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Solar Electric Generating System II Finite Element Analysis

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ABSTRACT

On June 2 1992, Landers' earthquake struck the Solar Electric Generating System II, located in Daggett, California. The 30 megawatt power station, operated by the Daggett Leasing Corporation (DLC), suffered substantial damage due to structural failures in the solar farm. These failures consisted of the separation of sliding joints supporting a distribution of parabolic glass mirrors. At separation, the mirrors fell to the ground and broke. It was the desire of the DLC and the Solar Thermal Design Assistance Center (STDAC) of Sandia National Laboratories (SNL) to redesign these joints so that, in the event of future quakes, costly breakage will be avoided.

To accomplish this task, drawings of collector components were developed by the STDAC, from which a detailed finite element computer model of a solar collector was produced. This nonlinear dynamic model, which consisted of over 8560 degrees of freedom, underwent model reduction to form a low order nonlinear dynamic model containing only 40 degrees of freedom. This model was then used as a design tool to estimate joint dynamics. Using this design tool, joint configurations were modified, and an acceptable joint redesign determined.

The results of this analysis showed that the implementation of metal stops welded to support shafts for the purpose of preventing joint separation is a suitable joint redesign. Moreover, it was found that, for quakes of Landers' magnitude, mirror breakage due to enhanced vibration in the trough assembly is unlikely.

1. BACKGROUND

This section describes the Solar Electric Generating System II (SEGS II) facility, the Landers' Earthquake, and the technical problem at hand.

1.1 The SEGS II Facility

The SEGS II facility consists of a farm of glass mirrored, single axis, parabolic trough solar collectors (see Figure 1). These collectors use solar energy to heat oil; which is used to produce steam; which, in turn, drives a turbine-generator to produce electricity.

A collector, shown in Figure 1, consists of a flexible truss structure supporting a distribution of mirrors. This flexible truss structure can be defined in terms of a number of substructures. *In this report*, the following terminology will be used to define these substructures. The tubular substructure which runs the length of the collector will be called the *torque tube.* The truss substructures between the mirrors and the torque tube will be called *mirror supports*. The flat plates attached at each end of the torque tube will be called *end plates,* and the tubular shafts projecting from these end plates will be called *support shafts*. The support shafts fit into journals which are attached to another set of substructures which will be called the *support trusses.* The support trusses transfer the weight of the entire collector to a concrete foundation. Vibration at this foundation can be transmitted through the support trusses, the journals, the support shafts, the end plates, the torque tube, and mirror supports to produce vibration in the mirrors. The end plates, torque tube, mirror supports, and mirrors will collectively comprise another substructure called the *trough assembly.*

In SEGS II, collectors are attached end to end to form rows which rotate about a north-south axis. The assembly of all collectors in all rows is referred to as the *solar farm.*

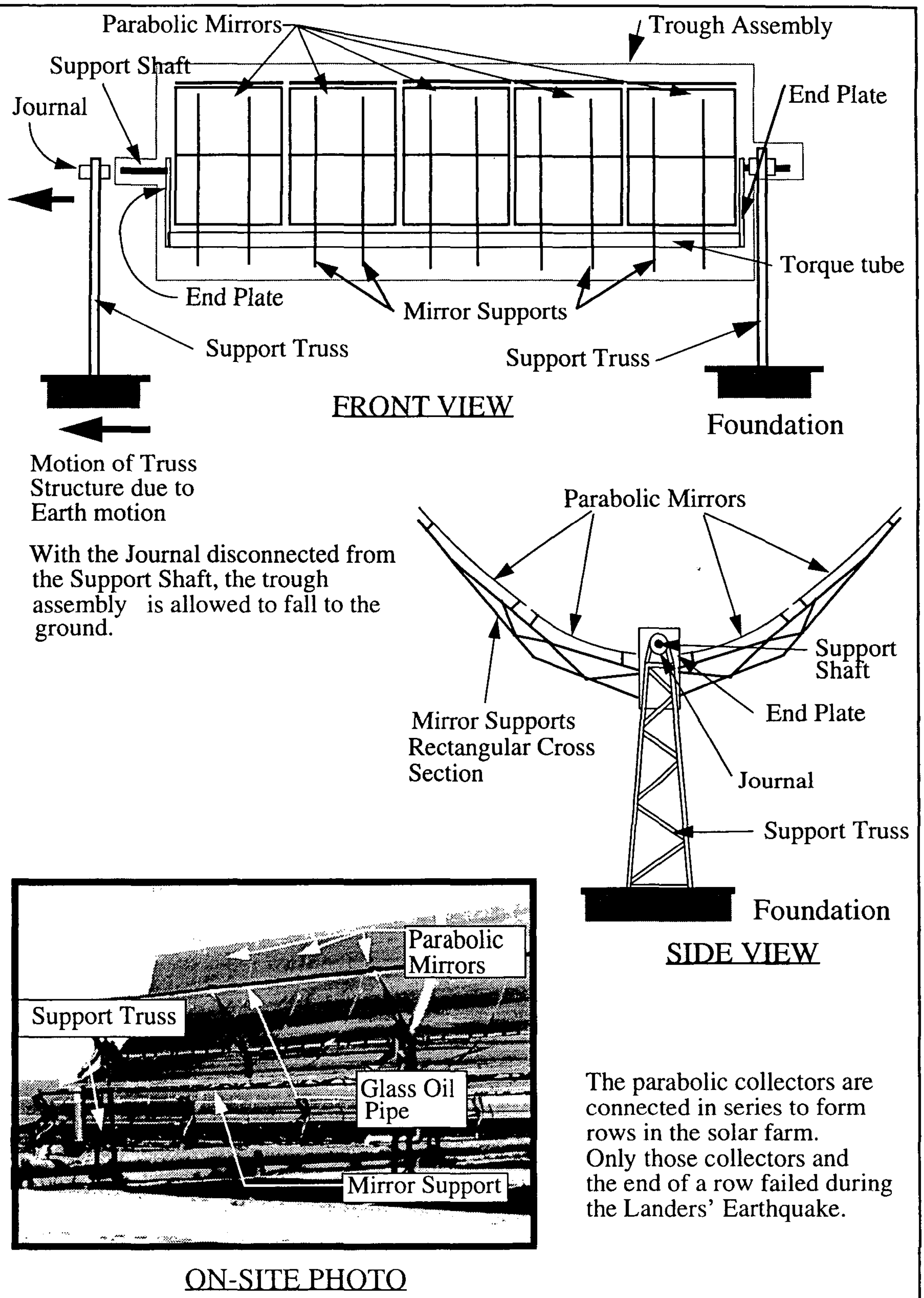

Figure 1. Drawing and Photo of Solar Collector.

1.2 The Landers' Earthquake

On June 2 of 1992, the Landers' earthquake struck the SEGS II facility in Dagget, California. The displacement of the quake was large enough such that some troughs separated from their support trusses allowing one end of the trough assembly to fall to the ground and shattered several mirrors of each fallen collector.

