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LARGE SPACECRAFT POINTING AND SHAPE CONTROL

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This presentation summarizes work performed under contract to the Flight Dynamics Laboratory (FIGC), Air Force Wright Aeronautical Laboratories. The contract, entitled Large Spacecraft Pointing and Shape Control (LSPSC), was initiated in September 1983. Technical work was completed in August 1986.

The major objectives and the scope of the study are listed below. The overall objective was the development of control algorithms that allow the concurrent operation of slewing, pointing, vibration, and shape control subsystems. This objective is important for near-term space surveillance missions that require the rapid-retargeting and precise pointing of large flexible satellites. The success of these missions requires the design and concurrent operation of the various interacting control subsystems.

LSPSC PROGRAM

MAJOR OBJECTIVES

- DEVELOP TECHNIQUES NECESSARY TO DESIGN A CONTROL SYSTEM TO SLEW AND PRECISELY SETTLE A LARGE FLEXIBLE ANTENNA SPACECRAFT
- EXPLORE THE INTEGRATION OF AND INTERACTIONS BETWEEN THE DIFFERENT CONCURRENTLY OPERATING CONTROL SUBSYSTEMS ONBOARD

  CONTROL SUBSYSTEMS:

  - SLEW
  - POINT/TRACK
  - VIBRATION SUPPRESSION
  - SHAPE

- IDENTIFY GAPS IN THE TECHNOLOGY REQUIRED FOR CONTROLLING A LARGE ANTENNA SPACECRAFT

SCOPE

- AN UNCLASSIFIED THEORETICAL STUDY, NOT A SYSTEMS STUDY
- LEVEL OF DETAIL CONSISTENT WITH A PREDESIGN EFFORT
- SUFFICIENT REALISM TO GUARANTEE THE RELEVANCE AND ACCURACY OF MAJOR CONCLUSIONS

The program was conducted in two phases. Phase I was primarily mathematical model development, while Phase II was primarily control development.

LSPSC PROGRAM TASKS

PHASE I

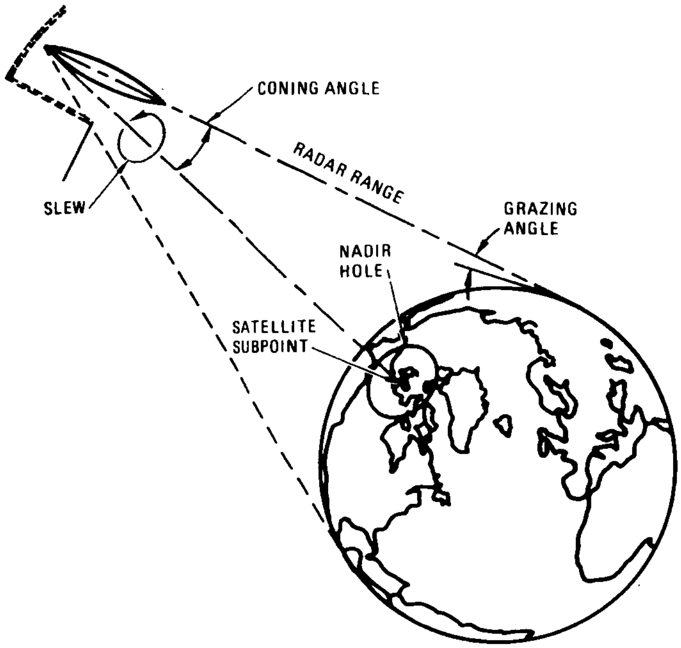
- REVIEW THREATS AND MISSIONS
- DEFINE MATHEMATICAL MODEL OF ANTENNA SPACECRAFT
- DEFINE CONTROL REQUIREMENTS AND GOALS
- EVALUATE EXTERNAL AND INTERNAL DISTURBANCES
- EVALUATE ACTUATORS/SENSORS FOR LSS CONTROL APPLICATIONS

PHASE II

- REVIEW LSS CONTROLS LITERATURE AND ON-GOING PROGRAMS
- DEVELOP CONTROLLERS USING HEURISTIC LOCATIONS OF ACTUATORS/SENSORS FOR:
  - SLEWING
  - POINTING/TRACKING
  - VIBRATION SUPPRESSION
  - SHAPE CONTROL
- DETERMINE OPTIMAL LOCATIONS OF ACTUATORS/SENSORS AND REPEAT CONTROLLER DEVELOPMENT
- EVALUATE ROBUSTNESS OF BOTH CONTROLLERS
- EXAMINE THE INFLUENCE OF PASSIVE DAMPING

The baseline generic mission for the study was a tactical surveillance mission for a space based radar. The satellite was to be in a 5600 n.mi. polar orbit and have a chase mode slew rate of 2 deg/sec. Both a coning mode of operation and a star-scan mode were examined initially. Due to the very high momentum requirements of a coning mode, the staring mode was chosen for the control development phase. For the staring mode, target acquisition and target tracking were required. A slow reorientation was required at least once per orbit. An occasional fast slew was required for surveying multiple targets.

MISSION GEOMETRY AND REQUIREMENTS



SYSTEM PARAMETERS	
ORBIT ALTITUDE	5,600 N.MI.
ORBIT PLANE	POLAR
STRUCTURE	
• TYPE	DISH ANTENNA
• DIAMETER	100M
• SLEW RATES	2 DEG/SEC (0.8 DEG/SEC)
OPERATING FREQUENCY	10 GHz (3 CM)
CONING ANGLE	22.4 DEG
DERIVED PARAMETERS	
ANTENNA DIRECTIVITY GAIN	80 dB
ANTENNA BEAMWIDTH	0.02 DEG
ACCESS RADIUS	4,060 N.MI.
INSTANTANEOUS COVERAGE·	
• MAXIMUM LENGTH	460 N.MI.
• OPERATIONAL LENGTH	170 N.MI.*
• WIDTH	2.7 N.MI.
SATELLITE SUBPOINT VELOCITY	3,600 KTS
MAXIMUM RADAR RANGE	8,360 N.MI.
OPERATIONAL RADAR RANGE	8,065 N.MI.*
NOMINAL SEARCH RATES	19,300 N.MI.^{2}/SEC* 8,700 N.MI.^{2}/SEC
PRIME POWER	20-50 KILOWATTS

*5 DEGREE GRAZING ANGLE MINIMUM

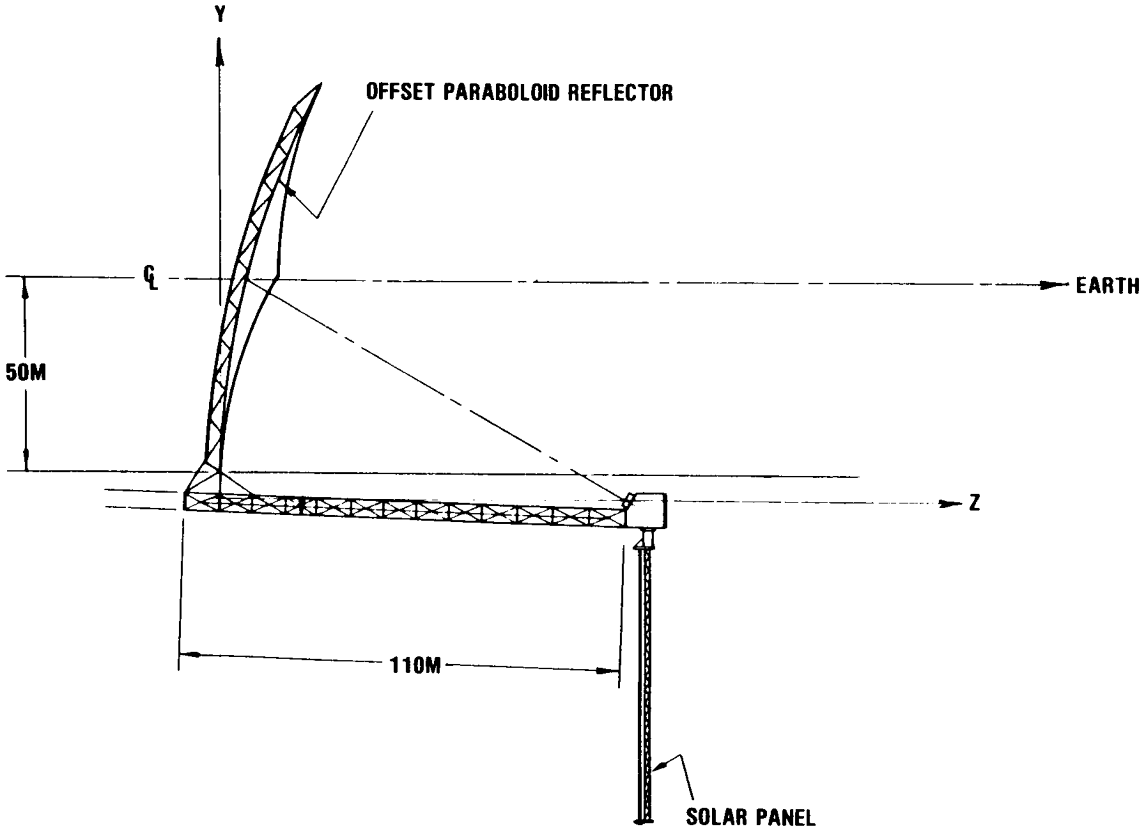
The table below summarizes pointing and surface accuracy requirements for the generic mission. The requirements for X-band operation were chosen in order to create the most challenging control problem.

