ROLE OF PASSIVE DAMPING IN ACTIVE STRUCTURAL CONTROL

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OBJECTIVE:

Large space structures are vunerable to vibration problems due to rapid maneuvering disturbances. To obtain stability and pointing accuracy, a vibration control method (i.e. the integration of passive damping with an active structural control system) has been used to study structural responses.

OBJECTIVE

TO INVESTIGATE:

- **THE ROLE OF PASSIVE DAMPING ON DYNAMIC RESPONSE OF ACTIVELY CONTROLLED LARGE SPACE STRUCTURES, e.g. SBL**
- **EFFECT OF PASSIVE DAMPING ON THE COMPLEXITY OF ACTIVE STRUCTURAL CONTROL SYSTEM DESIGNS**

ADVANCED MATERIALS FOR SPACE STRUCTURES - VIBRATION SUPP.

The structural response of a large SBL spacecraft can be studied by establishing a simple finite element model. A control system using 21 collocated sensors and actuators were mounted on the structure and parametric studies were performed wherein control forces and control effort were calculated versus passive system damping (modal damping). The LOS requirement was assumed to be 50 nanoradians within 3.0 seconds of end of slew.

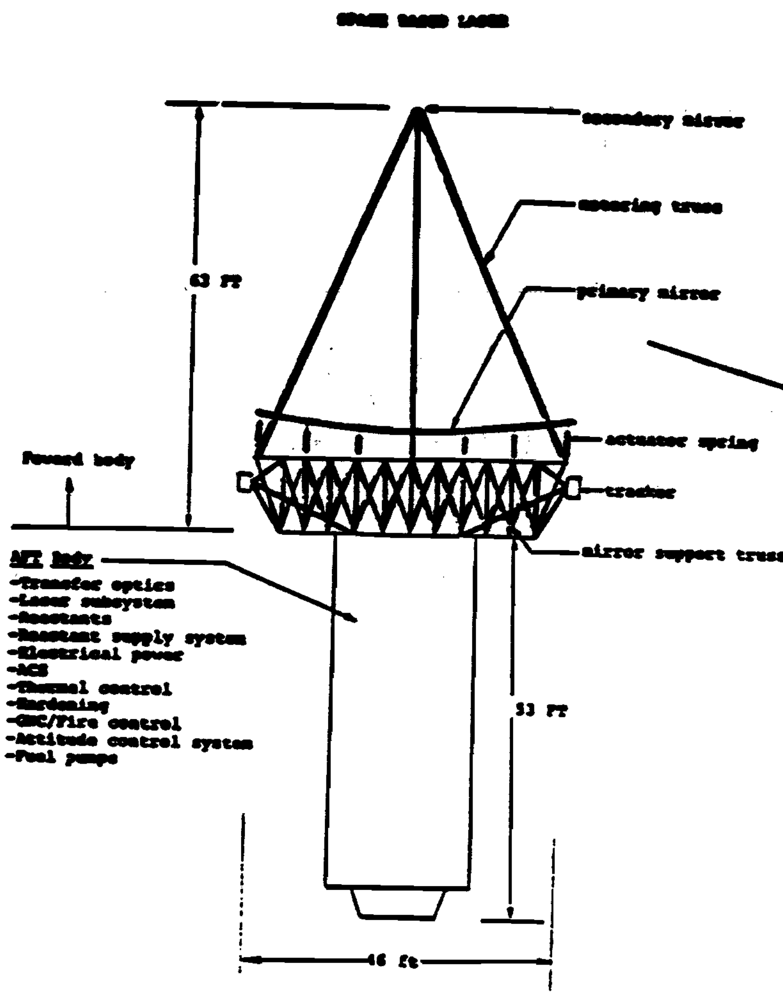
ACTIVE CONTROL / PASSIVE DAMPING SYNERGISM STUDY

- VERY PRELIMINARY RESULTS
- GENERIC SBL RETARGET
- 24 FLEXIBLE MODES
- 21 COLLOCATED SENSOR/ACTUATORS
- 1 SEC. BANG-BANG TORQUE PROFILE
- AT 1.1 SEC. ACTIVE CONTROL TURNED ON
- REQUIREMENT AT 3.0 SEC. IS 50 NANORADIANS

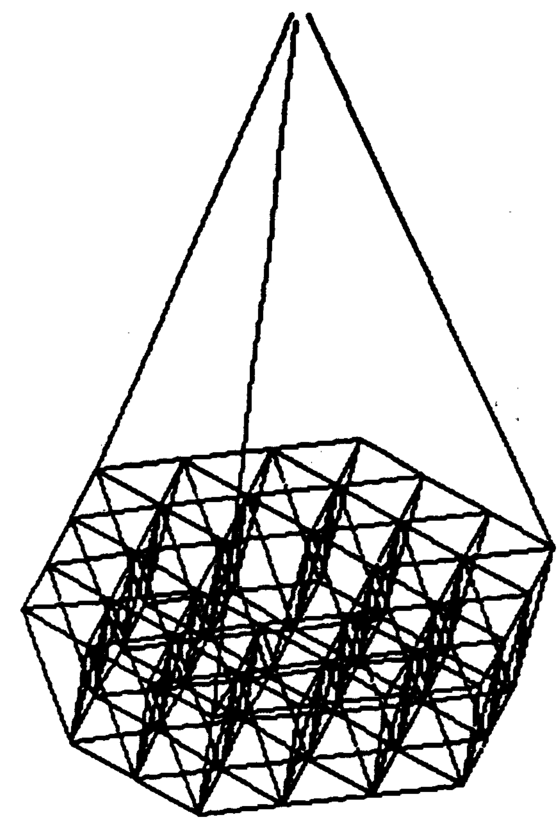
FINITE ELEMENT MODEL - SBL

The design of this SBL system is based on a generic design that was generated by the AFWL. The SBL structure is 116 feet long; it consists of two parts: a forward body and an AFT body. The forward body section consists of a secondary mirror, metering truss, and primary mirror. The metering structure is a tripod structure that provides accurate alignment/separation of the primary and secondary mirrors.

