Final Report on Vibration Suppression in a Large Space Structure

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The collaboration between the Center for Systems Science and NASA Johnson Space Center took place during the period January 1985 to August 1987. The research proposal submitted by the Center to NASA concerned disturbance isolation in flexible space structures. The general objective of the proposal was to create within the Center a critical mass of expertise on problems related to the dynamics and control of large flexible space structures. A specific objective was to formulate both passive and active control strategies for the disturbance isolation problem. Both objectives were achieved during the period of the contract.

While an extensive literature exists on the control of flexible space structures, it is generally acknowledged that many important questions remain open at even a fundamental level. Hence, instead of studying grossly simplified models of complex structural systems, it was decided as a first step to confine attention to detailed and thorough analyses of simple structures.

1. Control of a Flexible Beam

The cantilever beam, the beam in free motion, and two beams attached to a rigid body capture most of the important features of the problems encountered in more complex structures. The first set of problems that was studied included passive control, active control, indirect adaptive control, and direct adaptive control of these simple systems.

1.1 Passive Control of a Cantilever Beam

The effect of passive elements on the dynamics of a cantilever beam was extensively studied during the summer of '85. The study was undertaken with the belief that simulations with discrete oscillators at various locations on the beam would provide valuable data regarding effective actuator and sensor locations and efficient control strategies. As an auxiliary benefit of these simulations, a considerable amount of information was also obtained on the extent to which vibration control of a beam can be effected through passive elements. In particular, the optimal oscillator frequency, the damping ratio, as well as the location of the oscillator were determined for the dynamic response of a cantilever beam in the presence of transverse loads.

A cantilever beam of unit mass-density was modeled on Microvax Station I using Finite Element Methods (FEM) with 20 nodes, 40 degrees of freedom and 80 state variables.

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The natural frequencies, mode shapes and the response of an unforced undamped cantiler to an impulse load were included in a preliminary report. The behavior of the beam with a discrete oscillator attached to it was also shown. The oscillator lowered the beam frequencies while attenuating the gain at the resonant frequencies. When the oscillator was damped, all oscilations slowly decayed to zero. The optimal dynamic response was determined by varying the oscillator location, frequency and damping ratio. While the behavior of continuous systems was studied with the cantilever beam as a specific case, parallel work on rigid body dynamics was carried out as well.

1.2 Active Control of a Cantilever Beam

Our studies on active control were guided by the results obtained for a beam with optimal passive controllers. In particular, the actuator and sensor were located at the optimal node (tip). It was found that the response (displacement) of the beam with active control (colocated rate feedback) is considerably improved when compared to that with passive control.

1.3 Identification of a Flexible Beam

In indirect adaptive control the unknown plant is first identified using input-output data and this in turn is used to determine the parameters of an adaptive controller. To identifying the plant, however, a specific model structure has to be assumed. For the identification of a flexible beam, an ARMA model was used. Assuming that the force $u(t)$ applied at one point of the beam is the input and the transversal displacement $y(t)$ of another point of the beam is the output, the ARMA model may be considered to describe the input-output behavior of the unknown system. The ARMA model may be described as follows:

$$A(q^{-1})y(t) = B(q^{-1})u(t)$$

where

$$\begin{aligned} A(q^{-1}) &= 1 + a_1 q^{-1} + \cdots + a_{n_A} q^{-n_A} \\ B(q^{-1}) &= b_1 q^{-1} + \cdots + b_{n_B} q^{-n_B} \end{aligned}$$

where q^{-1}= unit delay operator $(q^{-1}y(t) = y(t-1))$. The choice of the parameter estimation technique to be used wil depend, among other factors, on the type of data processing needed, e.g. batch (off-line) or recursive (on-line) methods, the cost function to be minimized, and the accuracy desired for the estimates.

