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FACILITY TO STUDY THE CONTROL OF FLEXIBLE SPACE STRUCTURES

CONCEPTUAL DESIGN

[DOC-CR-SP-83-022]

DYNACON

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G. B. Sincarsin

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SUMMARY

A conceptual design is presented for a facility to study the control of flexible space structures. Attention is focused primarily on an appropriate flexible structure, a set of sensors and a set of actuators. Each of these components is chosen by subjecting a variety of possible candidates to both a preliminary and a detailed evaluation process. Extensive use is made of decision matrices.

PREFACE

Acknowledgments

The author would like to thank a number of people for their assistance in preparing this report. The insight gained through discussion with D. A. Staley (of ANCON Space Technology Corp.) and C. H. Hersom (of HERSOM Engineering Services) into the difficulties associated with, and suitability of, various hardware and electronics-support devices is gratefully acknowledged. The author would also like to thank P. C. Hughes (of Dynacon) for his critical review of the manuscript and for penning the introduction. Thanks are also due Mrs. J. Hughes for typing this document. Finally, the author is pleased to acknowledge the encouragement and timely advice of A. H. Reynaud of the Communications Research Centre, who acted as the Scientific Authority for this contract.

Proprietary Rights

Dynacon Enterprises Ltd. does not claim "proprietary rights" to the material in this report. Indeed, the hope is that the analyses, results, ideas and opinions in this report will be useful to others. In this event, a reference to this report would be appreciated.

Units and Spelling

This report uses S.I. units and (for the most part) North American spelling.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

1. INTRODUCTION

It has been recognized for many years that one of the challenges in maintaining strict control over the orientation of many Earth satellites is the challenge of structural flexibility. Moreover, current age-of-the-shuttle spacecraft designs tend to be larger and more flimsy than ever before, and structural flexibility for such designs becomes much more than a minor annoyance — it often poses the chief limitation to attitude-control-system (ACS) performance. To cope with structural flexibility in the process of designing an ACS for what have come to be known as 'third-generation' satellites, one must distribute a variety of sensors and actuators throughout the (flexible) vehicle. In addition to the obvious perplexities associated with choosing the type, number, and location of these sensors and actuators, there is, as well, a profound theoretical implication: 'classical' control theory, restricted as it is to a single 'input' to (and a single 'output' from) the controller, is no longer adequate. The control analyst must apply 'modern' control theory — a family of methods developed expressly to handle many inputs and many outputs.

This state of affairs has prompted a great deal of attention. In 1981, for example, there were at least 21 technical conferences in North America alone that had one or more sessions devoted to the attitude control of third-generation spacecraft. In fact, several of these conferences dealt exclusively with this subject. At present, over 200 papers per year are published on this topic. (And this figure does not include the many limited-circulation reports also being produced.) One might conclude from all this activity that 'the problem' must surely be, by now, essentially solved. However, this is not in fact the case. Only the first phase of the problem solution process is nearing completion — what might be termed the 'idealistic analysis' phase. In the current literature, a large number of idealistic assumptions are made over and over again — so many times, in fact, that one is tempted to forget the shaky premises on which one's conclusions tend to be based.

Only very recently has there been any significant activity in what might be called the 'second phase' of third-generation-satellite-control research: the 'laboratory development' phase. In the crucible of actual experience with physical hardware, really promising control techniques can be distinguished from merely optimistic ones. Indeed, in at least two respects — the presence of gravity and the absence of vacuum — ground-based experiments are *more* challenging than their counterparts in space. Yet it is virtually self-evident that a significant research and development effort in the laboratory is an essential prerequisite to the successful control of large structures in space.

In view of these facts, the following objectives have been identified for this work:

- A laboratory facility must be developed whose primary purpose is to study the stabilization and control of structurally flexible communications satellites.
- Fundamental concepts in the control of flexible space structures must be investigated and evaluated. Some concepts, although of theoretical appeal, may not all prove to be viable in practice.
- 'Hands-on' experience must be developed with realistic sensors, actuators, structures, and control electronics in order to ensure the practicality of proposed control techniques.
- Control approaches must be developed that are especially adapted to the unusual and challenging requirements of large, flexible *space* structures (e.g., system eigenvalues lightly damped, many system modes, clustered frequencies, zero gravity, vacuum, many-year lifetime with no chance to correct design errors, etc.).
- The results of this research must be of direct relevance to the attitude control of the next generation of Canadian communications satellites.

This report outlines the conceptual design of a facility that meets these objectives.

2. EVALUATION PHILOSOPHY

2.1 Evaluation Scheme

The objectives of the laboratory development program cited in Section 1 can only be achieved if sufficient care is taken during the conceptual design phase to assess and choose a design--'flexible structure', actuators, and sensors--that is consistent with the constraints inherent in those objectives. The primary concern must, of course, be the choice of a suitable structure. The nature of the structure then dictates, at least to some extent, the choice of actuators and sensors. It is possible, however, to apply a common evaluation scheme to each of the three design components, even though the details of each evaluation may be somewhat different. In particular, a three-step procedure is used to assess the candidate structures, actuators and sensors. The first step is to identify the aims of the evaluation. A preliminary qualitative selection of suitable candidates, based on these aims, then completes the second step. Finally, a detailed selection (both qualitative and quantitative) is performed to choose the most acceptable candidate(s).

2.2 Detailed Selection

The quantitative portion of the detailed-selection step takes the form of a *decision matrix*. This evaluation technique is well established and can be found in a number of texts (see, for example, [Hill, 1970] and [Middendorf, 1969]). Essentially, one chooses a set of evaluation criteria $[c_i]$ ($i = 1,2, \ldots, n$, where n is the number of criteria) that reflect the aims of the evaluation. A weight w_i is then assigned to each criterion c_i according to its perceived importance relative to the remaining criteria $[c_k]$, $k \neq i$. Each alternative candidate for evaluation C_j ($j = 1,2,\ldots,m$, where m is the total number of candidates) is then assigned a rating for each criterion c_i according to how well it meets that criterion. Hence, one obtains a 'matrix' of ratings $[r_{ij}]$, where r_{ij} is the measure of how well c_i is met by C_j. The total rating of each candidate is then found by forming the weighted sum

$$R_j = \sum_{i=1}^{n} w_i r_{ij} \tag{2.1}$$

with the most acceptable candidate being that yielding the highest rating. In what follows, w_i and r_{ij} each have the domain (1,2,3,4,5), with 1 denoting the lowest weight or rating, and 5 the highest. The highest possible rating is, therefore,

$$R_{max} = 5 \sum_{i=1}^{n} w_i \tag{2.2}$$

The usual practice of nomalizing the weights w_i according to

$$\sum_{i=1}^{n} w_i = 1 \tag{2.3}$$

is not performed here, in order to limit the number of digit entries required to perform the calculations for some of the more complex decision matrices. However, as is common practice, the final total ratings are forced to lie within the interval $[r_{ij_{min}}, r_{ij_{max}}] \equiv [1,5]$ by forming

$$R_{j_{new}} = R_{j_{old}} (5/R_{max}) \tag{2.4}$$

where $R_{j_{old}}$ and R_{max} are given by (2.1) and (2.2)

3. FLEXIBLE-STRUCTURE EVALUATION

3.1 Evaluation Aims

As stated in Section 1 the prime objective of the laboratory development program is to develop and assess control laws for third-generation spacecraft. This intent motivates the primary aim of the flexible-structure evaluation, namely, to choose a structure which can emulate third-generation spacecraft. In this regard, ideally, the structure would have its first nonzero frequency (assuming rigid-body modes are present) in the range 0.06 to 3.14 rad/s (0.01 to 0.5 Hz). This range is suggested by some of the more recent predictions for the dynamic characteristics of such structures (see, for example, [Hedgepeth, 1981] and [Sincarsin and Hughes, 1982]). Achieving this goal would also provide an opportunity to design control system demonstrations within the realm of realistic characteristic response times for the structure, as well as for the actuators and sensors.

The tendency to use repeated structural components, which are only weakly coupled through the dynamics of other secondary (in the sense of dynamical importance, either in mass or in stiffness) structural components, in the design of third-generation spacecraft results in the phenomenon of *clustered* frequencies. For example, the wrap-rib reflector being considered for use on MSAT exhibits this tendency [Tolivar, 1982]. In this structure the flexible ribs affixed to the rigid central hub are weakly coupled dynamically by the mesh gores stretched between each rib. The result is that many of the reflector's natural frequencies occur 'clustered' in narrow frequency bands. In fact, in the limit as the dynamical significance of the mesh goes to zero, the flexible ribs would all vibrate independently and have identical natural frequency spectra (assuming identical ribs). Therefore, the desire to choose a flexible structure for the control-system demonstration laboratory, which is capable of producing clustered frequencies, also becomes an important aim of the flexible-structure evaluation. This requirement is reinforced by the fourth objective of the laboratory development program cited in Section 1.

Another aim of the flexible-structure evaluation is to define an original structure — original in the sense that the structure does not replicate previous structural designs used in similar demonstration experiments in the U.S.A. A summary of these efforts is given in Table 1. This search for originality is motivated by the desire to provide new information built upon experiences of others rather than simply mimic their results.

One characteristic common to all the structures cited in Table 1 is their lack of adaptability. While in several of the experiments

Table 1

Relevant Control Demonstration Experiments

Company	Type	Description	Sensor	Actuator	Demonstration
Draper	Beam	Fixed-free 1/4" x 1' x 60" Aluminum	Piezoelectric Accelerometers	Electrodynamic Shaker	Observation/control spillover modern modal control
Lockheed	Beam	Fixed-free 40" Magnesium	Optical rate sensor	Proof-mass	Low-authority control
	I-Beam	Fixed-free 25' x 16" (400 lbs) Aluminum	Optical rate sensor	Single gimbal CMG	Low-authority control
	Vertical Beam (Thin-Walled)	Fixed-free 6' Aluminum lead tip masses	Accelerometers, quad-detector photo diodes	Pivoted proof-mass	Low-authority control System Identification
	Circular plate	Suspended, 2 meter diameter, Aluminum	Multi-channel micro-phase optics	Pivoted proof-mass	Low-authority control Low/High-authority control System Identification
	Frame	Suspended, 2 meter diameter, Aluminum tubes	Accelerometers, optics	Pivoted proof-mass	
	Toysat	Suspended rigid body 1.6 m cantilever beams Aluminum	Accelerometers, LVDT velocity pick-offs	Electrosesis Actuators	Open-Loop torque profile High-authority control
	Proof of Concept	Air Bearing Suspension 25' long, dish diameter 20', Aluminum	Laser Attitude Sensors Rate Gyros Accelerometers	3-CMG Cluster Pivoted proof- mass	Central- and Distributed-Actuator Control, Optimal Slewing, Dynamic Figure Control
Convair	Plate	Fixed-free 68" x 103" Aluminum 4" x 5/16" welded beams	Rate Gyros	Torque wheels	Model error sensitivity suppression
JPL	Beam	Pinned-free 150" x 6" x 1/32" Stainless steel	Eddy current position sensor	Brushless dc torque motor	Modern modal control
LaRC	Beam	Suspended 12' x 6" x 3/16" Aluminum	Noncontacting deflection sensor, load sensor	Electrodynamic shaker	

Dynacon Enterprises Ltd. is indebted to Dr. G. Rodriguez (JPL) and Dr. R. Strunce (Draper) provisions have been made to enable the relocation of actuators and sensors, the flexible structure cannot be changed (except for minor redistributions of the mass by the addition of rigid weights). Ideally, one would like to design a structure which could 'grow' with the demonstrated control capability. For example, assume a suitable controller is designed for a structure with a given set of structural parameters, one in which there are no structural or material asymmetries. Obviously, it would be advantageous to be able to adapt the existing structure to permit a change in symmetry and hence, assess the robustness of the controller. In fact, once the performance limit of the 'existing' controller is reached, the adapted structure then becomes the basis for a new controller design. It may also be possible to 'tune' an adaptable structure to produce dynamic behavior more suited to a particular control demonstration (e.g. control spillover).

Finally, while the first two aims of the structure evaluation are intended to assure that the chosen structure is at least indirectly related to proposed third-generation spacecraft, in that the results of the eventual control demonstrations should be applicable to this class of spacecraft (since the structure will resemble dynamically third-generation spacecraft), it still is desirable, ideally, to choose a structure geometrically or materially similar to actual proposed spacecraft. This desire for physical realism, however, is of secondary concern here. One realistic physical characteristic, that of being able to withstand testing in an vacuum will be given some consideration. However, for practical reasons, it is not anticipated that such testing will be performed, unless air drag poses a problem, or a particular control scheme warrants testing in such an environment (e.g. qualifying tests).

3.2 Preliminary Evaluation

In an attempt to make the evaluation as comprehensive as possible, very few structures were eliminated during the preliminary evaluation stage. It is interesting to note, however, that not all the structures previously chosen for control demonstration (see Table 1) have their first nonzero frequency fall within the preferred frequency range specified in the previous section. At present, based on the literature, TOYSAT [Breakwell and Chamber, 1981], the JPL beam [Schaechter, 1981] and the LaRC beam [Montgomery, 1980] have appropriate first modal frequencies, while the Lockheed Proof of Concept (POC) [Breakwell, 1981] and vertical beam [Aubrun, Breakwell and Chambers, 1981] do not. (Information concerning the remaining structures is still being sought.) This suggests that, for the present laboratory development program, certain of the structures previously considered are inappropriate, not from a conceptual viewpoint, but rather from a dynamical viewpoint. A choice of different structural material or a different geometry for a particular structure could possibly reverse this conclusion.

To assess the extent to which one is free to change the material properties of a structure in order to 'tune' the lowest nonzero natural frequency to some desired value, a number of simple structures were subjected to dimensional analysis using the Buckingham Pi Theorem [Baker, Westine and Dodge, 1973]. The case for a beam is given in some detail in Appendix A. The conclusion cited there applies equally well to membranes and plates, or to any general structure. Simply, if one wishes to scale structure A down in size to obtain structure B, while retaining *geometric similarity*, then the required nondimensional relationship to guarantee *dynamic similarity* takes the form

$$\omega_A^i \frac{\ell_A}{\alpha_A^i} \left(\frac{\rho_A}{E_A} \right)^{\frac{1}{2}} = \omega_B^i \frac{\ell_B}{\alpha_B^i} \left(\frac{\rho_B}{E_B} \right)^{\frac{1}{2}} \qquad \text{(B.1)}$$

where ℓ_j, ρ_j and E_j are a characteristic length, the 'effective' density and the 'effective modulus of elasticity for structure j (e.g. for a circular plate, ℓ_j is the radius, ρ_j is the material density, and $E_j = E/(1 - \nu^2)$, where E is Young's modulus and ν is Poisson's ratio). The quantities ω_j^i and α_j^i (which is dimensionless) are the ith natural frequency and the corresponding boundary-condition factor for that frequency (e.g. for the first natural mode of vibration of a circular plate fixed on the boundary, $\alpha_j^1 = 10.21$, while for a free boundary $\alpha_j^1 = 5.251$ [Timoshenko, Young and Weaver, 1974]).

Now, assuming that only the material properties of the original structure are changed in order to force the first natural frequency to lie within the preferred range, then (3.1) becomes

$$\omega_B^i = \omega_A^i \frac{(\rho_A/E_A)^{\frac{1}{2}}}{(\rho_B/E_B)^{\frac{1}{2}}} \tag{3.2}$$

where the boundary conditions on the structure are unaltered. Typically, for structures in Table 1 which fall outside the preferred frequency range, a reduction of approximately an order of magnitude in the lowest frequency is required. This implies a need for a new material such that ρ_B/E_B is 100 times larger than ρ_A/E_A. This is not easily achieved given the material properties of most common materials. The values cited in Table 2 demonstrate the difficulty.

One experiences a similar problem when a large existing structure is scaled down to produce a geometrically similar structure of much smaller dimensions, but possessing the same frequency spectrum as the original structure. Assuming identical boundary conditions for the pre-scaled structure A and the post-scaled structure B, and matching frequency spectra, the appropriate form of (3.1) becomes

$$\ell_B = \ell_A \frac{(\rho_A/E_A)^{\frac{1}{2}}}{(\rho_B/E_B)^{\frac{1}{2}}} \tag{3.3}$$

Once again, the material properties of commonly available substances greatly limit the achievable reduction in size (normally by a factor < 10, see Table 2). As a consequence, the exercise of scaling a 55 m ($\ell_B = 27.5$ m) graphite/epoxy wrap-rib reflector [Wade, Sinka and Singh, 1979] to one 3 m ($\ell_B = 1.5$ m) in diameter (for example, to meet test facility space limitations), while maintaining geometric and dynamic similarity, would not even be possible with a rib material change from graphite/epoxy to polyethylene ($\ell_B = 4.15$ m > 1.5 m).

Assume for the present that one is willing to sacrifice some increase in the lowest natural frequency to achieve a diameter of 3 m after scaling the above reflector (i.e., $\omega_B^i = 2.8\ \omega_A^i$). The problem

Table 2

Material Properties of Some Common Substances*†

Material Property Substance	Density, ρ (kg/m^3)	Modulus of Elasticity, E (N/m^2)	Possion's Ratio, ν	ρ/E	$\rho(1 - \nu^2)$/E
Steel	7.83^{3}	2.07^{11}	0.33	3.78^{-8}	3.37^{-8}
Aluminum	2.77^{3}	7.31^{10}	0.35	3.79^{-8}	3.33^{-8}
Magnesium	1.80^{3}	4.48^{10}	0.31	4.02^{-8}	3.63^{-8}
Wood	5.40^{2}	1.21^{10}	-	4.46^{-8}	-
Graphite/Epoxy	2.51^{3}	9.65^{10}	-	2.60^{-8}	-
Polycarbonates	1.20^{3}	2.12^{9}	-	5.66^{-7}	-
Fiber-glass (Nylon)	1.41^{3}	1.10^{10}	-	1.28^{-7}	-
Polystyrene	1.28^{3}	8.27^{9}	-	1.55^{-7}	-
Polyethylene	9.53^{2}	8.45^{8}	0.46	1.13^{-6}	8.89^{-7}

* value taken from [Popov, 1968] and [Bolz and Tuve, 1970]

† $N^{e} = N \times 10^{e}$

then becomes: can such a scaled structure support itself adequately under the influence of gravity? The answer is no! Consider a reflector oriented with its planform horizontally and assume that the length of each cantilevered rib is approximately one-half the diameter of the reflector. Then, as shown in Appendix B, gravity will cause a static tip deflection of 1.3 m for a polyethylene rib 1.5 m in length. Obviously this result is senseless within the context of the small-deflection assumption inherent in simple beam theory. The rib, in fact, will suffer lateral buckling, as the distributed gravitational load exceeds the critical lateral-buckling load by almost a factor of 2 (see Appendix B).

