:

- Weight
- Volume
- Eigenvalues
- Buckling load factor
- Static displacement, stress, strain, and element force
- Composite stress, strain, and failure criterion
- Frequency response displacement, velocity, acceleration, stress, and force
- Transient response displacement, velocity, acceleration, stress, and force
- Damping level in a flutter analysis
- Trim and stability derivative responses for static aeroelastic analysis

For more information on optimization capabilities, see Introduction to Design Sensitivity and Optimization in the *Design Sensitivity and Optimization User's Guide*.

A frequent task is to minimize the total structural weight while ensuring that the design still satisfies all of the design-related and performance-related constraints. The structural weight is termed the design objective, which the optimizer attempts to minimize. The design variables are the properties that can be changed to minimize the weight. The design variables shown in **Figure 12-6** are the height and width of the flange and web. Constraints on the design variables may be a minimum allowable flange width  $w$  and a maximum allowable beam height  $h$ . With the tip load shown in the figure, one performance constraint on the design may be the maximum allowable transverse displacement at the tip, while another constraint may be the maximum allowable bending stress. The design-related and performance-related constraints place limits on the optimizer's ability to reduce the weight.



**Figure 12-6. Cantilever I-Beam Example**

The optimizer in Simcenter Nastran uses sensitivity analysis. The sensitivity matrix  $[S]$  defines the change in response for a perturbation in design variable. A term in  $[S]$  is given by

$$S_{ij} = \frac{\partial R_i}{\partial X_j}$$

**Equation 12-3.**

where:

$S_{ij}$	=	$i, j$ term in $[S]$
$\partial$	=	partial derivative
$R_i$	=	$i$ -th response
$X_j$	=	$j$ -th design variable

The sensitivity is depicted in **Figure 12-7**, in which the sensitivity is the slope of the curve of response versus the design variable value. Design optimization and design sensitivity are described in greater depth in the *Simcenter Nastran Design Sensitivity and Optimization User's Guide*.

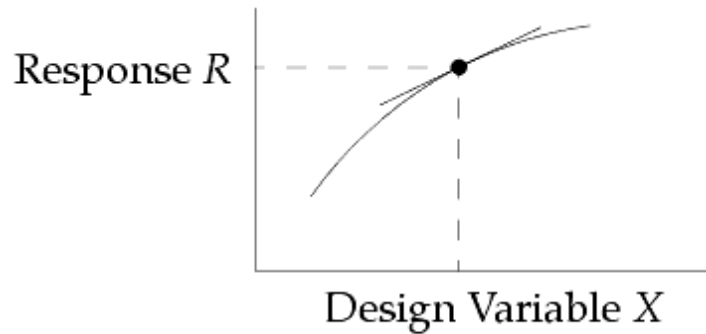


Figure 12-7. Response Sensitivity

## 12.11 Control System Analysis

A control system provides feedback (output) to an input. In aircraft, sample control systems are flap settings and actuator positions.

A control system involves a general input-output relationship called a transfer function. Transfer functions are represented in Simcenter Nastran by the TF Bulk Data entry. Transfer functions can utilize grid points, extra points (EPOINT), and scalar points (SPOINT). Nonlinearities can be simulated via the NOLINI entries with which nonlinear transient loads are expressed as functions of displacements or velocities. Complex eigenvalue analysis can be used to determine stability when control systems include damping and unsymmetrical matrices.

## 12.12 Aeroelastic Analysis

Simcenter Nastran provides efficient solutions of the problems of aeroelasticity, which is a branch of applied mechanics that deals with the interaction of aerodynamic, inertial, and structural forces. It is important in the design of airplanes, helicopters, missiles, suspension bridges, tall chimneys, and power lines. Aeroservoelasticity is a variation in which the interaction of automatic controls requires additional consideration.

### Note:

For more information on aeroelasticity capabilities, see Introduction to Aeroelastic Analysis and Design in the *Aeroelastic Analysis User's Guide*.

The primary concerns of aeroelasticity include flying qualities (that is, stability and control), flutter, and structural loads arising from maneuvers and atmospheric turbulence. Methods of aeroelastic analysis differ according to the time dependence of the inertial and aerodynamic forces that are involved. For the



analysis of flying qualities and maneuvering loads wherein the aerodynamic loads vary relatively slowly, quasi-static methods are applicable. The remaining problems are dynamic, and the methods of analysis differ depending on whether the time dependence is arbitrary (that is, transient or random) or simply oscillatory in the steady state.

Simcenter Nastran considers two classes of problems in dynamic aeroelasticity: aerodynamic flutter and dynamic aeroelastic response.

## Aerodynamic Flutter

Flutter is the oscillatory aeroelastic instability that occurs at some airspeed at which the energy extracted from the airstream during a period of oscillation is exactly dissipated by the hysteretic damping of the structure. The motion is divergent in a range of speeds above the flutter speed. Flutter analysis utilizes complex eigenvalue analysis to determine the combination of airspeed and frequency for which the neutrally damped motion is sustained (see [Figure 12-8](#)).

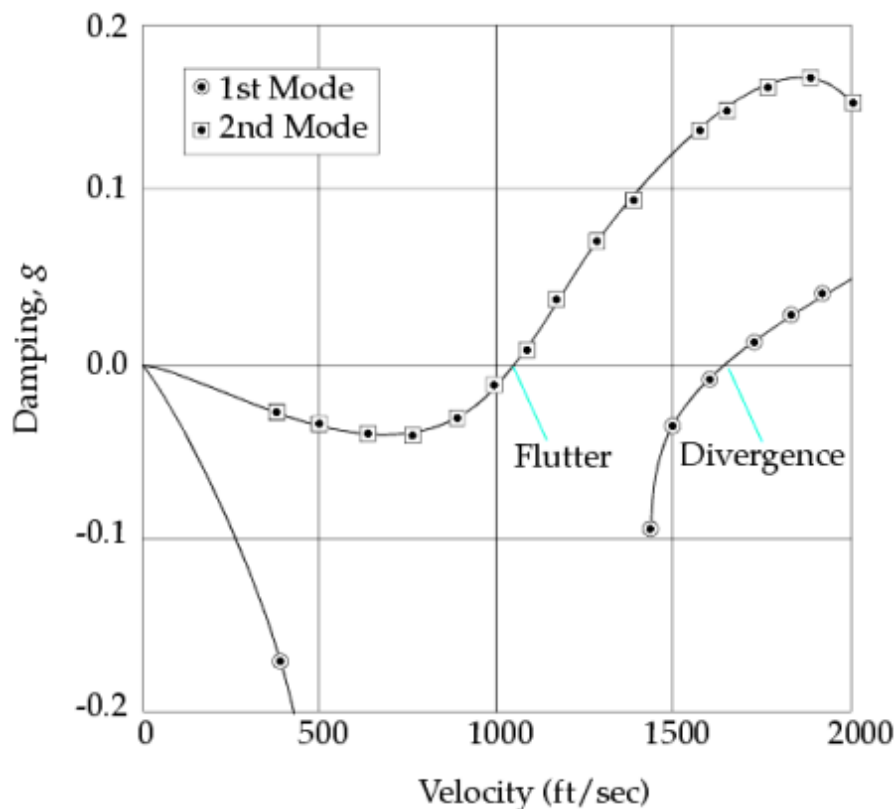


Figure 12-8. Flutter Stability Curve

Three methods of flutter analysis are provided in Simcenter Nastran: the American flutter method (called the K method in Simcenter Nastran), an efficient K method (called the KE method) for rapid flutter evaluations, and the British flutter method (called the PK method) for more realistic representation of the unsteady aerodynamic influence as frequency-dependent stiffness and damping terms. Complex eigenvalue analysis is used with the K method, and the QR transformation method is used with the KE and PK methods.

## Dynamic Aeroelastic Response

The aeroelastic dynamic response problem determines the response of the aircraft to time- or frequency-varying excitations. Atmospheric turbulence is the primary example of this type of excitation, but wind shear and control surface motion can also have an aeroelastic component. Methods of generalized harmonic (Fourier) analysis are applied to the linear system to obtain the response to the excitation in the time domain. The gust response analysis may be regarded either as a stationary random loading or as a discrete gust.

The gust analysis capability computes the response to random atmospheric turbulence, discrete one-dimensional gust fields, and control surface motion and other dynamic loading. The random response parameters calculated are the power spectral density, root mean square response, and mean frequency of zero-crossings. The response to the discrete gust and control surface motion is calculated by direct and inverse Fourier transform methods since the oscillatory aerodynamics are known only in the frequency domain. The time histories of response quantities are the output in the discrete case (see [Figure 12-9](#)).

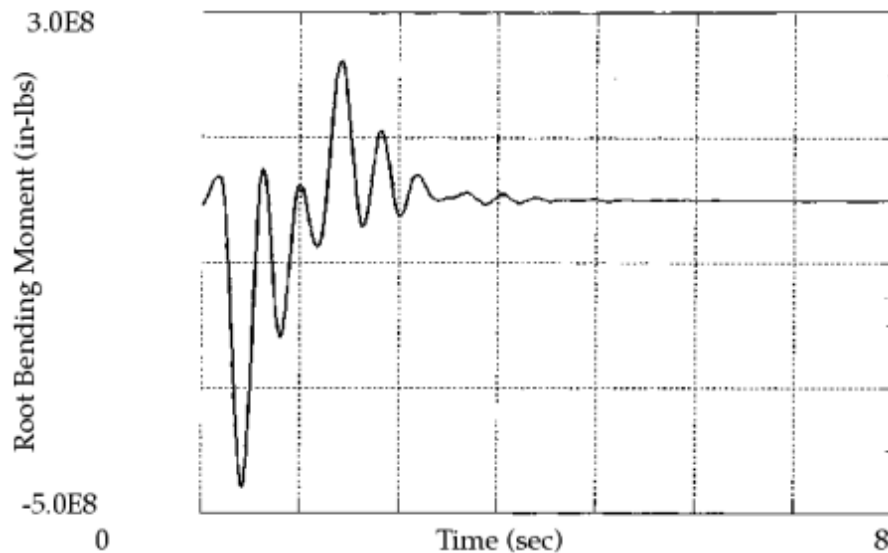


Figure 12-9. Transient Response Resulting from a Gust

### 12.13 DMAP

Simcenter Nastran provides a series of solution sequences (SOL 103 for normal modes analysis, for example) written in their own language tailored to matrix manipulation. These solution sequences consist of a series of DMAP (direct matrix abstraction program) statements. The DMAP capability enables you to modify these solution sequences and write your own solution sequences.

DMAP is a high-level language with its own compiler and grammatical rules. DMAP statements contain data blocks and parameters and operate on them in a specified manner. For example, the DMAP statement

```
ADD U1,U2/U3 $
```

adds matrices U1 and U2 together and calls the output U3. The DMAP statement

```
MATPRN U3// $
```

prints the matrix U3.

Numerous DMAP alters for dynamic analysis are provided (see the *Simcenter Nastran Release Guide* for further information). Many of these DMAP alters are for dynamic analysis. There are alters for model checkout, modal initial conditions, the addition of static results to transient response results, frequency-dependent impedance, and modal test-analysis correlation, among others. In summary, DMAP is a powerful capability that allows you to modify Simcenter Nastran to satisfy your needs.

For more information on DMAP, see Introduction to DMAP in the *DMAP Programmer's Guide*.



# A. Glossary of Terms

<b>CMS</b>	Component mode synthesis.
<b>Critical Damping</b>	The lowest value of damping for which oscillation does not occur when the structure is displaced from its rest position. Values of damping less than the critical damping value create an underdamped system for which oscillatory motion occurs.
<b>Damping</b>	Energy dissipation in vibrating structures.
<b>DOF</b>	Degree-of-freedom.
<b>Dynamic Amplification Factor</b>	Ratio of dynamic response to static response, which is a function of the forcing frequency, natural frequency, and damping.
<b>Forced Vibration Analysis</b>	Vibration response due to applied time-varying forces.
<b>Free Vibration Analysis</b>	Vibration response when there is no applied force. Normal modes analysis and transient response to initial conditions are examples of free vibration analysis.
<b>Frequency Response Analysis</b>	Computation of the steady-state response to simple harmonic excitation.
<b>LMT</b>	Lagrange multiplier technique.
<b>Mechanism Mode</b>	Stress-free, zero-frequency motions of a portion of the structure. A mechanism can be caused by an internal hinge.
<b>Mode Shape</b>	The deformed shape of a structure when vibrating at one of its natural frequencies.
<b>Natural Frequency</b>	The frequency with which a structure vibrates during free vibration response. Structures have multiple natural frequencies.
<b>Phase Angle</b>	The angle between the applied force and the response. In Simcenter Nastran this angle is a phase lead, whereby the response leads the force.
<b>Positive Definite</b>	A matrix whose eigenvalues are all greater than zero.
<b>Positive Semi-Definite</b>	A matrix whose eigenvalues are greater than or equal to zero.
<b>Repeated Roots</b>	Two or more identical natural frequencies.
<b>Resonance</b>	Large-amplitude vibrations that can grow without bound. At resonance, energy is added to the system. This occurs, for example, when the harmonic excitation frequency is equal to one of the natural frequencies. The response at resonance is controlled entirely by damping.
<b>Rigid-Body Drift</b>	Transient displacements of an unconstrained model that grow continuously with time. This is often caused by the accumulation

<b>Rigid-Body Mode</b>	of small numerical errors when integrating the equations of motion.
<b>SDOF</b>	Stress-free, zero-frequency motions of the entire structure.
<b>Structural Damping</b>	Single degree-of-freedom.
<b>Transient Response Analysis</b>	Damping that is proportional to displacement.
<b>Viscous Damping</b>	Computation of the response to general time-varying excitation.
	Damping that is proportional to velocity.

# B. Nomenclature for Dynamic Analysis

The appendix provides nomenclature for terms commonly used in dynamic analysis.

## B.1 General

$\cdot$	Multiplication
$\approx$	Approximately
[ ]	Matrix
{ }	Vector
$i$	$\sqrt{-1}$
$g$	Acceleration of Gravity
$\Delta t$	Time Step
$\Delta f$	Frequency Step
$i, j, k$	Subscripts (Indices)
$\infty$	Infinity
$u$	Displacement
$u(0)$	Initial Displacement
$\dot{u}$	Velocity
$\dot{u}(0)$	Initial Velocity
$\ddot{u}$	Acceleration
$m$	Mass
$m_0$	Large Mass
$b$	Damping
$b_{cr}$	Critical Damping

$k$	Stiffness
$p$	Applied Force
$\omega$	Circular Frequency
$\omega_n$	Circular Natural Frequency
$\omega^2$	Eigenvalue
$\lambda$	Eigenvalue
$f$	Frequency
$f_n$	Natural Frequency
$T_n$	Period
$\omega_d$	Damped Circular Natural Frequency
$\zeta$	Damping Ratio
$Q$	Quality Factor
$\theta$	Phase Angle
$\Delta$	Logarithmic Decrement

## B.2 Structural Properties

### Geometry

$L$	Length
$A$	Area



**Stiffness**

$E$	Young's Modulus
$G$	Shear Modulus
$J$	Torsional Constant
$\nu$	Poisson's Ratio
$I$	Area Moment of Inertia

**Mass**

$\rho$	Mass Density
$\rho_m$	Mass Density
$\rho_w$	Weight Density
$I_p$	Polar Moment of Inertia

**Damping**

$G$	Overall Structural Damping Coefficient
$G_E$	Element Structural Damping Coefficient

**B.3 Multiple Degree-of-Freedom System**

$\{u\}$	Displacement Vector
$\{\dot{u}\}$	Velocity Vector
$\{\ddot{u}\}$	Acceleration Vector
$\{\phi\}$	Mode Shape Vector
$\{\phi_{rig}\}$	Rigid-Body Modes

$\xi_i$	i-th Modal Displacement
$\dot{\xi}_i$	i-th Modal Velocity
$\ddot{\xi}_i$	i-th Modal Acceleration
$m_j$	j-th Generalized Mass
$k_j$	j-th Generalized Stiffness
$\lambda_k$	k-th Eigenvalue
$[M]$	Matrix
$[M_r]$	Rigid-Body Mass Matrix
$[K]$	Stiffness Matrix
$[K_E]$	Element Stiffness Matrix
$[B]$	Damping Matrix
$[P]$	Force Matrix

# C. The Set Notation System Used in Dynamic Analysis

The set notation system used for dynamic analysis in Simcenter Nastran continues and expands the set notation system for static analysis. Because of the great variety of physical quantities and displacement sets used in dynamic analysis, you should understand the set notation system in Simcenter Nastran.

## C.1 Displacement Vector Sets

When you create a model in Simcenter Nastran, equations are allocated for the purpose of assembling the necessary structural equations. These equations are rows and columns in the matrix equations that describe the structural behavior. Six equations are created per grid point, one equation is created per SPOINT or EPOINT. The model details (elements, properties, loads, etc.) are used to create the appropriate row and column entries in the matrices (e.g., stiffness coefficients). Certain data entries (i.e., SPCs, MPCs, ASETs, etc.) cause matrix operations to be performed in the various stages of the solution process. To organize the matrix operations, Simcenter Nastran assigns each DOF to displacement sets.

