THE RESULTS OF A LIMITED STUDY OF APPROACHES TO THE DESIGN, FABRICATION, AND TESTING OF A DYNAMIC MODEL OF THE NASA IOC SPACE STATION

EXECUTIVE SUMMARY

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ENGINEERING INCORPORATED
Hampton, Virginia (804) 865-0100
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Due to the breadth of the study, it was necessary to discuss numerous topics with many individuals within NASA, the aerospace community, the advanced composites industry, and the rubber specialists industry. All of them gave freely of their time, and their advice is deeply appreciated. The writer would particularly like to note the contributions of members of NASA-Langleys Structures and Dynamics Division and its Fabrication Division. Information they provided was particularly helpful in defining planned model test facilities and state-of-the-art techniques in model construction as well as insights into developmental trends for future full scale space station hardware.
INTRODUCTION

For many decades, structural dynamicists have sought simple, expedient, and cost effective means to better understand the dynamic response of complex structures. This search has frequently led to dynamic models for reasons including the following:

1. The forces and the manner in which they interact to produce dynamic phenomena, including mechanical, friction, or fluid driven instabilities, are not adequately understood.
2. The ability to analytically formulate and solve the governing equations is limited or uncertain.
3. The gap between the analyst and physical reality is often difficult to bridge without some experience with representative hardware.

Despite a high pace of progress with computer oriented analysis, experiments will continue to be necessary in the foreseeable future to check the adequacy of theoretical derivations, interpretations, and applications. Because the space station will be designed for fractional g operations, the dynamic model provides the only realistic option for assembling and testing it as an integrated system. Such systems studies would appear propitious for the world's largest flimsy structure which must be oriented and stabilized to accuracies of the order of 0.1 degree arc.

The dynamic model also provides a convenient and effective means to evaluate the dynamic response of major subassemblies which represent the station during the various phases of on-orbit construction and is also a valuable tool for assessing the impact of changes in the basic configuration, due to growth or redirection, on systems responses.

This report covers the results of limited studies which explore various options relative to the design, fabrication, and testing of a dynamic model of the IOC space station. An attempt was made to review as many aspects of the task as feasible and to evolve practical approaches which will aid in the model design, fabrication, and testing phases, and broaden the base of organizations capable of providing an effective model to NASA.
SCOPE OF STUDY

A limited study was made to evaluate options for the design, construction, and testing of a dynamic model of the space station. Since the definition of the space station structure is still evolving, the IOC reference configuration was used as the general guideline.

The results of the studies, as given in the main report, treat: general considerations of the need for and use of a dynamic model; factors which deal with the model design and construction; and a proposed system for supporting the dynamic model in the planned Large Spacecraft Laboratory.

Consideration was given to various topics under these three general headings as follows:

1. GENERAL CONSIDERATIONS
   1.1 The role of a dynamic model in the prediction of the structural dynamics of the space station.
   1.2 Approach to model design, construction, and testing.
   1.3 Selection of model scale and scale factors.

2. DESIGN AND FABRICATION OF MODEL
   2.1 Primary Truss Structure
      2.1.1 Approximation of allowance for joint free play for pointing accuracy.
      2.1.2 Considerations for a tube connector device to vary and control joint stiffness.
      2.1.3 Determination of effective stiffness of a structure and a joint in series.
      2.1.4 Review of scaling of extensions within a joint and an approximation of relative motions.
      2.1.5 Considerations for supporting the model for testing by attachment of tangs from the truss joints.
2.1.6 Feasibility of fabrication and testing of graphite epoxy tubes.

2.1.7 Use of air or water pressure to remove thin walled composite tubes from cylindrical mandrels.

2.1.8 Technique for model tube selection / grading.

2.2 Modules and other masses.

2.3 Solar arrays and large antenna dishes.

3. DESIGN AND FABRICATION OF MODEL SUPPORT SYSTEM

3.1 Minimization of gravitation effects.

3.2 Convenience, simplicity, and minimum costs of model tests.

3.3 Safety of model structures and personnel during model assembly and testing.

3.4 Discussion of factors relating to influence of gravitational effects on model support system.

3.5 Determination of model support frequencies on cable mounting system.

3.5.1 Determination of pendular natural frequencies.

3.5.2 Determination of plunging and rotational natural frequencies.

3.6 Determination of composition of cable below support platform.

3.7 Summary of frequency separation for a 1/4 inch scale model with a suspension length of 120 feet.

3.8 Nature of cables and their properties.

3.9 Considerations on selection of rubber for use in space station model supports.

3.10 Calculation of amount of rubber cord for model support.

3.11 Approximation of weight of rubber cord for model support.

3.12 Approximation of lateral natural frequencies of model support cables.

3.13 Summary of experimental data obtained from static and dynamic tests of a rubber sample.

3.14 Considerations relative to the number of elastic cables employed for model support.
3.15 Investigation of use of coil and reversed loop springs for model support.

Appendix I - Results of Experimental Tests of Additional Rubber Samples
Appendix II - Analysis of a Beam Suspended by Cables and Undergoing Combined Bending and Pendular Motions
A. THE ROLE OF A DYNAMIC MODEL IN THE PREDICTION OF THE STRUCTURAL DYNAMICS OF A SPACE STATION

As currently conceived, the space station will consist of an assembly of special purpose structures. These include the shuttle orbiter (when attached); pressurized vessels for personnel habitat, laboratories and supplies; solar panels for energy collection and radiators for thermal control; antennae for communications; and truss structures for interconnection and support of all of these components. When these components, all designed for minimum weight, are assembled in orbit, they will cover an area approximately the size of a baseball field. Because of its size, configuration, and the need for high structural efficiency, the integrated structure will be characterized by slow body movements and low frequency structural responses.

The space station will be continually subjected to unsteady (time dependent) forces during its assembly and operational use in space. A major concern is the reaction of the space station to external forces used to reposition, reorient, or stabilize it. If these forces are coupled to the structure in such a way that they are dependent on the displacement, velocity, or acceleration of the deformations of the structure, proper phasing of the control forces with respect to the structural deformations is necessary to avoid feeding energy into the structural deformations and driving the structure to unacceptable amplitudes or failure. The analyses necessary to design the integrated structural / propulsive systems to avoid unstable coupling requires a means for expressing the spatial relationships for the motions of the structure. Any of several closed sets of functions can be used for this purpose but the most convenient set is the set of natural mode shapes for the undamped structure. This closed set of functions, the infinity of specific shapes wherein the inertial forces generated by the vibrations of the structure at the corresponding natural frequency exactly balance the elastic forces, offers the advantages that they are orthogonal and characteristic. Orthogonality reduces the
mathematical coupling by the vanishing of all integrals which involve products of deformations of more than one mode - a substantial simplification for the analyst. The characteristics property is advantageous because the natural mode shapes are readily excited and "stand out" when the structure is shaken at or near the natural frequency corresponding to the mode of interest.

What is the impact of the foregoing statements? First, prediction of the response of the space station structure to external applied forces is critically dependent on a correct definition of the structural properties of the integrated station in each and all of its operational configurations. The correctness of the structural definition is reflected in the ability of the analyst to predict the natural frequencies and mode shapes of the integrated space station structure as determined by comparison of experimental and analytical results. Second, upon achievement of agreement between the calculated and measured natural mode shapes and their corresponding natural frequencies, the motions of the structure can be represented by linear superposition of a "limited" number of these natural modes. As a guideline to determining what constitutes a limited number, a reasonable approach is to include all modes whose natural frequencies range between 0.2 and 5 times the frequency of the exciting or driving force. However, it should be noted that finite element representations of the structure which adequately predict its characteristics will also adequately predict its dynamic response since the structural characteristics are the principal unknowns in the response problem. The dynamic model provides the best and perhaps the only tool available to the designer to verify the equations, and the values of the physical parameters in them, used to analytically define the space station in its actual flight condition. It can also be used to study any subcase such as those associated with partial construction during assembly, changes in configuration such as those associated with movements of the shuttle orbiter, or changes in payloads.
B. APPROACH TO MODEL DESIGN, CONSTRUCTION AND TESTING

The actual configuration of the space station which will ultimately fly is not yet known but the general consensus seems to be that it will be quite similar to the IOC configuration, Figure 1. A desirable dynamic model would be one which provides opportunities for study of the overall dynamic characteristics of the "current" configuration at the time the model is built plus the flexibility to be easily modified to reflect changes in configuration as the program progresses. In many cases, model test results highlight the need for, and guide development of, changes in full scale structures. The modular concept proposed in the main report for the model provides such options.

Because of the large size of the model and the high flexibility of its structure, it appears impractical to obtain model support frequencies low enough to eliminate interference between the model support system and the model natural modes. Interference implies coupling in cases where motions of the model are partially restrained by the support system. In other cases, proximity of frequencies make it difficult to establish motions of the model which do not involve the superposition of elastic and rigid body modes. Two approaches to alleviation of this problem are recommended. First, minimize the interference by making the support system cables as long as possible and by attachment of model excitation equipment in such ways as to minimize the excitation of rigid body motions of the model on the support cables; and, second, include the gravitational restraint forces in the differential equations of motion used to predict the model (and full scale) characteristics and forced responses. All of the gravitationally induced terms in the equations will contain $g$, which, when it exists enables prediction of the model responses, and when it vanishes, enables prediction of the scaled full scale station responses.
The testing of the space station model will be a unique experience because of its large size, its slow response, and its fragility. The approach outlined for the design and fabrication of the model support system appears to offer the only practical means for housing, supporting and testing the model as an integrated system. It will be a difficult but feasible task, the difficulty primarily arising from the need to minimize the effects of the support system on the dynamic characteristics of the model.

C. SELECTION OF MODEL SCALE AND SCALE FACTORS

Theoretically, the limitations on the scaling of a dynamic model reduce to the fact that both the model and the full scale structure must satisfy the same dimensionless equations of motion for the phenomena under study. Stated another way, the ratios of corresponding pairs of forces (and moments) on the model must equal those for the full scale vehicle. From the mathematical viewpoint, this is a straightforward task achievable with any dynamically similar model, replica or distorted, large or small, capable of generating all significant forces and moments in the correct ratios. But the model which satisfies these necessary conditions must satisfy some tough physical conditions to provide data which will identify, or improve the understanding of the dynamic response of the full scale space station in orbit. The two more important physical considerations are brought about by the fact that the space station will fly principally under zero gravity conditions and outside the atmosphere, whereas the model tests must be conducted at 1 g and in air at atmospheric pressure.

The fact that the model must be tested at 1 g means that it must be supported in some manner which imposes restraints on its dynamic response. The effects of these restraints on the response can be measured in terms of the ratios of the model's natural frequencies (assuming g = 0) to the model's support frequencies. It is desirable to make these ratios as high as possible to minimize model restraint interference.
High ratios mean small models and long, soft support systems, i.e.:

\[
\left( \frac{w_{\text{structure}}}{w_{\text{support}}} \right) \propto \left( \frac{\sqrt{\text{support length}}}{\text{model length}} \right).
\]

The practical need to test the model in air at atmospheric pressure leads to the imposition of aerodynamic damping forces and apparent air mass forces on the model which have no counterpart for the full scale space station flying in orbit. But, for replica scaling where the natural frequency of the model is inversely proportioned to its size, the ratios of the unwanted aerodynamic forces (apparent mass and damping) to the model inertial forces associated with vibrations are independent of model size, or scale. Hence, the aerodynamic forces do not impact the selection of the model size - their minimization requires the model designer to select structures such as screens, rods, cables, etc., to properly simulate the mass and stiffness distributions of structures such as solar panels, and radiators which have high area to mass ratios.

Thus, the selection of model scale reduces to trade-offs between the ability to build the model and the ability to test it. The ability to build the model is a function only of the model; the ability to test it is also contingent upon the provision of a facility to provide an adequate test volume. Also, because of the lack of experience in dynamic analysis of large, flimsy, joint dominated structures, it is desirable to make the model as large as test capabilities will permit. The combination of these and other factors as discussed in the main report and elsewhere leads the writer to recommend a 1/4 scale model. A summary of key factors in this recommendation includes the following:

1. The model can be supported in the planned Large Spacecraft Laboratory with a minimum of interference between model characteristics and model restraints.

2. The principal model structural elements are expected to be graphite epoxy tubes. The 1/4 scale tubes will be about 1/2 inch diameter with wall thickness of about 0.010 inch. On the basis of his recent review of the technology for the manufacture of graphite epoxy tubes, the writer believes the technology exists to make suitable tubes for the 1/4 scale dynamic model.
3. The proposed joint structure for the model truss is feasible at 1/4 scale and offers the opportunity to "tailor" the model mass and stiffness, attach modular and payload masses to the truss structure, and attach the elastic cables for supporting the model.

4. The 1/4 scale space station model will be compatible with the existing 1/4 scale model of the shuttle orbiter. This could represent considerable cost savings.

5. The 1/4 scale model will span approximately 100 feet by 75 feet in planform and weigh about 10,000 lbs. under maximum loading conditions. Its lowest natural frequency will be about 0.5 Hz. The writer believes that if the model is carefully built and tested it should be possible to extrapolate the results and experience from a 1/4 scale model to the prediction and understanding of the dynamic response of the full scale space station. It is noted in passing that 1/10 scale dynamic models of numerous smaller aerospace structures ranging from helicopters to launch vehicles have been eminently successful.

The scale factors for the model are based on replica scaling, i.e., those properties of each model element which is necessarily scaled should be scaled as though the element were a replica. For example, the model elements which would represent the habitability modules for a complete replica model would be so stiff that treating them as rigid elements would have negligible impact on the overall dynamic response of the model. But their masses, mass moments of inertia, and stiffness of the attachments of the masses to the keel are significant and must be scaled as though they were replica elements. Using these design guidelines, the model scale factors are as given in Figure 2.
SCALE FACTORS FOR PROPOSED MODEL OF IOC SPACE STATION

Primary Factors - Replically Scaled Elements

Length \( (L_M/L_F) \) \( \lambda \)

Mass \( (\rho_M/\rho_F) (L_M/L_F)^3 \) \( \rho_M = \rho_F \) \( \lambda^3 \)

Time \( (T_M/T_F) \) \( \lambda \)

Derived Factors

Area \( (L_M/L_F)^2 \) \( \lambda^2 \)

Volume \( (L_M/L_F)^3 \) \( \lambda^3 \)

Area Moment of Inertia \( (L_M/L_F)^4 \) \( \lambda^4 \)

Displacement \( (L_M/L_F) \) \( \lambda \)

Velocity \( (L_M/L_F) (T_F/T_M) \) \( 1 \)

Linear Acceleration \( (L_M/L_F) (T_F/T_M)^2 \) \( \lambda^{-1} \)

Angular Acceleration \( (T_F/T_M)^2 \) \( \lambda^{-2} \)

Structural Frequency \( (T_F/T_M) \) \( \lambda^{-1} \)

Pendular Frequency \( (g_M/g_F) (L_F/L_M)^{1/2} \) \( g_M = g_F \) \( \lambda^{-1/2} \)

Force \( (M_M/M_F) (L_M/L_F) (T_F/T_M)^2 \) \( \lambda^2 \)

Torque \( (M_M/M_F) (L_M/L_F)^2 (T_F/T_M)^2 \) \( \lambda^3 \)

Stress \( (M_M/M_F) (L_M/L_F) (T_F/T_M)^2 (L_F/L_M)^2 \) \( 1 \)

Mass Movement of Inertia \( (M_M/M_F) (L_M/L_F)^2 \) \( \lambda^5 \)

Gravity Beam Column Effect \( (M_M/M_F) (g_M/g_F) (L_F/L_M)^2 \) \( \lambda \)

Figure 2. -- Scale Factors for Replically Scaled Model.
D. DESIGN AND FABRICATION OF MODEL SUPPORT SYSTEM

Current plans for the design of the large spacecraft structures laboratory permit the installation of the model in the orientation shown in Figure 3. This is the recommended orientation for several reasons including: minimization of orientational effects, convenience, simplicity, minimum costs of model tests, and safety of model structures and personnel during model assembly and testing.

As shown in the derivations given in the main report, all natural frequencies of the model support system are proportioned to $1/\sqrt{L}$. Since the frequencies of the elastic modes of the model will be higher than the support frequencies, large values of $L$ produce wider separations between natural frequencies for elastic modes and rigid body support modes. The recommended model test configuration shown in Figure 3 provides the highest values for $L$ and thus minimizes coupling.

The recommended model test configuration offers the advantage that nearly all of the model assembly is accomplished with personnel positioned on the floor and working at levels between the floor and shoulder height. In a few instances, it will be necessary to work from a low mobile platform, but no situations are envisioned where model technicians or research personnel are required to work at heights above about 20 feet. This is primarily accomplished by suspending the model from the overhead platform which can be moved as needed from floor to ceiling.

The assembly and testing of the space station model will be a unique experience because of its large size and fragility. These factors impact the safety of test personnel and the utility of an expensive piece of test hardware.

The full scale space station will be designed to function under accelerations of the order of 0.04 g, and as a consequence of the need to minimize the weight to orbit, little structural "fat" is expected. Hence the model, scaled to the same stress level as the prototype, will not be able to support itself under 1 g loads except in small sections. The proposed, essentially continuous support system, effectively eliminates that problem.
Figure 3. Outline of Recommended Model Support Arrangement in the Large Spacecraft Laboratory.
Also, because of the fragility of the joints and the tubular members of the truss structure, model test technicians must work with extreme caution to avoid application of damaging model loads. Ground based access to most parts of the model will permit the exercise of reasonable precautions while expediting execution of the model assembly and testing tasks.

E. CONCLUSIONS AND RECOMMENDATIONS

The results of the studies lead to the following conclusions and recommendations:

1. It is proposed that the model be 1/4 scale and that replica scaling be used, i.e., that the natural frequencies of the model be four times the corresponding values for the full scale vehicle.

2. It is proposed that the tubular truss elements (keel, extended keel, transverse boom, etc.) be made as nearly replica as technology and available resources will permit. An alternative to replica joints is proposed which will enable parametric investigation of joint stiffness, free-play, non-linearity, and damping as desired.

3. It is recommended that all modules and other lumped masses which have characteristic natural frequencies substantially higher than the fundamental frequencies of the integrated space station be represented on the model by rigid bodies which have appropriately scaled masses, inertias, and attachment stiffnesses.

4. Because of the high apparent mass ratio of the air surrounding model solar array and antenna components during tests, it is recommended that these components be simulated by open grid structures having appropriate mass and stiffness distributions.

5. The combination of many factors associated with supporting the model for testing suggests that the best, and only necessary, model support configuration is the one which places the plane of the keel and transverse boom near and parallel to the floor. In this orientation, the model will be supported by approximately 100 elastic cables which will maintain the rigid body model frequencies substantially below the frequencies of the lower elastic modes.

6. Apparent air mass, support system masses and gravitational force restraints will all impact the model test results to some degree. It is believed that the proposed model design and test procedures will minimize these effects to the extent that full scale hardware responses in their absence will be highly predictable from model test results.
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211 RESEARCH DRIVE

HAMPTON, VIRGINIA 23666

Dr. George Brooks
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REFERENCES

1. VARIOUS AUTHORS: SPACE STATION REFERENCE CONFIGURATION DESCRIPTION, NASA, Flight Operations Directorate, Johnson Space Center, Houston, Texas, 1984


3. SCAVILLI, JOAN L.; MURRERITTO, ROBERT; AND PAPP, RICHARDS: VIBRATION STUDIES OF A LIGHTWEIGHT THREE SIDED MEMBRANE SUITABLE FOR SPACE APPLICATION, NASA TECHNICAL PAPER 2095, 1985


5. HOOCHMAN, CHARLES: HANDBOOK FOR CHEMISTRY AND PHYSICS, THE CHEMICAL RUBBER PUBLISHING CO., CLEVELAND, OHIO, 4TH EDITION 1962

6. MAIEUV, E.V.: MACHINE DESIGN, INTERNATIONAL TEXT BOOK COMPANY, SCRANTON, PA, 1946
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A limited study was made to evaluate options for the design, construction and testing of a dynamic model of the space station. Since the definition of the space station structure is still evolving, the IOC reference configuration was used as the general guideline.

The results of the studies, as given in the report, treat general considerations of the need for and use of a dynamic model, factors which deal with the model design and construction, and a proposed system for supporting the dynamic model in the planned large spacecraft laboratory.

The results of the studies lead to the following recommendations:

1. It is proposed that the model be 1/4 scale and that replica scaling be used, i.e. that the natural frequencies of the model be 4 times the corresponding values for the full scale vehicle.

2. It is proposed that the tubular truss elements (i.e., extended keel, transverse boom, etc.) be made as nearly replica as technology and available resources will permit. An alternative to replica joints is proposed which will enable asymmetry investigation of joint stiffness, free play, non-linearity and damping as desired.

3. It is recommended that all modules and other lumped masses which have characteristic natural frequencies substantially higher than the fundamental frequencies of the integrated space station be represented on the model by rigid bodies which have appropriately scaled masses, inertias and attachments stiffnesses.
4. Because of the high apparent mass ratio of the high subsonic model, solar array and antenna components during tests, it is recommended that these components be simulated by an open grid having appropriate mass and stiffness distributions.

5. The combination of many factors associated with supporting the model for testing suggest that the best, and only necessary, model support configuration is the one which places the plane of the keel and transverse boom near and parallel to the tail. In this orientation, the model will be supported by approximately 160 elastic cables which will maintain the rigid body model frequencies substantially below the frequencies of the lower elastic modes.

6. Apparent air mass, support system masses and gravitational force restrictions will all impact the model test results to some degree. It is believed that the present model design and test procedures will minimize these effects to the extent that full scale hardware responses in their absence will be highly predictable from model test results.
INTRODUCTION

FOR MANY DECADES, STRUCTURAL DYNAMICISTS HAVE
Sought SIMPLE, EXPEDITIOUS AND COST EFFECTIVE MEANS
TO BETTER UNDERSTAND THE DYNAMIC RESPONSE OF COMPLEX
STRUCTURES. THIS SEARCH HAS FREQUENTLY LED TO DYNAMIC
MODELS FOR REASONS INCLUDING THE FOLLOWING:

1. THE FORCES AND THE MANNER IN WHICH THEY INTERACT
   TO PRODUCE DYNAMIC PHENOMENA, INCLUDING MECHANICAL,
   ELECTRICAL, OR FLUID DRIVEN INSTABILITIES, ARE NOT
   ADEQUATELY UNDERSTOOD.

2. THE ABILITY TO INHERENTLY FORMULATE AND SOLVE
   THE GOVERNING EQUATIONS IS LIMITED OR UNCERTAIN.

3. THE GAP BETWEEN THE ANALYST AND PHYSICAL REALITY
   IS OFTEN DIFFICULT TO BRIDGE WITHOUT SOME EXPERIENCE
   WITH REPRESENTATIVE HARDWARE.

DESPITE ALL OF THE AVAILABLE COMPUTERS, EXPERIMENTS WILL
CONTINUE TO BE NECESSARY IN THE FORESEEABLE FUTURE TO
CHECK THE INTEGRITY OF THEORETICAL DEVELOPMENTS, INTERPRETATIONS
AND APPLICATIONS. BECAUSE THE SPACE STATION WILL BE DESIGNED
FOR FRACTIONAL OPERATIONS, THE DYNAMIC MODEL PROVIDES
THE ONLY REALISTIC OPTION FOR ASSEMBLING AND TESTING.
IT IS AN INTEGRATED SYSTEM. SUCH SYSTEMS STUDIES WOULD
ADVERSELY INFLUENCE THE WORLD'S SMALLEST FLIGHT STRUCTURE
WHICH MUST BE ORIENTED AND STABILIZED TO ACCURACIES
OF LESS THAN 0.1 DEGREE ARC.

THE DYNAMIC MODEL ALSO PROVIDES A CONVENIENT AND
EFFECTIVE MEANS TO EVALUATE THE DYNAMIC RESPONSE OF
MAJOR SUBASSEMBLIES WHICH REPRESENT THE STATION DURING
THE VARIOUS PHASES OF ON-ORBIT CONSTRUCTION AND IS
IS ALSO A VALUABLE TOOL FOR ASSESSING THE IMPACT OF
CHANGES IN THE BASIC CONFIGURATION, DUE TO GROWTH OR
RESTRICTION, ON SYSTEMS RESPONSES.