Faced with these results one might be tempted to resort to a somewhat more 'creative' scaling technique, whereby, rather than maintaining the same boundary conditions in the pre- and post-scaled structures, a different boundary condition is assumed for the post-scaled structure. If this scenario is adopted then (3.3) becomes

$$\ell_B = \ell_A \left(\frac{\alpha_B^i}{\alpha_A^i} \right) \frac{(\rho_A/E_A)^{\frac{1}{2}}}{(\rho_B/E_B)^{\frac{1}{2}}} \tag{3.4}$$

The implication is that by choosing the appropriate boundary condition for the scaled structure B, the factor α_B^i/α_A^i can be made to compensate for the shortcomings associated with the material property factor in (3.4). Unfortunately, this view is short-sighted. Firstly, the factor α_B^i/α_A^i is not likely, in general, to be a constant for all modes i. In the case of simple structures (e.g. a beam, a membrane or a plate), it is clear from standard tables [Volterra and Zachmanoglou, 1965] that α_B^i/α_A^i is not independent of i, where, now, α_B^i and α_A^i are boundary factors for two distinct boundary conditions on the same structure. One has no reason to believe this tendency should reverse itself if one structure is scaled and the other is not, especially since geometric similarity is retained. As a consequence, one can not truly match the frequency spectrum of the pre- and post-scaled structure if this technique is employed. The best one can do is to match a single frequency. One might still conclude that this is an acceptable restriction if, in fact, the frequency so matched is the first modal frequency. The problem with this reasoning is that, while the scaled structure 'emulates' the actual structure for one frequency, the remaining frequency spectrum will likely bear little resemblance to that of the original structure, nor will the mode shapes be the same either. Hence, the ability to 'tune', or scale, a particular structure to produce a particular first modal frequency must rely on suitable choices of ℓ_B, ρ_B and E_B, unless one is willing to sacrifice geometric similarity. From the above arguments, this appears to be essentially what must be done. Therefore, it is unlikely that the flexible structure finally recommended below will resemble exactly any particular existing structure, but instead will incorporate *ideas* from those structures in its design. It is within this context that the detailed structure evaluation is now undertaken.

3.3 Detailed Evaluation

As described in the previous section, the use of a decision matrix plays a key role in the detailed evaluation procedure. The first step in establishing this matrix is to choose the criteria upon which the structures to be evaluated will be judged. This choice is based on the aims cited in Section 2.1 and the constraints inherent in designing and modeling a suitable flexible structure for testing in a ground-based laboratory. The criteria judged most important, and the ideal situation in each case, are shown in Table 3.

Rather than attempt to evaluate every available flexible structure, the various existing and proposed designs are sorted into categories according to the essential geometric features of each structure. The chosen categories are shown in Table 4, accompanied by a number of representative structures belonging to each category.

This narrowing of candidates into eight rather broad categories was precipitated by the need to evaluate fairly a myriad of designs within a reasonable time. Consider, for example, the sub-categories Truss Beams and Antenna Reflectors. Figure 1 shows nine alternative truss beam designs and lists ten different deployable antenna reflector concepts. It is anticipated, however, that the dynamical performance of these structures should not be markedly different, given that their designers are all subject to the

Table 3

Evaluation Criteria for Flexible-Structure Selection

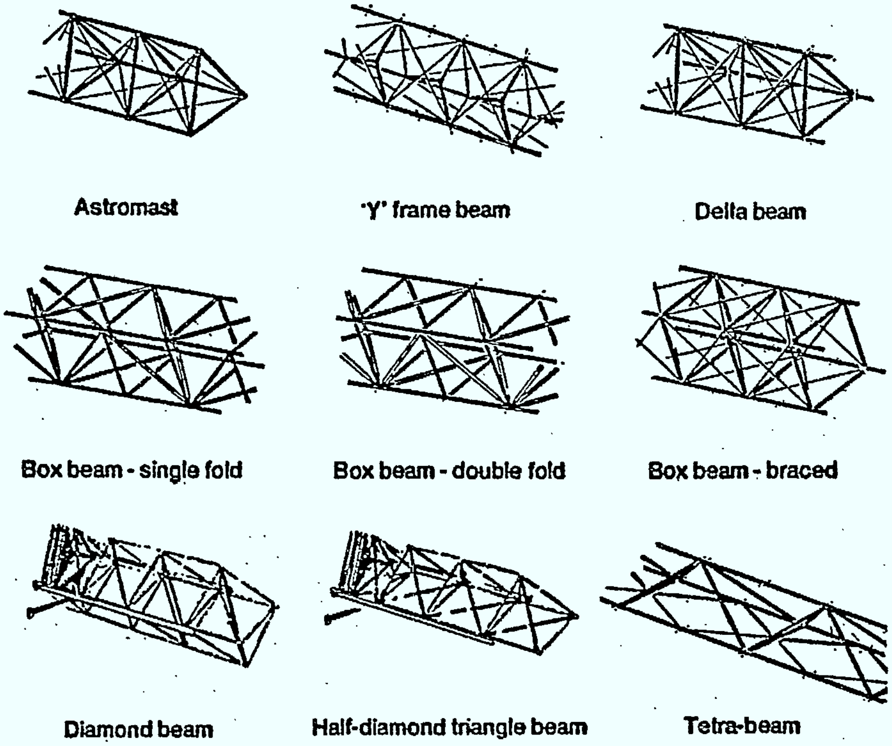
EVALUATION CRITERIA	THE IDEAL
1. Structural Flexibility	- structure has modal frequencies in the same range as Third Generation spacecraft - Frequencies can occur in clusters
2. Originality	- Experiments would build on, not repeat, earlier work in the U.S.A., thus providing new information
3. Adaptability	- Structure can be easily modified to provide a variety of different dynamic characteristics - It can demonstrate a number of different control strategies
4. Size/Weight	- Structure is a manageable size and its weight does not require an excessively large support structure
5. Complexity	- Structure is simple with the number of different structural components kept to a minimum
6. Cost	- Structure is inexpensive to manufacture and maintain
7. Reliability/Life	- Structure is not prone to structural fatigue - It consists of time proven, rather than novel, structural components - These components are insensitive to the working environment - A life span of over 5 years
8. Ease of Modeling	- Structure can be easily modeled analytically
9. Ease of Manufacture	- Structure consists of 'off-the-shelf' components and materials - It can be fabricated using common techniques and using equipment found in a well equipped machine shop - Structure is easy to build and can be built within a reasonable period of time
10. 1 - g Suitability	- Structure can withstand 1 - g test environment - There is no static deflection, or whatever deflection there is can be 'trimmed' away
11. Acceptance of Hardware	- Structure can accept a variety of actuators and sensors - Placement of hardware is unlimited
12. Space Related	- Structure emulates existing (or proposed) spacecraft - Experimental results can be extrapolated, or are directly applicable, to actual spacecraft

Table 4

Flexible-Spacecraft Categories

CATEGORY	BASIC	ADVANCED
1. Beams	Common Beams (AR[†] ≐ 1) Thin Beams (AR >> 1) Thin-Walled Beams I - Beams	Truss Beams Telescoping Cylindrical Beams Bellows Beams
2. Membranes	Rectangular/Square Elliptical/Circular	Pretensioned Meshes (wrap-rib) Stringer Tensioned Meshes (Hoop-Column) Electrostatically Tensioned Plastic Membranes
3. Plates	Rectangular/Square Elliptical/Circular	Toroidal Box Truss Geodesic Truss 'Planar' Truss (tetrahedral, diamond, box) Pretensioned Trusses
4. Compound Structures	Flexible and Rigid Body combinations (e.g. TOYSAT, Lockheed POC) Compound Frames (stayed columns)	Solar Power Collectors Space Platforms Antenna Reflectors

[†] AR = characteristic height/characteristic width (or inverse)



Astromast — 'Y' frame beam — Delta beam

Box beam - single fold — Box beam - double fold — Box beam - braced

Diamond beam — Half-diamond triangle beam — Tetra-beam

Figure 1a. Truss Beams
(taken from [Jenkins, 1981])

	CONCEPT	ORIGINATOR	ANTENNA TYPE	DIAMETER RANGE (M)
1.	EXPANDABLE TRUSS	GENERAL DYNAMICS	REFLECTOR	10-300
2.	WIRE WHEEL	GRUMMAN	PHASED ARRAY (REFLECTOR/LENS)	50-300
3.	HOOP/COLUMN	HARRIS	REFLECTOR	15-100
4.	ARTICULATED RIB	HARRIS	REFLECTOR	15-30
5.	CURVED ASTROMAST	HARRIS	REFLECTOR	15-<100
6.	RADIAL COLUMN	HARRIS	REFLECTOR	15-<100
7.	WRAP RIB	LOCKHEED	REFLECTOR	9-200
8.	POLYCONIC	LOCKHEED	REFLECTOR	30-300
9.	MAYPOLE	LOCKHEED	REFLECTOR	30-300
10.	SUNFLOWER	TRW	REFLECTOR	5

Figure 1b. Deployable Reflector Concepts
(taken from [Campbell, Croswell, Deaton and Dobrotin, 1978])

same structural design requirements. More often, the important factors differentiating between two particular designs are governed by considerations such as stored mass and size, ease of deployment, specific uses (for example, the ability to turn corners to create an 'elbow' joint during deployment), developmental costs, the ability to experimentally qualify the design 'on the ground' and the ultimate cost for an actual spacecraft constructed using a particular design. None of these factors is overriding for a control demonstration. Furthermore, the preliminary evaluation conducted in the previous section indicates that adopting an exact scaled 'copy' of a particular structural design is not likely to be fruitful. As a consequence, little is gained by assessing individual designs; instead, their overall geometrical, structural and dynamical characteristics should be evaluated.

A decision matrix based on the above criteria (Table 3) and categories (Table 4) is given in Table 5 (see Section 2 for details concerning the construction of decision matrices). Based on this matrix, the best structure for the proposed control demonstration laboratory is a *Basic Compound Structure*. The total ratings of advanced 'plates' and advanced compound structures are also very high. As such, it is likely that a good basic compound structure might resemble somewhat an advanced plate or some advanced compound structure. This is taken into consideration in the design of the flexible structure being proposed in this report.

3.4 Basis for Proposed Flexible Structure

Prior to describing in some detail the proposed structure, it is timely to summarize the requirements and findings of the previous three sections. The detailed evaluation indicates that a basic compound structure is the preferred choice, while the preliminary evaluation suggests that an original concept rather than a scaled design will be required. Within these guidelines, the following characteristics are also deemed desirable:

(1) simplicity, while emulating third generation spacecraft.

(2) lowest natural frequency can be specified.

Table 5

Flexible Structure Decision Matrix

Flexible Structures / Evaluation Criteria	Criteria Weights	Basic Beams	Advanced 'Beams'	Basic Membranes	Advanced 'Membranes'	Basic Plates	Advanced 'Plates'	Basic Compound Structures	Advanced Compound Structures
Structural Flexibility	5	2	3	1	1	2	3	4	4
Originality	4	1	4	5	4	3	5	4	5
Adaptability	4	1	2	1	1	1	3	4	4
Size/ Weight	3	2	4	5	4	1	4	3	3
Complexity	2	5	3	4	3	4	3	2	1
Cost	1	5	2	4	2	3	2	2	1
Reliability/ Life	4	5	4	5	4	5	4	3	2
Ease of Modeling	3	5	2	4	2	3	2	2	1
Ease of Manufacture	2	5	3	4	3	4	3	2	1
1 - g Suitability	4	3	4	1	2	3	4	5	5
Acceptance of Hardware	4	5	3	1	1	5	3	4	4
Space Related	5	2	4	3	4	1	4	4	5
Total Rating	R_{max} 205	3.07	3.29	2.90	2.66	2.78	3.49	3.54	3.46

(3) clustered frequencies.

(4) structural adaptability.

(5) rotational rigid-body modes.

(6) structure permits both symmetric and asymmetric motions.

(7) easily discretized for analytical modeling.

(8) static deflection (caused by gravity) removable.

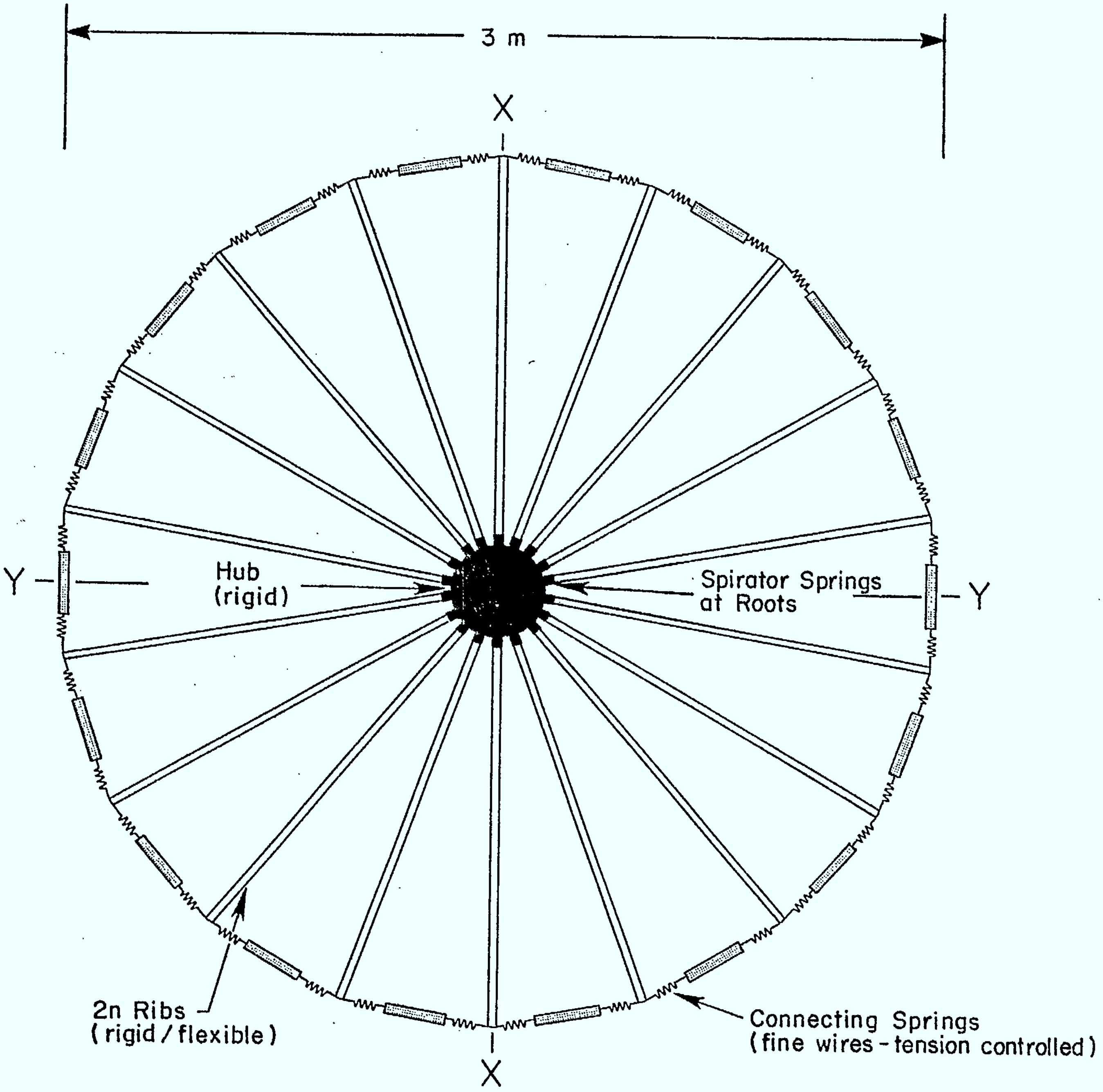
(9) supportable with a minimum of external contact (boundaries as free as possible).

(10) results applicable to most recent Canadian spacecraft project — MSAT.

3.5 Proposed Flexible Structure

A plan view of the proposed flexible structure (DAISY) is shown in Fig. 2. It consists of four basic components: a hub, a rib substructure, a hub-rib interface and a set of peripheral mass/spring connecting elements. The hub is rigid, but hollow to accept hardware (e.g. actuators). It is capable of accepting a maximum of 2n ribs where n is odd (tentatively $n = 9$). The decision to make n odd permits both symmetric and asymmetric rib patterns. For example, with a total of 18 ribs attached to the hub DAISY is symmetric about the lines X - X and Y - Y; however, the rib patterns about each line are not the same. If instead, only 9 ribs are attached at the hub, with the first rib attached to the hub along one of the two hub-Y lines and with the remaining ribs spared at 40^o intervals, then DAISY remains symmetric about Y - Y, but becomes asymmetric about X - X. The hub also contains a gimbal interface with which to mount the structure onto its support (see Section 3.6).

As stated above, the rib superstructure consists of a maximum of 2n ribs. Furthermore, as can be partially inferred from above, the ribs are removable and interchangeable. Nominally, each rib will be a rigid, 'planar' rod of constant tubular cross-section (to reduce weight) and consist of a homogeneous material. In the future, flexible (preloaded, if desired), curved rods of variable cross-section, and consisting of nonhomogeneous materials (including composites), may be considered. In fact, these two cases reflect



DAISY

Figure 2. Basic Components of Flexible Structure for Control Experiment

the limits of the possible rib designs. It is anticipated that the latter would only be achieved by progressing systematically from the former. The ability to introduce ribs of different geometric and structural properties greatly enhances the adaptability of the structure. In particular, given the removability of the ribs, one can introduce material asymmetry into DAISY while retaining geometric symmetry.