As shown in Figure 2,[1] the epicenter of this quake was locate near Landers, California - a small town approximated 50 miles south east of SEGS II. The magnitude of the quake was M7.5 (on the Richter scale). Thus, making it the largest quake to strike southern California in the last 40 years.[2] Approximately three and one half hours later, a second quake of magnitude M6.5 also struck SEGS II. Although its epicenter was closer, the

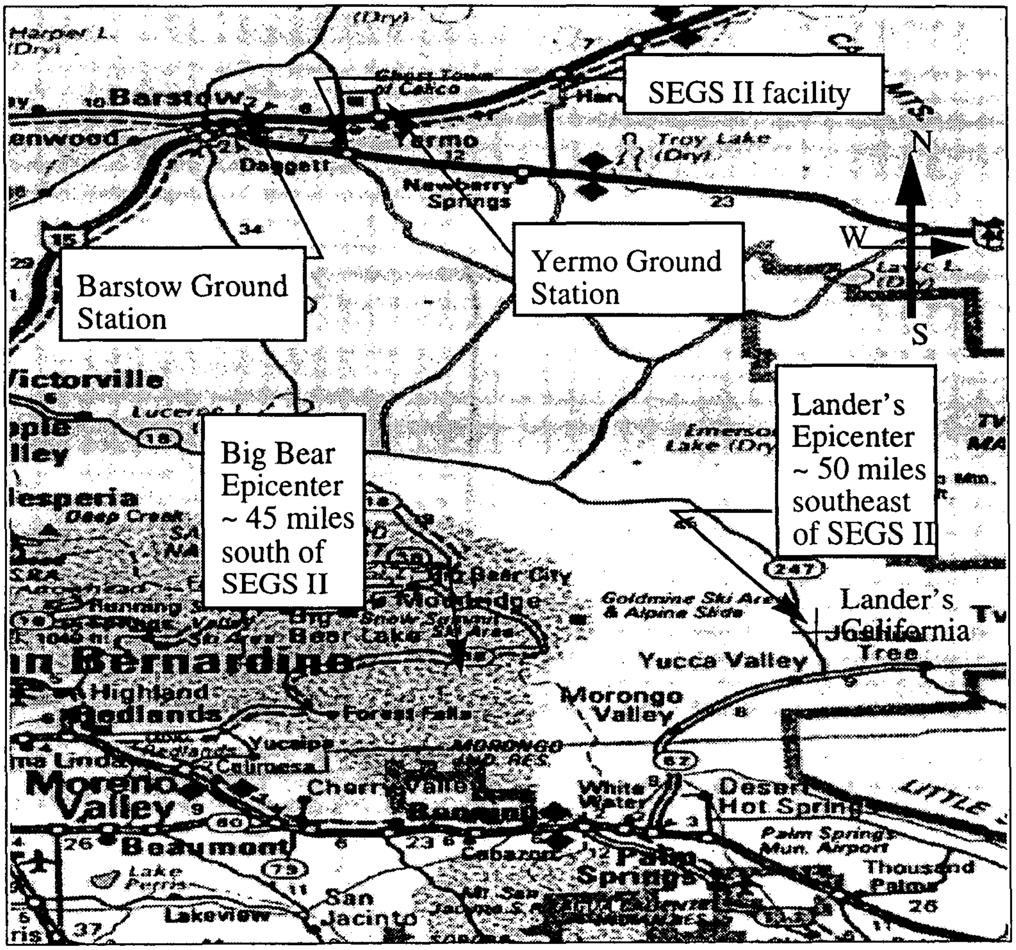

Figure 2. Map of Daggett, California Area.

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1. Rand McNally Road Atlas, 65th Edition, 1989
2. Ad Hoc Working Group, Future Seismic Hazards in Southern California, Phase I: Implications of the 1992 Landers' Earthquake Sequence, report by the California Division of Mines and Geology, P.O. Box 2980, Sacramento, CA 95812-2980

effects of this quake were minimal compared to the effects of the Landers' quake. The epicenter of this quake was near the small town of Big Bear Lake City - only 45 miles south of Daggett.

There are two ground-response stations[3] within 10 miles of Daggett. The first station, at Barstow, is located approximately eight miles to the west, and the second station, at Yermo, is located approximately four miles miles to the east. Due to their close proximity, it was assumed that earth motion at Dagget and Yermo was similar to earth motion at Barstow. Therefore, recorded Barstow data could be used as input to numerical simulations.

The Barstow data was limited in both frequency and direction. This data was band passed filtered to frequencies below 23Hz and above 0.07Hz. Three channels recorded acceleration in two perpendicular directions parallel to ground and in one direction perpendicular to ground. Rotations in these three directions where not measured, and therefore, without alternative, were assumed zero. From the acceleration data, displacement data was deduced. Figures 3a and 3b show recorded Barstow acceleration and deduced displacement for one channel.

As is shown in this data, the frequency response of acceleration is very high; however, the frequency response of displacement is low. Since both acceleration and displacement are excitations to the numerical problem (discussed below), and since the problem contained non-linear components (also discussed below), the frequency response in the mirrors could not be bounded without the use of a simulation. High frequencies creeping into mirror dynamics could break the mirrors.

1.3 Objective

After the Landers' earthquake, the Dagget Leasing Corporations (DLC), which operates SEGS II, contacted the Solar Thermal Design Assistance Center (STDAC) of Sandia National Laboratories (SNL) for assistance in determining a low cost solution to the journal separation problem. Personnel of DLC suggested welding stops at the end of each journal to prohibit the journals from separating from the support structure. STDAC engineers concurred that the stops would indeed prohibit separation but cautioned that the solution should be analyzed to ensure that this remedy would not cause subsequent failures. STDAC then contacted SNL's structural dynamics department and asked for their assistance in performing a Finite Element Analysis to determine the acceptability of the proposed solution.

3. A ground-response station is a location where seismic activity is recorded.

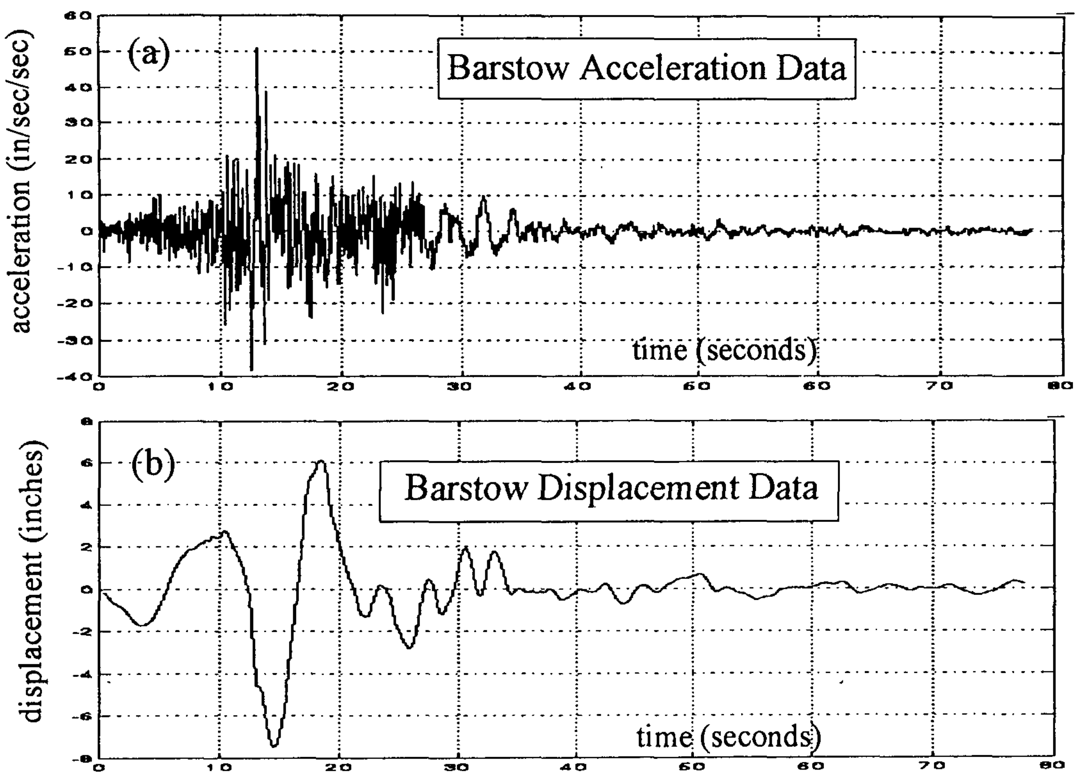

Figure 3. Barstow Data.