The parameter estimation techniques utilized in this study were:

- Least Squares Method (Off-line and Recursive)
- Instrumental Variable Method (Off-line and Recursive)
- Prediction Error Method (Recursive)

The final choice of the specific method used will depend on the computational capacity available for identification and control purposes. Some of the advantages and disadvantages of each of the three method are shown in the following table:

Method	Advantage	Disadvantage
Least Squares	Easy Implementation and low computational requirements	Accuracy depends on correlation between noise (disturbance) and data vector.
Instrumental Variable	Minimize Correlation between noise and data vector choosing a suitable IV vector.	It is not possible to reach lower bounds for the covariance of the estimation error.
Prediction Error	Reaches lower bounds for the covariance of estimation error (Cramer -Rao bound in Gaussian Case)	High computational requirements.

The real beam was analyzed using FEM with 40 degrees of freedom and 80 state variables. This model was adequate to represent the modes of the beam accurately atleast up to th 4th mode. The motion of the beam was simulated using three-mode model (6 state variables) obtained from finite element analysis. Parameter estimates were found for different values of n_A and n_B, using the recursive least squares method.

2. Software Package

For the solution of problems in structural dynamics, a large number of finite element programs are commercially available. However, these programs have generally been designed for the analysis of the dynamical behavior of structures under given loading conditions rather than for the simulation of actively controlled structures. Further, it is our opinion that a considerable familarity with the details of the software will be necessary to efficiently implement and evaluate any proposed schemes for active control of space structures. These considerations led to our decision that an extensive computer software package developed by Professor Maewal should be suitably modified for our purposes.

The program is based on the finite element method with nodal displacements as the primary variables. The software is highly modular and contains all the elements (such as beams, plates and shells) that we foresee using in the near future. The dynamic analysis capabilities of the program include the calculation of (i) the natural modes and frequencies, (ii) the response of the structure to arbitrary time-dependent loads, and (iii) the response of the structure to arbitrary time-dependent loads with a user-defined control scheme that prescribes the actuator forces as a function of generalized nodal displacements.

3. Graduate Course on Flexible Structures

A graduate course entitled "Control of Flexible Structures," was given in the spring term (1986) jointly by Professors Narendra and Maewal and Dr. Annaswamy. The emphasis of the course was on the generation of models using the finite element method and the use of optimal control theory to control such models effectively. In particular LQG methods using observers and Kalman filters, modal cost analysis, internal balancing and aggregation methods were studied during this period. Since the dimension of the model obtained by using the finite element method is usually very large, the above methods were also used to obtain reasonable reduced order models of the system. As a result of these theoretical efforts, the members of the group are now well equipped to proceed to the next stage in the control of large flexible structures.

4. Flexural Motion of A Flat Plate

Around April 1986, it was decided to apply several of the theoretical methods discussed in the class to one complex practical problem, to enable all members of the group to gain hands-on experience. The control of a flexible plate was chosen for this purpose. Using the finite element method, a 512th order model and the corresponding mass and stiffness matrices were determined. A reduced order model of dimension 16 was then derived using the first eight modes of the model. Passive as well as active control methods were tried out on the latter model.

a) The Model:

Consider a flat plate occupying a domain Ω in the $x-y$ plane. The classical theory for the flexural deformation of the plate can be formulated in terms of the displacement $\mathrm{w}(x, y, t)$ normal to the $x-y$ plane. The flexural strain components are related to the transverse displacement according to

$$\kappa_{xx} = -\frac{\partial^2 \mathrm{w}}{\partial x^2} \tag{1a}$$

$$\kappa_{yy} = -\frac{\partial^2 \mathrm{w}}{\partial y^2} \tag{1b}$$

$$\kappa_{xy} = -2\frac{\partial^2 \mathrm{w}}{\partial x \partial y} \tag{1c}$$

The strain energy of the plate (assumed to be made of a linearly elastic, isotropic material) is given by

$$U = \frac{1}{2}D \int_\Omega \left[\kappa_{xx}^2 + 2\nu\kappa_{xx}\kappa_{yy} + \kappa_{yy}^2 + \frac{1-\nu}{2}\kappa_{xy}^2\right] d\Omega. \tag{2}$$

Here ν is the Poisson's ratio of the material of the plate, and D is its bending stiffness given by

$$D = \frac{Eh^3}{12(1-\nu^2)}$$

with E and h being respectively, the Young's modulus of the material of the plate and the plate thickness.