At this juncture, the reader may be wondering what the source of DAISY's structural flexibility is. The majority of it is to be found in the 'spirator' springs of the hub-rib interface. This motor spring is the torsional equivalent of a linear 'negator' spring which supplies a constant force over a reasonably large range of deflections. As diagramed in Fig. 3 the spirator spring requires several 'turns' to reach its maximum output torque. Furthermore, for an output torque T_0 near this maximum it is relatively insensitive to changes in the angular displacement θ. That is, the torsional spring stiffness ($dT_0/d\theta$) is small. Now, assume that each rib is pivoted at the hub-rib interface so that it can only move out-of-plane (i.e. out of and into the page, with DAISY oriented as shown in Fig. 2), and that the only elastic contact between the hub and the rib is an interconnecting spirator spring. Furthermore, assume that DAISY is suspended horizontally relative to the ground so that gravity causes DAISY's ribs to droop towards the ground. The 'multiple-turn' property of the spirator spring could then be used to generate a counteracting torque about the pivot to remove this static deflection while providing a low operating spring stiffness at each hub-rib interface. Hopefully, this spring stiffness could be tailored, through careful spring design, to achieve a first modal frequency for DAISY in the desired frequency range (0.06 to 3.14 rad/sec). This is the anticipated nominal scenario. Future hub-rib interfaces could incorporportate an additional spring to permit in-plane (i.e. rotary) motion of the ribs about the hub axis, or even a third spring to permit 'twisting' of the ribs as well. (For flexible ribs, these additions may not be necessary.)

The last major structural component, the set of peripheral mass/spring elements, also plays a key role in obtaining the desired dynamical characteristics from DAISY. These elements provide the weak dynamical coupling which prevents the ribs from all vibrating independently. With an

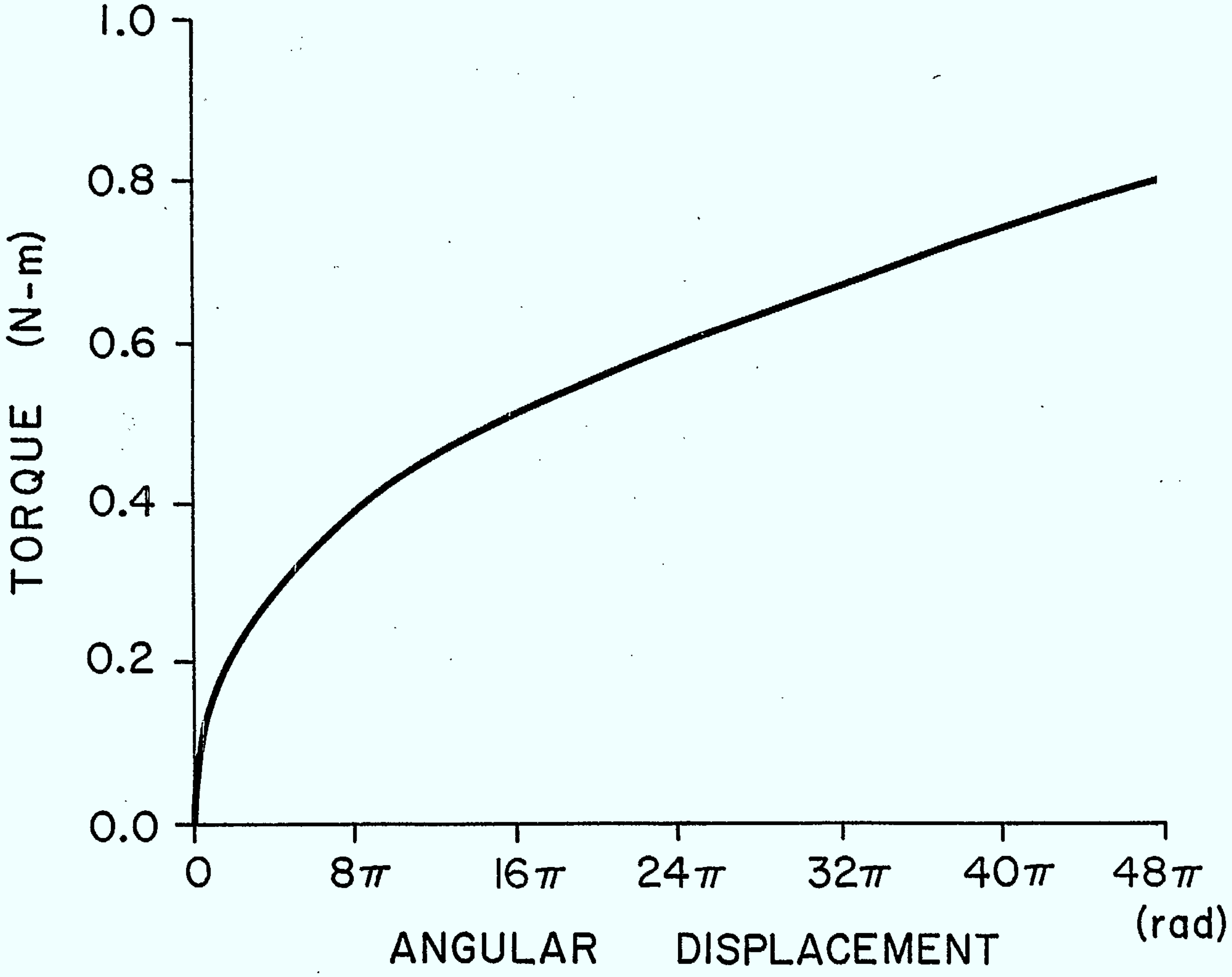


Figure 3. Typical Torque versus Angular Displacement Characteristic for a 'Spirator Spring' (after [Votta and Leidigh, 1959])

appropriate selection of masses and springs this should produce the desired 'clustering' of frequencies. The rigid circumferential masses are intended to be interchangeable and removable, while the connecting springs are to be adjustable (e.g. fine wire tensioned using a turnbuckle). Once again, this enables one to introduce structural asymmetries (either through material or stiffness variations). Finally, the intent is to have a variety of options for attachment of the peripheral elements to the ribs. Two possibilities are (i) fix the end of each element to each rib tip or (ii) interconnect the elements and then 'pass' the resulting 'loop' through eyelets at the tips of the ribs. In the future, interconnecting mass/spring elements could also be placed in concentric circles at selected radii between the hub and the tips of the ribs.

The initial estimate for the diameter of DAISY is between 2 and 3 m, with the hub taking up 10 per cent of the diameter. This size meets most laboratory space limitations while providing a workable structure from the viewpoint of human interaction. Also, a structure of this size should be capable of supporting the necessary actuator and sensor hardware associated with the control experiments.

3.6 Proposed Support Structure

It is proposed that DAISY be suspended (supported) horizontally (undeflected planform 'parallel' to the ground), from (by) a rigid support structure, using a two-degree-of-freedom *free* gimbal at the hub (i.e. no drive or gear mechanisms are present at the gimbal). The gimbal is intended to be as frictionless as possible, to permit two rotational rigid-body modes. This choice of gimbal is consistent with the anticipated choice of rib motions. For example, if the ribs were to experience rotary motion about the hub, as well as out-of-plane motions, then a three-degree-of-freedom gimbal would be more appropriate. It is expected that adding this extra degree of freedom would greatly complicate the gimbal design; however, a three-degree-of-freedom gimbal may be implemented in the future.

For the laboratory demonstration, a rigid support structure is one which has its first natural frequency well outside the anticipated control bandwidth and the relevant modal frequency range of DAISY. In this regard, it may be necessary to 'isolate' the support structure from disturbing input. Most of this 'noise' should have a frequency content much higher than either DAISY's structural frequencies or the control bandwidth. Hence, filtering, rather than active pneumatic isolation, should be adequate.

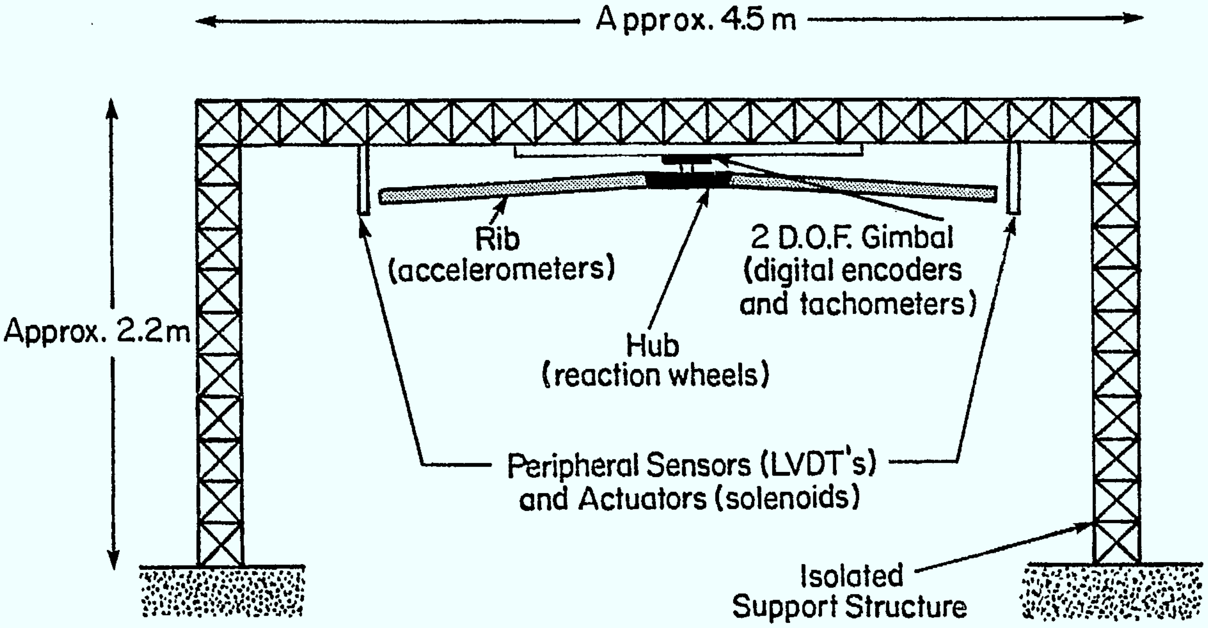
Two potential support-structure configurations are conceptualized in Fig. 4. The geometrical and structural design details and the exact dimensions have not yet been finalized; however, approximate dimensions are given, assuming a 3 m diameter for DAISY. Both options shown in the figure employ a support-mounted platform to accept the gimbal used to suspend (support) DAISY. This platform will also be used to support any sensor and actuator electronics which must be kept close to the actual hardware because of problems associated with line resistance, capacitance and inductance. For the present, this platform is viewed as being stationary; however, in the future, it could possibly be 'track'-mounted to permit two translational rigid-body modes.

Secondary platforms will also be required to support peripheral actuators and sensors. The actuators and sensors depicted in Fig. 4 are the culmination of evaluations detailed in the next two chapters. They are the suggested actuators and sensors for the control demonstration laboratory.

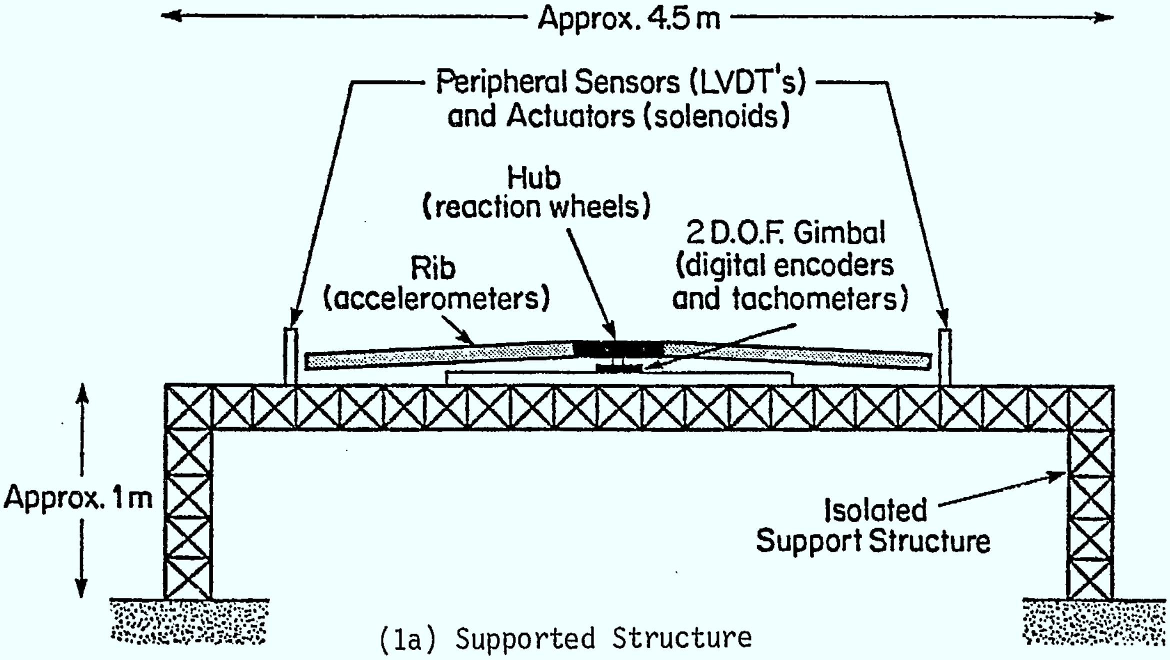
4. ACTUATOR EVALUATION

4.1 Evaluation Aims

One of the major objectives of this development program (see Section 1) is to gain 'hands on' experience with actual actuator and sensor hardware devices; however, these devices must be selected within the confines of the other major objectives. To be specific, these devices must be 'suitable' for use in control strategies applicable to Third-Generation spacecraft. For example, a 'suitable' actuator must supply a useful control input (force or torque) which, ideally, is representative (in character and in magnitude) of that anticipated for use in Third-Generation spacecraft (see, for example, [Lang and



(a) Suspended Structure



(1a) Supported Structure

Figure 4. Alternative Mountings for DAISY

Yuan, 1981]). In what follows, the 'suitability' of various candidate actuators will be assessed, while a similar procedure is performed, in the next section, to select sensors.

The selection of 'suitable' actuators is, of course, the primary aim of the actuator evaluation. There are, however, two secondary aims which should also be noted. The first is the desire to use well-established technology. It is felt that developing a new actuator is beyond the scope of the present development program. Furthermore, the experience to be gained with existing devices can only aid in such a future endeavor. The second desire is to avoid, initially, 'space qualified' actuators, because of their cost.

4.2 Preliminary Evaluation

As a preliminary stage to eliminating totally unacceptable actuators, a number of potential devices were categorized according to whether they were inertial reaction or relative motion actuators. An inertial reaction actuator provides an input to a structure by reaction against a mass which is either expelled from, or retained within, the structure (e.g. thrusters or reaction wheels). A relative motion actuator provides an input to a structure by the interaction of the structure with an external device or the external environment (e.g. shakers or magnetic torquers). Some relative motion actuators can also be used as intrastructural actuators to move one structural component relative to another, rather than to move the structure relative to inertial space. Here, both of these applications are considered.

A further sub-categorization of the candidate actuators was then performed based on whether the principal input supplied was a force or a torque. For example, while thrusters are often used to generate a torque to control spacecraft attitude, they do so by providing a force at some preselected 'moment-arm' location. Hence, for evaluation purposes, they are considered to be force rather than torque actuators.

At this juncture totally unacceptable actuating devices were discarded. For example, motors utilizing combustible fuels, heat engines, turbines and linear induction motors are of little use in this application.

Less obscure devices, such as variable-pressure 'air-bearing' tables and dual momentum ring actuators [Montgomery, 1979] were also excluded, based on such factors as being capable of functioning in a vacuum or being commercially available. The final list of actuator candidates subjected to a detailed evaluation is shown in Table 6. A brief description of each device cited in the table, and a number of pertinent references, can be found in Appendix C.

4.3 Detailed Evaluation

As in the previous section, prior to preparing decision matrices (see Section 2) for the actuators given in Table 6, it is necessary to establish the evaluation criteria upon which each actuator is to be judged. The chosen criteria, and the ideal situation in each case, are specified in Table 7. Based on these criteria, the decision matrices for the four actuator categories from Table 6 — Inertial-Reaction Force and Torque Actuators and Relative-Motion Force and Torque Actuators — are given in Tables 8, 9, 10 and 11, respectively. The *pivoted proof-mass* actuator proves to be the preferred inertial-reaction force actuator, while *reaction wheels* are the best inertial-reaction torque actuators. The highest rated relative-motion force and torque actuators are *solenoid drivers* and *brushless DC motors*, respectively. The evaluations conducted in Tables 8 through 10, were performed assuming a flexible-structure with the characteristics attributed to DAISY (see the previous section). The choice of a different structure would have consequences for some of the entries in the various decision matrices.

4.4. Proposed Actuators

Based on the above actuator selections and the proposed design for DAISY (see previous section), it is proposed that the primary actuators be three mutually-perpendicular reaction wheels located in the hub. Ideally, each reaction wheel axis will be aligned along one of DAISY's axes of symmetry, the X-X, Y-Y and hub axes (see Fig. 2). This configuration enables one to apply a torque in an arbitrary direction, depending on the magnitude of the torque applied to each wheel, while relating the input torque

Table 6

Actuator Candidates

Actuator Types	Force Actuators	Torque Actuators
1. Inertial Reaction Actuators	Linear-stroke Proof-mass Actuator Pivoted Proof-mass Actuator Gas Thrusters Compressed-Air Thrusters Electric Thrusters	Reaction Wheels Control Moment Gyros Momentum Wheels Pivoted Proof-mass Actuators
2. Relative Motion Actuators	Solenoid Drivers (low frey, input) Shakers (medium to high freq. input) Hydraulic Actuators Compressed-Air Devices Spring Devices Magnetic Coil Actuators	DC - Motors Brushless DC - Motors AC - Synchronous Motors AC - Squirrel-Cage Induction Motor AC - Wound-Rotor Induction Motor Spring Devices Magnetic Coil Actuators

Table 7

Evaluation Criteria for Actuator* Selection

EVALUATION CRITERIA	THE IDEAL
1. Input Supplied	- Supplied input appropriate to control strategies - Device is perfectly suited physically to applying particular input - Actuator does not disturb working environment (electrically, chemically)
2. Power Consumption	- Actuator uses as little power as possible (especially from viewpoint of applicability to spacecraft)
3. Efficiency	- Actuator is 100 per cent efficient in electrical and mechanical energy conversions (power in equals power out)
4. Contact/Non-Contact	- Actuator makes the minimal possible contact with the structure - Actuator does not interfere with motion of the structure
5. Size/Weight	- Actuator is as small and light as possible - Addition of actuator does not change the dynamics of the structure at all
6. Complexity	- Actuator is very simple with few moving parts
7. Cost	- Actuator is inexpensive
8. Reliability/Life	- Actuator very reliable - Longevity equal to at least that of the structure
9. Ease of Implementation	- Actuator can be incorporated into structural design with minimal amount of effort
10. Ease of Modeling	- Actuator can be easily modeled analytically
11. Availability	- Actuator is commercially available - No delivery delay - Service facility nearby
12. Space Related	- Actuator directly applicable to, or emulates, an actual spacecraft application

* 'Actuator' means the hardware device, including any accompanying electronics.