The following section describes the Finite Element (FE) model used in assessing acceptability.

2. FINITE ELEMENT (FE) MODEL DEVELOPMENT

There is a 60% probability that within the next thirty years an earthquake as large as or larger than the Landers' earthquake will strike southern California.[4] Therefore, the solar collectors, which failed in the Landers' earthquake, must be redesigned to survive this imminent geological threat.

The redesign process can be experimental, analytical or both. The disadvantage of using an experimental process is that experiments are relatively costly, whereas the advantage of an analytical process is that redesign costs are lower and design flexibility higher. Nevertheless, the disadvantage of using an analytical process is that when dynamics are complex, the accuracy of results unsupported by experiment is questionable. Therefore,

4. see Ref. 2 on page 5.

a process which uses both experiment and analysis is preferred. In this report, only an analytical approach is discussed.

The following section describes the Finite Element (FE) model used in assessing acceptability.

The redesign process consist of constructing a model of collector dynamics, simulating that model with recorded input data (the Barstow data), and varying modeled joint parameters to determine a set of viable joint configurations. The complexity and detail of the model is dependent on the characteristics of the excitation, and the capacity of the computational implementation relative to the complexity of dynamics.

In this redesign process, a FE model of a single solar collector was produced. This model was nonlinear and of high order. The difficulty with assimilating a high order nonlinear model into the joint redesign process is that such models require excessive central processing unit run times. Nevertheless, for redesign, the model must run quickly. This high order, nonlinear model was massaged into workable form via substructure modeling, model reduction, and substructure assembly. This resulted in a low order nonlinear model sufficient as a design tool.

2.1 Substructure Modeling

Substructure modeling is the dynamic modeling of an individual or a group of substructures. If a substructure model is linear, model reduction can be performed, and the reduced order substructure model can be combined with other substructure models to form a reduced order model of the total system. The process of recombining all substructure models will be called substructure assembly.

Collector dynamics were modeled using five separate substructure models. These models were:

<u>Left Support Truss Model</u> - a linear FE model representing left support truss dynamics,

<u>Left Joint Model</u> - a constraint model representing the dynamics of the joint between the left support truss and the trough assembly,

<u>Trough Assembly Model</u> - a linear FE model representing trough assembly dynamics,

<u>Right Joint Model</u> - a constraint model representing the dynamics of the joint between the right support truss and the trough assembly, and

<u>Right Support Truss Model</u> - a linear FE model representing right support truss dynamics.

A concise discussion of the production of each of these models is given below.

2.1.1 Left and Right Support Truss Models

Support trusses, composed of a distribution of welded angle iron, were modeled in PATRAN 3 as a collection of beams. Figure 4 shows a drawing of a support truss. Notice that this model does not include a journal. This is because journal dynamics are included into the left and right joint models. The left and right support trusses where identical in geometry and construction and therefore did not have to be modeled separately. Each support truss weighs about 200 lb_m, and each support truss model contained 378 Degrees of Freedom (DOF).

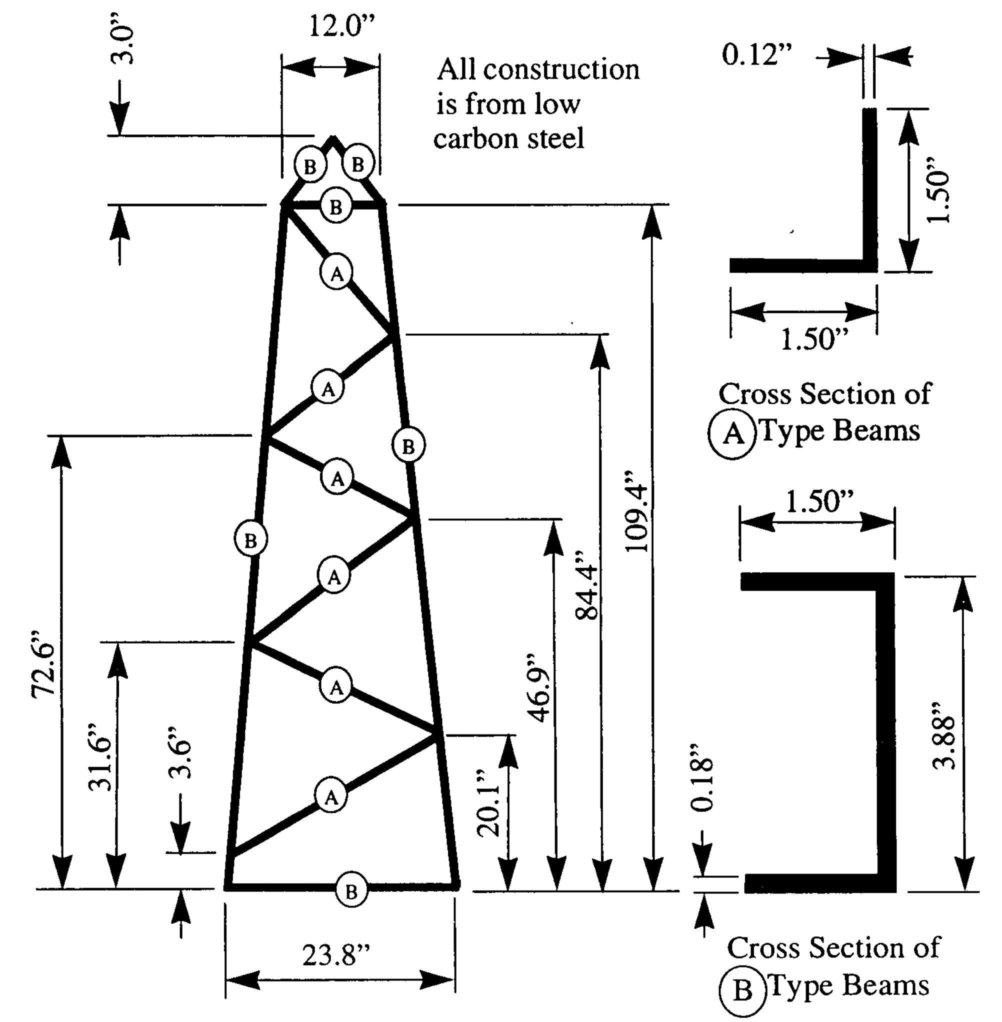

Figure 4. Geometry of Support Truss.

2.1.2 Trough Assembly Model

The trough assembly contains the end plates, the torque tube, the mirror supports, and the mirrors. This assembly was also modeled in PATRAN 3. The end plates and mirrors were modeled using thin shell elements, and the torque tube and mirror supports were modeled using beam elements. Two problems evolved from this modeling. First, since thin shell elements carry no normal angular rotation, it was not possible to model the transmission of normal torsional loads into the end plates via shell elements alone. Therefore, a collection of rigid bar constraints were attached to the plate elements to allow for torsional loading. Second, since the finite diameter torque tube was represented by a bar element of equivalent stiffness and mass but of infinitesimal diameter, massless rigid bar constraints had to be applied to couple torque tube dynamics to support truss dynamics.

Trough component dimensions are shown in Figure 5. The total mass of the trough assembly was 2160 lb_m. The trough assembly model consisted of 7812 DOF.