SPACECRAFT POINTING REQUIREMENTS

	L-Band	S-Band	X-Band
Band			
• Wavelength (frequency)	24 CM (1.25 GHz)	10 CM (3 GHz)	3 CM (10 GHz)
• Gain	64 dB	72 dB	80 dB
• Beamwidth	0.1° (1,750μr)	0.04° (700μr)	0.02° (350μr)
Antenna pointing accuracy			
• Threshold	0.01° (175μr)	0.004° (70μr)	**0.002° (35μr)**
• Goal	0.001° (17.5μr)	0.0004° (7μr)	**0.0002° (3.5μr)**
Feed angular orientation			
• Threshold	0.01° (175μr)	0.004° (70μr)	**0.002° (35μr)**
— Lateral movement/120M	2 CM (0.08λ)	0.8 CM (0.08λ)	0.4 CM (0.13λ)
• Goal	0.001° (17.5μr)	0.0004° (7μr)	**0.0002° (3.5μr)**
— Lateral movement/120M	0.2 CM (0.008λ)	0.08 CM (0.008λ)	0.04 CM (0.013λ)
Search mode slew rate	5.0°/sec	1.2°/sec	0.8°/sec
Tracking mode slew rate	0.004°/sec	0.004°/sec	0.004°/sec
Tracking mode pointing accuracy	0.0025° (44μr)	0.001° (18μr)	0.0005° (8.8μr)
Surface accuracy			
• Surface tolerance (RMS)	1.2 CM (0.05λ)	0.5 CM (0.05λ)	0.15 CM (0.05 λ)
• Surface accuracy (absolute)			
— Threshold	1.7 CM (0.07λ)	0.7 CM (0.07λ)	**0.35 CM (0.10λ)**
— Goal	0.17 CM (0.007λ)	0.07 CM (0.007λ)	**0.035 CM (0.01λ)**

The spacecraft model itself was chosen to be a geodetic-truss, 100-meter diameter, offset-feed antenna.

SPACECRAFT MODEL — OFFSET CONFIGURATION



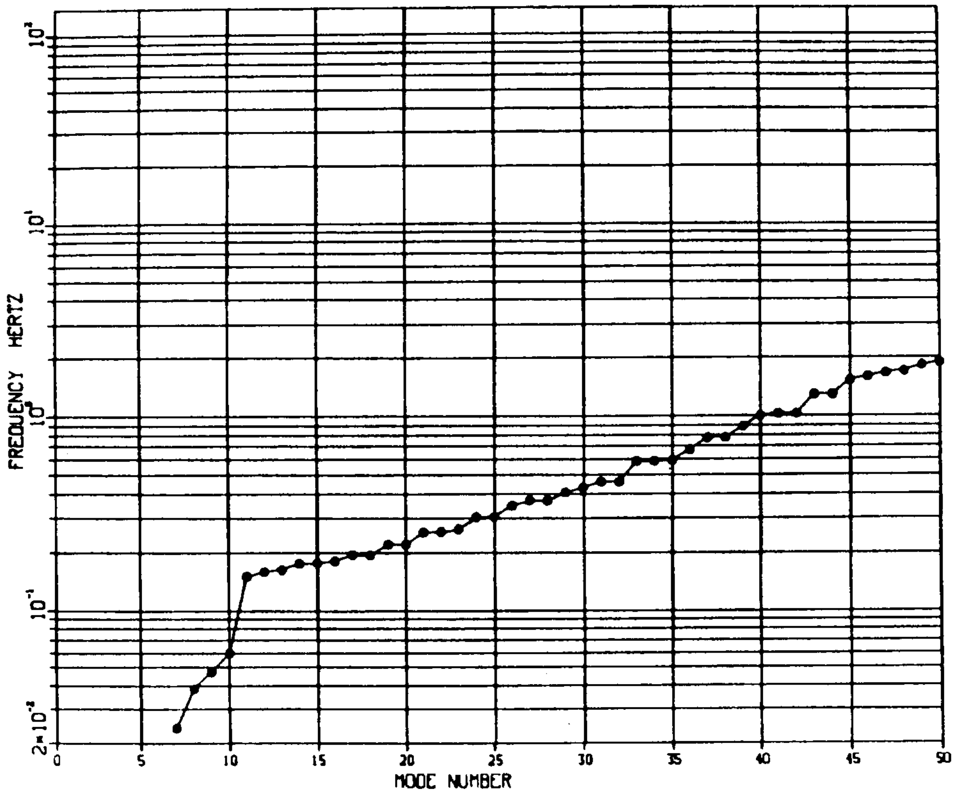
An extensive parametric study of unattached (free-free) truss reflectors was conducted. The goal was to investigate and provide data concerning low-frequency truss-reflector behavior. A strawman objective was to achieve a reflector with a first-mode frequency on the order of 0.1 Hz. This objective could not be achieved using standard geo-truss design practices to obtain a reasonable design. Consequently, a reasonably designed 100-meter reflector was chosen. The reflector's lowest free-free modal frequency is 1.7 Hz.

PARAMETRIC STUDY: UNATTACHED REFLECTOR DISH

	Trial No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
	E (mpsi)	20	15	10	10	2	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	34	34
Independent variables	No. of bays	12	12	12	16	20	20	24	28	20	16	20	16	20	20	20	20	20	16	20	16	12	12
	Strut angle (degree)	30	30	30	24	24	24	24	24	15	15	12	12	10	8	24	24	24	24	24	24	40	40
	F/D_p	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.8	0.8	1.0	1.0	1.2	1.5	0.5	0.5	0.8	0.8	0.8	0.8	0.5	0.5
	Diameter (m)	50	50	50	50	50	50	50	50	50	50	50	50	50	50	80	100	100	100	150	150	50	100
	Truss depth (m)	1.1	1.1	1.1	0.4	0.3	0.3	0.3	0.3	0.1	0.1	0.1	0.1	0.1	0.03	0.5	0.7	1.2	1.4	1.7	2.1	2.0	3.9
	Diagonal length (m)	3.2	3.2	3.2	2.3	1.8	1.8	1.5	1.3	1.7	2.2	1.7	2.1	1.7	1.7	2.9	3.6	3.6	4.6	5.5	6.8	3.6	7.3
	Tube diameter (cm)	2.2	2.2	2.2	2.7	2.3	2.3	2.0	1.8	2.2	2.6	2.2	2.5	2.2	2.2	3.1	3.6	3.6	4.2	4.8	5.5	2.4	3.8
	Weight (kg)	1,193	1,193	1,193	2,040	2,234	2,234	2,412	2,570	2,139	1,956	2,117	1,932	2,100	2,095	4,733	6,768	6,587	6,047	12,721	11,746	1,236	3,945
	Package diameter (cm)	282	282	282	448	481	481	510	535	472	441	470	438	468	468	653	754	743	693	967	905	283	442
	Package height (cm)	494	494	494	357	285	285	238	203	272	340	269	336	267	265	456	571	563	705	845	1,057	536	1,071
	1st vib mode (Hz) (free-free)	1.66	1.44	1.17	0.604	0.233	0.498	0.422	0.365	0.256	0.306	0.196	0.231	0.157	0.118	0.316	0.254	0.332	0.406	0.223	0.271	3.43	1.70

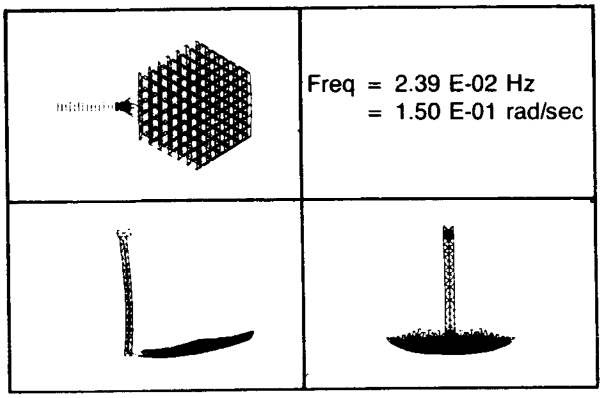
A quite flexible feed-boom was coupled to the reflector. A simulated solar array and a feed-bus structure were attached to the end of the feed-boom opposite the reflector. The lowest frequency of vibration of the vehicle is 0.024 Hz. There are 33 elastic modes below 1 Hz. The flexible feed-boom was chosen to facilitate technological development by creating a challenging control problem.