Table 8

Inertial-Reaction Force Actuators

Force Actuators Evaluation Criteria	Criteria Weights	Linear-Stroke Proof Mass	Pivoted Proof Mass	Gas Thrusters	Compressed-Air Thrusters	Electric Thrusters
Input Supplied	5	4	5	1	3	1
Power Consumption	5	4	4	3	3	2
Efficiency	4	3	3	5	4	4
Contact/ Non-Contact	4	4	4	3	3	4
Size/ Weight	4	3	4	3	3	2
Complexity	3	4	4	3	3	3
Cost	4	4	4	2	3	1
Reliability/ Life	2	4	4	3	3	4
Ease of Implementation	3	3	4	2	2	2
Ease of Modeling	1	4	4	3	3	2
Availability	2	1	1	4	3	3
Space Related	3	5	5	5	3	5
Total Rating	R_{max} 200	3.65	3.95	2.98	3.03	2.63

Table 9

Inertial-Reaction Torque Actuators

Torque Actuators Evaluation Criteria	Criteria Weights	Reaction Wheels	Control Moment Gyros	Momentum Wheels	Pivoted Proof Mass
Input Supplied	5	4	1	3	2
Power Consumption	5	3	2	3	4
Efficiency	4	3	3	3	3
Contact/ Non-Contact	4	3	3	3	3
Size/ Weight	4	4	2	3	4
Complexity	3	3	2	3	4
Cost	4	3	1	3	4
Reliability/ Life	2	3	2	3	3
Ease of Implementation	3	3	3	3	3
Ease of Modeling	1	3	3	3	3
Availability	2	4	4	4	1
Space Related	3	5	5	5	5
Total Rating	R_{max} 200	3.43	2.40	3.20	3.33

Table 10

Relative-Motion Force Actuators

Force Actuators Evaluation Criteria	Criteria Weights	Solenoid Drivers	Shakers	Hydraulic Devices	Compressed-Air Devices	Spring Devices	Magnetic Coils
Input Supplied	5	4	3	2	1	2	3
Power Consumption	5	4	3	3	2	4	3
Efficiency	4	4	4	3	2	2	3
Contact/ Non-Contact	4	4	4	3	3	3	3
Size/ Weight	4	4	3	3	3	5	4
Complexity	3	4	4	2	2	4	3
Cost	4	4	4	2	2	4	3
Reliability/ Life	2	4	3	3	3	3	5
Ease of Implementation	3	3	3	2	2	3	3
Ease of Modeling	1	4	4	3	3	3	3
Availability	2	4	4	4	3	3	2
Space Related	3	3	2	1	1	2	5
Total Rating	R_{max} 200	3.85	3.38	2.53	2.13	3.20	3.30

Table 11

Relative-Motion Torque Actuators

Torque Actuators / Evaluation Criteria	Criteria Weights	DC Motor	Brushless DC Motor	AC Synchronous Motor	AC Squirrel-Cage Induction Motor	AC Wound-Rotor Induction Motor	Spring Devices	Magnetic Coils
Input Supplied	5	3	3	1	2	1	2	3
Power Consumption	5	3	3	3	3	3	4	3
Efficiency	4	4	4	3	3	3	2	3
Contact/ Non-Contact	4	3	4	3	4	4	3	3
Size/ Weight	4	4	4	4	4	4	4	3
Complexity	3	3	3	3	3	3	4	3
Cost	4	2	2	2	2	2	4	3
Reliability/ Life	2	4	4	3	3	3	3	5
Ease of Implementation	3	2	2	2	2	2	3	3
Ease of Modeling	1	3	3	3	3	3	3	3
Availability	2	4	3	4	4	4	3	2
Space Related	3	4	5	3	4	4	1	5
Total Rating	R_{max} 200	3.20	3.33	2.73	3.03	2.90	3.03	3.20

directly to DAISY's body axes. This should simplify the analytics involved in mathematically modeling the dynamics. The use of three reaction wheels also recognizes the ultimate aim of providing a three-degree-of-freedom free gimbal at the hub.

Solenoid drivers, on the rib periphery, are suggested as secondary actuators. Potentially, each actuator could be made to simulate thrust inputs, by controlling the stroke of the device. Alternatively, these devices can be used to apply (known) disturbance inputs to the structure as an aid to assessing various control stategies. The exact solenoid configuration has not yet been established, but it will be consistent with the anticipated nominal out-of-plane motion of the ribs.

Future possibilities for actuators include pivoted proof-mass actuators located on the ribs and brushless DC motors to automatically adjust the hub-rib interface springs to remove DAISY's static deflection. Conceivably, these motors could be used as active control devices; however, this possibility is not presently being considered.

A brief tabular summary of the proposed actuators and their probable locations is provided, for easy reference, in Table 12.

5. SENSOR EVALUATION

5.1 Evaluation Aims

It is the primary aim of the sensor evaluation to select sensors 'suitable' for control systems demonstration. By 'suitable', it is implied, once again, that the chosen devices will provide useful outputs from the viewpoint of control system design, as well as be representative of sensors anticipated for use on Third Generation spacecraft (again, see, for example, [Lang and Yuan, 1981]). Also, since it is the dynamics of DAISY (see Section 3) that ultimately must be controlled, only motion sensors will be considered.

Again, the intent is to use 'time-tested' sensor designs, rather than attempting to develop new sensors. It is furthermore intended that the

Table 12

Probable Actuator Locations

Actuator / Component of DAISY	Inertial Reaction		Relative Motion	
	Force	Torque	Force	Torque
Gimbal				
Hub		Reaction wheels		
Hub/Rib Interface				(Brushless DC Motors)
Rib Structure	(Pivoted Proof-mass Actuators)			
Rib Periphery			Solenoid Drivers	
Peripheral Mass/ Sping Elements				

( ) - possible future actuators

chosen sensors avoid the integration or differentiation of their primary signals to provide a secondary output. (The former technique smooths the final output while the latter is often a source of high frequency 'noise'.) For example, a sensor providing a velocity-proportional output obtained by differentiating the signal from some displacement sensor would be deemed unacceptable, unless, of course, there is no other way of obtaining the velocity. Finally, while 'space qualified' sensors need not necessarily be used, the chosen sensors should be compatible with the space environment. This eliminates a number of flow, pressure, and cumbersome mechanical devices.

5.2. Preliminary Evaluation

Both translational and rotational motion sensors can be conveniently classified according to the dynamical quantity sensed. The three major classifications are displacement sensors, velocity sensors and acceleration sensors. There are also jerk sensors and devices which provide the integral of the displacement directly; however, they are not considered here. Each of the three major motion-sensor categories can further be subdivided according to whether the quantity sensed is a local relative measurement, a remote relative measurement or an absolute measurement. The concepts of relative and absolute measurement are well known. An absolute measurement is one made relative to an inertial frame, while a relative measurement is made relative to a noninertial frame; it is equivalent to the difference between two absolute measurements. Here, relative measurements will normally occur between two structural components of DAISY or between DAISY and the support structure. Within this context, the terms 'local' and 'remote' refer to the proximity of the two structural elements between which the related measurements are being made.

Given the above motion-sensor classifications and the aims cited in the previous section, a number of sensors can be excluded from detailed evaluation. These devices are summarized in Table 13. Those sensors judged acceptable and subjected to a further evaluation are given in Table 14. Also, a brief description of the acceptable sensor candidates is provided in Appendix D; however, no further details are provided for those sensors deemed

Table 13

Sensors Excluded

Sensor Type	Local Relative Sensors	Remote Relative Sensors	Absolute Sensors
1. Displacement Sensors	Sensors using flow or pressure pickups Sensors incorporating fluid, other than as a damping element Photoelastic and Brittle Coatings Sensors using chemical decomposition Signal integrating devices Ionization transducers Interferometric sensors Mechanical encoders	Radar sensors Earth-referenced sensors Photographic techniques Interferometric sensors Triangulation systems Radiometric sensors	Proof-mass sensors using pickups from first column
2. Velocity Sensors	Mechanical revolution counters Timers Mechanical fly-balls Signal integrating and differentiating devices	Doppler Radars Laser Velocimeters Stroboscopic devices	Proof-mass sensors using pickups from first column
3. Acceleration Sensors			Acceleration Threshold Switch Gravity-Reference Pendulous Gyros Liquid crystal Accelerometers Laser and interferometric devices Photo-electric Tranducers Electrostatic Accelerometers

Table 14

Sensor Candidates

Sensor Type	Local Relative Sensors	Remote Relative Sensors	Absolute Sensors
1. Displacement Sensors	Resistive Polentiometers Differential Transformers Variable Inductance/ Reluctance Devices Synchros and Induction Potentiometers Capacitance Pickups Piezoelectric Transducers Strain Gauges Piezoresistive Transducers Electro-optical devices Digital Encoders	Optically-Referenced Sys. Radio-Referenced Systems Difference Between Two Absolute Sensors	Basic Gryo Rate-integrating Gyr Proof-mass Pickups
2. Velocity Sensors	Electromagnetic Transducers Reluctive Pickups Capacitance Pickups Piezoelectric pickups Strain Gauge Piezoresistive pickups DC Tachometer AC Induction Tachometer AC Permanent Tachometer Eddy-Current Drag Cup Tachometer Digital Tachometer	Difference Between Two Absolute Sensors	Rate Gyros Proof-mass Pickups
3. Acceleration Sensors	Modified AC Tachometer	Difference Between Two Absolute Sensors	Potentiometric Inductive/LVDT Capacitive Piezoelectric Strain Gauge Piezoresistive Vibrating Wire Pendulous Gyroscope Inverse Wiedman Effect Re- or Null-Balanced

unacceptable. A number of useful references are also cited in the appendix.

5.3 Detailed Evaluation

Once more, the technique of using decision matrices (see Section 2) as a selection tool will be applied. And as before, the first stage in this procedure, after identifying the devices to be evaluated, is to specify the evaluation criteria by which the devices are to be judged. Several of the criteria applied to the actuators in the previous section are equally acceptable for assessing the merits of the various sensors. In particular, Criteria 4 through 12 of Table 7 apply directly, provided that the word 'actuator' is replaced by the word 'sensor'. Except for possibly the second criterion, the first three criteria in that table do not apply and should be replaced by those shown in Table 15. In fact, a rating for the power consumption of a particular sensor forms an implicit portion of the rating assigned to Criterion 3 of that table.

Tables 16 and 17 show the decision matrices corresponding to the displacement sensors given in Table 14. The preferred local-relative-displacement sensors are *differential transformers* for translational motions and *digital encoders* for rotational motions. For remote-relative-displacement measurements, *optically referenced systems* are the choice for both translational and rotational motions. Finally, absolute translational displacements are best measured using *proof-mass displacement pickups*, while (rebalanced) *rate-integrating gyros* are best suited for absolute-rotational-displacement measurements.

As is apparent from above, each sensor is rated according to how well it can sense translation and rotational motion. These ratings are performed independently and usually on different sensor designs, even though the same basic technical principles apply in each case. For example, a differential transformer designed to sense rotational motion is physically quite different from that used to sense translational motion; however, both devices are still differential transformers. To permit this dual rating of sensors to occur within a single decision matrix, a convention is adopted, whereby each entry in the matrix is divided into two triangles. The upper triangle contains the rating for the sensor when it is used to measure trans-

Table 15

Evaluation Criteria for Sensor* Selection

EVALUATION CRITERIA	THE IDEAL
1. Quantity Sensed	- Sensor input appropriate to control strategies - Device is perfectly suited physically to sensing desired quantity - Senses desired quantity directly
2. Performance/Accuracy	- Sensor has large range, is very accurate, has a high sensitivity and excellent resolution - Sensor is linear with no (zero and scale-factor) drift, no hysteresis or dead zones and has a zero threshold - Sensor has an excellent frequency response from DC to ultra-high frequencies and is void of any noise
3. Inpedance/Admittance	- Sensor has a low output impedance or a high output admittance relative to the input impedance/admittance of the device used to measure sensor output (ensures a valid measurement of sensor output) - Sensor input impedance high to ensure low power requirement for a given voltage

* 'Sensor' means the hardware device, including any accompanying electronics.

Table 16

Local Relative-Displacement Sensors

trans / rotn

Displacement Sensors / Evaluation Criteria	Criteria Weights	Resistive Potentiometers	Differential Transformers	Variable Inductance Reluctance Devices	Synchros and Induction Potentiometers	Capacitance Pickups	Piezoelectric Transducers	Strain Gauges	Piezoresistive Transducers	Electro-Optical Devices	Digital Encoders
Quantity Sensed	5	2 / 4	5 / 2	5 / 2	- / 4	4 / 3	4 / 3	4 / 3	4 / 3	4 / 2	3 / 5
Performance/ Accuracy	5	2 / 3	4 / 3	3 / 2	- / 4	3 / 2	1 / 1	3 / 3	4 / 4	4 / 3	5 / 5
Impedance/ Admittance	4	4 / 4	3 / 3	3 / 3	- / 3	1 / 1	1 / 1	1 / 1	2 / 2	3 / 3	4 / 4
Contact/ Non-Contact	4	2 / 2	4 / 4	4 / 4	- / 3	4 / 4	3 / 3	3 / 3	3 / 3	4 / 4	4 / 4
Size/ Weight	4	3 / 4	4 / 3	4 / 3	- / 4	5 / 4	5 / 4	5 / 4	5 / 4	2 / 1	2 / 4
Complexity	3	5 / 5	4 / 4	4 / 4	- / 4	4 / 4	4 / 4	4 / 4	4 / 4	3 / 3	3 / 3
Cost	4	5 / 5	3 / 3	3 / 3	- / 3	3 / 3	3 / 3	4 / 4	3 / 3	2 / 2	2 / 2
Reliability/ Life	2	3 / 3	4 / 4	4 / 4	- / 4	4 / 4	3 / 3	3 / 3	3 / 3	4 / 4	4 / 4
Ease of Implementation	3	2 / 2	3 / 3	3 / 3	- / 2	4 / 3	4 / 3	4 / 3	4 / 3	5 / 5	2 / 2
Ease of Modeling	1	5 / 5	4 / 4	4 / 4	- / 4	4 / 3	4 / 3	4 / 3	4 / 3	2 / 2	2 / 2
Availability	2	4 / 5	4 / 3	4 / 3	- / 4	4 / 1	4 / 1	5 / 1	4 / 1	3 / 1	4 / 4
Space Related	3	2 / 3	3 / 2	3 / 2	- / 3	1 / 1	2 / 2	3 / 3	3 / 3	4 / 3	2 / 4
Total Rating	R_{max} 200	3.05 / 3.65	3.78 / 3.05	3.65 / 2.93	- / 3.48	3.35 / 2.75	3.03 / 2.55	3.50 / 2.98	3.58 / 3.10	3.40 / 2.75	3.18 / 3.78

Table 17

Remote Relative-Displacement Sensors

trans / rotn

Displacement Sensors / Evaluation Criteria	Criteria Weights	Optically Referenced	Radio Referenced	Difference Between Two Absolute Sensors
Quantity Sensed	5	4 / 3	- / 3	3 / 3
Performance/ Accuracy	5	5 / 5	- / 4	3 / 3
Impedance/ Admittance	4	4 / 4	- / 4	3 / 3
Contact Non-Contact	4	4 / 4	- / 3	3 / 3
Size/ Weight	4	2 / 2	- / 3	3 / 3
Complexity	3	2 / 2	- / 2	3 / 3
Cost	4	1 / 1	- / 1	3 / 3
Reliability/ Life	2	4 / 4	- / 4	2 / 2
Ease of Implementation	3	2 / 2	- / 2	2 / 2
Ease of Modeling	1	1 / 1	- / 1	3 / 3
Availability	2	2 / 1	- / 3	4 / 3
Space Related	3	5 / 5	- / 4	4 / 4
Total Rating	R_{max} 200	3.23 / 3.05	- / 2.95	3.00 / 2.95

Absolute Displacement Sensors

Displacement Sensors / Evaluation Criteria	Criteria Weights	Basic Gyro	Rate-Integrating Gyro	Proof-Mass Displacement Pickups
Quantity Sensed	5	- / 4	- / 4	4 / 3
Performance/ Accuracy	5	- / 3	- / 4	3 / 2
Impedance/ Admittance	4	- / 3	- / 3	3 / 3
Contact Non-Contact	4	- / 4	- / 4	4 / 4
Size/ Weight	4	- / 2	- / 2	3 / 3
Complexity	3	- / 3	- / 2	4 / 4
Cost	4	- / 2	- / 2	3 / 3
Reliability/ Life	2	- / 4	- / 4	3 / 3
Ease of Implementation	3	- / 3	- / 3	2 / 2
Ease of Modeling	1	- / 4	- / 4	3 / 3
Availability	2	- / 4	- / 4	4 / 3
Space Related	3	- / 5	- / 5	4 / 4
Total Rating	R_{max} 200	- / 3.30	- / 3.43	3.35 / 3.10

lational motion, while the lower triangle contains the rating when it is used to measure rotational motion. It should also be stressed that, whenever a particular type of sensor can be designed using a variety of different techniques (e.g. electrical, magnetic and optical digital encoders), then the design which yields the highest total rating is the one represented in the decision matrix (for the above example, the ratings for optical digital encoders are given).