Most matrix operations used in a structural analysis involve partitioning, merging, and/or transforming matrix arrays from one subset to another. All components of motion of a given type form a vector set which is distinguished from other vector sets. A given component of motion can belong to many sets.

In Simcenter Nastran, there are two basic types of sets:

- combined sets
- mutually exclusive sets

Combined sets are formed through the union (i.e., combination) of two or more sets. Mutually exclusive sets are important in the solution process because if a DOF is a member of a mutually exclusive set, it cannot be a member of any of the other mutually exclusive sets. The importance of this property is demonstrated as the sets are described below.

When Simcenter Nastran starts to assemble the structural equations, it allocates six equations for each grid point (GRID) and 1 equation for each scalar point (SPOINT). These equations are associated with a displacement set defined as the g-set. Fundamentally, the g-set represents an unconstrained set of structural equations.

The next step in the solution process is to partition the g-set into two subsets: the m-set and the n-set. The dependent DOFs of all multipoint constraint relations (MPCs, RBEs, etc.) that define the m-set are condensed into a set of independent DOFs (the n-set). The n-set represents all of the independent DOFs that remain after the dependent DOFs were removed from the active set of equations.

Using the n-set, SPCs are applied to the independent equations to further partition the equations. The degrees-of-freedom defined on SPC entries define the DOFs in the s-set. When you reduce the n-set by

applying the s-set constraints, the f-set remains. The f-set is the "free" DOF of the structure. At this stage of the solution, the f-set is comprised of the remaining equations that represent a constrained structure. If the applied constraints are applied properly, the f-set equations represent a statically stable solution.

If static condensation is to be performed, the f-set is partitioned into the o-set and the a-set. The o-set degrees-of-freedom are those that are to be eliminated from the active solution through a static condensation. The remaining DOFs reside in the a-set. The a-set is termed the "analysis" set. The a-set is often the partition at which the solution is performed.

If the SUPORT entry is used, the degrees-of-freedom defined on the SUPORT entry are placed in the r-set. When the r-set is partitioned from the t-set, the l-set remains. This final set is termed the "leftover" set and is the lowest level of partitioning performed in Simcenter Nastran static analysis. The l-set partition is the matrix on which the final solution is performed. Under special circumstances, the l-set is divided into two types of DOFs. The DOFs that are held fixed in component mode synthesis (CMS) are called b-set points, and those DOFs that are free to move in CMS are called c-set points.

**C-1** demonstrates the basic partitioning operations. When a particular set above has no DOFs associated with it, it is a null set. Its partition is then applied, and the DOFs are moved to the subsequent partition.

**Table C-1. Basic Partitioning Operations**

g-set – m-set = n-set
n-set – s-set = f-set
f-set – o-set = a-set
a-set – r-set = l-set

In the above table, the m-, s-, o-, and r-sets form the mutually exclusive sets. Physically, the mutually exclusive set partitioning ensures that operations cannot be performed on DOFs that are no longer active. For example, if you apply an SPC to a DOF which is a dependent degree-of-freedom on an RBAR, a FATAL error is issued. Using an SPC entry moves a DOF to the s-set, but this cannot occur if the same DOF is already a member of the m-set. Both the m-set and s-set are mutually exclusive.

**Table C-1** is a representation of all sets, set partitions, and set combinations used in Simcenter Nastran.

A number of additional, mutually independent sets of physical displacements (namely, q-set, c-set, b-set and e-set) are used in dynamic analysis to supplement the sets used in static analysis. The q-, c-, and b-sets facilitate component mode synthesis. The e-set is used to represent control systems and other nonstructural physical variables. The p-set is created by combining the g-set with the e-set variables.

In addition to the combined sets described above, the v-set is a combined set created by combining the c-, r- and o-sets. The DOFs in these sets are the DOFs free to vibrate during component mode synthesis.

Some additional sets (sa-, k-, ps-, and pa-sets) represented in **Figure C-1** are defined and used in aeroelastic analysis.

The modal coordinate set  $\xi$  is separated into zero frequency modes  $\xi_o$  and elastic (finite frequency) modes  $\xi_f$ . For dynamic analysis by the modal method, the extra points  $u_e$  are added to the modal coordinate set  $\xi_f$  to form the h-set as shown in [Figure C-2](#).

The parameter PARAM,USETPRT can be used to print of lists of degrees-of-freedom and the sets to which they belong. Different values of the PARAM create various tables in the printed output.

The supersets formed by the union of other sets have the following definitions.

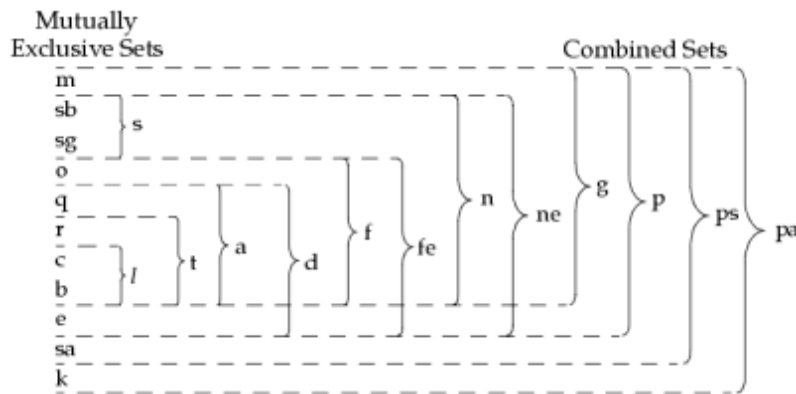


Figure C-1. Combined Sets Formed from Mutually Exclusive Sets

The set names have the definitions described in [Table C-2](#).

Table C-2. Sets in Simcenter Nastran

Mutually Exclusive Sets	
Set Name	Meaning
m	Points eliminated by multipoint constraints.
sb <sup>1</sup>	Points eliminated by single-point constraints that are included in boundary condition changes and by the automatic SPC feature.
sg <sup>1</sup>	Points eliminated by single-point constraints that are listed on grid point Bulk Data entries.
o	Points omitted by structural matrix partitioning.
q	Generalized coordinates for dynamic reduction or component mode synthesis.
r	Reference points used to determine free body motion.
c	The free boundary set for component mode synthesis or dynamic reduction.
b	Coordinates fixed for component mode analysis or dynamic reduction.

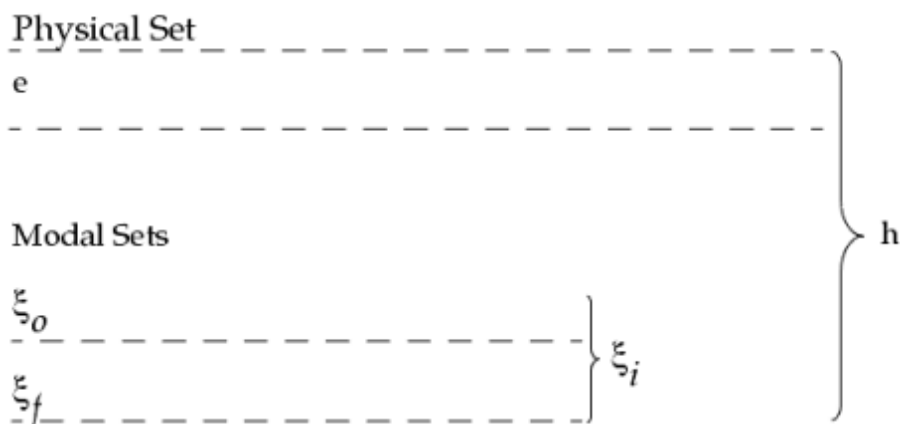
Mutually Exclusive Sets	
Set Name	Meaning
e	Extra degrees-of-freedom introduced in dynamic analysis.
sa	Permanently constrained aerodynamic points.
k	Aerodynamic points.

<sup>1</sup>Strictly speaking, sb and sg are not exclusive with respect to one another. Degrees-of-freedom may exist in both sets simultaneously. These sets are exclusive, however, from the other mutually exclusive sets.

**Table C-3. Sets in Simcenter Nastran**

Combined Sets (+ Indicates Union of Two Sets)	
Set Name	Meaning
$s = sb + sg$	All points eliminated by single-point constraints.
$l = b + c$	The structural coordinates remaining after the reference coordinates are removed (points left over).
$t = l + r$	The total set of physical boundary points for superelements.
$a = t + q$	The set assembled in superelement analysis.
$d = a + e$	The set used in dynamic analysis by the direct method.
$f = a + o$	Unconstrained (free) structural points.
$fe = f + e$	Free structural points plus extra points.
$n = f + s$	All structural points not constrained by multipoint constraints.
$ne = n + e$	All structural points not constrained by multipoint constraints plus extra points.
$g = n + m$	All structural (grid) points including scalar points.
$p = g + e$	All physical points.

Combined Sets (+ Indicates Union of Two Sets)	
Set Name	Meaning
$ps = p + sa$	Physical and constrained aerodynamic points.
$pa = ps + k$	Physical set for aerodynamics.
$fr = o + l$	Statically independent set minus the statically determinate supports ( $o + l = f - q - r$ ).
$v = o + c + r$	The set free to vibrate in dynamic reduction and component mode synthesis.



where:

- $\xi_0$  = rigid-body (zero frequency) modal coordinates
- $\xi_f$  = finite frequency modal coordinates
- $\xi_i$  =  $\xi_0 + \xi_f$  = the set of all modal coordinates
- $u_h$  =  $\xi_i + u_e$  = the set used in dynamic analysis by the modal method

Figure C-2. Set Notation for Physical and Modal Sets





## D. Solution Sequences for Dynamic Analysis

The solution sequences (SOLs) for the dynamic analysis types are listed below. The SOLs are listed for the (preferred) structured solution sequences, rigid formats, and unstructured superelement solution sequences.

SOL Number	SOL Name	Description
103	SEMODES	Normal Modes
108	SEDFREQ	Direct Frequency Response
109	SEDTRAN	Direct Transient Response
111	SEMFREQ	Modal Frequency Response
112	SEMTRAN	Modal Transient Response

The *Simcenter Nastran Quick Reference Guide* lists all of the solution sequences for all of the Simcenter Nastran analysis types.



# E. Case Control Commands for Dynamic Analysis

This section lists the Case Control commands that are often used for dynamic analysis. Commands that apply to statics, such as FORCE and STRESS, are not listed. The dynamic analysis Case Control commands are listed alphabetically. The description of each command is similar to that found in the *Simcenter Nastran Quick Reference Guide*. The descriptions in this guide have been edited to apply specifically to the dynamic analysis capabilities described herein.

The *Simcenter Nastran Quick Reference Guide* describes all of the Case Control commands.

The Case Control commands described in this appendix are summarized as follows:

## Input Specification

B2GG	Direct Input Viscous Damping Matrix Selection
K2GG	Direct Input Stiffness Matrix Selection
K42GG	Direct Input Structural Damping Matrix Selection
M2GG	Direct Input Mass Matrix Selection

## Analysis Specification

BC	Boundary Condition Identification
DLOAD	Dynamic Load Set Selection
FREQUENCY	Frequency Set Selection
IC	Transient Initial Condition Set Selection
METHOD	Real Eigenvalue Extraction Method Selection
SDAMPING	Structural Damping Selection
SUPPORT1	Fictitious Support Set Selection
TSTEP	Transient Time Step Set Selection for Linear Analysis

## Output Specification

ACCELERATION	Acceleration Output Request
DISPLACEMENT	Displacement Output Request
MODES	Subcase Repeater

OFREQUENCY	Output Frequency Set
OLOAD	Applied Load Output Request
OTIME	Output Time Set
SACCELERATION	Solution Set Acceleration Output Request
SDISPLACEMENT	Solution Set Displacement Output Request
SVECTOR	Solution Set Eigenvector Output Request
SVELOCITY	Solution Set Velocity Output Request
VELOCITY	Velocity Output Request

# F. Bulk Data Entries for Dynamic Analysis

This appendix lists the Bulk Data entries that are often used for dynamic analysis. Entries that apply to generic modeling or statics, such as GRID or FORCE, are not listed. Bulk Data entries are listed alphabetically. The description of each entry is similar to that found in the Simcenter Nastran Quick Reference Guide. The descriptions in this guide have been edited to apply specifically to the dynamic analysis capabilities described herein.

The *Simcenter Nastran Quick Reference Guide* describes all of the Bulk Data entries.

The Bulk Data entries described in this appendix are summarized as follows:

## Mass Properties

CMASS1	Scalar Mass Connection
CMASS2	Scalar Mass Property and Connection
CMASS3	Scalar Mass Connection to Scalar Points Only
CMASS4	Scalar Mass Property and Connection to Scalar Points Only
CONM1	Concentrated Mass Element Connection, General Form
CONM2	Concentrated Mass Element Connection, Rigid Body Form
PMASS	Scalar Mass Property

## Damping Properties

CDAMP1	Scalar Damper Connection
CDAMP2	Scalar Damper Property and Connection
CDAMP3	Scalar Damper Connection to Scalar Points Only
CDAMP4	Scalar Damper Property and Connection to Scalar Points Only
CVISC	Viscous Damper Connection
PDAMP	Scalar Damper Property
PVISC	Viscous Element Property
TABDMP1	Modal Damping Table

## Normal Modes

EIGR	Real Eigenvalue Extraction Data
EIGRL	Real Eigenvalue Extraction Data, Lanczos Method

## Dynamic Loading

DAREA	Dynamic Load Scale Factor
DELAY	Dynamic Load Time Delay
DLOAD	Dynamic Load Combination or Superposition
DPHASE	Dynamic Load Phase Lead
LSEQ	Static Load Set Definition
TABLED1	Dynamic Load Tabular Function, Form 1
TABLED2	Dynamic Load Tabular Function, Form 2
TABLED3	Dynamic Load Tabular Function, Form 3
TABLED4	Dynamic Load Tabular Function, Form 4

## Frequency Response

FREQ	Frequency List
FREQ1	Frequency List, Alternate Form 1
FREQ2	Frequency List, Alternate Form 2
FREQ3	Frequency List, Alternate Form 3
FREQ4	Frequency List, Alternate Form 4
FREQ5	Frequency List, Alternate Form 5
RLOAD1	Frequency Response Dynamic Load, Form 1
RLOAD2	Frequency Response Dynamic Load, Form 2

## Transient Response

TIC	Transient Initial Condition
TLOAD1	Transient Response Dynamic Load, Form 1
TLOAD2	Transient Response Dynamic Load, Form 2
TSTEP	Transient Time Step

## Miscellaneous

DMIG	Direct Matrix Input at Points
SUPPORT	Fictitious Support
SUPPORT1	Fictitious Support, Alternate Form





# G. Parameters for Dynamic Analysis

A variety of parameters are used for dynamic analysis. Some of the more frequently used parameters are listed below alphabetically. For a description of these parameters, see the *Simcenter Nastran Quick Reference Guide*.

ASING

AUTOSPC

BAILOUT

CB1, CB2

CK1, CK2

CK41, CK42

CM1, CM2

COUPMASS

CP1, CP2

CURVPLOT

DDRMM

DYNSPCF

EPZERO

G

HFREQ

KDAMP

LFREQ

LMODES

MAXRATIO

MODACC

NONCUP

NOSORT1

WMODAL

WTMASS

W3, W4, W3FL, W4FL

# H. File Management Section

## H.1 Overview of File Management Section

This appendix provides an overview of how the database structure and the File Management Section (FMS) work so that you can allocate your computer resources efficiently, especially for large models. For many problems, due to the default values, the use or knowledge of the FMS is either transparent or not required by you.

## H.2 Definitions

Before presenting the details of the database description, it is helpful to define some of the basic terms that are used throughout this appendix.

DBset	Database set. This consists of a Simcenter Nastran logical name, such as MASTER, DBALL, etc., which refers to an entity within the Simcenter Nastran database. It consists of a collection of physical files.
Database	Collection of all DBsets assigned to a run.
Data block	Matrix or table (e.g., KAA, LAMA) that is stored in the database.
Logical name	Local internal name (log-name) used by Simcenter Nastran for a file or DBset.
Word	For 32- and 64-bit machines, each word is equivalent to four and eight bytes, respectively.
Buffsize	Length of a Simcenter Nastran buffer in terms of words (32- or 64-bit words). It controls the physical record size for data storage/transfer that is contained in many Simcenter Nastran logical units. The default and maximum allowable buffsize is machine dependent. The default value is recommended except for large problems. It can be modified by using the following NASTRAN statement:

```
NASTRAN BUFFSIZE = xxxxx
```

where xxxxx is the desired buffsize.