The equation of motion of the plate when it is subjected to transverse loads $p(x, y, t)$ can be written in the form of the variational statement

$$\int_\Omega \rho h\, \ddot{\mathrm{w}}\, \partial \mathrm{w}\, d\Omega + \partial U + \int_\Omega p\, \partial \mathrm{w}\, d\Omega = 0, \tag{3}$$

where ρ is the density of the material of the plate.

It is convenient to introduce scaled variables

$$\begin{aligned} (\overline{x},\overline{y}) &= (x,y)/\ell \\ \overline{\mathrm{w}} &= \mathrm{w}/h \\ \overline{t} &= \sqrt{\frac{Dh}{\rho\ell^2}}\cdot t \\ \overline{p} &= \frac{p\ell}{Dh^2} \end{aligned}$$

where ℓ is a typical length scale associated with the domain Ω. In the scaled system, the variational statement (3) becomes

$$\int_{\overline{\Omega}} \ddot{\mathrm{w}}\, \partial \mathrm{w}\; d\Omega + \partial V + \int_{\overline{\Omega}} p\, \partial \mathrm{w}\; d\overline{\Omega} = 0, \tag{4}$$

where $\ddot{(\cdot)} \triangleq \frac{\partial^2(\cdot)}{\partial \overline{t}^2}$, $\overline{\Omega}$ denotes the scaled version of the domain Ω and

$$V = \frac{1}{2}\int_{\overline{\Omega}} \left[\mathrm{w}_{,xx}^2 + 2\nu \mathrm{w}_{,xx}\mathrm{w}_{,yy} + \mathrm{w}_{,yy}^2 + 2(1-\nu)\mathrm{w}_{,xy}^2\right] d\overline{\Omega}. \tag{5}$$

Equations (4-5) are the basis of the scheme which has been used to discretize the problem.

b) Discretization:

Since the variational problem (5-6) contains second derivatives of the displacement w, the basis functions that can be used to obtain a discrete approximation must be at least of the class C^1 (i.e., the functions must be continuous and have continuous first derivatives). This criterion is satisfied by bicubic Hermite Polynomials over a rectangle; the choice of these functions, however, restricts the class of problems that can be analyzed to those with the domains Ω whose boundary lines are parallel to one of the coordinate axes.

Within a rectangular element, the bicubic Hermite polynomials interpolate a function in terms of the values of the quantities (w,$\mathrm{w}_{,x}$,$\mathrm{w}_{,y}$,$\mathrm{w}_{,xy}$) at the four nodes (corners) of the rectangle. Thus the element has 16 degrees of freedom, with four degrees of freedom per node.

Let q_e^i denote the vector of degrees of freedom of the i^{th} element, and $F^T(x,y)$ the vector of Hermite bicubics. Then with (5), the strain energy of an element can be written as

$$V^i \approx \frac{1}{2}\left(q_e^i\right)^T K_e^i q_e^i \tag{6}$$

where

$$\begin{aligned} K_e^i &= \int_{\overline{\Omega}_i^e} \left[F_{,xx}F_{,xx}^T + \nu\left(F_{,xx}F_{,yy}^T + F_{yy}F_{,xx}^T\right) + F_{,yy}F_{,yy}^T + 2(1-\nu)F_{,xy}F_{,xy}^T\right] d\overline{\Omega}, &(7)\\ \mathrm{w} &= F^T q_e^i &(8) \end{aligned}$$

and $\overline{\Omega}_e^i$ denotes the domain occupied by the i^{th} element. Similarly the discrete approximation to the first and the last term in (4) can be obtained by first noting that

$$\begin{aligned} \int_{\overline{\Omega}_e^i} \ddot{\mathrm{w}}\, \partial \mathrm{w}\; d\overline{\Omega} &\approx \left(\partial q_e^i\right)^T M_e^i q_e^i, &(9)\\ \int_{\overline{\Omega}_e^i} p\, \partial \mathrm{w}\; d\overline{\Omega} &\approx -\left(\partial q_e^i\right)^T f_e^i &(10) \end{aligned}$$

where

$$M_e^i = \int_{\overline{\Omega}_e^i} FF^T d\overline{\Omega}, \tag{11}$$

$$f_e^i = \int_{\overline{\Omega}_e^i} pF^T d\overline{\Omega}. \tag{12}$$

The element matrices K_e^i and M_e^i, and the load vector f_e^i can be assembled by using the usual finite element assembly process to obtain the discrete approximation to (4) in the familiar form

$$M\ddot{q} + Kq = f. \tag{13}$$

c) The Transfer Matrix:

Since the matrices M and K in equation (13) are respectively positive definite and positive semi-definite, it is well known that a linear transformation exists which reduces the equation to the modal form. In particular, if $\Phi r = q$, substituting in equation (13) and premultiplying both sides by Φ^T, we have

$$\Phi^T M \Phi \ddot{r} + \Phi^T K \Phi r = \Phi^T f$$

or

$$\ddot{r} + \Omega^2 r = \Phi^T f \tag{14}$$

where Ω^2 is a diagonal matrix whose elements $\omega_1^2, \omega_2^2, \ldots \omega_n^2$ are arranged in ascending order. The vector f is determined by the location of the actuators as well as the external input vector u; similarly, the output vector y is determined by the location of the position and velocity sensors. The $(n \times n)$ modal matrix Φ contains all the relevant information about the modes, since each column ϕ_k corresponds to one mode (the k^{th} mode) of the system.

From the above it is clear that each actuator affects all the modes and that the latter in turn affect all the outputs. If there are a actuators, a columns of Φ^T are retained; similarly if there are s sensors, s rows of Φ are retained. Denoting these by the matrices Φ_a^T and Φ_s respectively, the $(s \times a)$ transfer matrix of the overall system (of order $2n$) is given by

$$\Phi_s (s^2 I + \Omega^2)^{-1} \Phi_a^T \tag{15}$$

if only position sensors are used. If, however, only velocity sensors are used, the transfer matrix is given by

$$s\Phi_s (s^2 I + \Omega^2)^{-1} \Phi_a^T. \tag{16}$$

In general, since both position and velocity sensors are used, the transfer matrix is a linear combination of the two given in (15) and (16).

d) The Reduced Order Model:

When the transfer matrix of the system is obtained as described above, its characteristic polynomial is of degree $2n$. Since n is generally large (256 in the case of the plate in the simulations), this poses a serious problem computationally. So, the physical model obtained by the finite element method is successively reduced for both evaluation and design purposes. This process is briefly described in this section.

The principal idea behind the reduction procedure is that only a small number of the modes of the system be retained in the final model. The vector r and matrix Ω^2 are denoted as

$$r = \begin{bmatrix} r_c \\ r_R \end{bmatrix}; \qquad \Omega^2 = \begin{bmatrix} \Omega_c^2 & 0 \\ 0 & \Omega_R^2 \end{bmatrix} \tag{17}$$

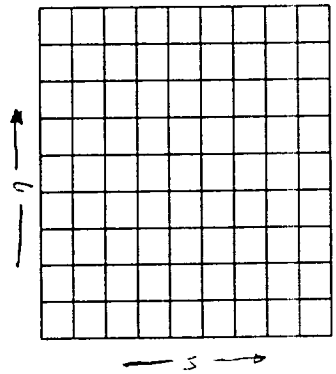

Figure 1

so that r_c denotes the controlled modal vector of dimension $n_1(< n)$ and r_R the residual vector of dimension $(n - n_1)$. The transfer matrices given by (15) and (16) are modified as follows:

Let

$$\Phi_s = [\Phi_{c,s}, \ \ \Phi_{R,s}] \qquad \text{and } \Phi_r = \begin{bmatrix} \Phi_{c,a}^T \\ \Phi_{R,a}^T \end{bmatrix}. \tag{18}$$

Assuming that $\Phi_{R,s}$ and $\Phi_{R,a}$ are sufficiently small, they are replaced by null matrices to yield the transfer matrix of the reduced order model (of order $2n_1$) as

$$\Phi_{c,s}(s^2 I + \Omega_C^2)^{-1} \Phi_{c,a}^T \tag{19}$$

for position sensors and

$$s\Phi_{c,s}(s^2 I + \Omega_C^2)^{-1} \Phi_{c,a}^T \tag{20}$$

for velocity sensors. For the plate under consideration, eight modes were retained in the reduced order model, yielding a sixteenth order system.