The decision matrices for the velocity motion sensors are provided in Tables 18 and 19. The preferred sensors, in this case, are *electromagnetic transducers* for detecting translational velocities and *digital 'tachometers'* for measuring rotational velocities. After preliminary evaluations, the only reasonable method for measuring remote relative velocities is the *difference between two absolute velocity sensors* — hence no detailed evaluation was needed to select this option. In this regard, a *proof-mass velocity* pickup is the chosen option for measurement of absolute translational velocity. The selected absolute rotational velocity sensor is a *rate gyro*.

The choice of sensors to measure relative accelerations is very limited. In fact, the *modified AC Tachometer* cited in Table 14 was never taken beyond the conceptual stage and is the only local-relative-acceleration sensor known to the author. Similarly, remote-relative-acceleration measurements can only be obtained by taking *the difference between two absolute acceleration sensors* (accelerometers). Luckily, a variety of absolute-acceleration-sensor designs exists. A decision matrix encompassing the contending accelerometer candidates is given in Table 20. The preferred accelerometers for measuring translational accelerations are those employing *piezoresistive* displacement pickups, while actively *rebalanced* accelerometers are preferred for detecting angular accelerations.

5.4 Proposed Sensors

A variety of potential motion sensors has been identified in the previous section. It is not the intent, here, to adopt all of these sensors for use in control demonstrations; instead, a selected few, capable of

Table 18

Local Relative-Velocity Sensors

trans / rotn

Velocity Sensor Evaluation Criteria	Criteria Weights	Electromagnetic Transducers	Reluctive Pickups	Capacitance Pickups	Piezoelectric Pickups	Strain Gauges	Piezoresistive Pickups	DC Tachometer	AC Induction Tachometer	AC Permanent Magnet Tachometer	Eddy-Current Drag Cup Tachometer	Digital Tachometer
Quantity Sensed	5	4 / 3	4 / 3	3 / 2	3 / 2	3 / 2	3 / 2	- / 5	- / 5	- / 5	- / 5	3 / 5
Performance/ Accuracy	5	4 / 3	3 / 3	3 / 2	1 / 1	3 / 3	4 / 4	- / 5	- / 3	- / 4	- / 5	5 / 5
Impedance/ Admittance	4	3 / 3	3 / 3	1 / 1	1 / 1	1 / 1	2 / 2	- / 3	- / 3	- / 3	- / 3	4 / 4
Contact Non-Contact	4	4 / 4	4 / 4	3 / 3	2 / 2	2 / 2	2 / 2	- / 3	- / 3	- / 3	- / 4	4 / 4
Size/ Weight	4	4 / 3	4 / 3	5 / 4	5 / 4	5 / 4	5 / 4	- / 3	- / 3	- / 3	- / 4	2 / 4
Complexity	3	4 / 3	4 / 3	3 / 3	3 / 3	3 / 3	3 / 3	- / 3	- / 3	- / 3	- / 3	3 / 3
Cost	4	3 / 3	3 / 3	3 / 3	3 / 3	4 / 4	3 / 3	- / 2	- / 2	- / 2	- / 2	2 / 2
Reliability/ Life	2	4 / 4	4 / 4	4 / 4	3 / 3	3 / 3	3 / 3	- / 4	- / 4	- / 4	- / 4	4 / 4
Ease of Implementation	3	4 / 3	4 / 3	4 / 3	3 / 2	3 / 2	3 / 2	- / 3	- / 3	- / 3	- / 3	2 / 2
Ease of Modeling	1	4 / 4	4 / 4	4 / 3	4 / 3	4 / 3	4 / 3	- / 3	- / 3	- / 3	- / 3	2 / 2
Availability	2	4 / 4	2 / 2	1 / 1	1 / 1	1 / 1	1 / 1	- / 5	- / 5	- / 5	- / 4	4 / 4
Space Related	3	2 / 3	2 / 3	1 / 1	2 / 2	3 / 3	3 / 3	- / 3	- / 3	- / 3	- / 3	2 / 4
Total Rating	R_{max} 200	3.65 / 3.23	3.43 / 3.00	2.90 / 2.45	2.50 / 2.18	2.93 / 2.60	3.05 / 2.73	- / 3.55	- / 3.30	- / 3.43	- / 3.70	3.18 / 3.78

Table 19

Absolute Velocity Sensors

trans / rotn

Velocity Sensor Evaluation Criteria	Criteria Weights	Rate Gyro	Proof-Mass Velocity Pickups
Quantity Sensed	5	- / 4	4 / 3
Performance/ Accuracy	5	- / 4	3 / 2
Impedance/ Admittance	4	- / 3	3 / 3
Contact/ Non-Contact	4	- / 4	4 / 4
Size/ Weight	4	- / 2	3 / 3
Complexity	3	- / 3	4 / 4
Cost	4	- / 2	3 / 3
Reliability/ Life	2	- / 4	3 / 3
Ease of Implementation	3	- / 3	2 / 2
Ease of Modeling	1	- / 4	3 / 3
Availability	2	- / 4	4 / 4
Space Related	3	- / 5	4 / 4
Total Rating	R_{max} 200	- / 3.43	3.35 / 3.10

Table 20

Absolute Acceleration Sensors

trans / rotn

Accelerometer Evaluation Criteria	Criteria Weights	Potentiometric	Inductive/ LVDT	Capacitive	Piezoelectric	Strain Gauge	Piezoresistive	Vibrating Wire	Pendulous Gyroscope	Inverse Wiedman Effect	Re- or Null-Balanced
Quantity Sensed	5	2 / 4	5 / 2	5 / 2	4 / 3	4 / 3	4 / 3	4 / 3	4 / -	- / 4	4 / 4
Performance/ Accuracy	5	1 / 1	4 / 3	3 / 2	1 / 1	4 / 3	5 / 4	5 / 4	5 / -	- / 3	5 / 5
Impedance/ Admittance	4	4 / 4	3 / 3	1 / 1	1 / 1	1 / 1	2 / 2	4 / 4	4 / -	- / 4	4 / 4
Contact Non-Contact	4	3 / 3	3 / 3	3 / 3	3 / 3	3 / 3	3 / 3	3 / 3	3 / -	- / 3	3 / 3
Size/ Weight	4	3 / 4	3 / 2	5 / 4	5 / 4	5 / 4	5 / 4	4 / 3	2 / -	- / 4	3 / 2
Complexity	3	4 / 4	4 / 4	4 / 4	4 / 4	4 / 4	4 / 4	3 / 3	2 / -	- / 4	3 / 3
Cost	4	4 / 4	3 / 3	3 / 3	3 / 3	4 / 4	3 / 3	2 / 2	1 / -	- / 4	2 / 2
Reliability/ Life	2	3 / 3	4 / 4	4 / 4	3 / 3	3 / 3	3 / 3	4 / 4	4 / -	- / 4	4 / 4
Ease of Implementation	3	4 / 4	4 / 4	4 / 4	4 / 4	4 / 4	4 / 4	4 / 4	4 / -	- / 4	4 / 4
Ease of Modeling	1	5 / 5	4 / 4	4 / 3	4 / 3	4 / 3	4 / 3	4 / 3	3 / -	- / 2	4 / 4
Availability	2	4 / 3	4 / 3	4 / 3	4 / 3	4 / 3	4 / 3	4 / 3	4 / -	- / 1	4 / 3
Space Related	3	1 / 1	2 / 2	2 / 2	2 / 2	3 / 3	4 / 4	4 / 4	4 / -	- / 4	4 / 4
Total Rating	R max 200	2.93 / 3.23	3.58 / 2.93	3.45 / 2.78	3.03 / 2.73	3.58 / 3.15	3.78 / 3.35	3.75 / 3.33	3.35 / -	- / 3.58	3.65 / 3.50

satisfying the anticipated control needs, are chosen. To measure the rigid-body motions, angular displacements and angular rates, of DAISY, at the gimbal, three digital encoders and three digital 'tachometers' will ultimately be required. If, initially, only a two-degree-of-freedom gimbal is adopted, then only two of each of these devices will be necessary. The other primary sensors will be piezoresistive accelerometers located on the rib structure (and possibly on the hub, to provide a remote relative measurement). The sensing axis of each accelerometer will be aligned with the hub axis, to be consistent with the planned nominal out-of-plane rib motion. In the future, if other rib motions are permitted, additional accelerometers would be required.

It should be mentioned that the chosen accelerometer design is based substantially on the information provided by [Lang, 1982]. Furthermore, it is understood by the author that such devices are to be procured for use in the controls laboratory at the Communications Research Centre in Ottawa. This provides an excellent opportunity to use thoroughly tested and accurately characterized sensors in the proposed control demonstrations.

To complement the solenoid actuators situated around the rib periphery, a secondary set of sensors comprised of linear variable differential transformers (LVDTs) is proposed. These sensors could be used in control strategies, or to ensure that 'known' disturbance inputs from the solenoid drivers are being supplied as expected. Again, these sensors would be oriented to be consistent with the planned rib motions and rigid-body motions of DAISY. In this respect, to obtain rotational information if, for example, DAISY rotates rigidly about the Y - Y axis in Fig. 2 and the tip rotation of the rib along the hub - Y line is required (for a rigid rib this should be the same as the angle measured by the encoders of the gimbal), paired LVDTs with differenced outputs would be required.

Potential future sensors include either rate-integrating or rate gyros on the hub and possibly a hub/rib-structure optical system to sense relative displacements. It may also prove useful to mount accelerometers on the peripheral mass/spring elements; however, for the present this possibility is not being entertained. Finally, in the event that the adjustment of the hub/rib-interface springs is ever automated, digital encoders are expected to play a role.

In a manner analogous to Table 12 for the actuators, a tabular summary of the proposed sensors and their probable locations is provided in Table 21.

6. CONCLUDING REMARKS

A conceptual design has been presented for a facility to study the control of flexible space structures, based on the objectives set out in Section 1. The chosen structure, "DAISY", illustrated in Fig. 3 of Section 3, should be capable of producing dynamical traits — a fundamental frequency in the range 0.06 to 3.14 rad/s (0.01 to 0.5 Hz), clustered frequencies and both rigid-body and flexible modes — that emulate Third-Generation spacecraft. Furthermore, the primary sensors and actuators adopted — digital encoders at the hub, accelerometers on the ribs and reaction wheels in the hub — are also directly applicable for use in this class of spacecraft. While the secondary sensors and actuators specified in the conceptual design are not directly applicable to spacecraft, their function is analogous. They are used to apply 'known' external disturbances to DAISY. Using solenoid drivers to apply these disturbances and linear differential transformers to detect their exact forms, it should be possible to simulate a variety of space-related inputs (e.g. thruster firings).

The combination of structure, sensors and actuators proposed herein forms a good conceptual design for a facility to study the control of flexible spacecraft, a design upon which detailed design can proceed.

Table 21

Probable Sensor Locations

Motion Sensor / Component of DAISY	Displacement			Velocity			Acceleration		
	Local Relative	Remote Relative	Absolute	Local Relative	Remote Relative	Absolute Relative	Local Relative	Remote Relative	Absolute
Gimbal	Digital Encoder		(Rate-Integrating Gyro)	Digital Tachometer		(Rate Gyro)			
Hub		(Optical System)						*	Accelerometers
Hub/Rib Interface	(Digital Encoders)								
Rib Structure		(Optical System)						*	Accelerometers
Rib Periphery	LVDTs								
Peripheral Mass/ Spring Elements									(Accelerometers)

( ) - possible future sensors
* - difference between absolute acceleration measurements on Hub and Ribs

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Appendix A

Dimensional Analysis for a Beam

In this appendix the Buckingham Pi Theorem is applied to obtain the important dimensionless quantities governing the frequencies of a vibrating beam. Formally, this theorem states that any function of n variables

$$f(q_1, q_2, q_3, \ldots q_n) = 0 \tag{A.1}$$

can be expressed in terms of (n - k) π-products

$$f(\pi_1, \pi_2, \pi_3, \ldots, \pi_{n-k}) = 0 \tag{A.2}$$

where each π-product is a dimensionless product of a selected set of k variables and one other; that is

$$\begin{aligned} \pi_1 &= f(q_1, q_2, \ldots, q_k, q_{k+1}) \\ \pi_2 &= f(q_1, q_2, \ldots, q_k, q_{k+2}) \\ &\vdots \qquad\qquad \vdots \\ \pi_{n-k} &= f(q_1, q_2, \ldots, q_k, q_n) \end{aligned} \tag{A.3}$$

and k is usually equal to the number of fundamental dimensions; however, under certain circumstances k can be less [Baker, Westine and Dodge, 1973]. One restriction, not normally mentioned, is that the chosen k variables must together contain all the fundamental dimensions.

For a beam, it is well known that the natural frequency ω depends on the mass m of the beam, the moment of area of the beam cross-section I, the elastic modulus of the beam material E, the length of the beam ℓ and the beam boundary conditions, which, for each modal frequency, can be represented by a nondimensional boundary-condition factor α (see, for example, [Thomson, 1972], where, $\alpha = \beta^2\ell^2$). Actually, once the length of the beam is specified, the density of the beam material ρ and the beam cross-sectional area A become more fundamental variables than the mass m. In any case, all of these variables involve at most three fundamental dimensions, namely, mass M, length L and time T. As a consequence, for the present analysis, $k = 3$. A summary of dimensions for the various beam variables (except for $m = \rho A\ell$) is given in Table A-1.

The equation corresponding to (A.1) for the above beam variables is

$$f(\omega, \ell, \rho, E, A, I, \alpha) = 0 \tag{A.4}$$

If we choose the k set of variables to be (ω, ℓ, ρ), then the $(n - k)$ π-products $(n = 7)$ become

$$\begin{aligned} \pi_1 &= f(\omega, \ell, \rho, E) \\ \pi_2 &= f(\omega, \ell, \rho, A) \\ \pi_3 &= f(\omega, \ell, \rho, I) \\ \pi_4 &= f(\omega, \ell, \rho, \alpha) \end{aligned} \tag{A.5}$$

Since each of these products must be dimensionless, it follows that

$$\begin{aligned} \dim(\omega^{a1}, \ell^{a2}, \rho^{a3}, E^{a4}) &\triangleq M^0L^0T^0 \\ \dim(\omega^{b1}, \ell^{b2}, \rho^{b3}, A^{b4}) &\triangleq M^0L^0T^0 \\ \dim(\omega^{c1}, \ell^{c2}, \rho^{c3}, I^{c4}) &\triangleq M^0L^0T^0 \\ \dim(\omega^{d1}, \ell^{d2}, \rho^{d3}, I^{d4}) &\triangleq M^0L^0T^0 \end{aligned} \tag{A.6}$$

Table A - 1

Dimensions $M^aL^bT^c$ for the Beam Variables

VARIABLE	SYMBOL	a	b	c
Natural Frequency	ω	0	0	-1
Length	ℓ	0	1	0
Density	ρ	1	-3	0
Elastic Modulus	E	1	-1	-2
Cross-sectional Area	A	0	2	0
Moment of Area	I	0	4	0
Boundary Factor	α	0	0	0

where dim (x) denotes the dimensions ($M^aL^bT^c$) of the product x. Now, to illustrate the procedure, consider the first relation in (A.6). Applying the dimensions from Table (A - 1) to this relation, one obtains

$$(T^{-1})^{a1}(L)^{a2}(ML^{-3})^{a3}(M^1L^{-1}T^{-2})^{a4} = M^oL^oT^o \qquad (A.7)$$

The implication is that

$$\begin{aligned} a_3 + a_4 &= 0 \\ a_2 - 3a_3 - a_4 &= 0 \\ - a_1 - 2a_4 &= 0 \end{aligned} \qquad (A.8)$$

from which it follows that

$$a_4 = -\tfrac{1}{2} a_1 \quad ; \quad a_2 = a_1 \quad ; \quad a_3 = \tfrac{1}{2} a_1 \qquad (A.9)$$

Letting $a_1 = 1$, the resulting π-product is

$$\pi_1 = \omega \ell (\rho/E)^{\frac{1}{2}} \qquad (A.10)$$

Similarly,

$$\pi_2 = A/\ell^2 \quad ; \quad \pi_3 = I/\ell^4 \quad ; \quad \pi_4 = \alpha \qquad (A.11)$$

so that (A.4) can be written in the form

$$f(\omega\ell(\rho/E)^{\frac{1}{2}}, A/\ell^2, I/\ell^4, \alpha) = 0 \qquad (A.12)$$

It should be apparent that the choice of π-products is not unique, given the free exponent a_1 in the above procedure. Furthermore, (A.8) could have been solved in terms of a_2, a_3 or a_4 just as easily as in terms of a_1. Another property of π-products is that new products can be formed by multiplying two existing products. Since they are non-dimensional they can also be inverted or raised to any power. In particular, it is useful to combine the first and fourth products in (A.12) to obtain

$$f(\omega \frac{\ell}{\alpha} (\rho/E)^{\frac{1}{2}}, \quad A/\ell^2, \quad I/\ell^4, \alpha) = 0 \tag{A.13}$$

It is also possible to manipulate the first, second and third products in (A.13), to obtain the more familiar form

$$f(\omega \frac{\ell^2}{\alpha} \left(\frac{m}{EI}\right)^{\frac{1}{2}}, A/\ell^2, I/\ell^4, \alpha) = 0 \; ; \tag{A.14}$$

However, when considering a variety of different structures, including membranes and plates, the first π-product in (A.13) is more useful.

For two different beams, A and B, (A.13) implies that *dynamic similarity* can be retained, for each mode i, provided that

$$\omega_A^i \frac{\ell_A}{\alpha_A^i} \left(\frac{\rho_A}{E_A}\right)^{\frac{1}{2}} = \omega_B^i \frac{\ell_B}{\alpha_B^i} \left(\frac{\rho_B}{E_B}\right)^{\frac{1}{2}} \tag{A.15}$$

for all i, and

$$\frac{A_A}{\ell_A^2} = \frac{A_B}{\ell_B^2} \; ; \quad \frac{I_A}{\ell_A^4} = \frac{I_B}{\ell_B^4} \; ; \quad \alpha_A^i = \alpha_B^i \tag{A.16}$$

In fact, (A.16) implies that the two beams are *geometrically similar.*

In general, (A.15) can be applied to a variety of structures, with ρ and E often taking on the roles of an 'effective' density and an 'effective' elastic modulus. However, depending on the structure chosen, the relations guaranteeing geometric similarity will vary.

