NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

Technical Memorandum 33-729

Experiences in Using Modal Synthesis Within Project Requirements

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(NASA-CR-143132) EXPERIENCES IN USING MODAL SYNTHESIS WITHIN PROJECT REQUIREMENTS (Jet Propulsion Lab.) 37 p HC $3.75 CSCL 20K

N75-27428

Unclas

G3/39 29266

JET PROPULSION LABORATORY

CALIFORNIA INSTITUTE OF TECHNOLOGY

PASADENA, CALIFORNIA

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PREFACE

The work described in this report was performed by the Applied Mechanics Division of the Jet Propulsion Laboratory.

English units were used for the principal measurements and calculations in this report.

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ABSTRACT

Modal systhesis methods have been developed for use by engineers for cost-effective solutions of large structural dynamics problems. Different methods have been proposed by various investigators based upon comparative solutions of relatively simple structures using different forms of displacement functions. The paper describes the experiences in the application of modal synthesis methods to a large complex structure in a project environment. The considerations include analysis, hardware interfaces, organizational interfaces, schedules, tests, resources, and other project requirements. Good technical results were obtained through the use of an integrated analysis/test modal synthesis on both substructures and systems. These experiences should be beneficial for engineers contemplating the use of modal synthesis concepts for future projects.

Introduction

Modal synthesis concepts have been attractive to many engineers involved with the solution of eigenvalues and eigenvectors for large, complex dynamic problems. It is a method that retains the few significant independent coordinates of the various substructures which are combined into the system dynamic equations.

The reduced independent coordinates are selected to provide accuracy in the lower eigenvalues which are of significance to the structural dynamicists. The initial developments were motivated by a requirement for lower eigenvalues of large structures for reasonable computer costs.

The first publication popularizing modal synthesis was a report by W. Hurty [1].[1] Since then, several other documents on its different aspects, such as optimum selection of displacement functions, have been published [2, 3, 4, 5, and 6]. Other studies [7 and 8] have been made on Shuttle type structures to establish the "best" displacement functions that converge to the correct solution with the least number of independent coordinates. A current research program at Langley Research Center includes a task on a 1/8-scale Shuttle model to verify modal synthesis concepts as applied to the Shuttle.

[1]Numbers in brackets designate references at end of paper.

Although versatile computer codes have been available for eigenvalue solutions of a structure, and a few modal synthesis computer programs [9] have been developed, very few projects have successfully implemented a complete modal synthesis substructure and system analysis and test program.

This paper presents the experience at the Jet Propulsion Laboratory (JPL) in a complete modal synthesis analysis and test program. The "best" method for the project was not determined by analytical studies [2 and 3] but by project requirements. The results of the effort are based upon its application to six space projects [10 and 11] at JPL, including the most recent one, Viking[2]. The Viking results will be used to illustrate the salient points because it incorporated all of the experiences at JPL. The information should be of interest for engineers contemplating the use of modal synthesis concepts for future projects.

The proper use of modal synthesis will result in an efficient and cost-effective support to many projects.

Project Requirements

The "best" modal synthesis method and/or a modal synthesis computer program cannot be specified because of their dependence on the particular structure and the project requirements. The Project requirements for Viking [12] should be similar to those for future projects.

Since labor is often the largest cost element, most projects minimize the time for the design, fabrication, and test phases, as shown in Fig. 1. For a

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[2]JPL is responsible for the Viking Orbiter System, which is part of the overall Viking Project managed by the Viking Project Office at Langley Research Center (LRC) for NASA.

series of projects with minor modifications to the basic structure, higher confidence in the design and minimization of the test program can be achieved by initially developing a good mathematical structures and dynamics model that is verified by a test program. The analysis based on the initially developed mathematical model can be used in subsequent projects for reliable data and elimination of tests.

Schedule — Observations

The Viking Orbiter (VO) schedule pertinent to this discussion is shown in Fig. 2.

The current trend is to establish initial design loads using load analyses which are transient analyses based on spacecraft and launch vehicle mathemetical models and launch vehicle engine forcing functions. The final model, verified by a system modal test, is used in a load analysis to establish the final flight loads. The results are used to establish the adequacy of the structure and/or the ultimate static qualification test loads. Emphasis is placed on the calculation of spacecraft member dynamic loads rather than accelerations.

The time available between the system modal test and the test-correlated mathematical model is usually a minimum. Thus, if the test and analysis do not correlate, modification of the large system mathematical model to match the test results is an almost insurmountable task. Criteria for the correlation of analysis and test [13] are lacking, and a proven algorithm to automatically modify the mathematical model to match the test data is not available. Consequently, the results from the system modal test are often not used effectively.

The effective use of resources is made possible by

1. Verifying the mathematical models of the substructures with substructure tests. (The term substructure is deliberately used instead of subsystem.)

2. Increasing the probability of a good system test by recognition of potential problems that may be caused by selected substructures. The potential problems include nonlinearities, instrumentation requirements, mass matrix estimations, and identification of significant dynamic characteristics.

3. Distributing the instrumentation and engineer workload over a longer time period.

4. Performing a good substructure analysis and test program to eliminate the requirements for system tests.

Dynamic Model Requirements

The dynamic model requirements for different projects will vary along with the appropriate modal synthesis approach. The different project uses for a dynamic model are to evaluate the

1. Attitude control interaction

2. Spacecraft/launch vehicle interface loads or accelerations

3. Spacecraft accelerations

4. Spacecraft member forces

Another consideration is whether the significant dynamic forcing function is superimposed with a quasi-static acceleration.

System Modal Test

The need for an accurate analytical model prior to and after a system modal test is dependent on the project requirements. Other than to obtain good early load estimates and to help guide the system modal test itself, an analytical model that correlates with test results would be superfluous if the system test results could be used directly. This is rarely the case, since it is difficult to determine force coefficients experimentally and experimentally determined models often result in numerical difficulties. For attitude control interaction studies, or for the evaluation of spacecraft/launch vehicle interface loads or accelerations, modal test data are probably sufficient.

However, frequently a modal-test-correlated analytical model is desirable for the following reasons:

1. The modal test configuration may not duplicate the final flight configuration because

   a. Referee propellants are used for safety considerations. The density of the referee propellants differs from that of the flight propellants.

   b. The test configuration may exclude a few select substructures. Substructures (e.g., nonlinear ones) may be excluded to assure a good modal test to provide physical space to attach the shakers, or to minimize the number of system eigenvalues to those that require verification.

c. The test configuration is modified to allow a meaningful modal test. For instance, discrete dampers may be inactivated or sliding joints may be prevented from sliding.

d. Design changes have been made since the fabrication of the hardware for the test.

2. More detailed information is available from the mathematical model than from the test measurements. For instance, detailed modal force distribution can be obtained from the mathematical model, whereas its measurement during the test is impractical.

3. A variety of configurations must be evaluated for its dynamic characteristics. For instance, attitude control studies with various positions of appendages, launch analyses, and ground condition tests are typical of configurations required for one project.