4.1 Vibration Control of Plate

Following the procedure outlined in section 2, eight modal vectors of dimension 256 were first determined corresponding to the eight principal modes. These in turn were truncated to yield $(8 \times a)$ and $(s \times 8)$ matrices for the inputs and outputs respectively. The s rows of the second and the a columns of the first can be chosen from two hundred and fifty six possible values and depend upon the location of the sensors and actuators. Using this model, both passive and active control of the plate were attempted. The results obtained were included in a progress report submitted to NASA.

Figure 1 shows a 10×10 grid representation of the plate. The boundaries of the plate are clamped and there are 64 free nodes. At each free node, the displacement w, the slope w_ς in the ς direction, the slope w_η in the η direction, and the second mixed derivative $w_{\varsigma\eta}$, are four independent variables, making 256 variables in all. Along with their time derivatives, they constitute the 512 state variables of the system.

a) Passive Control of Plate: Of the various methods known for disturbance isolation, the simplest involves the use of passive elements. Auxiliary systems, which are mass-spring-dashpot combinations, are attached at various nodes and their parameters tuned so that the effect of a disturbance is damped out relatively rapidly.

The optimal parameters of the auxiliary system were chosen based on the experience gained with the similar problem involving a beam. This involved choosing a mass which is approximately one tenth of that of the plate and locating it at the point at which the first mode has a maximum. The spring constant was chosen so that the natural frequency of the auxiliary system coincided
ant frequency of the plate. The auxiliary system serves to dissipate energy and
ut the vibrations in the system due to initial conditions.

tuators and Sensors: The main aim of this study was to determine the location of
ensor-actuator combination to yield the optimal response. Due to the symmetry
y ten possible positions need to be considered and these are indicated in Figure
nsor is located at a node (say 5,3 which corresponds to location 9) and a force
the negative of the velocity is applied at the same point. It is well known that
e introduction of dynamic damping into the system so that the overall system
mptotically stable. The effect of this control on both the position and velocity
easured. An exhaustive search was made by performing 200 experiments. In all
gain of $k = -1$ was used. These correspond to the cases where the colocated
ator are at locations 1,4, and 10 respectively. The initial condition of the plate

$$\begin{aligned} w(0) &= 16\varsigma(\varsigma\text{-}1)\eta(\eta\text{-}1) \\ w_\varsigma(0) &= 0 \\ w_\eta(0) &= 0 \\ w_{\varsigma\eta}(0) &= 0 \end{aligned}$$

r the specified initial conditions, the best location of the actuator-sensor com-
ation 4 (corresponding to $\varsigma = 5, \eta = 5$). In such a case the position as well as
at all points are damped out rapidly. In contrast to this, when the actuator-
ion is at 1 (i.e., 2,2) the best response is at the same node. At locations 2,3
equency components persist, while along 5,8 and 10, high frequency signals are
nt. Finally, when the actuator and sensor are located at 9, the damping effect
satisfactory.

ons also indicate that the best location for the sensor-actuator combination is
this case, several other experiments were carried out using different feedback
that the optimal responses both in terms of speed as well as accuracy are
es of k around -2.3.

he results obtained by the active control described in this section to those
ive control in the previous section, it is clear that the former is superior both in
ed of response as well as its effect on the displacements at all the other nodes.
noting that the optimal location of the actuator-sensor combination will also be
large extent by the nature of the anticipated disturbance (in the present case,
cement).

ge of colocation is that the stability properties of the overall system are improved
e number of sensor-actuator pairs. Even in such a multivariable case, feeding the
velocity sensor to the actuator at the same node can be theoretically shown to
otic stability. The theory can also be extended to include more general feedback
sors and the actuators.

assuring global asymptotic stability, colocated actuators and sensors are simple

to are quite robust. However, as mentioned earlier, their effectiveness is over a
wi nd hence is quite limited.