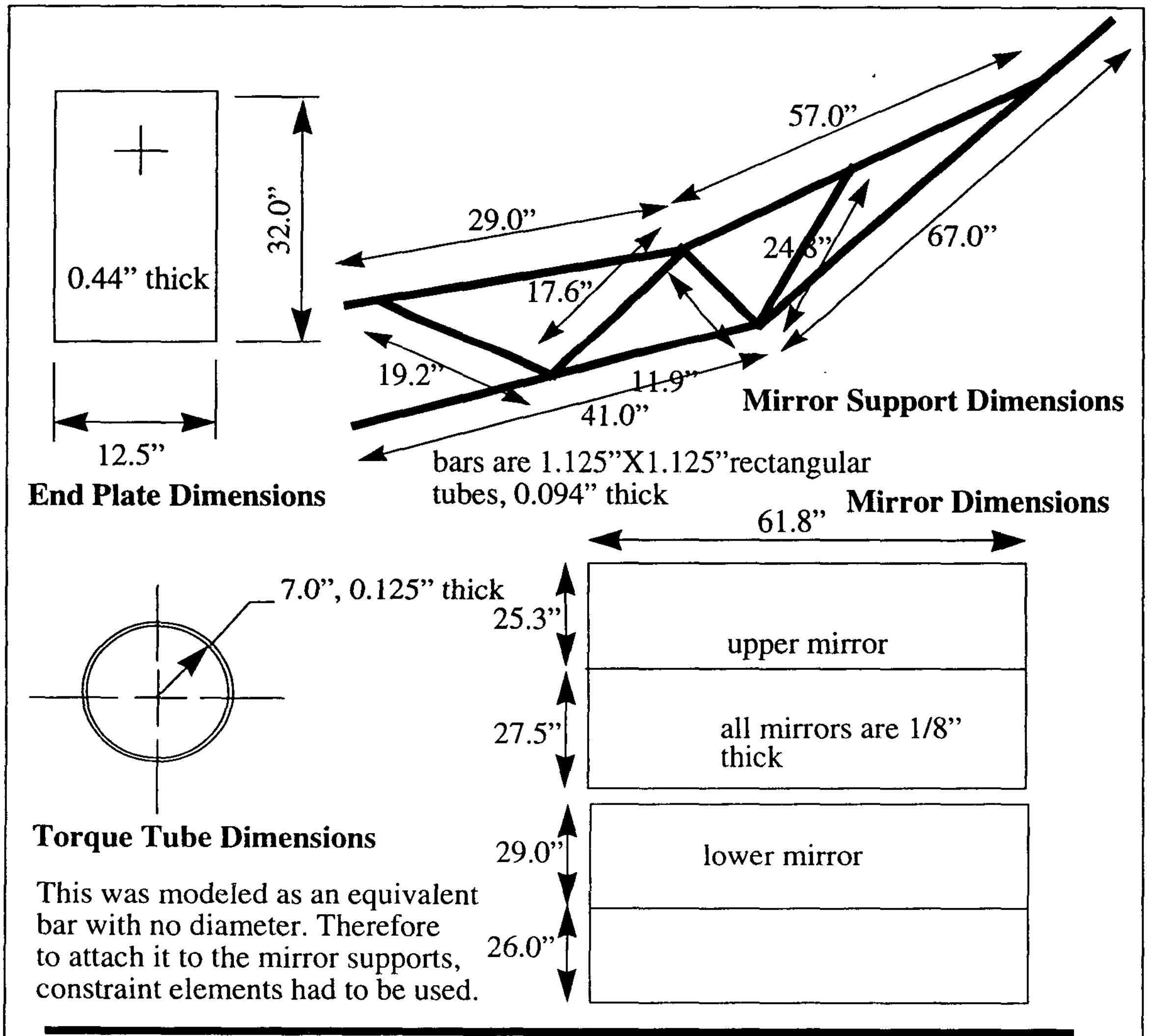

Figure 5. Geometry of Trough Assembly Components.

A NASTRAN modal analysis of the trough assembly model produced six rigid body modes and a closely spaced distribution of structural modes. The natural frequencies of the first seven structural modes were 9.94, 11.53, 12.01,16.39,16.41,16.53, and 16.81Hz. Due to high modal density, the accuracy of the model above 16Hz was questionable. This is because model accuracy is inversely related to modal density. As modal density increases, the ability for a deterministic FE model to represent reality accurately is questionable. Nevertheless, since the lower order modes were well spaced, and since these are the modes which were significant in determining joint dynamics, the model was deemed sufficient as a design tool.

Figure 6 shows the mode shape of the first structural mode at 9.94Hz. For sufficiently large amplitude excitations, this mode produces mirror collisions. If the mirrors collide, they will break. This mode represents a dominate mechanism of mirror collision.

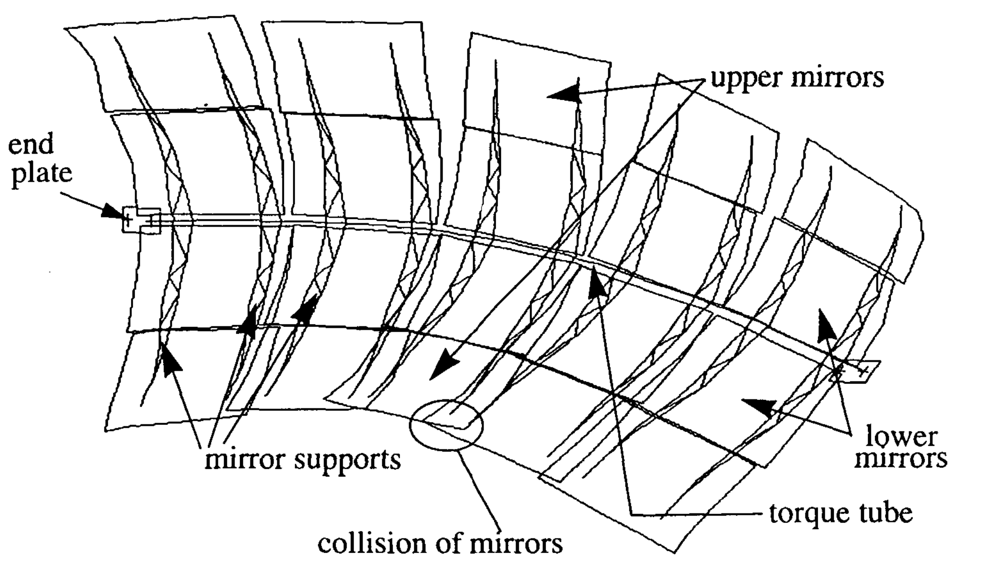

Figure 6. First Structural Model of Trough Assembly.

2.1.3 Left and Right Joint Models

Joints were modeled as mathematical constraints. Figure 7 shows the DOF and loads needed to define these constraints. The journal, modeled as a single point at the apex of the support truss, can rotate and displace in all directions. The DOF and the loads at the journal can be related to the DOF and the loads at the end plate attachment point. The attachment point is the location where the support shaft is connected to the end plate. The mathematical constraints between DOF and loads at the journal, and DOF and loads at the attachment point, comprise the joint model.