Block	This is often referred to as a Simcenter Nastran GINO block. Each GINO block contains one Simcenter Nastran buffer.
{ }	A brace indicates that the quantity within this bracket is mandatory. The underlined item within { } is the default value.
[ ]	A square bracket indicates that the quantity within this bracket is optional.

## H.3 Simcenter Nastran Database

When submitting a Simcenter Nastran job, a series of standard output files is created (e.g., F06 file). Conventions for the filenames are machine dependent. Furthermore, four permanent (MASTER, DBALL, USRSOU, USROBJ) and one scratch (SCRATCH) DBsets are created during the run.

MASTER	This is the “directory” DBset that contains a list of all the DBsets used in the job, all the physical file names assigned, and an index pointing to all the data blocks created and where they are stored. In addition, it also contains the Simcenter Nastran Data Definition Language (NDDL) used. NDDL is the internal Simcenter Nastran language that describes the database. You do not need to understand NDDL to use Simcenter Nastran effectively. The default maximum size for MASTER is 5000 blocks.
DBALL	This is the DBset where the permanent data blocks are stored by default. The default maximum size is 25000 blocks.
USRSOU	This DBset stores the source file for user-created DMAP. The default maximum size is 5000 blocks.
USROBJ	This DBset stores the object file for user-created DMAP. The default maximum size is 5000 blocks.
SCRATCH	This DBset is used as the temporary workspace for Simcenter Nastran. In general, this DBset is deleted at the end of the run. The default maximum size is 350100 blocks.

For most solutions, USRSOU and USROBJ are not needed and may be deleted or assigned as temporary for the duration of the run (see **“ASSIGN”**).

For a Simcenter Nastran input file named “dyn1.dat”, the following sample submittal command can be used:

```
nastran dyn1 scr=no
```

where nastran is the name of shell script for executing Simcenter Nastran. The following four physical database files are created as a result of the above command.

```
dyn1.MASTER
dyn1.DBALL
dyn1.USROBJ
dyn1.USRSOU
```

Unless otherwise stated, the input filename is assumed to be “dyn1.dat” in this appendix.

## H.4 File Management Commands

The File Management Section is intended primarily for the attachment and initialization of Simcenter Nastran database sets (DBsets) and FORTRAN files. For many problems, due to the default values, the use of the FMS is either transparent or not required by you. At first glance the FMS may be

overwhelming due in part to its flexibility and the many options it has to offer. However, in its most commonly used form, it can be quite simple. Examples in the most commonly used format are presented throughout this appendix.

If an FMS statement is longer than 72 characters, it can be continued to the next line by using a comma as long as the comma is preceded by one or more blank spaces.

## INIT

### Purpose

Creates/initializes permanent and/or temporary DBsets. The INIT statement has two basic formats: one for all the DBsets and one specifically for the SCRATCH DBsets.

### Format (Simplified) for All DBsets Except SCRATCH

```
INIT DBset-name [LOGICAL=(log-name1(max-size1) ,
    log-name2(max-size2) , ...lognamei(max-sizei) ,
    ...lognamen(max-sizen) ]
```

DBset-name	Logical name of the DBset being used (e.g., DBALL).
log-name <sub>i</sub>	i-th logical name for the DBset-name referenced in this INIT statement. You can have up to 10 logical names for each DBset ( $1 \leq i \leq 10$ ). The i-th physical file is assigned with the "assignment" statement.
max-size <sub>i</sub>	Maximum allowable number of Simcenter Nastran blocks which may be written to the i-th file.

### Example

```
INIT DBALL LOGICAL=(DBALL(50000))
```

This statement creates the DBALL DBset with a logical name of DBALL and the maximum size of 50000 Simcenter Nastran blocks instead of 25000 blocks, which is the default value for DBALL. Unless an ASSIGN statement is also used, the physical file is given the name dyn1.DBALL assuming that your input file is called "dyn1.dat".

```
INIT DBALL LOGICAL=(DB1(35000) , DBTWO(60000))
```

This statement creates the DBALL DBset with logical names of DB1 and DBTWO. Two physical files, dyn1.DB1 and dyn1.DBTWO, are created with a maximum size of 35000 and 60000 Simcenter Nastran blocks, respectively.

## Format (Simplified) for the SCRATCH DBset

```
INIT SCRATCH [LOGICAL=(log-name1(max-size1) ,
  log-name2(max-size2) , ...log-namei(max-sizei) ) ,
  SCR300=(log-namei+1(max-sizei+1) , ...log-namen(max-sizen))] ]
```

Log-name1 through log-namei are allocated for regular scratch files as temporary workspace. This temporary workspace is not released until the end of the job. SCR300 is a special keyword which indicates that the log-names are members reserved for DMAP module internal scratch files. The space occupied by these SCR300 files is for the duration of the execution of the module. This SCR300 space is released at the end of the module execution. You can have up to a combined total of 10 logical names for the SCRATCH DBset ( $1 \leq i \leq 10$ ).

### Example

```
INIT SCRATCH LOGICAL=(SCR1(150000) , SCR2(100000) ) ,
  SCR300=(SCRA(250000) , SCRB(300000) )
```

This statement creates the SCRATCH DBset with logical names of SCR1, SCR2, SCRA, and SCRB. Two physical files, dyn1.SCR1 and dyn1.SCR2, are created with a maximum size of 150000 and 100000 blocks, respectively. These two files are regular scratch files. Two additional physical files, dyn1.SCRA and dyn1.SCRB, are created with a maximum size of 250000 and 300000 blocks, respectively. These two files are SCR300 type files.

## ASSIGN

### Purpose

Assigns physical filenames to logical filenames or special FORTRAN files that are used by other FMS statements or DMAP modules.

### Format (Simplified) to Assign Logical Files

```
ASSIGN log-namei='filenamei' [TEMP DELETE]
```

log-namei	The i-th logical name for the DBset created by the INIT statement.
TEMP	Requests that filenamei be deleted at the end of the job. This is optional and is often used for USRSOU and USROBJ.
DELETE	Requests that filenamei be deleted if it exists before the start of the run. This is optional; however, if this option is not used and the FORTRAN file exists prior to the current run, then the job may fail with the following messages:

```
*** USER FATAL MESSAGE 773 (DBDEF)
THE FOLLOWING PHYSICAL FILE ALREADY EXISTS
LOGICAL NAME = XXXX
```

```

PHYSICAL FILE = YYYY
USER INFORMATION: NO ASSOCIATED DEFAULT FILES OR
ASSIGNED DBSETS CAN
EXIST PRIOR TO THE DATA BASE INITIALIZATION RUN,
USER ACTION: DELETE THIS FILE AND RESUBMIT THE JOB

```

DELETE is not a suggested option if you are using RESTART since you can delete your database inadvertently. Manual deletion of unwanted databases is a safer approach.

## Example

```

ASSIGN DB1='sample.DB1'
INIT DBALL LOGICAL=(DB1(50000))

```

These statements create a physical file called sample.DB1 for the logical name DB1 in the current directory. Without the ASSIGN statement, the physical file name created is called dyn1.DB1, assuming once again that your input file is called dyn1.dat.

```

ASSIGN DB1='/mydisk1/se/sample.DB1'
ASSIGN DB2='/mydisk2/sample.DB2'
INIT DBALL LOGICAL=(DB1(50000),DB2(40000))

```

Two logical names DB1 and DB2 are created for the DBset DBALL. DB1 points to a physical file called sample.DB1 that resides in the file system (directory) /mydisk1/se. DB2 points to a physical file called sample.DB2 that resides in the file system (directory) /mydisk2.

## Format (Simplified) to Assign FORTRAN Files

```

ASSIGN DB1='/mydisk1/se/sample.DB1'
ASSIGN DB2='/mydisk2/sample.DB2'
INIT DBALL LOGICAL=(DB1(50000),DB2(40000))

```

**log-key** This is the logical keyword for the FORTRAN file being created. The default values depend on the keyword. Acceptable keywords are

```

DBC, DBMIG, INPUTT2, INPUTT4, OUTPUT2, OUTPUT4
,
DBUNLOAD, DBLOAD, and USERFILE.

```

You should reference the *Simcenter Nastran Quick Reference Guide* for detailed descriptions and the defaults for these keywords.

filenamef	This is the physical name of the FORTRAN file.
STATUS	Specifies whether the FORTRAN file will be created (STATUS = new) or is an existing file (STATUS = old).
UNIT	Specifies the FORTRAN unit (e.g., UNIT = 12).
FORM	Specifies whether the file written is in ASCII (FORM = FORMATTED) or binary (FORM = UNFORMATTED) format.
DELETE	Requests that filenamef be deleted if it exists before the start of the run.

### Example

```
ASSIGN OUTPUT2='sample.out', STATUS=NEW, UNIT=11, FORM=FORMATTED,
DELETE
```

This example creates a new FORTRAN file to be used for OUTPUT2 operations. This file is in ASCII format with a physical filename of sample.out and is assigned to unit 11.

## EXPAND

### Purpose

Concatenates files into an existing DBset in order to increase the allowable disk space. The EXPAND statement is normally used in a restart run when you run out of disk space in your previous run.

### Format

```
EXPAND DBset-name LOGICAL=(log-namei(max-sizei), ... )
```

DBset-name	Logical name of the DBset to be expanded by the addition of new members to this existing DBset.
log-namei	Logical name of the i-th member of the DBset. An ASSIGN statement should be used to point this logical name to a physical file.
max-sizei	Maximum allowable number of Simcenter Nastran blocks that may be written to the i-th member.

### Example

The original run creates a database with a name dyn1.DBALL. This database is filled and the job fails with the following error messages in the F06 file:

```
*** USER FATAL MESSAGE 1012 (GALLOC)
DBSET DBALL IS FULL AND NEEDS TO BE EXPANDED.
```



For small to medium size problems, it is best to rerun the job from the beginning with a larger file allocation. For large problems, if rerunning the job is not practical, then the database can be expanded with the following statements:

```
RESTART
ASSIGN MASTER='dyn1.MASTER'
ASSIGN DBADD='morespace.DB'
EXPAND DBALL LOGICAL=(DBADD(50000))
```

These statements create an additional member (with a logical name of DBADD) to the existing DBset DBALL. This member points to a new physical file called morespace.DB, which may contain up to a maximum of 50000 Simcenter Nastran blocks. In this case, you are restarting from "dyn1.MASTER".

## RESTART

### Purpose

Allows you to continue from the end of the previous run to the current run without resolving the problem from the beginning.

### Format

```
RESTART [PROJECT='proj-ID', VERSION={version-ID, LAST},
{KEEP, NOKEEP}]
```

proj-ID	Project identifier used in the original run, which can have up to 40 characters. This is optional and is normally not used. The default proj-ID is blank.
version-ID	The version number you are restarting from. The default is the last version.
KEEP	If this option is used, then the version that you are restarting from is also saved at the end of the current run.
NOKEEP	If this option is used, then the version that you are restarting from is deleted at the end of the current run. This is the default.

### Example

```
RESTART
```

The current run uses the last version in the database for restart. At the end of the run, this last version is deleted from the database. This statement is probably the most commonly used format for RESTART.

```
RESTART VERSION=5, KEEP
```

The current run (version 6 or higher) uses version 5 in the database for restart. At the end of the run, version 5 is also retained in the database. This format is used most often when you want to ensure that a specific version is saved in the database (i.e., a large normal modes run).

```
RESTART PROJ='xyz' VERSION=3
```

The current run uses version 3 with a proj-ID of xyz in the database for restart. At the end of the run, version 3 with a proj-ID of xyz is deleted from the database.

## INCLUDE

### Purpose

Inserts an external file at the location where this “include” statement is used. This is not a pure FMS statement because it can be used anywhere in the input file, not just in the FMS Section. The “include” statement must not be nested; in other words, you cannot attach a file that contains an “include” statement.

### Format

```
Include 'filename'
```

filename            Physical filename of the external file to be inserted at this location.

### Example

```
Sol 101
time 10
cend
include 'sub1.dat'
begin bulk
$
include 'bulk1.dat'
include 'bulk2.dat'
$
$           rest of bulk data file
$
           .
           .
enddata
```

This run reads a file called sub1.dat with all the Case Control commands contained in it. It also reads two additional files (bulk1.dat and bulk2.dat) in the Bulk Data Section. You may, for example, want to include all your grid Bulk Data entries in the file bulk1.dat and all your element connectivities in bulk2.dat. As you can see, the “include” statement can be a handy tool. For parametric studies, you can

potentially save a tremendous amount of disk space by using the “include” statement instead of having multiple files with duplicate input data.

## Summary

Due to the default values, very little knowledge of the Simcenter Nastran FMS statements and database structure is required for small to medium size problems. For large problems, however, some knowledge of the FMS statements and database structure can help you to optimize your computer resources.



# I. Numerical Accuracy Considerations

## I.1 Overview of Numerical Accuracy Considerations

Simcenter Nastran is an advanced finite element analysis program. Because speed and accuracy are essential, the numerical analysis capabilities are continually enhanced to provide the highest level of each.

This appendix provides a brief overview for detecting and avoiding numerical ill-conditioning problems, especially as they relate to dynamic analysis. For more information regarding numerical analysis algorithms, see the Simcenter Nastran Numerical Methods User's Guide.

## I.2 Linear Equation Solution

The basic statement of the linear equation solution is

$$[A]\{x\} = \{b\}$$

Equation I-1.

where  $[A]$  is a square matrix of known coefficients (and usually symmetric for structural models),  $\{b\}$  is a known vector, and  $\{x\}$  is the unknown vector to be determined.

The methods used for solution in Simcenter Nastran are based on a decomposition of  $[A]$  to triangular matrices and are followed by forward-backward substitution to get  $\{x\}$ . The equations for this solution technique are

$$[A] = [L][U]$$

Equation I-2.

where  $[L]$  is a lower-triangular matrix and  $[U]$  an upper-triangular matrix, and

$$[L]\{y\} = \{b\}$$

Equation I-3.

where  $\{y\}$  is an intermediate vector. **Equation I-3** is called the forward pass because the solution starts with the first row where there is only one unknown due to the triangular form of  $[L]$ . The backward pass starts with the last row and provides the solution

$$[U]\{x\} = \{y\}$$

Equation I-4.

### I.3 Eigenvalue Analysis

The general eigensolution equation is

$$[K + pB + p^2M]\{u\} = 0$$

Equation I-5.

where  $p$  is the complex eigenvalue. This equation can always be transformed to a special eigenvalue problem for a matrix  $[A]$

$$[A - \lambda I]\{\phi\} = 0$$

Equation I-6.

where  $[I]$  is the identity matrix. **Equation I-6** is the basis of all the transformation methods of Simcenter Nastran (HOU, etc.). The iterative method (SINV) work directly from **Equation I-5**. The Lanczos method uses both. If  $[A]$  is a symmetric matrix, the eigenvectors are orthogonal, and they can be normalized such that

$$[\phi^T][\phi] = [I]$$

Equation I-7.

where  $[\phi]$  is a square matrix whose columns contain the eigenvectors  $\{\phi_i\}$ . With this normalization convention, then

$$[\phi^T][A][\phi] = [\lambda]$$

Equation I-8.

and

$$[A] = [\Phi][\lambda][\Phi^T]$$

Equation I-9.

where  $[\lambda]$  is the eigenvalue diagonal matrix.

## I.4 Matrix Conditioning

Reordering the previous equations, any matrix  $[I]$  can be expressed as a sum of its eigenvalues multiplied by dyadic eigenvector products

$$[A] = \lambda_1 \{\phi_1\} \{\phi_1\}^T + \lambda_2 \{\phi_2\} \{\phi_2\}^T + \dots + \lambda_n \{\phi_n\} \left\{ \phi_n^T \right\}$$

Equation I-10.

Defining  $[B_i] = \{\phi_i\} \left\{ \phi_i^T \right\}$ , which is a full rank 1 matrix, then

$$[A] = \sum_{k=1}^n \lambda_k [B_k]$$

Equation I-11.

where  $n$  is the dimension of  $[A]$ . On the average, an element of  $[B_k]$  has the same magnitude as the corresponding element of  $[B_{k+1}]$ . Let  $B_{max}$  be the magnitude of the largest coefficient of all  $[B_k]$  matrices. Then

$$|a_{ij}| \leq B_{max} (\lambda_1 + \lambda_2 + \dots + \lambda_n)$$

Equation I-12.

This equation shows that the terms of  $[A]$  are dominated by the largest eigenvalues. Unfortunately, the smallest eigenvalues are those of greatest interest for structural models. These small eigenvalues must be calculated by taking the differences of coefficients that are dominated by the largest eigenvalue. For this reason, the ratio  $\lambda_n/\lambda_1$  is called a numerical conditioning number. If this number is too large, numerical truncation causes a loss of accuracy when computing the lowest eigenvalues of a system.