Appendix B

Comments on the Geometric Scaling of a Cantilevered Rib

The ramifications of attempting to scale down a cantilevered rib 27.5 m in length to one only 1.5 m in length, while retaining geometric and dynamic similarity, but not the same rib material, are explored in what follows.

The pre-scaled rib, A, is assumed to be made of graphite/epoxy possessing a density $\rho_A = 2.51 \times 10^3$ kg/m^3 and an elastic modulus $E_A = 9.65 \times 10^{10}$ N/m^2, while the scaled rib, B, is to be composed of polyethylene plastic possessing a material density $\rho_B = 9.53 \times 10^2$ kg/m^3 and an elastic modulus of $E_B = 8.45 \times 10^8$ N/m^2. Given this information, and (A.15) from the previous appendix, the natural frequencies of rib B are related to those of rib A by the relation

$$\omega_B^i = \frac{\ell_A}{\ell_B}\left(\frac{\rho_A/E_A}{\rho_B/E_B}\right)^{\frac{1}{2}} \omega_A^i = 2.78\ \omega_A^i \qquad \text{(B.1)}$$

assuming identical rib boundary conditions. As a consequence, the lowest natural frequency of the scaled-down rib will be greater than twice that of the unscaled rib.

It is also interesting to inquire as to the anticipated static deflection for rib B under the influence of gravity. If one approximates the lenticular cross-section of the original rib by a thin-walled elliptical cross-section of height h, width w and thickness t, then the appropriate moment of area for the rib about its horizontal mid-plane is given by

$$I = \frac{\pi}{64} (wh^3 - w_o h_o^3) \tag{B.2}$$

where

$$w_o = w - 2t \quad ; \quad h_o = h - 2t \tag{B.3}$$

Now, if the rib is assumed cantilevered such that its horizontal mid-plane is parallel to the ground, it will 'droop' downward under the acceleration of gravity, with a resultant tip deflection of [Popov, 1968]

$$v_o = \frac{P_o \ell^4}{8\,EI} \tag{B.4}$$

assuming simple beam theory applies. The only undefined variable in (B.4) is the distributed force per unit length P_o acting on the rib:

$$P_o = \rho\,\pi\,(hw - h_o w_o)\,g \tag{B.5}$$

where the acceleration of gravity $g = 9.8$ m/sec^2. Equation (B.4) applies equally well to both the pre- and post-scaled ribs. Now, while it was never intended that the original rib should support its own weight under the influence of gravity, it is required that the scaled rib have this capability if it is to be used in a terrestial control laboratory. Based on dimensional considerations (see the previous appendix) it can be shown that, for geometrical similarity, the dimensions of ribs A and B obey the relations

$$\frac{h_B}{\ell_B} = \frac{h_A}{\ell_A} \quad ; \quad \frac{w_B}{\ell_B} = \frac{w_A}{\ell_A} \quad ; \quad \frac{t_B}{\ell_B} = \frac{t_A}{\ell_A} \tag{B.6}$$

Hence, given the average dimensions of rib A to be $(h_A, w_A, t_A) = (0.28, 0.102, 0.005)\,m$, the static deflection for rib B, from (B.4), is $v_{oB} = 1.3\,m$. This value is totally unbelievable for a rib 1.5 m in length! Furthermore, it violates the small deflection assumption inherent in linear beam theory. (Given that $v_{oA} = (\omega_B^i/\omega_A^i)^2 v_{oB}$, the static deflection for the original rib is $v_{oA} = 10.1\,m$.)

The absurdity of the above result suggests that the rib has likely buckled. This suspicion can be confirmed by considering the formula for the critical distributed lateral-buckling load of a thin (h/w small) cantilevered beam [Sechler, 1952]:

$$P_{cr} = \frac{12.85}{\ell^3} (B'C)^{\frac{1}{2}} \qquad \text{(B.7)}$$

where

$$B' = EI' \quad ; \quad C = (GJ)_e \qquad \text{(B.8)}$$

Here, I' is the moment of area about the axis parallel to the applied load and perpendicular to the axis about which the beam bends prior to buckling, while C, the 'effective' torsional stiffness, equals the product of the shear modulus G and the polar moment of area J. For both ribs A and B, w/h = 0.36, so that one is reasonably justified in applying (B.7). The required moments of area are, simply,

$$I' = \frac{\pi}{64} (hw^3 - h_o w_o^3) \qquad \text{(B.9)}$$

and

$$J = \frac{\pi}{64} [hw(h^2 + w^2) - h_o w_o (h_o^2 + w_o^2)] \qquad \text{(B.10)}$$

It therefore remains only to specify the shear modulus for polyethylene to be able to evaluate (B.7) for rib B. Unfortunately, a published value could not be found; however, for most substances $G \doteq E/2.5$. Assuming this relation to be valid for polyethylene, (B.7) yields a critical distributed lateral-buckling load of $P_{cr} = 0.19$ N/m for rib B. In comparison, form (B.5), the distributed load caused by gravity is $P_o = 0.32$ N/m, a load 1.7 times greater than P_{cr}. As expected then, rib B buckles under the influence of gravity. (The P_o/P_{cr} ratio for the original rib is given by $P_{oA}/P_{crA} = [(\omega_B^i/\omega_A^i)(\rho_A/G_A)^{\frac{1}{2}}/(\rho_B/G_B)^{\frac{1}{2}}]P_{oB}/P_{crB}$ and equals 2.2, for $G_A = 4.14 \times 10^9$ kg/m^3 [Wade, Sinha and Singh, 1979].)

The above results all suggest that to geometrically scale down an actual space structure to obtain a structure that is both dynamically similar and suitable for ground tests is not an easily achievable goal.

Appendix C

Actuator Candidates

This appendix complements Table 6 of Chapter 4, which lists the actuator candidates subjected to detailed evaluation. Here, a brief description of each actuator listed in the table is presented. To be consistent, the actuator categories cited in Table 6 are adopted in what follows.

C.1 Inertial-Reaction Force Actuators

C.1.1 Linear-Stroke Proof-Mass Actuator

This device is described by [Aubrun and Margulies, 1982] and is essentially a modified Ling Shaker (Mode 102). To quote the reference, "this actuator consists mainly of a powerful, heavy, permanent magnet and a cylindrical coil free to move in the magnet gap and attached to an output rod (see Fig. C.1). A bellows-type suspension system maintains the coil centered, while allowing for small axial motions." Unfortunately, this bellows suspension is stiff enough to cause striction problems at low force levels, but not stiff enough to support the weight of the magnetic under the influence of gravity. The first problem is solved using a velocity-feedback control system, while the second is solved by supporting the actuator in a pivoted cradle. The actuator output is a sinusoidally varing force which has no constant bias (i.e. the average force over one period is, ideally, zero).

C.1.2 Pivoted Proof-Mass Actuator

This device also is described by [Aubrun and Margulies, 1982]. It

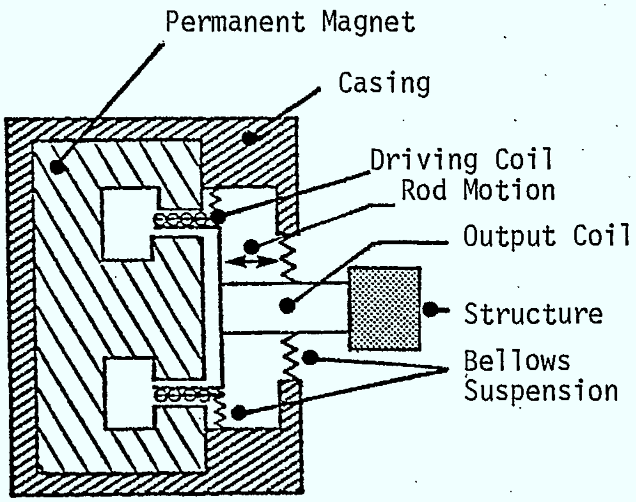


Figure C.1. Linear Proof-Mass Actuator (taken from [Aubrun and Margulies, 1982])

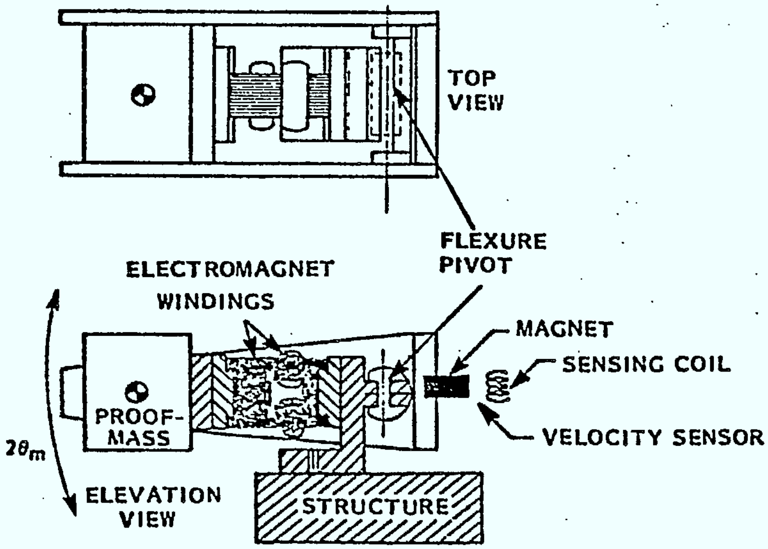


Figure C.2. Pivoted Proof-Mass Actuator (taken from [Aubrun and Margulies, 1982]) is a novel design that attempts to solve the problems of weight, stiff suspension, and susceptibility to transverse loads which are associated with the linear proof-mass actuator. A schematic of this actuator is given in Fig. C.2. Again to quote the above reference, "in this actuator the linear motion of the proof-mass is approximated by a small circle of arc about a pivot point realized by a flexure." The light electrodynamic motor shown in Fig. C.2 provides the actuation. Sinusoidally varying forces and torques can be produced about the structure attachment point; however, by correctly selecting the distance from this point to flexure point, zero torque is produced. A velocity-feedback control system is also required with this device. Both the linear and pivoted proof-mass actuators were developed by Lockheed Missiles and Space Company.

C.1.3 Gas Thrusters

As it is expected that the reader is familiar with the concept of a gas thruster, whereby a gaseous mass is expelled through a nozzle to elicit a reaction force, these devices will not be described in any detail here. Otherwise, the reader may consult the literature. Three very helpful references are [Sutherland and Maes, 1966], [Holcomb and Lee, 1972] and [Wertz, 1978].

C.1.4 Compressed-Air Thrusters

These devices are conceptually the same as gas thrusters, where the working chemical(s) in the latter are replaced by compressed air. For non-space applications these devices have the major advantage of not contaminating the work environment (unless that environment is an artificially maintained vacuum). However, the 'plumbing' requirements for these devices are not substantially less than those for gas thrusters.

C.1.5 Electric Thrusters

As stated [Barker, 1981], there are essentially three types of electric thrusters--electrothermal, electromagnetic and electrostatic (see Fig. C.3). Electrothermal thrusters expel an electrically heated propellent through an expansion nozzle to produce a reaction force, whereas electromagnetic thrusters expel an ionized gas of (both positively and negatively) charged particles accelerated by a magnetic field, and electrostatic thrusters expel charged particles (of one polarity) accelerated by a high voltage aperture. Details of various designs can be found in the above reference, as well as in [Sutherland and Maes, 1966] and [Holcomb and Lee, 1972].

C.2 Inertial-Reaction Torque Actuators

C.2.1 Reaction Wheels

A reaction wheel is a device which can store or transfer momentum between itself and the vehicle in which it is located. To accomplish this, a reaction wheel consists of two main components: a wheel spinning about a vehicle-fixed axis, and a torquing mechanism (e.g. a brushless direct-current motor) to change the spin rate relative to the vehicle and hence transfer momentum. There is, of course, a reaction torque applied to the vehicle. It is this characteristic that provides the actuation. (A minimum of three reaction wheels is required to produce

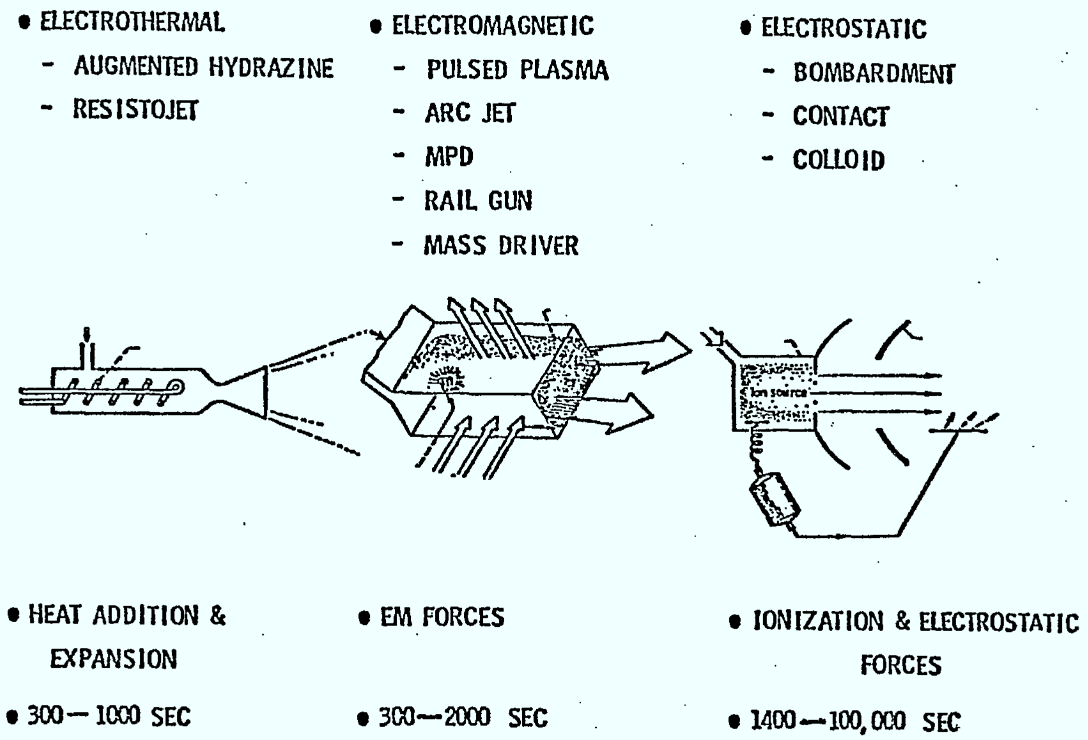


Figure C.3. Electric Thruster Concepts (taken from [Barker, 1981])

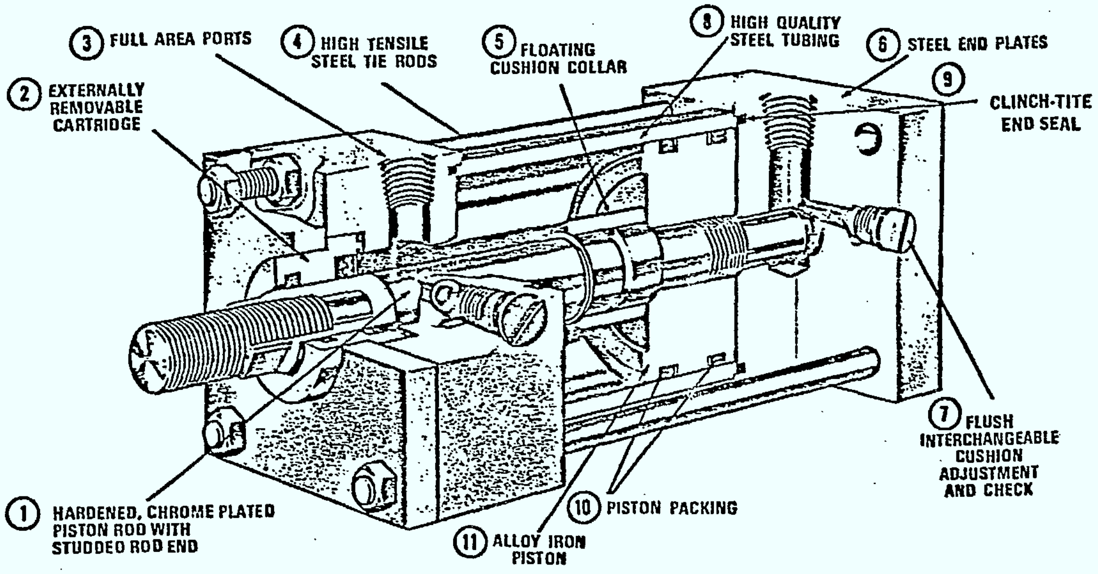


Figure C.4. A Typical Hydraulic Actuator (taken from [SP Manufacturing, 1976]) a torque in an arbitrary direction.) A reaction wheel is designed to operate at zero-bias; that is, the average stored angular momentum is zero. [Staley, 1981] and [Wertz, 1978] discuss a number of wheel designs which have 'flown' on actual spacecraft.

C.2.2 Control Moment Gyros (CMG)

A control moment gyro consists of a single- or a double-gimballed wheel spinning at a constant rate. By rotation about the input gimbal axes, the direction of the stored angular momentum vector can be changed, thus creating a control torque parallel to the output axis of the CMG (for a discussion of basic gyroscope principles, see [Greensite, 1970]). The magnitude of the applied torque is a function of the wheel spin rate and the gimbal rotation rates. As with reaction wheels, more than one CMG will normally be required to produce an arbitrary torque; however, this is not because they are intrinsically one-degree-of-freedom devices but because gimbal angles and rates may be restricted by design considerations. [Wertz, 1978] cites a number of illustrative examples.

C.2.3 Momentum Wheels

These devices are similar to reaction wheels; however, they are designed to operate at some non-zero bias. The examples mentioned by [Staley, 1981] and [Wertz, 1978] provide a reasonable cross-section of the range of available momentum wheels, their operating characteristics and their physical characteristics.