FINITE ELEMENT MODEL SBL



SBL DESIGN

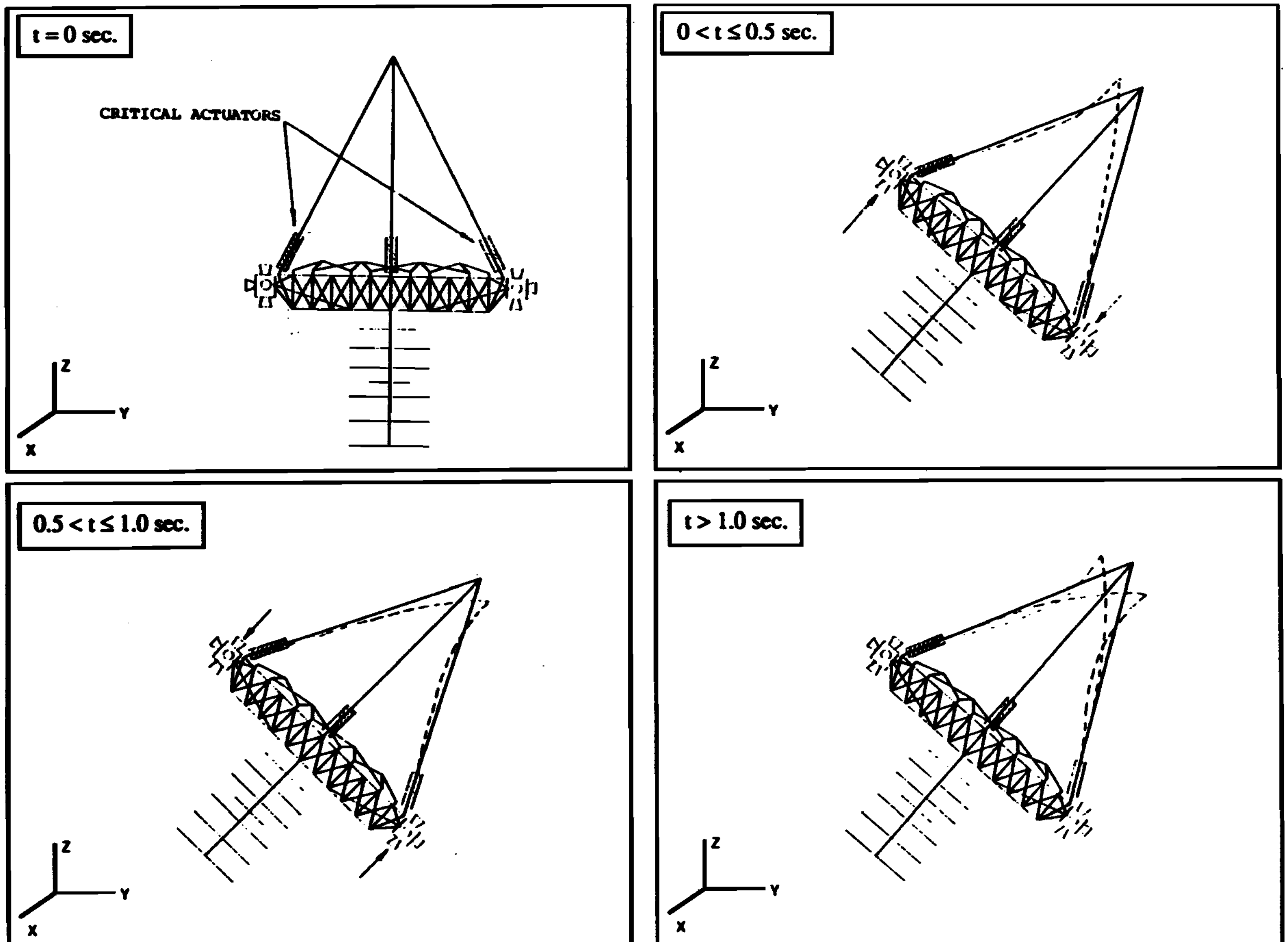


FEM MODEL

ADVANCED MATERIALS FOR SPACE STRUCTURES - VIBRATION SUPP.

The slewing and retargeting is mathematically generated by firing the jets in equal and opposite directions to create enough thrust to turn the beam expander 10 degrees from the original position. The time interval for forward firing and reverse firing is a total of 1 second.

ADVANCED MATERIALS FOR SPACE STRUCTURES
VIBRATION SUPPRESSION



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SPACE BASED LASER - MODAL ANALYSIS DESCRIPTION

The modal analysis was performed on this SBL structural model with three different specific stiffness values. The eigenvectors are calculated by the unit mass normalized method (phi * M * phi-transposed = I) where phi = eigenvector matrix and M = mass matrix and they are subsequently used for generating the state variables.

SPACE BASED LASER
MODAL ANALYSIS DESCRIPTION

- **SET UP FINITE ELEMENT MODEL**
  - 164 NODES
  - 253 BAR ELEMENTS
  - 21 ELASTIC SPRING ELEMENTS
  - 51 CONCENTRATED MASS ELEMENTS
  - 22 RIGID BODY ELEMENTS

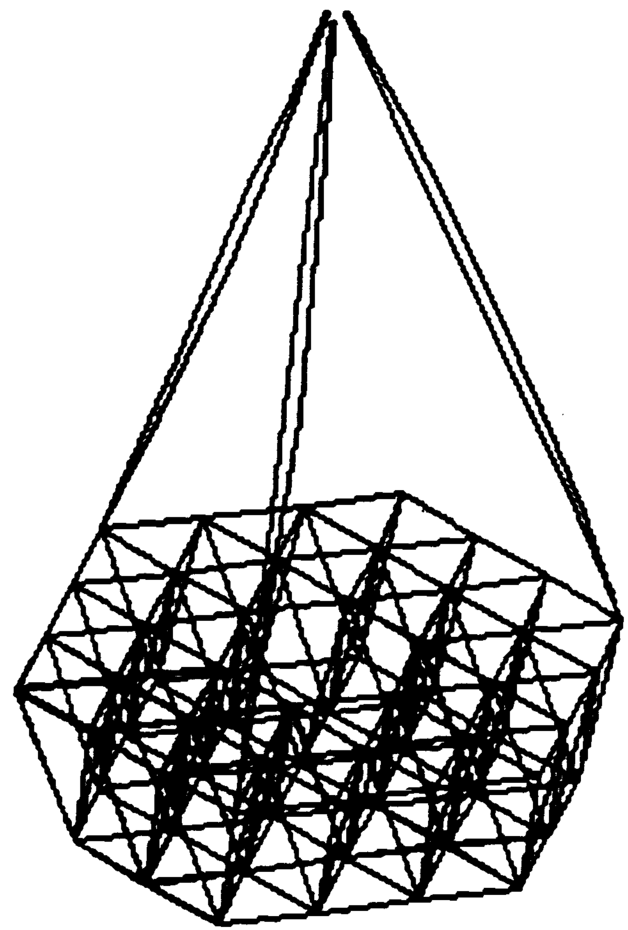
- **CALCULATE NATURAL FREQUENCIES AND MODE SHAPES**
  - MASS NORMALIZED MODE SHAPE CALCULATION
  - SIGNIFICANT MODES ARE PLOTTED FOR CONTROL SYSTEM DESIGN