The assumptions that allow this simple analysis are often pessimistic in practice, that is, the bounds predicted by the error analysis are conservative. However, the effects it predicts do occur eventually so that models that produce acceptable results for one mesh size may produce unacceptable results with a finer mesh size due to the higher eigenvalues included in the larger-sized matrices occurring from the finer mesh.

## I.5 Definiteness of Matrices

A matrix whose eigenvalues are all greater than zero is said to be positive definite. If some eigenvalues are zero but none are less than zero, the matrix is positive semi-definite. A stiffness matrix assembled from elements is at least positive semi-definite. If all of the structure's rigid-body modes are constrained, the stiffness matrix is positive definite. Another category is the indefinite matrices category. These matrices have zeroes or blocks of zeroes on the diagonal.

Although definiteness is most concisely defined in terms of eigenvalues, it is not a practical test for large matrices because of the computational cost of extracting all of the eigenvalues. However, other operations, particularly linear equation solution and dynamic reduction, may detect nonpositive definite matrices and provide diagnostics using these terms as described later in this appendix.

## I.6 Numerical Accuracy Issues

The numerical operations of Simcenter Nastran are executed in a finite 64-bit floating point arithmetic. Depending on the specific computer's word structure (number of bits for mantissa versus exponent), different roundoff errors may occur. The sophistication level of the actual hardware (or software) arithmetic units also has an influence on the numerical accuracy. To attain the most numerical accuracy possible, the following strategies are used in Simcenter Nastran.

In the decomposition of positive definite matrices, the Gaussian elimination process (which is the basis of all decomposition algorithms in Simcenter Nastran) does not require numerical pivots. Since some of the matrices are not positive definite, sparse decomposition (both symmetric and unsymmetric) employs a numerical pivoting (row/column interchange) algorithm. These methods consider a pivot term suitable if

$$|a_{kk}| \geq 10^{\text{THRESH}} \cdot \text{MAX}_i |a_{ki}|$$

Equation I-13.

which means that a diagonal term is accepted as a pivot if it is greater than the maximum term in that row multiplied by  $10^{\text{THRESH}}$  (the default for THRESH is  $-6$ ).

To ensure numerical accuracy in eigenvalue calculations, most Simcenter Nastran methods use a spectral transformation of



$$\mu = \frac{1}{\lambda - \lambda_s}$$

Equation I-14.

where  $\lambda_s$  is an eigenvalue shift.

This transformation is shown in **Figure I-1**.

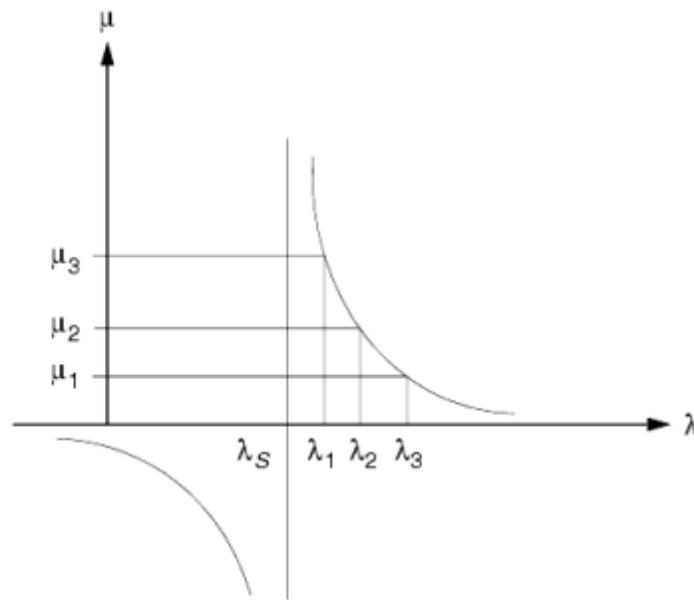


Figure I-1. Spectral Transformation

The spectral transformation ensures uniform accuracy throughout the frequency by “shifting” to the area of interest. Another effect of this transformation is the welcomed  $\mu$  space separation of closely-spaced eigenvalues. When  $\lambda_s$  is close to an eigenvalue  $\lambda_i$ , the decomposition of the “shifted matrix”

$$[K - \lambda_s M] = [L][D][L^T]$$

Equation I-15.

may produce high MAXRATIO messages. Automatic logic to perturb the  $\lambda_s$  value in this case is implemented in Simcenter Nastran.

## I.7 Sources of Mechanisms

In all of the decomposition methods, a null row or column in  $[A]$  causes a fatal error message. All other causes of singularity are not distinguishable from near-singularity because of the effects of numerical truncation. Only warning messages are usually provided for these cases.

In standard decomposition

$$[A] = [L][D][L^T]$$

Equation I-16.

the process starts to compute the first term of  $[D]$  with the first internal degree of freedom and then processes each additional degree of freedom and its associated terms. It can be shown that when processing the  $k$ -th row, the  $k$ -th row and all rows above it are in effect free, and all rows below it are constrained to ground. The term of  $[D]$  at the  $k$ -th row is proportional to the amount of coupling between that degree of freedom and the degree-of-freedom with a higher value of  $k$ . If the terms of the  $k$ -th row and above are not connected to the remaining rows, the  $k$ -th term of  $[D]$  goes to zero. Because of numerical truncation, the term may be a small positive or negative number instead. If a term of  $[D]$  is calculated to be identically zero, it is reset to a small number because of the indeterminacy of its calculation. The existence of such a small term defines a mechanism in static analysis in Simcenter Nastran. A mechanism is a group of degrees-of-freedom that may move independently from the rest of the structure as a rigid body without causing internal loads in the structure. A hinged door, for example, is a mechanism with one rigid-body freedom. If the hinges are disconnected, the door mechanism has six rigid-body freedoms.

Mechanisms are characterized by nondimensional numbers derived by dividing the terms of  $[D]$  into the corresponding diagonal term of  $[A]$ . If these “matrix diagonal to factor diagonal” ratios are large numbers, various warning and fatal messages are produced, depending on the context.

## I.8 Sources of Nonpositive Definite Matrices

A negative semidefinite element stiffness matrix, which is defined as one whose eigenvalues are all negative or zero, implies that the element has an energy source within it. Simcenter Nastran discourages using such elements by giving fatal messages when negative element thicknesses or section properties are input. However, there are applications where such elements are useful, such as when using an element with negative stiffness in parallel with a passive element to model a damaged or thinned-down element. For this reason, negative material property coefficients and negative stiffnesses for scalar elements are allowed. Also, some incorrect modeling techniques, such as the misuse of the membrane-bending coupling term on PSHELL entries (MID4), can lead to negative eigenvalues of the element stiffness matrix.

Stiffness matrices with negative eigenvalues cause negative terms in  $[D]$ . The number of such terms is automatically output by the standard symmetric decomposition subroutine. Their existence again causes various warning and fatal messages, depending on the context. The most common cause of negative

terms in  $[D]$  is true mechanisms, whose terms are small negative numbers and are actually computational zeroes.

Indefinite matrices occur when using the Lagrange multiplier method for constraint processing. In this method, the constraints are not eliminated from but are concatenated to the  $g$ -size system of equations. The constraint equations have zeroes on the diagonal, resulting in indefinite matrices. These systems can be solved using the block pivoting scheme of the sparse decomposition.

## 1.9 Detection and Avoidance of Numerical Problems

### Static Analysis

Models used in static analysis must be constrained to ground in at least a statically determinate manner even for unloaded directions. For example, a model intended for only gravity loading must be constrained in horizontal directions as well as vertical directions. The evidence of unconstrained directions is that the entire model is a mechanism, that is, the large ratio occurs at the last grid point in the internal sequence.

Another source of high ratios arises from connecting soft elements to stiff elements. Local stiffness is a function of element thickness (moment per unit rotation) through element thickness cubed (force per unit deflection) and is inversely proportional to mesh spacing, again in linear through cubic ratios. Some relief is possible by sequencing the soft degrees-of-freedom first in the internal sequence, although this is difficult to control in the presence of automatic resequencing. More reliable corrections are to replace the very stiff elements with rigid elements or to place the soft and stiff elements in different superelements.

A third source of high ratios is the elements omitted through oversight. The corrective action here is to start with the grid points listed in the diagnostics and track back through the elements connected to them through the upstream grid points. The missing elements may be anywhere upstream. PARAM,GPECT,1 output and undeformed structure plots all provide useful data for detecting missing elements.

At present, there are two major methods of identifying large ratios and nonpositive-definite matrices. In some solutions, the largest matrix diagonal to factor diagonal ratio greater than 105 (MAXRATIO default) is identified by its internal sequence number, and the number of negative factor diagonal terms is output. The best method to identify mechanisms here is to apply checkout loads that cause internal loads in all of the elements. Then inspect the displacement output for groups of grid points that move together with implausibly large displacements and common values of grid point rotation. The only condition that causes a fatal error is a true null column, and NASTRAN SYSTEM(69)=16 avoids this fatal error by placing a unit spring coefficient on the degrees-of-freedom with null columns. This option is recommended only for diagnostic runs because it may mask modeling errors.

In other solution sequences, all ratios greater than  $10^5$  are printed in a matrix format named the MECH table. The external sequence number of each large ratio is also printed, which is the grid point and degree-of-freedom number. If any such ratios exist, the action taken depends on the value of PARAM,BAILOUT. In the conventional solution sequences, its default value causes the program to continue after printing the MECH matrix. In the superelement solution sequences, a different default

causes a fatal error exit after printing the MECH matrix. For both types of solution sequences, the opposite action may be requested by setting the value of PARAM,BAILOUT explicitly. Also, the criterion used for identifying large ratios may be changed using PARAM,MAXRATIO.

For static analysis, values between  $10^3$  and  $10^6$  are almost always acceptable. Values between  $10^7$  and  $10^8$  are questionable. When investigating structures after finding these values, some types of structures may be found to be properly modeled. It is still worthwhile to investigate the structures with questionable values.

The solutions with differential stiffness effects offer another method to obtain nonpositive definite stiffness matrices. For example, a column undergoing compressive gravity loading has a potential energy source in the gravity load. A lateral load that is stabilizing in the absence of gravity (i.e., a decreased load causes a decreased deflection) is destabilizing when applied in a postbuckled state.

## Eigensolutions Using the Inverse Iteration and Lanczos Methods

The matrix  $[K - \lambda_s M]$  is decomposed where  $\lambda_s$  is an eigenvalue shift and  $[K]$  and  $[M]$  are the stiffness and mass matrices, respectively. This condition allows the solution of models with rigid-body modes since the singularities in  $[K]$  are suppressed if there are compensating terms in  $[M]$ . The only conditions that should cause fatal messages due to singularity are

1. The same null column in both  $[K]$  and  $[M]$ . These columns and rows are given an uncoupled unit stiffness by the auto-omit operation if the default value (0) for PARAM,ASING is used. If this value is set to  $-1$ , a null column in both matrices is regarded as an undefined degree of freedom and causes a fatal error exit.
2. A massless mechanism. One commonly encountered example is a straight-line, pinned rod structure made from CBAR elements with no torsional stiffness (J) defined. The structural mass generated for CBAR elements does not include inertia for the torsion degrees-of-freedom. The natural frequency for this torsion mode approaches the limit of zero torsional stiffness divided by zero mass, which is an undefined quantity. If the elements lie along a global coordinate axis, the mass term is identically zero, which leads to very large negative or positive eigenvalues and is usually beyond any reasonable search region. If the elements are skewed from the global axes, the eigenvalues may be computed at any value (including negative) because of the indeterminacy caused by numerical truncation.

The “negative terms on factor diagonal” message generally occurs for every decomposition performed in the iteration. It can be shown from Sturm sequence theory that the number of negative terms is exactly equal to the number of eigenvalues below  $\lambda_s$ . This condition is a means of determining if all roots in the range have truly been found.

## Eigensolutions Using Transformation Methods

If the HOU method (and variations) are selected, one of the first operations performed is a Cholesky decomposition, which is used to reduce the problem to a special eigenvalue problem. The mass matrix is given this operation for the straight HOU method. Columns that are identically null are eliminated by the auto-omit operation. Poor conditioning can result from several sources. One example is a point mass

input on a CONM2 entry that uses offsets in three directions. The grid point to which it is attached has nonzero mass coefficients for all six degrees-of-freedom. However, only three independent mass degrees-of-freedom exist, not six. Another example results in a superelement analysis when most of the elements in a superelement do not have mass and all interior masses are restricted to only a few structural or mass elements. The boundary matrix produced for the superelement is generally full, no matter what its rank, i.e., regardless of the number of independent mass degrees-of-freedom that it contains.

The presence of a rank-deficient mass matrix when using the HOU method produces fatal messages due to the singularity of the mass matrix or produces solutions with poor numerical stability. Poor stability is most commonly detected when making small changes to a model and then observing large changes in the solution. Either the MHOU method or the AHOU method is the preferred methods to use when nearly singular mass matrices are expected because these methods decompose the matrix  $[K - \lambda_s M]$  instead of the mass matrix. The shift parameter  $\lambda_s$  is automatically set to be large enough to control rigid-body modes. Better modeling practices also reduce the costs and increase the reliability for the two examples cited above. If an offset point mass is significant in any mode, it is better to attach it to an extra grid point at its center of gravity and model the offset with a rigid element. The auto-omit feature then eliminates the rotational degrees-of-freedom. Similarly, if only a few interior points of a superelement have mass, it may be more economical to convert them to exterior points, which also eliminates the singular boundary mass matrix.

It is possible to input negative terms into a mass matrix directly with DMIG terms or scalar mass elements. This class of problem causes fatal errors due to nonpositive definite mass matrices for the transformation methods, fatal errors with the Lanczos method. The complex eigenvalue methods should be used for this type of problem, which infers that modal or dynamic reduction methods may not be used.

A similar but quite different problem arises because Cholesky decomposition is used on the generalized mass matrix (named MI in the diagnostics) when orthogonalizing the eigenvectors with respect to the mass matrix. The existence of negative terms here indicates that poor eigenvectors were computed. The row number where the negative term occurs is printed in the diagnostic. This row number does not refer to a physical degree of freedom but refers instead to an eigenvector number. The usual cause is computing eigenvectors for computational infinite roots in the modified transformation methods. This problem can be avoided after the fact by setting ND on the EIGR entry to a value less than the row number that appears in the diagnostics or before the fact by setting F2 on the EIGR entry to a realistic value instead.

## Frequency Response Analysis

Negative terms on the factor and high factor to matrix diagonal ratios can be expected when using coupled methods and may often be safely ignored. These messages are merely an indication that the excitation frequency is above one or some of the natural frequencies of the system (negative terms), or is near a natural frequency (high ratio).

## Transient Response Analysis

A well-conditioned model should have neither negative terms nor high ratios on its factor terms. The causes of such messages include all of the effects described above. Negative mass terms can be detected by rapidly diverging oscillations in the transient solution.

### The Large Mass Method

One of the methods for enforcing motion is the large mass method. It may be used with direct methods as well as all of the reduction methods. The basis of the method is to attach artificial masses to the structure at the degrees-of-freedom where the motion is to be enforced. These large masses should be orders of magnitude larger than the total mass or the moment of inertia of the structure. A survey of the literature shows recommendations for mass ratios ranging from  $10^3$  to  $10^8$  with a value of  $10^6$  as the most common recommendation.

The mass ratio affects both the accuracy and numerical conditioning, and must be adjusted in a compromise that meets both criteria. With regard to load accuracy, the error in the approximation is inversely proportional to the ratio. A ratio of  $10^3$  causes an error of ten percent at resonance for a mode with one-half percent of damping, which represents an extreme case. Off-resonance excitation or higher damping ratios result in lower errors. Numerical conditioning problems are much more difficult to predict.

### The Lagrange Multiplier Method

Another method to enforce motion is the Lagrange multiplier method. In this method, the input motion function is described by a constraint equation. This method provides better accuracy than the large mass method where the numerical error introduced is proportional to the large mass.

The formulation introduces an indefinite system of linear equations where some numerical problems may arise from the fact that the system matrix contains terms that have dimensions of stiffness as well as nondimensional (constraint) terms. The decomposition of this matrix with the sparse decomposition methods pivoting strategy is stable.

You may control the pivoting strategy with the THRESH DMAP parameter mentioned earlier. The default value ( $-6$ ) is adequate for most Lagrange multiplier solutions. In fact, higher (ranging up to  $-2$ ) values provide better accuracy, while increasing the number of pivots may result in performance degradation.

# J. Grid Point Weight Generator

## J.1 Overview of the Grid Point Weight Generator

The grid point weight generator (GPWG) calculates the masses, centers of gravity, and inertias of the mathematical model of the structure. The data are extracted from the mass matrix by using a rigid-body transformation calculation. Computing the mass properties is somewhat complex because a finite element model may have directional mass properties, that is, the mass may differ in each of the three coordinate directions. From a mathematical point of view, the Simcenter Nastran mass may have tensor properties similar to the inertia tensor. This complexity is reflected in the GPWG output. All of the transformations used in calculating the mass properties are shown for the general case. Since most models have the same mass in each of the three coordinate directions, the GPWG output provides more information than you generally need.