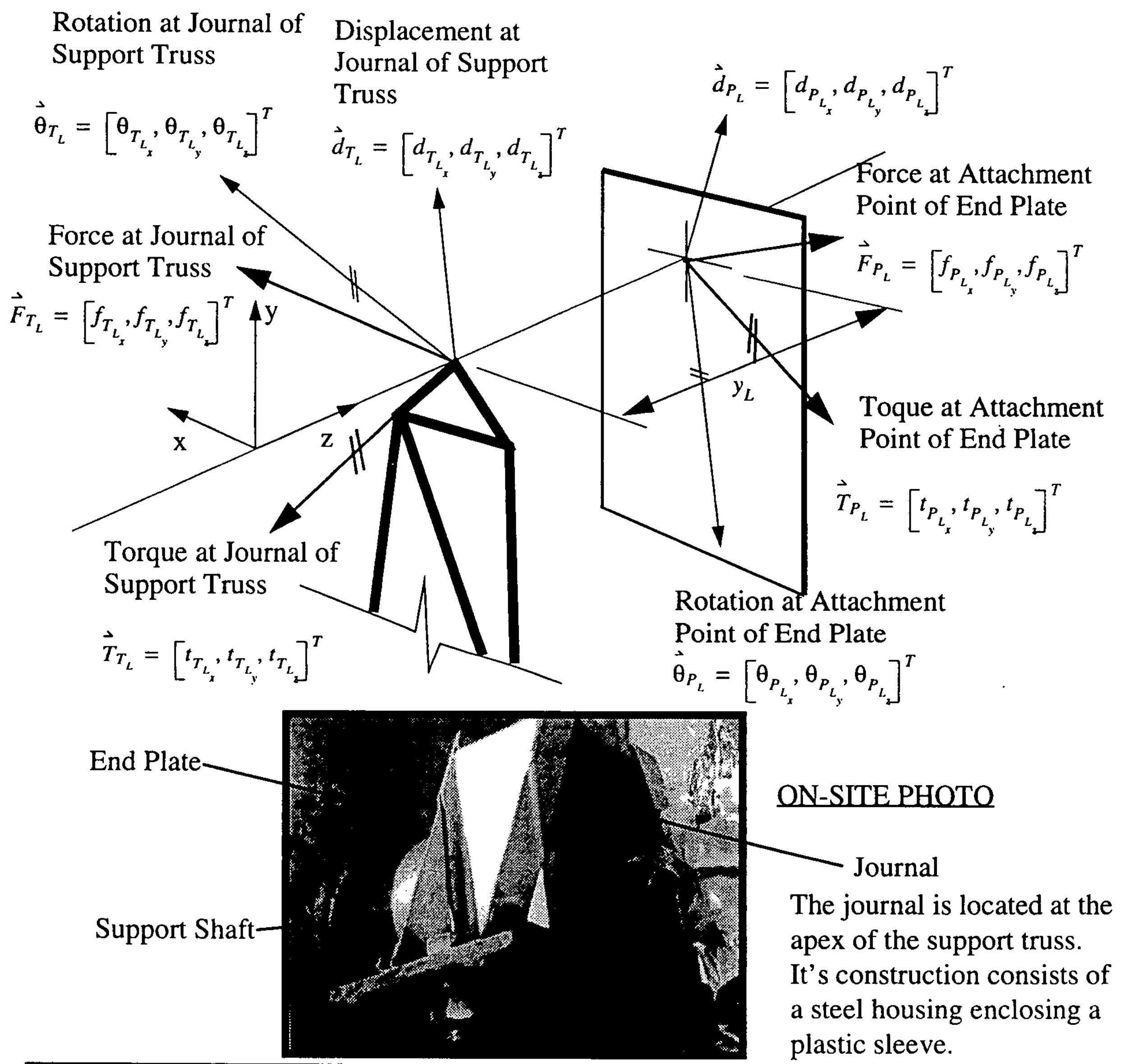

Figure 7. Joint Constraint Conditions for Left Joint.

Since rotations in the x and y directions are not significantly large, it was assumed that support shafts were always parallel to the z axis. Therefore, a joint can be modeled by equating displacements and rotations in the x and y directions, allowing rotations in z direction to be free, and relating displacements in the z direction via a nonlinear constraint equation representing journal/stop interactions. If the rotation, displacement, force, and torque at and on the journal of the left support truss is given by $\vec{\theta}_{T_L} = \left[\theta_{T_{L_x}}, \theta_{T_{L_y}}, \theta_{T_{L_z}}\right]^T$, $\vec{d}_{T_L} = \left[d_{T_{L_x}}, d_{T_{L_y}}, d_{T_{L_z}}\right]^T$, $\vec{F}_{T_L} = \left[f_{T_{L_x}}, f_{T_{L_y}}, f_{T_{L_z}}\right]^T$, and $\vec{T}_{T_L} = \left[t_{T_{L_x}}, t_{T_{L_y}}, t_{T_{L_z}}\right]^T$ respectively, and if the rotation, displacement, force, and torque at and on the attachment point of the left end plate is given by $\vec{\theta}_{P_L} = \left[\theta_{P_{L_x}}, \theta_{P_{L_y}}, \theta_{P_{L_z}}\right]^T$, $\vec{d}_{P_L} = \left[d_{P_{L_x}}, d_{P_{L_y}}, d_{P_{L_z}}\right]^T$ $\vec{F}_{P_L} = \left[f_{P_{L_x}}, f_{P_{L_y}}, f_{P_{L_z}}\right]^T$, and $\vec{T}_{P_L} = \left[t_{P_{L_x}}, t_{P_{L_y}}, t_{P_{L_z}}\right]^T$, and if the variable y_L is the z-displacement between the journal of the support truss and its attachment point, then the mathematical constraints describing the left joint are given by

$$\vec{X}_{T_{2_L}} = \vec{X}_{m_1} \ , \quad \vec{F}_{T_{2_L}} = -\vec{F}_{m_1} \ , \qquad \text{(Eq. 1)}$$

where

$$\vec{X}_{T_{2_L}} = \left[\theta_{T_{L_x}}, \theta_{T_{L_y}}, d_{T_{L_x}}, d_{T_{L_y}}\right]^T \ ,$$

$$\vec{X}_{m_1} = \left[\theta_{P_{L_x}}, \theta_{P_{L_y}}, d_{P_{L_x}}, d_{P_{L_y}}\right]^T \ ,$$

$$\vec{F}_{T_{2_L}} = \left[t_{T_{L_x}}, t_{T_{L_y}}, f_{T_{L_x}}, f_{T_{L_y}}\right]^T \ ,$$

$$\vec{F}_{m_1} = \left[t_{P_{L_x}}, t_{P_{L_y}}, f_{P_{L_x}}, f_{P_{L_y}}\right]^T \ ,$$

$$\vec{F}_{T_{3_L}} = \left[f_{T_{L_z}}\right] = -\left[f_{P_{L_z}}\right] = -\vec{F}_{m_3} = \sigma_L(y_L) + \delta_L(\dot{y}_L, y_L) \ . \qquad \text{(Eq. 2)}$$

The nonlinear functions $\sigma_L(y_L)$ and $\delta_L(\dot{y}_L, y_L)$ are force/displacement relationships which will be discussed below. The variables $\vec{F}_{m_1}$, $\vec{F}_{m_3}$, $\vec{F}_{T_{2_L}}$, $\vec{F}_{T_{3_L}}$, $\vec{X}_{T_{2_L}}$ and $\vec{X}_{m_1}$ are also defined above for later use. Again, since the left and right joints are dynamically identical, the constraint equations for the right joint are the same as those for the left joint except that the subscript L is replaced by the subscript R.

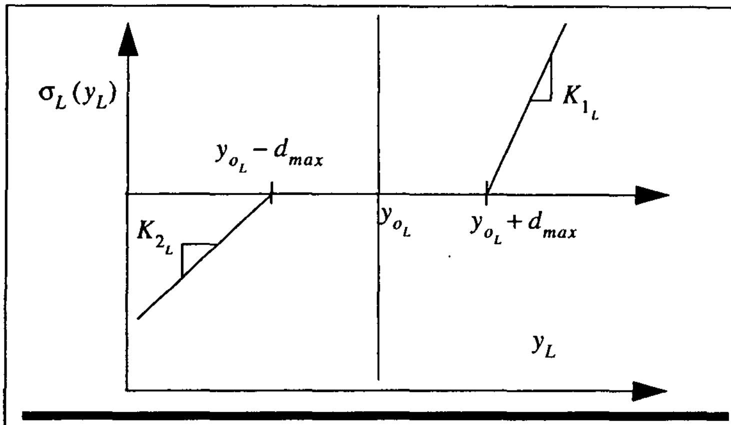

Figure 8. Nonlinear Joint Model.