TYPICAL MODE SHAPE

The mode shape plot reveals the natural responses of all structural members. The deflected shape of the structure is used for determining critical vibration locations, so that the control system designer knows where to install the collocated sensors and actuators in order to obtain optimal damping, thereby achieving the LOS requirement effectively.

TYPICAL MODE SHAPE

(THIRD FLEXIBLE MODE - FREQUENCY = 12.856 Hz.)



PREPARATION FOR CONTROL DESIGN AND ANALYSIS

MATLAB was used in the control system design procedure. Prior to using MATLAB, the control system designer must generate the eigenvector matrix, eigenvalue matrix, system damping matrix, directional cosine matrix (nodal connectivity of linear actuators and sensors), and control distribution matrix.

PREPARATION FOR CONTROL DESIGN AND ANALYSIS

- ANALYTICAL TOOLS
  - MATLAB
- PRE-PROCESSING PROCEDURE
  - CONVERT FINITE ELEMENT ANALYSIS RESULTS INTO MATLAB FORMAT
  - LOCATE ACTUATORS AND SENSORS AT THE CRITICAL (MAXIMUM DEFLECTION) LOCATIONS
  - COMPUTE DIRECTION COSINES AT EACH ACTUATOR LOCATION; GENERATE CONTROL DISTRIBUTION MATRIX

CONTROL SYSTEM DESIGN & ANALYSIS PROCEDURE

In this parametric study, 30 modes are used for the control design and analysis. There are 6 rigid body modes and 24 flexible modes. The 6 rigid body modes are considered as slew motion (i.e. the entire SBL structure moves from one reference position to another). The time interval for slewing the SBL structure 10 degrees was assumed to be 1.0 second. At 1.1 seconds the vibration control system is activated to damp out the vibration due to the 24 flexible modes. The actuators/sensors control system continue to function until the LOS requirement is achieved (e.g., LOS error is less than or equal to 50 nanoradians within 2.0 seconds).

CONTROL SYSTEM DESIGN AND ANALYSIS PROCEDURE

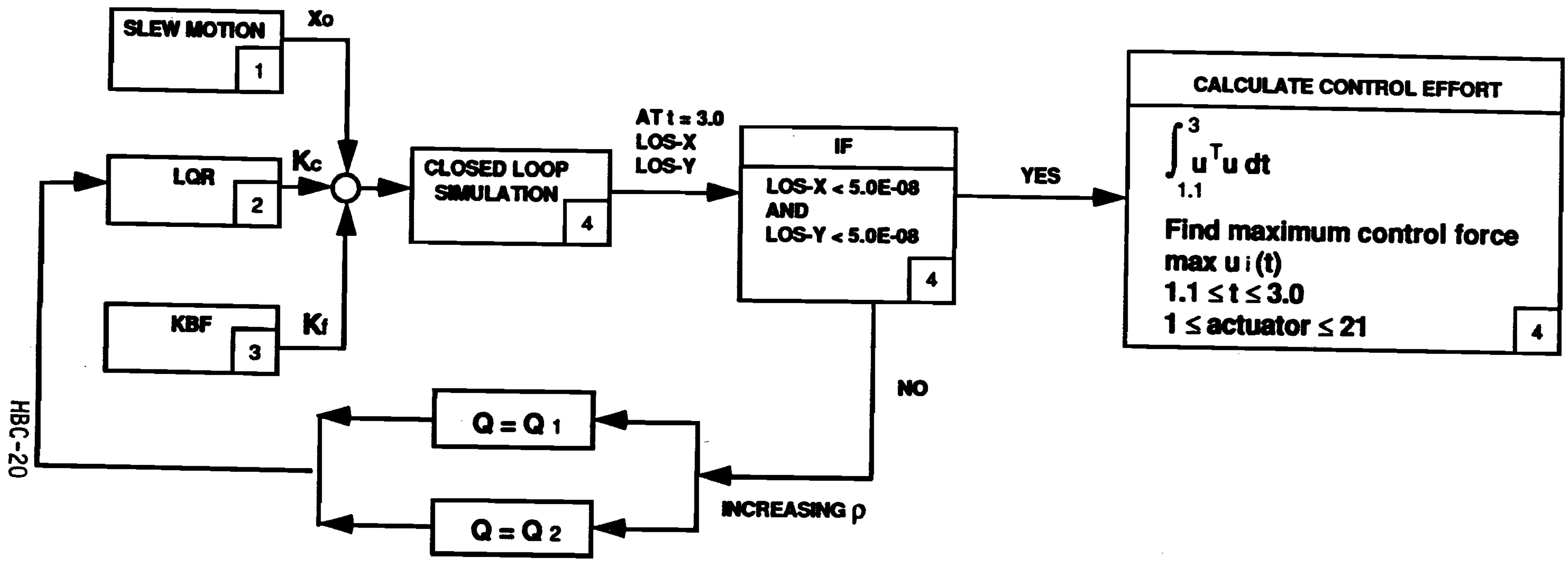
- **SLEW MOTION (RIGID BODY MODES)**
  - SLEW TO A DESIRED POSITION (10°) USING JET FORCES (IN 1 SECOND)
  - CONSIDER 6 RIGID BODY MODES AND 24 FLEXIBLE MODES
  - LET THE RIGID BODY MOTION STOP AT THE END OF THE SLEW
  - FIND THE STATE AT 0.1 SECOND AFTER THE SLEW

- **ACTIVE CONTROL DESIGN (FLEXIBLE MODES)**
  - SUPPRESS THE VIBRATION DUE TO THE SLEW BELOW 50 NANORADIANS AFTER $t = 3$ SECONDS
  - CONSIDER ONLY 24 FLEXIBLE MODES
  - USE THE STATE OF THE FLEXIBLE MODES AT $t = 1.1$ SEC. AS THE INITIAL STATE OF THE ACTIVE CONTROL DESIGN

CLOSED LOOP SIMULATION

This chart shows the control system design/analysis procedure. Two different expressions were used for representing performance in the objective function, leading to slightly different results as shown in the following charts.