4. The mathematical model is valuable for follow-on projects to provide fundamental information which may allow elimination of some analyses and tests.

Two difficulties that currently exist are a lack of

1. A criterion for the degree of correlation [13 and 14] required by an engineer.

2. A proven algorithm to upgrade a mathematical model [15 and 16] to correlate with test data.

Although several methods are available to reconstruct the stiffness and mass matrix from the test data [17, 18, and 19], they appear to lose their physical significance; thus, their use for analyzing other configurations is limited.

Viking Requirements

The Viking requirements [12] are described to provide a background for the modal synthesis approach. In general, the requirements were more extensive than those of previous JPL projects.

Dynamics Data

Since the design and qualification test loads were established by load analysis, the detailed loads in the various structural members resulting from combined quasi-static and staging transients were required. Load analysis consists of developing a complex finite element model (approximately 32,000 degrees-of-freedom) of the VO (Figs. 3 and 4), which is reduced to about 250 dynamic degrees-of-freedom by modal synthesis. The model is coupled by modal synthesis methods to the Viking Lander capsule and the Titan IIIE/Centaur D-IT launch vehicle (Fig. 5) and excited by the forcing functions shown in Fig. 6. Loads resulting from eigenvalues less than 40 Hz were of interest. (The first Viking spacecraft eigenvalue is 4.42 Hz and the 40th eigenvalue is 43.53 Hz.)

Schedule

As shown in Fig. 1, the modal-test-correlated mathematical model was required within a few weeks after the completion of the VO modal test [20]. The model was required for load analysis to establish the ultimate qualification static test loads prior to the scheduled test date. Fortunately, good substructure modal tests and the analyses program directly contributed to a successful system test and correlation of the analytical model. The schedule would not have permitted an update of the mathematical model.

A high probability for a good mathematical model, modal test, correlation, and successful ultimate static test was required because a redesign and requalification of the structure prior to launch in the third quarter of 1975 would have been costly.

Dynamic Model Requirements

A mathematical model of the VO was required. The modal-test-verified mathematical model was used to establish ten different models representing a variety of test and flight conditions and approximately six models for attitude control studies in the VO cruise configuration (Fig. 4). All VO models were generated using the JPL Structural Analysis and Matrix Interpretive System (SAMIS) computer program. The computer used was the Univac 1108.

Modal Synthesis

Definition of Problem

The major step required to obtain a solution to the system equations of motion

$$[M]\{\ddot{U}\} + [C]\{\dot{U}\} + [K]\{U\} = \{F(t)\}$$
$$\{f\} = [S]\{U\} \tag{1}$$

where

$[M]$ = mass matrix of the system

$\{U\}$ = independent coordinates of the system

$[C]$ = damping matrix of the system

$[K]$ = stiffness matrix of the system

$\{F(t)\}$ = forcing function

$\{f\}$ = member forces

$[S]$ = force transformation

is to reduce the degrees-of-freedom of Eq. (1) by representing $\{U\}$ in terms of various displacement functions. The relationship is

$$\{U\} = [\phi]\{Q\} \tag{2}$$

where

$[\phi]$ = displacement function matrix

$\{Q\}$ = independent generalized coordinates of the system

Substitution of Eq. (2) into (1) and premultiplication by $[\phi]^T$ results in

$$[\phi]^T [M] [\phi] \{\ddot{Q}\} + [\phi]^T [C] [\phi] \{\dot{Q}\} + [\phi]^T [K] [\phi] \{Q\} = [\phi]^T \{F(t)\} \tag{3}$$

or

$$[\bar{M}] \{\ddot{Q}\} + [\bar{C}] \{\dot{Q}\} + [\bar{K}] \{Q\} = \{\bar{F}(t)\} \tag{4}$$

The eigenvalue solution of (4) for $[\bar{C}] = 0$ and $\{\bar{F}(t)\} = 0$ results in the system eigenvalues and eigenvectors that are verified by the system modal test.

The main advantage of modal synthesis or the proper establishment of relationship (2) is that the order of the equations of motion is reduced from 32,000 for (1) to 250 for (3), with little loss in accuracy of the desired information.

Displacement Functions

The various displacement functions for a substructure can be obtained from the following two equations for any substructure:

$$\begin{bmatrix} k_{II} k_{IO} \\ k_{OI} k_{OO} \end{bmatrix} \begin{bmatrix} u_I \\ u_O \end{bmatrix} = \begin{bmatrix} f_I \\ f_O \end{bmatrix} \tag{5}$$

or

$$[u_O] = [k_{OO}]^{-1} \left([f_O] - [k_{OI}][u_I]\right) \tag{6}$$

and

$$[m]\{\ddot{u}\} + [k]\{u\} = \{0\} \tag{7}$$

where

$\{u\}$ = substructure displacements

$[k]$ = substructure stiffness matrix

$[f]$ = forces on the substructure

$[m]$ = substructure mass

I = subscript to denote interface degrees-of-freedom

O = subscript to denote degrees-of-freedom other than the interface.

The various displacement functions are discussed below.

1. Rigid Body Displacement Function [9]. Rigid body functions represent the motion $[\phi_R]$ when a degree-of-freedom $[u_I]$ is displaced by an arbitrary value without force. The $[\phi_R]$ is a solution to Eq. (6), where $[f_O] = 0$ and $[u_I]$ is a unit matrix (or linear combination of unit matrices) in the degrees-of-freedom associated with the rigid body displacements.

$$\{u_R\} = [\phi_R]\{q_R\} \tag{8}$$

The number of rigid body motions may range from 1 to ∞. Rigid body motions in excess of 6 are related to linkages within the substructure.

2. Constraint Displacement Functions [1]. Constraint functions represent displacements $[\phi_C]$ of the substructure when a unit displacement of an interface degree-of-freedom requires force as the other interface degrees-of-freedom are restrained. If the interface degrees-of-freedom are statically determinate, the constraint functions are equal to the rigid body functions. The constraint

functions are solutions to Eq. (6), where $\lfloor f_O \rfloor$ = 0 and $\lfloor u_I \rfloor$ is a matrix of displacement vectors with unity associated with the degrees-of-freedom defining a constraint mode. (Other terms of the vector are zero.)

$$\left\{u_C\right\} = [\phi_C]\left\{q_C\right\} \tag{9}$$

3. Attachment Displacement Functions [9, 5, and 6]. Attachment functions are displacements $[\phi_A]$ of the substructure resulting from concentrated loads $\lfloor f_A \rfloor$ on the substructure. Displacements $[\phi_A]$ result from the solution to (6), where $[f_O] = \lfloor f_A \rfloor$ and $[u_I]$ = u.

$$\left\{u_A\right\} = [\phi_A]\left\{q_A\right\} \tag{10}$$

The $\lfloor f_A \rfloor$ can be quasi-static inertia loading of the substructure or various combinations of concentrated loads.