The function $\sigma_L(y_L)$ is the force/displacement relationship for a journal sliding on a frictionless shaft whose motion is limited by stops at $y_{o_L} - d_{max}$ and $y_{o_L} + d_{max}$ where y_{o_L} is the mean distance between stops. When the journal is not in contact with a stop, no force is produced. When the journal is in contact with a stop, the support shaft elongates by δl. The force produced by this elongation is $K_{1_L}\delta l$ or $K_{2_L}\delta l$ where K_{1_L} is the contact stiffness for the stop at $y_{o_L} + d_{max}$ and K_{2_L} is the contact stiffness for the stop at $y_{o_L} - d_{max}$. These two stiffness are given by lE/A where l is the length of the support shaft in stress, E is Young's Modulus, and A is the cross sectional area of the support shaft. Strictly speaking, two stiffnesses are required since different lengths of the support shaft are stressed relative to which stop is in contact with the journal. Nevertheless, since these stiffnesses were similar, their values were assumed equal. A graphical representation of this force/displacement relationship is shown in Figure 8.

The function $\delta_L(\dot{y}_L, y_L)$ is the force/displacement relationship for stick/slip friction between the journal and the support shaft. This friction is not only dependent upon joint location, but also upon relative joint velocity.

When the relative velocity between the joint and shaft is zero, friction is equal to that required to resist slip. Numerically this can be approximated via a very stiff spring (in the infinite limit the approximation is exact). When friction exceeds the coefficient of static friction, μ_s, times supported weight, W, its value is reduced to the coefficient of kinetic friction, μ_k, times supported weight. Therefore, its magnitude is always bounded. A block diagram representation of these dynamics is shown in Figure 9.

Friction in the left and right joints is different considering that the weight that each support truss carries is different. If the modeled solar collector is attached at the end of a row of collectors, the weight that the left support truss carries will be half the weight of the modeled trough assemble, whereas the weight which the right support truss carries will be half the weight of the modeled trough assemble plus half the weight of the trough assemble of the next collector in the row.

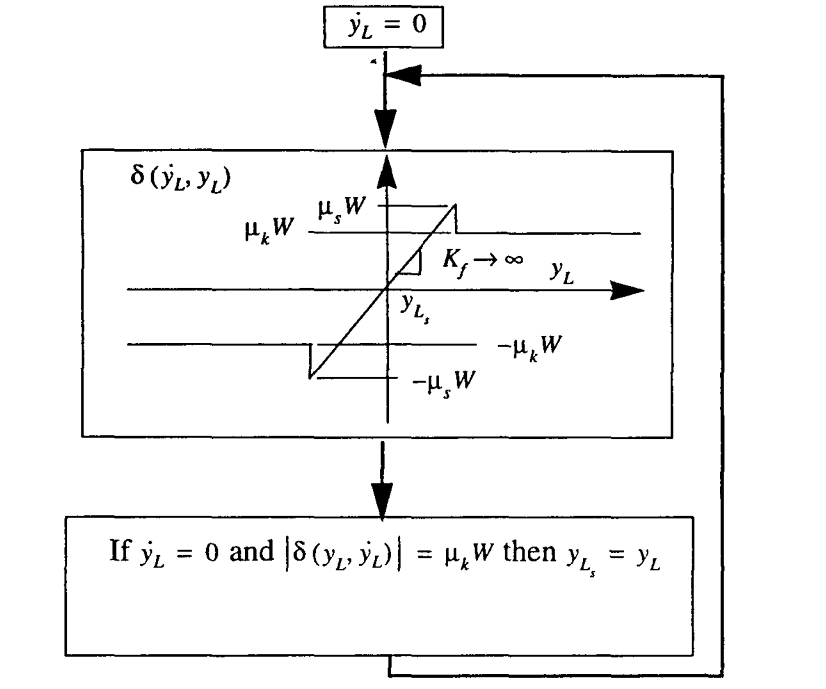

Figure 9. Block Diagram of Stick/Slip Frictional Loading.

The coefficients of static and kinetic friction at the time of the Landers' quake are unknown. Therefore, μ_s and μ_k can only be approximated. The coefficient of static friction was determined from the coefficient of kinetic friction via the assumed relationship, $\mu_s = 1.5\mu_k$. This relationship was made for lack of a better alternative. The coefficient of kinetic friction was determined via numerical iteration. If μ_k is large, there will be no relative motion between the journal and the support shaft. As μ_k is decreased, more motion will occur. At the time of the Lander's quake, the relative motion between the journal and support shaft had to be at least 4" for the joint to fail. Therefore, μ_k was decreased until 4" of relative motion occurred. This represented the maximum possible amount of joint friction under assumed conditions.

2.2 FE Model Reduction

Without model reduction, the number of DOF in the unified model would be excessive. A non-reduced assembled model would contain over 8560 DOF. A model of this order, using direct simulation,[5] will require excessive cpu time. Thus, to reduce cpu run

5. Direct simulation means that all DOF are calculated during numerical integration.

time, model reduction was performed. The preferred method of model reduction is Craig-Bampton.[7] A brief overview of this method is found in Appendix A.

Numerically, this reduction was performed in NASTRAN via D-mapping and the results transferred into MATLAB via the NASMAT translator. Numerical simulations were performed in MATLAB.

2.3 Substructure Assembly

Three FE substructure models underwent model reduction. These models were the left support truss model, the right support truss model, and the trough assembly model.

The reduced second order matrix equation for the dynamics of the left support truss is given by

$$\begin{bmatrix} m_{T_{11}} & m_{T_{12}} & m_{T_{13}} & m_{T_{14}} \\ m_{T_{21}} & m_{T_{22}} & m_{T_{23}} & m_{T_{24}} \\ m_{T_{31}} & m_{T_{32}} & m_{T_{33}} & m_{T_{34}} \\ m_{T_{41}} & m_{T_{42}} & m_{T_{43}} & m_{T_{44}} \end{bmatrix} \begin{bmatrix} \vec{x}_{T_{1_L}} \\ \vec{x}_{T_{2_L}} \\ \vec{x}_{T_{3_L}} \\ \vec{x}_{T_{4_L}} \end{bmatrix} + \begin{bmatrix} k_{T_{11}} & k_{T_{12}} & k_{T_{13}} & k_{T_{14}} \\ k_{T_{21}} & k_{T_{22}} & k_{T_{23}} & k_{T_{24}} \\ k_{T_{31}} & k_{T_{32}} & k_{T_{33}} & k_{T_{34}} \\ k_{T_{41}} & k_{T_{42}} & k_{T_{43}} & k_{T_{44}} \end{bmatrix} \begin{bmatrix} \vec{x}_{T_{1_L}} \\ \vec{x}_{T_{2_L}} \\ \vec{x}_{T_{3_L}} \\ \vec{x}_{T_{4_L}} \end{bmatrix} = \begin{bmatrix} \vec{F}_{T_{1_L}} \\ \vec{F}_{T_{2_L}} \\ \vec{F}_{T_{3_L}} \\ 0 \end{bmatrix}$$

where

$\vec{x}_{T_{1_L}}$ - are DOF in the left support truss model representing displacements and accelerations at the concrete foundation,

$\vec{x}_{T_{2_L}}$ - are DOF in the left support truss model to be equated to DOF in the trough assembly model (Equation 1),

$\vec{x}_{T_{3_L}}$ - are DOF in the left support truss model to be coupled to DOF in the trough assembly model via the nonlinear functional relationships $\sigma_L(y_L)$ and $\sigma_L(y_L, \dot{y}_L)$ (Equation 2),

$\vec{x}_{T_{4_L}}$ - are generalized DOF in the left support truss model which can be used for model reduction.

The subscript L represents the left truss. If the subscript L is replaced by the subscript R the equations for the dynamics of the right truss result.

7. R.R. Craig, Jr, Structural Dynamics, *An Introduction to Computer Methods,* John Wiley & Sons, New York, NY, 1981, pp. 475-478.