To avoid unnecessary confusion and at the same time provide the necessary information for the advanced user, the discussion of the GPWG is separated into two sections. In **Commonly Used Features**, a basic discussion is given that should satisfy most users. If you need additional information, read **Example with Direction Dependent Masses**.

In both sections, a simple model consisting of four concentrated masses is used to demonstrate the GPWG output. In the first section, the mass is the same in each direction. For the second section, the mass is different in each of the three directions.

## J.2 Commonly Used Features

To demonstrate the typical output generated by the GPWG, a small model consisting of four concentrated masses as shown in **Figure J-1** is used. This model is typical of most models because the mass is the same in each coordinate direction. The number of masses has been kept small so you can better understand the physics.

Concentrated masses are located at four different grid points. The displacement coordinate system for each of the grid points is the basic coordinate system.

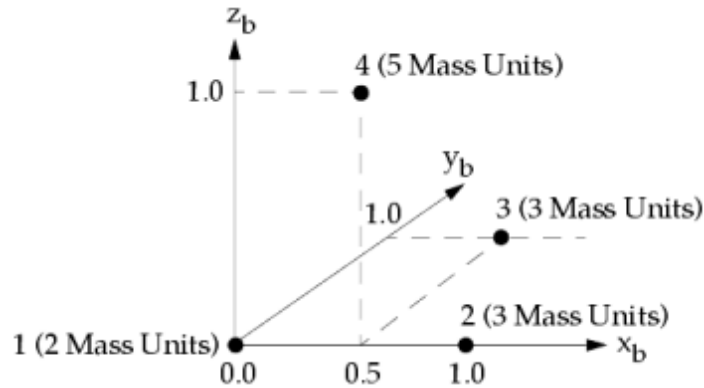


Figure J-1. Four Concentrated Mass Model

To request the GPWG output, you must add parameter GRDPNT in either the Bulk Data Section or the Case Control Section as follows:

```
PARAM, GRDPNT, x
```

If

- x = -1      GPWG is skipped (default).
- x = 0      The mass properties are computed relative to the origin of the basic coordinate system.
- x > 0      The mass properties are computed relative to grid point x. If grid point x does not exist, the properties are computed relative to the basic coordinate system.

For the four masses shown in **Figure J-1**, the resulting GPWG output is given in **Figure J-2**.

```

O U T P U T F R O M G R I D P O I N T W E I G H T G E N E R A T O R
R E F E R E N C E P O I N T = 0
M O
* 1.300000E+01 0.000000E+00 0.000000E+00 0.000000E+00 5.000000E+00 -3.000000E+00 *
* 0.000000E+00 1.300000E+01 0.000000E+00 -5.000000E+00 0.000000E+00 7.000000E+00 *
* 0.000000E+00 0.000000E+00 1.300000E+01 3.000000E+00 -7.000000E+00 0.000000E+00 *
* 0.000000E+00 -5.000000E+00 3.000000E+00 8.000000E+00 -1.500000E+00 -2.500000E+00 *
* 5.000000E+00 0.000000E+00 -7.000000E+00 -1.500000E+00 1.000000E+01 0.000000E+00 *
* -3.000000E+00 7.000000E+00 0.000000E+00 -2.500000E+00 0.000000E+00 8.000000E+00 *
S
* 1.000000E+00 0.000000E+00 0.000000E+00 *
* 0.000000E+00 1.000000E+00 0.000000E+00 *
* 0.000000E+00 0.000000E+00 1.000000E+00 *
D I R E C T I O N
M A S S A X I S S Y S T E M ( S ) M A S S X - C . G . Y - C . G . Z - C . G .
X 1.300000E+01 0.000000E+00 2.307692E-01 3.846154E-01
Y 1.300000E+01 5.384616E-01 0.000000E+00 3.846154E-01
Z 1.300000E+01 5.384616E-01 2.307692E-01 0.000000E+00
I ( S )
* 5.384615E+00 -1.153847E-01 -1.923079E-01 *
* -1.153847E-01 4.307692E+00 -1.153846E+00 *
* -1.923079E-01 -1.153846E+00 3.538461E+00 *

```



```

I (Q)
* 5.503882E+00 *
* 5.023013E+00 *
* 2.703873E+00 *
Q
* 8.702303E-01 4.915230E-01 3.323378E-02 *
* 3.829170E-01 -7.173043E-01 5.821075E-01 *
* 3.099580E-01 -4.938418E-01 -8.124324E-01 *

```

**Figure J-2. GPWG Output for the Four Concentrated Mass Model**

The  $[MO]$  matrix represents the rigid-body mass properties of the structure and is generally not needed for model checkout. This matrix represents an intermediate step in computing the inertia properties of the structure. The  $[S]$  matrix should always be equal to the identity matrix when the mass is the same in each coordinate direction, which is the typical case. If this matrix is not the identity matrix, inspect the model for inconsistent masses.

Following the  $[S]$  matrix are the mass and center of gravity locations. These are the most commonly used information of the GPWG output. Because the mass may be different in the three translational directions, the mass is printed for every coordinate direction. For the same reason, the center of gravity location is given for each of the three translational masses. If the mass is the same in all directions, a unique center of gravity exists and is located at the x-component of the y (or z) mass, the y-component of the x (or z) mass, and the z-component of the x (or y) mass. If the mass is not the same in all three directions, it is likely due to the CONM1, CMASSi, or DMIG input. If the reference point is specified with PARAM,GRDPNT,0 and  $[S]$  is the identity matrix, then the center of gravity location is given in the basic coordinate system. For the example, the mass of the structure is 13.0 and the center of gravity location is (0.5384, 0.2307, 0.3846) in the basic coordinate system.

If a grid point ID is used for the reference point, and  $[S]$  is an identity matrix, then the center of gravity location is in a coordinate system parallel to the basic coordinate system with an origin located at the grid point.

If the  $[S]$  matrix is equal to the identity matrix, then the  $[I(S)]$  matrix represents the inertia matrix of structure for the center of gravity with respect to the basic coordinate system, the  $[I(Q)]$  matrix is the corresponding principal moments of inertia matrix, and  $[Q]$  represents the transformation from the principal directions to the basic coordinate system.

The following additional comments on the GPWG should be noted.

- The scale factor entered with parameter WTMASS is applied to the assembled element mass before the GPWG. The GPWG module, however, converts mass back to the original input units that existed prior to the scaling effect of the parameter WTMASS. (Note that the parameter WTMASS is not applied to M2GG or M2PP input, but the M2GG mass is assembled into the mass matrix prior to GPWG. Therefore, for GPWG output only, the M2GG mass is scaled by the same parameter as the element mass. M2GG input may be scaled independently using the CM2 parameter.)
- The GPWG is performed on the g-size mass matrix, which is the mass matrix prior to the processing of the rigid elements, MPCs, and SPCs.

- The mass at scalar points and fluid-related masses are not included in the GPWG calculation.
- The GPWG for a superelement does not include the mass from upstream superelements. Therefore, the GPWG for the residual structure includes only the mass on the residual points. The center of gravity location is also based on the mass of the current superelement only.
- If a large mass is used for enforced motion, the large mass dominates the structural mass. For model checkout, it is recommended to remove the large mass and constrain the driving point. A static analysis is a convenient way to generate a mass matrix and obtain output from the GPWG.
- The output from the GPWG is for information purposes only and is not used in the analysis.

### J.3 Example with Direction Dependent Masses

In the previous section, the mass was the same in each of the three coordinate directions thereby producing a unique center of gravity location. However, if scalar masses are used, the total mass may have different properties in each direction, and the center of gravity may not be a unique location. This effect is shown in the output by providing the direction and center of gravity for each of the three mass principal components.

When using directional mass, the axes about which the inertia matrix  $i[S]$  is calculated are referred to as the principal mass axes. The principal mass axes may not necessarily intersect. However, these axes provide uncoupled rotation and translation mass properties. If the structural model is constructed using only real masses, the three principal mass values printed out are equal, the center of gravity is unique, and the principal mass axes intersect at the center of gravity.

To demonstrate all of the features of the GPWG module, the four-mass sample problem discussed in the previous section is modified so that the mass is not equal in each of the three translational directions (see [Figure J-3](#)). Furthermore, different displacement coordinate systems are used for the grid points. The displacement coordinate system for grid point 1 is the local rectangular system 1, which is oriented at an angle of 45 degrees (about the  $z_b$  axis). The displacement coordinate system for grid point 3 is the local rectangular system 3, which is oriented at an angle of 60 degrees (about the  $z_b$  axis). The grid point locations and masses are summarized in [Table J-2](#).

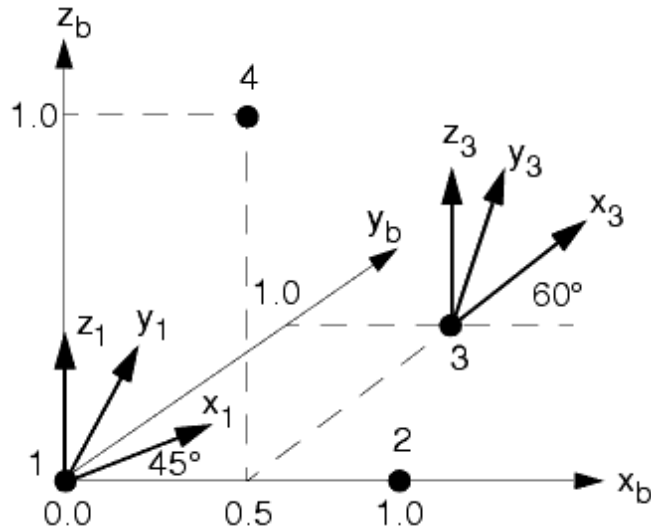


Figure J-3. Four Concentrated Mass Model

Table J-2. Location and Size of Masses

Grid ID	Location Basic System (CP Fields)			Mass Global System (CD Fields)		
	$x_b$	$y_b$	$z_b$	$x_{CD}$	$y_{CD}$	$z_{CD}$
1	0	0	0	2	3	5
2	1	0	0	2	3	5
3	0.5	1	0	2	3	5
4	0.5	0	1	2	3	5

The GPWG output for the four mass model is shown in J-3.

Table J-3. Output from the Grid Point Weight Generator

<pre> *** USER WARNING MESSAGE 3042 MODULE = GPWG INCONSISTENT SCALAR MASSES HAVE BEEN USED. EPSILON/DELTA = 3.8054429E-02  REFERENCE POINT = 1 M O * 9.250000E+00 -9.330128E-01 0.000000E+00 0.000000E+00 2.000000E+00 -2.966506E+00 * * -9.330128E-01 1.075000E+01 0.000000E+00 -3.000000E+00 0.000000E+00 6.058013E+00 * * 0.000000E+00 0.000000E+00 2.000000E+01 5.000000E+00 -1.000000E+01 0.000000E+00 * * 0.000000E+00 -3.000000E+00 5.000000E+00 8.000000E+00 -2.500000E+00 -1.500000E+00 * * 2.000000E+00 0.000000E+00 -1.000000E+01 -2.500000E+00 9.500000E+00 0.000000E+00 *                 </pre>	<p>Rigid-Body Mass Properties Matrix for the Reference Point</p>
--	--

* -2.966506E+00 6.058013E+00 0.000000E+00 -1.500000E+00 0.000000E+00 7.495513E+00 *	
S * 4.321332E-01 9.018098E-01 0.000000E+00 * * -9.018098E-01 4.321332E-01 0.000000E+00 * * 0.000000E+00 0.000000E+00 1.000000E+00 *	Transformation from the Principal Mass to the Basic Direction
DIRECTION MASS AXIS SYSTEM (S) MASS X-C.G. Y-C.G. Z-C.G. X 1.119709E+01 3.480388E-02 6.023980E-01 2.512494E-01 Y 8.802916E+00 -6.515555E-03 -4.426965E-02 2.484108E-01 Z 2.000000E+01 -9.385824E-03 5.589382E-01 0.000000E+00	Center of Gravity Relative to the Reference Point in the Principal Mass Axes System
I (S) * 4.376953E+00 -8.768300E-01 6.624477E-01 * * -8.768300E-01 5.623007E+00 -3.419880E-01 * * 6.624477E-01 -3.419880E-01 3.431904E+00 *	Moments of Inertia with Respect to Principal Mass Axes for the Center of Gravity
I (Q) * 4.463246E+00 * * 6.075616E+00 * * 2.893001E+00 *	Principal Moments of Inertia
Q * 7.201511E-01 4.586845E-01 5.205678E-01 * * -3.717336E-01 8.885992E-01 -2.687111E-01 * * -5.858298E-01 0.000000E+00 8.104341E-01 *	Transformation from the Principal Direction of the Momental Ellipsoid to the Principal Mass Axes

Before showing how each of the matrices are computed, a few items should be noted for this model:

- User Warning Message 3042 is printed to inform you that inconsistent scalar masses were used. This message occurs because there are different scalar masses in the three components. In general, if you are using structural mass and/or CONM2s, you should not get this message.
- The rigid-body mass matrix [MO] is computed with respect to the reference grid point1 in the basic coordinate system. Grid point 1 is used for this example because PARAM,GNDPNT,1 is entered in the Bulk Data Section.

- The mass and center of gravity location shown are not in the basic coordinate system for this example. The mass and center of gravity are computed and printed in the principal mass coordinate system. This principal mass coordinate system should not be confused with the principal axes discussed in most text books. The principal mass axes in Simcenter Nastran are the axes that have no coupling terms between the translational masses (diagonal translational mass matrix). Also, the Simcenter Nastran principal mass axes are not the axes of the inertia ellipsoid. The  $[S]$  matrix is the transformation from the principal mass direction to the basic coordinate system. This additional step may sound confusing, but it is necessary. In real structures, the mass of structure is generally the same in all directions, so there is no coupling between the translational mass terms. Since text books are written to solve real structural problems, there is no need to discuss the principal mass axes, as they are called in Simcenter Nastran. However, with Simcenter Nastran you are not restricted to the same mass in each coordinate direction—a situation that may not be physically realizable but still is quite useful for certain modeling situations. Therefore, the additional step of computing principal mass axes is necessary.
- If your model has the same mass in all coordinate directions, then the  $[S]$  matrix is the identity matrix indicating that the principal mass axes is the basic coordinate system. Always check the  $[S]$  matrix. If it is not the identity matrix, verify that the mass distribution is correct. Do not use the directional mass and center of gravity location blindly. Remember, these quantities are in the principal mass axes.

To fully understand how the GPWG module works, it is useful to trace the steps Simcenter Nastran follows to generate the output shown in the figure **GPWG Output for the Four Concentrated Mass Model**. The following shows the step-by-step procedure for the four mass example.

1. The GPWG module uses the global mass matrix, which is the mass matrix before any constraints are applied. In this example, there are four grids, each with six degrees-of-freedom, resulting in a total of 24 degrees-of-freedom in the mass  $[MJJ]$  matrix. The matrix is shown in **Figure J-4**.



(Reference Point)

The transformation matrix  $[D]$  is assembled from the individual transformation matrices  $[d_i]$  computed for each grid point. Each individual transformation matrix  $[d_i]$  consists of two transformations:  $[Tr]_i$ , which relates the location of the grid point to the reference grid point in the basic coordinate system, and  $[Ti]_i$ , which relates the global coordinate system at the grid point to the basic coordinate system.

The  $[Tr]_i$  transformation matrices are first computed by constructing the location vectors in the basic coordinate system  $r_i$  for each grid point in the model relative to the reference point as shown in **Equation J-2**.

$$\{r\}_i = \{R\}_i - \{R\}_0 = \begin{Bmatrix} r_1 \\ r_2 \\ r_3 \end{Bmatrix}$$

**Equation J-2.**

The location vectors for the example are as follows:

$$\{r\}_1 = \begin{Bmatrix} 0 \\ 0 \\ 0 \end{Bmatrix}; \{r\}_2 = \begin{Bmatrix} 1 \\ 0 \\ 0 \end{Bmatrix}; \{r\}_3 = \begin{Bmatrix} .5 \\ 1 \\ 0 \end{Bmatrix}; \{r\}_4 = \begin{Bmatrix} .5 \\ 0 \\ 1 \end{Bmatrix}$$

**Equation J-3.**

Using the location vectors, the grid point transformation matrix  $[Tr]_i$  is computed for each grid point by expanding the location vectors to a 3x3 matrix as shown in **Equation J-4**.

$$[Tr]_i = \begin{bmatrix} 0 & r_3 & -r_2 \\ -r_3 & 0 & r_1 \\ r_2 & -r_1 & 0 \end{bmatrix}$$

**Equation J-4.**

For the example problem, the grid point transformation matrices are

$$[Tr]_1 = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}; [Tr]_2 = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 1 \\ 0 & -1 & 0 \end{bmatrix}; [Tr]_3 = \begin{bmatrix} 0 & 0 & -1 \\ 0 & 0 & .5 \\ 1 & -.5 & 0 \end{bmatrix}; [Tr]_4 = \begin{bmatrix} 0 & 1 & 0 \\ -1 & 0 & .5 \\ 0 & -.5 & 0 \end{bmatrix}$$

Equation J-5.