4. Imposed Displacement Functions. Imposed functions represent motions $\lfloor \phi_{IM} \rfloor$ that engineers consider relevant to describe a structural deformation. The displacements are not necessarily a result of any realistic external loads, but are usually directly related to such loads.

$$\left\{u_{IM}\right\} = [\phi_{IM}]\left\{q_{IM}\right\} \tag{11}$$

5. Eigenvector Displacement Functions. The eigenvector functions are the eigenvector solution of Eq. (7).

$$\left\{u_E\right\} = [\phi_E]\left\{q_E\right\} \tag{12}$$

Total Displacement Function

The displacements of any substructure are represented by any combination of the displacement functions

$$\{u\} = \left[[\phi_R][\phi_C][\phi_A][\phi_{IM}][\phi_E]\right] \begin{Bmatrix} q_R \\ q_C \\ q_A \\ q_{IM} \\ q_E \end{Bmatrix} \tag{13}$$

or

$$\{u\}_i = [\phi_i]\{q_i\} \tag{14}$$

for the i^{th} substructure.

Equation of Motion of Total System

The independent generalized coordinates of the system are selected by the engineer, who combines the generalized coordinates of the substructures through compatibility relations representing the interconnections. Figure 7 shows that the substructures are attached to the bus for Viking. The selected displacement functions of the substructures (Eq. 13) are combined to obtain the system equations (4).

Modal Synthesis--VO Project Requirements

Project Requirements

The experience gained with the modal synthesis process for VO should be applicable to most projects. At the initiation of the Viking Project, the modal synthesis plan was closely integrated with other Project constraints and requirements that included

1. Organizational interfaces external to JPL (Figs. 3 and 5)
2. Organizational interfaces internal to JPL
3. Substructure design, fabrication, and delivery schedules
4. System-related hardware and test program
5. Load analysis definition and requirements

A modal synthesis plan based solely upon the "best" technical approach could not be practically implemented and would not have been acceptable. Two decisions, partially involving analysis and test considerations, were:

1. The responsibility for the Viking Lander capsule adapter (Fig. 3) was assigned to JPL. JPL was cognizant of the hardware which was important in the modal synthesis and the system test plans.
2. The Viking transition adapter (VTA) and Centaur truss adapter (CTA) that interconnect the Viking Orbiter System and Centaur were included in the JPL VO mathematical model, modal, and static qualification test program. The VTA and CTA were designed and fabricated by General Dynamics/Convair Astronautic (GD/CA).

A goal was to minimize the analysis, design, and test interfaces between organizations and people to decrease the coordination effort. Modal synthesis methods minimized interactions and provided a means to effectively obtain good technical results. The complexities associated with the inclusion of the VTA/CTA into the VO effort for technical considerations clearly demonstrated the advantages of minimizing the interfaces whenever possible.

A more detailed schedule is presented in Fig. 8, and the interfaces of the substructures and system are shown in Fig. 9.

All the structural development and structural qualification testing was performed by the JPL Structures and Dynamics Section.

Selection of Substructures

The substructures were defined by the interfaces shown in Fig. 9 rather than an analytical consideration. As noted in the schedule (Fig. 8) and the interfaces (Fig. 9), the substructures were under the cognizance of different engineers and organizations. They were delivered and tested at different times. To provide effective support, analysis, design, and test efforts were performed for each substructure with a minimum of interaction. This naturally resulted in the substructure/modal synthesis approach.

Advantages of Substructure Approach

The goal was to effectively support the VO and develop a mathematical model that would be verified by the system modal and static tests. The plan was to develop the system mathematical model (used for both static and dynamic models) by testing the substructures at the earliest possible time and incorporating the results into the system model.

The advantages of modal synthesis to VO were:

1. A structures engineer provided static and dynamic analysis and test support to his assigned substructure with a minimum of interaction with other substructures.

2. Each engineer and/or organization developed the mathematical model independently of the others. The number of degrees-of-freedom, numbering of nodes, and computer program were selected by each engineer. Thus, a special dynamic model with "reduced" degrees-of-freedom was not required. The model was used to evaluate substructure responses, interface distortion, and other parameters.

3. A better engineering estimate of substructures prior to the test was possible when experience on similar hardware was available. Estimates of damping on substructures made of honeycomb or composites are typical examples.

4. An engineer developed a substructure mathematical model, which was correlated and corrected to match the test data. Corrections were made to relatively small mathematical models prior to their incorporation into the system model.

5. Ninety percent of the strain gages required for the system model and static tests were installed for the substructure tests. This distributed the instrumentation workload and simplified the work by allowing hardware to be instrumented in the instrumentation laboratory. Additionally, the instruments

and their calibration were validated during the substructure tests.

6. The substructure tests revealed design deficiencies early in the VO schedule.

7. The potential problems caused by the various substructures during the system modal tests were recognized. For instance, structural nonlinearity would invalidate the system modal test based upon linear theory. The troublesome substructures were modified or eliminated from the system test.

8. The system modal [20], vibration [21 and 22], and qualification static [23] test results were excellent. Good test results were due to instrumentation and mathematical model verification during the substructure tests.

9. System parameters, such as the pressure in the propellant tank, were established based upon the influence of the pressures on the tank stress and structural nonlinearity.

10. A variety of different system configurations were effectively analyzed by changing only the affected substructures. The configurations analyzed included:

    a. Two different launch configurations with different propellant loadings.

    b. Different launch configurations to establish the maximum or minimum propellant loading which would affect the structural design.

c. Modal test configuration without selected substructures and referee propellants in the propellant tanks.

d. Forced vibration test configurations.

e. Attitude control cruise configurations.

Description of VO Substructures and Their Displacement Functions

General. All the dynamic and static tests to be described were performed at different magnitudes of loading to establish the nonlinearities and high-level damping trends. Since loads were required when members were subjected to combined quasi-static and dynamic loading, modal acceleration methods (rather than modal displacement) were used. In addition, the selected displacement functions should result in a small residual mass [24] across an interface for which quasi-static loads were important. (This can be achieved by selection of quasi-static attachment functions.) Strain gage readings were measured during the substructure tests to partially verify the eigenvalue force coefficients.

Individual Members and Joints. Whenever feasible, individual truss type members, and occasionally the joints with their instrumentation, were tested in a uniaxial testing machine. The primary objective was to detect a poor design or deficient hardware early in the Project schedule. Additionally, the stiffness of the substructure was verified and the instrumentation was calibrated. A test of the Viking spacecraft adapter truss/fitting revealed a joint that "gapped" when a tension load was applied.

Solar Panels. Eigenvector and constraint functions were used for the solar panel. The solar panel with the relay antenna was verified with a modal and static test in the launch configuration, and a modal test in the cruise or extended configuration. After correlating the mathematical model with the test results, the antenna was removed from the mathematical model to obtain the solar panel model without the relay antenna.

In addition to the solar panel tests, dynamic and static tests were performed on the aluminum honeycomb substrate to verify its structural integrity and the structural properties for the mathematical model.

The analysis indicated more eigenvalues in the frequency range of interest than the test results. Only the analytical eigenvalues corresponding to test eigenvalues were retained, and the eigenvalues were modified to match the test results. The analytical eigenvectors were retained.