Since $\vec{X}_{T_{1_L}}$ and $\vec{X}_{T_{1_L}}$ are known inputs (the Barstow measurements), the above equation can be written as

$$\begin{bmatrix} m_{T_{22}} & m_{T_{23}} & m_{T_{24}} \\ m_{T_{32}} & m_{T_{33}} & m_{T_{34}} \\ m_{T_{42}} & m_{T_{43}} & m_{T_{44}} \end{bmatrix} \begin{bmatrix} \vec{X}_{T_{2_L}} \\ \vec{X}_{T_{3_L}} \\ \vec{X}_{T_{4_L}} \end{bmatrix} + \begin{bmatrix} k_{T_{22}} & k_{T_{23}} & k_{T_{24}} \\ k_{T_{32}} & k_{T_{33}} & k_{T_{34}} \\ k_{T_{42}} & k_{T_{43}} & k_{T_{44}} \end{bmatrix} \begin{bmatrix} \vec{X}_{T_{2_L}} \\ \vec{X}_{T_{3_L}} \\ \vec{X}_{T_{4_L}} \end{bmatrix} = \begin{bmatrix} \vec{F}_{T_{2_L}} \\ \vec{F}_{T_{3_L}} \\ 0 \end{bmatrix} - \begin{bmatrix} m_{T_{21}} & k_{T_{21}} \\ m_{T_{31}} & k_{T_{31}} \\ m_{T_{41}} & k_{T_{41}} \end{bmatrix} \begin{bmatrix} \vec{X}_{T_{1_L}} \\ \vec{X}_{T_{1_L}} \end{bmatrix} \qquad \text{(Eq. 3)}$$

where all forcing functions are now on the right hand side.

The reduced second order matrix equation for the dynamics of the trough assemble is given by

$$\begin{bmatrix} m_{m_{11}} & m_{m_{12}} & m_{m_{13}} & m_{m_{14}} & m_{m_{15}} \\ m_{m_{21}} & m_{m_{22}} & m_{m_{23}} & m_{m_{24}} & m_{m_{25}} \\ m_{m_{31}} & m_{m_{32}} & m_{m_{33}} & m_{m_{34}} & m_{m_{35}} \\ m_{m_{41}} & m_{m_{42}} & m_{m_{43}} & m_{m_{44}} & m_{m_{45}} \\ m_{m_{51}} & m_{m_{52}} & m_{m_{53}} & m_{m_{54}} & m_{m_{55}} \end{bmatrix} \begin{bmatrix} \vec{X}_{m_1} \\ \vec{X}_{m_2} \\ \vec{X}_{m_3} \\ \vec{X}_{m_4} \\ \vec{X}_{m_5} \end{bmatrix} + \begin{bmatrix} k_{m_{11}} & k_{m_{12}} & k_{m_{13}} & k_{m_{14}} & k_{m_{15}} \\ k_{m_{21}} & k_{m_{22}} & k_{m_{23}} & k_{m_{24}} & k_{m_{25}} \\ k_{m_{31}} & k_{m_{32}} & k_{m_{33}} & k_{m_{34}} & k_{m_{35}} \\ k_{m_{41}} & k_{m_{42}} & k_{m_{43}} & k_{m_{44}} & k_{m_{45}} \\ k_{m_{51}} & k_{m_{52}} & k_{m_{53}} & k_{m_{54}} & k_{m_{55}} \end{bmatrix} \begin{bmatrix} \vec{X}_{m_1} \\ \vec{X}_{m_2} \\ \vec{X}_{m_3} \\ \vec{X}_{m_4} \\ \vec{X}_{m_5} \end{bmatrix} = \begin{bmatrix} \vec{F}_{m_1} \\ \vec{F}_{m_2} \\ \vec{F}_{m_3} \\ \vec{F}_{m_4} \\ 0 \end{bmatrix} \qquad \text{(Eq. 4)}$$

where

- $\vec{X}_{m_1}$ - are DOF in the trough assembly model to be equated to DOF in the left support truss model,
- $\vec{X}_{m_2}$ - are DOF in the trough assembly model to be equated to DOF in the right support truss model,
- $\vec{X}_{m_3}$ - are DOF in the trough assembly model to be coupled to DOF in the left support model by the nonlinear functions $\sigma_L\,(y_L)$ and $\sigma_L(y_L,\, y_L)$,
- $\vec{X}_{m_4}$ - are DOF in the trough assembly model to be coupled to DOF in the right support model by the nonlinear function $\sigma_R\,(y_R)$ and $\sigma_R\,(y_R,\, y_R)$,
- $\vec{X}_{m_5}$ - are generalized DOF.

To couple equation (3) to equation (4), equations (1, 2, 3) are used. This results in a combined relation as follows.

$$
\begin{bmatrix}
m_{m_{11}}+m_{T_{22}} & m_{m_{12}} & m_{m_{13}} & m_{m_{14}} & m_{m_{15}} & m_{T_{23}} & m_{T_{24}} & 0 & 0 \\
m_{m_{21}} & m_{m_{22}}+m_{T_{22}} & m_{m_{23}} & m_{m_{24}} & m_{m_{25}} & 0 & 0 & m_{T_{23}} & m_{T_{24}} \\
m_{m_{31}} & m_{m_{32}} & m_{m_{33}} & m_{m_{34}} & m_{m_{35}} & 0 & 0 & 0 & 0 \\
m_{m_{41}} & m_{m_{42}} & m_{m_{43}} & m_{m_{44}} & m_{m_{45}} & 0 & 0 & 0 & 0 \\
m_{m_{51}} & m_{m_{52}} & m_{m_{53}} & m_{m_{54}} & m_{m_{55}} & 0 & 0 & 0 & 0 \\
m_{T_{32}} & 0 & 0 & 0 & 0 & m_{T_{33}} & m_{T_{34}} & 0 & 0 \\
m_{T_{42}} & 0 & 0 & 0 & 0 & m_{T_{43}} & m_{T_{44}} & 0 & 0 \\
0 & m_{T_{32}} & 0 & 0 & 0 & 0 & 0 & m_{T_{33}} & m_{T_{34}} \\
0 & m_{T_{42}} & 0 & 0 & 0 & 0 & 0 & m_{T_{43}} & m_{T_{44}}
\end{bmatrix}
\begin{bmatrix}
\vec{X}_{m_1} \\ \vec{X}_{m_2} \\ \vec{X}_{m_3} \\ \vec{X}_{m_4} \\ \vec{X}_{m_5} \\ \vec{X}_{T_{3_L}} \\ \vec{X}_{T_{4_L}} \\ \vec{X}_{T_{3_R}} \\ \vec{X}_{T_{4_R}}
\end{bmatrix}
$$

$$
+\begin{bmatrix}
k_{m_{11}}+k_{T_{22}} & k_{m_{12}} & k_{m_{13}} & k_{m_{14}} & k_{m_{15}} & k_{T_{23}} & k_{T_{24}} & 0 & 0 \\
k_{m_{21}} & k_{m_{22}}+k_{T_{22}} & k_{m_{23}} & k_{m_{24}} & k_{m_{25}} & 0 & 0 & k_{T_{23}} & k_{T_{24}} \\
k_{m_{31}} & k_{m_{32}} & k_{m_{33}} & k_{m_{34}} & k_{m_{35}} & 0 & 0 & 0 & 0 \\
k_{m_{41}} & k_{m_{42}} & k_{m_{43}} & k_{m_{44}} & k_{m_{45}} & 0 & 0 & 0 & 0 \\
k_{m_{51}} & k_{m_{52}} & k_{m_{53}} & k_{m_{54}} & k_{m_{55}} & 0 & 0 & 0 & 0 \\
k_{T_{32}} & 0 & 0 & 0 & 0 & k_{T_{33}} & k_{T_{34}} & 0 & 0 \\
k_{T_{42}} & 0 & 0 & 0 & 0 & k_{T_{43}} & k_{T_{44}} & 0 & 0 \\
0 & k_{T_{32}} & 0 & 0 & 0 & 0 & 0 & k_{T_{33}} & k_{T_{34}} \\
0 & k_{T_{42}} & 0 & 0 & 0 & 0 & 0 & k_{T_{43}} & k_{T_{44}}
\end{bmatrix}
\begin{bmatrix}
\vec{X}_{m_1} \\ \vec{X}_{m_2} \\ \vec{X}_{m_3} \\ \vec{X}_{m_4} \\ \vec{X}_{m_5} \\ \vec{X}_{T_{3_L}} \\ \vec{X}_{T_{4_L}} \\ \vec{X}_{T_{3_R}} \\ \vec{X}_{T_{4_R}}
\end{bmatrix}
=
$$