VIBRATION MODES

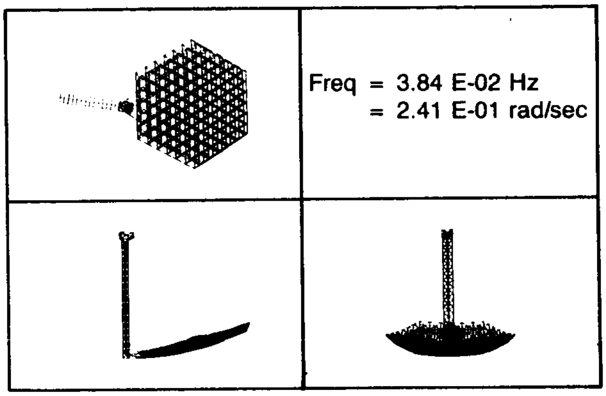


The lowest 4 elastic modes are significantly excited by maneuvering disturbances. The first elastic mode, mode 7, is primarily boom bending in the Y-Z plane. Mode 8 is primarily a torsion mode of the feed-boom. Mode 9 is primarily a boom bending mode in the Y-Z plane coupled with solar array bending. Finally, mode 10 is primarily a reflector rocking mode with boom bending in the X-Z plane.

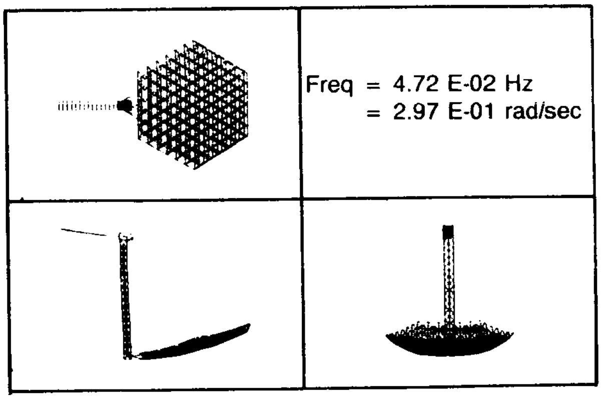
DEFORMED SHAPE — MODE 7



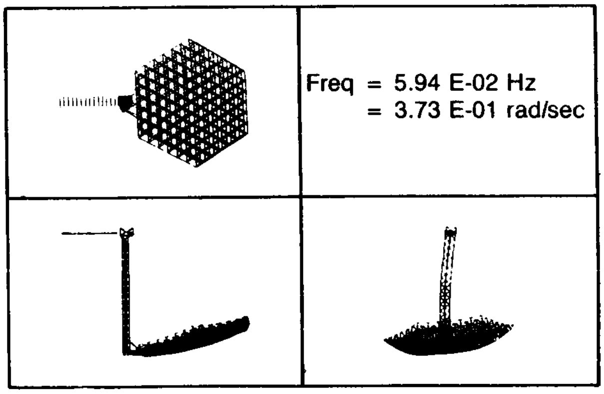
DEFORMED SHAPE — MODE 8



DEFORMED SHAPE — MODE 9



DEFORMED SHAPE — MODE 10



Conclusions of the structural model development task are summarized below.

STRUCTURAL MODEL DEVELOPMENT SUMMARY

- Geodetic-truss reflector was chosen for:
  - Ability to accommodate fast slewing maneuvers
  - High achievable surface accuracy
  - High failure & attack survivability (structural redundancy)
- Parametric studies of the reflector show that very low natural frequencies are not inherent (even for 100-meter diameter reflectors)
- A "reasonably designed" 100-meter diameter (1.7Hz) reflector was chosen as representative of this class of reflectors
- An offset antenna configuration was chosen over center-fed because it offers a more challenging control problem
- The truss-boom's bending stiffness was chosen to be small (mode 7 frequency = 0.024 Hz) to provide a challenging slewing/vibration/pointing/shape control problem

Many disturbances, both internal and external, affect the spacecraft. The table shows that by far the dominant disturbances are due to the slewing maneuvers. The effect of gravity gradient torques is comparable to that of pointing/tracking torques for this spacecraft with a flexible boom.

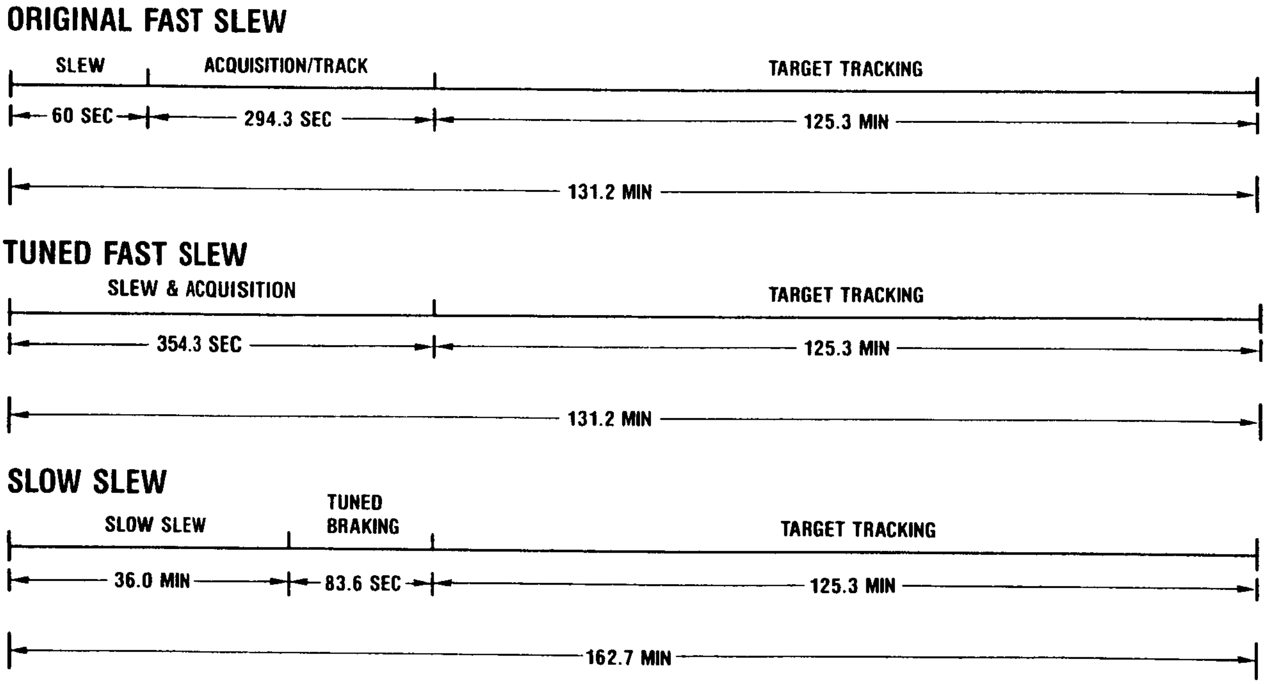
LSPSC FAST-SLEWING DISTURBANCE DOMINATES ALL OTHER DISTURBANCES

Disturbances	LOS Error/LOS* goal
Thermal gradient	<<1.0
Solar pressure	<1.0
Gravity gradient	1.1 - 4.0
Pointing/tracking torques (CMGs)	0.1 - 7.2
Reboost (RCS)	490
Slow slewing (CMGs)	500
Fast slewing (RCS)	56 - 39,000

*Line of sight (LOS).