C.2.4 Pivoted Proof-Mass Actuator

For a description of this device, see section C.1.2. In this application, the torque generating capability of the device is retained; however, there is no means of counteracting the force applied by this actuator and hence it cannot actually apply a pure torque.

C.3 Relative-Motion Force Actuators

C.3.1 Solenoid Drivers

These are low-frequency actuation devices, many of which are similar in construction to linear proof-mass actuators (see Fig. C.1). They often do not have bellows suspensions, the magnet and the rod being independently attached to a stationary reference and the object to be actuated. The simplest designs involve an electromagnet which attracts a ferromagnetic rod; however, these tend to be 'one way' devices, with a restoring spring to 'reset' the rod. As a consequence, this type of device is only suited to applying an 'impulsive' disturbing force to the chosen structure, while the 'shaker' solenoid drivers can be made to perform this task and to provide a time-varying force as well.

C.3.2 Shakers

By the term 'shaker' we mean an actuator which is designed to provide a time-varying force of medium to very-high frequency. Such devices are commonly used in modal-testing to provide a variety of different inputs [Straud, Bonner and Chambers, 1978]. It may be somewhat misleading to differentiate between these devices and 'shaker' solenoid drivers; however, the distinction is made here to emphasize the difference in the frequency of the applied force, even though the two devices are virtually the same in construction, if not necessarily in size and weight.

C.3.3 Hydraulic Actuators

A typical hydraulic actuator is shown in Fig. C.4. The piston rod is activated by pumping fluid into one area port while simultaneously controlling the output of fluid from the other area port. Thus the piston-cylinder is always full of fluid being redistributed to cause the desired piston action. In truth, a reservoir and a valve system are also required, as is a powerful 'pump'. A variety of devices, with various operating pressures, outside diameters, and stroke lengths, are available (see, for example, [SP Manufacturing, 1976]). Some of these devices are no larger than the pivoted-proof-mass actuator described in C.1.2 (approximately 127 mm).

C.3.4 Compressed-Air Devices

These devices are essentially the same as hydraulic actuators with compressed air replacing the hydraulic fluid. Unfortunately, the 'stroke' of compressed-air devices cannot be controlled as accurately as for hydraulic devices. They are, however, smaller, lighter and provide a cleaner 'leak' environment than do hydraulic cylinders. Also, compressed-air actuators tend to have a smaller operating pressure (again, see, for example, [SP Manufacturing, 1976]).

C.3.5 Spring Devices

A number of 'motor' springs are available. These springs store potential (spring) energy when initially extended or compressed and, upon activation, they release this stored energy as kinetic energy. Depending on the spring design, either a variable or a constant force can be applied [Votta and Lansdale, 1952].

C.3.6 Magnetic Coils

Normally, in spacecraft applications, electromagnetic coils are used to interact with Earth's magnetic field [Wertz, 1978]. In the present application, localized permanent magnets on the support structure could be used to interact (attract/repel) with localized electromagnets on DAISY, or vice versa, to produce an input force.

C.4 Relative-Motion Torque Actuators

C.4.1 Direct Current (DC) Motors

These well-known devices operate on the principle of reversing the magnetic field of an armature as it rotates in a constant external magnetic field. The field reversals are performed to maintain the rotary motion of the armature. The designation "direct current" simply applies to the current supplied to the armature. To reverse the polarity of the armature, this current is supplied through a rotating commutator, using a system of brushes. Direct current motors are classified according to their type of magnetic field winding--series wound, shunt wound or compound wound [Anderson and Miller, 1981]. Each design has somewhat different starting characteristics, operating efficiencies and applied torque histories (see Fig. C.5).

C.4.2 Brushless Direct Current Motors

In this design of DC motor the brush commutators used in conventional DC motors are replaced by electronic or 'brushless' comutators. As with

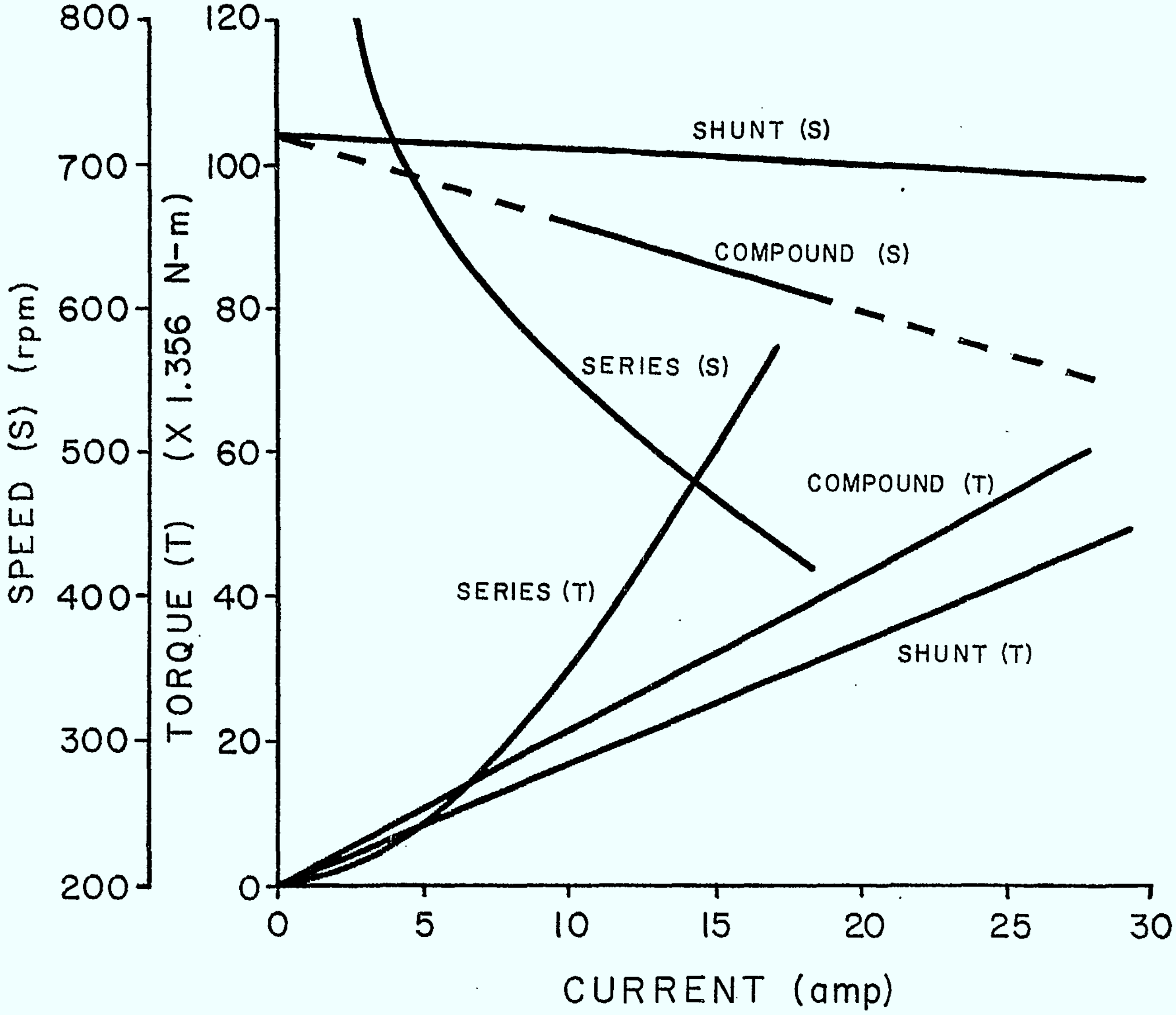


Figure C.5. Comparison of Direct Current Motors

(taken from [Anderson and Miller, 1981]) their conventional counterparts, these motors are efficient, provide a high torque at low speed and are easily controlled. These characteristics and the non-contact nature of the commutator have made the brushless DC motor one of the preferred torquing devices for use in reaction and momentum wheels [Wertz, 1978].

C.4.3 Alternating Current (AC) Synchronous Motors

In an alternating current motor the current is supplied directly to the armature by a pair of slip rings. The reversal in current then causes the desired reversal in the magnetic field of the armature. For most practical motors, however, the roles of the armature and external electric field are reversed. The armature is powered by direct current while the external magnetic field is segmented and supplied by alternating current, thus producing a 'revolving' magnetic field that 'drags' the armature along. The number of segments in the external field is equal to the number of poles (occurring in North-South pairs) on the armature. This number is important in determining the speed of the motor:

$$\text{rpm} = \frac{\text{frequency (Hz)} \times 60}{\text{number of paired poles}} \tag{C.1}$$

This speed is 'synchronized' with the frequency of the applied alternating current, hence the name "AC synchronous" motor.

C.4.4 AC Squirrel-Cage Induction Motors

In an induction motor no current is supplied directly to the armature of the AC motor. Instead, the currents in the revolving magnetic field induce a magnetic field in the armature. This armature always lags behind and can never rotate quite as fast as the revolving magnetic field. The difference in speed is called the 'slip'. A squirrel-cage induction motor derives its name from the fact that the armature resembles the wheel of a squirrel cage. This motor does not change its speed appreciably under loading and is suitable for applications requiring a medium or low starting torque.

C.4.5 AC Wound-Rotor Induction Motor

A wound-rotor induction motor differs from a squirrel-cage induction motor only in the design of its armature (or rotor). In wound-rotor motors, the armature consists of insulated coils of wire that are not permanently short-circuited, but are connected in regular succession to form a definite polar area having the *same* number of segments as the revolving magnetic field (stator) [Anderson and Miller, 1981]. For this motor any change in load results in a considerable change in speed. It does, however, have the capability of applying a large starting torque. In general, AC motors are less efficient than DC motors and their higher operating speeds tend to make them less reliable [Wertz, 1978].

C.4.6 Spring Devices

The comments in Section C.3.5 apply equally here, except that the resultant output is a torque rather than a force. [Votta and Leidigh, 1959] compare variable and 'constant' torque motor springs and cite several applications, one of which is as a primary driver unit.

C.4.7 Magnetic Coils

Again, the comments of a previous section, C.3.6, are relevant here;

however, now the magnetic dipole moment of a current loop would be used, within an external magnetic field, to generate a torque.

Appendix D

Sensor Candidates

A list of the sensor candidates subjected to detailed evaluation is provided in Table 14 of Section 4. This appendix provides a brief description of each sensor cited in that table. To be consistent, the sensor classifications adopted there will be retained in what follows. Also, rather than frequently referring to the three texts found most useful in assessing the various sensors, it is simplest to list them here. They are [Doebelin, 1975], [Norton, 1969] and [Norton, 1982]. This last reference is a revised version of Norton's 1969 text; however, each book contains unique information. Two references of secondary importance are [Holman, 1971] and [Cook and Rabinowicz, 1963]. Whenever necessary, a number of specialized references will also be given.

D.1 Local-Relative-Displacement Sensors

D.1.1 Resistive Potentiometers

A resistive potentiometer consists of a resistance element and a movable contact. The contact motion can be translational, rotational, or a combination of the two (helical motion devices). The distribution of the resistance element with respect to the contact governs the output voltage, which ideally is proportional to the displacement of the contact.

D.1.2 Differential Transformers

A typical differential transformer is shown in Fig. D.1. It consists of three windings, one which is excited at some frequency, and two

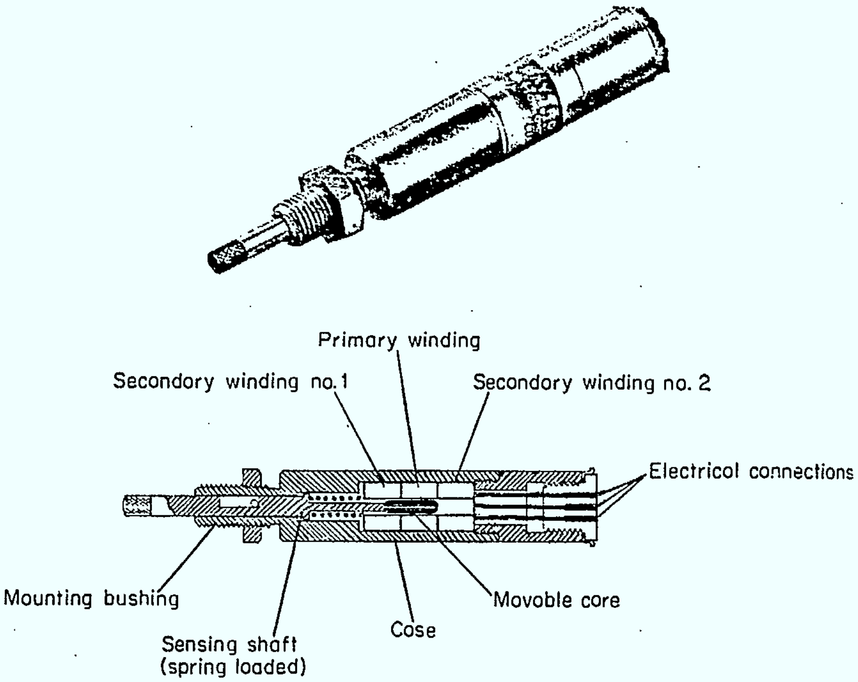


Figure D.1. A Typical Differential Transformer
[(taken from [Norton, 1969])

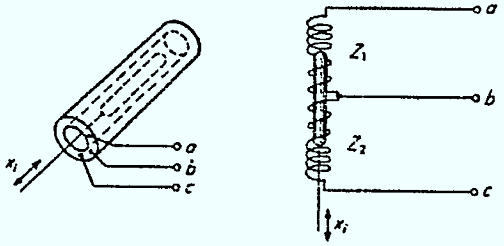


Figure D.2. A Typical Variable Inductance Device
(taken from [Doebelin, 1975])

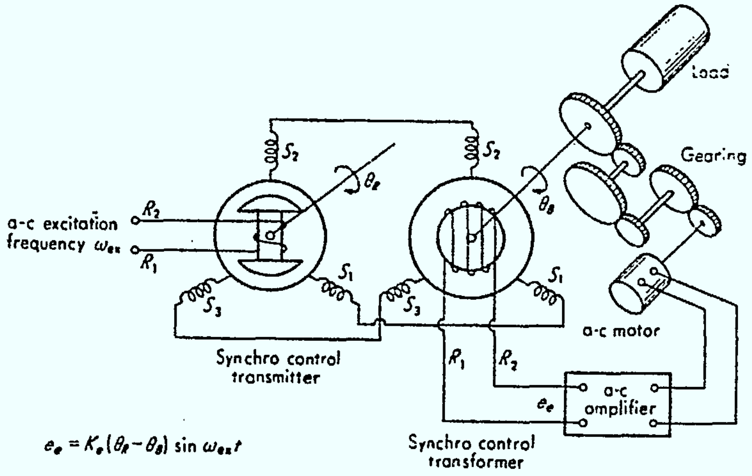
which have induced voltages at the same frequency; however, the position of the iron core relative to these two windings changes the amplitude of the induced voltage in each. The rod causes a larger mutual inductance in one coil than in the other. When connected in series opposition the resultant voltage from these two secondary coils is proportional to the displacement of the rod. This device is not 'null sensitive': the sign of a displacement cannot be detected unless a phase sensitive demodulator is used.

D.1.3 Variable Inductance/Reluctance Devices

In a variable inductance device the primary coil of the LVDT is missing while the two secondary coils are connected in a non-differencing bridge and excited directly (see Fig. D.2). A change in rod position changes the reluctance of the magnetic path in each coil, increasing one and decreasing the other. The change in reluctance causes a proportional change in inductance for each coil, a bridge unbalance and an output voltage ideally proportional to the displacement of the rod. The number of coils varies from device to device and from application to application. Again, a phase-sensitive demodulator is required to differentiate between a positive and a negative displacement.

D.1.4 Synchros and Induction Potentiometers

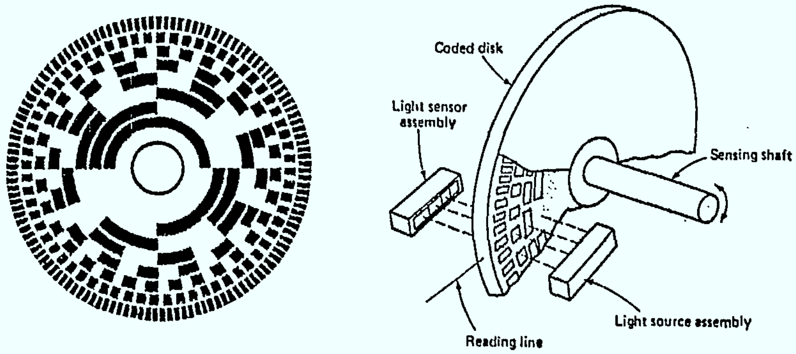
In a synchro-pair (see Fig. D.3) the primary winding on the rotor of the transmitting synchro is excited at some frequency. As this device senses the angle θ_R, the rotor interacts inductively with the three-phase stator whose windings are physically 120^o apart. The resulting electrical



$e_e = K_e(\theta_r - \theta_B) \sin \omega_{ex} t$

S_1, S_2, S_3 are stator coils
R_1, R_2 are rotor coils

Figure D.3. A Typical Synchro Pair
(taken from [Doebelin, 1975])



(a) Encoder Disc

Arabic number	(Natural) Binary		Gray (Binary)		Binary Coded Decimal (BCD)			
	Digital number	Code pattern	Digital number	Code pattern	Digital number		Code pattern	
					Tens	Units	Tens	Units
	8 4 2 1	2^3 2^2 2^1 2^0		G_3 G_2 G_1 G_0	8 4 2 1	8 4 2 1	2^0	2^3 2^2 2^1 2^0
0	0000	□□□□	0000	□□□□	0000	0000	□	□□□□
1	0001	□□□■	0001	□□□■	↑	0001	□	□□□■
2	0010	□□■□	0011	□□■■		0010	□	□□■□
3	0011	□□■■	0010	□□■□		0011	□	□□■■
4	0100	□■□□	0110	□■■□		0100	□	□■□□
5	0101	□■□■	0111	□■■■		0101	□	□■□■
6	0110	□■■□	0101	□■□■		0110	□	□■■□
7	0111	□■■■	0100	□■□□		0111	□	□■■■
8	1000	■□□□	1100	■■□□	↓	1000	□	■□□□
9	1001	■□□■	1101	■■□■	0000	1001	□	■□□■
10	1010	■□■□	1111	■■■■	0001	0000	■	□□□□

(b) Digital Code Structures

Figure D.4. A Typical Digital Encoder
(taken from [Norton, 1982])

output causes a torque on the rotor of the receiving (or control) synchro. This torque is only zero when the angular displacement θ_B of the second synchro equals θ_R. The output from the second synchro, the error voltage $e_e = \theta_R - \theta_B$, can then be measured directly or used in a feedback control system. Inductive potentiometers involve one (or two) rotor(s) and one (or two) stator coil(s). The rotor coil is excited, which induces a voltage in the stator. The relative position of the two coils changes their mutual inductance and hence an output voltage proportional to angular displacement is obtained.