The solar panels are connected to the Viking spacecraft adapter with "viscous dampers" that critically damp the panel lowest normal mode. Since the system eigenvalues were limited to real eigenvalues, the dampers were not included. The influence of damping from the viscous dampers was treated as solar panel eigenvalue damping.

Effective Mass Determination of the Propellant. A forced vibration test was performed on a single propellant tank with the propellant management device to measure the effective mass of the propellant in the lateral, longitudinal, and pitch directions for various ullages. The data were used for both the modal test and analysis.

Propellant Tank Tab. The reduced stiffness matrices of the propellant tabs as attached to the members supporting the tanks were calculated and verified by applying static loads similar to attachment functions to confirm the mathematical model. Imposed displacements similar to constraint functions were then applied to the mathematical model to verify the reduced stiffness matrices.

Propulsion Substructure. Eigenvalue and constraint functions were used to describe the motion of the propulsion substructure. Modal and static tests were performed to verify the functions. Tests were made at various internal pressure levels and ullages to establish the threshold of nonlinearity caused by propellants and structural nonlinearities. Zero ullage tests were run in the event that ullages resulted in nonlinearities which would compromise the system modal test. Although a zero ullage condition test was included in the system modal test plan, it was canceled because the ullage conditions did not introduce significant nonlinearities.

Components mounted on isolation pads to the propulsion substructure resulted in local eigenvectors that did not affect the overall significant eigenvectors.

High-Gain Antenna. Eigenvalue and constraint functions were used to describe the motion of the high-gain antenna. Modal and force vibration sine tests were performed to verify the analytical model. The design, including "snubbers," resulted in a nonlinear (frequency vs force) structure.

Additional modal tests were run for the antenna deployed positions.

Science Platform. Eigenvalue and constraint functions were used to describe the platform motion. Modal tests in the stowed and deployed conditions were performed to verify the model. The design included serrated joints to allow slippage at high loads. However, since the magnitude of the forces in the modal test did not allow slippage, the modal analyses excluded joint slippage, but the model for load analyses did include joint slippage.

During the test, the fixture was not sufficiently rigid, and it rotated. Because of schedule and cost considerations, the test was not repeated. The influence of base rotation could not be eliminated from the test results; thus no experimental results were available for correlation with analyses. An uncorrelated analytical model was used for the system modal test. The science-platform-related eigenvalues and eigenvectors revealed the worst correlation in the system modal test. This deficiency was, however, understood and allowances were made in the loads calculation to cover these uncertainties.

Cable Trough. Prior to the buildup of the structure for the system modal test, a quick modal test of the cable trough was performed to identify the eigenvectors and establish the adequacy of the experimental mass distribution.

Electronic Chassis. Imposed functions were selected for the electronic chassis to establish its generalized stiffness and mass matrix based upon its distortion when it was integrated with the bus. Tests were not run because of the difficulty in imposing the boundary conditions.

Bus. The bus structure included the rigid mass simulation of the Viking Lander capsule (VLC), Viking Lander capsule adapter (VLCA), Viking spacecraft adapter (V-S/C-A), and the VTA/CTA. The rigid mass of the VLC was included to allow the substructure function to be more representative of its motion in the system response in order to simplify and minimize the selection of bus functions. The inclusion of the rigid mass is identical to mass loading the interface [2]. As mentioned before, the GD/CA hardware, referred to as the VTA/CTA between the Centaur and Viking spacecraft, was included for analysis purposes. The VTA/CTA at the Viking interface was a flexible structure, whereas at the Centaur interface it was considered a rigid plane. The attachment or constraint functions to attach the Viking to the Centaur were eliminated by the inclusion of the CTA/VTA in the bus model.

Three types of functions as independent coordinates were tried for the bus:

1. Attachment functions related to the forces from the substructures.

2. Eigenvector functions with the interfaces to the substructures mass loaded and stiffness loaded (if statically indeterminate). They were linearly combined to obtain super-elements [11] compatible with the substructure degrees-of-freedom.

3. Degrees-of-freedom associated with the bus mass points.

The bus functions were verified by a static test. The static displacements and internal member forces were used to verify the mathematical model.

System Model

The effort to generate a mathematical system model using attachment and eigenvector functions resulted in failure. The resulting mass matrix of Eq. (1) could not be decomposed for the eigenvalue solution. This may be attributed to single-precision arithmetic. It is a limitation of SAMIS using the Univac 1108 computer.

The use of the bus's original degrees-of-freedom and the substructure functions (mixed coordinate system [14]) was successful. The model was verified by a system modal test and a system static test. Eigenvalues, eigenvectors, static displacements, and eigenvalue force coefficients were verified. The system test did not include all the substructures but only those necessary to verify the model and structure.

The eigenvalue damping was estimated by calculating the kinetic energy participation of various substructures in each eigenvalue and proportioning the substructure damping in relation to their contributions.

Conclusion

Modal synthesis concepts are valuable in the solution of large dynamic problems as well as effective in the support of a project. However the approach or selection of the methodology must not be based solely on "theory" but must be closely integrated with the overall project plan. Fortunately, the project objectives and modal synthesis desires are often similar (e.g., simplify interfaces). The selected methodology should also consider the ability to verify the mathematical model by an experimental program. A difficulty in modal synthesis is that the dynamicist must have a good understanding of structural dynamics to combine the substructures; automated computer programs to select the "best" methods are not available and may not be feasible for a general type of structure.

The use of modal synthesis resulted in an excellent mathematical model and meaningful test results, as well as a good correlation of analysis and test, for the VO. Its advantages for multiple mission projects are even greater because of the potential savings by elimination of analysis and tests. Using the verified mathematical model, only those substructures to be changed for a mission will require analysis for incorporation into the system model. Modal synthesis provides a means by which past experiences of a project can be fully utilized.

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22. Fortenberry, J.W., and Rader, W.P., "Fail Safe Vibration Testing of the Viking 1975 Developmental Spacecraft," presented at the 45th Shock and Vibration Symposium, Dayton, Ohio, October 1974.