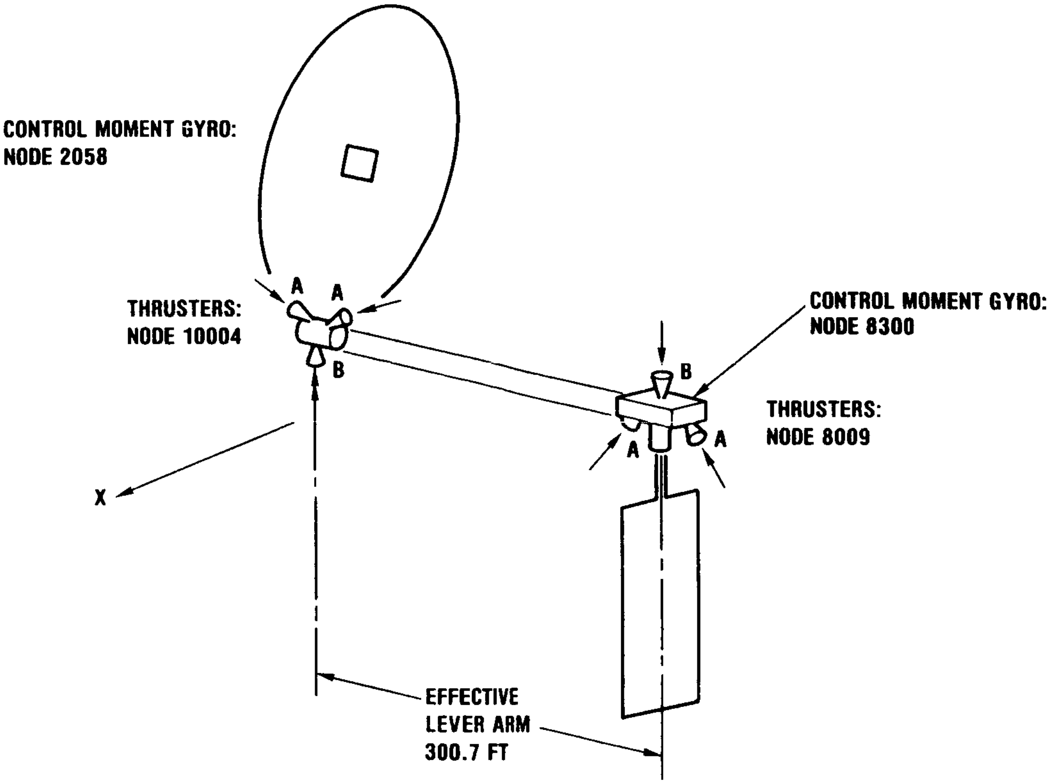
All the generic orbit scenarios considered include a slew and target acquisition phase followed by an operational phase in which a target is tracked. RCS-thrusters were used to perform the fast slewing maneuvers, while CMGs were used to perform the slow slewing and target tracking maneuvers. In the case of a fast slewing maneuver, settling of vibrations must be completed during the acquisition phase. To reduce the elastic excitation following the fast slewing maneuver, the RCS pulses were tuned to periods of the lower modes.

ORBIT SCENARIO SEQUENCES
(Not to Scale)



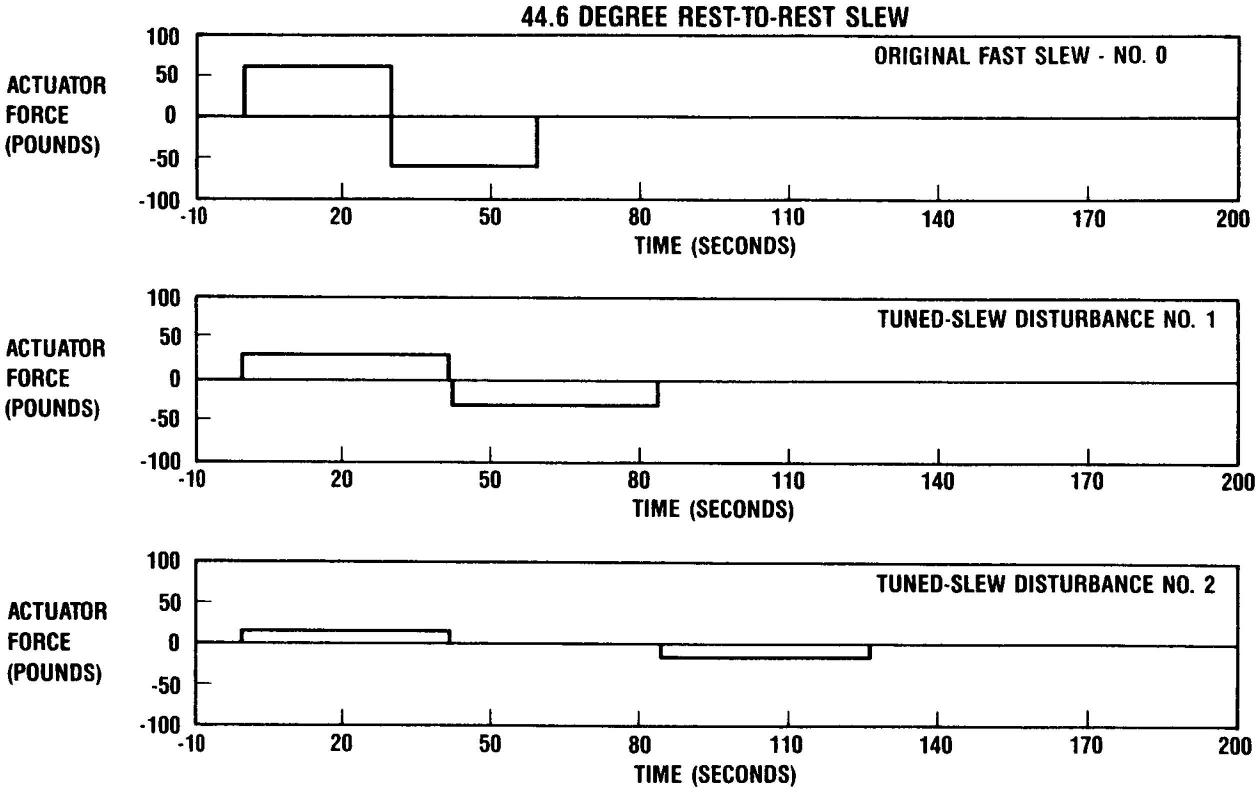
Locations of the RCS-thrusters and the CMGs are shown below.

LOCATION OF SLEWING DEVICES



As mentioned, the fast slewing torque profile was tuned to the periods of modes 7 and 9. Two "tuned" torque profiles were compared to an original profile.

FAST-SLEW DISTURBANCES



Tuning the slewing pulses is seen to significantly reduce the post-slew dynamic response. This is important as it reduces the vibration control torques required to settle the vehicle. Tuned slew number 1 was chosen as a baseline.

COMPARISON OF POST FAST-SLEW EXCITATION LEVELS CLEARLY SHOWS THE BENEFITS OF TUNING

	PERFORMANCE (PEAK NEAREST T = 130 SEC)		
DESCRIPTION	TOTAL LOS ERRORS (ARC-SEC)	RMS SURFACE ERRORS (10^{-3} IN.)	PATH LENGTH Δ (10^{-3} IN.)
ORIGINAL FAST SLEW BANG/BANG (29.6/29.6)	38,785	56	55,000
TUNED SLEW NO. 1 BANG/COAST/BANG (41.7/0.64/4.17)	402	2	50
TUNED SLEW NO. 2 BANG/COAST/BANG (41.7/42.98/41.7)	56	2	85
PERFORMANCE SPECIFICATIONS	7	59	59

Conclusions of the disturbance evaluation task are summarized below.

EVALUATION OF DISTURBANCES

- FAST SLEWING DISTURBANCE DOMINATES
  - ORDERS OF MAGNITUDE LARGER THAN ALL OTHERS EXCEPT SLOW SLEW
  - SLOW-SLEW IMPULSE IS HIGH BUT TIME TO DAMP IS LONG
- VIBRATION CONTROL REQUIREMENTS DRIVEN BY
  - ELASTIC MODE RESPONSE TO FAST SLEW
  - TIME AVAILABLE IN ACQUISITION PHASE FOR DAMPING
- ORIGINAL FAST SLEW LEADS TO VERY LARGE (UNREALISTIC) VIBRATION-CONTROL TORQUES
- TUNING THE FAST-SLEW PULSES TO PERIODS OF FUNDAMENTAL ELASTIC MODES
  - LEADS TO A REALISTIC VIBRATION CONTROL PROBLEM
  - IS PRACTICALLY IMPLEMENTED

The control system development task designed decentralized control subsystems for vibration suppression, three-axis pointing, and required shape control. Fast slewing was taken to be open loop.

CONTROL SYSTEM DEVELOPMENT

Tasks

- Review LSS controls literature & on-going programs
- Develop decentralized pointing/vibration/shape controllers using:
  - Heuristically located actuators & sensors
  - Optimally located actuators & sensors

Approach

- Fast-slewing is open loop
- Vibration suppression system designed using filter-accommodated MESS
  - Control lower elastic modes, suppress rigid-body modes & a few higher elastic modes
  - Collocated actuators (reaction wheels) & sensors (rate gyros)
  - Filter rigid-body rates from rate gyro measurements
- Three-axis attitude controller for pointing & tracking
  - Each axis designed independently
  - Low-gain "coarse pointing" controller for target acquisition
  - High-gain "fine pointing" controller for target tracking
- Shape control consists of aligning the antenna feed over the reflector
  - Alignment for the tracking maneuver was demonstrated by simulations
  - The same controller will accommodate solar pressure & gravity gradient torques (these disturbance torques are comparable to the tracking torques)