c) \ctuators and Sensors: As mentioned in the introduction, practical constraints
m se of actuators and sensors at the same point. Further, in many situations, the
lo trol (LAC) resulting from colocation may not be adequate to meet specifica-
tic es high authority control, generated using estimates of the states of the system
to criterion function, may be desirable. It should also be mentioned here that
hi ntrol (HAC) generated by optimal control techniques can also be used when
th sensor are colocated. The essential difference between HAC and LAC is that
wl back is used in the latter case, the control input is computed using principles
of l in the former to achieve a specific performance objective over some portion of
th 1 this section typical examples of HAC when actuators and sensors are located
at are presented.

rs and sensors are not colocated and direct feedback is used, there is a distinct
pc ability, particularly when the feedback gain is high. Hence, caution must be
ex rectly feeding back the signals to the actuators, if stability is to be guaranteed.

d) ack: In this case the measured output is processed dynamically before being
fe tuator. The normal procedure is to use an observer to generate an estimate
$\hat{x}$ f the system and use $\hat{x}$ in turn to generate a control input $u(\cdot)$ to the flexible
st timal control minimizes a quadratic performance index of the form

$$\int_0^\infty x^T Q x + u^T R u dt$$

w i-definite matrix. In particular, the displacements of all the nodes are included
in ce index. Experiments were performed by locating a velocity sensor at (3,3)
ai at the same node. The resulting performances when colocated rate feedback
ai al feedback which depends on $\hat{x}$ were used, were compared.

he stability of the system, colocated sensors and actuators should be used
w e. However, when high performance is needed, quadratic optimization theory
ap rticularly effective. In high dimensional problems, however, the effect of spill
ov deled parts of the system have to be studied before implementing the control.

the flexural vibrations of a plate, while seemingly a simple problem, has several
in s to it. It provided the group at Yale an opportunity to get acquainted with
m culties encountered in problems of high dimensionality. The variety of criteria
th l for the evaluation of the performance, as well as the different approaches
th control, force the designer to limit his choices at every stage. Based on the
ex l while simulating this problem, the Yale group became ready to deal with the
cc of a model whose dimension is approximately one thousand.

5 ance Isolation in a Space Station Model

D d June 1, 1986 to October 1987, the technical team members including Pro-
fe and Maewal, Dr. Annaswamy and Mr. Khamkoon, Mr. Duarte, and Mr.

Be the problem of developing active control strategies for disturbance isolation
in ion. Ms. Cunningham, the systems manager of the Center assisted in the
pr lies.

Th oncerning the space station was received from NASA Johnson Space Center.
ra chematic diagram of the proposed NASA space station with dual keel configu-
sir station consists of 128 nodes with 732 active degrees of freedom. To facilitate
be , a new schematic representation was prepared with all nodes properly num-
N nation concerning the mass and stiffness matrices M and K were provided in
an ut 4 format on a tape. After a considerable interval of time the eigenvalues
24 ding eigenvectors were also received. These included 6 rigid body modes and
we nodes. From the M and K matrices, the first 70 eigenvalues and eigenvectors
Th the Center using a subspace iteration scheme developed by Professor Maewal.
pr ared with those received from NASA and were found to be identical. A new
ten to convert the differential equations

$$M\ddot{q} + Kq = f$$

to r form

$$\begin{aligned} \dot{x} &= Ax + Bu \\ y &= Cx \end{aligned}$$

so variables correspond to the modal amplitudes and velocities in this represen-
tat

50 response of the space station was studied for momentary disturbances at node
Ex effect of the behavior of elastic modes, the rigid body modes were neglected.
at ier simulations were carried out to determine the sensitivity of the response
ra due to initial displacements at other nodes. The response due to colocated
ma studied for different values of gains. The effect of control was found to be a
ma he sensor-actuator pairs were located at node 236 where the first mode has a
str ude. The responses due to sinusoidal excitation at different nodes were also

tio or accomplishments of this period concerned the three dimensional representa-
dev rough 55. These had been received earlier from NASA. But the representations
hav Center proved far superior. They provided greater insights into the modal be-
the e station. It became clear from these representations that only modes beyond
tly affect the habitat.