THIS REPORT COVERS THE RESULTS OF LIMITED STUDIES
WHICH EXPLORE VARIOUS OPTIONS RELATIVE TO THE DESIGN,
FABRICATION AND TESTING OF A DYNAMIC MODEL OF THE ILC SPACE STATION. AN ATTEMPT WAS MADE TO REVIEW AS MANY ASPECTS OF THE TASK AS FEASIBLE AND TO EVOLVE PRACTICAL APPROACHES WHICH WILL AID IN THE MODEL DESIGN, FABRICATION AND TESTING PHASES, AND BROADEN THE BASE OF ORGANIZATIONS CAPABLE OF PROVIDING AN EFFECTIVE MODEL TO NASA.
1. GENERAL CONSIDERATIONS


ORIGINAL PAGE IS OF POOR QUALITY
1.1 THE ROLE OF A DYNAMIC MODEL IN THE PREDICTION OF
THE STRUCTURAL DYNAMICS OF A SPACE STATION

THE PURPOSE OF THIS NOTE IS TO DISCUSS THE ROLE OF
A DYNAMICALLY SIMILAR MODEL IN THE PREDICTION OF THE DYNAMIC
RESPONSE OF A SPACE STATION. THE DYNAMICALLY SIMILAR
MODEL MAY OR MAY NOT BE A STRUCTURAL REPLICA (WHERE
DIFFERENCES BETWEEN THE MODEL AND FULL SCALE
STRUCTURES ARE ESSENTIALLY A MATTER OF SCALE OR
SIZE) BUT IT MUST FAITHFULLY REPRESENT THE MASS
AND STIFFNESS DISTRIBUTIONS OF THE SPACE STATION
IN INSTANCES WHERE THESE DISTRIBUTIONS RESULT IN
NATURAL FREQUENCIES AND MODE SHAPES IN THE
FREQUENCY REGION WHICH ENCOMPASSES PERTINENT
FULL SCALE STRUCTURAL DEFORMATIONS. IT IS ALSO
DESIRABLE THAT THE MODEL REFLECT THE DAMPING
DISTRIBUTION OF THE FULL SCALE VEHICLE BUT THIS
IS PROBABLY NOT ACHIEVABLE AND NOT REALLY ESSENTIAL
FOR APPLICATION OF MODEL TEST RESULTS FOR PREDICTION
OF FULL SCALE RESPONSES.

AS CURRENTLY CONCEIVED, THE SPACE STATION WILL
CONSIST OF AN ASSEMBLY OF SPECIAL PURPOSE STRUCTURES.
these include the shuttle orbiter (when attached);
pressurized vessels for personal habitat, laboratories
and supplies; solar panels for energy collection and
radiators for thermal control; antennae for communications;
and truss structures for interconnection and support
of all of these components. When these components,
all designed for minimum weight, are assembled
in orbit, they will cover an area approximately
the size of a baseball field. Because of its size,
configuration, and the need for high structural efficiency,
the integrated structure will be characterized by slow
body movements and low frequency structural responses.

The space station will be continuously subjected to
UNSTEADY (TIME DEPENDENT) FORCES DURING ITS ASSEMBLY AND OPERATIONAL USE IN SPACE, THESE FORCES WILL BE OF THREE BASIC TYPES: GRAVITY, GRADIENT BODY FORCES WHICH TEND TO KEEP THE MAJOR AXES OF THE STATION ORIENTED ALONG THE EARTH'S RADIUS; FORCES DUE TO CHANGES IN THE INTERNAL MOMENTUM OF THE SYSTEM; AND EXTERNALLY APPLIED IMPULSIVE FORCES PROVIDED BY JUMPING OR PROPELLING SYSTEMS AS MAY BE NECESSARY TO REORIENT OR REPOSITION THE STATION. FROM THE STRUCTURAL DYNAMICS VIEWPOINT, THE LATER TWO ARE OF PRIMARY INTEREST.

FORCES REPRESENTING CHANGES IN THE INTERNAL MOMENTUM OF THE STATION ARE GENERALLY IMPULSIVE AND THE MAJOR CONCERN IS THAT THE DISTURBANCES THEY CAUSE BE SMALL RELATIVE TO ALLOWABLE ON-BOARD LIMITS FOR RESEARCH AND HABITABILITY, AND THAT THE DAMPING OF THE STRUCTURES CAUSE THEM TO DECREASE QUICKLY.

THE MAJOR CONCERN IS THE REACTION OF THE SPACE STATION TO EXTERNAL FORCES USED TO REPOSITION, REORIENT, OR STABILIZE IT. IF THESE FORCES ARE COUPLED TO THE STRUCTURE IN SUCH A WAY THAT THEY ARE DEPENDENT ON THE DISPLACEMENT, VELOCITY, OR ACCELERATION OF THE DEFORMATIONS OF THE STRUCTURE, PROPER PHASING OF THE CONTROL FORCES WITH RESPECT TO THE STRUCTURAL DEFORMATIONS IS NECESSARY TO AVOID FEEDING ENERGY INTO THE STRUCTURAL DEFORMATIONS AND DRIVING THE STRUCTURES TO UNACCEPTABLE AMPLITUDES OR FAILURE. THE ANALYSES NECESSARY TO DESIGN THE INTEGRATED STRUCTURAL/PROPELLING SYSTEMS TO AVOID UNSTABLE COUPLING REQUIRES A MEANS FOR EXPRESSING THE SPATIAL RELATIONSHIPS FOR THE MOTIONS OF THE STRUCTURE. AN I OF SEVERAL CLOSED SETS OF FUNCTIONS CAN BE USED FOR THIS PURPOSE BUT THE MOST CONVENIENT SET IS THE SET OF NATURAL MODE SHAPES FOR THE UNDAMPED STRUCTURE. THIS CLOSED SET OF FUNCTIONS, THE INFINITY OF SPECIFIC SHAPES WHEREIN THE INERTIAL FORCES GENERATED BY THE VIBRATIONS OF THE STRUCTURE AT THE CORRESPONDING NATURAL FREQUENCY
EXACTLY BALANCE THE ELASTIC FORCES, OFFERS THE ADVANTAGES
THAT THEY ARE ORTHOGONAL AND CHARACTERISTIC. ORTHOGONALITY
REDUCES THE MATHEMATICAL COUPLING BY THE VANISHING OF
ALL INTEGRALS WHICH INVOLVE PRODUCTS OF DEFORMATIONS OF
MORE THAN ONE MODE — A SUBSTANTIAL SIMPLIFICATION FOR THE
ANALYST. THE CHARACTERISTICS PROPERTY IS ADVANTAGEOUS
BECAUSE THE NATURAL MODE SHAPES ARE READILY EXCITED
AND "STAND OUT" WHEN THE STRUCTURE IS SHAKEN AT OR NEAR
THE NATURAL FREQUENCY CORRESPONDING TO THE MODE OF
INTEREST.

WHAT IS THE IMPACT OF THE FOREGOING STATEMENTS?
FIRST, PREDICTION OF THE RESPONSE OF THE SPACE
STATION STRUCTURE TO EXTERNAL APPLIED FORCES IS
CRITICALLY DEPENDENT ON A CORRECT DEFINITION OF THE
STRUCTURAL PROPERTIES OF THE INTEGRATED STATION IN
EACH AND ALL OF ITS OPERATIONAL CONFIGURATIONS. THE
CORRECTNESS OF THE STRUCTURAL DEFINITION IS REFLECTED
IN THE ABILITY OF THE ANALYST TO PREDICT THE NATURAL
FREQUENCIES AND MODE SHAPES OF THE INTEGRATED SPACE
STATION STRUCTURE AS DETERMINED BY
COMPARISON OF EXPERIMENTAL AND ANALYTICAL RESULTS.
SECOND, UDTHE ACHIEVEMENT OF AGREEMENT BETWEEN THE
CALCULATED AND MEASURED NATURAL MODE SHAPES AND
THEIR CORRESPONDING NATURAL FREQUENCIES, THE MOTIONS
OF THE STRUCTURE CAN BE REPRESENTED BY LINEAR
SUPERPOSITION OF A "LIMITED" NUMBER OF THESE NATURAL
MODES. AS A GUIDELINE TO DETERMINING WHAT CONSTITUTES
A LIMITED NUMBER, A REASONABLE APPROACH IS TO INCLUDE
ALL MODES WHOSE NATURAL FREQUENCIES RANGE BETWEEN
0.2 AND 5 TIMES THE FREQUENCY OF THE EXCITING OR
DRIVING FORCE. HOWEVER, IT SHOULD BE NOTED THAT FINITE
ELEMENT REPRESENTATIONS OF THE STRUCTURE WHICH ADEQUATELY
PREDICT ITS CHARACTERISTICS WILL ALSO ADEQUATELY PREDICT
ITS DYNAMIC RESPONSE SINCE THE STRUCTURAL CHARACTERISTICS
ARE THE PRINCIPAL UNKNOWNS IN THE RESPONSE PROBLEM.
Thus the dynamic model provides the best and perhaps the only tool available to the designer to verify the equations and the values of the physical parameters in them, used to non-linearly define the space shuttle in its actual flight condition. It can also be used to study any subcase such as those associated with partial construction during assembly, changes in configuration such as those associated with movements of the shuttle orbiter, or changes in payloads.
1.2 APPROACH TO MODEL DESIGN, CONSTRUCTION AND TESTING

The initial configuration of the Space Station which will ultimately fly is not yet known but the general consensus seems to be that it will be quite similar to the IOC configuration outlined in Figures 1 to 3, extracted from Reference 1. The spacecraft consists of a backbone truss system (keel, keel extensions and booms) for interconnection of major and minor substructures including habitability, laboratory, and logistics modules, solar arrays and antennas, and radiator systems. Since the physical properties of the station are not yet defined, and when defined are subject to change by growth and redirection, a desirable dynamic model would be one which provides opportunities for study of the overall dynamic characteristics of the "current" configuration at the time this model is built plus the flexibility to be easily modified to reflect changes in configuration as the program progresses. In many cases, model test results highlight the need for and guide developmental changes in full scale structures. The modular concept proposed for the model, as discussed in Part B—Design and Fabrication of the Model, provides such options.

Owing to the large size of the model and the high flexibility of its structure, it appears impractical to obtain model support frequencies low enough to eliminate interference between the model support system and the model natural modes. Interference implies coupling in cases where motions of the model are partially restrained by the support system. In other cases, proximity of frequencies make it
Figure 3.- 10C Reference Space Station - Component Definition

*NOTE: SOLAR DYNAMICS SYSTEM A PRIME OPTION (NOT SHOWN)*

1.3 SELECTION OF MODEL SCALE AND SCALE FACTORS

Theoretically, the limitations on the scaling of a dynamic model reduce to the fact that both the model and the full scale structure must satisfy the same dimensionless equations of motions for the phenomenon under study. Stated another way, the ratios of corresponding pairs of forces (and moments) on the model must equal those for the full scale vehicle. From the mathematical viewpoint, this is a straightforward task achievable with any dynamically similar model, replica, or distrippes, large or small, capable of generating all significant forces and moments in the correct ratios. But the model which satisfies these necessary conditions must satisfy some tough physical conditions to provide data which will identify, or improve the understanding of, the dynamic response of the full scale space station in orbit. The two more important physical considerations are brought about by the fact that the same station will fly principally under zero gravity conditions and outside the atmosphere whereas the model tests must be conducted at 1 g and in air at atmospheric pressure.

The fact that the model must be tested at 1 g means that it must be supported in some manner which imposes restraints on its dynamic response. The effects of these responses can be measured in terms of the ratios of the model's natural frequencies (assuming $g = 0$) to the models support frequencies. It is desirable to make these ratios as high as possible to minimize model restraint interference. High ratios mean similar models and long, soft support systems.
THE PRACTICAL NEED TO TEST THE MODEL IN AIR AT
ATMOSPHERIC PRESSURE LEADS TO THE IMPOSITION OF
AERODYNAMIC DAMPING FORCES AND APPARENT AIR MASS
FORCES ON THE MODEL WHICH HAVE NO COUNTERPART FOR
THE FULL SCALE SPACE STATION FLYING IN ORBIT. BUT,
FOR REDUCED SCALES (W = 1), THE RATIO OF THE
UNWANTED AERODYNAMIC FORCES (APPARENT MASS AND
DAMPING) TO THE MODEL INERTIA FORCES ASSOCIATED
WITH VIBRATIONS IS INDEPENDENT OF MODEL SIZE OR
SCALE. HENCE, THE AERODYNAMIC FORCES DO NOT IMPACT THE
SELECTION OF THE MODEL SIZE — THEIR MINIMIZATION FORCES
THE MODEL DESIGNER TO SELECT STRUCTURES SUCH AS SCREENS,
RODS, CABLES, ETC., TO PROPERLY SIMULATE THE MASS AND
STIFFNESS DISTRIBUTIONS OF STRUCTURES SUCH AS SOLAR PANELS,
AND RODS WHICH HAVE HIGH AREA TO MASSES RATIO.

THUS, THE SELECTION OF MODEL SCALE REDUCES TO
TRADEOFFS BETWEEN THE ABILITY TO BUILD THE MODEL AND
THE ABILITY TO TEST IT. THE ABILITY TO BUILD THE MODEL IS
A FUNCTION ONLY OF THE MODEL; THE ABILITY TO TEST IT IS
ALSO DEPENDENT ON THE PRESENCE OF A FACILITY TO PROVIDE
AN ACCURATE TEST VOLUME. ALSO, BECAUSE OF THE LACK OF
EXPERIENCE IN DYNAMIC ANALYSIS OF LARGE, FLUSH, JOINT
UNIMATED STRUCTURES, IT IS DESIRABLE TO MAKE THE
MODEL AS LARGE AS TEST CAPABILITIES WILL PERMIT. THE
COMBINATION OF THESE TWO FACTORS AS DISCUSSED IN
THIS REPORT AND ELSEWHERE LEADS THE WRITER TO RECOMMEND
A 1/4 SCALE MODEL. A SUMMARY OF KEY FACTORS IN THIS
RECOMMENDATION INCLUDES THE FOLLOWING:

1. THE MODEL CAN BE SUPPORTED IN THE PLANNED LARGE
SPACECRAFT LABORATORY WITH A MINIMUM OF INTERFERENCE
BETWEEN MODEL CHARACTERISTICS AND MODEL RESTRAINTS.
2. THE PRINCIPAL MODEL STRUCTURAL ELEMENTS ARE
EXPECTED TO BE SIBONITE EPOXY TUBES. THE 1/4
SCALE TUBES WILL BE ABOUT 1/2 INCH DIAMETER WITH
WALL THICKNESS OF ABOUT 0.010 INCH. ON THE BASIS
OF HIS RECENT REVIEW OF THE TECHNOLOGY FOR THE MANUFACTURE OF GANITIC EPTI TUBES, THE WRITER BELIEVES THE TECHNOLOGY EXISTS TO MAKE SUITABLE TUBES FOR THE 1/4 SCALE DYNAMIC MODEL.

3. THE PROPOSED JOINT STRUCTURE FOR THE MODEL TRUSS IS FEASIBLE AT 1/4 SCALE AND OFFERS THE OPPORTUNITY TO "TAILORED" THE MODEL MASS AND STIFFNESS, ATTACH MODULAR AND PRECISE MASSES TO THE TRUSS STRUCTURE, AND ATTACH THE ELASTIC CABLES FOR SUPPORTING THE MODEL.

4. THE 1/4 SCALE SPACESTATION MODEL WILL BE COMPATIBLE WITH THE EXISTING 1/4 SCALE MODEL OF THE SHUTTLE ORBITER. THIS COULD REPRESENT CONSIDERABLE COST SAVINGS.

5. THE 1/4 SCALE MODEL WILL SPAN APPROXIMATELY 100 FT. BY 75 FT. IN PLANFORM AND WEIGH ABOUT 10,000 LBS. UNDER MAXIMUM LOADING CONDITIONS. ITS LOWEST NATURAL FREQUENCY WILL BE ABOUT 0.5 HZ. THE WRITER BELIEVES THAT IF THE MODEL IS CAREFULLY BUILT AND TESTED IT SHOULD BE POSSIBLE TO EXTRAPOLATE THE RESULTS AND EXPERIENCE FROM A 1/4 SCALE MODEL TO THE PRECISE ONES UNDERSTANDING OF THE DYNAMIC RESPONSE OF THE FULL SCALE SPACE STATION. IT IS NOTED IN PASSING THAT 1/10 SCALE MODELS OF NUMEROUS SMALLER AEROSPACE STRUCTURES RANGING FROM HELICOPTERS TO LAUNCH VEHICLES HAVE BEEN EMINENTLY SUCCESSFUL.

THE SCALE FACTORS FOR THE MODEL ARE BASED ON REPLICA SIMULATING - I.E., THOSE PROPERTIES OF EACH MODEL ELEMENT WHICH IS NECESSARILY SCALLED SHOULD BE SCALLED AS THOUGH THE ELEMENT WERE REPLICA. FOR EXAMPLE, THE MODEL ELEMENTS WHICH WOULD REPRESENT THE HABITABILITY MODULES FOR A COMPLETE REPLICA MODEL WOULD BE SO STIFF THAT TREATING THEM AS RIGID ELEMENTS WOULD

THE FOLLOWING TABLE C

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SCALE FACTORS FOR PROPOSED MODEL OF IGS SPACE STATION

PRIMARY FACTORS - SIMILARLY SCALED ELEMENTS

LENGTH \( (\frac{L_1}{L_F}) \)

\( \lambda \)

MASS \( (\frac{M_1}{M_F})(\frac{L_1}{L_F})^3 \)

\( \beta_1 = \beta_F \)

TIME \( (\frac{T_1}{T_F}) \)

\( \lambda \)

DERIVED FACTORS

AREA \( (\frac{A_1}{A_F})^2 \)

\( \lambda^2 \)

VOLUME \( (\frac{V_1}{V_F})^3 \)

\( \lambda^3 \)

AREA MOMENT OF INERTIA \( (\frac{I_1}{I_F})^4 \)

\( \lambda^4 \)

DISPLACEMENT \( (\frac{D_1}{D_F}) \)

\( \lambda \)

VELOCITY \( (\frac{V_1}{V_F})(\frac{T_F}{T_1}) \)

\( \lambda^{-1} \)

LINEAR ACCELERATION \( (\frac{a_1}{a_F})(\frac{T_F}{T_1})^2 \)

\( \lambda^{-2} \)

ANGULAR ACCELERATION \( (\frac{\omega_1}{\omega_F})(\frac{T_F}{T_1})^2 \)

\( \lambda^{-2} \)

STRUCTURAL FREQUENCY \( (\frac{T_F}{T_1}) \)

\( \lambda^{-\frac{1}{2}} \)

PRECURSOR FREQUENCY \( [2\pi\sqrt{\frac{g_1}{g_F}}(\frac{L_1}{L_F})]^{\frac{1}{2}} \)

\( g_1 = g_F \)

FORCE \( (\frac{F_1}{F_F})(\frac{M_1}{M_F})(\frac{T_F}{T_1})^2 \)

\( \lambda^2 \)

TORQUE \( (\frac{M_1}{M_F})(\frac{L_1}{L_F})^2(\frac{T_F}{T_1})^2 \)

\( \lambda^3 \)

STRESS \( (\frac{S_1}{S_F})(\frac{M_1}{M_F})(\frac{T_F}{T_1})^2(\frac{L_1}{L_F})^2 \)

\( \lambda^5 \)

MASS MOMENT OF INERTIA \( (\frac{M_1}{M_F})(\frac{I_1}{I_F})^2 \)

\( \lambda \)

GRAVITY DEAM COLUMN EFFECT \( (\frac{M_1}{M_F})(\frac{g_1}{g_F})(\frac{L_1}{L_F})^2 \)

\( \lambda \)

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FIGURE 4: SCALE FACTORS FOR PROPOSED SIMILARLY SCALED MODEL
2. DESIGN AND FABRICATION OF THE MODEL

As noted in Section 1.2 and Figures 1 to 3, the IOL Reference Space Station consists of a primary truss structure to which special purpose components or elements will be attached. These include the labs, the Shuttle Orbiter, the Solar Power Systems, Radiators, Antennas, etc. It seems highly probable that the actual IOL Space Station will be constructed in similar fashion because this general configuration offers the potential of high strength-to-weight ratio structures, ease of access for on-orbit assembly by deployment or erection, and a versatile base for payload attachment and servicing. Thus, it is assumed that the dynamic model of the Space Station will simulate, and replicate where appropriate, structures very similar to the IOL Reference Configuration.
2.1 Primary Truss Structure

The primary truss structure of the space station will probably consist of tubular elements made of graphite-epoxy composite materials joined in a tetrahedral arrangement similar to that shown in Figure 4. The structure is a repetitive arrangement of 8 cells where each cell may be considered as consisting of 4 longerons, 4 battens and 4 diagonals. Whether the structure is deployed or erected in orbit will only impact the model to the extent that the method of assembly may influence the mass and stiffness of some flight structures, hence, model structures.

On the basis of the previous assumptions, the model designer is principally interested in the following questions relative to the truss:

1. How do you build the model truss structure so it dynamically simulates the properties of the full scale truss structure?
2. How do you provide for the imposition of the loads imposed by the various masses which constitute the remainder of the station?
3. How do you provide for attachment of the model support system required for model tests?

Implicit in the phrase "dynamically simulates" is the decision for appropriate distributions of mass, stiffness, and damping, with emphasis on the possibility of free play and flexibility in the joints if assembled in a manner similar to that shown in Figure 4.

As noted in section 2.1.1, the space station pointing requirements dictate that the free play in the joints of the truss must be extremely small, i.e., averaging less than 0.001 inches per joint. In all probability, the joints will be designed with a free play
LOCKOUT SYSTEM. THE PROBLEM OF FREE PLAY FOR THE MODEL IS EVEN MORE DIFFICULT FOR TWO REASONS—ONE, THE TOLERANCES MUST BE REDUCED BY THE MODEL SCALE FACTOR, AND TWO, IT WOULD BE MUCH EASIER TO HOLD CLOSE TOLERANCES ON JOINT ELEMENTS THE SIZE OF THE FULL SCALE STRUCTURE THAN IT WOULD BE ON ELEMENTS THE SIZE OF THE MODEL.
2.1.1 APPROXIMATION OF ALLOWABLE FOR JOINT FREE PLAY FOR POINTING ACCURACY

As noted on page 1-26 of Ref. 2, the Rockwell International report which covers studies made for NASA MSEC for development of deployable structures for large space platforms, a pointing accuracy of 0.05 to 0.10 deg is representative. Assuming the keel has 44 sections (43 joints) as shown in the 10C reference configuration description (Fig. 2), that the pointing accuracy is uniformly relative to the case (laboratory), and that the jib effects produced in each joint are additive as expected for low frequency motion, the allowable angle per joint for the average pointing accuracy is:

\[ \alpha_j = \frac{0.075}{43} \times 57.3 = 3.04 \times 10^{-5} \text{ rad} \]

Assume that half of this joint rotation is a result of free play and half is a result of elastic deformation. Then for a nine-foot deep truss, the free-play must be limited to:

\[ \Delta_j = \frac{1}{2} \times \text{free play} \times \alpha_j \]

\[ = \frac{1}{2} \left( \frac{9 \times 12}{2} \right) \times 3.04 \times 10^{-5} \text{ in} \]

\[ = 0.00082 \text{ in} \]

With the model makes allowance

\[ \left( \frac{\Delta_j}{\alpha_j} \right)_M = \lambda = \frac{1}{4} \]

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Hence, \[ \left( \frac{\Delta_j}{\alpha_j} \right)_F \approx 0.000825 \] - probably impossible to achieve in a reasonable model construction environment.
3.1.2 CONSIDERATIONS FOR A TUBE CONNECTOR
DEVICE TO VARY AND CONTROL JOINT STIFFNESS.

SINCE FULL SCALE TRUSS AND JOINT DETAILS ARE
NOT YET KNOWN, AND SINCE IT IS HIGHLY DESIRABLE
TO BUILD RESEARCH VERSATILITY INTO THE MODEL,
IT IS RECOMMENDED THAT A TECHNIQUE FOR MODEL
CONSTRUCTION BE EMPLOYED WHICH ENABLES CHANGES
IN THE DYNAMIC CHARACTERISTICS OF THE MODEL
STRUCTURE. THIS TECHNIQUE IS ILLUSTRATED IN
THIS SECTION. THE IDEA IS TO BUILD THE STRUCTURAL
ELEMENTS AS LIGHT, STRONG AND STIFF AS POSSIBLE
AND INCORPORATE FACTORS SUCH AS JOINT FREE-
PLAY, RESILIENCE AND DAMPING IN A CONTROL ELEMENT
AS APPROPRIATE. THIS THE BASIC ELEMENTS OF THE
TRUSS STRUCTURE WOULD BE THE TUBES, THE JOINTS
AND THE CONNECTOR. THE TUBES WOULD BE AS CLOSE
TO REPLICA CONSTRUCTION AS TECHNICALLY FEASIBLE;
THE JOINTS WOULD BE INVESTMENT CASTINGS MADE
WITH MULTIPLE PRONGS OR CONNECTORS TO INTERCONNECT
THE VARIOUS TRUSS MEMBERS, TO ATTACH OTHER SPACE
STATION COMPONENTS, AND TO ATTACH THE SUPPORT CABLES;
AND THE CONNECTORS WOULD PROVIDE FOR TUBE
INSTALLATION AND REPLACEMENT AS WELL AS FOR
"TUNING" THE DYNAMIC CHARACTERISTICS OF THE TRUSS
ELEMENTS.