CLOSED LOOP SIMULATION



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CODES:

1	SIMJET.M
2	LQRGEN.M
3	LQEGEN.M
4	ENR.M

(1)

$$(LOS\,x)^2 + (LOS\,y)^2 + (LOS\,\dot{x})^2 + (LOS\,\dot{y})^2 = x^T \bar{Q}_1 x$$

$$Q_1 \triangleq \text{diag.}\,(\bar{Q}_1)$$

(2)

$$(LOS\,x)^2 + (LOS\,y)^2 = x^T \bar{Q}_2 x$$

$$Q_2 \triangleq \bar{Q}_2 + \varepsilon I$$

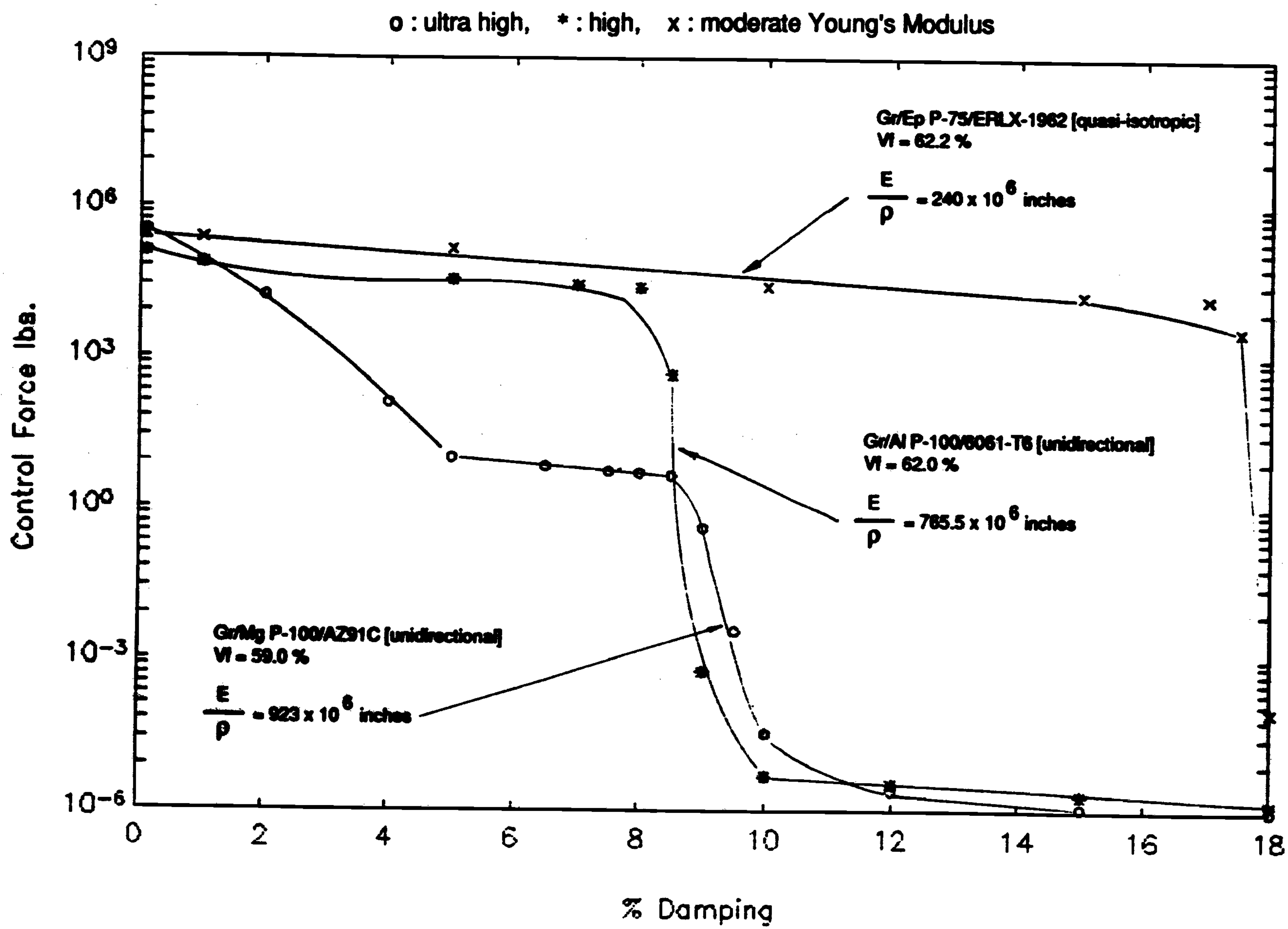
where $\varepsilon = 1.0 \times 10^{-18}$

CONTROL FORCE VS.% DAMPING

Case Q = Q1

In this case the control force provided by the actuators decreases as the system damping increases. For the case of high specific modulus, the control force tends to go near zero at 8.5 % system damping.