The coordinate system transformation matrices from the global coordinate system to the basic system coordinates are given by the direction cosine matrices as follows:

$$[Ti]_1 = \begin{bmatrix} \cos(45^\circ) & -\sin(45^\circ) & 0 \\ \sin(45^\circ) & \cos(45^\circ) & 0 \\ 0 & 0 & 1 \end{bmatrix}; [Ti]_2 = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix};$$

$$[Ti]_3 = \begin{bmatrix} \cos(60^\circ) & -\sin(60^\circ) & 0 \\ \sin(60^\circ) & \cos(60^\circ) & 0 \\ 0 & 0 & 1 \end{bmatrix}; [Ti]_4 = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

Equation J-6.

The grid point transformation  $[Tr]_i$  and the coordinate system transformation  $[Ti]_i$  are combined to form the individual grid point transformation matrix  $[d]_i$  for each grid point using [Equation J-7](#).

$$[d]_i = \begin{bmatrix} Ti^T & Ti^T \cdot Tr \\ 0 & Ti^T \end{bmatrix}$$

Equation J-7.

The rows of each  $[d]_i$  form the columns of the global transformation matrix  $[D]^T$  as shown in [Equation J-8](#).



$$[D]^T = \left[ d_1^T \mid d_2^T \mid \dots \mid d_n^T \right]$$

Equation J-8.

Using Equation J-9, the global transformation matrix for the example is

$$[D]^T = \begin{bmatrix} .707 & -.707 & 0 & 0 & 0 & | & 0 & 1 & 0 & 0 & 0 & 0 & | & .5 & -.866 & 0 & 0 & 0 & 0 & | & 1 & 0 & 0 & 0 & 0 & 0 \\ .707 & .707 & 0 & 0 & 0 & | & 0 & 0 & 1 & 0 & 0 & 0 & | & .866 & .5 & 0 & 0 & 0 & 0 & | & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & | & 0 & 0 & 0 & 1 & 0 & 0 & | & 0 & 0 & 1 & 0 & 0 & 0 & | & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & .707 & -.707 & | & 0 & 0 & 0 & 0 & 1 & 0 & | & 0 & 0 & 1 & .5 & -.866 & 0 & | & 0 & -1 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & .707 & .707 & | & 0 & 0 & 0 & -1 & 0 & 1 & | & 0 & 0 & -.5 & .866 & .5 & 0 & | & 0 & 1 & 0 & -.5 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & | & 1 & 0 & 1 & 0 & 0 & 0 & | & 1 & -.067 & 1.116 & 0 & 0 & 0 & | & 0 & 1 & 0 & .5 & 0 & 0 & 0 & 1 \end{bmatrix}$$

Grid 1
Grid 2
Grid 3
Grid 4

Equation J-9.

Using the global transformation matrix  $[D]$ , the rigid-body mass matrix about the reference point in the basic coordinate system  $[M0]$  is obtained by Equation J-10.

$$[M0] = [D]^T [MJJ] [D]$$

Equation J-10.

For the example,  $[M0]$  is determined to be

$$[M0] = \begin{bmatrix} 9.25 & -.933 & 0. & 0. & 2. & -2.967 \\ -.933 & 10.75 & 0. & -3. & 0. & 6.058 \\ 0. & 0 & 20. & 5. & -10. & 0. \\ 0. & -3 & 5. & 8. & -2.5 & -1.5 \\ 2. & 0 & -10. & -2.5 & 9.5 & 0. \\ -2.967 & 6.058 & 0. & -1.5 & 0. & 7.496 \end{bmatrix}$$

Equation J-11.

Comparing the results shown in **Equation J-11** to  $[MO]$  generated by the GPWG module (in the figure **GPWG Output for the Four Concentrated Mass Model**) shows the matrices to be numerically the same.

- The next step is to inspect the  $[MO]$  to determine whether the basic coordinate system can be used as the principal mass directions. The principal mass axes are axes that have no coupling between the translational mass components. For real structures, there is no coupling in the translational mass terms in the inertia matrix. However, with inconsistent scalar masses (CONM1, CMASSi, or DMIG), you may define any type of mass matrix you desire. To determine whether coupling exists between the translational mass terms,  $[MO]$  is partitioned into four 3x3 matrices as shown in **Equation J-12**.

$$[MO] = \begin{bmatrix} \bar{M}^t & | & \bar{M}^{tr} \\ \hline \bar{M}^{rt} & | & \bar{M}^r \end{bmatrix}$$

Equation J-12.

where the superscripts  $t$  and  $r$  refer to translation and rotation, respectively. For this example, the translational mass partition is given by

$$[\bar{M}^t] = \begin{bmatrix} 9.25 & -.933 & 0 \\ -.933 & 10.75 & 0 \\ 0 & 0 & 20. \end{bmatrix}$$

Equation J-13.

A check is made for coupling as follows:

$$\delta = \sqrt{\sum (\bar{M}_{ij}^t)^2} \quad i = j$$

$$\epsilon = \sqrt{\sum (\bar{M}_{ij}^t)^2} \quad i > j$$

Equation J-14.

If  $\epsilon/\delta$  is greater than .001, then excessive coupling exists preventing the basic coordinate system from being used for the principal mass directions and User Warning Message 3042 is printed. For this problem,  $\delta = (9.25^2 + 10.75^2 + 20^2)^{.5}$  and  $\epsilon = (.933^2 + 0.^2 + 0.^2)^{.5}$ . The ratio  $\epsilon/\delta$

= .038 agrees with the System Warning Message 3042 shown in the figure **GPWG Output for the Four Concentrated Mass Model**.

If needed, the principal mass directions are computed by performing an eigensolution with the translational mass components. The eigenvectors of  $[\bar{M}^t]$  are the columns of the transformation matrix  $[S]$

$$[S] = [\{e_1\}, \{e_2\}, \{e_3\}]$$

Equation J-15.

Using this eigenvector matrix, the partitions of the rigid-body mass matrix with respect to the principal mass direction are computed by **Equation J-16**.

$$[M^t] = [S]^T [\bar{M}^t] [S]$$

$$[M^{tr}] = [S]^T [\bar{M}^{tr}] [S]$$

$$[M^r] = [S]^T [\bar{M}^r] [S]$$

Equation J-16.

The  $[S]$  matrix for the example is given by

$$[S] = \begin{bmatrix} .432 & .902 & 0. \\ -.902 & .432 & 0. \\ 0. & 0. & 1. \end{bmatrix}$$

Equation J-17.

The  $[S]$  matrix is printed after the  $[M0]$  matrix as shown in the figure **GPWG Output for the Four Concentrated Mass Model**. This represents the transformation relating the basic coordinate system to the principal mass axes. Again, if there is no coupling between the translational mass component, which is the case for most problems, the eigensolution is not required, and the  $[S]$  matrix is set equal to the identity matrix. This example was selected to demonstrate all of the features of the GPWG module, but it is not a typical problem.

4. The next step is to determine the principal masses and the center of gravity location in the principal mass axes system as shown in **Equations J-18**.

$$M_x = M_{11}^t$$

$$M_y = M_{22}^t$$

$$M_z = M_{33}^t$$

The “centers of gravity” are

$$X_x = \frac{M_{11}^{tr}}{M_x}, \quad Y_x = \frac{-M_{13}^{tr}}{M_x}, \quad Z_x = \frac{M_{12}^{tr}}{M_x},$$

$$X_y = \frac{M_{23}^{tr}}{M_y}, \quad Y_y = \frac{M_{22}^{tr}}{M_y}, \quad Z_y = \frac{-M_{21}^{tr}}{M_y},$$

$$X_z = \frac{-M_{32}^{tr}}{M_z}, \quad Y_z = \frac{M_{31}^{tr}}{M_z}, \quad Z_z = \frac{M_{33}^{tr}}{M_z}$$

**Equation J-18.**

As can be seen, the center of gravity location is not a unique location. The center of gravity location is computed separately for the for x-, y-, and z-directions relative to the principal mass axes. Only if the mass is the same in each direction (which is typical) is there a unique center of gravity location, which is relative to the reference point in the basic coordinate system.

For this example problem, the center of gravity locations are determined to be

Table J-4. Mass Center of Gravity Locations

Mass Component	Center of Gravity Location in Principal Mass Axes		
	X	Y	L
x - mass	.035	-0.602	0.251
y - mass	-0.007	-0.044	0.248
z - mass	-0.009	-0.559	0

The center of gravity location given in **Figure J-4** is the same as shown in the figure **GPWG Output for the Four Concentrated Mass Model**.

- Following the center of gravity calculation is the calculation to determine the moments of inertia for the center of gravity with respect to the principal mass axes  $[I(S)]$  as shown in **Equation J-19**.

$$I_{11}^{(S)} = M_{11}^r - M_y Z_y^2 - M_z Y_z^2$$

$$I_{12}^{(S)} = I_{21}^{(S)} = -M_{12}^r - M_z X_z Y_z$$

$$I_{13}^{(S)} = I_{31}^{(S)} = -M_{13}^r - M_y X_y Z_y$$

$$I_{22}^{(S)} = M_{22}^r - M_z X_z^2 - M_x Z_x^2$$

$$I_{32}^{(S)} = I_{23}^{(S)} = -M_{23}^r - M_x Y_x Z_x$$

$$I_{33}^{(S)} = M_{33}^r - M_x Y_x^2 - M_y X_y^2$$

**Equation J-19.**

For the example, the inertia matrix  $[I(S)]$  is given by

$$[I(S)] = \begin{bmatrix} 4.377 & -.877 & .662 \\ -.877 & 5.623 & -.342 \\ .662 & -.342 & 3.432 \end{bmatrix}$$

Equation J-20.

6. The final step is to compute the principal moments of inertia and the principal directions of the momental ellipsoid (commonly referred to as the principal axes in text books). An intermediate inertia matrix  $[\tilde{I}]$  is generated by reversing the sign on the off diagonal terms of  $[I(S)]$ . For the inertia matrix given in **Equation J-20**,  $[\tilde{I}]$  is given by

$$[\tilde{I}] = \begin{bmatrix} 4.377 & .877 & -.662 \\ .877 & 5.623 & .342 \\ -.662 & .342 & 3.432 \end{bmatrix}$$

Equation J-21.

An eigensolution is performed on the  $[I(Q)]$  matrix to determine the principal directions. The resulting eigenvalues are the principal moments of inertia, which are assumed to be the diagonal terms of the principal inertia matrix  $[I(Q)]$ . The eigenvectors form the columns of the matrix  $[Q]$ , which is the transformation relating the intermediate inertia matrix  $[\tilde{I}]$  to the principal inertia matrix  $[I(Q)]$  as shown in **Equation J-22**.

$$[I(Q)] = [Q]^T [\tilde{I}] [Q]$$

Equation J-22.

For the example,  $[Q]$  and  $[I(Q)]$  are given by

$$[I(Q)] = \begin{bmatrix} 4.463 & 0. & 0. \\ 0. & 6.076 & 0. \\ 0. & 0. & 2.893 \end{bmatrix}$$
$$[Q] = \begin{bmatrix} .720 & .459 & .521 \\ -.372 & .889 & -.269 \\ -.586 & 0 & .810 \end{bmatrix}$$

Equation J-23.

The matrices  $[S]$  and  $[Q]$  matrices are the coordinate rotation matrices, which when taken together, relate the principal directions of the momental ellipsoid to the basic coordinate system. The matrices given in **Equation J-23** are in agreement with those given in the figure **GPWG Output for the Four Concentrated Mass Model**. The example for the four mass model is now complete.





# K. Diagnostic Messages for Dynamic Analysis

This appendix lists common diagnostic messages for dynamic analysis. The text for each message is given in uppercase letters and is followed by additional explanatory material, including suggestions for remedial action..

The messages in this section have the following format:

\*\*\* (SYSTEM/USER) (FATAL/WARNING/INFORMATION) MESSAGE ID, text

where "ID" is a unique message identification number and "text" is the message as indicated in capital letters for each of the diagnostic messages. Four asterisks (\*\*\*\*) in the message text indicates information that is filled in for a specific use of the message, such as the number of a grid point or the name of a Bulk Data entry. Some of the messages are followed by additional explanatory material, including suggestions for remedial action.

Fatal messages cause the termination of the execution following the printing of the message text. These messages always appear at the end of the Simcenter Nastran output. Warning and information messages appear at various places in the output stream. Such messages only convey warnings or information to the user. Consequently, the execution continues in a normal manner following the printing of the message text.

As an example, consider message number 2025, which appears in the printed output as follows:

\*\*\* USER FATAL MESSAGE 2025, UNDEFINED COORDINATE SYSTEM 102

The three leading asterisks (\*\*\*) are always present in numbered user and system diagnostic messages. The word USER indicates that this is a user problem rather than a system problem. The word FATAL indicates that this is a fatal message rather than a warning or information message. The number 2025 is the identification number for this message. The text of the message follows the comma. The number 102 replaces the asterisks (\*\*\*\*) in the general message text and indicates that 102 is the identification number of the undefined coordinate system. The abbreviation UFM refers to User Fatal Message, UWM refers to User Warning Message, UIM refers to User Information Message, and SFM refers to System Fatal Message.

UFM 2066      \*\*\* USER FATAL MESSAGE 2066, UNDEFINED GRID POINT \*\*\*\* ON DAREA CARD.

A dynamic loading entry references an undefined grid point.

UFM 2069      \*\*\* USER FATAL MESSAGE 2069, UNDEFINED GRID POINT \*\*\*\* IN TRANSIENT INITIAL CONDITION SET \*\*\*\*.

An attempt has been made to specify initial conditions for an undefined grid point. All degrees-of-freedom with initial conditions must be in the analysis set.

- UFM 2071      \*\*\* USER FATAL MESSAGE 2071, DYNAMIC LOAD SET \*\*\*\* REFERENCES UNDEFINED TABLE \*\*\*\*.
- A referenced dynamic load table was not present in the Bulk Data.
- UFM 2074      \*\*\* USER FATAL MESSAGE 2074, UNDEFINED TRANSFER FUNCTION SET \*\*\*\*.
- A transfer function set was selected in the Case Control but was not present in the Bulk Data.
- UFM 2079      \*\*\* USER FATAL MESSAGE 2079, GRID OR SCALAR POINT \*\*\*\* HAS AN UNDEFINED COORDINATE REFERENCED ON A DAREA, DELAY, DPHASE CARD.
- The "C" or component value for scalar-type points must be zero or one.
- UFM 2088      \*\*\* USER FATAL MESSAGE 2088, DUPLICATE TABLE ID \*\*\*\*.
- All tables must have unique numbers. Check for uniqueness.
- UFM 2101A     \*\*\* USER FATAL MESSAGE 2101A, GRID POINT \*\*\*\* COMPONENT \*\*\* ILLEGALLY DEFINED IN SETS \*\*\*\*.
- The above grid point and component are defined in each of the above independent subsets. A point may belong to a maximum of one independent subset. This error occurs when a DOF is defined as belonging to two mutually exclusive sets (see *Simcenter Nastran User's Guide*).
- A common example of this occurs when a DOF is defined as dependent on an MPC (M-set) as well as being constrained (s-set) on an SPC entry. The message for this states that the component is illegally defined in the um (user-defined m-set) and us (user-defined s-set) sets. These two sets are mutually exclusive because all MPC equations are processed before the SPCs are applied and the m-set DOFs are removed from the matrix. When the program attempts to apply the SPC, the DOF is no longer available, and the fatal message is issued. The normal correction for this is to modify the MPC so that the DOF in question is independent (n-set). Then there is no conflict.
- UFM 2107      \*\*\* USER FATAL MESSAGE 2107, EIGR-CARD FROM SET \*\*\*\* REFERENCES DEPENDENT COORDINATE OR GRID POINT \*\*\*\*.
- When the point option is used on an EIGR entry, the referenced point and component must be in the analysis set (a-set) for use in normalization.
- UFM 2109      \*\*\* USER FATAL MESSAGE 2109, NO GRID, SCALAR OR EXTRA POINTS DEFINED.
- Dynamics problems must have at least one grid, scalar, or extra point.
- UFM 2133      \*\*\* USER FATAL MESSAGE 2133, INITIAL CONDITION IN SET \*\*\*\* SPECIFIED FOR POINT NOT IN ANALYSIS SET.
- Initial conditions can only be specified for analysis set points. Therefore the point/component mentioned on TIC entries must belong to the analysis set.
- UFM 2135      \*\*\* USER FATAL MESSAGE 2135, DLOAD CARD \*\*\* HAS A DUPLICATE SET ID FOR SET ID \*\*\*.