D.1.5 Capacitance Pickups

Very simply these devices rely on the change in capacitance brought about by a change in position. The change in capacitance can be converted into an electrical output related to the original change in position. The simplest design measures changes in the air-gap between two parallel 'plates' caused by the motion (displacement) of one plate relative to the other. Often, the 'plate' which moves is a conducting surface on the object whose displacement is to be detected, while the second 'plate' is a probe rigidly mounted on some reference structure. Generally, capacitance pickups require high-input-impedance electronics and are sensitive to the length and position of connecting cables.

D.1.6 Piezoelectric Transducers

When deformed, certain materials generate an electric charge. This is known as the piezoelectric effect. It is reversible, in that the application of an electrical charge will result in a deformation for these same materials (e.g. quartz). As a consequence, piezoelectric transducers are used in both actuators and sensors. A major drawback of these types of sensors is that they cannot detect a static displacement; nor is their response at low frequencies acceptable. Normally, they are used in accelerometer designs for high-frequency and shock vibration studies.

D.1.7 Strain Gauges (Metallic)

Sensors employing strain gauges rely on the fact that the resistance of a conductor changes when an applied stress causes a dimensional change in the conductor. The actual quantity measured is strain, which can be converted to an output voltage proportional to displacement by using an electrical bridge. It is also necessary to include a temperature compensation bridge, complete with a dummy (non-transducing) strain gauge, to remove spurious thermal strain readings. Also, the output levels from these devices are usually low enough that electronic amplification is required. A common technique used to amplify the signal mechanically, prior to introducing electronics, is to use the tip-deflection of a flexible cantilevered beam, with strain gauges mounted at its root, to measure displacement.

D.1.8 Piezoresistive Transducers

While a change in the dimensions (length and cross-sectional area) of a actuator will result in a change in resistance, there is also a fundamental property of materials, known as piezoresistance, which relates a change in the resisitivity directly to the mechanical strain present. For metallic strain gauges, the former effect dominates. For semi-conductor strain gauges, it is the piezoresistance of the material that is most important. Semi-conductor strain gauges tend to have a larger gauge factor (change in resistance normalized by original resistance and divided by strain) than metallic strain gauges; however, amplification and temperature compensation are still required. Piezoresistive transducers, unlike piezoelectric transducers, can detect a static displacement.

D.1.9 Electro-Optical Devices

These devices use optics and electronics to detect displacements over short distances. Early designs required a light beam to be focused on a moving edge. Depending on the intensity of the light reflected from this edge, it could be deduced whether the edge had been displaced and by how much. Later designs use targets and election imaging tubes with rebalancing electronics (electronics to force a deflected target image to return to its undeflected location on the imaging tube). By detecting the voltage required to rebalance the image, the displacement of the target can be deduced. [Smalley, Tessarzik and Badgley, 1975] give basic dimensions and characteristics for commercially-available photo-optical proximity sensors based on the earlier technology, but updated to use fiber-optics rather than using lens-optics and specially constructed cathode-ray tubes.

D.1.10 Digital Encoders

There are essentially four types of encoders: mechanical, electrical, magnetic and optical. Each device 'encodes' a displacement using a binary coding bit pattern. For a particular displacement a certain number of bits are 'turned-on' within this pattern, while the others remain 'turned-off' (see Fig. D.4). In an optical encoder a bit is 'turned-on' when the light intensity from a known source exceeds the threshold level for the appropriate photosensitive portion of the binary coding bit pattern. Unfortunately, misalignments can cause bits to be erroneously 'turned-on'. To avoid this, a cyclic binary code (e.g. Gray code) is often used. A detailed explanation of these devices can be found in [Woolvet, 1977].

D.2 Remote-Relative-Displacement Sensors

D.2.1 Optical-Referenced Systems

A number of systems capable of detecting the distance to remote targets have recently been developed. Some detect the repetition rate of light pulses returning from passive targets [Berdahl, 1981]; others compare the returning pulse pattern with a reference pattern (much like a star mapper) to detect displacement [McLauchlan, 1981], while yet others use sun-sensor photosensitivity arrays to detect angular displacement [Collyer, 1980]. Systems employing active targets have also been suggested [Neiswander, 1981]. Unfortunately, the majority of these devices are still at the development stage and are not yet commercially available.

D.2.2 Radio-Referenced Systems

Here we are concerned only with radio frequency (RF) sensors which can detect angular displacement. Conceptually, an antenna is aligned with some transmitting beacon of known location (relative to the non-gimbaled position of the antenna reflector). The antenna reflector gimbal angles are then measured using local-relative displacement sensors, from which the angular displacement of the vehicle supporting the reflector can be deduced. A discussion of these sensors as applied to communications spacecraft can be found in [Perrotta, 1976] and [Staley, 1979].

D.2.3 Difference Between Two Absolute Sensors

Very simply, absolute sensors employ a proof mass, whose motion (relative to some case that has been displaced relative to inertial space) is measured using a local-relative-displacement sensor. That is, the proof mass remains essentially stationary while the absolute displacement of the case (and the structure to which it is attached) is measured. The difference between the measurements from two such sensors located on different portions of a given structure is a measurement of the motion of one sensor relative to the other. To permit the measurement of large displacements, often the proof-mass is rebalanced (i.e. forced to remain stationary relative to its case). The input required to accomplish this can then be processed to obtain a displacement measurement.

D.3 Absolute-Displacement Sensors

D.3.1 Basic Gyros

A good description of various gyroscope sensors is given by [Savage, 1978], including electrostatic, tuned-rotor and laser gyros; however, here we are only concerned with the basic gyro. A two-axis position gyro is shown in Fig. D.5. This gyro can measure the angles θ and ϕ about two perpendicular directions, because the spin-axis of the gyroscope rotor remains fixed relative to inertial space (neglecting bearing friction). A good description of the detailed equations governing gyroscopes can be found in [Greensite, 1970]. Again, to permit the measurement of larger angular displacements, these devices are often rebalanced using a feedback control loop, the rebalancing output from which can be converted into a measurement of

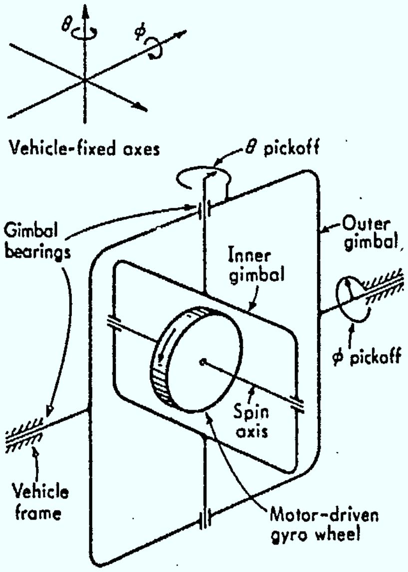


Figure D.5. A Two-Axis Free Gyro (taken from [Doebelin, 1975])

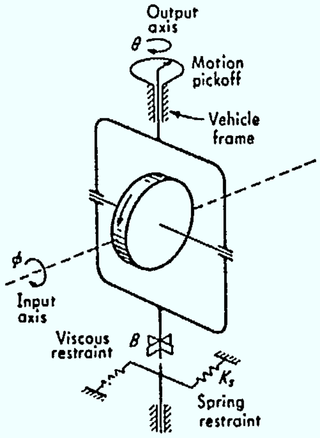


Figure D.6. A Single-Axis Restrained Gyro (taken from [Doebelin, 1975])

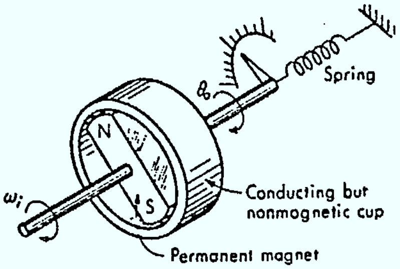


Figure D.7. An Eddy-Current Drag-Cup Tachometer (taken from [Doebelin, 1975]) angular displacement. This technique also produces greater accuracy.

D.3.2 Rate-Integrating Gyro

These devices are single-axis gyroscopes restrained by a spring and some damping about the output (or sensing) axis. By choosing a weak spring the damping term can be made to be dominant in the transfer function. This produces an output angle θ (see Fig. D.6) proportional to the integral of the angular velocity about the input axis. Hence the name, "rate integrating gyro". To obtain three-dimensional measurements, at least three such gyros are required. The tendency to use four gyros is motivated by the desire to increase reliability through redundancy. Both [Wertz, 1978] and [Greensite, 1970] contain good descriptions of this type of gyro. Again, rebalanced designs provide the best accuracy.

D.3.3 Proof-Mass Displacement Pickups

The basic concepts governing these devices have been discussed in Section D.2.3 and are not repeated here; however, it should be emphasized that the proof-mass in these devices is supported by a suspension system with a known spring stiffness and damping coefficient (both of which are small).

D.4 Local-Relative-Velocity Sensors

D.4.1 Electromagnetic Transducers

Electromagnetic transducers are similar to variable inductance/ reluctance devices. For example, if one replaces the ferromagnetic rod in Fig. D.2 with a permanent magnet, then the resultant output is proportional to velocity rather than to position.

D.4.2 Reluctive Pickups

In this sensor a toothed rotor is passed between a C-core transformer with an excitation winding on one leg of the core and an output winding on the other leg. When a tooth of the rotor passes between the legs, the change in reluctance causes a change in the induced output. The frequency of this change is then a measure of the rotational speed, since the number of teeth on the rotor is known a priori. Sensors equipped to measure linear speed also exist. One drawback of these sensors is that they cannot detect direction.

D.4.3 Capacitance Pickups

These devices, like capacitance displacement sensors, use the change in capacitance of a variable capacitor to detect the desired physical quantity—here, velocity. This requires an AC bridge that is periodically unbalanced by the change in capacitance in one of its two arms. The frequency of this change is related to the speed. Again, a secondary device is required to detect direction.

D.4.4 Piezoelectric Pickups

These sensors are conceptually similar to capacitance velocity sensors, except that they use piezoelectric pickups as the transducing elements (see Section D.1.6).

D.4.5 Strain Gauges (Metallic)

Little can be added to the discussion of strain gauges given in Section

D.1.7, except that, by design, velocity sensors detect the change in the strain with respect to time rather than the absolute strain.

D.4.6 Piezoresistive Pickups

Velocity sensors based on the use of piezoresistive pickups have been developed. They simply rely on the time variation in material resistance to detect velocity (see D.1.8).

D.4.7 DC Tachometer

A direct-current tachometer uses either a permanent magnet or a separately excited winding as its stator and a conventional generator winding on the commutator-equipped rotor to produce an output voltage proportional to speed. This type of tachometer detects a change in direction by a change in the polarity of the output voltage.

D.4.8 AC Induction Tachometer

An alternating current induction tachometer is essentially a squirrel-cage induction motor (see Section C.4.4) in which the primary (input) stator coil is excited using alternating current and the secondary (output) stator coil produces an output voltage, with the same frequency as the excitation, but with an amplitude proportional to the speed of the squirrel-cage rotor. A phase demodulator is necessary to detect direction, which appears as a 180^{0} shift of phase in the output voltage.

D.4.9 AC Permanent-Magnet Tachometer

These tachometers use the flux changes between a permanent-magnet rotor and a stator winding to provide an alternating output which varies with rotary speed both in amplitude and frequency. The output signal can be signal-conditioned to remove either one of these dependences while leaving the remaining one intact. Again, phase-demodulation is required to detect direction.

D.4.10 Eddy-Current Drag-Cup Tachometer

An eddy-current drag-cup tachnometer is shown in Fig. D.7. A voltage is induced in the conducting cup by the rotation of the permanent magnet. This voltage produces eddy currents within the cup which interact with the magnet to cause a torque on the cup. The cup turns through an angle θ_0 until balanced by the torque from the restraining spring. In the steady state, θ_0 is proportional to the angular rate ω_i. The device can be operated dynamically; however, its response is governed by the spring stiffness, the rotary inertia of the cup portion of the tachometer and the damping caused by the eddy-current coupling between the magnet and the cup.

D.4.11 Digital Tachometers

These devices 'count' the number of pulses reflected from a moving disk of fixed targets, or they count the number of pulses from an *incremental* digital encoder, over a period of time. An incremental digital encoder produces equally spaced pulses from one or more tracks; the pulses are fed to and accumulated in an up/down counter. The resultant count is a measure of the displacement. An *absolute* digital encoder is described in Section D.1.10. For measuring displacement it has the advantage that if an error occurs, it occurs only once. In contrast, once an error occurs in the count of an incremental encoder, it perpetuates. Still, most tachometer designs are better suited to the use of an incremental encoder. Also, while the above counting technique produces an average velocity, techniques do exist for measuring instantaneous velocity [Woolvet, 1977].

D.5 Remote-Relative-Velocity Sensors

D.5.1 Difference Between Two Absolute Sensors

The comments of Section D.2.3 apply equally here, except that a local-relative-velocity sensor is used to sense the velocity between the proof mass and its case.

D.6 Absolute-Velocity Sensors

D.6.1 Rate Gyros

Rate gyroscopes are single-axis, restrained devices very similar to rate-integrating gyros (see Section D.3.2); however, in these gyroscopes the stiffness of the restraining spring (while very small) is chosen to dominate the damping in the gyroscope. As a consequence, the output of the gyroscope is proportional to the angular velocity about the input axis, rather the integral of the rate. At least three gyroscopes are required to define completely the angular velocity vector and once again the best performance is achieved by rebalancing the gyroscope.

D.6.2 Proof-Mass Velocity Pickups

Again, the underlying principles involved in these devices are outlined in Section D.2.3; however, here, a local-relative-velocity sensor replaces the local-relative-displacement sensor. Recall that the proof mass is supported by a suspension system with a known spring stiffness and damping coefficient. These quantities are important in determining the frequency response of the sensor.

D.7 Local-Relative-Acceleration Sensors

D.7.1 Modified AC Tachometer

This sensor is based on the tachometer described in Section D.4.8. If the primary stator coil of that design is supplied with direct current rather than alternating current, then the output would be a varying direct current proportional to the relative acceleration of the rotor and the stator; however, this device is not commercially available, nor is there enough detailed information available to assess the capabilities of such a sensor.

D.8 Remote-Relative-Acceleration Sensors

D.8.1 Difference Between Two Absolute Sensors

Once again, the reader is referred to Section D.2.3 where now the absolute sensor is one of the accelerometers discussed in the next section.

D.9 Absolute-Acceleration Sensors (Accelerometers)

A comprehensive discussion of all of the accelerometers considered below can be found in [Lang, 1982]. This document cites a number of excellent references and gives examples of commercially available accelerometers. An accelerometer consists of a proof-mass supported by a suspension system with a known equivalent spring stiffness and damping coefficient. For a constant acceleration the displacement x of the proof mass m is proportional to the acceleration a by virtue of the fact that the force f on the proof-mass obeys the relation $f = ma = kx$, where k is the spring stiffness of the suspension system. (When the acceleration is changing, damping plays a role.) Hence, for accelerometers which are not rebalanced (i.e. do not include a feedback control loop that applies f to keep the proof mass undeflected, with the acceleration being inferred from this applied force) a local-relative-displacement sensor is required to detect acceleration. As a consequence, accelerometers can be classified according to the displacement sensor incorporated in their designs.

D.9.1 – D.9.6 Potentiometric, Inductive/LVDT, Capacitive, Piezoelectric, Strain Gauge and Piezoresistive Accelerometers

The local-relative-displacement sensor used in each of the above accelerometers has been described previously (see Sections D.1.1, D.1.3/D.1.2, D.1.5, D.1.6, D.1.7 and D.1.8), and as such, no further comment is required.

D.9.7 Vibrating Wire Accelerometer

In this transducer acceleration causes a change in the resonant frequency of a wire kept in tension within a permanent magnetic field. The wire is caused to vibrate by passing a current through it. The displacement of a proof mass attached to the wire creates an asymmetric tension distribution, from which the applied acceleration can be deduced. The output is frequency-modulated and requires demodulation to obtain a DC output.

D.9.8 Pendulous Gyroscope Accelerometer

This device uses the acceleration of a pendulous proof mass to generate a torque about the input axis of a rate-integrating gyroscope. The resulting precession rate of the gyro is proportional to the applied acceleration.

D.9.9 Inverse Wiedman-Effect Accelerometers

These accelerometers detect a shift in the direction of an induced magnetic field as the consequence of an applied torque. As described in [Lang, 1982] one simple configuration uses alternating current passed along a ferromagnetic rod to produce a circumferential magnetic field. An applied torque alters this field with the result that a voltage is induced in a solenoid coil wrapped about the rod. This voltage, which is caused by the axial component of the skewed magnetic field, is proportional to the amplitude of torque and hence to the applied acceleration.

D.9.10 Rebalanced or Null-Balanced Accelerometers

The concept of a rebalanced accelerometer was mentioned in the introduction to Section D.9. Rather than allowing the proof mass to deflect, a balancing force (torque) is provided, using a feedback control loop, to hold the proof mass at null. The force (torque) required to accomplish this can be directly related to the applied acceleration. The majority of the accelerometer designs discussed in this section can be rebalanced. These types of accelerometers are more sensitive and more linear than non-rebalanced designs.