CONTROL FORCE VS. % DAMPING



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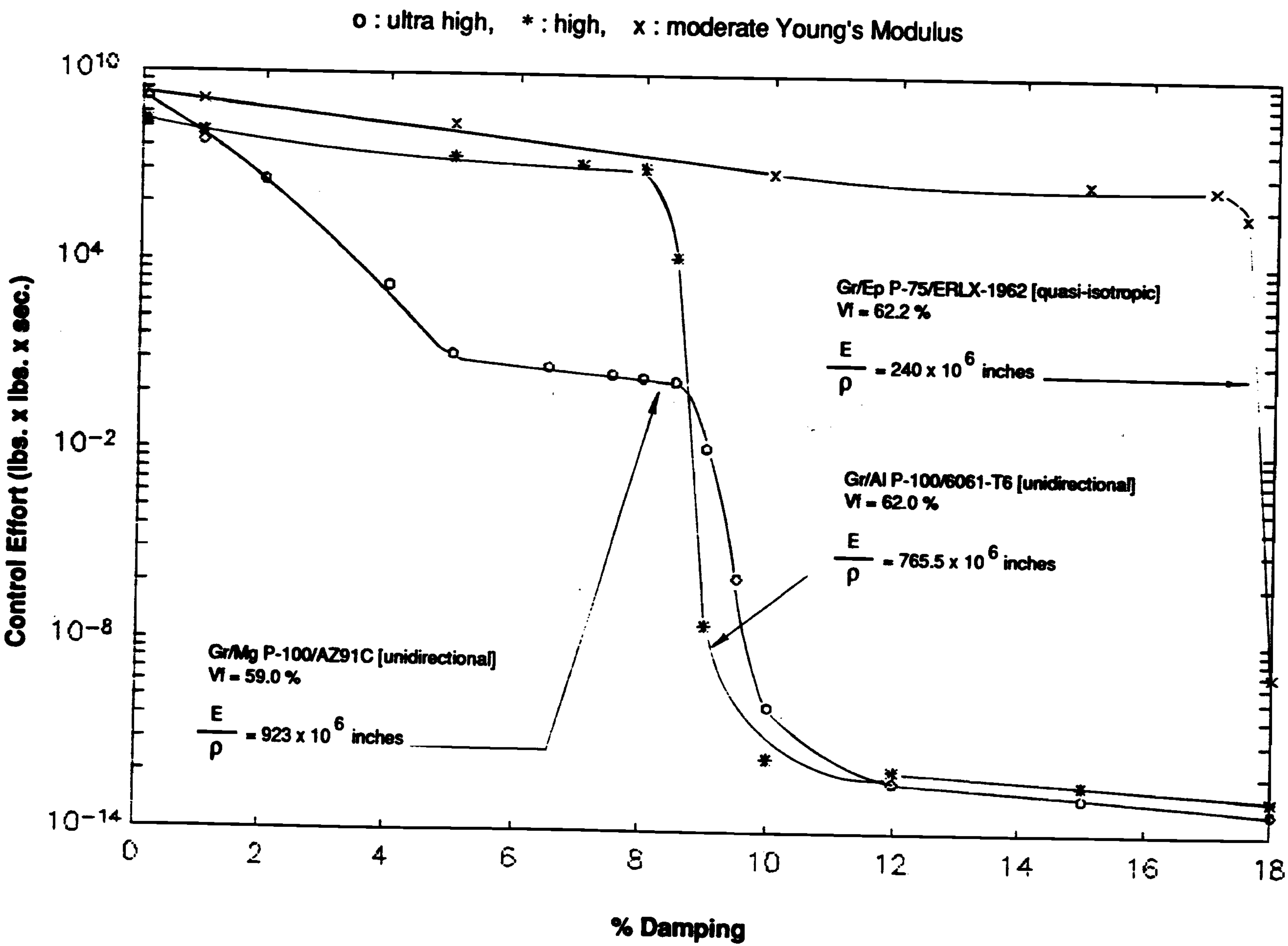
CONTROL EFFORT VS. % DAMPING

Case Q = Q1

For the case Q=Q1, the control effort required to control the vibration within the 50 nanoradian limit is plotted against the percent system damping.

CONTROL EFFORT VS. % DAMPING

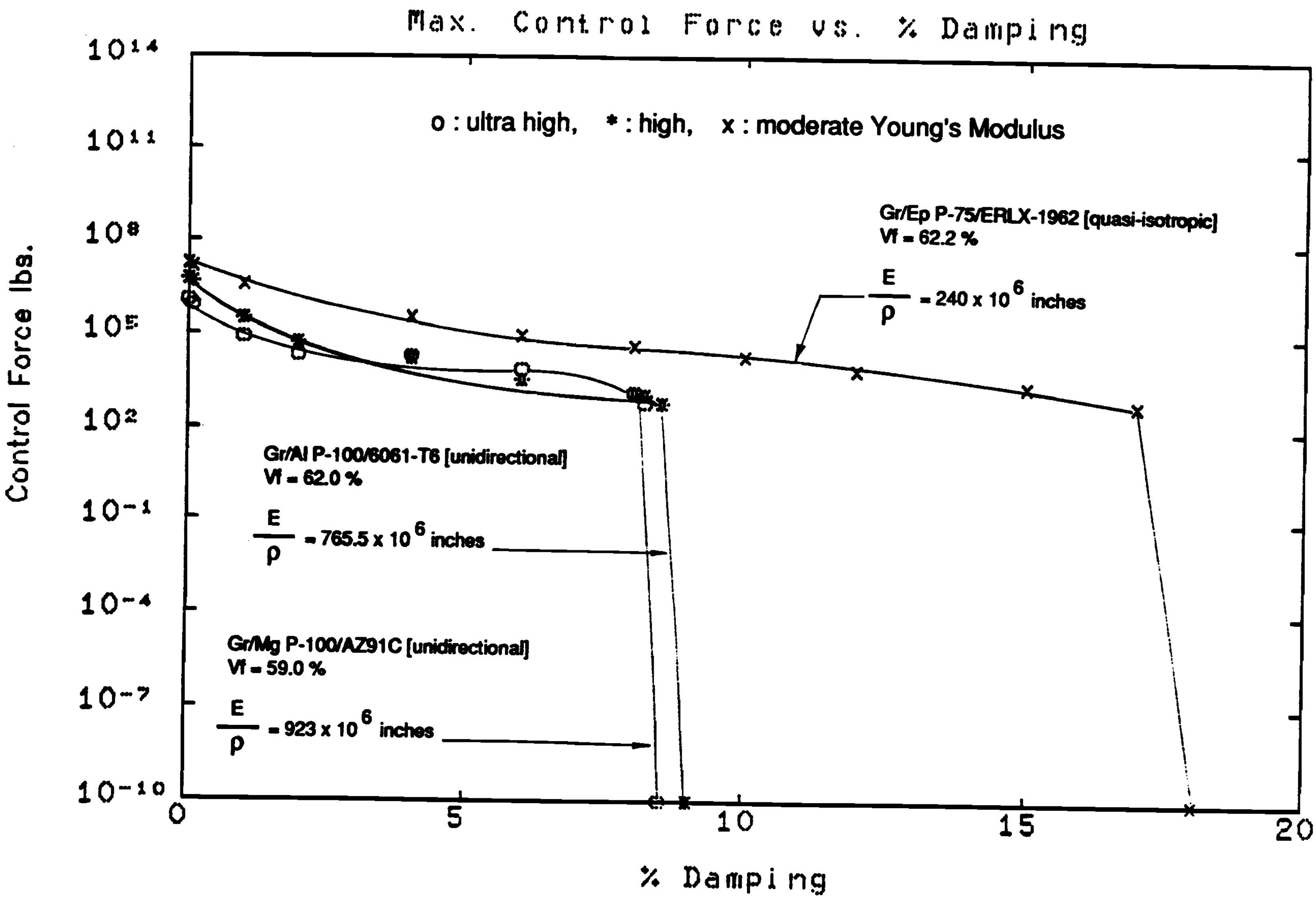


CONTROL FORCE VS. % DAMPING

Case Q = Q2

The control force provided by the actuators decreases as the system damping increases. For the case of high specific modulus, the control force tends to go near zero at 8.5% system damping.