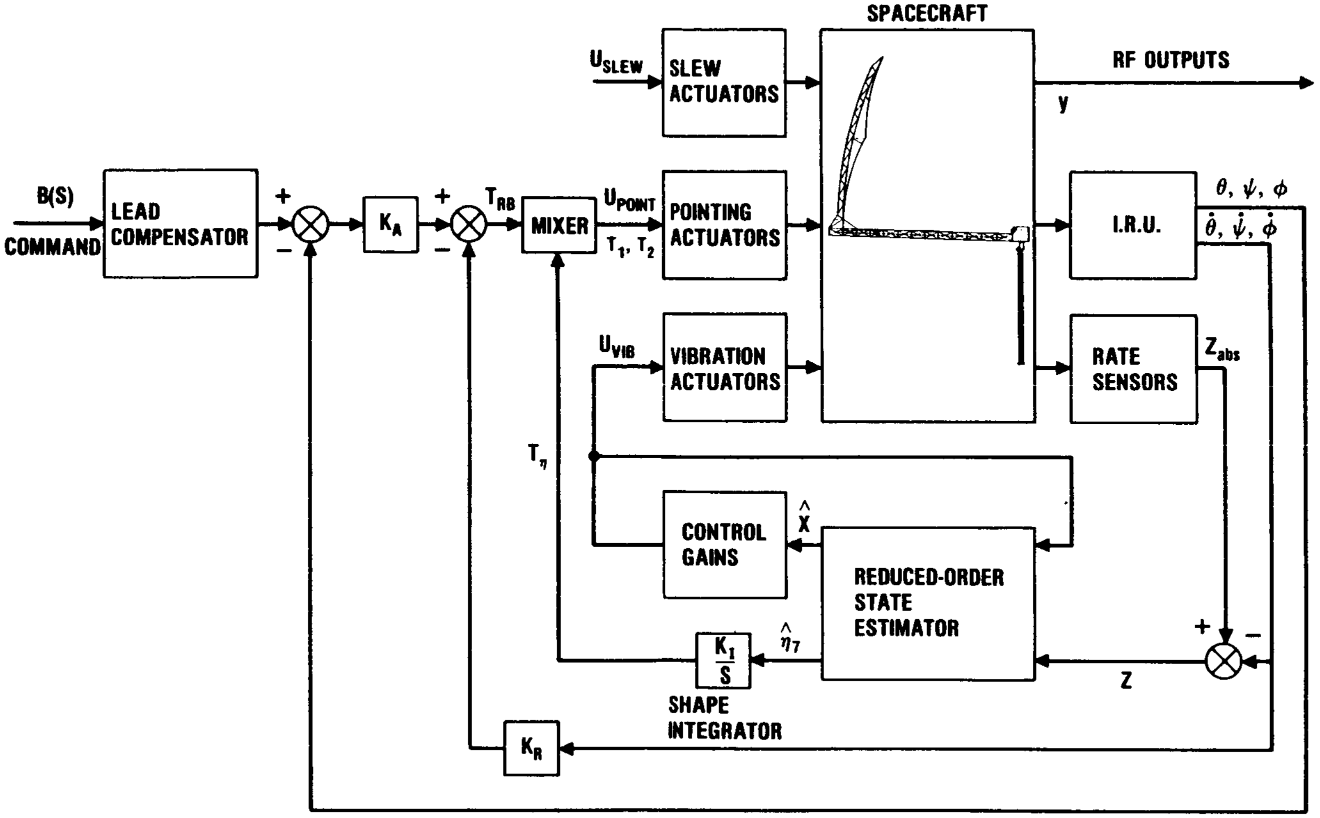
The Large Space Structures (LSS) controls literature was reviewed and the Model Error Sensitivity Suppression (MESS) design method was chosen as a method for designing the vibration control subsystem.

COMPARISON OF SOME LSS CONTROL DESIGN APPROACHES

TECHNIQUE	DESCRIPTION	ADVANTAGES	DISADVANTAGES
MESS	LQG – BASED APPROACH EXTENDED TO ACCOUNT FOR TRUNCATION OF KNOWN DYNAMICS; HEAVILY PENALIZES UNCONTROLLED DYNAMICS IN COST FUNCTION; CAN INCORPORATE ROLL-OFF FILTERS TO DECREASE EXCITATION OF UNKNOWN DYNAMICS.	• HIGH PERFORMANCE • ALLOWS DECENTRALIZED CONTROL • DIRECT METHODOLOGY TO SUPPRESS SUBSYSTEM INTERACTION	• DECOUPLING MECHANISM REQUIRES KNOWN DYNAMICS • MAY REQUIRE ADDITIONAL ACTUATORS TO ACHIEVE DECOUPLING • LQG ROBUSTNESS CONCERNS
IMSC	TRANSFORMATION APPLIED TO THE CONTROL INFLUENCE MATRIX SUCH THAT PRODUCT OF IT AND GAIN MATRIX IS DIAGONAL; EACH MODE CONTROLLED INDEPENDENTLY.	• CONTROLLED MODES ARE COMPLETELY DECOUPLED • EASY TO DESIGN	• FOR COMPLETE DECOUPLING, REQUIRES ONE ACTUATOR PER CONTROLLED MODE • "MODAL FILTERS" REQUIRE MANY SPATIALLY DISTRIBUTED SENSORS
HAC/LAC	HAC CONTROLLER DESIGNED VIA FREQUENCY – SHAPED LQG; LAC CONTROLLER DESIGNED USING OUTPUT FEEDBACK; FREQUENCY SHAPING PROVIDES A MEANS TO DECREASE EXCITATION OF UNKNOWN DYNAMICS.	• HIGH PERFORMANCE • FREQUENCY SHAPING ALLOWS INCORPORATION OF COMMON FREQUENCY DOMAIN CONSTRAINTS INTO STATE-SPACE FORMULATION	• HAC MAY DESTABILIZE LAC • FREQUENCY SHAPING MAY RESULT IN HIGH-ORDER SYSTEM • LQG ROBUSTNESS CONCERNS
POSITIVE REAL	A POSITIVE REAL COMPENSATOR APPLIED TO A LSS WITH FORCE ACTUATORS AND COLOCATED LINEAR VELOCITY SENSORS REMAINS POSITIVE REAL AND THUS STABLE REGARDLESS OF MODEL UNCERTAINTY	•TOTALLY STABILITY-ROBUST CONTROL DESIGN DUE TO PARAMETER INDEPENDENT STABILITY	• ACTUATOR DYNAMICS DESTROY POSITIVITY • DIGITAL IMPLEMENTATION ALSO DEGRADES STABILITY THROUGH THE ELIMINATION OF POSITIVITY USUALLY LOW PERFORMANCE CONTROL
MATHEMATICAL PROGRAMMING	LINEAR AND NONLINEAR MATHEMATICAL OPTIMIZATION TECHNIQUES USED TO DESIGN CONTROLLER; DESIGN CONSTRAINTS AND POSSIBLY AN OBJECTIVE FUNCTION ARE INCORPORATED INTO A CONSTRAINED MINIMIZATION PROBLEM SUBJECT TO THE LSS DYNAMICS.	• OPTIMIZES THE ACTUAL DESIGN VARIABLES • MECHANIZES THE ACTUAL ENGINEERING PROCESS • HANDLES NONLINEAR PROBLEMS • VERY GENERAL APPROACH	• SINCE THE TECHNIQUE EMULATES THE ENGINEER, THE ALGORITHM AND INTERFACE SOFTWARE CAN BE DIFFICULT TO DEVELOP • SENSITIVITY COMPUTATION CAN BE COSTLY
ALGEBRAIC METHODS (ESPECIALLY H_{00})	DESIGN THE COMPENSATOR DIRECTLY RATHER THAN A CONTROL LAW PLUS AN ESTIMATOR; FUNCTIONAL ANALYSIS METHOD OFTEN USED.	• ROBUSTNESS OF DESIGN EMPHASIZED • DESIGN CONSTRAINTS BASED ON FREQUENCY DOMAIN MEASURES	• COMPUTATIONALLY INTENSIVE • OFTEN RESULTS IN HIGH-ORDER COMPENSATORS • IMMATURE STATE OF DEVELOPMENT

Each of the concurrently operating subsystems is shown in the block diagram below.

LSPSC DECENTRALIZED CONTROL CONFIGURATION



Only the lowest 4 elastic modes (modes 7-10) contribute significantly to the LOS error. They are the modes that are actively controlled in the vibration control subsystem.