The Li Set IDs on a DLOAD entry are not unique. See the DLOAD Bulk Data description.

UFM 2136

\*\*\* USER FATAL MESSAGE 2136, DUPLICATE DLOAD, RLOAD, OR TLOAD SET ID NUMBER = \*\*\*\*\* HAS BEEN ENCOUNTERED FOR DLOAD SET = \*\*\*\*\*.

Dynamic loads may not be defined by giving multiple data entries with the same ID. Use unique IDs.

UIM 2141

\*\*\* USER INFORMATION MESSAGE 2141, TIME ESTIMATE IS \*\*\*\*\* SECONDS. PROBLEM SIZE IS \*\*\*\*\*, SPILL WILL OCCUR FOR THIS CORE AT A PROBLEM SIZE OF \*\*\*\*\*.

The time estimate includes the time of the tridiagonalization and eigenvalue calculation when the HOU method is used. If ND is given on the EIGR entry, it also includes the time of the eigenvector generation. If F1 and F2 are used instead, the eigenvector times are not estimated. This condition can underestimate the time when the range includes many eigenvectors.

UFM 2200

\*\*\* USER FATAL MESSAGE 2200, INCONSISTENT RIGID BODY SYSTEM.

This message occurs if a SUPORT is used and the rigid-body mass matrix is not positive definite. Possible causes are unconstrained mechanisms or input of negative mass terms. A diagnostic method is to remove all SUPORT entries and inspect the resulting eigenvectors for implausible behavior.

UFM 3031

\*\*\* USER FATAL MESSAGE 3031, UNABLE TO FIND SELECTED SET (\*\*\*\*) IN TABLE (\*\*\*\*) IN SUBROUTINE (\*\*\*\*).

A particular set used in the problem was not included in the data. Good examples are loads, initial conditions, or frequency sets. Include the required data or change the Case Control commands to select data already in the problem. Set zero (0) has a special meaning. A set selection was required, but none was made. For example, no METHOD was selected for an eigenvalue extraction problem, or no FREQ was selected for frequency response.

This message can also indicate that a LOAD entry references another LOAD entry, which is not permitted.

This message can also occur if a DLOAD entry references a nonexisting LOAD entry, e.g., RLOAD1.

UWM 3034

\*\*\* USER WARNING MESSAGE 3034, ORTHOGONALITY TEST FAILED. LARGEST TERM = \*\*\*\*\*, NUMBER FAILED = \*\*\*\*\*, PAIR = \*\*\*\*\*, \*\*\*\*\*, EPSILON = \*\*\*\*\*.

This indicates that eigenvector accuracy is in doubt. This message is printed only when the off-diagonal terms of the modal mass matrix are larger than 1.0E-10. The eigenvectors are not orthogonal to this extent. This nonorthogonality is especially important if a modal formulation is used. The pair of eigenvectors listed exhibit the worst behavior. The number failed is the number of pairs above the criteria. You can improve the numerical conditioning of the problem by reducing the range of mass ratios, stiffness ratios, and eigenvalue range.

- UIM 3035      \*\*\* USER INFORMATION MESSAGE 3035, FOR DATA BLOCKS \*\*\*\*. SUPORT PT. NO. EPSILON STRAIN ENERGY EPSILONS LARGER THAN .001 ARE FLAGGED WITH ASTERISKS.
- One line of output is printed for each component on a SUPORT entry. Large values of either EPSILON or STRAIN ENERGY indicate errors in the constraints, MPCs, or offsets.
- UWM 3045      \*\*\* USER WARNING MESSAGE 3045, INSUFFICIENT TIME TO COMPLETE THE REMAINING \*\* SOLUTION(S) IN MODULE \*\*\*.
- The estimated time for completion of the module is less than the time remaining, as specified on the Executive Control TIME statement. The module computes one solution (for example, one excitation frequency in frequency response analysis) and then processes all output requests. The remaining frequencies can be obtained on restart by adding or changing a FREQ command.
- UFM 3046      \*\*\* USER FATAL MESSAGE 3046, YOUR SELECTED LOADING CONDITION, INITIAL CONDITION, AND NONLINEAR FORCES ARE NULL. A ZERO SOLUTION WILL RESULT.
- Transient solutions must have one of the above nonzero loading condition, initial condition, or nonlinear forces. Also, make sure that LOADSET is spelled correctly.
- UFM 3047      \*\*\* USER FATAL MESSAGE 3047, NO MODES WITHIN RANGE AND LMODES=0. A MODAL FORMULATION CANNOT BE MADE.
- The modes used for a modal formulation must be selected by a PARAM entry. Set LFREQ, HFREQ, or LMODES to request modes.
- UWM 3053      \*\*\* USER WARNING MESSAGE 3053, THE ACCURACY OF EIGENVALUE \*\*\*\* IS IN DOUBT. HOU QR FAILED TO CONVERGE IN \*\*\*\* ITERATIONS.
- Each eigenvalue is computed to the precision limits of each machine consistent with the maximum number of iterations allowed.
- UFM 3057      \*\*\* USER FATAL MESSAGE 3057, MATRIX \*\*\*\* IS NOT POSITIVE DEFINITE.
- A Cholesky decomposition was attempted on the above matrix, but a diagonal term of the factor was imaginary or equal to zero such that the decomposition failed. This message is from the regular (as opposed to sparse) decomposition method.
- This message may be produced because of constraint problems. Check the output for UWM 4698 for large factor diagonal ratios and constrain appropriately.
- UWM 4193      \*\*\* USER WARNING MESSAGE 4193, A GRID AND COMPONENT SPECIFICATION ON A (DPHASE/DELAY) SID = \*\*, DOES NOT APPEAR ON A DAREA CARD.
- The area specification is set to zero.
- SFM 4276      \*\*\* SYSTEM FATAL MESSAGE 4276, \*\*\*\* ERROR CODE \*\*\*\*
- This message occurs when Simcenter Nastran encounters errors that are not otherwise trapped (including system errors). There are various error codes (EC), each of which has a different meaning.
- In nearly every case, the log file contains further information about the problem, so you should look there for further clues. In addition, because these are errors that

most likely should be caught in another manner (and with a more explicit error message), you should look at the recent Error Reports (search for "4276").

In many cases, increasing memory, BUFFSIZE, or disk space resolves the problem, especially when attempting to run large models on workstations.

This error often indicates machine underflow or overflow, although it is impossible to list specific reasons for this error code. Observed problems include:

- Modeling problems. This condition is usually accompanied by an "arithmetic fault, floating overflow" type message in the log file on some machines. Perform a static analysis and verify that the value for the maximum factor diagonal ratio is acceptable.
- When SFM 4276 is followed by an "access violation", then it is often due to a lack of memory or disk space or due to a coding error in Simcenter Nastran.

UWM 4312

\*\*\* USER WARNING MESSAGE 4312, CONM2 \*\*\*\* HAS NONPOSITIVE-DEFINITE INERTIA MATRIX.

Most dynamic analysis methods require positive-definite mass matrices. Inserting inertia data into the wrong fields can result in nonpositive-definite systems, which are not physically realistic for normal modeling practices. Reduce the size of the off-diagonal terms to provide a positive-definite determinant.

UFM 4346

\*\*\* USER FATAL MESSAGE 4346, FREQUENCY RESPONSE SET, ID = \*\*\* IS UNDEFINED.

Define the set of frequencies to be used for the analysis.

UFM 4391

\*\*\* USER FATAL MESSAGE 4391, NONUNIQUE DAREA SET \*\*\*\*\* HAS BEEN SPECIFIED FOR LSEQ DEFINED VECTOR \*\*\*\*\*.

Each LSEQ Bulk Data entry must define a unique DAREA set specification.

UFM 4392

\*\*\* USER FATAL MESSAGE 4392, CONTINUATION CARD ERRORS. EXPLANATIONS FOLLOW LIST OF CARDS IN ERROR.

ERROR NUMBER .1..2..3... \*\*\* Input echo

EXPLANATION OF ERROR CODES ABOVE FOLLOWS...

1. FIELD 1 IS NOT UNIQUE.
2. MORE THAN ONE CARD HAS FIELD 10 WHICH IS THE SAME AS FIELD 1 OF THIS CARD.
3. CARD IS AN ORPHAN (I.E., NO PARENT CARD EXISTS).

Continuation mnemonics in field 10 of a parent entry and field 1 of its continuation entry must be unique. Each continuation entry must have a parent entry. Check all continuation mnemonics.

- UFM 4405      \*\*\* USER FATAL MESSAGE 4405, NO EIGENVECTORS COMPUTED FOR COMPONENT MODE SYNTHESIS OR SYSTEM SOLUTION.
- The eigenvectors computed in component mode synthesis (CMS) are used to approximate the motion of the component. Some vectors must be present to perform this reduction. For the system solution, an exit is taken if eigenvalues are requested but not eigenvectors. Reset the "range of frequency" and/or the "number desired" on the EIGR or EIGRL entry. If the superelement does not have any eigenvalues in the range of interest, remove the CMS request for that particular superelement.
- This message is also issued when the eigenvectors calculated with a transformation method do not pass internal orthogonality checks. This is indicative of a modeling error.
- This message can also be issued if insufficient memory is available for the Lanczos method with sparse decomposition.
- This could occur with UFM 5401 and be related to UWM 5411.
- UFM 4407      \*\*\* USER FATAL MESSAGE 4407, MR MATRIX HAS NULL DIAGONAL TERM.
- The MR matrix contains the rigid-body mass matrix of the structure as measured at the degrees-of-freedom listed on the SUPORT entry. If any of these degrees-of-freedom have null mass, they result in invalid eigenvectors. Specify enough masses to define all rigid-body modes and check attachments between the SUPORT and the rest of the model.
- UIM 4415      \*\*\* USER INFORMATION MESSAGE 4415, THE FOLLOWING A-SET DEGREES-OF-FREEDOM HAVE EITHER NULL MASSES OR NULL MASSES AND STIFFNESSES.
- If the listed degrees-of-freedom have null mass for the HOU or MHOUE methods, they are automatically omitted. For direct frequency or direct transient response, the null degrees-of-freedom are given a very small mass or stiffness. Inspect the listed degrees-of-freedom to ensure that masses or stiffnesses are not left out inadvertently.
- UFM 4416      \*\*\* USER FATAL MESSAGE 4416, NO DYNAMIC LOAD TABLE AVAILABLE.
- A frequency response or transient response analysis was requested, but no dynamic load data is available. Include dynamic load data (DLOAD, RLOADi, TLOADi) in model.
- UFM 4417      \*\*\* USER FATAL MESSAGE 4417, NO TRANSIENT RESPONSE LIST AVAILABLE.
- A transient response dynamic analysis was requested, but no transient response list is available. Include a TSTEP entry in the Bulk Data Section.
- UFM 4418      \*\*\* USER FATAL MESSAGE 4418, NO EIGENVALUE EXTRACTION DATA IS AVAILABLE.
- A dynamic analysis was requested, but no eigenvalue extraction data was available. Include eigenvalue extraction data EIGR or EIGRL in the Bulk Data.
- Possible causes are



- The METHOD command in the Case Control Section but no EIGR or EIGRL entry in the Bulk Data Section.
- No correspondence between Set IDs on the METHOD command and the EIGR/ EIGRL entries.

UWM 4420 \*\*\* USER WARNING MESSAGE 4420, THE FOLLOWING DEGREES-OF-FREEDOM ARE POTENTIALLY SINGULAR.

During decomposition, the degrees-of-freedom listed had pivot ratios greater than MAXRATIO. Verify that the degrees-of-freedom are not part of a mechanism and that elements do not have excessive stiffness. In SOLs 100 and higher, this condition causes run termination. PARAM,BAILOUT may be used to continue the run. See the *Simcenter Nastran Numerical Methods User's Guide*.

UFM 4421 \*\*\* USER FATAL MESSAGE 4421, NO FREQUENCY RESPONSE LIST AVAILABLE.

A frequency response dynamic analysis was requested, but no frequency data is available. Include frequency data (FREQ, FREQ1, FREQ2) in the Bulk Data.

UFM 4501 \*\*\* USER FATAL MESSAGE 4501, RLOADi CARD SELECTED IN TRANSIENT ANALYSIS. USE TLOADi.

RLOADi entries are used in frequency response analysis. These entries have no meaning in transient analysis. Replace RLOADi with TLOADi entries.

UWM 4561 \*\*\* USER WARNING MESSAGE 4561, INSUFFICIENT MEMORY FOR MODE ORTHOGONALITY CHECKS.

The amount of memory needed for eigenvector orthogonalization is  $1/2 \cdot [\text{number of eigenvectors} \cdot (\text{number of eigenvectors} + 1)] + 2 \cdot \text{BUFFSIZE} + \text{number of eigenvalues}$ . If this equation is not met, the modes are orthogonalized, but the checking function is not performed. However, all outputs from the module are provided. If the check is desired, you should either increase memory or decrease the number of eigenvectors to satisfy the above equation.

UFM 4562 \*\*\* USER FATAL MESSAGE 4562, TSTEP (TIME STEPS) DATA IS MISSING.

Transient analysis requires the time step data. Add a TSTEP Bulk Data entry, and select it with a TSTEP Case Control command.

UWM 4582 \*\*\* USER WARNING MESSAGE 4582, LSEQ CARD SID = \*\*\* REFERS TO A NONEXISTENT STATIC LOAD MATRIX COLUMN \*\*\* (NCOLS = \*\*\*).

The most likely cause occurs when changing an LSEQ entry on a restart without regenerating and assembling the static load matrix.

UFM 4603 \*\*\* USER FATAL MESSAGE 4603, THE LSEQ SET ID \*\*\*\* IS NOT UNIQUE WITH RESPECT TO OTHER STATIC LOAD IDS.

LSEQ set IDs must be unique with respect to all other static load set IDs.

UFM 4645 \*\*\* USER FATAL MESSAGE 4645, THE SHIFTED STIFFNESS MATRIX IS NOT POSITIVE DEFINITE.

The matrix sum  $K + \lambda \cdot M$  is decomposed by the Cholesky method at the start of the MGIV method of eigensolution. This decomposition requires that the matrix be positive-definite. A condition that prevents this is a massless mechanism (for example, a point mass on an offset with no rotational stiffness).

UFM 4646

\*\*\* USER FATAL MESSAGE 4646, THE MASS MATRIX IS NOT POSITIVE DEFINITE, USING THE HOU METHOD. USE MHOU INSTEAD.

The reduced mass matrix has columns that are not linearly independent. Common causes are rotation degrees-of-freedom whose only inertia terms result from point masses on offsets. Use the MHOU method instead since it does not require a positive definite mass matrix.

UFM 4647

\*\*\* USER FATAL MESSAGE 4647, INSUFFICIENT TIME TO COMPLETE \*\*\*\*\*.

CPU ESTIMATE = \*\*\*\*\* SEC CPU REMAINING = \*\*\*\*\* SEC I/O ESTIMATE = \*\*\*\*\* SEC I/O REMAINING = \*\*\*\*\* SEC

CPU and I/O limits are supplied on the Executive Control statement TIME (in minutes). The module where the terminated program is listed. If the time to completion appears reasonable, you should increase the estimates on the TIME statement and resubmit the run. For large models, an increase in the system memory request should also be considered.

UWM 4648

\*\*\* USER WARNING MESSAGE 4648, THE MODAL MASS MATRIX IS NOT POSITIVE DEFINITE.

The modal mass matrix cannot be decomposed by the Cholesky algorithm after merging elastic and free-body modes. (Cholesky decomposition is used to orthogonalize the eigenvectors with respect to the mass matrix.) The causes include the input of negative masses and the calculation of eigenvectors for eigenvalues approaching machine infinity. Inspect the model or ask for fewer eigenvectors using the F2 option. When this condition occurs, the eigenvectors are not orthogonalized or normalized. The second parameter of the READ or REIGL module is given a negative sign. This parameter is used in the solution sequences to branch to an error exit after printing the real eigenvalue table. You may use a DMAP Alter to print these eigenvectors if the cause of the problem is not apparent in the eigenvalues. The solution can be forced to completion by changing the sign of this parameter. You should be aware that a poor-quality solution is provided for this case. This poor solution may be useful for diagnosing the problem but should not be used for other purposes.

A possible cause of this error is when large offsets (large relative to the element length) are used for the BEAM element and coupled mass is selected.

UFM 4671

\*\*\* USER FATAL MESSAGE 4671, LOAD COMBINATION REQUESTED BUT LSEQ CARDS DO NOT EXIST FOR SID = \*\*\*\*\*.

Check the LSEQ entries.

UFM 4683

\*\*\* USER FATAL MESSAGE 4683, MASS/STIFFNESS MATRIX NEEDED FOR EIGENVALUE ANALYSIS.