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THE FOLLOWING PICTURES PROVIDE FURTHER DETAILS ON THE PROPOSED JOINT SYSTEM.

ASSUME THAT EACH JOINT IS A ONE PIECE CAST OR MOLDED SYSTEM WHICH HAS NO ARTICULATION BUT HAS TUBE CONNECTIONS COMMENSURATE WITH CONFIGURATION OF FULL SCALE JOINTS.

ONE PIECE JOINT, CAST OR MOLDED CONSTRUCTION, AND REPEATABLE IN THE NECESSARY NUMBER OF DIFFERENT CONFIGURATIONS THROUGHOUT THE STRUCTURE.

ASSUME THAT EACH END OF EACH TUBE IS FITTED WITH A TUBE CONNECTOR WHICH, WHEN COMBINED WITH THE JOINT AND TUBE PROVIDES THE DESIRED TUBE STIFFNESS, NON-LINEARITY, DAMPING, ETC. EACH CONNECTOR MUST HAVE THE FOLLOWING PROPERTIES:

1. ALLOW ANY TUBE TO BE REMOVED AND REPLACED OR ADJUSTED IN LENGTH WITH EASE.
2. INERTIA'S, HIGHLY REPRODUCIBLE

3. STIFFNESS CHARACTERISTICS UNIFORM & PREDICTABLE

4. LIGHTWEIGHT

5. LOW INHERENT DAMPING

6. OPTIONS FOR BUILDING IN JOINT NON-LINEARITY IF DESIRED

IT IS BELIEVED THAT THESE PROPERTIES ARE PROVIDED TO A HIGH DEGREE BY THE CONNECTOR SYSTEM SKETCHED BELOW

THREADED NUTS (ONE BONDED IN EACH PLUG)

THREAD LoS STUD (RIGHT HAND THREAD)

THREAD LoS STUD (LEFT HAND THREAD)

SPRING (TWO STEPS OR TWO DISCS WELDED TOGETHER AT EDGES)

WELD OR BRAKE

-PRESSED FIT

SPRING DETAIL

STUD
THE CONNECTORS SHOWN PROVIDE SEVERAL OPTIONS FOR VARIATIONS IN JOINT PROPERTIES INCLUDING THE FOLLOWING:

1. STIFFNESS

VARIATIONS IN SPRING DIAMETER, MATERIAL, THICKNESS, ETC.

2. MASS

MAKE SYSTEM AS LIGHT AS POSSIBLE. ADD TAPE TO TUBES TO INCREASE MASS AS DESIRED

3. DAMPING

INHERENTLY LOW. FILL AND COAT SPRINGS WITH VISCOS ELASTOPLASTS TO INCREASE DAMPING

4. FREE PLAY

CONTROL BY THREAD CLEARANCE BETWEEN STUDS AND NUTS. PLACE STUDS AS NECESSARY. USE STUDS AND NUTS FROM SAME LOT FOR UNIFORMITY

5. NON-LINEARITY

PUT LATE VARIATIONS ON OUTSIDE OF SPRINGS
FIGURE 5 shows a sketch of a proposed model joint which was sized on the basis of the following calculations.

The assumption is made that a force acting at the center of mass of the space station will accelerate the station at a rate of 0.04 g. Then, for an all-up station weight of 600,000 lb and considering that the forces are transmitted through 4 longerons, the area required to carry the compression and tension forces in each longeron is

\[ A = \frac{F}{q_a} = \frac{600,000 \times 0.04}{g_a} \times \frac{1}{4} \cdot \frac{1}{\sigma} \]

Assuming that the material is aluminum and that \( q_a = 12,500 \text{ psi} \),

\[ A = \frac{600,000 \times 0.04}{2 \times 12,500 \times 4} = 0.24 \text{ in}^2 / \text{strut} \]

\[ = \frac{\pi d_f^2}{4} \]

Therefore, \( d_f = \sqrt{\frac{24 \times 4}{\pi}} = 0.55 \text{ in} \)

For a 1/4 scale replica model,

\[ \frac{d_f}{4} = \frac{0.55}{4} = 0.1375 \text{ - say } \frac{1}{16} \text{ in} \]

Original page is of poor quality.
SKETCH OF SPACE STATION MODEL JOINT

MATERIAL - CASTING ALUMINUM

CROSS SECTION OF TUBE

CROSS SECTION OF STEM

ORIGINAL PAGE IS OF POOR QUALITY

OTHER PRONGS WILL ALSO EXIST PERPENDICULAR TO THE PLANE OF THE PAPER BUT MAY BE ADDED BY WELDING THE WAX MOLDS DURING THE CASTING PROCESS

\[ \pi \cdot d^2 = \frac{\pi}{4} \cdot d^2 \quad \text{or} \quad t = \frac{d^2}{4D} = 0.0078" \]

FIGURE 5: SKETCH OF SPACE STATION MODEL JOINT
FIGURE 6 SHOWS AN INVESTMENT CASTING OF CASTING ALUMINUM WHICH WAS MADE BY THE NASA-LANGLEY FABRICATION DIVISION TO ASSESS THE CAPABILITY OF BUILDING JOINTS SIMILAR TO THAT SKETCHED IN FIGURE 5. THE MODELMAKERS FOUND THAT THE WALL THICKNESS OF 0.008" AS CHILLED FOR IN FIGURE 5 WAS NOT ACHIEVABLE BY THE LAST-WAY PROCESS AND HAD TO INCREASE THE THICKNESS TO ABOUT 0.013" TO ACHIEVE A GOOD REPEATABLE PRODUCT. EVEN SO, THE WEIGHT OF THE JOINT, WHICH ACCOMODATES ALL THE TUBES EXPECTED FOR A TYPICAL JOINT, IS APPROXIMATELY EQUIVALENT TO THAT OF AN ALUMINUM ROD 5/16" D AND 7.5" L, OR ABOUT 1/4 OZ.

JOINTS OF THE TYPE SHOWN IN FIGURE 6 SATISFY SEVERAL OTHER REQUIREMENTS. FIRST, BY WELDING ANY DESIRABLE ELEMENTS TO THE JOINT DURING THE WAX STAGE, HIGH STRENGTH COMPONENTS CAN BE PROVIDED FOR ATTACHING THE OTHER MOSESSES (PRYLODOS, ANTENNAS, ETC) TO THE KEEL STRUCTURE AND FOR ATTACHING THE MODEL SUPPORT CABLES.

A SECOND FACTOR TO BE DEALT WITH IS MODEL COST. THE QUESTION WHICH SHOULD RIGHTEOUSLY BE ASKED IS: WHY CAN'T YOU BUILD REPLICA JOINTS OF THE FULL SCALE HARDWARE? THE ANSWER IS, YOU CAN. FINE SWISS WATCHES ARE MORE DIFFICULT TO BUILD AND THEY ARE BUILT EVERY DAY. BUT THE HIGH COSTS OF TOOLING IS SPREAD OVER TENS OF THOUSANDS OF WATCHES AND ONLY ONE SPACE STATION MODEL IS ANTICIPATED. THE COST OF REPLICA JOINTS WOULD EXCEED THE COSTS OF FULL SCALE JOINTS. FURTHERMORE, COMPROMISES MUST BE MADE. THROUGH MODEL TESTS IN THE EARTH'S ATMOSPHERE AND GRAVITY FIELD WHICH IMPOSE HIGHER DYNAMIC CONSTRAINTS, MODEL JOINTS SUCH AS THESE APPEAR TECNICALLY RESPONSIVE AND COST EFFECTIVE.
FIGURE 6. - INVESTMENT CASTING OF ALUMINUM JOINT OF TYPE PROPOSED FOR SPACE STATION MODEL.
AN APPROXIMATION OF VARIABLE SPRING CHARACTERISTICS
REQUIRED FOR THE MODEL TUBE CONNECTOR TO SIMULATE THE
FULL SCALE JOINT PROPERTIES IS GIVEN AS FOLLOWS.

\[ \delta = \delta_1 + \delta_2 \]

\[ \delta_1 = 4 \delta_2 \]

ASSUME THAT 80% OF THE STRETCH TAKES PLACE IN THE
JOINT, I.E., \( \delta_1 = 4 \delta_2 \)

ASSUME THAT \( \delta_2 \) IS THE DEFLECTION FOR A 1/2" 0
GRAPHITE TUBE WITH WALL THICKNESS OF 0.015" AND A LENGTH
OF 13.5" (A TUBE CONNECTOR ASSUMED AT EACH END OF THE TUBE)

THE SPRING \( \delta_1 \) IS AS SHOWN. IT CAN BE REPRESENTED
BY TWO FIXED-FIXED BEAMS IN SERIES.

\[ \delta_1 = 2 \frac{PL^3}{9EI_s} \]

THE DEFLECTION OF THE TUBE IS DERIVED BY

\[ \delta_2 = \frac{P}{A} \frac{L}{4E_t} \]

\[ \delta_2 = \frac{L}{2} \frac{P}{AE_t} \]

NOW BY EQUATING \( \delta_1 \) TO \( 4 \delta_2 \)
\[ \frac{2PL^3}{192EI_s} = A \frac{L}{3AE} \]

\[ I_s = \frac{1}{192} \left( \frac{E_s}{E} \right) \left( \frac{b}{2} \right)^3 \]
\[ A = \pi \left( \frac{1}{2} \right) t_s \]
\[ = \frac{1}{192} \left( \frac{21 \times 10^6 (0.5)^3}{30 \times 10^6} \right) \frac{21}{3} (0.15) \]
\[ = 3.977 \times 10^{-7} = \frac{1}{12} \Delta t_s^3 \]
\[ \Delta t_s = t_s \]
\[ \frac{1}{4} \quad 0.0267 \]
\[ \frac{3}{4} \quad 0.0233 \]
\[ \frac{3}{8} \quad 0.0212 \]
\[ \frac{1}{2} \quad 0.0267 \]
\[ \frac{1}{2} \quad 0.0267 \]
\[ W_s = \frac{L_s^3}{3} = \frac{21.774 \times 10^{-6}}{3} \]

The conclusion from these calculations is that the springs could be made from steel spring stock of apprx. 0.025 thickness.

Other factors reviewed relative to joint details include the effective stiffness of a structure and a joint in series, the relative motions due to joint and tube deformations, and the behaviors of neighboring within a hinge joint. Details of these analyses are given in Sections 4.1.3 - 4.1.5 which follow.
2.1.3 Determination of Effective Stiffness of a Structure and a Joint in Series

The effective stiffness is determined as follows:

\[ S_{\text{eff}} = S_{c} + S_{j} = \frac{S_{c} E_{c}}{E_{c}} + \frac{S_{j} E_{j}}{E_{j}} = \frac{S_{c} E_{c}}{E_{c}} + \frac{S_{j} E_{j}}{E_{j}} \]

\[ = \frac{P}{(EA)_{c}} \left( \frac{L}{(EA)_{c}} + \frac{L}{(EA)_{j}} \right) = \frac{PL}{(EA)_{\text{eff}}} \]

\[ \frac{(EA)_{\text{eff}}}{(EA)_{c}} = \frac{1}{1 + \frac{L}{4} \left( \frac{(EA)_{c}}{(EA)_{j}} - 1 \right)} \]

What are reasonable values?

Let \( L = 1'' \), \( L = 9'' = 108'' \), \( \frac{L}{h} = 0.0093 \)

\( (EA)_{c} = 10 (EA)_{j} \)

\[ \frac{(EA)_{\text{eff}}}{(EA)_{c}} = \frac{1}{1 + 0.0093 (10-1)} = \frac{1}{1 + 0.084} = 0.92 \]

A NASA derived chart plot is shown in Figure 7.
EFFECTIVE STRUT STIFFNESS CONSIDERING JOINT EFFECTS

\[
\frac{(AE)_{\text{J}}}{(AE)_{\text{S}}} = 1.0
\]

REGION OF PROBABLE INTEREST

\[
\frac{1}{L} = 0.1
\]

\[
\frac{\overline{AE}_{\text{EFF}}}{AE_{\text{STRUT}}} = \frac{1}{1 + \frac{\phi}{L}(\frac{AE_{\text{S}}}{AE_{\text{J}}} - 1)}
\]

FIGURE 7.- EFFECT OF JOINT STIFFNESS ON THE EFFECTIVE STIFFNESS OF A STRUT.
2.1.4 REVIEW OF SCALING OF EXTENSIONS WITHIN A JOINT (HINGE) AND AN APPROXIMATION OF RELATIVE MOTIONS

\[ \delta = \sum_{i=1}^{n} \delta_i \]

AND

\[ \delta_1 = \text{extension of tube } \alpha \frac{k_1 P L_1}{d_2 E_2} \]

\[ \delta_2 = \text{extension due to pin bending} \]

\[ \alpha \frac{L_2 P L_2^3}{E_2 d_2^4} \]

\[ \delta_3 = \text{extension due to pin shear} \]

\[ \alpha \frac{b_2 P}{d_2 G_2} \]

\[ \delta_4 = \text{extension due to clevis stretching at pin} \]

\[ \alpha \frac{F_4 d_2^2 P}{(\omega - \phi) t E_2} \]
\( \delta_5 = \text{EXTENSION DUE TO PIN SHEARING} \)
\[
\alpha \frac{E_s d_2 P}{E_2 d_2 t}
\]

\( \delta_6 = \text{EXTENSION DUE TO CLEWS BEARING} \)
\[
\alpha \frac{E_6 d_2 P}{E_3 d_2 t}
\]

THE FORCE \( P \propto M_0 \cdot \alpha M_0^{3/2} \) FOR REPLICA SCALING

\[
\frac{H_M}{M_F} = \lambda^3 = \left( \frac{H_M}{H_F} \right)^3 \frac{\omega_M^2}{\omega_F^2} = \frac{1}{\lambda^2}
\]

\[
\frac{P_M}{P_F} = \left( \frac{M_M}{M_F} \right) \left( \frac{\omega_M}{\omega_F} \right)^2 \left( \frac{H_M}{H_F} \right) = \left( \lambda^3 \frac{1}{\lambda^2} \lambda \right) = \lambda^2 \text{ AND}
\]

IT FOLLOWS THAT

\[
\frac{\delta_{5,F}}{\delta_{5,M}} = \left( \frac{P_M}{P_F} \right) = 1 \text{ WHEN } E_M = E_F \neq G_M = G_F
\]

AND \( \lambda \) IS ONE OF THE CHARACTERISTIC LENGTHS, THUS ALL EXTENSIONS OF THE JOINT OF A REPLICA MODEL SCALE DIRECTLY AS THE SIZE OF THE JOINT.
APPROXIMATION OF THE MOTIONS DUE TO JOINT AND TUBE DEFLECTIONS MAY BE ACHIEVED AS FOLLOWS.

CONSIDERING THE SIMPLE HINGED JOINT SHOWN IN THE SKETCH ON PAGE 37, EXAMINE THE RELATIVE DEFLECTIONS OF A 1/4 INCH STEEL PIN IN BENDING WITH THE DEFLECTIONS OF 1/2 OF A REPRESENTATIVE TUBE.

(a) DEFLECTION DUE TO PIN BENDING

\[ \delta = \frac{P/2}{k} \]

Center of Claws

\[ \frac{1}{3} \]

Center of Hinge

\[ \delta = \frac{P L^3}{3 E I} = \left( \frac{P}{2} \right) \left( \frac{1}{3} \right) \left( \frac{3}{30 \times 10^6} \right) \left( \frac{1}{\frac{4}{8}} \right) \]

\[ \delta / P_2 = (0.422) \left( \frac{1}{3} \right) \left( \frac{1}{30} \right) \times 10^{-6} \left( \frac{1}{0.001917} \right) \]

\[ = 0.46 \times 10^{-6} \text{ in/16} \]

(b) DEFLECTION DUE TO TUBE STRETCHING

\[ \delta = \frac{E L}{2} = \frac{4}{E} \frac{L}{2} = \left( \frac{E}{4E} \right) \left( \frac{L}{2} \right) \]

\[ \frac{\delta}{F} = \left( \frac{1}{A} \right) \left( \frac{1}{E} \right) \left( \frac{L}{2} \right) = \left( \frac{1}{17 	imes 2 \times 0.06} \right) \left( \frac{1}{2 \times 10^6} \right) \left( 54 \right) = 6.82 \times 10^{-6} \]

CONCLUSION: DEFLECTION DUE TO BENDING OF THE 1/4" PIN IS ABOUT 4 TIMES AS MUCH AS DUE TO TUBE STRETCHING.
2.1.5 Considerations for supporting the model by attachments to things from the truss joints.

The all-up weight of the model will be supported from a series of SCFT cables. For a 1/4 scale model, which will be about 100 ft. long, the maximum weight will be about 10,000 lb. The forces representing this weight must be carried through the joints of the truss structure.

The IOC configuration of prime interest contains about 416 keel sections. Thus the keel has about 68 points for attachment of cables to carry loads in a given direction. The keel extensions, the transverse boom and the upper boom provide another 140 points. If we assume that the weight is carried by one fourth of these 228 possibilities, the force per joint would be 10,000/57 or 175/165 lb.

If a long attachment tang in the form of a projection from each joint is provided to carry the loads, the necessary diameter will be

\[ d = \left( \frac{F}{E \cdot F_{tu}} \right)^{1/2} = \left( \frac{H}{F_{tu}} \frac{175}{32 \times 10^3} \right)^{1/2} = 0.106 \text{ in} \]

Assuming the use of an aluminum casting alloy such as 355 heat treated to T-6. If we assume a nominal size of \( \frac{1}{8} \) in diameter and \( \frac{1}{2} \) in length, and assume that each joint is equipped with 2 tangs (for support in either of 2 directions), the total weight will be 0.23 lb.
2.1.6 THE FEASIBILITY OF FABRICATION AND TESTING OF GRAPHITE EPOXY COMPOSITE TUBES

ON THE BASIS OF DISCUSSIONS WITH POTENTIAL MATERIAL SUPPLIERS, SPACE STATION STRUCTURAL ENGINEERS, COMMERCIAL SUPPLIERS OF GRAPHITE EPOXY TUBULAR PRODUCTS AND MODEL MANUFACTURERS, THE WRITER BELIEVES THAT A 1/4 SCALE SPACE STATION MODEL WOULD REQUIRE TUBES ABOUT 1/2 IN. DIAMETER AND 21 TO 54 IN. LONG. THE TUBES WILL PROBABLY REQUIRE ABOUT 4 LAMINATIONS OF PRE-PREGS HAVING A THICKNESS OF ABOUT 0.005 INCH EACH AND COMBINING TO PRODUCE A TOTL WALL THICKNESS OF ABOUT 0.010 IN. MATERIALS SIMILAR TO RT/55/EP20 GRAPHITE/EPOXY ARE EXPECTED TO BE USED BECAUSE HIGHER NITROUS GRAPHITE FIBERS ARE TOO BRITTLE FOR FABRICATIONS REQUIRING LARGE CURVATURES SUCH AS FOR SMALL TUBES.

TO MAXIMIZE THE LONGITUDINAL STIFFNESS OF THE TUBES WHILE MAINTAINING ADEQUATE RESISTANCE TO TUBE SPLITTING DURING COMPRESSIVE LOADINGS, THE ORIENTATIONS OF THE FIBERS ARE EXPECTED TO BE ABOUT ± 30° RELATIVE TO THE AXIS OF THE TUBE.

OF THE SEVERAL WAYS TO FABRICATE THE TUBES, IT IS EXPECTED THAT HAND LAYUP OF PREARRANGED PLIES INCLUDING A MINICELL WILL PROBABLY BE THE CHOICE BECAUSE OF THE CONTROL WHICH CAN BE EXERTED DURING THE MANUFACTURING PROCESS AND THE FACT THAT MOST GRAPHITE EPOXY STRUCTURES OF THIS SIZE ARE CURRENTLY MADE IN THIS MANNER. THE EXPENSES FOR TELLING ARE MINIMAL AND THE TETIS BSL (MEASURED BY COMMERCIAL PRODUCTS STANDARDS) IS SMALL. IN THIS PROCESS, THE USUAL TECHNIQUE IS TO COAT THE MINICELL WITH A RELEASE AGENT (E.G., SILICON), ROLL THE MINICELL OVER THE PREARRANGED AND PRECUT PLIES, COVER WITH SHRINK TAPE,
AND LUTE AT ABOUT 350°F IN AN OVEN.

THE REMOVAL OF THE CORED TUBES FROM THE MANOREL
CAN BE ACHIEVED IN SEVERAL WAYS. A SMALL AMOUNT OF
THREK IN THE TUBES IS VERY HELPFUL AND IS OFTEN USED
WHERE ONLY THE AVERAGE PROPERTIES OF THE TUBE ARE
SIGNIFICANT. THIS MAY BE ACCEPTABLE IN THIS CASE
BECAUSE THE CHARACTERISTIC MODES OF THE SPACE STATION
MODEL WILL ONLY REFLECT THE AVERAGE VALUE OF
TUBE STIFFNESS IN TENSION AND COMPRESSION. SPLIT
MANORELS ALSO PROVIDE FEASIBLE OPTIONS AND
THE WRITER SUGGESTS THAT THE TECHNIQUE DESCRIBED
IN SECTION 2.1.7 MAY BE THE SIMPLEST PROCEDURE.

AFTER THE TUBES ARE MANUFACTURED, SOME EFFECTIVE MEANS
WILL BE NEEDED TO CLASSIFY THEM IN A GO/NO-GO
SITUATION FOR ACCEPTANCE AND FOR MATCHING THEM
SO THE TUBES INSTALLED IN A GIVEN TRUSS BAY ARE
OF EQUAL MASS AND STIFFNESS. A TECHNIQUE FOR
ACHIEVING THIS GOAL IS OUTLINED IN SECTION 2.1.8.
THE TECHNIQUE INvolVES WEIGHING THE TUBES AND
VIBRATING THEM IN A SIMPLE GRIP DEVICE TO
OBTAIN THE NECESSARY DATA FOR TUBE CLASSIFICATION.
ENGINEERING INCORPORATED
41 Research Dr. • Langley Research Park
HAMPTON, VIRGINIA 23666
(804) 865-0100

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2.1.7 USE OF AIR OR WATER PRESSURE TO REMOVE
THIN-WALLED COMPOSITE TUBES FROM CYLINDRICAL MANORELS

GRAPHITE REINFORCED COMPOSITE TUBES HAVING
DIMENSIONS OF APPROXIMATELY 1/2 IN DIAMETER,
0.015 IN WALL THICKNESS, AND 45 INCH LENGTH
ARE OF INTEREST FOR CONSTRUCTING A DYNAMIC
MODEL OF THE SPIKE STATION. SUCH TUBES ARE
USUALLY MADE BY WRAPPNG COMPOSITE MATERIALS
(USUALLY CARBON-FIBER) AROUND A CYLINDRICAL
MANOREL, OVERWRAPPING THE COMPOSITES WITH
A HEAT SHRINK TAPE, AND CURING IN AN OVEN.
THE RESULT OF THE BORED MANOREL IS A SNUG
FIT TO THE TUBE ON THE MANOREL, AND SINCE
THE WALL IS VERY THIN, REMOVAL OF THE TUBE FROM
THE MANOREL WITHOUT DAMAGING THE TUBE IS OFTEN
CHALLENGING. THIS NOTE SUGGESTS THAT THIS CAN
BE DONE BY EXPANDING THE TUBE DIAMETER WITH AIR
OR WATER UNDER PRESSURE.