CONTROL FORCE VS. % DAMPING

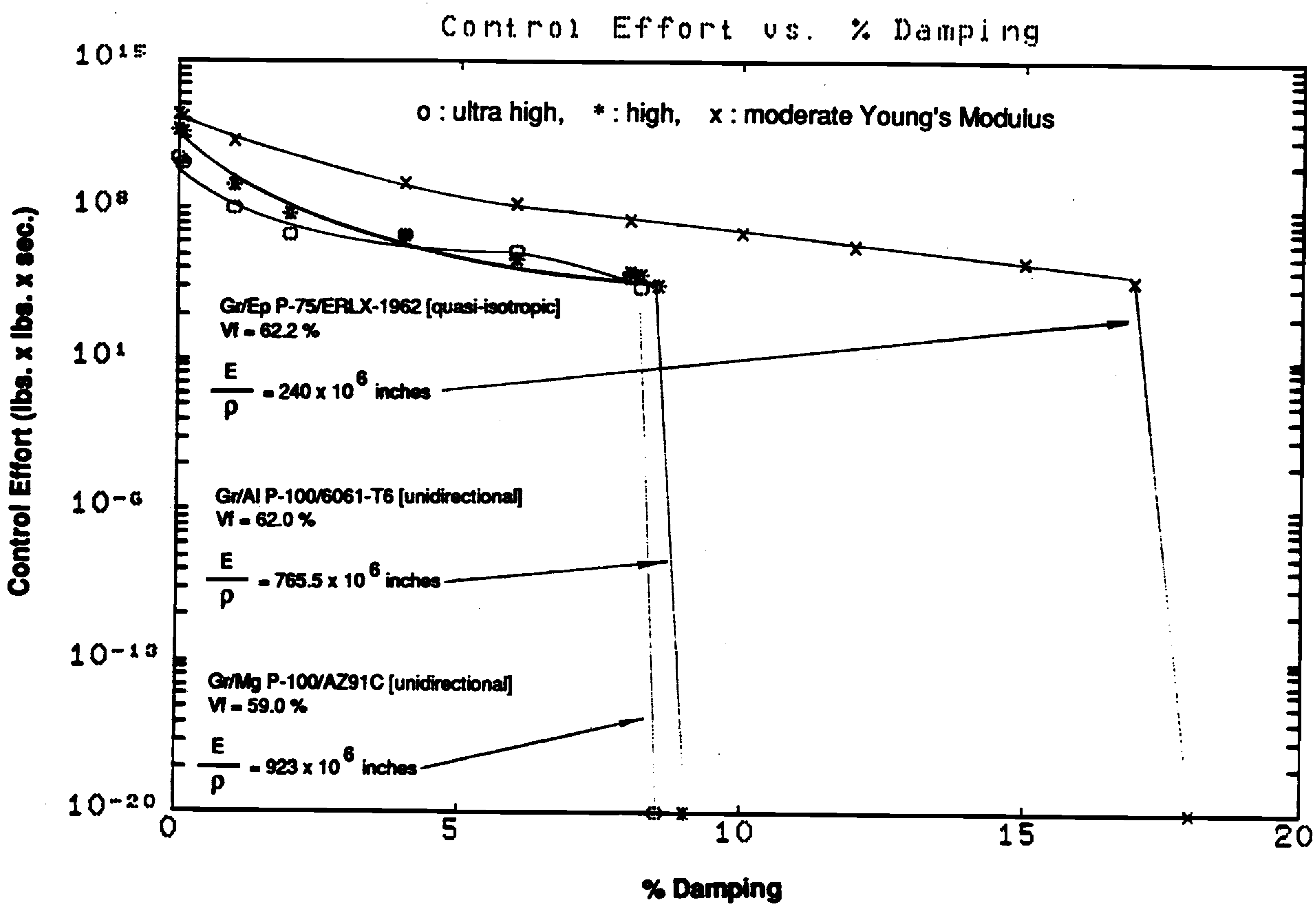


CONTROL EFFORT VS. % DAMPING

Case Q = Q2

For the case Q=Q2 the control effort required to control the vibration within the 50 nanoradian limit is plotted against the percent system damping.

CONTROL EFFORT VS. % DAMPING



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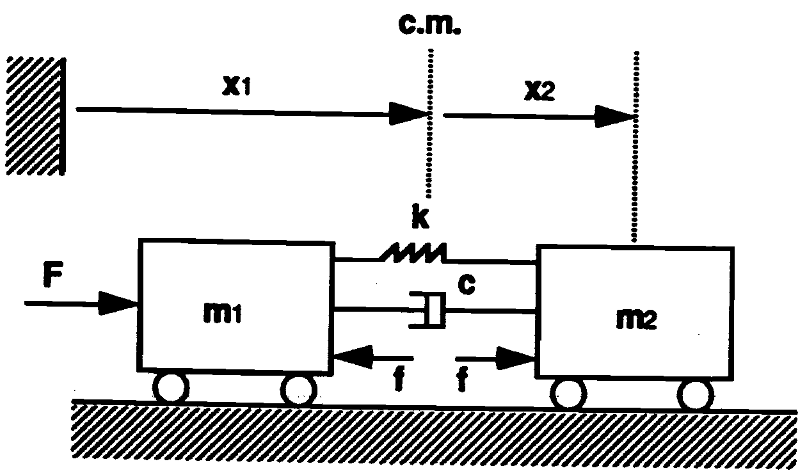
TWO MASS SYSTEM

The role of passive damping together with an active control system on dynamic structural response was also investigated by using a two degree of freedom spring mass system. This two mass system provides insight into the role passive techniques can play in vibration suppression. The provided actuator force for this two mass system is denoted as "f".