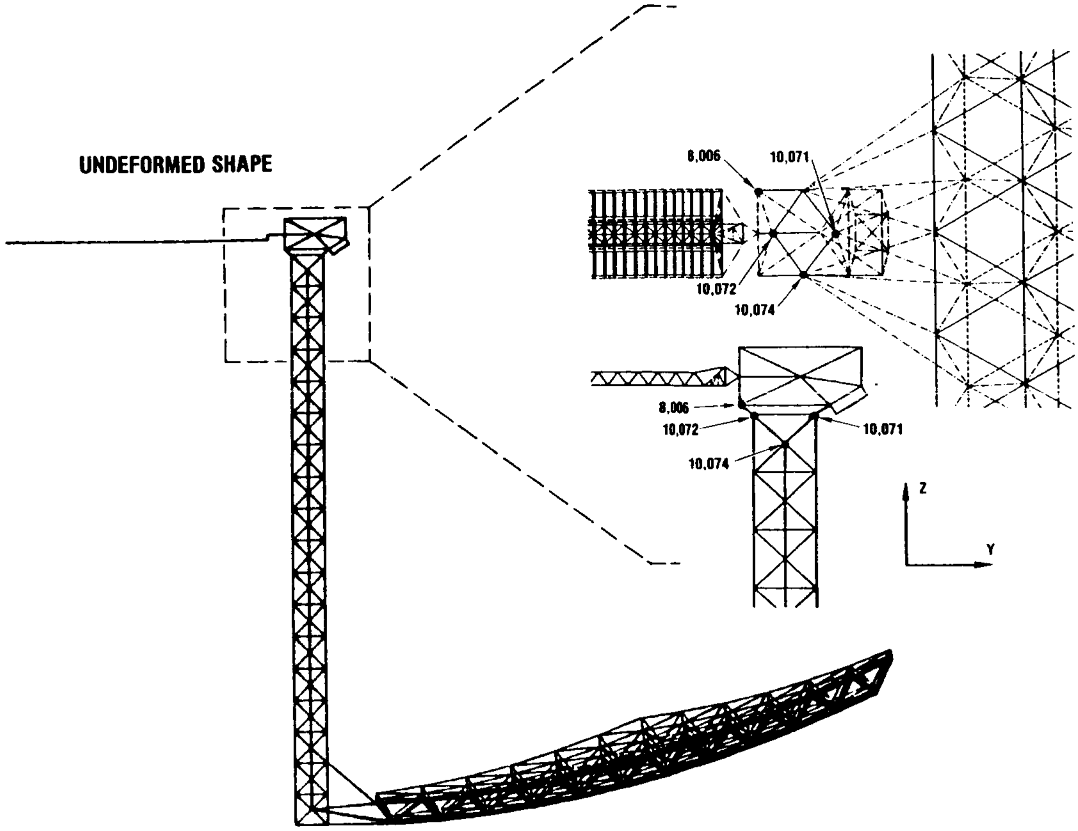
INDIVIDUAL MODAL CONTRIBUTIONS TO TOTAL LOS ERROR (PEAK NEAREST T = 130 SECONDS)

SLEW DESCRIPTION	MODE NUMBER			
	MODE 7 (.024 Hz)	MODE 8 (.038 Hz)	MODE 9 (.047 Hz)	MODE 10 (.059 Hz)
ORIGINAL FAST SLEW BANG/BANG (29.6/29.6 SEC)	37500 (96.7)	2 (.005)	1000 (2.6)	283 (.695)
TUNED SLEW NO. 1 BANG/COAST/BANG (41.7/0.64/41.75 SEC)	21 (5.2)	4 (1.0)	2 (.5)	375 (93.3)
TUNED SLEW NO. 2 BANG/COAST/BANG (41.7/42.98/41.7 SEC)	28 (50.0)	2 (3.6)	1 (1.8)	25 (44.6)

NOTE: ENTRIES ARE IN ARC-SECONDS. NUMBER IN PARENTHESIS INDICATES APPROXIMATE PERCENT OF TOTAL LOS ERROR

Both heuristically and optimally located actuators and sensors were investigated. Ten collocated actuators and sensors were used in each case. Ten actuators were needed since the torque per actuator was constrained.

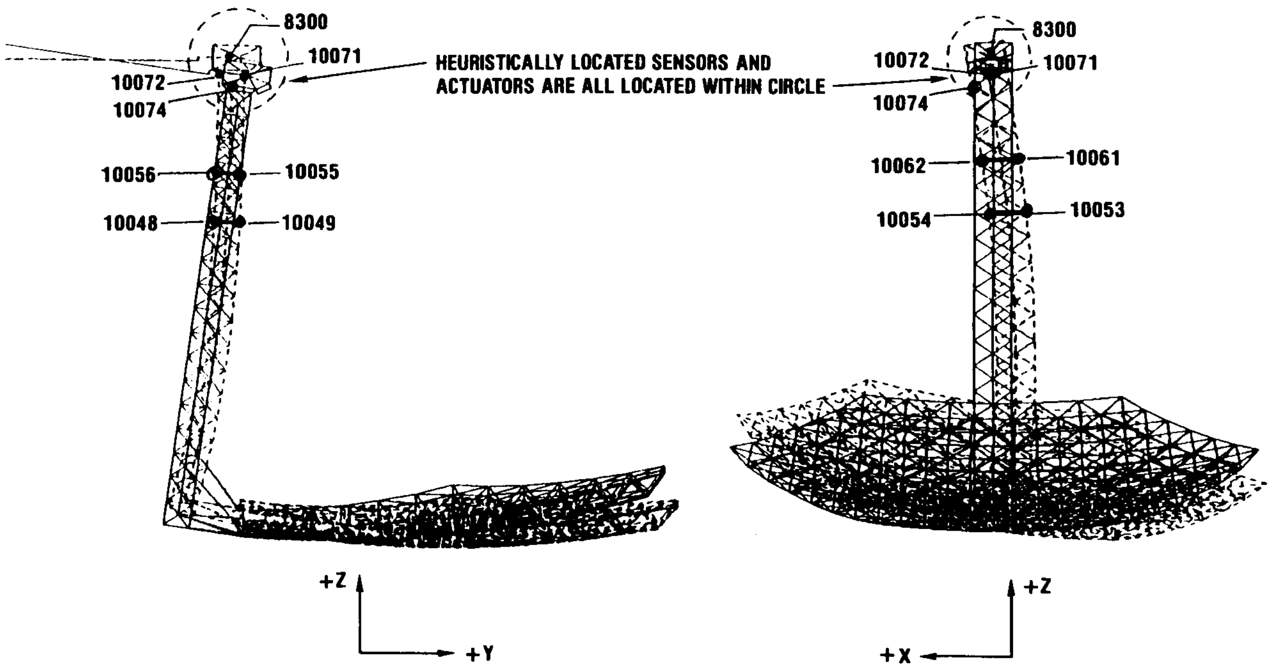
HEURISTICALLY LOCATED ACTUATORS FOR ACTIVE VIBRATION SUPPRESSION



NODE	TORQUE DIRECTIONS
8,006	X,Y,Z
10,071	X,Y,Z
10,072	X,Y,Z
10,074	Y

Optimizing the locations of actuators and sensors led to distributing them to locations of high modal kinetic energy.

OPTIMALLY LOCATED VIBRATION CONTROL SENSORS AND ACTUATORS SUPERIMPOSED ON MODES 7 AND 10 DEFLECTIONS



MODAL DEFORMATION: MODE 7 — 0.024 Hz

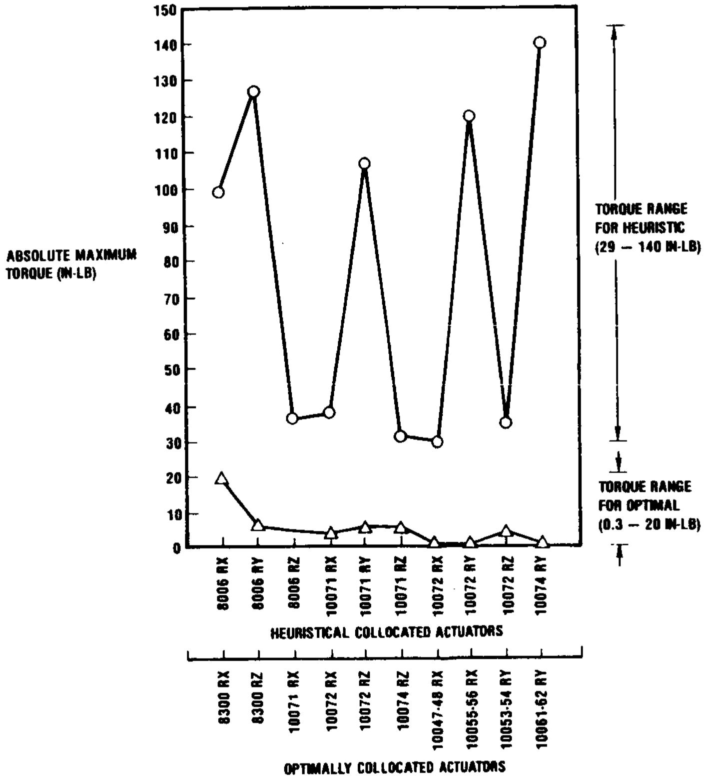
MODAL DEFORMATION: MODE 10 — 0.059 Hz

NOTE: BOOM MOUNTED SENSORS/ACTUATORS LOCATED AT POSITIONS OF MAXIMUM MODE 7 AND 10 SLOPES

The torque per actuator was substantially smaller for the optimally located actuators.