THE SPLITTING STRESS IN A TUBE UNDER INTERNAL
PRESSURE IS:

\[ \sigma = \frac{pd}{\text{t}^2} \]

AND

\[ \sigma = \frac{pd}{\text{t}^2} = \frac{E}{\text{r}^2} = \frac{pd}{\text{r}^2} \]

SINCE \( d = \pi d \) AND \( d = \pi \text{r} \)

\[ \frac{\text{d}}{\text{r}} = \pi = \frac{pd}{\text{r}^2} \]
FOR PURPOSES OF ANALYSIS, ASSUME THE WORST CASE, i.e., ALL FORCES ARE CIRCUMFERENTIAL, AND THE MATERIAL IS A HIGH MODULUS STEEL/GRAPHITE PROXY (P755/E934) WITH THE FOLLOWING PROPERTIES

\[ F_{th} = 1000 \text{ MPa} = 1.45 \times 10^6 \text{ psi} \]
\[ E_{th} = 365 \text{ GPa} = 53.4 \times 10^6 \text{ psi} \]

ASSUME A SAFETY FACTOR ON STRESS OF 1.5, THE ALLOWABLE STRESS IS THEN

\[ F_{th}^a = 96.7 \times 10^3 \text{ psi} \]

AND THE ALLOWABLE STRAIN IS

\[ \varepsilon = \frac{F_{th}^a}{E_{th}} = \frac{96.7}{53.4 \times 10^6} = 1.81 \times 10^{-3} \text{ in/in} \]
\[ \cong 0.002 \text{ in/in} \]

THE INTERNAL PRESSURE REQUIRED TO PRODUCE THIS STRAIN IS

\[ P = \frac{2 \varepsilon F}{d} = \frac{2 \times 96.7 \times 10^3}{0.5} \]

FOR A TUBE WITH \( d = 0.5 \text{ in} \) \( \varepsilon = 0.01 \), THE ALLOWABLE PRESSURE IS

\[ P = \frac{2 \times 0.01 \times 96.7 \times 10^3}{0.5} = 3.87 \times 10^3 \text{ psi} \]

NOTE THAT FOR A MATERIAL SUCH AS THORNELL OR CELION 3K OR 6K, THE ALLOWABLE PRESSURE WOULD BE PROPORTIONAL TO THE ALLOWABLE STRESS, e.g., FOR A SIMILAR TUBE.
\[
P_a = \frac{2 \times 0.01 \times (825 \times 10^3/1.5)}{0.5} = 6 \times 10^3 \text{ psi}
\]

However, since the modulus is substantially lower (\( E = 20.7 \times 10^6 \text{ psi} \)), the allowable strain produced by the allowable stress is much higher, i.e.

\[
\varepsilon_a = \frac{\sigma_a}{E} = \frac{(245/1.5) \times 10^3}{20.7 \times 10^6} = 0.0072
\]

The effect of this is that the tube can be enlarged 3.5 times as much to get it off the mandrel.

The proposed design of the mandrel consists of making it from a tube which is fitted with very small radially drilled holes. The tube is then fitted with a pressure regulated fluid tap at one end and plugged at the other.

During the process of making & curing the tube, it is expected that some resin will bleed into the small holes unless they are covered in some way. It is believed that a suitable thin tape could be placed over the holes to adequately stop the bleeding. The tape would be readily removed by the pressure. However, since the holes are very small, the shearing forces to shear the resin plugs would be minimal since leakage would be expected at the removal pressures, the plugs may not create any problems whatsoever.
2.1.8 Technique for Needle Tube Selection/Graining:

1. Measure with length gauge.

2. Insert end on tapered ring gauge for inside diameter.

3. Weigh tube.

4. Place one end of tube in a gripper and add a mass to the other end.

5. Deflect end and release to measure the natural frequency.

6. Rotate gripper 90° and repeat item 5.

7. Accept or reject tube on basis of results from items 3, 5, and 6.
2.2. MODULI AND OTHER MASSES

It seems probable that most of the components that make up the major portion of the mass of the space station can be treated as rigid bodies when analyzing the overall dynamics of the station. Since they are pressurized vessels or other relatively compact systems, their lowest natural frequencies will be much higher than the natural frequencies of the highest overall vehicle modes of interest from the standpoint of vehicle guidance, control, or stabilization. However, proper dynamic scaling of these masses is essential and must include at least the following:

1. Mass
2. Mass distribution relative to the vehicle coordinate axes
3. Mass moments of inertia about the principal axes of the body in question
4. The spatial distribution of connections between the body and the vehicle structure or other points of attachment
5. The effective stiffness of all attachments in the three mutually perpendicular coordinate directions and about these axes.
6. The damping distributions throughout the mass attachment system.

The significance of these factors can be realized by review of a typical system such as shown by the sketches of potential ICG structures shown in figure 8. The effect of such masses on bending or torsion modes of the overall structure is readily apparent by visualizing the motions of the mass and the mass attachments under conditions where the mass may lie near a node or an antinode of a bending mode.
OR TORSION MODE.

IT APPEARS THAT THE MASSES, LOCATED IN THE MODULES USED FOR LABORATORIES, HABITABILITY MODULES AND LOGISTICS WILL BE ACCOMPANIED AROUND THE PERIMETERS AND MOSTLY ATTACHED TO THE CORE SHELL. THIS WILL MEAN THAT METALLIC SHELLS WITH ATTACHMENTS OR OUTLETS TO SIMULATE THE MASSES AND MASS MOMENTS OF INERTIA, AND WELDMENTS FOR ATTACHMENT OF STRUTS TO THE TRUSS STRUCTURES, WOULD PROVIDE ATTRACTIVE MODELING OPTIONS. SUCH TECHNIQUES WILL BE NECESSARY TO KEEP THE DUMPING OF THE MODEL STRUCTURES DOWN. THE ADDITION OF THE DUMPING IF DESIRED IS EASILY ACCOMPLISHED.
2.3 Solar Arrays and Large Antenna Dishes

The principal difficulty of modeling the dynamics of large lightweight structures such as the solar panels or antenna dishes arises from the fact that they must be tested in a high-atmospheric-pressure environment. The vibrations of such structures are imposed by the surrounding air in the form of apparent mass forces and damping forces. Such forces have no counterpart for the full scale space station motions in orbit and the objective is to reduce them to the maximum extent possible on the model.

The apparent mass of the air surrounding a plate is generally measured in terms of the ratio of the mass of the air in a cylinder surrounding the plate to the mass of the plate. As shown by Figure 8, each of the 16 solar panels has dimensions of about 15 ft. by 80 ft. and it is expected that each panel will weigh about 600 lbs. For a 1/4 scale model, each panel would have dimensions of about 3.75 ft. by 20 ft. and would weigh about (600/64) or about 9.38 lbs. Thus the ratio of the apparent air mass to the panel mass would be approx.

\[ K = \frac{4 \pi d^2 \rho g}{4} = \frac{\pi (3.75)^2 (20)}{9.38} \approx 1.80 \]

Or, the mass of the surrounding air is about twice the mass of the panel.

The literature does not show much information on the effects of apparent mass ratios of this size. However, much smaller ratios have significant impact on aircraft flutter, the work of Sewall, Misermentino and Dappa, ref. 3, indicates that for
A lightweight triangular structure having a mass ratio of about three, the surrounding air dominated the mass of the system for vibrations in the fundamental mode. The results also show that the apparent mass of the air, as determined from the frequencies of the first mode vibrations, also approximates the mass of the air contained in three intersecting cones originating from the three corners of the triangle and having diameters equal to the distances between the adjacent edges of the triangle.

The intent of the aforementioned statements is that it will be necessary to simulate the solar panels and probably the antenna dishes by some structures which duplicate the mass and stiffness distributions of the panels but minimize blocking of air by permitting it to flow through the panel structure. A gridwork of suitably chosen rods or cables would appear to offer a solution.

To minimize the damping of the simulated panel structures which will arise from the flow-through of air, care must be taken to minimize the generation of vorticity. As shown in Reference 4, a sharp edged flexible device which creates vorticity is a much more effective damper than a rounded rigid body which merely redirects the air. The implication is clearly that the elements of the grid should be as widely separated as possible and should have smooth rounded surfaces normal to the planes of the panels.
J. Design and Fabrication of Model Support System

Current plans for the design of the large spaceframe structures inhibitor were discussed with Mr. Robert Matzner of the U.S. Structural Dynamics Branch on 5/4/65. The synthesis of these discussions is that the laboratory outline permits the installation of the model in the orientation shown. This is the recommended orientation for several reasons including:

1. Minimization of gravitational effects,
2. Conveniences, simplicity, and minimum costs of model tests,
3. Safety of model structural and personnel during model assembly and testing.

Some considerations relative to these terms are elaborated in the following sections.
3.1 Minimization of Gravitational Effects

Since models tested in earth based laboratories will be subjected to gravitational forces which have no counterpart during orbital flight of the spacecraft, it is desirable to reduce the effects of these gravitational forces as much as possible. If the model is hung as a pendulum, which appears to be the only attractive option, the gravitational forces always tend to restore the model to a condition of minimum potential energy. The integrated effect of gravitational forces is the creation of three rigid body modes (two translational modes and one rotational mode) in a plane normal to the support cables.

The two translational modes (lateral & longitudinal) have the frequency of a simple pendulum

\[ \omega = \sqrt{\frac{g}{L}} \]

where \( g \) is the gravitational constant and \( L \) is the length of the support cable.

As demonstrated on page 63 in this report, the rotational mode is the bi-planar pendulum mode where

\[ \omega = \sqrt{\frac{m}{I} \cdot \frac{g}{L}} \approx \sqrt{\frac{3}{L}} \]

Note that \( m \) is the length of the model, \( I \) is the radius of gyration, and that \( m/I \approx \sqrt{3} \).

It will be shown that \( L \) can be made sufficiently large that the support frequencies will lie below the band of natural frequencies of the bi-planar modes of interest, thus...
BY MAXIMIZING $\theta$, THE COUPLING OF THE MODEL ELASTIC, INERTIA AND DUMPING FORCES WITH THE CONVOLUTIONAL FORCES IS MINIMIZED.

WITH REFERENCE TO THE SKETCH OF THE FACILITY SHOWN ON PAGE 52, THE PROPOSED HORIZONTAL MODEL TEST CONFIGURATION WILL PERMIT THE CENTER OF THE MODEL KEEL TO BE PLACED APPROX. 25 FEET ABOVE THE FACILITY FLOOR AND ALLOW FOR THE CASE WHERE THE SOLAR PANELS ARE ORIEN TED AT $\theta = 90$ DEGREES. THE ALLOWANCE OF 12 FEET FOR THE MODEL SUPPORT PLATFORM AND 5 FEET FOR KEEL THICKNESS AND ATTACHMENT OF CABLES TO THE KEEL SUPPORTED COMPONENTS LEAVES A CLEAR CABLE LENGTH OF APPROXIMATELY 120 FEET FOR MODEL SUSPENSION.


$$w = \sqrt{\frac{g}{\delta_{st}}}$$

WHERE $\delta_{st}$ IS THE STATIC DEFLECTION OF THE MODEL ON THE ELASTIC SUPPORT CABLES.

SUBSEQUENT SECTIONS OF THE REPORT PRESENT THE RESULTS OF ANALYSES WHICH EXAMINE VARIOUS ASPECTS
OF THIS SUPPORT SYSTEM, INCLUDING THE DERIVATION AND DISCUSSION OF THE FREQUENCY SEPARATIONS BETWEEN THE MODEL ELASTIC MODOES AND THE SEVERAL RIGID BODY SUPPORT MODALS.
3.2 CONVENIENCE, SIMPLICITY AND MINIMUM COST OF MODEL TESTS

THE RECOMMENDED MODEL TEST CONFIGURATION AS SHOWN BY THE SKETCH ON PAGE 52 OFFERS THE ADVANTAGE THAT NEARLY ALL OF THE MODEL ASSEMBLY IS ACCOMPLISHED WITH PERSONNEL POSITIONED ON THE FLOOR AND WORKING AT LEVELS BETWEEN THE FLOOR AND SHOULDER HEIGHT. IN A FEW INSTANCES, IT WILL BE NECESSARY TO WORK FROM A LOW MOBILE PLATFORM BUT NO SITUATIONS ARE ENVISAGED WHERE MODEL TECHNICIANS OR RESEARCH PERSONNEL ARE REQUIRED TO WORK AT HEIGHTS ABOVE ABOUT 20 FEET.

THE ANTICIPATED MODEL INSTALLATION AND TEST PROCEDURE IS AS FOLLOWS:

6. THE MODEL SUPPORT PLATFORM IS REMOVED FROM STORAGE, ASSEMBLED (ASSUMED TO BE MADE IN SEVERAL PIECES FOR EASE OF STORAGE), AND ATTACHED TO THE VERTICAL HOIST SYSTEM BY Riggers. IT IS THEN OPERABLE IN AN UP AND DOWN SENSE BY TEST TECHNICIANS.

6. THE MODEL SUPPORT PLATFORM IS THEN LOWERED TO A CONVENIENT HEIGHT AND ALL SUPPORT CABLES ARE ATTACHED IN A PREARRANGED PATTERN FOR THE MODEL TEST CONFIGURATION OF INTEREST.

C. THE PLATFORM IS THEN RAISED TO PLACE ALL THE SUSPENSION CABLES IN LIGHT TENSION AS THEY ARE ATTACHED TO THE FLOOR. WHEN ALL SUSPENSION CABLES ARE ATTACHED, THE PLATFORM IS RAISED TO THE HEIGHT WHERE THE TENSION IN A CABLE WILL SUPPORT IT'S RESPECTIVE MASS, AT THE DESIRED MODEL ASSEMBLY HEIGHT.
d. Consistent with a prearranged plan, the various components of the model are taken to their respective points for assembly, the appropriate cables are attached to the components, and the components are then joined together to form the model. Assembly would start from the base and initially involve the joining of the heavy components including the habituation modules, the labs and the orbiter to each other and to the base truss elements. Assembly would then proceed outwards to incorporate the keel, the traverse boom, the solar panels and other model components. It may also be desirable to form several sub-assemblies to assure their balance and orientation before assembling them together to form the complete structure. In this manner, the model would be subjected to a very low gravity induced state of stress and should provide the best opportunity for simulating zero-G conditions relative to joint non-linearities and imbalances of structural responses.

b. Once the model is assembled and instrumented to the extent possible at ground level, it is raised to this necessary height by raising the support platform to complete the addition of antennae and to rotate the solar panels to θ = 90° when necessary. (The latter configuration represents the extreme model test height as shown on the sketch on page 51.)

f. All model tests are then conducted at the highest height possible for that configuration. This minimizes the complexity and costs of instrumenting and monitoring the model and its convenience will substantially reduce the costs of test fixtures and the conduct of the tests.
3.3 SAFETY OF MODEL STRUCTURES AND PERSONNEL DURING MODEL ASSEMBLY AND TESTING


A. THE FULL SCALE SPACE STATION WILL BE DESIGNED TO FUNCTION UNDER ACCELERATIONS OF THE ORDER OF 0.04-g, AND AS A CONSEQUENCE OF THE NEED TO MINIMIZE THE WEIGHT TO ORBIT, LITTLE STRUCTURAL "FAT" IS EXPECTED. HENCE, THE MODEL, SCALLED TO THE SAME STRESS LEVEL AS THE Prototype, WILL NOT BE ABLE TO SUPPORT ITSELF UNDER 1-g LOADS EXCEPT IN SMALL SECTIONS. THE PRECISELY, ESSENTIALLY CONTINUOUS SUPPORT SYSTEM EFFECTIVELY ELIMINATES THAT PROBLEM. ALSO, BECAUSE OF THE FRAGILITY OF THE JOINTS AND THE TUBULAR MEMBERS OF THE TRUSS STRUCTURE, MODEL TEST TECHNICIANS MUST WORK WITH EXTREME CAUTION TO AVOID APPLICATION OF DAMAGING LOADS. GROUND BASED ACCESS TO MOST PARTS OF THE MODEL WILL PERMIT THE EXECUTION OF REASONABLE PRECAUTIONS WHILE EXPEDIENT EXECUTION OF THE MODEL ASSEMBLY AND TESTING TASKS.

B. ANY VERTICAL ORIENTATION OF THE MODEL KEEL OR TRANSVERSE BOOM WILL REJECT IN MODEL TEST PERSONNEL WORKING AT HEIGHTS OF ABOUT 100 FEET TO SERVICE A 1/4 SCALE MODEL. THE GANTRY AND SAFETY PRECAUTIONS TO MAKE THIS POSSIBLE FOR A SOFTLY SPRUNG MOVABLE MODEL, EVEN FOR WORKERS NOT SUBJECT TO DISCOMFORT WHILE WORKING AT HEIGHTS UP TO 10 STORIES, WOULD PRESENT A VERY STRONG IMPEDIMENT.
TO DYNAMIC TESTING OF A FIXEDILE MODEL. THE PROPOSED, ESSENTIALLY GROUND LEVEL, MODEL PREPARATION AND TESTING PROCEDURE WILL NOT ONLY REMOVE THE PERSONNEL HAZARDS BUT IT MAY BE THE ONLY FEASIBLE OPTION FOR ACHIEVING A SATISFACTORY TEST PROGRAM FOR THE COMPLEX AND SENSITIVE SAME STATION MODEL.
3.4 DEGREES OF FREEDOM RELATING TO INFLUENCE OF GRAVITATIONAL EFFECTS ON MODEL SUPPORT SYSTEM

THE NEAR COINCIDENCE OF FREQUENCIES OF DIFFERENT NATURAL MODES OF VIBRATION CAUSES COMPLICATIONS IN TESTING AND DATA ANALYSIS. IN SOME CASES, THE PROBLEMS INVOLVE CROSSING OF STRUCTURAL ACTIONS; IN OTHER CASES THEY ONLY INVOLVE INTERFERENCE DUE TO SIMULATANEOUS ACTIONS. IN EITHER CASE, WHEN INTERFERENCE EFFECTS ARE THE RESULT OF EXTRINSIC FORCES, SUCH AS GRAVITY FORCES ON THE MODEL, MINIMIZATION OF THE INTERFERENCE, USUALLY BY FREQUENCY SEPARATION, IS DESIRABLE.

FOR GENERAL CONSIDERATIONS, ASSUME THAT THE LOWEST NATURAL ELASTIC FREQUENCY OF INTEREST FOR STRUCTURAL MODS OF THE SPACE STATION IS:

\[ f = f_E \sqrt{\frac{1}{\omega}} \]

\[ \omega = f_E (2\pi) \]

FOR REPLICA SCALING, \( \omega_{m,e} = \frac{1}{\lambda} \omega_{f,e} \) WHERE THE SUBSCRIPTS \( m,e \) DENOTE MODEL AND FULL SCALE VALUES RESPECTIVELY, \( \lambda \) IS THE SCALE FACTOR (\( \lambda \ll 1 \)).

THE FREQUENCY SEPARATION IS THEN

\[ \omega_{m,s} = \frac{1}{\lambda} \omega_{m,e} = \frac{1}{\lambda} \frac{1}{\lambda} \omega_{f,e} \]

\[ \frac{1}{\lambda} \omega_{f,e} \]

IN SUBSEQUENT SECTIONS, \( \omega_{m,s} \) AND \( \omega_{f,e} \) WILL BE EXAMINED TO FURTHER REFINING \( \lambda \).
3.5 DETERMINATION OF MODEL SUPPORT FREQUENCIES ON CABLE MOUNTING SYSTEM

As shown on page 53, the proposed cable mounting system for the model involves suspending it from an overhead platform by numerous elastic cables. The model is oriented so that the keel and the transverse boom are parallel to the floor (Y-Z plane) and the flight direction (X-axis) is up. The model will thus be permitted to undergo motions under elastic restraints in six degrees of freedom. For purposes of analysis, it is convenient to group these motions as follows:

1. Pendulum motions in the Y-Z plane including rotations (θ, bifilar rotations) about the X axis.

2. Plunging motions in the X-direction including rotations about the Y-axis (θ, pitching rotations) and Z-axis (ψ, rolling rotations)

For simplicity of computation of the model motions on the support system, the following assumptions are made:

1. The model is rigid.

2. The distribution of elastic supports is the same as the distribution of mass across the X-Y plane.
3.5.1. Determination of500064 natural frequencies. Motions are along y or z axes and about x-axis. Consider case for motions along y-axis.

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The model consists of a series of masses connected by a truss and supported by a series of elastic cables. The system is assumed to undergo lateral motions composed of superimposed translations $\xi$ and rotations $\phi$. As a result of the fixed length of the supports, these motions cause the model to move up and down as it moves laterally. The resulting and translational frequencies will be determined by applying energy methods and using Lagrange's equations.

\[
\frac{\partial}{\partial t} \left( \frac{\partial T}{\partial q} \right) - \frac{\partial T}{\partial q} + \frac{\partial V}{\partial q} = 0
\]

Where $T$ and $V$ have the kinetic and potential energies, respectively, and $q_0$ is a generalized coordinate, y or $\phi$.

\[
T = \frac{1}{2} \sum_{i=1}^{n} M_i \dot{y}_i^2 \quad \frac{1}{2} \sum_{i=1}^{n} M_i (\dot{\phi}_i + \dot{\phi}_i) \phi_i^2 \quad = \frac{1}{2} \sum_{i=1}^{n} M_i \left( \dot{\phi}_i^2 + \dot{y}_i^2 + \dot{\phi}_i^2 \right)
\]

Since $\phi_i = \phi_2 = \phi$; $\dot{y}_i = \dot{y}_2 = \dot{y}$; $\sum_{i=1}^{n} M_i \dot{y}_i^2 = I_{y}$; $\sum_{i=1}^{n} M_i \phi_i^2 = I_{\phi}$.

\[
T = \frac{1}{2} M \dot{y}^2 + \frac{1}{2} \sum_{i=1}^{n} M_i \dot{\phi}_i^2 + \frac{1}{2} I_{y} \phi^2
\]

\[
= 0 \quad \text{since } \phi_i \text{ is measured from } O_A.
\]
Since, for this calculation, the model is assumed to be rigid, the upward motions of all masses are equal to the upward motions of the masses $m_0$ and $m_n$. Hence

$$v = \frac{Mq}{2} \frac{v_0^2}{2} + \frac{Mq}{2} \frac{v_n^2}{2}$$

But

$$v_0^2 = \frac{1}{k^2} \left( \gamma + \frac{\gamma_0}{\phi} \right)^2 \quad \gamma_0^2 = \frac{1}{k^2} \left( \gamma - \frac{\gamma_0}{\phi} \right)^2$$

And assuming that $|\gamma_0| = |\gamma_0|$

$$v = \frac{Mq}{2} \left( \gamma^2 + \frac{\gamma_0^2}{\phi} \right)$$

Applying Lagrange's equations, we obtain

$$M\ddot{y} + \frac{Mq}{2} y = 0$$

And

$$I_0 \ddot{\phi} + \frac{Mq}{2} \phi^2 \dot{\phi} = 0$$

For a uniform beam, $\gamma_0 = \frac{1}{2} \gamma$ and $I_0 = \frac{1}{12} ML^2$.

The above equations reduce to

$$\ddot{y} + \frac{\gamma_0}{2} y = 0$$

And

$$\ddot{\phi} + \frac{3\gamma_0}{2} \phi = 0$$

The resulting uncoupled natural frequencies are

$$\omega_{1,2} = \sqrt{\frac{2}{L}} \quad \text{and} \quad \omega_{3,4} = \sqrt{\frac{3\gamma_0}{L}}$$
3.5.2 DETERMINATION OF PLUNGING AND
ROTATIONAL NATURAL FREQUENCIES OF MODEL ON
CABLES. MOTIONS ARE ALONG X AND ABOUT YZ AXES.