fro ig disturbances due to crew members (one man forceful soaring) was received
Th tape during this period. This in turn was transferred to the Yale model.
am ra of the given functions were computed and were found to have maximum
vicinity of 1 Hz.

be necessary information regarding the space station and the disturbances had
Ho emphasis during the second stage was on generating active control strategies.
the ormation concerning the number of actuators and sensors that could be used,
sta well as their location and the criteria by which the performance of the space
out judged, had not been received from NASA, simulation studies were carried
am using reasonable assumptions.

inil ; of simulations the free vibrations of the space station was studied. An
noc ıt was introduced in node 236 and the corresponding response at the same
sta [o study the effect of approximations various evaluation models of the space
difl ,5, 10, 20 and 40 modes were simulated. Not surprisingly, the response in the
inc cated that the accuracy of the model improved as more and more modes were
fiv ed feedback was used at node 236 and the experiments were repeated for the
Co ıer experiments were carried out using different gains in the feedback loop.
wa k was also attempted at other nodes besides node 236. Colocated feedback
wh nd to be most effective when the actuator-sensor pair is placed at node 236 de has maximum amplitude.

int orced vibrations were studied next. Disturbances at various frequencies were
exp 236 and their effects were observed at different locations on the station. These
Wl repeated with colocated feedback at node 236 with a feedback gain of unity.
4% ance is at node 236 the effect at the habitat was found to be approximately de of the disturbance without control and .2% with control.

Th listurbances were introduced which may be generated by crew member motion.
26(disturbances introduced at node 288 was observed at nodes 60, 65, 250, 255,
wit 298, 302, 304, 2060, 2250, 2255, 2288, and 2302. The experiment was repeated e feedback at node 236 and node 288 with a feedback gain of 200.

Th ced order LQ controller was used in the place of a colocated rate feedback.
is ı while sensor and actuator are located at the same node, the control input
an(o rate feedback but obtained by minimizing a quadratic performance index
we ; all the modes were accessible. The responses at various nodes in the habitat
ba th state feedback with a reduced order controller located at node 236 and
eva twenty flexible modes. In all cases, the simulation was carried out using an
Th with 38 modes and the Q and R matrices were chosen to be unit matrices.
a s as repeated with the reduced order controller based on modes 25-44. Finally,
an(as applied at node 288 using a reduced order controller based on modes 25-44
acc = [1]. Assuming that the location that is of interest for obtaining minimum
6 × 3, it was observed that with such a feedback, a steady state acceleration of obtained.

Wi ol, the acceleration magnitude at the habitat was of the order of $1.56 \times 10^{-6}g$.
cor te feedback this was reduced to approximately by a factor of 2. With an LQ
use ı further reduced by a factor of 25 to $6.5 \times 10^{-8}g$. In all cases, node 298 was he various acceleration responses.

rec pts were made to determine appropriate controls for the space station using
the dels. The evaluation model of the space station includes 10 modes, i.e. of
res ıg a force actuator and velocity sensor at node 236 as the input and output
spa feedback using a linear quadratic regulator was implemented to control the formance indices of the form

$$\int_0^\infty \left[x^T Q x + u^2\right] \, dt$$

wi were used and the response was observed at node 236 in the habitat.

a reduced order controller was built based on the first m flexible modes, for m = th an initial displacement at 236, the velocity response at 236 was observed usir k. It is seen that as m increases, the quality of feedback also increases. Next, assu tates were not accessible for measurement, an mth order observer was built to e: es and the state estimates were fed back.

r e of an LQ controller when the entire state is accessible is obviously the best that d. This is followed closely in performance by an LQ controller using a full state siderable care has to be taken when reduced order observers are used in the cont: instability may arise. Using rate feedback in addition to an LQ controller impr nt performance to a certain extent. Finally, the problem of damping higher mod controller is not straightforward. The number of actuators and sensors that are u ocation become quite critical.

(celeration of less than $10^{-7}g$ in the habitat due to crew disturbance appears poss ore experimentation with several actuators and sensors which are judiciously loca ded.