COMPARISON OF MAXIMUM ABSOLUTE VIBRATION-CONTROL TORQUE LEVELS (MESS-COMPENSATORS)

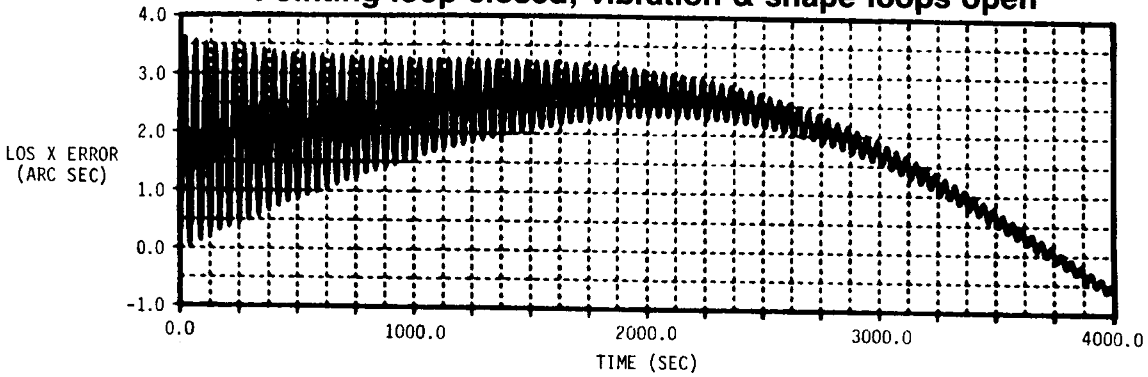
Heuristic Vs Optimal Locations



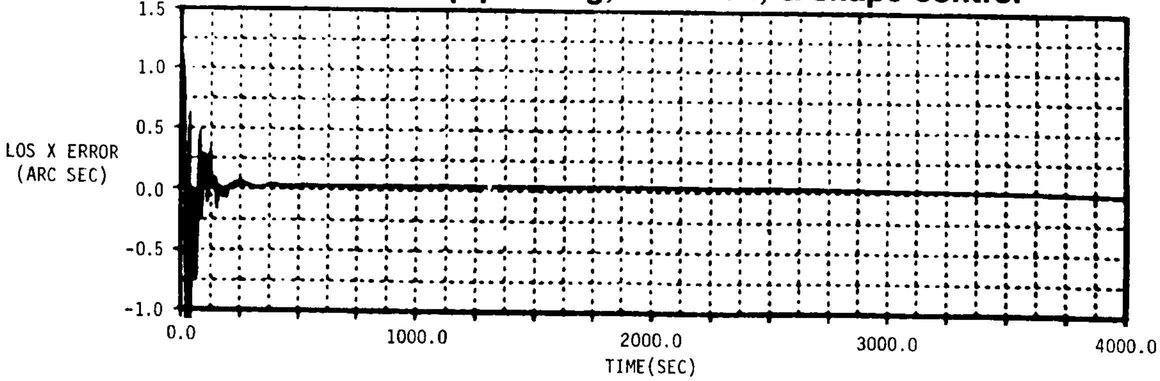
Open- and closed-loop LOS response is compared in the plots below. The open-loop response shows a significant slowly varying LOS error which is corrected by the shape control loop. The closed-loop response is well within our threshold for LOS error and also within our goal.

TRACKING MANEUVER RESPONSE

Pointing loop closed, vibration & shape loops open



Closed-loop pointing, vibration, & shape control



Conclusions from the control system design and nominal evaluation task are summarized below.

CONTROL SYSTEM DEVELOPMENT SUMMARY

- For this LSS with 0.5% assumed modal damping, only the lowest four elastic modes (modes 7, 8, 9 & 10) require active vibration suppression
- Distributed (optimally) actuators & sensors are able to suppress vibrations using much less control torque
- For this class of LSS, a larger number of actuators & sensors may be required than previously expected for the heuristically located actuator
  - — Driven by performance, maximum torque level, & hardware failure constraints
  - — We needed more actuators than controlled modes
- The nominal performance of the final closed-loop pointing/vibration/shape controller is within the goal
- Redesigns of each subsystem were required to achieve the performance goal; this suggests that a centralized approach may be more efficient

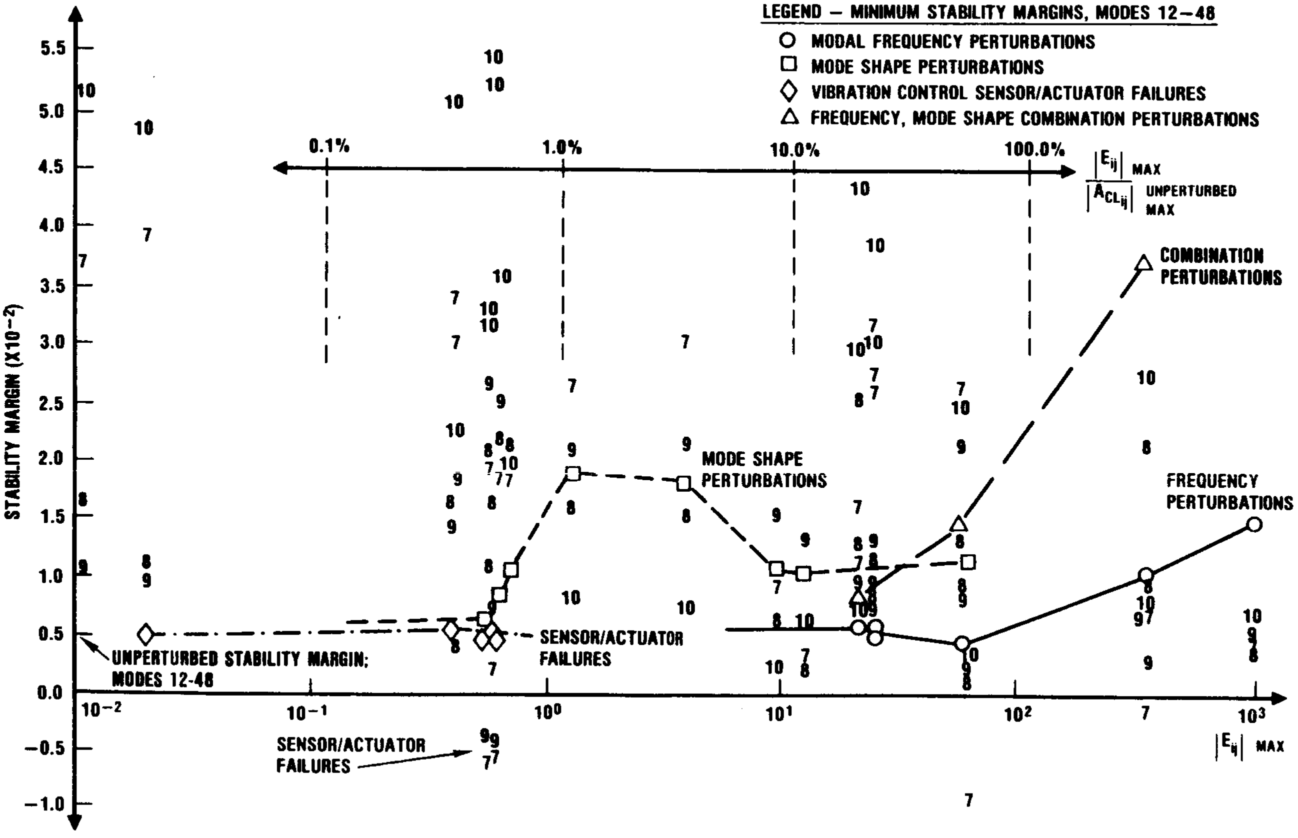
To evaluate the performance and stability robustness of each control system, both direct perturbations and frequency-domain singular value analysis were used.

ROBUSTNESS MEASURES

- Perturbation case studies — parameter variations made directly on the evaluation model; closed-loop stability & performance directly assessed
- Frequency domain singular value analysis (G_0, G stable)
  - — Stability robustness
    - Additive perturbations
      $\bar{\sigma}(G(J\omega)) < \underline{\sigma}(I + G_0(J\omega))$, $\omega \geq 0$
    - Multiplicative perturbations
      $\bar{\sigma}(G(J\omega)) < 1 / \bar{\sigma}[G_0 (I + G_0)^{-1}]$, $\omega \geq 0$
  - — Sensitivity
    $\Delta Y = (I + G_0)^{-1} G$ $\Rightarrow$ Make $(I + G_0)$ Large

The vibration control system is most sensitive to actuator and sensor failures.