THE SYSTEM CONSISTS OF A SERIES OF MASSES
CONNECTED BY A TRUSS AND SUPPORTED BY A SERIES
OF ELASTIC CABLES. THE SYSTEM IS ASSUMED TO UNDERGO
SIMULTANEOUS TRANSLATIONAL MOTIONS (X) AND PITCHING
MOTION'S θ. THE TWO NATURAL FREQUENCIES OF THE SYSTEM
WILL BE DETERMINED BY APPLYING ENERGY METHODS
AND LAGRANGE'S EQUATIONS. (RESULTS FOR θ ARE IDENTICAL)

\[
\frac{d}{dt} \left( \frac{d}{dt} \phi \right) = 0
\]

WHERE T & V ARE THE KINETIC & POTENTIAL ENERGIES, RESPECTI-
VELY, AND θS IS A GENERALIZED COORDINATE, θ, OR θ.

\[
T = \frac{1}{2} \sum_{i=1}^{n} M_i \dot{x}_i^2 = \frac{1}{2} \sum_{i=1}^{n} M_i (\dot{x}_i + \dot{\phi} \dot{\theta})^2 = \frac{1}{2} \sum_{i=1}^{n} M_i (\dot{x}_i^2 + 2 \dot{x}_i \dot{\theta} \dot{\phi} + \dot{\phi}^2)
\]

Since \( \dot{\theta}_i = \dot{\phi}_i = 0 \); \( \dot{x}_i = \dot{x}_i \); \( \sum \dot{x}_i = 0 \); \( \sum M_i \dot{x}_i = M \dot{x}_c \).

\[
T = \frac{1}{2} M \dot{x}_c^2 + \dot{\theta} \sum M_i \dot{x}_i + \frac{1}{2} I_{cg} \dot{\theta}^2
\]

= 0 since \( \dot{x}_c \) IS MEASURED FROM C.G.
\[ V = \frac{1}{2} \sum_{i=1}^{n} k_i \Theta_i^2 \]

WHERE \( k_i \) IS THE SPRING CONSTANT OF CABLE \( i \) AND \( \Theta_i \) IS ITS TOTAL DEFLECTION; THEN

\[ V = \frac{1}{2} \sum_{i=1}^{n} k_i (x_i + L_i \Theta_i)^2 = \frac{1}{2} \sum_{i=1}^{n} k_i (x_i^2 + 2L_i x_i \Theta_i + L_i^2 \Theta_i^2) \]

\[
= \frac{1}{2} \sum_{i=1}^{n} k_i^2 x_i^2 + k_i L_i \Theta_i \sum_{i=1}^{n} k_i L_i^2 + \frac{1}{2} \sum_{i=1}^{n} k_i L_i^2 \Theta_i^2
\]

TO REDUCE THE TRANSMISSION OF GRAVITY LOADS THROUGH THE RELATIVELY WEAK KEEL STRUCTURE IT IS DESIRABLE TO HAVE \( k_i \) PROPORTIONAL TO \( M_i \), i.e., \( k_i = \epsilon M_i \)

\[ V = \frac{1}{2} x^2 \epsilon M + \kappa \Theta \sum_{i=1}^{n} M_i \Theta_i + \frac{1}{2} \Theta^2 \epsilon I_{cg} \]

= 0 SINCE \( L_i \) IS MEASURED FROM C.G.

APPLYING LAGRANGES EQUATION WE OBTAIN

\[ M \ddot{x} + \kappa M x = 0 \quad \text{or} \quad \ddot{x} + \kappa x = 0 \quad \text{ORIGINAL PAGE IS OF POOR QUALITY} \]

AND

\[ I_{cg} \ddot{\Theta} + \kappa I_{cg} \Theta = 0 \quad \text{or} \quad \ddot{\Theta} + \kappa \Theta = 0 \]

THUS BOTH THE TRANSLATIONAL AND PITCHING MODES HAVE THE SAME NATURAL FREQUENCY

\[ \omega_{n_i} = \sqrt{\frac{k_i}{M_i}} = \sqrt{\frac{\kappa}{\Theta}} \]

THIS IS BECAUSE THE SPRING FORCE FOR PITCHING IS PROPORTIONAL TO THE DISTANCE FROM THE C.G. WHEREAS FOR BIFLAR ROTATIONS IT IS THE SAME FOR ALL POINTS OF THE STRUCTURE - THEY ALL RISE THE SAME AMOUNT FOR A GIVEN \( \Theta \).
RELATIVE TO THE INFLUENCE OF SUPPORT SYSTEM RIGID BODY MODES ON THE MODEL ELASTIC MODES, THE FOLLOWING ASSUMPTIONS SEEM REASONABLE:

1. ROTARY RIGID BODY MODES PRIMARILY INFLUENCE ANTI-SYMMETRIC ELASTIC MODES

2. PLUNGING AND PENDULAR RIGID BODY MODES PRIMARILY INFLUENCE SYMMETRIC ELASTIC MODES

3. THE NATURAL FREQUENCY OF THE LOWEST ANTI-SYMMETRIC ELASTIC MODE > 2.5X NATURAL FREQUENCY OF THE LOWEST SYMMETRIC MODE

4. THE NATURAL FREQUENCY OF THE LOWEST FULL-SCALE SYMMETRIC ELASTIC MODE IS 0.1 Hz IN X-Z OR Y-Z PLANE

5. FOR PURPOSES OF FIRST APPROXIMATION, THE SPACE STATION CAN BE TREATED AS A BEAM.

FOR THESE APPROXIMATIONS, THE FREQUENCY SEPARATION FOR THE CASES OF INTEREST ARE

PENDULAR & LATERAL SYMMETRIC BENDING

\[
\alpha_1 = \frac{\omega_{m,e}}{\omega_{m,s}} = \frac{\sqrt{2}}{\sqrt{18}} \frac{0.625}{0.628} = \frac{11.07 \times 10^{-2}}{2} \frac{\sqrt{2}}{\sqrt{18}} = 10.97 \times 10^{-2} \frac{\sqrt{2}}{\sqrt{18}}
\]

BIFILAR PENDULAR & LATERAL ANTISYMMETRIC BENDING

\[
\alpha_2 = \frac{\omega_{m,e}}{2} \left( \frac{\omega_{m,s}}{\omega_{m,s}} \right)^2 = \frac{\sqrt{2}}{2} \frac{(2.5)(0.628)}{\sqrt{2}} = \sqrt{\frac{2}{2}} \frac{(2.5)(0.625)}{\sqrt{2}/2} = 15.77 \times 10^{-2} \frac{\sqrt{2}}{\sqrt{18}}
\]
RELATIVE TO THE INFLUENCE OF SUPPORT SYSTEM RIGID BODY MODES ON THE MODEL ELASTIC MODES, THE FOLLOWING ASSUMPTIONS SEEM REASONABLE:

1. ROTARY RIGID BODY MODES PRIMARILY INFLUENCE ANTI-SYMMETRIC ELASTIC MODES

2. PLUNGING AND PENDULAR RIGID BODY MODES PRIMARILY INFLUENCE SYMMETRIC ELASTIC MODES

3. THE NATURAL FREQUENCY OF THE LOWEST ANTI-SYMMETRIC ELASTIC MODE > 2.5 X NATURAL FREQUENCY OF THE LOWEST SYMMETRIC MODE

4. THE NATURAL FREQUENCY OF THE LOWEST FULL-SCALE SYMMETRIC ELASTIC MODE IS 0.1 Hz IN X-Z OR Y-Z PLANE

5. FOR PURPOSES OF FIRST APPROXIMATION, THE SPACE STATION CAN BE TREATED AS A BEAM

FOR THESE APPROXIMATIONS, THE FREQUENCY SEPARATION FOR THE CASES OF INTEREST ARE

PENDULAR & LATERAL SYMMETRIC BENDING

\[ \alpha_1 = \frac{(\omega_{ME})}{(\omega_{MS})_1} = \frac{\frac{1}{2}(0.625) \sqrt{\frac{12}{1816}}}{0.0238} = \frac{11.07 \times 10^{-2} \sqrt{2}}{\lambda} \]

BIFILAR PENDULAR & LATERAL ANTI-SYMMETRIC BENDING

\[ \alpha = \frac{(\omega_{ME})}{(\omega_{MS})_2} = \frac{(\frac{3}{2})(2.5)(0.625) \sqrt{\frac{\sqrt{38}}{2}}}{x} \frac{(2.5)(0.625)}{\sqrt{159}} = 15.97 \times 10^{-2} \frac{\sqrt{2}}{\lambda} \]
PLUNGING & VERTICAL SYMMETRIC BENDING

\[ \alpha_3 = \frac{\left( \frac{WME}{W_{4.5}^3} \right)}{\sqrt{\frac{1}{2.5}}} = \frac{\sqrt{3}}{\pi} \frac{0.628}{1.9} = 11.07 \times 10^{-2} \sqrt{3/5} \frac{\lambda}{y} \]

PITCHING AND VERTICAL ANTI-SYMMETRIC BENDING

\[ \alpha_4 = \frac{\left( \frac{WME}{W_{4.5}^4} \right)}{\sqrt{\frac{1}{2.5}}} = \frac{\sqrt{3}}{\pi} \frac{(2.5)0.628}{1.9} = 27.8 \times 10^{-2} \sqrt{3/5} \frac{\lambda}{y} \]

Noting \( 1 < \sqrt{3/5} < 1 \), \( \alpha_3 \) is the smallest frequency. Separation is not possible to make all \( \omega_3 \) is large as possible for a given \( \lambda \) and therefore it is desirable to have \( \omega_3 \) is large as possible. If the length \( (\lambda_{sa}) \) is reduced, \( \sqrt{3/5} \) is reduced, and the frequencies effective, \( \omega_3 = \lambda \). But this would create a problem where the ging material must be assumed beyond the length of the cable. A more desirable situation would involve \( \lambda = \infty \) where all the elastic material plus a limited amount of inelastic material for cable movement is assumed below the support platform. The suggested procedure is outlined in the following section which shows the derivation of the bending of the various elements of the support cables. Note that the recommended procedure for supporting the model (see Page 64) employs numerous parallel cables of essentially the same length.

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2.4 DETERMINATION OF COMPOSITION OF CABLE BELOW SUPPORT PLATFORM

IT IS ASSUMED THAT THE CABLE CONSISTS OF AN ELASTIC MEMBER AND AN INELASTIC MEMBER.

<table>
<thead>
<tr>
<th>PLATFORM</th>
<th>PLATFORM</th>
<th>L - test length</th>
</tr>
</thead>
<tbody>
<tr>
<td>( L_{IE} )</td>
<td>( L_{IE} )</td>
<td>( L_{IE} )- length of inelastic material</td>
</tr>
<tr>
<td>( L_{E} )</td>
<td>( L_{E} )</td>
<td>( L_{E,0} )</td>
</tr>
<tr>
<td>( \delta_{ST} )</td>
<td>( \delta_{ST} )</td>
<td>( \delta_{E,0} ) - length of elastic material - no load</td>
</tr>
<tr>
<td>MODEL</td>
<td>MODEL</td>
<td>( L_{E} )- length of elastic material - loaded</td>
</tr>
</tbody>
</table>

\[ \beta = \frac{\text{percent elongation}}{100} \]  
\[ (\text{Sec p. 21}) \]

ASSUMPTIONS:

1. \( L_{IE} = \varepsilon L \)
2. \( \delta_{ST} = \beta L_{E,0} \)

THEN

\[ L = L_{IE} + L_{E,0} + \delta_{ST} \]
\[ = \varepsilon L + L_{E,0} + \beta L_{E,0} \]

OR

\[ L_{E,0} = \frac{L (1-\varepsilon)}{1+\beta} = \frac{\delta_{ST}}{\beta} \]
For convenience of hanging and adjusting, the model from the platform, an inelastic length of the cables of the type of 0.12 (ε = 0.1) seems reasonable. Also, information from rubber suppliers suggest that a value of 3 for β is reasonable. Then
\[ \varepsilon_{E0} = \frac{\varepsilon (1-\varepsilon)}{(1+3)} = \frac{(1-0.1)}{(1+3)} \varepsilon = 0.225 \varepsilon \]
\[ \delta_{ST} = \beta \varepsilon_{E0} = 0.675 \varepsilon \]
\[ \delta_{LE} = 0.10 \varepsilon \]
3.1 SUMMARY OF FREQUENCY SEPARATIONS FOR 1/4 SCALE MODEL WITH A SUSPENSION LENGTH OF 120 FEET AND A STATIC DEFORMATION OF 0.015%.

Substitution of $x=1/4$, $l=120$ feet, and $d_{ct}=0.015$ (61 feet and the assumptions made on page 66 of the following values for frequency separations $\alpha_i$ throughout $\alpha_4$.

$$\alpha_1 = \frac{11.07 \times 10^{-2}}{0.25} \frac{\sqrt{E}}{l} = \frac{11.07 \times 10^{-2} \sqrt{120}}{0.25} = 4.85$$

$$\alpha_2 = \frac{15.95 \times 10^{-2}}{0.25} \frac{\sqrt{E}}{l} = \frac{15.95 \times 10^{-2} \sqrt{120}}{0.25} = 6.98$$

$$\alpha_3 = \frac{11.07 \times 10^{-2}}{0.25} \frac{\sqrt{E}}{d_{ct}} = \frac{11.07 \times 10^{-2} \sqrt{81}}{0.25} = 3.98$$

$$\alpha_4 = \frac{21.80 \times 10^{-2}}{0.25} \frac{\sqrt{E}}{d_{ct}} = \frac{21.80 \times 10^{-2} \sqrt{81}}{0.25} = 10.01$$

Where $\alpha_1$ through $\alpha_4$ are defined as on pages 66 and 67.

The above values for the frequency separations should substantially minimize the influence of the rigid body motions, induced by gravitational forces, on the elastic modes of the space station structure.

The minimum gap situation to be encountered is the one where all rigid body modes interfere with all elastic modes. In this case, the frequency separation of interest is the one which compares the lowest frequency elastic mode with the highest frequency support mode, i.e., a model symmetric elastic bending with similar rotations, inserting some elastic extension, this results in the following value of $\alpha$ for the aforementioned test conditions.

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\[
\chi = \frac{1}{\lambda} \frac{\omega_{RE}}{\omega_{MS}} = \frac{1}{\lambda} \frac{\omega_{RE}}{\sqrt{3 \vartheta}} = \frac{1}{0.25 \sqrt{96.6}} = 2.8
\]

It is anticipated that the actual situation will be better because the finiteness of the cables will probably cause some reduction in the Eigen support frequencies.

It should be noted that because of nonlinearities of the rubber support cables which will probably be used to support the model, experimental data indicate that the effective spring constant of the plunging motions will vary and may as high as \((2g/\vartheta)^{1/2}\) instead of \((g/\vartheta)^{1/2}\). The impact of this would be a reduction of \(\Delta_3\) and \(\Delta_4\) as given on pages 65, 67, and to by \(\sqrt{a}\). It also is an indication to try to work the cables at a lower percentages of elongation, which tend to increase cable mass. Thus some tradeoffs may be necessary.
3.6 NATURE OF CABLES AND THEIR PROPERTIES

In the previous discussions, the cables were treated as elastic members, with effective masses (at the point of attachment to the model) which are small relative to the model masses they support. These cables must have large static deflections per unit stress or high resilience. This property is achieved by either using the inherent properties of a suitable material, such as a highly resilient rubber, or by configuring other materials such as high-strength steel into resilient structures such as coil springs.

The question is, will either approach yield sufficiently large static deflections for acceptable ratios of cable weight to suspended weight, to provide model support frequencies low enough to achieve acceptable model test conditions. As the results in the following sections will show, appropriate high-quality rubber offers a suitable solution; steel springs, because of their excessive weight, do not.

The sections which follow represent the results of analyses and experiments on the properties of various types of rubber samples. Also shown are the results for calculations on two types of high-strength steel springs.
3.7 CONSIDERATIONS ON SELECTION OF RUBBER FOR USE IN SPACE STATION MODEL SUPPORTS

RUBBERS MANUFACTURERS NOTE THAT THE EXTRUSION OF RUBBER IN THE RANGE OF ELASTICITY OF INTEREST FOR SUPPORTING THE MODEL OF THE SPACE STATION IS A VERY IMPRECISE TASK. MECHANICAL PROPERTIES SHOULD BE VERIFIED BY SIMPLE ELONGATION TESTS AND FREQUENT MEASUREMENTS USING SAMPLES OF THE SELECTED RUBBER STOCK. CREEP STUDIES ARE ALSO ADVISABLE.

MANUFACTURER RUBBERS ARE VERY SENSITIVE TO DAMAGE BY OXIDE AND ULTRAVIOLET. AMONG THOSE WHICH HAVE GOOD TO EXCELLENT RESISTANCE TO THESE AND OTHER ENVIRONMENTAL USE FACTORS ARE SOME SILICONGLES, NEOPRENE AND ACRICEL. PAPERS 74 THROUGH 82 PRESENT ASSESSMENT DATA ON REPRESENTATIVE TYPES OF RUBBERS INCLUDING PHYSICAL PROPERTIES.

IN ADDITION TO REASONABLE ENVIRONMENTAL RESISTANCE, THE RUBBERS NEEDED FOR THE SPACE STATION MODEL MUST HAVE GOOD RESILIENCE, LOW SHRINKING, AND HIGH STRENGTH TO ASSURE A SOFT, LIGHTWEIGHT SUPPORTING SYSTEM.

THE FOLLOWING SECTIONS PRESENT THE RESULTS OF SEVERAL SERIES OF TESTS ON RUBBER SAMPLES TO GAIN A BETTER UNDERSTANDING OF RUBBER BEHAVIOR IN GENERAL AND TO IDENTIFY THE CHARACTERISTICS AND SPECIFIC RUBBER TYPES NEEDED FOR THE SPACE STATION MODEL TESTS.
6. Elastomers

Since 1932, Du Pont has met the requirements of advancing technology with the development of new and better polymers. In 1951, HYPALON® synthetic rubber was introduced; in 1957, WITON™ fluoroelastomer; in 1970, ADVANCE™ u.s.thane rubber, and in 1983, NORDIENE® hydrocarbon rubber.

FAIRFRIEN® elastomer coated fabrics utilize all of these Du Pont elastomers, but also use other available polymers of their properties offer advantages. Table 5 compares the Du Pont elastomers with each other and with our major polymers that are used today.

7. Fabric Coating Methods

There are three methods of applying the elastomer to the fabric that are commonly used today. They are as follows:

- Spread Coating
- Dip Coating
- Calender Coating.

Spread Coating (fig. 1)

In the operation, base fabric is passed beneath a stationary blade, then through a drying oven where solvent is driven off. One or more passes are used to build the coating to the desired thickness.

Dip Coating (fig. 2)

Fabric is passed through a tank containing the coating solution. After the fabric passes through the solution, excess compound is removed by scraping with bars or rods, then the fabric is passed through a drying oven.

Calendering (fig. 3)

In this form of coating, the rubber compound is formed into a film by the top three rolls and transferred to the fabric as passes between the two lower rolls.

---

Table 5: Comparative Properties of Natural & Synthetic Rubbers

<table>
<thead>
<tr>
<th>Property</th>
<th>Natural Rubber</th>
<th>Hypalon®</th>
<th>VITON®</th>
<th>WITON® Fluoroelastomer</th>
<th>ADVANCE™</th>
<th>NORDIENE®</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tear Strength (Pounds Per Inch)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Abrasion Resistance</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Compression Set</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ductile Strength</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Electrical Insulation</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Permeability to Gases</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Acid Resistance</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Base Resistance</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

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DE POOR QUALITY
temperatures, due to the reversion of the outer parts of the article during the long time necessary to obtain adequate internal vulcanization. Where the properties demanded by the end application permit, reversion can largely be overcome by the use of the efficient vulcanization, EV, systems (46). The technique of injection molding offers both advantage and disadvantage for natural rubber (47). In this process, rubber compound is passed through a heated barrel, typically by a screw. The screw can reciprocate and force heated compound through a nozzle into a mold at high temperature (180-200°C). The flow properties of natural rubber compounds are such that considerable heat is generated by work done during injection, resulting in rapid vulcanization times (30 sec for thin articles, perhaps 4 min for a large item of 1 kg). Such rapid vulcanization reduces the danger of surface reversion in thick articles. By proper manipulation of the processing conditions, quite conventional vulcanization systems can be used, although for items with thick sections it may still be necessary to use the EV or semi-EV systems. Calendering demonstrates one of the most outstanding merits of natural rubber, ie, its building tack, an ability to stick to itself rapidly and firmly. This property is invaluable in the formation of composite items which have to be built up, eg, a conveyor belt made by calendering and plying up; for this reason alone at least a portion of natural rubber is commonly used in such articles. The other requirements of calendering, ie, smoothness and consistent dimensions, are readily obtained by control of compound viscosity and calendering conditions. It is one of the consequences of the ready breakdown of natural rubber that compound viscosity can easily be reduced to a suitable value by milling. Extrusion is performed as usual. When die swell is high and extruded surfaces tend to be rough, as with unfilled compounds, superior processing rubbers or process aids PASO or PA57, all of which are made from prevulcanized latex, can be used (48). In addition to reducing die swell and smoothing extrusion, these materials reduce collapse of extruded sections during further processing. PASO is a masterbatch material made from a mixture of 4 parts prevulcanized and 1 part natural latex (28); PA57 is PASO plus 40 phr of a light-

<table>
<thead>
<tr>
<th>Table 4. Physical Constants of Vulcanized Natural Rubber</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unfilled gum rubber</td>
</tr>
<tr>
<td>---------------------</td>
</tr>
<tr>
<td>IRHD* hardness</td>
</tr>
<tr>
<td>Tensile strength, kg/cm²</td>
</tr>
<tr>
<td>Elongation at break, %</td>
</tr>
<tr>
<td>Young's modulus, kg/cm²</td>
</tr>
<tr>
<td>Shear modulus, kg/cm²</td>
</tr>
<tr>
<td>Bulk modulus, kg/cm²</td>
</tr>
<tr>
<td>Poisson's ratio</td>
</tr>
<tr>
<td>Resilience, %</td>
</tr>
<tr>
<td>Velocity of sound, ft/sec</td>
</tr>
<tr>
<td>Specific gravity</td>
</tr>
<tr>
<td>Specific heat</td>
</tr>
<tr>
<td>Thermal conductivity, relation to water</td>
</tr>
<tr>
<td>Coefficient of cubic expansion, °C</td>
</tr>
<tr>
<td>Electrical resistivity, ohm/cm²</td>
</tr>
<tr>
<td>Dielectric constant</td>
</tr>
<tr>
<td>Power factor</td>
</tr>
</tbody>
</table>

* IRHD, International Rubber Hardness Degrees.
Table 5. Typical Natural Rubber Compounds

<table>
<thead>
<tr>
<th>Type of compound</th>
<th>Conveyor belt cover, earthmover tread, and sidewall</th>
<th>Bridge bearing, other engineering items</th>
<th>White-filled compound</th>
<th>Temperature-resistant compound</th>
</tr>
</thead>
<tbody>
<tr>
<td>natural rubber, sheet or block</td>
<td>100</td>
<td>100</td>
<td>100</td>
<td></td>
</tr>
<tr>
<td>natural rubber, pale crepe</td>
<td>45</td>
<td>-</td>
<td>100</td>
<td></td>
</tr>
<tr>
<td>low-structure HAF*-black</td>
<td>-</td>
<td>-</td>
<td>45</td>
<td></td>
</tr>
<tr>
<td>SRF*-black</td>
<td>-</td>
<td>-</td>
<td>60</td>
<td></td>
</tr>
<tr>
<td>aluminum silicate</td>
<td>-</td>
<td>-</td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>process oil</td>
<td>3</td>
<td>2</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>zinc oxide</td>
<td>5</td>
<td>5</td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>stearic acid</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>antioxidant/antioxidant</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>wax</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>CBS†</td>
<td>0.5</td>
<td>1</td>
<td>0.7</td>
<td></td>
</tr>
<tr>
<td>MOL§</td>
<td>-</td>
<td>-</td>
<td>1.4</td>
<td></td>
</tr>
<tr>
<td>TMTD∥</td>
<td>-</td>
<td>-</td>
<td>0.4</td>
<td></td>
</tr>
<tr>
<td>sulfur</td>
<td>2.5</td>
<td>2</td>
<td>2.5</td>
<td></td>
</tr>
<tr>
<td>cured</td>
<td>30/141°C</td>
<td>29/141°C</td>
<td>30/141°C</td>
<td>30/141°C</td>
</tr>
<tr>
<td>IRHD hardness</td>
<td>62</td>
<td>58</td>
<td>63</td>
<td>65</td>
</tr>
<tr>
<td>tensile strength, kg/cm²</td>
<td>300</td>
<td>264</td>
<td>240</td>
<td>288</td>
</tr>
<tr>
<td>elongation at break, %</td>
<td>500</td>
<td>550</td>
<td>625</td>
<td>563</td>
</tr>
<tr>
<td>tear resistance, kg/min at 21°C</td>
<td>4</td>
<td>3</td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>after aging 21 days at 100°C</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>tensile strength, kg/cm²</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>elongation at break, %</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

* HAF, high-abrasion furnace black.
† SRF, semi-reinforcing furnace black.
‡ CBS, N-cyclohexylbenzothiazole-2-sulfenamide.
§ MOL, 2-(4-morpholino-N-mercapto)-benzothiazole.
∥ TMTD, tetramethylthiuram disulfide.

Some important physical constants of vulcanized natural rubber are listed in Table 4; Table 5 lists recipes and characteristics of some typical natural rubber compounds.