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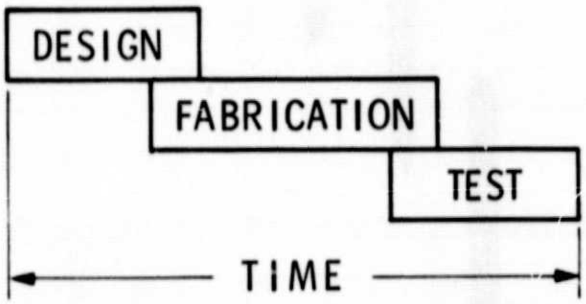


Fig. 1. Typical project

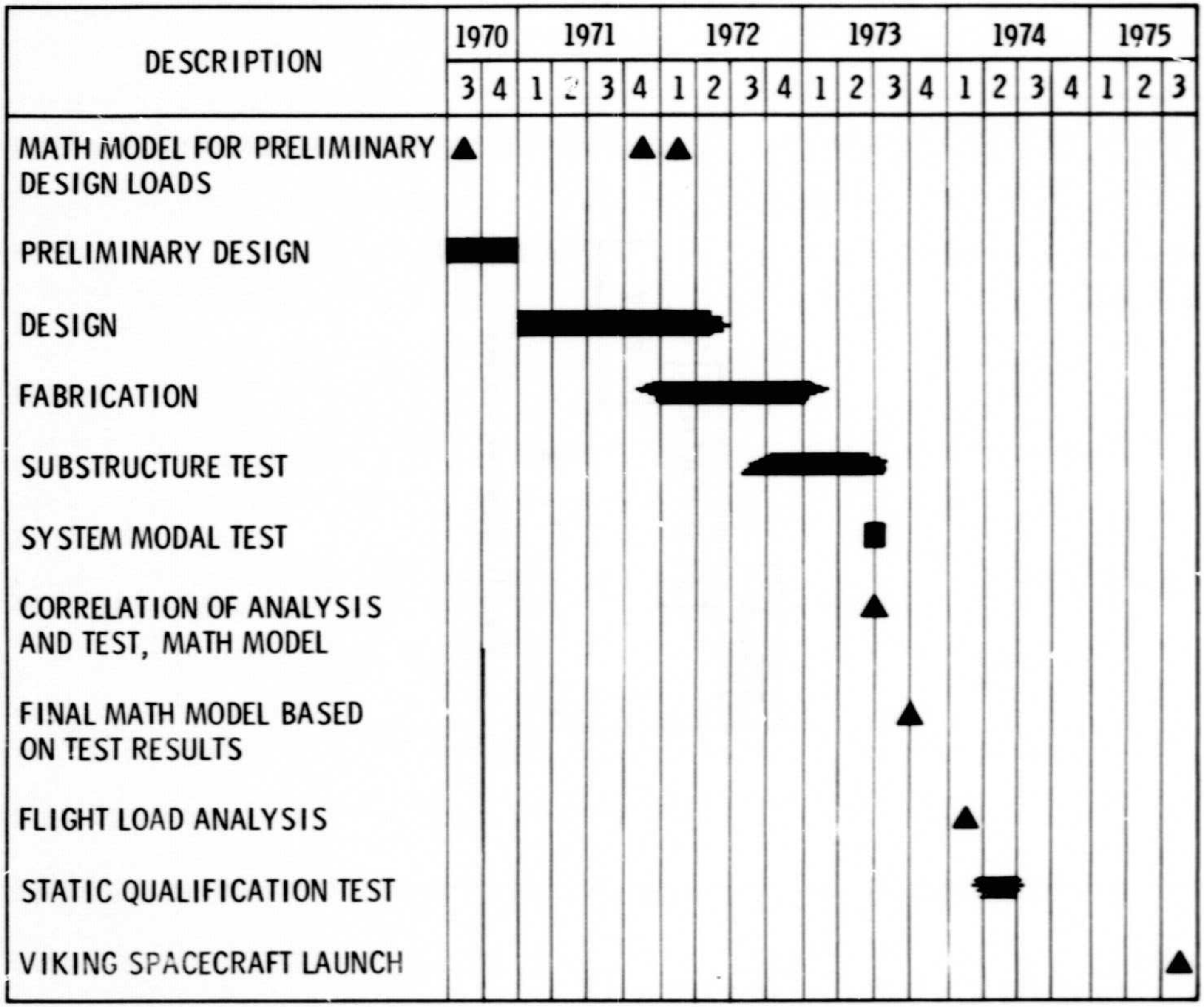