The eigensolution module was given a purged (that is, nonexistent) mass or stiffness matrix. Common causes include the deletion of mass density input on MATi entries, user restart errors in the superelement solution sequences. It therefore sets the number of generalized coordinates to zero. This condition can be detected from UIM 4181. Provide mass matrix generating data by any of several means including a mass density entry on material entries, concentrated masses, and g-type DMIG entries.

Possibly no mass matrix is defined. Check for the following:

- RHO entry on MATi
- NSM entry on element properties (i.e., PSHELL, PBAR)
- CONMi or CMASSi

Simcenter Nastran needs at least one of the above to compute the mass matrix. Incorrect cross-sectional properties may also lead to this error but typically show up as another error.

UWM 4698

\*\*\* USER WARNING MESSAGE 4698, STATISTICS FOR DECOMPOSITION OF MATRIX \*\*\*\*. THE FOLLOWING DEGREES-OF-FREEDOM HAVE FACTOR DIAGONAL RATIOS GREATER THAN \*\*\*\*, OR HAVE NEGATIVE TERMS ON THE FACTOR DIAGONAL.

During decomposition, the degrees-of-freedom listed have pivot ratios that are greater than maxratio or are negative. Verify that the degrees-of-freedom are not part of a mechanism and that elements do not have excessive stiffness. In SOLs 100 and higher this condition causes run termination. PARAM,BAILOUT may be used to continue the run to obtain messages issued by subsequent modules. See the *Simcenter Nastran Numerical Methods User's Guide*.

UIM 5010

\*\*\* USER INFORMATION MESSAGE 5010, STURM SEQUENCE DATA FOR EIGENVALUE EXTRACTION. TRIAL EIGENVALUE = (real), CYCLES = (real), NUMBER OF EIGENVALUES BELOW THIS VALUE = (integer).

This message is automatic output during eigenvalue extraction using the Lanczos and SINV methods. This message can be used, along with the list of eigenvalues, to identify the modes found. See the *Simcenter Nastran Numerical Methods User's Guide*.

UFM 5025

\*\*\* USER FATAL MESSAGE 5025, LAMA PURGED. DSTA MODULE TERMINATED.

The LAMA data block contains a list of natural frequencies and may be purged because no eigenvalues were computed or the data block was not properly recovered on restart.

UIM 5218

\*\*\* USER INFORMATION MESSAGE 5218, EIGENVALUE APPROACHING INFINITY AT \*\*\*\* TH MODE. EIGENVECTORS WILL NOT BE COMPUTED BEYOND THIS POINT.

The MHOu and AHOu methods substitute a very large number for eigenvalues that approach machine infinity. If eigenvectors are computed for these artificial values, they may be numerical noise, or they may cause overflows. Eigenvector

computation is halted at the first machine infinity instead even if you requested eigenvectors in this range.

UIM 5222

\*\*\* USER INFORMATION MESSAGE 5222, COUPLED/UNCOUPLED SOLUTION ALGORITHM USED.

The modal methods use uncoupled solution algorithms, if possible. The uncoupled algorithms are considerably more economical than the coupled algorithms. Coupled algorithms are required when any of the following effects are present: transfer functions, DMIG requests of the p-type, element damping, and PARAM,G.

Consider the use of modal damping (TABDMP1 entry) to reduce the cost of your analysis in modal solutions.

UFM 5225

\*\*\* USER FATAL MESSAGE 5225, ATTEMPT TO OPERATE ON THE SINGULAR MATRIX \*\*\*\*\* IN SUBROUTINE DCMP.

This message is preceded by the listing of the grid point ID and degrees-of-freedom for any null columns.

UIM 5236

\*\*\* USER INFORMATION MESSAGE 5236, THE FREQUENCY RANGE HAS BEEN SPLIT INTO \*\*\*\*\* SUBREGIONS.

The overall frequency range for eigenanalysis is split into several smaller ranges when using the SINV option to calculate modes and frequencies.

UFM 5238

\*\*\* USER FATAL MESSAGE 5238, THE NUMBER OF ROOTS IN THE DEFINED FREQUENCY RANGE IS GREATER THAN 600.

More than 600 roots are in the desired frequency range, which is greater than the maximum allowed using SINV. Decrease the size of the frequency range.

UIM 5239

\*\*\* USER INFORMATION MESSAGE 5239, BISECTIONING IN THE \*\*\*-\*\*\* INTERVAL.

The frequency subregion encompassing eigenvalues xx-yy is cut in half in order to find the remaining roots.

UIM 5240

\*\*\* USER INFORMATION MESSAGE 5240, THE BISECTION VALUE IS: \*\*\*\*\*

The selected value is midway between the lowest and highest frequencies in the frequency subregion.

UIM 5241

\*\*\* USER INFORMATION MESSAGE 5241, MISSING ROOT(S) IN THE \*\*\*-\*\*\* INTERVAL.

The Sturm sequence check has indicated that roots are missing in the frequency range, and they cannot be found by further bisectioning.

If the run terminates with missing roots, decrease the frequency range.

UIM 5242

\*\*\* USER INFORMATION MESSAGE 5242, THE ROOT FOUND IS NOT THE LOWEST ONE ABOVE FMIN.

The Sturm sequence check indicates that at least one unfounded root exists between FMIN and the lowest frequency root found. Set FMAX close to the lowest frequency found, so that the lower roots can be found.

UIM 5274	<p>*** USER INFORMATION MESSAGE 5274, THE ACTUAL TIME OF (****) TRIDIAGONALIZATION IS: ****, THE ACTUAL TIME OF EIGENVALUE ITERATION IS: ****, THE ACTUAL TIME OF EIGENVECTOR GENERATION IS: ****.</p> <p>The time spent in the major operations of the real eigensolution module are output. Note that the number of eigenvectors requested has a large effect on solution cost.</p>
UFM 5288	<p>*** USER FATAL MESSAGE 5288, NO ROOT EXISTS ABOVE FMIN.</p> <p>This message occurs when the eigenproblem of finding any number of roots above FMIN is expected, but the Sturm number indicates that there is no root above FMIN. Reduce FMIN in order to attempt to find a root.</p>
SFM 5299	<p>*** SYSTEM FATAL MESSAGE 5299 (This text varies depending on the reason for termination; see the description given below.)</p> <ol style="list-style-type: none"> <li>1. Insufficient storage for the Lanczos method.</li> <li>2. Factorization error on three consecutive shifts.</li> <li>3. Stack overflow in the Lanczos method.</li> <li>4. Unrecoverable termination from the Lanczos method.</li> <li>5. Insufficient working storage.</li> <li>6. Finite interval analysis error (see UIM 6361).</li> </ol> <p>This message can also occur for models with two or more widely separated groups of repeated roots. An avoidance is to search each group separately. This error may also be caused by a massless mechanism, which can be confirmed by performing a static analysis.</p> <p>See the <i>Simcenter Nastran Numerical Methods User's Guide</i> for more information.</p>
UFM 5400	<p>*** USER FATAL MESSAGE 5400, INCORRECT RELATIONSHIP BETWEEN FREQUENCY LIMITS.</p> <p>You have incorrectly specified <math>V1 &gt; V2</math>. Check <math>V1</math>, <math>V2</math> specified on the EIGRL Bulk Data entry.</p>
SFM 5401	<p>*** SYSTEM FATAL MESSAGE 5401 (REIGL), LANCZOS METHOD IS UNABLE TO FIND ALL EIGENVALUES IN RANGE. ACCEPTED EIGENVALUES AND ADDITIONAL ERROR MESSAGES MAY BE LISTED ABOVE. USER ACTION: RERUN WITH ANOTHER METHOD OR ANOTHER SETTING ON EIGRL ENTRY.</p> <p>This message can be issued if insufficient memory is available for Lanczos with sparse decomposition. It can also be issued if UFM 5299 occurs. See the <i>Simcenter Nastran Numerical Methods User's Guide</i>.</p> <p>This condition can be related to the occurrence of UWM 5411.</p>

- UWM 5402      \*\*\* USER WARNING MESSAGE 5402, ---THE PROBLEM HAS NO STIFFNESS MATRIX.  
The problem requires a stiffness matrix. Verify that property entries are specified correctly.
- UIM 5403      \*\*\* USER INFORMATION MESSAGE 5403, CPU TIME AT START OF LANCZOS ITERATION \*\*\*\*.  
Since several Lanczos iterations may be executed during one application of the Lanczos method (each shift is followed by at least one iteration), this information is given to measure the time required for the individual iterations.
- UWM 5404      \*\*\* USER WARNING MESSAGE 5404, NEGATIVE MODAL MASS TERM, IS ENCOUNTERED DURING INVERSE ITERATION. PROCESS ABORTED.  
The modal mass matrix should have unit diagonal terms (for mass normalization). Negative terms may indicate negative eigenvalues. If these negative terms are computational zeroes (rigid-body modes, for example), then the negative terms are acceptable. If the negative terms are finite values, there may be a modeling problem.
- UWM 5405      \*\*\* USER WARNING MESSAGE 5405, ERROR OCCURRED DURING ITERATION. ERROR NUMBER IS : Y (SEE DESCRIPTION FOR VALUES OF Y AND USER ACTION.)  
This message marks the breakdown of the inverse iteration process in the Lanczos method. See the *Simcenter Nastran Numerical Methods User's Guide* for additional values and actions.

Y Value		User Action
-11	File open error in interface; see GINO error message.	This error should not occur; report error to Siemens Digital Industries Software.
-12	File open error in postprocessing; see GINO error message.	This error should not occur; report error to Siemens Digital Industries Software.
-13	File read error; see GINO error message.	This error should not occur; report error to Siemens Digital Industries Software.
-21	Insufficient space for blocksize = 1.	Increase memory.
-22	Three consecutive factorizations failed at a shift.	Possible ill-conditioning; check model.
-23	Lanczos internal table overflow due to enormous number of shifts.	Specify smaller internal; may be necessary to have several runs.
-31	Internal error in Lanczos (REIGL) module.	This error should not occur; report error to Siemens Digital Industries Software.

Y Value		User Action
-32	No convergence in solving the tridiagonal problem.	Possible ill-conditioning; check model.
-33	Too many eigenvalues were found; inconsistency between the roots found and Sturm number.	Check the orthogonality of the eigenvectors; if it is good, then ignore this warning.

UWM 5406

\*\*\* USER WARNING MESSAGE 5406 NO CONVERGENCE IN SOLVING THE TRIDIAGONAL PROBLEM.

This message signals eigensolution problems in the Lanczos method. There is possible ill-conditioning; check your model.

UWM 5407

\*\*\* USER WARNING MESSAGE 5407, INERTIA (STURM SEQUENCE) COUNT DISAGREES WITH THE NUMBER OF MODES ACTUALLY COMPUTED IN AN (SUB) INTERVAL

This message shows a serious problem. Spurious modes were found in the Lanczos method. Check the multiplicity of the roots given in the interval. See the *Simcenter Nastran Numerical Methods User's Guide*.

UWM 5408

\*\*\* USER WARNING MESSAGE 5408, FACTORIZATION FAILED. SHIFT CHANGED TO \*\*\*\*.

No user action to be taken. This message occurs only for the Lanczos method.

UWM 5411

\*\*\* USER WARNING MESSAGE 5411, NEGATIVE TERM ON DIAGONAL OF MASS MATRIX (VIBRATION) OR STIFFNESS (BUCKLING), ROW \*\*\*\*, VALUE = \*\*\*\*

The message is given from the REIGL module which performs a necessary (but not sufficient) check on the positive semi-definiteness of the indicated matrix. Look for evidence of negative mass, such as minus signs on input. Negative terms on the factor of the indicated matrix must be removed for correct answers.

Something has caused a negative term on the diagonal of the mass or stiffness matrix. Look for explicitly defined negative mass and/or stiffness terms. Also, check the continuation entries on the PBEAM entry. An incorrect entry for the SO field may lead to improper mass definition. For example, if SO is set to NO at a particular X/XB location, the continuation entry for defining four stress locations on the cross section (C, D, E, F) is not used. If SO is NO but the C, D, E, and F points are entered in error, negative mass terms can result if either E1 or E2 entries are entered. The offending DOF can be traced using the USET tables.

The Lanczos method gives wrong answers for indefinite matrices. The existence of negative diagonal terms indicates a subclass of indefinite matrix.

See the *Simcenter Nastran Numerical Methods User's Guide* for more information.

UIM 5458

\*\*\* USER INFORMATION MESSAGE 5458, (\*\*\*\*) METHOD IS SELECTED. or (\*\*\*\*) METHOD IS (\*\*\*\*)

The exact text of this message depends on the METHOD field on the selected EIGR Bulk Data entry. This message indicates the eigensolution status (all eigenvalues found, not all found, etc.).

UFM 6133

\*\*\* USER FATAL MESSAGE 6133 (DFMSDD), SINGULAR MATRIX IN SPARSE DECOMPOSITION.

USER ACTION: CHECK MODEL.

This message is often followed by UFM 4645, UFM 4646, or UWM 4648.

UFM 6134

\*\*\* USER FATAL MESSAGE 6134 (DFMSDD), MATRIX IS NOT POSITIVE DEFINITE IN SPARSE DECOMPOSITION.

USER ACTION: CHECK MODEL.

This message is often followed by UFM 4645, UFM 4646, or UWM 4648.

SFM 6135

\*\*\* SYSTEM FATAL MESSAGE 6135, ERROR IN READING SYMBOLIC FACTOR IN SPARSE FBS.

This message may be issued if the FBS module is using a sparse method to solve factors which are not decomposed by the sparse method. This message can also be caused by a compatibility or database integrity problem.

UFM 6136

\*\*\* USER FATAL MESSAGE 6136 (\*\*\*\*), INSUFFICIENT CORE FOR (SYMBOLIC/ NUMERIC) PHASE OF SPARSE DECOMPOSITION.

USER ACTION: INCREASE CORE BY \*\*\*\* WORDS.

USER INFORMATION: !!! NOW REVERTING BACK TO ACTIVE COLUMN DECOMPOSITION UPON USER REQUEST !!!

If this message is issued in the symbolic phase, the memory estimate is not necessarily conservative and even more memory may be required (although this estimate is fairly accurate for Version 68). Also, the memory increase required is only for the symbolic phase. It is not unusual for the decomposition phase to require more memory than the symbolic phase. To increase the chances for a successful run, increase the memory even more than the amount indicated in this message. After the run is complete, determine the amount of memory actually used and use this as a guideline for similar runs in the future.

The user information message is written if  $SYSTEM(166) = 1$  (that is, if there is not enough memory for sparse decomposition, and you should switch to regular decomposition).

UWM 6137

\*\*\* USER WARNING MESSAGE 6137 (DFMSDD), INPUT MATRIX IS RANK DEFICIENT, RANK = \*\*\*\*.

USER ACTION: CHECK MODEL.

One of your matrices is singular. See the *Simcenter Nastran Numerical Methods User's Guide* for a discussion of singularity.

UFM 6138

\*\*\* USER FATAL MESSAGE 6138 (DFMSB), INSUFFICIENT CORE FOR SPARSE FBS.

USER ACTION: INCREASE CORE BY \*\*\*\* WORDS.



See UFM 6136.

UIM 6214

\*\*\* USER INFORMATION MESSAGE 6214, FEWER THAN REQUESTED VECTORS CALCULATED, DUE TO INSUFFICIENT TIME.

This information message occurs in the READ module when there is insufficient time to compute eigenvectors. Resubmit the job with an increased time limit (TIME).

UWM 6243

\*\*\* USER WARNING MESSAGE 6243 (READ) ---- THE DEGREE OF FREEDOM (D.O.F) REQUESTED FOR POINT NORMALIZATION HAS NOT BEEN SPECIFIED ON THE EIGR OR EIGB ENTRY.

USER INFORMATION: THE D.O.F PRECEDING THE REQUESTED D.O.F. IN THE INTERNAL SEQUENCE LIST WILL BE USED

The point requested was not in the a-set, so another point was chosen.

UIM 6361

\*\*\* USER INFORMATION MESSAGE 6361-LANCZOS MODULE DIAGNOSTICS

This message prints various levels of diagnostics for the Lanczos method. The amount of print depends on the message level set on the EIGRL entry. See the *Simcenter Nastran Numerical Methods User's Guide* for more information.

UIM 6480

\*\*\* USER INFORMATION MESSAGE 6480 (REIGLA) --- EXTERNAL IDENTIFICATION TABLE FOR DECOMPOSITION MESSAGES FOR MATRIX \*\*\*\* ROW NUMBER \*\*\*\* = GRID ID \*\*\*\* + COMPONENT \*\*\*\*

This message is output from the REIGL module when using sparse decomposition to convert the internal (row number oriented) diagnostic messages to external (grid and component) form.





# L. References and Bibliography

## L.1 Overview

This appendix includes references of interest in the field of dynamic analysis. Two categories are included. The first category, General References, lists books that cover the general range of structural dynamic analysis. The second category, Bibliography, is an excerpt from the dynamic analysis section of the Simcenter Nastran Bibliography.

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