Latex. The compounding of natural rubber latex is, in principle, similar to that of dry natural rubber. There are, however, two major differences. First, because latex compounds are mixed at room temperature, highly active vulcanization systems can be used, which enable vulcanization to be performed at temperatures below 100°C. In dry rubber, such systems are difficult or impossible to use because the heat developed in mixing causes premature vulcanization, or scorch. Secondly, the reinforcing action of fillers, which is of major importance in dry rubber, is not obtainable in normally processed latex compounds. This is because reinforcement is developed only by masticating rubber and filler together. Although various means for reinforcing latex com-
Refractories to Silk
The need for an instrument to make measurements of some cure-dependent property continuously while cure is taking place and at the curing temperature so that a curing curve can be produced with good precision has been satisfied with the cure meters, e.g., the oscillating disk cure meter (ASTM D 2084-79). The Plasti-Corder is another instrument which can be used to measure the rheological properties of rubber and plastics and it is very versatile (96). The Plasti-Corder can be used to measure compound or polymer flow characteristics over a wide range of shear rates and temperatures. The relative power requirements for different stocks can be predicted by the instrument. The extrusion performance of rubber compounds can be predicted from the Brabender curves. Generally, as the trace bandwidth increases, the extrusion quality decreases.

The extrudability of unvulcanized elastomeric compounds (ASTM D 2320-78) can be determined using ASTM Extrusion Die No. 1, Garvey type, which has a triangular shape. Systems for rating extrusions are described, and formulas and their preparation are given for compounds of known extrusion characteristics to allow each laboratory to evaluate its own technique. Since extrusion machines differ from laboratory to laboratory, these methods outline techniques to minimize differences in testing approaches between tubers.

Capillary rheometry is one of the few techniques available that covers the total range of shear rates involved in rubber processing. In one industrial method, the output rate of the extrudate is determined by applying constant pressure on the piston. The corresponding pressure is measured when a constant rate of piston movement is applied to give a fixed extrusion rate. The Monsanto Processability Tester (MPT) is designed with an advanced constant-rate capillary rheometer with die swell and relaxation measuring capability.

Vulcanizates. There are two types of tests for vulcanizates. In the first type, the tests are either of specimens especially molded for the purpose or of specimens cut from a finished product. In the second type the tests are of the product itself, either in actual service or in machines designed to simulate or exaggerate conditions. The following discussion applies only to tests of the first type.

Tension. The stress-strain test in tension is the most widely used test in the rubber industry. It is extremely useful for analyzing compound development, aiding in manufacturing control, and determining a compound's susceptibility to natural and artificial aging. Tensile strength and ultimate elongation values, however, have little significance for design or application engineers, since they cannot be used in design calculations and they bear little relation to the ability of a rubber part to perform its function. Tensile strength and elongation properties serve as an index to the general quality of a rubber part. Rubber compounds less than 6.9 MPa (1000 psi) in tensile strength are usually poor in most mechanical properties and those with tensile strengths over 20.7 MPa (3000 psi) are usually good in most mechanical properties. In the middle range, which is applied to most rubber products, correlation is at best haphazard between tensile strength and such properties as flex life, compression set, abrasion resistance, and resilience. In the standard test the specimen has a dumbbell shape and is cut from a sheet with die C as described in ASTM D 412-80 and is ca 2.0 ± 0.2 mm thick. The test is conducted at room temperature and the jaws which grip the tab ends of the dumbbell specimens are separated at the rate of 8.5 mm/s. By means of suitable devices, the load required to elongate the specimen is recorded for each 100% extension of the restricted portion of the dumbbell, and both the elongation at
break and the tensile strength at break are recorded. The load required to elongate
to a specified elongation, eg, 300%, is referred to as the modulus of the material.
The values of stress at each increment of elongation and at break are calculated
on the basis of the original, unstressed cross section, rather than the actual cross section
at the time the measurements are made. If it is assumed that no change in volume
occurs during stretching, it follows that for a conventional tensile strength value of
20.7 MPa (300 psi) and an ultimate elongation of 900%, the actual stress at break on
the cross-sectional area at the instant of breaking is

\[ 20.7 \times \left( \frac{900}{100} + 1 \right) = 207 \text{ MPa (30,000 psi)} \]

Nonstandard tests are frequently made with specimens larger or smaller than
die C or thicker or thinner than 2.0 ± 0.2 mm. The rate of jaw separation can be varied
as can the testing temperature. All these conditions influence the results obtained.
In the United States much of the work of tensile testing is done on a Scott tensile-
testing machine which has a pendulum dynamometer. These machines are considered
accurate only at 15–85% of their maximum rated capacity. The Instron and Accr-O-
Meter include strain gauges in their weighing system; thus they are useful in studying
the low strain portion of stress–strain curves.

**Hardness.** Hardness (qv) is the relative resistance of the surface to indentation
by an indenter of specified dimensions under a specific load. The objective of a
hardness test is to measure the elastic modulus of the rubber compound under condi-
tions of small strain. This property is one which is closely related to product per-
f ormance, since most rubber products in use are subjected to relatively small strains.
The ASTM hardness testing methods are ASTM D 2240-75, D 1415-68 (1975), and
D 531-78. ASTM D 531 (Pusey and Jones Indentation) is used mainly for roll com-
 pounds.

The assumption that hardness is a close measure of stiffness may be problematic
when testing rubber products such as motor mounts. There is a stress–strain rela-
tionship between hardness and stiffness, but it is established for two entirely different
kinds of deformation. Hardness is derived from small deformations at the surface,
whereas stiffness measurements are derived from gross deformations of the entire mass.
Because of this difference hardness is not a reliable measure of stiffness.

**Set, Creep, and Hysteresis.** No rubber vulcanizate is perfectly elastic, and a great
many tests are employed to measure the extent to which a material fails to be perfectly
elastic. These can be grouped into static or long-time tests and dynamic or short-time
tests. Among the static tests are tests of permanent set in terms of tension or com-
pression, creep, and stress–relaxation. Perhaps the most common of these is the per-
manent-set test in compression (ASTM D 395-78), in which a specimen 12.7 mm thick
and 28.7 mm in diameter is compressed between flat plates and compressed for a
specified time at the desired test temperature, after which the compressing force is
released and the specimen allowed to recover for a specified period of time. The height
of the specimen is then measured and the permanent, unrecovered height is noted.
This type of test is useful in developing materials or predicting the performance of
a product which is utilized in compressive strain.

Permanent set in tension (ASTM D 412-80) is the permanent deformation caused
by tensional forces. The tension-set tests are seldom used in practice except in the wire
industry.

Creep is the increase of deformation with time under constant stress. Creep is
important in motor mounts, since it influences the space relationships between various parts of equipment. It is difficult to predict creep for a given application without conducting simulated service tests because several factors influence creep, especially the amount of strain, temperature, and changes in these two resulting from vibration. The higher the initial strain, the higher the creep; also the higher the temperature, the higher the creep. The degree of creep depends on the type of strain. Creep is greater under tension strain than under equal shear or compression strain. Creep is also greater under dynamic loading than under static loading.

Stress relaxation of a cured rubber is the loss in stress with time at a constant deformation. A method of measuring stress relaxation in compression is described in ASTM D 1390-76. Stress relaxation is an important characteristic of a rubber gasket in its ability to maintain a seal.

The dynamic group of tests includes rebound tests and free-vibration tests either at resonance or at a frequency avoiding resonance. The objective in these tests is generally to determine the hysteresis or energy lost under the particular conditions employed, although the determination of the dynamic stiffness of the material is also important. One of the most widely used of these tests is the free-vibration test with the Yezley oscillograph (ASTM D 945-79), whereby the specimen is vibrated either in compression or in shear and a damping curve is obtained from which the more important properties can be calculated. Another widely used method is the determination of hysteresis by means of the Goodrich Flexometer (method A of ASTM D 623-78). A cylindrical specimen 17.8 mm in diameter and 2.5 mm tall is vibrated at 30 Hz under controlled conditions of load, stroke, and ambient temperature. The temperature rise at the base of the specimen is measured and this is considered a measure of the hysteresis defect of the material under the particular conditions employed. Method B of ASTM D 623-78 describes the Firestone flexometer, which also vibrates the specimen at a constant amplitude. Method C of ASTM D 623 describes the St. Joe flexometer, which vibrates the specimen at either a constant load or a constant amplitude.

Impact resilience is determined in accordance with ASTM D 1054-79, also known as the Goodyear-Healy method. A free-falling pendulum hammer is dropped against a specimen. The resilience is the height to which it rebounds, expressed as the percentage of the height from which it was dropped.

**Cracking and Crack Growth.** The flexing resistance of a rubber compound is its ability to withstand fatigue resulting from repeated distortion by extension, bending, or compression. This flexing fatigue may result in several different types of failure. The most important fatigue failure is popularly called flex cracking. The cause of this failure is twofold: stress breaking of rubber chains and cross-links and, more important, oxidation accelerated by heat buildup in flexing. This type of cracking occurs in tires, shoe soling, and belting. Both flex cracking and ozone cracking can be considered in two parts: initiation of cracks and crack growth.

ASTM D 430-73 methods B and C describe procedures by which the initiation of cracks and their subsequent growth can be measured. With some materials the initiation of cracks, as measured in method B, is slow and erratic, but once initiated they grow quite rapidly. To measure the growth of initiated cracks, a specimen as used in the initiation test is cut or pierced with a sharp tool at the base of the groove, and the rate of growth of this cut is measured as a function of the number of flexures (ASTM D 813-59) (1976). Method C, involving a specially molded, grooved specimen is used outdoors and in an ozone box to determine crack initiation and growth.
3.10 Calculation of Amount of Rubber Cord for Model Support

Data extracted from Reference 5

For carbon filled vulcanized natural rubber.

Tensile Strength $\sigma = 280 \text{ Kg/㎠} = 1567 \text{ lb/in}^2$

Elongation at Break $\% = 600 \text{ percent}$

Specific Gravity $\gamma = 1$

Force at 200\% Elongation $= 120 \text{ Kg/㎠} = 672 \text{ lb/in}^2$

Assumptions:

1. Work at 200\% Elongation
2. Model Weight is 10,000 lb

Original Area: $A_{or}$

$$A_{or} = \left( \frac{\text{Area}}{\text{Weight}} \right) \times \left( \frac{\text{Weight}}{W} \right) = \frac{W}{G} = \frac{10,000}{120 \times 6.205 \times 0.394} = 14.89 \text{ in}^2$$

$$E = \frac{G}{\gamma} = \frac{9}{4} \left( \frac{120 \times 2.205}{0.394} \right) = 2241 \text{ lb/in}^2$$

$A_{or}$ is the Area of the Unstretched Rubber. This much area will hold the weight and allow the rubber to elongate 300\% percent. The elongation is equal to $5\%$ which equals three times the original length of the rubber, $L_0$.

*Based on original area $A_{or}$
3.11 APPROXIMATION OF THE WEIGHT OF RUBBER CORD FOR MODEL SUPPORT

VOLUME OF RUBBER

\[ V = \text{original area} \times \text{length of rubber under no. load} \]

\[ = A \times 24.0 \quad \text{(see page 24)} \]

\[ = 14.89 \times 0.225 \times (120) \times 12 = 4524 \text{ in}^3 \]

THE SPECIFIC GRAVITY \( G \) IS THE EFFECTIVE WEIGHT \( \leq \) 1/2 ACTUAL WEIGHT, THEREFORE THE EFFECTIVE WEIGHT \( W_{es} \)

\[ W_{es} \leq \frac{1}{2} \times 4524 \times 62.4 \leq 57.1 \text{ lb} \]

Since the chosen method for supporting the model results in the rubber's supports' weight always being proportional to the model's weight for the configuration being tested, we have

\[ \frac{W_{es}}{W_{m}} \leq 81.1 = 0.00871 \]

\[ W_{m} = 10,000 \]

\[ \text{I.e., the effective mass of the support elements is expected to always be less than 1\% of the model mass.} \]

It is recommended that samples of the rubber cord be tested to insure achievement of required mechanical properties. Rubber properties are highly variable, particularly when confined in the ranges of hardness or defined needed.
3.12 APPROXIMATION OF LATERAL NATURAL FREQUENCIES OF MODEL SUPPORT CABLES

IT IS RECOMMEND THAT THE MODEL BE SUPPORTED BY A NUMBER OF CABLES SO DISTRIBUTED THAT THE KEEL, KEEL EXTENSIONS, AND TRANSVERSE BOOM CARRY VERY LITTLE LOAD. EACH OF THESE CABLES WILL CARRY A TENSILE LOAD AND THEREFORE SIMILARLY TO A STRING UNDER TENSION, ITS LOWEST NATURAL FREQUENCY WILL BE

\[ \omega_1 = \frac{\pi}{T} \sqrt{\frac{T}{\mu_s L^2}} \]

WHERE

\[ T = \text{TENSILE FORCE / CABLE} = \frac{Mg}{N} \]

\[ \mu_s = \text{MASS OF CABLE / UNIT LENGTH} \approx \frac{Mr}{0.92N} = \frac{Mr}{L_N} \]

\[ L = \text{CABLE LENGTH} \]

THEN,

\[ \frac{\omega_1}{\omega_{0.9}} = \frac{\pi}{4} \sqrt{\frac{Mr}{L_N^2}} \left( \frac{9}{2} \right) = \pi \sqrt{\frac{M}{Mr}} \left( \frac{9}{2} \right) = \pi \sqrt{\frac{W}{W_r} \left( \frac{3}{2} \right)} \]

\[ = \pi \sqrt{\frac{10,000}{2 \times 87.1} \frac{9}{2}} = 33.8 \approx 0.518 \]

AND \[ \omega_1 = 11.70 \text{ RADIANS} \]

\[ f_1 = \frac{\omega_1}{2\pi} = 186 \text{ HZ} \]

NOTE THAT THIS (THE LOWEST LATERAL STRING) FREQUENCY IS 32.6 TIMES AS HIGH AS THE SIMPLE PENDULUM FREQUENCY AND FOR A GIVEN CABLE STRESS, IS INDEPENDENT OF THE NUMBER OF CABLES.
THE FREQUENCY SEPARATION FOR THE STRING FREQUENCY RELATIVE TO THE FIRST ELASTIC MODE IS:

\[ \alpha_{0} = \frac{\omega_{M, E}}{\omega_{1}} = \frac{f_{E, x_{1}}}{f_{1}} \]

\[ = \frac{0.1 \times 4}{1.66} = 0.22 \]

Thus the first string frequency is about 5 times as high as the first elastic frequency, but coincidence of frequencies is likely to occur for higher elastic modes of the model. The use of random height lateral ties between the cables (perhaps made of highly damped rubber) may eliminate lateral excitations when conditions of coincidence occur between string frequencies & model excitation frequencies.

It should be noted that the model mass distribution, the actual length, between the platform and model attachment points, and the inherent variations in the properties of extruded soft (30 to 40 durometer) rubber will be such that the natural frequencies of the various cables will be slightly different from one another, it is expected that this will eliminate gross lateral excitations of the strings even if the model is driven at a frequency equal to the mean value of the string frequencies.
3.13 SUMMARY OF EXPERIMENTAL DATA 
OBTAINED FROM STATIC AND DYNAMIC TESTS 
OF A RUBBER SAMPLE 

THE LOAD DEFORMATION CURVE OBTAINED FROM THE 
TESTS OF THE RUBBER SAMPLE ARE GIVEN IN FIGURE 9. 
OTHER PERTINENT DATA ARE SUMMARIZED AS FOLLOWS. 

STRESS AT 300% ELONGATION BASED ON ORIGINAL 
ARENA IS 

\[ \sigma_{300} = \frac{15.8}{16} \times \frac{1}{5.6 \times 10^{-3}} = 176 \text{ psi} \] 

THE SPRING CONSTANT AT 300% ELONGATION IS 

\[ K = \frac{A}{L} = \frac{0.25}{3.1} = 8.06 \times 10^{-3} \text{ lbs/ft} \] 

= 0.97 lbs/ft 

THE NATURAL FREQUENCY PREDICTED IS 

\[ f = \frac{1}{2 \pi} \sqrt{\frac{K}{M}} = \frac{1}{6.28} \sqrt{0.97 \times 0.25} = 0.90 \text{ Hz} \] 

THE MEASURED NATURAL FREQUENCY WAS APPROX. 

\[ f_{\text{MEASURED}} = 1.1 \text{ Hz} \] 

SOME DATA FROM TESTS OF SEVERAL ADDITIONAL 
SAMPLES OF VARIOUS TYPES OF RUBBER ARE GIVEN IN 
APPENDIX I. AS A RESULT OF THESE TESTS, IT BECAME 
CLEAR THAT IT WOULD BE EASIEST TO MANUFACTURE NATURAL RUBBER WAS 
REQUIRED AND SAMPLES WERE LOCATED AND ARE NOW 
UNTESTED BY NASA-LaNGLEY.
Figure 9: Results for tests on a rubber sample - thin apex

\[ \Delta = 5.6 \times 10^{-3} \text{ m}^2 \]

\[ d = 3.2 \text{ m} \]

FIBER AMOUNT: 100%

CALCULATION:

\[ A_d = \text{Area under the curve} \]

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3.14 Considerations Relative to the Number of Elastic Cables Employed for Model Support

On the basis of the allowable working stress (300 percent elongation of rubber) it was shown on page 83 that an initial area of 14.89 square inches of rubber is needed. For convenience of attachment to the inelastic cord which will connect to the model and to the platform, tubing is recommended. Rubber tubing is generally available in a variety of cross sectional sizes and lengths. The number of tubes for various outside and inside diameters is given as follows

$$N = \frac{4 \text{ AREA}}{\pi (d^2 - d'^2)} = \frac{4 \times 14.89}{\pi (0.75 - 0.125)(0.75 - 0.125)} = 18.97$$

<table>
<thead>
<tr>
<th>D in</th>
<th>d in</th>
<th>N</th>
<th>Load/Cord, lb</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/4</td>
<td>3/16</td>
<td>405</td>
<td>25</td>
</tr>
<tr>
<td>3/8</td>
<td>1/2</td>
<td>180</td>
<td>55</td>
</tr>
<tr>
<td>7/16</td>
<td>1/4</td>
<td>147</td>
<td>62</td>
</tr>
<tr>
<td>1/2</td>
<td>5/16</td>
<td>101</td>
<td>99</td>
</tr>
<tr>
<td>5/8</td>
<td>3/16</td>
<td>65</td>
<td>154</td>
</tr>
</tbody>
</table>

Compromises will be necessary between the added complexity of many cables and the desire to avoid transmission of large gusts. Induced loads through the structure. Also, for reasons of safety of the model and test personnel, it is desirable to limit the load carried per cable. Thus, it seems reasonable that 1/2 in x 1/4 in cables, each carrying about 100 lbs.,
WOULD PROVIDE A GOOD PRIMARY SYSTEM, PARTICULARLY FOR THE HEAVIER PARTS OF THE MODEL SUCH AS THE ORBITER AND BICHTICIES. PARTS OF THE HEEL STRUCTURE WILL BE LIGHTLY LOADED AND IT MIGHT BE DESIRABLE TO USE SMALLER CABLES IN THESE AREAS TO GET BETTER LOAD DISTRIBUTIONS; PERHAPS 3/8" X 3/16" CABLES.

AS A POINT OF REFERENCE, AN ALL-UP ORBITER WEIGHT OF 240,000 IBS WOULD RESULT IN A MODEL ORBITER WEIGHT OF 3750 IBS. SINCE THE ORBITER MODEL LENGTH WOULD BE ABOUT 30 FEET, IT WOULD BE SUPPORTED BY 1/2 INCH ELASTIC CABLES PLACED AT INTERVALS OF ABOUT 1 FOOT ALONG ITS LENGTH.

ALSO, IT IS NOTED THAT THE COMBINED WEIGHT OF THE 3 MODULES AND 2 LABS LOCATED PRIMARILY BELOW THE NOSE OF THE ORBITER WILL HAVE MODEL WEIGHTS TOTALING ANOTHER 3140 IBS. THESE WEIGHS MUST BE CARRIED BY ANOTHER 31 CABLES, SOME OF WHICH WILL BE LOCATED CLOSE TO THE ORBITER SUPPORTS. IT IS ALSO NOTED THAT THE ORBITER SUPPORT SYSTEM MUST BE COMPATIBLE WITH THE LIMITED CAPABILITY OF THE ORBITER/HABITATION MODULE DENTRY INTERFACE TO TRANSMIT MOMENTS RESULTING FROM GRAVITY-INDUCED FORCES.
3.15 INVESTIGATION OF USE OF COIL AND REVERSE WOP SPRINGS FOR MODEL SUPPORT

THE NATURAL FREQUENCY OF A MASS SUSPENDED BY A LINEAR COIL SPRING IS GIVEN BY

\[ \omega^2 = \frac{g}{2} \]

THE STATIC DEFORMATION OF A COIL SPRING IS

\[ \delta_{ST} = \frac{\pi^2 S_0 d^4}{4 E} \]

WHERE

\[ e = \text{NO. OF COILS} \]
\[ S_0 = \text{ALLOWABLE SHEAR STRESS} \]
\[ d = \text{COIL DIAMETER} = 2R \]
\[ d = \text{WIRE DIAMETER} \]
\[ G = \text{SHEAR MODULUS} \]
\[ \alpha = \text{AN EMPIRICAL DESIGN FACTOR (SEE p. 177, REF. 6)} \]

CHOOSE THE EQUALITY FOR STATIC DEFORMATION, AND THE FOLLOWING CONDITIONS

\[ \omega^2 = \frac{e g}{2} = \frac{g}{\delta_{ST}} \]
\[ S_0 = \frac{150,000}{1.5} = \text{YIELD STRESS / SAFETY FACTOR} = 100,000 \text{ psi} \]
\[ e = \frac{\delta_{ST}}{R} = \frac{\alpha}{3R} = \frac{\text{STRETCHED SPRING LENGTH}}{\text{COIL RADIUS}} \]
\[ \alpha = 1.1 \]

WE OBTAIN

(ORIGINAL PAGE IS OF POOR QUALITY)
\[
\frac{3_t}{3_t} = \pi \frac{23t}{R} \frac{S_t}{D_g^2} \frac{1}{k_g^6}
\]

From which

\[
\frac{d}{R} = \frac{4\pi^2 S_t}{k_g^2} = \frac{4\pi^2}{1.1 \times 10^6}
\]

\[
= 0.101
\]

\[
\frac{d}{D} = \frac{1}{C} = \frac{1}{2} \frac{d}{R} = 0.0505
\]

\[
\therefore C = 20 \quad \text{F.E.Q. 11 VERIFIED}
\]

See Ref. 6, Page 217

The force this spring will carry is

\[
F = \frac{S_t}{3} \frac{d^4}{64} = \frac{S_t}{3} \frac{6}{d} \frac{d^4}{8R} = \frac{6}{64} \frac{d^2}{d} \frac{2}{R} = \frac{6}{64} (1.101)^2 d^2
\]

Let \( F = 100 \text{ lb} \)

\[
d^2 = \frac{100 \times 64 \times 11.3 \times 10^6 (101)^2}{11.3 \times 10^6 (101)^2} = 5.55 \times 10^{-2}
\]

\[
\delta = 0.236 \quad D = 2R = 2 \frac{(R)^2}{d^2} = 2 \frac{1}{2} \times 0.236 = 4.67''
\]

Let \( \delta_t = \frac{R}{8} = \frac{120 \times 12}{8} = 480 \text{ in} \)

\[
\therefore \frac{R}{d} = \frac{480 \delta}{d} \frac{1}{1} = 480 \times 0.101 \times \frac{1}{0.236} = 205
\]

Spring weight

\[
W = \delta (W_d) = \frac{\pi d^2}{4} (0.25) = 0.07 \pi (7.1)^2 d^3
\]

\[
= 35.8 \text{ lb per 100 lb of model weight}
\]
\[ S_c = \frac{E A_d E}{12 I} = \frac{11 \times 3.1416 \times 10^6 \times 0.101}{11 \times 4 \times 4} \]
\[ = 0.1 \times 10^6 = 100,000 \text{ lb} \]

**Spring Constant**

\[ K = \frac{F}{x} = \frac{100}{40 \times 12} \]
\[ = \frac{146}{810^3} = \frac{6.1 \times 10^3}{8} = \frac{1131 \times 10^6}{8} = 0.207 \]

\[ \omega^2 = \frac{K}{M} = \frac{0.207 \times 386}{100} = 0.806 \text{ rad/sec}^2 \]
\[ = \frac{3 \times 386}{120 \times 12} = 0.804 \text{ rad/sec}^2 \]

**Note that the aforementioned calculations neglected the weight of the spring in the calculations of the frequency. But the approximation serves the purpose in pointing out that even for the stiffest spring (highest allowable frequency) the spring mass is of the order of \( \frac{1}{2} \) third of the total mass. For a softer system (one with a larger static deflection), the number of coils and hence the weight of the spring is even higher.**

**The conclusion is that coil springs are not a viable support option.**
ANALYSIS OF A HINGE-LEAF SPRING

EACH LENGTH L IS TREATED AS A BEAM WITH ZERO SLOPE AT EACH END.

\[ S = n \alpha \quad \text{where} \quad n \quad \text{is the number of half loops} \]

\[ \frac{dy}{dx} = \frac{P(x^2 - \frac{x^3}{6})}{E I} + c \]

Since \( \frac{dy}{dx} = 0 \) at \( x = 0 \), \( k = 2 \)

\[ c = 0 \quad \text{and} \quad M_x = P \frac{x^2}{2} \]

\[ y = \frac{P}{E I} \left( \frac{x^2}{2} - \frac{x^3}{6} - \frac{6}{x^2} \right) + c \]

Since \( y = 0 \) at \( x = 0 \)

\[ c = 0 \]

AND

\[ \varepsilon(x) = \frac{M_x}{E I} = \frac{P x^2}{12 E I} \]
THE STRESS $\sigma$ IS

$$\sigma = \frac{Mc}{I} = \left( P \left( \frac{L}{2} - x \right) - P \frac{L}{2} \right) \frac{1}{3} \frac{1}{L} = P \frac{\left( L - x \right)}{2} \frac{1}{L}, \text{ or}$$

$$\sigma_{\text{max}} = \frac{P}{L} \frac{L}{2}$$

AND

$$\frac{P}{L} = 4 \frac{\sigma_{\text{max}}}{E}$$

THE DEFLECTION CAN NOW BE EXPRESSED IN TERMS OF
THE MAXIMUM STRESS, i.e.

$$\delta_i = \frac{1}{12} \left( \frac{P}{L} \right) \frac{L}{6} = \frac{1}{12} \left( \frac{4 \sigma_{\text{max}}}{E} \right) \frac{L}{6} = \frac{1}{3} \sigma_{\text{max}} \frac{L}{E}$$

IF WE ASSUME THAT THE LEAF SPRING HAS A RECTANGULAR
CROSS-SECTION, THE WEIGHT OF A LEAF IS

$$w_i = \rho b l$$

AND THE DEFLECTION/WEIGHT FOR A LEAF IS

$$\frac{\delta_i}{w_i} = \frac{1}{3} \frac{\sigma_{\text{max}}}{E} \left( \frac{L}{b} \right) \frac{l}{L}$$

NOTE THAT THE DEFLECTION/WEIGHT IS MINIMIZED BY
LARGE $L$ AND SMALL $b$ AND $t$, GEOMETRICAL FACTORS. IT IS
ALSO MINIMIZED BY HAVING A HIGH VALUE OF THE MATERIALS
PROPERTY RATIO $(\frac{\text{Young's Modulus}}{\rho})$. FOR MANY REASONS, A HIGH
QUALITY SPRING STEEL APPEARS DESIRABLE BUT HIGH STRENGTH
ADVANCED COMPOSITES MIGHT ALSO PROVIDE AN OPTION.