TWO - MASS SYSTEM



$m_1 = m_2 = m = 1/2$

$\frac{2k}{m} = \omega^2, \ \frac{2c}{m} = 2\zeta\omega$

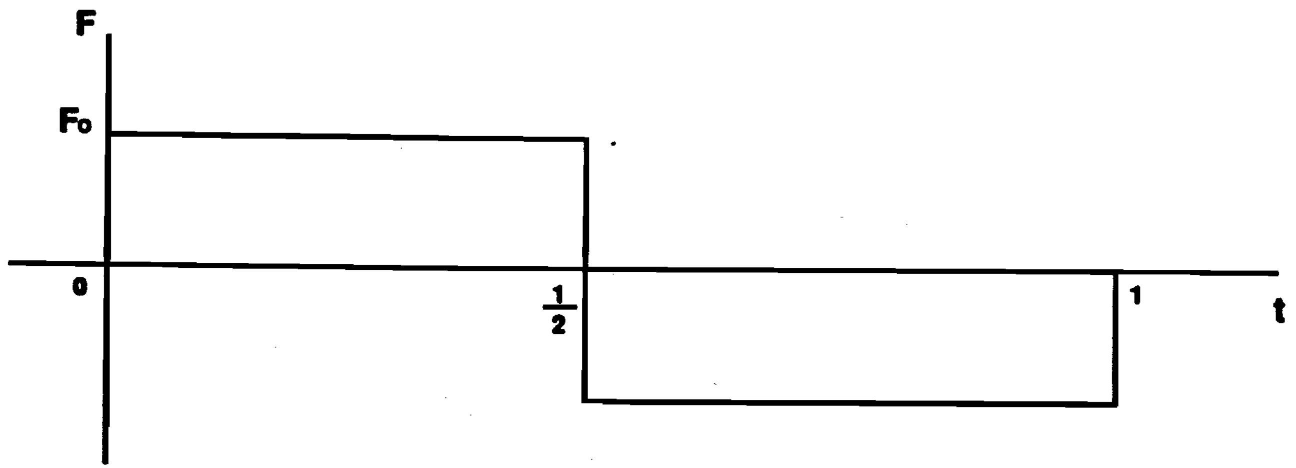
- **TWO MODES IN THIS SYSTEM**
  - **RIGID BODY MODE :** $\ddot{x}_1 = F$
  - **FLEXIBLE MODE :** $\ddot{x}_2 + 2\zeta\omega\dot{x}_2 + \omega^2 x_2 = f - F$

CONTROL SYSTEM DESIGN AND ANALYSIS

The "slew" motion of a two degree of freedom system is mathematically generated by using a step function of magnitude Fo, and the system is assumed to move 1 unit away from its reference point. The center of mass of the system stops at time equal to 1 second.