Fig. 2. Viking Orbiter structures and dynamics overall schedule

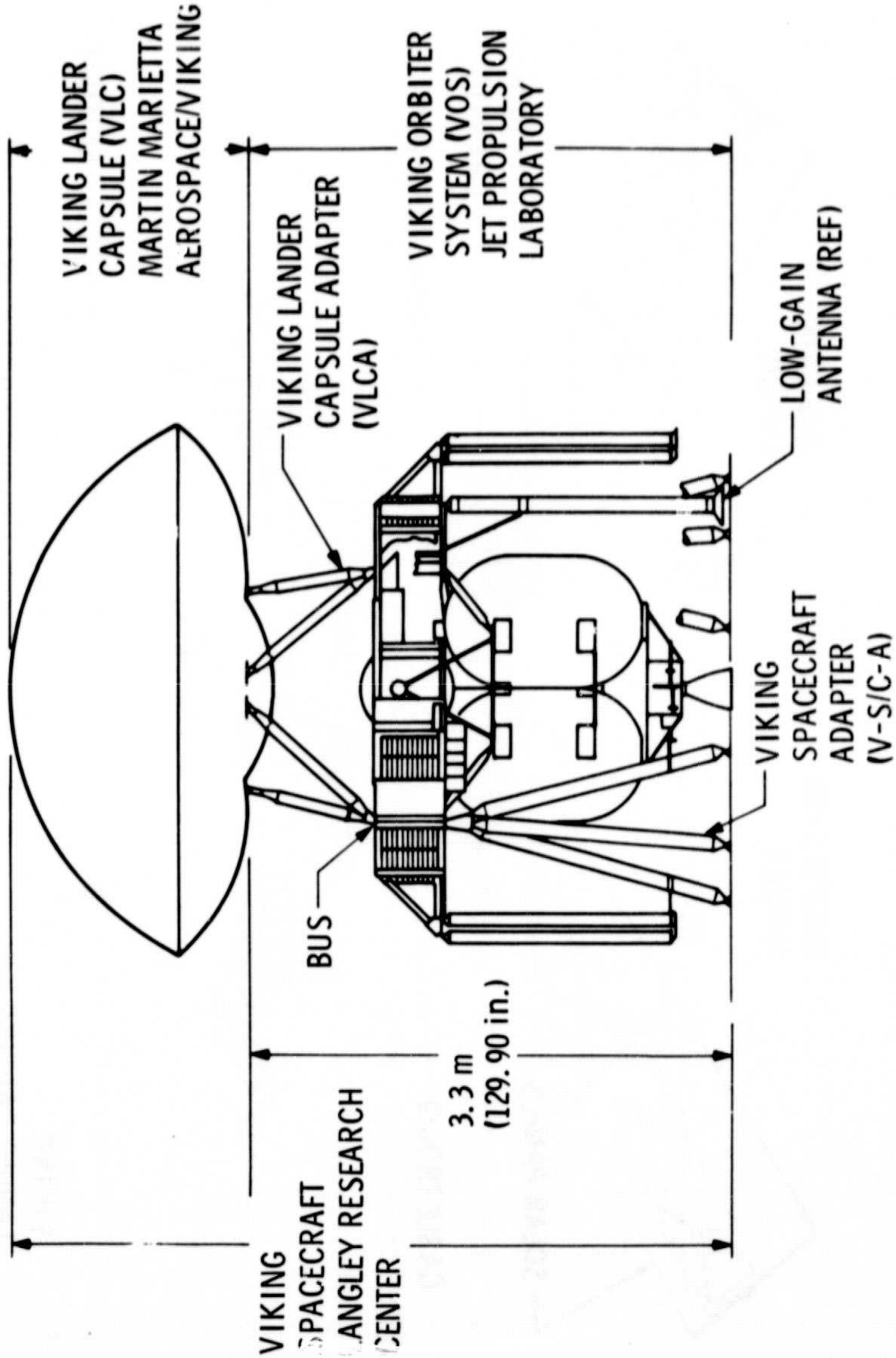


Fig. 3. Viking spacecraft configuration