THE WEIGHT OF THE SUPPORT SYSTEM IS THEN

GIVEN BY
\[ W = n_{s} \varphi_{t} = n \pi b \ell \]

WHERE

\[ n = \frac{33T}{5} = \frac{675 \times 120 \times 12}{5} \]

THEN

\[ W = \left( \frac{675 \times 120 \times 12}{5} \right) \frac{3TE}{\varphi_{t} \ell^{2}} \pi b \ell \]

\[ = 2.92 \times 10^{3} \frac{1}{\ell^{2}} \frac{PE}{\varphi_{t}} \]

FOR SPRING STEEL

\[ \frac{PE}{\varphi_{t}} = 0.28 \times 30 \times 10^{6} = 50.3 \]

AND, FOR \( \frac{\ell}{D} = 0.1 \), WE OBTAIN THE FOLLOWING TABLE

<table>
<thead>
<tr>
<th>t, \frac{\ell}{D}</th>
<th>W, \frac{PE}{\varphi_{t}}</th>
<th>P</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/16</td>
<td>9.19</td>
<td>3.1</td>
</tr>
<tr>
<td>0.025</td>
<td>36.3</td>
<td>13.7</td>
</tr>
<tr>
<td>0.050</td>
<td>82.6</td>
<td>31.4</td>
</tr>
<tr>
<td>0.100</td>
<td>147</td>
<td>64.3</td>
</tr>
</tbody>
</table>

THE OBVIOUS CONCLUSION IS THAT THE LOOP TYPE SPRING IS ALSO MUCH TOO HEAVY.
APPENDIX I

RESULTS OF EXPERIMENTAL TESTS OF ADDITIONAL RUBBER SAMPLES

TESTS CONDUCTED IN NASA-LANGLEY DYNAMICS RESEARCH LABORATORY ON 6/12 AND 6/13, 1985

BECAUSE OF THE UNCERTAINTY OF THE CHARACTERISTICS OF RUBBER WHICH MAY BE USED TO SUPPORT THE MODEL, SEVERAL SAMPLES OF RUBBER WERE OBTAINED AND TESTED IN THE DYNAMICS RESEARCH LABORATORY AT NASA-LANGLEY. DETAILS OF THESE SAMPLES (C), TOGETHER WITH THE DATA FROM THE TESTS, ARE GIVEN ON PAGES 98 THROUGH 110.

THE STRESS-STRAIN RELATIONSHIPS FOR THE RUBBER SAMPLES ARE PLOTTED ON FIGURE I-1 WHERE THE FOLLOWING DEFINITIONS WERE USED:

\[ \sigma = \frac{F}{A_0} = \text{STRESS} \]

\[ \varepsilon = \frac{\text{DEFLECTION}}{L_0} = \text{STRAIN} \]

THE MEASURED NATURAL FREQUENCIES, NORMALIZED BY DIVIDING BY THE NATURAL FREQUENCIES OF LINEAR SYSTEMS UNDER THE SAME STATIC DEFLECTIONS, ARE PLOTTED IN FIGURE I-3.
**Sample No. 1 - Latex Surgical Tubing**  
Source: Southampton Pharmacy  
Supplier: Mont Latex Products, Inc.

**Nominal Dimensions:** Inside Diameter: 0.25"  
Outside Diameter: 0.375"  
Original Length: 6.5"  
Final Length: 6.35"  
Original Area: 0.0613 in²

<table>
<thead>
<tr>
<th>T</th>
<th>F</th>
<th>T²</th>
<th>F/T</th>
<th>E</th>
<th>E'</th>
<th>o</th>
<th>o² Do</th>
<th>g</th>
</tr>
</thead>
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<td>201</td>
<td>6.91</td>
<td>0.62</td>
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<td></td>
</tr>
<tr>
<td>6</td>
<td>417.11</td>
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<td>121</td>
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<td>0.71</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>179.1</td>
<td>146.5</td>
<td>2.53</td>
<td>75.2</td>
<td>1.73</td>
<td>1.14</td>
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<td></td>
</tr>
<tr>
<td>16</td>
<td>398.0</td>
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<td>2.71</td>
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<td>273.2</td>
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<td>9.01</td>
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<tr>
<td>19</td>
<td>309.8</td>
<td>281.5</td>
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<td>67.9</td>
<td>3.77</td>
<td>10.9</td>
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<td></td>
</tr>
<tr>
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<td>342.0</td>
<td>284.5</td>
<td>4.83</td>
<td>71.0</td>
<td>3.93</td>
<td>11.3</td>
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<td>286.5</td>
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<td>71.6</td>
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</tr>
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<td>23</td>
<td>372.5</td>
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<td>6.99</td>
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<td>16.0</td>
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</tbody>
</table>

**Notes:** Original page 10 of poor quality.
SAMPLE NO. 2 - NATEX TUBING
SOURCE: HAMPTON RUBBER CO
SUPPLIER: KENT NATEX PRODUCTS, INC.

NOMINAL DIMENSIONS:
- INSIDE DIAMETER: 0.375"
- OUTSIDE DIAMETER: 0.625"
- ORIGINAL LENGTH: 23.25"
- FINAL LENGTH: 24.50"
- ORIGINAL AREA: 0.190 in²

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<th>$\frac{E}{\Delta e}$</th>
<th>$E'$</th>
<th>$\frac{w}{\frac{w_{2.5}}{g}}$</th>
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COMpletely UNCONDEd & RELOADED:

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<th>$E'$</th>
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<td>475.8</td>
<td>123.3</td>
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</table>
SAMPLE NO. 3 - LATEX TUBING
SOURCE: HAMPTON RUBBER
SUPPLIER: KENT LATEX PRODUCTS, INC.
AVAILABLE IN 50' LENGTHS

NOMINAL DIMENSIONS:

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<th>Outside Diameter</th>
<th>Original Length</th>
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<td>0.3125&quot;</td>
<td>26.50&quot;</td>
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<th>σ'</th>
<th>E</th>
<th>E'</th>
<th>ω</th>
<th>ω'</th>
<th>α</th>
<th>α'</th>
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<td>in</td>
<td>in/ln</td>
<td>16112</td>
<td>in</td>
<td>in/ln</td>
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<table>
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<th>σ'</th>
<th>E</th>
<th>E'</th>
<th>ω</th>
<th>ω'</th>
<th>α</th>
<th>α'</th>
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<tr>
<td>5.0</td>
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<td>1.63</td>
<td>92.8</td>
<td>3.14</td>
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<tr>
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<td>10.98</td>
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<tr>
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<td>366.6</td>
<td>125.6</td>
<td>4.72</td>
<td>77.7</td>
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<td>11.12</td>
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<tr>
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<td>128.0</td>
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<tr>
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<td>97.2</td>
<td>5.86</td>
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<tr>
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<td>5.02</td>
<td>96.2</td>
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<td>102.0</td>
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<tr>
<td>28.0</td>
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<td>138.8</td>
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<tr>
<td>33.0</td>
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<td>130.3</td>
<td>KNOT ROLLED OUT</td>
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</tr>
</tbody>
</table>
**Sample No. 4 - Red Gum**

**Source:** Hampton Rubber Co  
**Supplier:** Unknown  
**Available: Long Length Rolls**

**Nominal Dimensions:**  
- Inside Diameter: 0.250"  
- Outside Diameter: 0.375"  
- Original Length: 16.5"  
- Final Length: 16.5"

<table>
<thead>
<tr>
<th>F</th>
<th>d₁</th>
<th>d₂</th>
<th>ε₁</th>
<th>ε₂</th>
<th>E</th>
<th>E'</th>
<th>ω</th>
<th>( \omega^2 \cdot 10^3 )</th>
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</thead>
<tbody>
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<td>16</td>
<td>16</td>
<td>1</td>
<td>1</td>
<td>.028</td>
<td>532</td>
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<td>16</td>
<td>.99</td>
<td>.127</td>
<td>770</td>
<td>5.97</td>
<td>4.47</td>
<td></td>
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<tr>
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<td>16</td>
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<td>24</td>
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<td>1.83</td>
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<td>24</td>
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<td>2.28</td>
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<td>2.41</td>
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<td>9</td>
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<td>8.06</td>
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<tr>
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<td>32</td>
<td>32</td>
<td>9.75</td>
<td>2.75</td>
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<td>5.65</td>
<td>8.06</td>
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<td>11</td>
<td>32</td>
<td>32</td>
<td>9.75</td>
<td>2.75</td>
<td>219</td>
<td>5.65</td>
<td>8.06</td>
<td></td>
</tr>
</tbody>
</table>

**HELD LONG MOMENTARILY & FAILED**
**SAMPLE NO. 5**
**SILICONE RUBBER**
**SOURCE:** HAMPTON RUBBER CO
**SUPPLIER:** UNKNOWN
**AVAILABLE:** LONG-LENGTH ROLLS

**Nominal Dimensions:**
- Inside Diameter: 0.25"
- Outside Diameter: 0.375"
- Original Length, l: 29.0"
- Final Length: 
- Original Area, A₀: 0.0613 in²

<table>
<thead>
<tr>
<th>F</th>
<th>G</th>
<th>E₀</th>
<th>E₁</th>
<th>ω</th>
<th>ω²E₁</th>
<th>E₀/l₀</th>
<th>θ₀E₀/l₀</th>
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</thead>
<tbody>
<tr>
<td>16/10²</td>
<td>11</td>
<td>m/m</td>
<td>12/10²</td>
<td>rad/sec</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

| 1   | 16.3  | .75   | .026  | 62.7  |
| 6   | 97.8  | 9.25  | .318  | 308   | 6.28     | .945  |
| 11  | 17.9  | 17.75 | .595  | 301   | 8.79     | 1.45  |
| 14  | 24.6  | 22.0  | .759  | 394   | 10.05    | 5.76  |
| 21  | 34.3  | 27.0  | .931  | 363   | 10.05    | 7.06  |
| 26  | 42.3  | 31.0  | 1.172 | 361   | 10.05    | 8.90  |
| 31  | 50.5  |       |       |       |          | FAILED SHORTLY AFTER APPLICATON |

**ORIGINAL PAGE IS OF POOR QUALITY**
<table>
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<tr>
<th>Sample NO. 6</th>
<th>BLACK NEOFRIENE?</th>
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</thead>
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<td>SOURCE: NASH STOCK NO. 4720-00993-0392</td>
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</tbody>
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<table>
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<th>NOMINAL DIMENSIONS</th>
<th>INSIDE DIAMETER</th>
<th>0.3125</th>
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<tr>
<td>OUTSIDE DIAMETER</td>
<td>0.30&quot;</td>
<td></td>
</tr>
<tr>
<td>ORIGINAL LENGTH</td>
<td>10'-1&quot;</td>
<td></td>
</tr>
<tr>
<td>FINAL LENGTH</td>
<td></td>
<td></td>
</tr>
<tr>
<td>ORIGINAL AREA, $A_0$</td>
<td>0.120</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$F$</th>
<th>$\sigma$</th>
<th>$\frac{\sigma}{E'}$</th>
<th>$e$</th>
<th>$W$</th>
<th>$\frac{W^2}{E'}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>16</td>
<td>16112</td>
<td>in.</td>
<td>10/10</td>
<td>16/10</td>
<td>100/500</td>
</tr>
<tr>
<td>33.25</td>
<td>27.08</td>
<td>1.5</td>
<td>.0123</td>
<td>2201</td>
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<td>.2152</td>
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<td>49.8</td>
<td>.412</td>
<td>674</td>
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<td>63.25</td>
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<td>898</td>
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<tr>
<td>83.25</td>
<td>693.7</td>
<td>Healed for 1 minute &amp; Failed</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>


ORIGINAL PAGE IS OF POOR QUALITY

300% ELONGATION

DENOTES FAILURE
DENOTES KNOT PULL OUT

FIGURE I-1: STRESS-STRAIN VARIATIONS MEASURED FOR 6 RUBBER SAMPLES
FIGURE I-2 - VARIATION OF MEASURED FREQ. WITH STRAIN
**Figure I-3.** Variation of normalized frequency of latex samples with strain.

Latex rubber samples

Sample No.

\[ \frac{\omega^2}{g} \]

At 300% elongation

\[ \omega^2 \propto \frac{g}{E_g} \]
APPROXIMATION OF NATURAL FREQUENCY ON BASIS OF STRESS-STRAIN DATA FOR CLOTH: KIDDEY CHAMPS

FIGURE I-1 SHOWS THE STRESS-STRAIN DATA MEASURED FOR 6 SAMPLES OF RUBBER. THE 3 CLOTH SHAPES, SHOWN BY THE OPEN SYMBOLS, DIFFERED PRIMARILY IN SAMPLE SIZE (LENGTH X CROSS-SECTION AREA) AND THE DATA ARE SHOWN TO BE FAIRLY CONSISTENT.


ON THE BASIS OF SLOPE DATA, IT IS EXPECTED THAT

\[
\omega^2 = \frac{K}{M} = \frac{AF}{M} = \frac{\Delta \sigma \Delta \varepsilon}{\sigma' \Delta \varepsilon} \frac{\sigma'}{\epsilon} = \frac{\Delta \sigma}{\Delta \varepsilon} \frac{9}{\sigma' \Delta \varepsilon}
\]

(Note: If a system is linear, \( \frac{\Delta \sigma}{\Delta \varepsilon} = \sigma' = \frac{\sigma}{3} \) and \( \omega^2 = \frac{9}{\sigma' \Delta \varepsilon} \) as expected)

IF THE NATURAL FREQUENCY IS NORMALIZED BY THE NATURAL FREQUENCY MEASURED FOR A GIVEN SAMPLE AT A GIVEN LOAD LEVEL, THEN

\[
\left( \frac{\omega}{\omega_n} \right)^2 = \frac{\Delta \sigma}{\Delta \varepsilon} \frac{9}{\sigma' \Delta \varepsilon}
\]
AND, IN TABULAR FORM, THE RESULTS ARE

<table>
<thead>
<tr>
<th>ORIGINAL PAGE IS</th>
<th>SAMPLE 1</th>
<th>SAMPLE 2</th>
<th>SAMPLE 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>OF POOR QUALITY</td>
<td>( I_0 = 61.5'' )</td>
<td>( I_0 = 23.25'' )</td>
<td>( I_0 = 26.5'' )</td>
</tr>
<tr>
<td>( \epsilon )</td>
<td>( \frac{45}{\Delta \epsilon} )</td>
<td>( \frac{(\omega_2)}{\omega} )</td>
<td>( \frac{(\omega_2)}{\omega} )</td>
</tr>
<tr>
<td>12/11/2</td>
<td>16/11/2</td>
<td>(rad/sec)^2</td>
<td>(rad/sec)^2</td>
</tr>
<tr>
<td>0.50 0.65 90</td>
<td>8.67 0.80</td>
<td>23.0 1.00</td>
<td>20.2 0.88</td>
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<tr>
<td>3.00 18.15 34.4</td>
<td>1.15 0.31</td>
<td>3.05 0.26</td>
<td>2.67 0.23</td>
</tr>
<tr>
<td>5.00 48.5 48.8</td>
<td>6.32 0.40</td>
<td>14.7 0.41</td>
<td>14.7 0.41</td>
</tr>
</tbody>
</table>

WHERE THE EXPERIMENTAL FREQUENCY DATA ARE GIVEN BY FIGURE 1-2.

IN SUMMARY, THE RESULTS SHOW:

1. AT LOW STRAIN LEVELS (\( \epsilon = 0.50 \)), THE FREQUENCY OBTAINED FROM SLOPE DATA VARIES BETWEEN 90° AND 100º PERCENT OF THE MEASURED FREQUENCIES. AT INTERMEDIATE STRAIN LEVELS (\( \epsilon = 3.0 \)), THE MEASURED FREQUENCIES ARE ABOUT TWICE AS HIGH AS PREDICTED FROM SLOPE DATA AND AT HIGH STRAIN LEVELS (\( \epsilon = 5.0 \)), THE MEASURED FREQUENCIES ARE ABOUT ONE AND A HALF TIMES AS LARGE AS PREDICTED FROM SLOPE DATA.

2. MORE DETAILED INFORMATION IS NECESSARY, PARTICULARLY AT THE INTERMEDIATE STRAIN LEVELS AND FOR THE PARTICULAR RUBBER TYPES TO BE USED, TO ACCURATELY PREDICT THE NATURAL FREQUENCIES OF SUPPORTED SYSTEMS. NECESSARY DATA MUST INCLUDE EFFECTS OF CREEP.
(b) As shown on Figure 1-3, the natural frequency at
E = 3.25 closely approaches the value for a simple system.
Thus the natural frequency at E = 2.25 will be about
1/2 lower than at E = 3. However, the working stresses is
less (approx. 162.5 MPa vs. 181.5 MPa) and the rubber
is thinner. Consequently, a given weight of rubber
will be longer. The natural frequencies will change. For these samples,
the relative weights are segregated as follows.

\[
\frac{w_{1.25}}{w_{2.25}} = \frac{L_1 - L_2}{L_2} = \frac{\sqrt{162.5}}{\sqrt{187.5}} = \frac{1}{1.125} \approx 0.91
\]

OR

\[
\frac{w_{1.25}}{w_{2.25}} = \frac{L_1 - L_2}{L_2} = \frac{\sqrt{162.5}}{\sqrt{187.5}} = \frac{1}{1.125} \approx 0.91
\]

AND

\[
E_0 = \frac{E (1 - \nu)}{(1 + \nu) \mu} = \frac{9000}{3.25} = 2712.5
\]

\[
\Delta T = \beta C_0 = 2.25 \times 2.77 \times 2.4 = 16.23 \times 2.77
\]
DETERMINATION OF WEIGHT OF RUBBER FROM LATEX TEST RESULTS

STRESS AT 300% ELONGATION = 187.5 lb/in²

AREA = \( \frac{FORCE}{STRESS} = \frac{14,000}{187.5} = 75.33 \)

LENGTH = \( L_0 = 0.225L = 0.225(120)/2 = 32.4 \) in

VOLUME = \( L_0 \times A = 53.33 \times 32.4 = 1,721.8 \) in³

WEIGHT = \( V \times 62.4 \times g = 17,218 \times \frac{1}{1728} \times 62.4 \times .95 = 593.16 \)

EFFECTIVE WEIGHT = \( \frac{1}{2} \) WEIGHT

= 296.16 lbs

\( \times 396 = 3 \text{ percent of actual weight} \)

14,000
APPENDIX II

ANALYSIS OF A BEAM SUSPENDED BY CABLES AND UNDERGOING COMBINED BENDING AND PENDULAR MOTIONS

Assume that the bending motions include both symmetric and antisymmetric motions and that the pendular motions include both regular (translatory) and bifilar (rotary) motions.

![Diagram of a beam with cables and angles]

The governing differential equation is

\[ \frac{d^2}{dx^2} \left( EI \frac{d^2 \phi}{dx^2} \right) + \alpha \frac{d^2 \phi}{dx^2} = F(x,t) \]

where \( F(x,t) \) is the combination of applied external forces including the cable restraints.

Assume that

\[
\phi(x,t) = \sum_{i=1}^{\infty} \alpha_i(t) \phi_i(x)
\]

(1)

\[
= \alpha(t) + \sum_{j=1}^{p} \int_{j}^{j+1} \int_{j}^{j+1} \sum_{k=1}^{q} \int_{k}^{k+1} \frac{d^2 \phi_i(x)}{dx^2} \phi_i(x) dx
\]

(2)

where the \( \alpha_i(t) \)'s and \( \phi_i(x) \)'s are the natural coupled symmetric and antisymmetric modes of the beam, respectively.

Since our primary interest is the coupling between the pendular modes and the lower frequency elastic
MODES, WE CONSIDER ONLY THE FIRST SYMMETRIC MODE AND
THE FIRST ANTISYMMETRIC MODE FOR THE EQUILIBRIUM REPRESENTATION
OF THE BEAM, i.e.,

\[ \phi(x,t) = \alpha(t) + 60x + C(t) f(x) + a_d x g(x) \]  

which we can simplify to

\[ \phi = \alpha + bx + cf + dg \]  

and

\[ C \left( \frac{d^2\phi}{dx^2} \right) + d \left( \frac{d\phi}{dx} \right) + m \left( \frac{d^2\phi}{dt^2} + \frac{d\phi}{dt} + \frac{d^2f}{dt^2} + \frac{d^2g}{dt^2} \right) = F \]  

TO TAKE ADVANTAGE OF THE ORTHOGONALITY OF THE
NATURAL MODES, WE MAY MULTIPLY EACH EQUATION BY \( \phi \) AND
INTEGRATE OVER THE LENGTH. THEN

\[ \int_0^L \left( \frac{d^2\phi}{dx^2} \right) dx + \int_0^L \left( \frac{d\phi}{dx} \right) dx + \int_0^L m \phi dx + \int_0^L \frac{d\phi}{dt} dx \]

\[ + \int_0^L \frac{d^2f}{dt^2} dx + \int_0^L \frac{d^2g}{dt^2} dx = \int_0^L F dx \]  

\[ \int_0^L \left( \frac{d^2\phi}{dx^2} \right) dx + \int_0^L \left( \frac{d\phi}{dx} \right) dx + \int_0^L m \phi dx + \int_0^L \frac{d\phi}{dt} dx \]

\[ + \int_0^L \frac{d^2f}{dt^2} dx + \int_0^L \frac{d^2g}{dt^2} dx = \int_0^L F dx \]  

\[ \int_0^L \left( \frac{d^2\phi}{dx^2} \right) dx + \int_0^L \left( \frac{d\phi}{dx} \right) dx + \int_0^L m \phi dx + \int_0^L \frac{d\phi}{dt} dx \]

\[ + \int_0^L \frac{d^2f}{dt^2} dx + \int_0^L \frac{d^2g}{dt^2} dx = \int_0^L F dx + \int_0^L \frac{d\phi}{dt} dx \]  

\[ + \int_0^L \frac{d^2f}{dt^2} dx + \int_0^L \frac{d^2g}{dt^2} dx = \int_0^L F + \int_0^L \frac{d\phi}{dt} dx \]  

\[ + \int_0^L \frac{d^2f}{dt^2} dx + \int_0^L \frac{d^2g}{dt^2} dx = \int_0^L F + \int_0^L \frac{d\phi}{dt} dx \]

\[ + \int_0^L \frac{d^2f}{dt^2} dx + \int_0^L \frac{d^2g}{dt^2} dx = \int_0^L F + \int_0^L \frac{d\phi}{dt} dx \]

\[ + \int_0^L \frac{d^2f}{dt^2} dx + \int_0^L \frac{d^2g}{dt^2} dx = \int_0^L F + \int_0^L \frac{d\phi}{dt} dx \]
\[
\int (EF)^{n} q \, dx + d \int (Eg)^{n} q \, dx + a \int \sin \theta \, dx + b \int \sin \frac{\theta}{2} \, dx
\]

\[
+ c \int \sin \theta \, dx + d \int \sin^{2} \theta \, dx = \int F \, dx
\]

(9)

It is noted that in general, \( F \) may be a distributed force, and the \( \int F \, dx \) may be evaluated. For the cases of most interest here, \( F(x, t) \) is a series of concentrated forces represented by the flexure restraints and externally applied excitation forces.