VARIOUS PERTURBATIONS:* STABILITY MARGIN VS. MAXIMUM PERTURBATION MAGNITUDE



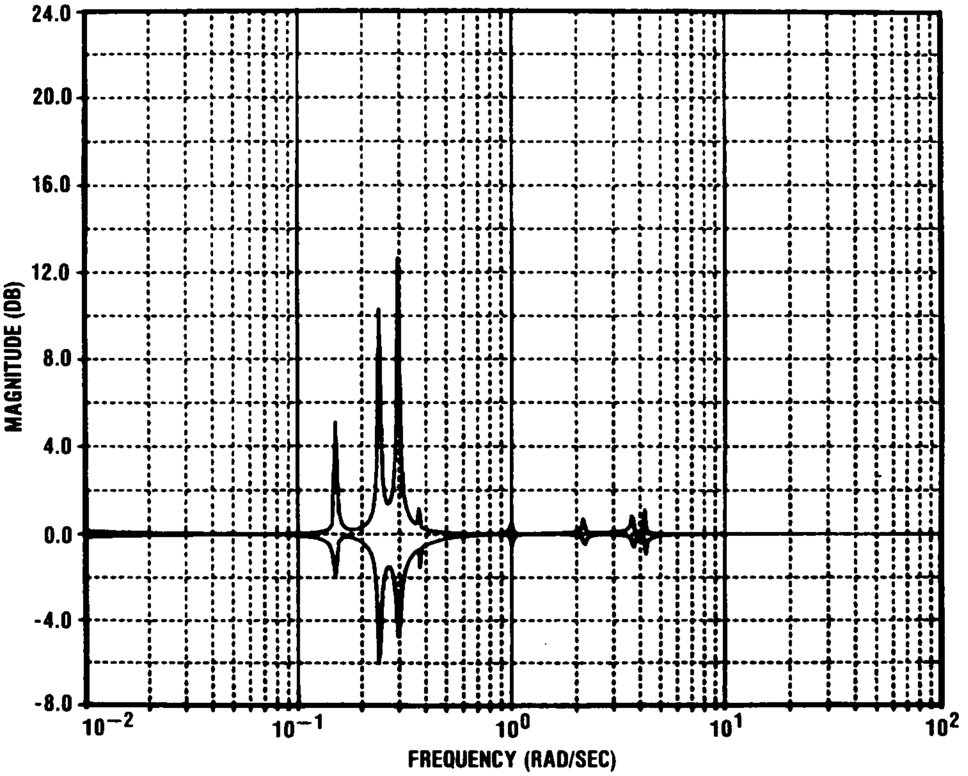
*PERTURBATION MODEL

The minimum singular value of the return difference matrix gives the distance from the critical point. The closer the minimum singular value gets to zero, i.e. minus infinity decibels, the closer the closed-loop system is to being unstable.

Comparing the minimum singular value of this plot with that on the following plot, one sees that the high-gain pointing loop increases the system's sensitivity to parameter variations by an order of magnitude.

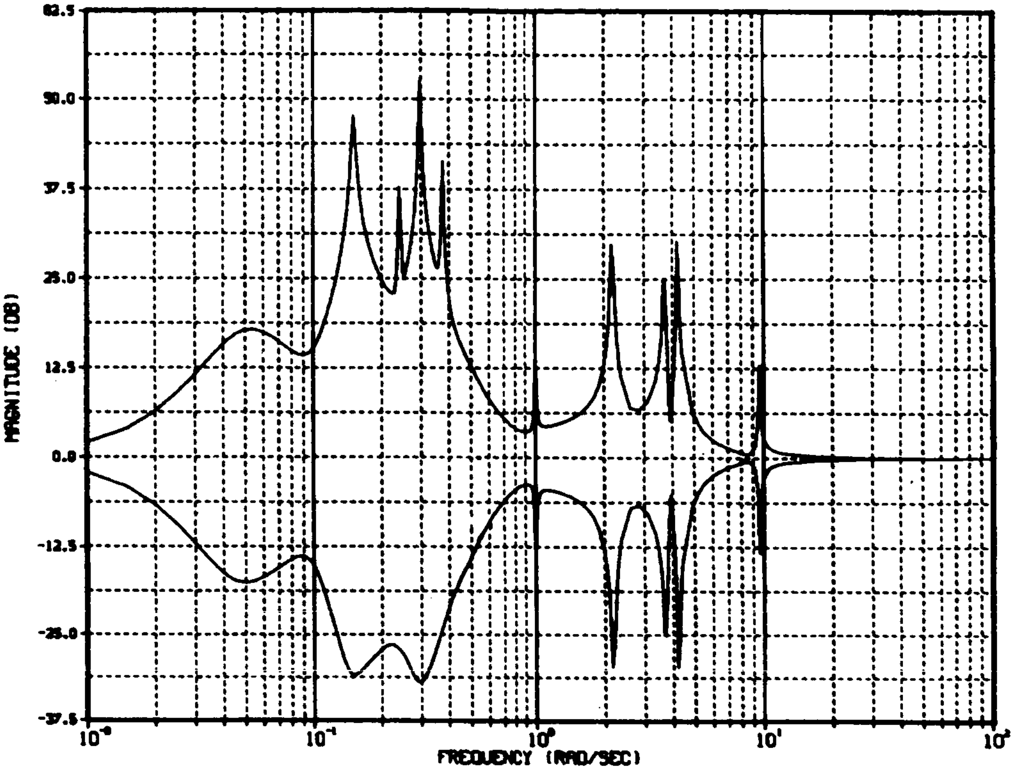
SINGULAR VALUES OF RETURN DIFFERENCE MATRIX VS. FREQUENCY

Closed-Loop Vibration Control Only



SINGULAR VALUES OF RETURN DIFFERENCE MATRIX VS. FREQUENCY

Closed-Loop Pointing, Vibration, and Shape Control



Conclusions concerning controller robustness are summarized below.

CONTROL ROBUSTNESS CONCLUSIONS

- The vibration suppression subsystem, when considered alone, possesses reasonable stability robustness qualities to modal frequency & mode shape perturbations
- The MESS compensator design is sensitive to certain actuator & sensor failures
  - The MESS algorithm depends on these sensors & actuators for subsystem decoupling
  - Collocated actuator & sensor failures do provide a degree of stability robustness, but not necessarily performance robustness
- Unstructed singular value analysis is useful in identifying frequencies at which sensitivity to perturbations is significant
- Interaction between the high-gain pointing & the flexible modes (primarily mode 9) in the perturbed system are extremely destabilizing to the integrated control system

The feasibility of adding passive damping to the vehicle was assessed and the effects of passive damping on the closed-loop system's performance were examined.

PASSIVE VS ACTIVE DAMPING TRADEOFFS

- An assessment of the LSPSC-spacecraft structure concludes that from 1% to 15% passive modal damping in the lower modes is achievable
- To achieve the highest levels of passive damping, it is important to consider it in the initial structural design
- For the LSPSC spacecraft, the optimum mix of passive & active damping is to use the highest achievable level and supplement it with active controls as necessary
- The slewing torque tuning we did is sensitive to passive damping levels
  - We actually found higher active-control torques with the addition of passive damping
  - This is considered a disadvantage of tuning the torques rather than a disadvantage of added passive damping

A number of important major conclusions resulted from the LSPSC study. The conclusions are summarized below.

LSPSC MAJOR CONCLUSIONS

Truss antenna structures are inherently stiff

- It takes "heroic" efforts to achieve reflector vibration frequencies less than 0.1 Hz, even with a reflector the size of 100 meters
- While the feed boom bending can have low frequencies, damping of these modes requires a different type control than does correction of reflector distortions

Slewing maneuvers are dominant design drivers

- Settling after fast-slew drives vibration control design
- Acquisition/tracking after fast-slew drives pointing control design

Rapid slewing/pointing of this size vehicle will require very large, fast responding actuators

- Large actuators add large nonstructural mass to the vehicle
- Locating the actuators leads to conflicting demands on minimizing vehicle moments of inertia & minimizing flexible-body modal excitation

LSPSC MAJOR CONCLUSIONS (continued)

Maturity of shape control technology is well behind other control technologies

- Actuators require development
- Sensors require a great deal of development

For a large truss antenna, only a few lower elastic modes require vibration control

- Slewing disturbances significantly excite only the fundamental boom bending & torsion modes
- RF parameters are most sensitive to these lowest modes

Spatially distributed actuators/sensors are advantageous

- The torque per actuator is reduced with more actuators
- Optimizing the locations of actuators/sensors leads to distributing to locations of high modal kinetic energy
- For same number actuators, torque per optimally located actuators is substantially smaller than the torque per heuristically located actuators

Decentralized control design leads to complex series of analyses

- Interaction among controllers with overlapping bandwidths is difficult to avoid
- Constant interaction analysis & subsystem redesign of decentralized controllers suggests that centralized approach may be more efficient
- Robustness of the integrated controllers should be considered from the outset

A significant level of passive damping is possible for truss structures (PACOSS conclusion)

- 5-15% passive modal damping reduces requirements for active vibration control
- Achieving 5% passive modal damping is very feasible
- With significant effort, can probably achieve 10%
- It is important to design for passive damping from the outset