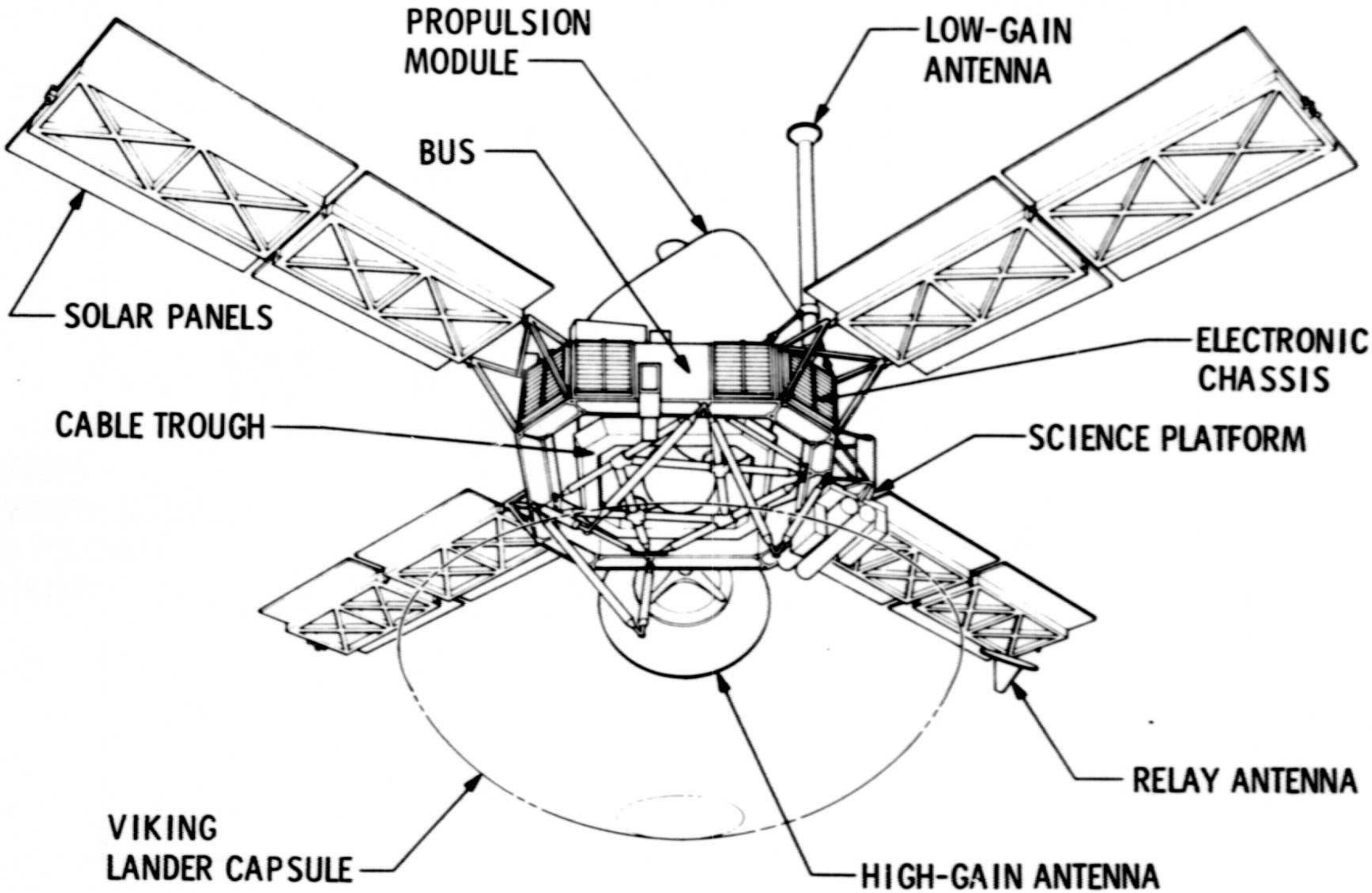


Fig. 4. Viking Orbiter substructures

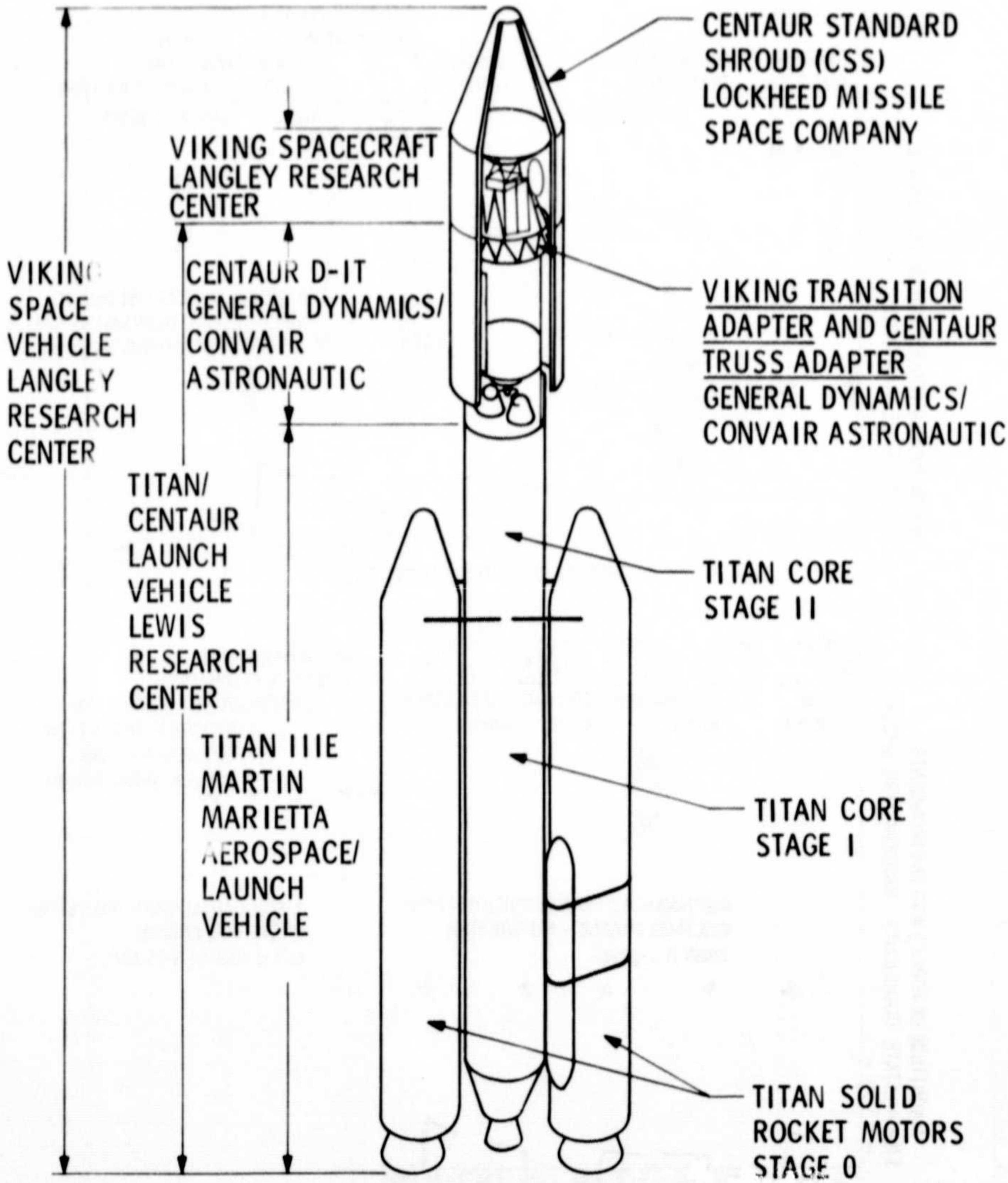
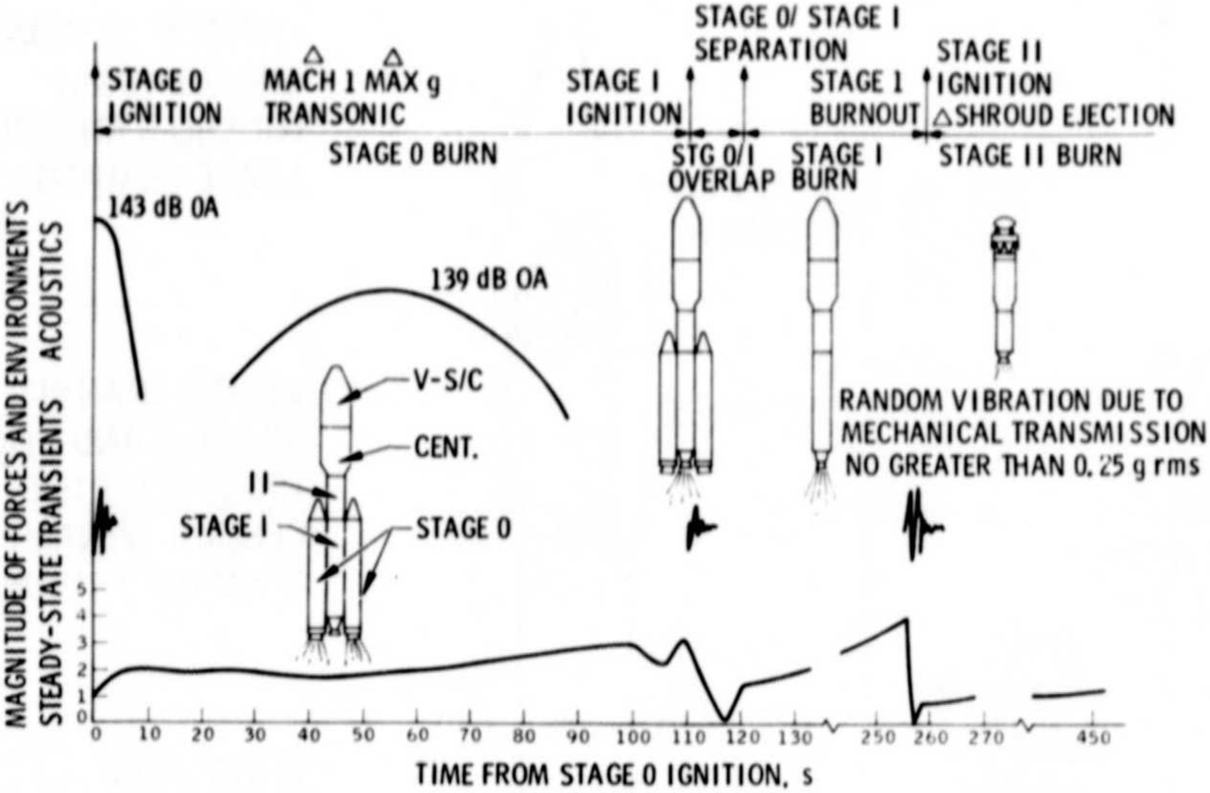