Consideration of subcase where beam is non-uniform and undergoes free translatory and rotary pendular motions while attached to cables at each end.

For this case, \( c = \theta = 0 \) and the equation reduces to

\[
\int \sin \theta \, dx + b \int \sin \frac{x}{2} \, dx = \int F \, dx
\]

(10)

\[
\int \sin \frac{x}{2} \, dx + b \int \sin \frac{x}{2} \, dx = \int F \, dx
\]

(11)

Evaluation of integrals

\[
\int \sin \theta \, dx = M \quad \int \sin \frac{x}{2} \, dx = \frac{M}{L} \quad \int \sin \frac{x}{2} \, dx = \frac{M}{L} \quad \int \sin \frac{x}{2} \, dx = \frac{M}{L}
\]

In general, each cable attached to the beam will support a percentage of the beam mass and will provide a force to the beam as follows

\[
F(i) = - \frac{M_{i} \phi_{i}^{2}}{2} \quad F(i) = \sum_{i=1}^{n} \phi_{i}\left( \psi_{i} - \phi_{i} \right)
\]

(12)
\[ A = \text{STATION 1 (} t = 0) \]

\[ -F_1 = M_1 \varphi_1 = M_1 \frac{a}{L} (a + b) = M_1 \frac{a}{L} \]

\[ -F_2 = M_2 \varphi_2 = M_2 \frac{a}{L} (a + b) = M_2 \frac{a}{L} (a + b) \]

\[ \int_0^L F_1 \, dx = M_1 \frac{a}{L} a + M_2 \frac{a}{L} a + M_2 \frac{a}{L} b \]

\[ \int_0^L F_2 \, dx = M_2 \frac{a}{L} a + M_2 \frac{a}{L} b \]

For simple harmonic motion, \( \omega \mathbf{z} = -\omega^2 \mathbf{z} \), and the equations may be written in matrix form as follows:

\[
\begin{bmatrix}
-\omega^2 M + M_1 \frac{a}{L} + M_2 \frac{a}{L} \\
-\omega^2 X_{eq} + M_2 \frac{a}{L}
\end{bmatrix}
\begin{bmatrix}
a_0 \\
b_0
\end{bmatrix}
= 0
\]

\[ \begin{bmatrix}
\omega^2 \frac{g}{L} - \frac{g}{L} \left( \frac{M_1 + M_2}{M} \right) + \omega^2 \frac{X_{eq}}{L} - \frac{g}{L} \frac{M_2}{M} \\
\omega^2 \frac{X_{eq}}{L} - \frac{g}{L} \frac{M_2}{M} + \omega^2 \left( \frac{X_{eq} + r^2}{L} \right) - \frac{g}{L} \frac{M_2}{M}
\end{bmatrix}
\begin{bmatrix}
a_0 \\
b_0
\end{bmatrix}
= 0
\]

The frequency equation is given by the vanishing of the determinant of (18):

\[
\left( \omega^2 - \frac{g}{L} \left( \frac{M_1 + M_2}{M} \right) \right) \left( \omega^2 \left( \frac{X_{eq} + r^2}{L} \right) - \frac{g}{L} \frac{M_2}{M} \right) - \left( \omega^2 \frac{X_{eq}}{L} - \frac{g}{L} \frac{M_2}{M} \right)^2 = 0
\]
A subcase of interest is that of a uniform bar where
\( M_1 = M_2 = M/L, \ x_{eq} = L/2, \ r^2 = L^2/12 \)

\[
\left( \omega^2 - \frac{g}{L} \right) \left( \omega^2 - \frac{L}{2} \right) - \frac{1}{4} \left( \omega^2 - \frac{g}{L} \right)^2 = 0
\]

(20)

OR

\[
\left( \omega^2 - \frac{g}{L} \right) \left( \omega^2 - \frac{3g}{2} \right) = 0
\]

(21)

Equation (21) gives the natural frequencies of the
two pendulum modes which are shown to be uncoupled for
a uniform beam.

Of more general interest is the case of a nonuniform
beam. As an example, consider the case where the beam
consists of two sections of equal length for which the
first section is twice as heavy as the second section.

\[
M_1 = \frac{2}{3} M_2 = \frac{2}{3} M
\]

\[
M_2 = \frac{1}{2} M_2 = \frac{1}{2} M
\]

\[
x_{eq} = \frac{L}{12}
\]

\[
\nu^2 = \frac{L^2}{12} \left( \frac{2}{3} M_2 \left( \frac{L}{3} \right)^2 + \frac{1}{2} M_2 h^2 \right) = \frac{3}{14} M L^2 \left( \frac{L^2}{3} + h^2 \right) = \frac{1}{4} h^2
\]
SUBSTITUTION of Eq. (22) INTO Eq. (19) YIELDS

\[
\left( \omega^2 - \frac{g}{l} \right) \left( \omega^2 - \frac{5g}{12l} \right) - \left( \frac{5}{12} \right) \left( \omega^2 - \frac{g}{l} \right)^2 = 0
\]  

(23)

\[
\left( \omega^2 - \frac{g}{l} \right) \left( \omega^2 - \frac{35g}{11l} \right) = 0
\]  

(24)

AND AGAIN WE SEE THAT THE MOTIONS ARE UNICOUPLED.

SINCE THE MOTIONS ARE UNICOUPLED, WE SHOULD BE ABLE TO DERIVE THE BIAREAL PENDULUM FREQUENCY DIRECTLY FROM CONSIDERATION OF ROTATIONS ABOUT THE CENTER OF MASS AND BY THE USE OF ENERGY PRINCIPLES AS FOLLOWS.

\[
U = M_1 g \frac{\theta_1^2}{2} + M_2 g \frac{\theta_2^2}{2} 
\]  

(25)

\[
T = \frac{1}{2} I \hat{\theta}^2 = \frac{1}{2} M_1 \theta^2 \hat{\theta}^2 
\]  

\[
\text{Since } \phi_1 \theta = \Theta \hat{x}_1, \quad \phi_2 \ell = \Theta \hat{x}_2
\]

\[
U = \frac{9}{2} \left( M_1 \hat{x}_1^2 + M_2 \hat{x}_2^2 \right) 
\]  

(27)
Since

\[ \dot{M}_x = M_z (x_1 + x_2) \quad \text{and} \quad \dot{M}_z = M_z (x_1 + x_2) \]

we can multiply \( \dot{M}_x \) by \( x_2 \) and \( \dot{M}_z \) by \( x_1 \) and add them together to obtain

\[ \dot{M}_x x_2 + \dot{M}_z x_1 + M_z x_1 x_2 = 2 \dot{M}_x x_2 \quad (28) \]

but \( M = M_1 + M_2 \) and we obtain

\[ M_1 x_2 + M_2 x_1 = M_2 x \quad (29) \]

which yields

\[ U = \frac{g}{2} \frac{\dot{M}_x x_2}{2} \quad (30) \]

Assuming simple harmonic motion and equating the maximum kinetic and potential energies, we obtain

\[ \omega^2 = \frac{g}{r^2} \frac{x_2}{2} = 0 \quad (31) \]

From Eq. (28) we find \( x_1 = x_2 = \frac{5}{12} L \), \( x_2 = \frac{7}{12} L \)

and \( x_1 x_2 = \frac{35}{144} L \).

Also, from Eq. (32), \( \omega^2 = -x_2^2 + \frac{g^2}{4} = -x_1^2 + \frac{L^2}{4} = -\frac{25}{144} L^2 + \frac{L^2}{4} = \frac{11}{144} L^2 \)

and Eq. (31) yields

\[ \omega^2 = \frac{25}{11} \frac{g}{L} \quad (32) \]

Thus \( \omega^2 \) derived from the bifilar pendulum analysis is identical to the corresponding root from Eq. (24). Hence the pendular motions do not couple and the bifilar frequency increases as the beam becomes heavier at one end.
SOLUTIONS OF THE GENERAL CASE FOR COMBINED
ELASTIC AND PENDULAR MOTIONS

THE MORE GENERAL CASE OF COMBINED ELASTIC AND
PENDULAR MOTIONS IS OF INTEREST BECAUSE WE NEED TO
KNOW THE EXTENT TO WHICH COUPLING OCCURS.

CASE 1. COMBINATION OF SYMMETRIC ELASTIC MODE
AND TRANSLATORY PENDULAR MOTIONS

THIS CASE IS GIVEN BY Eqs. (6) AND (8) WITH \( \Delta \neq 0 \), i.e.
\[
\phi = a_1 \cos f
\]

AND
\[
\int_0^L (EI\phi'')'' dx = \int_0^L F dx \tag{32}
\]
\[
\int_0^L (EI\phi'')' dx + \int_0^L M_1 L dx + \int_0^L fM_2 dx = \int_0^L F + F dx \tag{34}
\]

EVALUATION OF INTEGRALS FOR SIMPLE HARMONIC MOTION

\[
\int_0^L (EI\phi'')' dx = 0
\]
\[
\int_0^L M_1 L dx = M
\]
\[
\int_0^L fM_2 dx = 0
\]
\[
\int_0^L F dx = F(1) + F(2) = -M_1 \frac{d \phi(1)}{dx} - M_2 \frac{d \phi(2)}{dx} \tag{35}
\]
\[
\int_0^L F + F dx = F(1) \phi(1) + F(2) \phi(2) = -M_1 \frac{d \phi(1)}{dx} - M_2 \frac{d \phi(2)}{dx}
\]
ASSUME THAT THE BEAM SUPPORTS ARE LOCATED AT X=0, X=L, AND THAT \( f(x) = f(L) = 1 \). THEN FOR A UNIFORM BEAM \( M(x) = M_0 = \frac{M}{L} \) AND

\[
\phi(x) = \phi_1(x) + \phi_2(x) = a + c
\]

\[
\phi(L) = \phi_1(L) + \phi_2(L) = a + c
\]

THEN

\[
\int_0^L F dx = -\frac{M g}{L} (a + c) - \frac{M g}{L} (a + c) = -\frac{M g}{L} (a + c)
\]

\[
\int_0^L F dx = -\frac{M g}{L} (a + c) - \frac{M g}{L} (a + c) = -\frac{M g}{L} (a + c)
\]

THE EQUATIONS OF MOTIONS THEN REDUCE TO

\[
\begin{bmatrix}
-\omega^2 M + M \frac{g}{L} \\
+ M \frac{g}{L} \\
+ M \frac{g}{L} \\
+ M \frac{g}{L}
\end{bmatrix} \begin{bmatrix} a_1 \\ a_2 \\ a_3 \\ a_4 \\ \end{bmatrix} = 0
\]

AND THE DETERMINANT, AFTER DIVIDING EACH EQUATION BY \( M \), IS

\[
\left( -\omega^2 + \frac{g}{L} \right) - \frac{\omega^2 M_f}{M} + \frac{\omega^2 M_f}{M} + \frac{g}{L} = 0
\]

OR

\[
\omega^2 = \left( \frac{g}{L} + \omega_1^2 + \frac{M g}{M_f} \right) \omega^2 + \frac{g}{L} \omega_1^2 = 0
\]
CHECK BY USE OF ENERGY METHODS AND LAGRANGE'S Eqs.

\[ U = \frac{1}{2} \int_0^L \left( E I f''(x) \right) dx + \frac{M_0}{2} \frac{d^2}{dx^2} f(x) \left( \frac{d}{dx} f(x) \right) \]

(41)

\[ T = \frac{1}{2} \int_0^L m \phi^2 \phi \]

(42)

But, for our conditions, \( M_0 = M(2) = M/4 \) and

\[ \phi = a + cf \]

(43)

\[ \phi' = cf', \phi'' = cf'' \]

\[ \phi = -\omega^2 (a + cf) e^{i\omega t} \]

Lagrange's Eqs. which apply are

\[ \frac{d}{dt} \left( \frac{dL}{dx} \right) + \frac{dU}{dx} = 0 \]

\[ \frac{d}{dt} \left( \frac{dU}{dx} \right) + \frac{dU}{dx} = 0 \]

The resulting Eqs. are

\[ U = \frac{1}{2} \int_0^L E I f''(x) dx + \frac{M_0}{2} (a + cf)^2 \]

(44)

\[ T = \frac{1}{2} \int_0^L m \phi^2 \phi \]

(45)

\[ \frac{d}{dt} \left( \frac{dL}{dx} \right) = \frac{d}{dt} \left( a \int_0^L \sin \omega t \phi dx + \frac{d}{dx} \int_0^L \sin \phi dx \right) = \frac{d}{dx} \int_0^L \sin \omega t \phi dx + \frac{d}{dx} \int_0^L \sin \phi dx \]

\[ = -\omega^2 a \sin \omega t \]

(46)

\[ \frac{d}{dt} \left( \frac{dL}{dx} \right) = \frac{d}{dt} \left( a \int_0^L \sin \omega t \phi dx + \frac{d}{dx} \int_0^L \sin \phi dx \right) = \frac{d}{dx} \int_0^L \sin \omega t \phi dx + \frac{d}{dx} \int_0^L \sin \phi dx \]

\[ = -\omega^2 \phi \]

(47)
\[
\frac{dl}{da} = \frac{M_0 (a+c)}{E}
\]
\[
\frac{dw}{dc} = \frac{M_0 (a+c)}{E} + c \int_0^l \rho I (r')^2 \, dr'
\]
\[
= \frac{M_0 (a+c)}{E} + c w^2 M_f
\]

AND, UPON COMBINATION OF TERMS AND DIVIDING BY M

\[
\begin{bmatrix}
-\omega^2 + \frac{g}{c} \\
\frac{g}{c} \\
\end{bmatrix}
\begin{bmatrix}
q_0 \\
c_0 \\
\end{bmatrix}
= 0
\]

WHICH IS THE RESULT SHOWN IN EQ. 3B
COMPUTATION OF REPRESENTATIVE FREQUENCIES

LET: \( y = \frac{M}{M_i} \); \( \frac{g}{L} = \rho \); \( \omega = \lambda \Delta \); \( \omega = \xi \Delta \)

THEN EQ. (40) REDUCES TO

\[
e^y - (1 + y + \alpha^2) e^\xi + \alpha^2 = 0
\]

ORIGINAL PAGE IS OF POOR QUALITY

IF WE ASSUME \( y = 4 \)

\[
\alpha = \frac{5x^2}{2} \pm \frac{1}{21} \sqrt{25 + 6x^2 + x^4}
\]

AND THE FOLLOWING TABLE SHOWS THE FREQUENCY CALCULATIONS

<table>
<thead>
<tr>
<th>( \alpha )</th>
<th>( \alpha^2 )</th>
<th>( \frac{5x^2}{2} )</th>
<th>( \frac{1}{21} \sqrt{25 + 6x^2 + x^4} )</th>
<th>( \epsilon_1 )</th>
<th>( \epsilon_2 )</th>
<th>( \epsilon_3 )</th>
<th>( \epsilon_4 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>3</td>
<td>2.83</td>
<td>5.63</td>
<td>0.17</td>
<td>2.44</td>
<td>0.41</td>
</tr>
<tr>
<td>2</td>
<td>4</td>
<td>4.5</td>
<td>4.03</td>
<td>8.53</td>
<td>0.17</td>
<td>2.92</td>
<td>0.63</td>
</tr>
<tr>
<td>3</td>
<td>9</td>
<td>7</td>
<td>6.32</td>
<td>13.32</td>
<td>0.65</td>
<td>3.65</td>
<td>0.22</td>
</tr>
<tr>
<td>4</td>
<td>16</td>
<td>10.5</td>
<td>9.71</td>
<td>20.31</td>
<td>0.74</td>
<td>4.50</td>
<td>0.89</td>
</tr>
<tr>
<td>5</td>
<td>25</td>
<td>15</td>
<td>14.14</td>
<td>27.14</td>
<td>0.83</td>
<td>5.10</td>
<td>0.92</td>
</tr>
<tr>
<td>6</td>
<td>36</td>
<td>20.5</td>
<td>19.60</td>
<td>40.10</td>
<td>0.90</td>
<td>6.63</td>
<td>0.95</td>
</tr>
<tr>
<td>7</td>
<td>49</td>
<td>27</td>
<td>26.07</td>
<td>53.07</td>
<td>0.93</td>
<td>7.28</td>
<td>0.96</td>
</tr>
<tr>
<td>8</td>
<td>64</td>
<td>34.5</td>
<td>33.56</td>
<td>68.06</td>
<td>0.94</td>
<td>8.25</td>
<td>0.97</td>
</tr>
</tbody>
</table>

\[
\alpha^2 = \frac{\omega}{(9/2)^{1/2}}
\]

For \( \alpha = 5 \)

\[
\frac{\omega}{\omega_0} = 5.4 = 1.08
\]

\[
\alpha = 5.4
\]

\[
\alpha = 5
\]

\[
\omega = 5.4
\]

\[
\omega = 5
\]

\[
\frac{\omega}{\omega_0} = (9/2)^{1/2}
\]

\[
\alpha = (9/2)^{1/2}
\]
THE GENERAL CASE, Eqs. (6) THRU (9), WILL NOW BE CONSIDERED.

THE INTEGRALS

\[ \int F \phi_i(x) dx = \sum_{j=1}^{n} F_j \phi_i(j) \]  

(53)

\[ F_j = -M(i) \frac{q_j}{L} \phi(i) \text{ AS SHOWN IN Eqs. (12)} \]

(54)

\[ = -M(i) \frac{q_j}{x} \left( a + bx^i + cx^i + dx^3 \right) \]

(55)

CH} \text{ BE EVALUATED FOR ANY NUMBER OF CABLES } n

THEN

\[ \int F \phi_i(x) dx = - \sum_{j=1}^{n} M(i) \left( a \phi_i(i) + b \phi'(i) + c \phi''(i) + dx^3 \phi(i) \right) \]

(56)

IF WE AGAIN RESTRICT OUR SELVES TO TWO CABLES, ONE AT EACH END OF A UNIFORM BEAM

\[ \int F \phi_i(x) dx = - M(i) \left( a \phi(i) + b \phi'(i) + c \phi''(i) \right) \]

(57)

\[ + a \phi'(i) + b \phi''(i) + c \phi''(i) \]

(58)

\[ \text{BUT } \phi_1(x) = 1, \phi_2(x) = \frac{x}{L}, \phi_3(x) = \frac{x^3}{3}, \phi_4(x) = 8 \]

\[ \text{AND } \phi_1(x) = 1, \phi_2(x) = 1, \phi_3(x) = 0, \phi_4(x) = 1, \phi_5(x) = 1, \phi_6(x) = 1 \]
THEN

\[
\int F_1(x) \, dx = -\frac{M g}{L} \left( a + c + d + a + b + c - d \right) \tag{57}
\]

\[
= -\frac{M g}{L} \left( a + \frac{L}{2} + c \right) \tag{57}
\]

\[
\int F_2(x) \, dx = -\frac{M g}{L} \left( 0 + c + c + 0 + a + b + c - d \right)
\]

\[
= -\frac{M g}{L} (a + b + c - d) \tag{58}
\]

\[
\int F_3(x) \, dx = -\frac{M g}{L} \left( a + c + c + d + a + b + c - d \right)
\]

\[
= -\frac{M g}{L} \left( 2a + b + 2c \right) \tag{59}
\]

\[
\int F_4(x) \, dx = -\frac{M g}{L} \left( a + c + d - a - b - c + d \right)
\]

\[
= -\frac{M g}{L} \left( \frac{L}{2} \right) \tag{60}
\]

AND THE DIFFERENTIAL EQUATIONS (6) THRU (9) ARE

\[
\begin{bmatrix}
-\omega_t^2 + \frac{g}{L} & -\omega_t^2 \frac{g}{M L} + \frac{g}{M} & \frac{g}{M} & 0 \\
-\omega_t^2 \frac{g}{M L} + \frac{g}{M} & -\omega_t^2 \frac{1}{3} + \frac{g}{M L} & \frac{g}{M L} & \frac{-g}{M L} \\
\frac{g}{M L} & \frac{g}{M L} & \omega_t^2 \frac{M g}{M} - \omega_t^2 \frac{g}{M} + \frac{g}{L} & 0 \\
0 & \frac{-g}{M L} & \omega_t^2 \frac{M g}{M} - \omega_t^2 \frac{g}{M} + \frac{g}{L} & \omega_t^2 \frac{M g}{M} - \omega_t^2 \frac{g}{M} + \frac{g}{L}
\end{bmatrix}
\begin{bmatrix}
a_0 \\
b_0 \\
c_0 \\
d_0
\end{bmatrix} = 0
\]
IT IS NOTED THAT IN THE DEVELOPMENT OF THESE EQUATIONS, SEVERAL INTEGRALS VANISH. FOR A FREE-FREE BEAM,

\[ \int_0^L f \, dx = 0 \]

\[ \int_0^L (EI f')'' \, dx = k \int_0^L f' \, dx = 0 \]  
\[ \text{ORIGINAL PAGE IS OF POOR QUALITY} \]  

\[ \int_0^L mg \, dx = 0 \]

\[ \int_0^L (EI g'')'' \, dx = k^2 \int_0^L mg \, dx = 0 \]

BECAUSE THE CENTER OF MASS OF THE BEAM DOES NOT MOVE.

OTHER INTEGRALS OF INTEREST ARE

\[ \int_0^L f \, dx = \int_0^L f \left( \frac{x}{L} - \frac{1}{2} \right) \, dx + \frac{1}{2} \int_0^L f \, dx = 0 \]

\[ \int_0^L (EI f'')'' \, dx = \int_0^L (EI g'')'' \left( \frac{x}{L} - \frac{1}{2} \right) \, dx + \frac{1}{2} \int_0^L (EI f'')'' \, dx = 0 \]  
\[ \text{ORIGINAL PAGE IS OF POOR QUALITY} \]  

\[ \int_0^L mg \left( \frac{x}{L} - \frac{1}{2} \right) \, dx = \int_0^L mg \left( \frac{x}{L} - \frac{1}{2} \right) \, dx + \frac{1}{2} \int_0^L mg \, dx = 0 \]

\[ \int_0^L (EI g'')'' \left( \frac{x}{L} - \frac{1}{2} \right) \, dx = \int_0^L (EI g'')'' \left( \frac{x}{L} - \frac{1}{2} \right) \, dx + \frac{1}{2} \int_0^L (EI g'')'' \, dx = 0 \]

NOTE THAT IN THE EXPRESSIONS ABOVE, THE FIRST INTEGRALS ON THE RIGHT HAND SIDE VANISH IDENTICALLY AND THE SECOND INTEGRALS VANISH BECAUSE THE CENTER OF MASS DOES NOT MOVE. IF THE BEAM IS NOT UNIFORM, THE SAME SITUATION PREVAILS EXCEPT THE MULTIPLIER IS \( \left( \frac{x}{L} - \frac{1}{2} \right) \) WHERE \( \varphi \) IS A CONSTANT NOT NECESSARILY EQUAL TO \( \frac{1}{2} \).