CONTROL SYSTEM DESIGN AND ANALYSIS

- **SLEW MOTION ($f = 0$)**

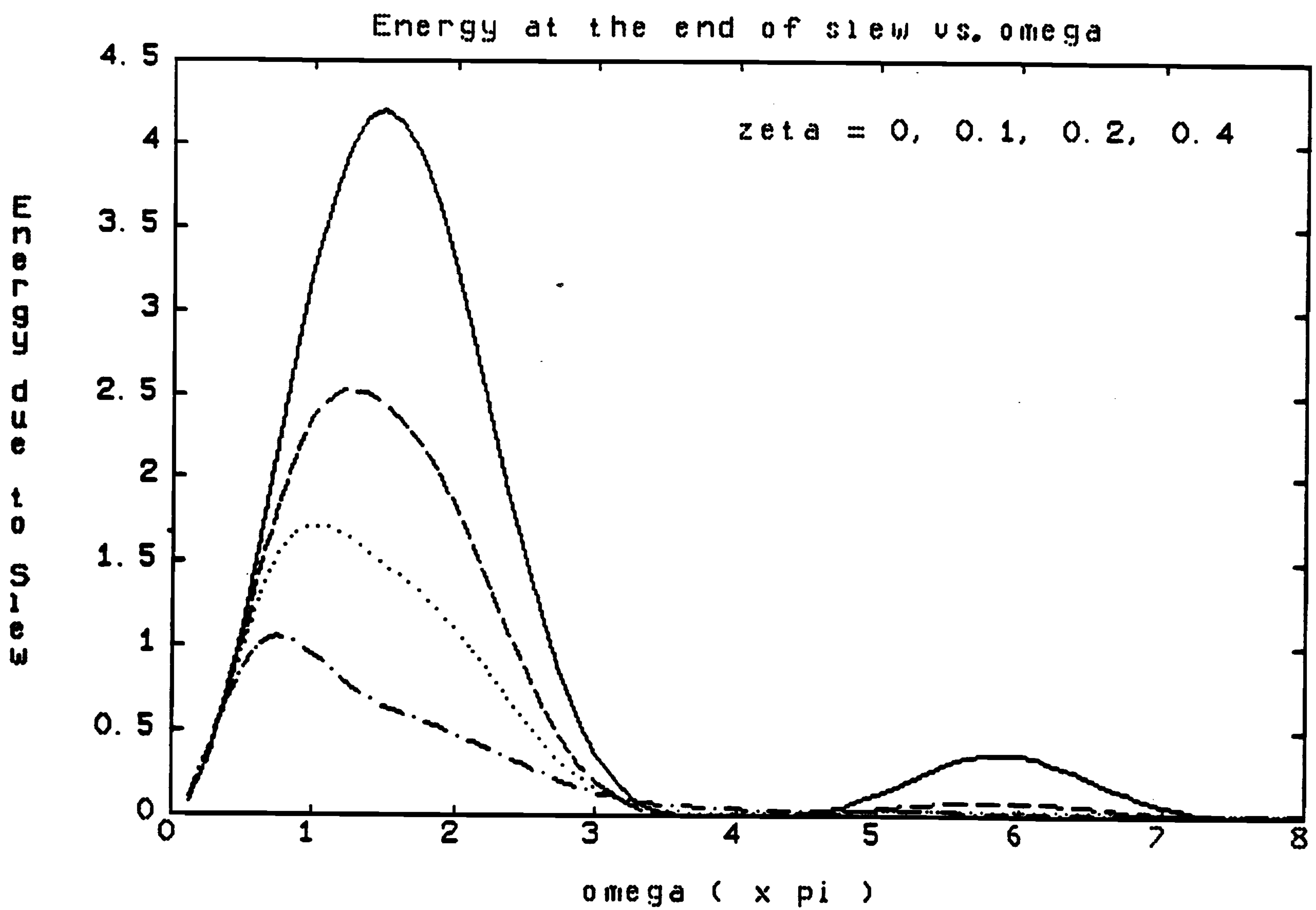


- **MOVE THE C.M. (CENTER OF MASS) 1 UNIT TO THE RIGHT IN $t = 1$ sec. ($F_0 = 4$)**
- $X_1(1) = 1$ **and** $\dot{X}_1(1) = 0$
- $\ddot{X}_1(t) = 0 \quad , \quad t \geq 1$
- **COMPUTE THE ENERGY STORED IN THE SYSTEM RIGHT AFTER THE SLEW MOTION**

ENERGY AT THE END OF SLEW VS. OMEGA

This chart shows the energy in the system at the end of the 1 second slew versus the system frequency for various values of damping. Note that at omega = 4 pi the energy is near zero for all values of damping.

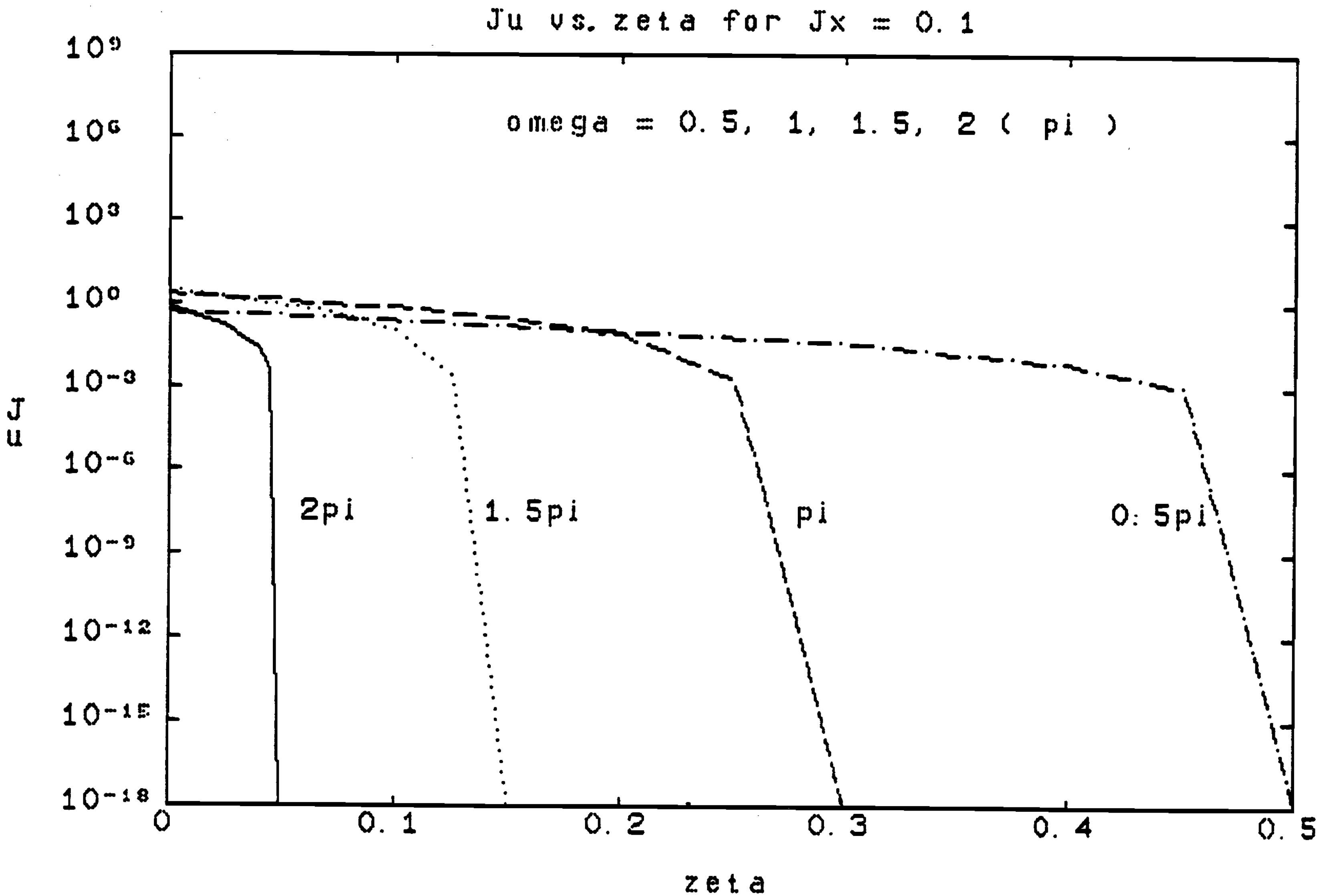
ENERGY AT THE END OF SLEW VS. OMEGA



Ju VS. ZETA FOR Jx = 0.1

This chart shows the control effort required to minimize the relative displacement of the two masses versus system damping for various values of system natural frequency. This shows a similar behavior to the results obtained from the SBL study.

Ju vs. ZETA FOR Jx = 0.1



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OBSERVATIONS

FOR THE CASES CONSIDERED :

- SUFFICIENT PASSIVE MODAL DAMPING TOGETHER WITH HIGH SPECIFIC STIFFNESS STRUCTURAL MATERIALS IS AN EFFECTIVE WAY TO ACHIEVE PERFORMANCE WHILE MINIMIZING CONTROL SYSTEM COMPLEXITY

- E.G., THE CONTROL EFFORT AND CONTROL FORCE REQUIRED TO ACHIEVE SBL PERFORMANCE DROPS CONSIDERABLY WITH DAMPING; FOR E/ρ = 760e+06 in. AT $\zeta \geq 9\%$ AND FOR E/ρ = 235e+06 in. AT $\zeta \geq 18\%$

- TRENDS FROM THE SBL CASES AND THE TWO DEGREE-OF-FREEDOM SYSTEM ARE SIMILAR