Fig. 5. Viking space vehicle configuration



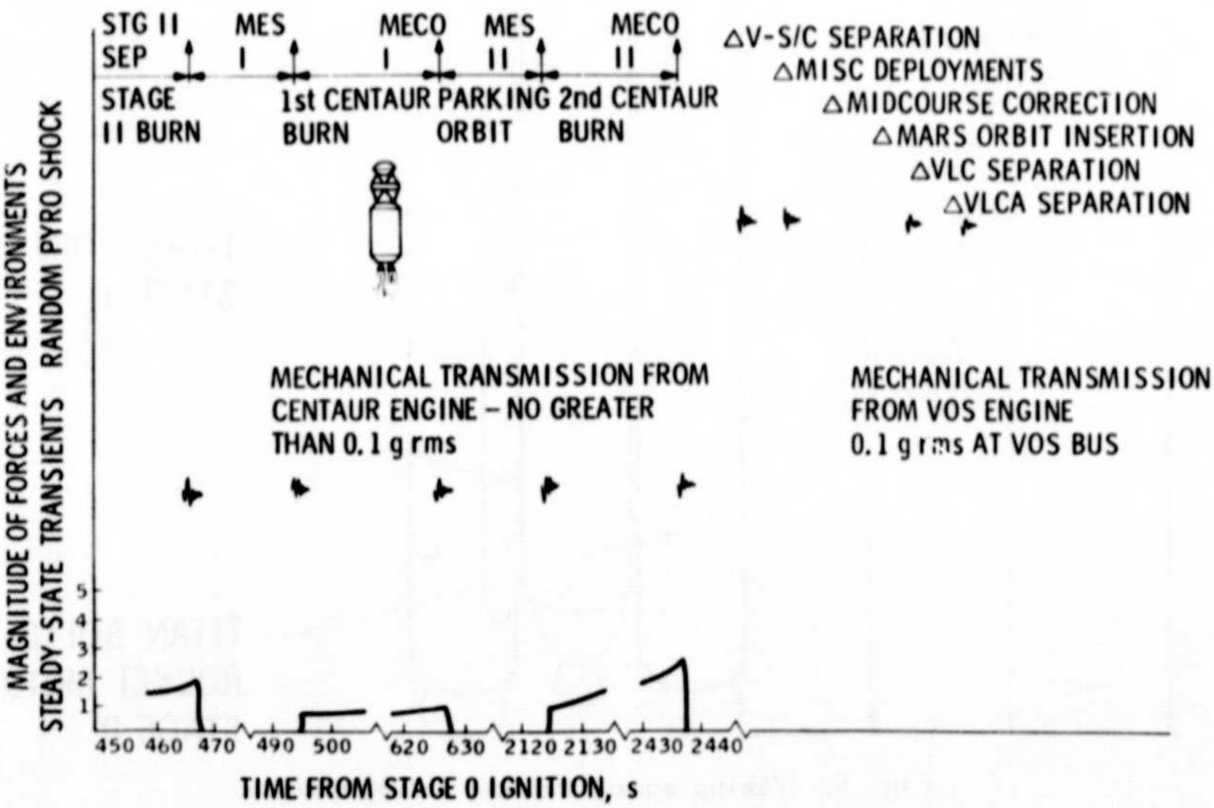


Fig. 6. Flight loads and environments

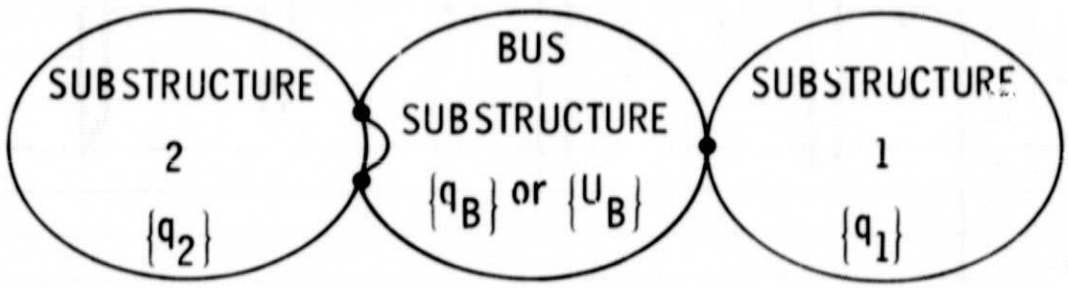


Fig. 7. Interconnections of substructure

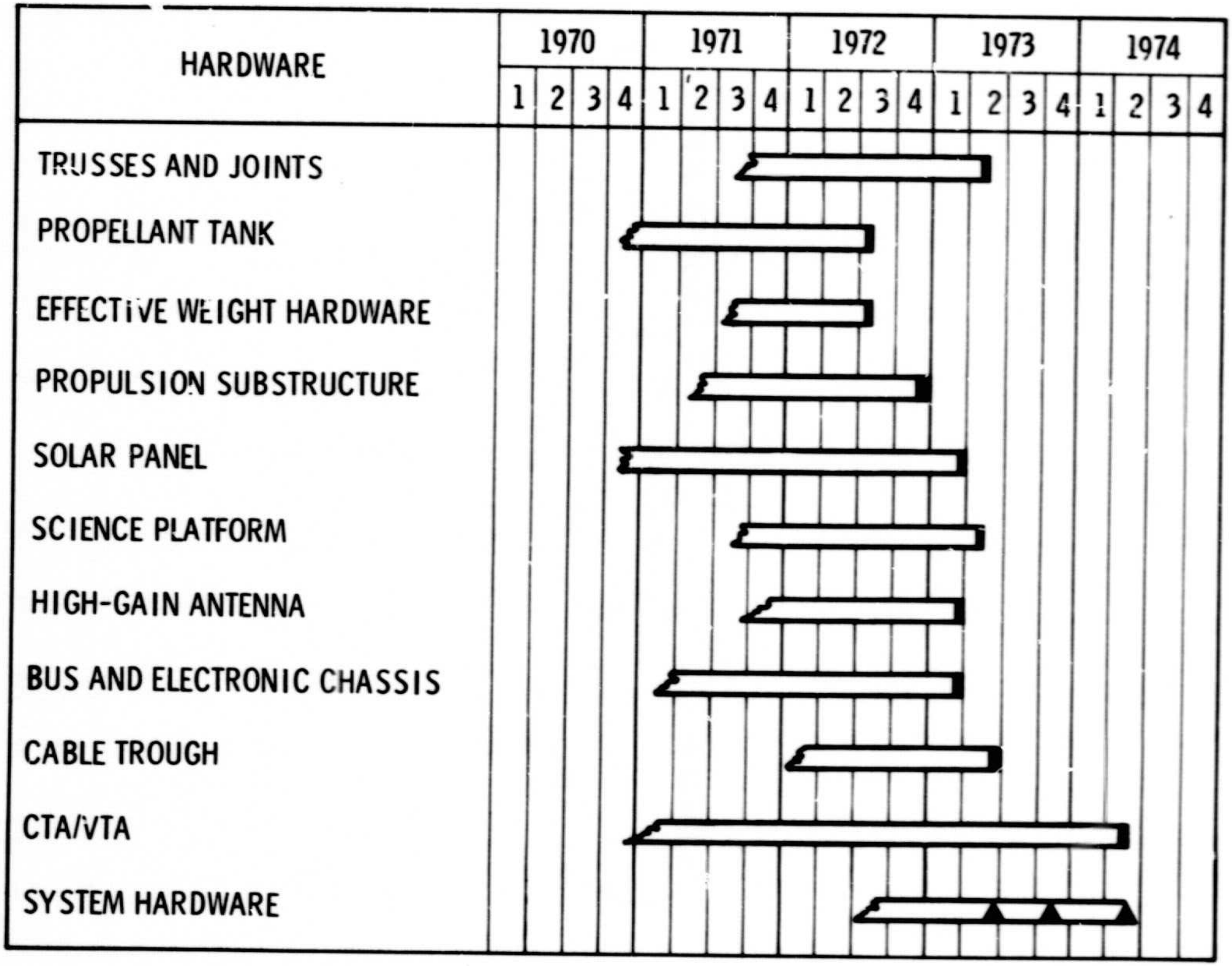


Fig. 8. Substructure and system schedule

HARDWARE	SECTION	DIVISION	EXTERNAL ORGANIZATION
TRUSSES AND JOINTS	352[a]		
PROPELLANT TANK	352	38	PRESSURE SYSTEMS, INC.
EFFECTIVE WEIGHT HARDWARE	354		
PROPULSION SUBSTRUCTURE	352	38	
SOLAR PANEL	352		PARSONS
SCIENCE PLATFORM	352		
HIGH-GAIN ANTENNA	352	33	PHILCO-FORD
BUS AND ELECTRONIC CHASSIS	352/357		
CABLE TROUGH	352/357		
CTA/VTA			GENERAL DYNAMICS/ CONVAIR ASTRONAUTIC
SYSTEM HARDWARE	352		

[a]THE NUMBERS DENOTE AN ORGANIZATION WITHIN JPL. WITHIN EACH SECTION, A COGNIZANT ENGINEER WAS ASSIGNED TO EACH OF THE HARDWARE ITEMS.

Fig. 9. Substructure and system hardware interface