$$
\begin{bmatrix}
-m_{T_{21}} & -k_{T_{21}} & 0 & 0 \\
0 & 0 & -m_{T_{21}} & -k_{T_{21}} \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
-m_{T_{31}} & -k_{T_{31}} & 0 & 0 \\
-m_{T_{41}} & -k_{T_{41}} & 0 & 0 \\
0 & 0 & -m_{T_{31}} & -k_{T_{31}} \\
0 & 0 & -m_{T_{41}} & -k_{T_{41}}
\end{bmatrix}
\begin{bmatrix}
\vec{X}_{T_{1_L}} \\ \vec{X}_{T_{1_L}} \\ \vec{X}_{T_{1_R}} \\ \vec{X}_{T_{1_R}}
\end{bmatrix}
+
\begin{bmatrix}
0 & 0 \\ 0 & 0 \\ -I & 0 \\ 0 & -I \\ 0 & 0 \\ I & 0 \\ 0 & I \\ 0 & 0 \\ 0 & 0
\end{bmatrix}
\begin{bmatrix}
\vec{F}_{T_{3_L}} \\ \vec{F}_{T_{3_R}}
\end{bmatrix}
\qquad \text{(Eq. 5)}
$$

$$y = \begin{bmatrix} y_L \\ y_R \end{bmatrix} = \begin{bmatrix} 0 & 0 & I & 0 & 0 & -I & 0 & 0 & 0 \\ 0 & 0 & 0 & I & 0 & 0 & 0 & -I & 0 \end{bmatrix} \vec{X} = Q\vec{X} \quad , \qquad \text{(Eq. 6)}$$

$$\vec{X} = \begin{bmatrix} \vec{X}_{m_1}^T & \vec{X}_{m_2}^T & \vec{X}_{m_3}^T & \vec{X}_{m_4}^T & \vec{X}_{m_5}^T & \vec{X}_{T_{3_L}}^T & \vec{X}_{T_{4_L}}^T & \vec{X}_{T_{3_R}}^T & \vec{X}_{T_{4_R}}^T \end{bmatrix}^T$$

where for this problem, $\vec{X}_{m_3}^T, \vec{X}_{m_4}^T, \vec{X}_{T_{3_L}}^T, \vec{X}_{T_{3_R}}^T, \vec{F}_{T_{3_L}}, \vec{F}_{T_{3_R}}, I \in R^{1x1}$ and the 0s take appropriate dimensions.

Written in condensed notation, equations (5) and (6) become $M\ddot{\vec{X}} + K\vec{X} = R\vec{u} + D\vec{F}$ and $y = Q\vec{X}$ where $\vec{u} = \begin{bmatrix} \vec{X}_{T_{1_L}}^T & \vec{X}_{T_{1_L}}^T & \vec{X}_{T_{1_R}}^T & \vec{X}_{T_{1_R}}^T \end{bmatrix}^T$ and $\vec{F} = \begin{bmatrix} \vec{F}_{T_{3_L}}^T & \vec{F}_{T_{3_R}}^T \end{bmatrix}^T = \begin{bmatrix} f_{T_{L_2}} & f_{T_{R_2}} \end{bmatrix}^T$. This can be collapsed into state space form as $\dot{\vec{x}} = A\vec{x} + B_u\vec{u} + B_F\vec{F}$, $y = C\vec{x}$, and $\dot{y} = C_v\vec{x}$ where $\vec{x} = \begin{bmatrix} \vec{X} \\ \dot{\vec{X}} \end{bmatrix}$, $A = \begin{bmatrix} 0 & I \\ -M^{-1}K & 0 \end{bmatrix}$, $B_u = \begin{bmatrix} 0 \\ M^{-1}R \end{bmatrix}$, $B_F = \begin{bmatrix} 0 \\ M^{-1}D \end{bmatrix}$, $C = \begin{bmatrix} Q & 0 \end{bmatrix}$, $C_v = \begin{bmatrix} 0 & Q \end{bmatrix}$ and again the 0s take appropriate dimensions. This state space relationship includes all the dynamics of the linear portion of the system.

Nonlinear joint dynamics,

$$\vec{F} = \begin{bmatrix} f_{T_{2_L}} \\ f_{T_{2_R}} \end{bmatrix} = \begin{bmatrix} \sigma_L(y_L) \\ \sigma_R(y_R) \end{bmatrix} + \begin{bmatrix} \delta_L(y_L, \dot{y}_L) \\ \delta_R(y_R, \dot{y}_R) \end{bmatrix} = K(\vec{y})\vec{y} + \vec{R}$$

can be implemented using joint constraint conditions given in Figures 9 and 10.

The assembled collector model can be drawn in block diagram form as shown in Figure 11.

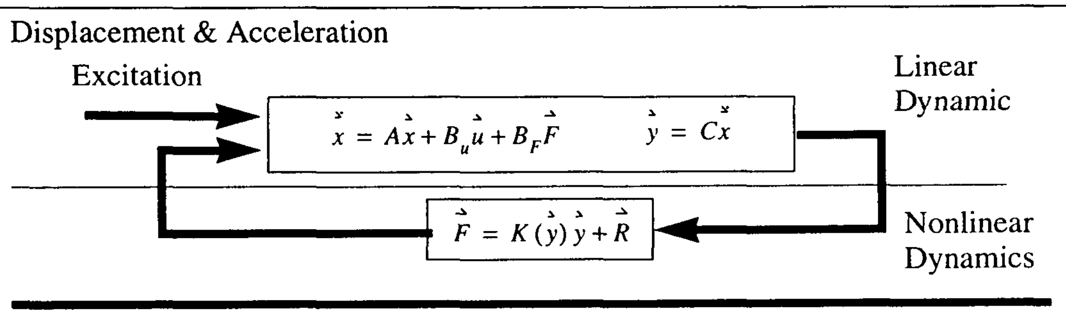

Figure 11. Nonlinear Collector Model in Block Diagram Form.

As shown in this diagram, nonlinear dynamics and linear dynamics can be separated into feedback and feedforward blocks. As discussed above, model reduction was used to reduce the order of all linear dynamics. Nonlinear dyanmics were already of low order. Thus, the assembled model was of low order. The assembled model is given by

$$\dot{\vec{x}} = (A + B_F K(\vec{y})\,(C))\,\vec{x} + B_u\vec{u} + B_F\vec{R}\,. \qquad \text{(Eq. 7)}$$

During simulations, equation (7) was updated every time step.

Also during simulations, variable time steps were used. Variability in time step was required since journal/stop impacts produced high frequency dynamics which could throw the numerical simulation into instability. When the journals and stops were far from contact, long time steps were used, and when the journal and stops were in or near contact, short time steps were used. Stability was also constrained by the stiffnesses used to impose stop conditions. The proper selection of these stiffnesses could only be determined by iteration.

3. RESULTS

The section below presents results on model reduction errors and collector dynamics.

3.1 *Model Reduction Error*

The assembled model was only 40th order. The natural frequencies of each reduced order models are given in Table 1.

	Support Truss Model			Trough Assembly Model		
Mode	Full order	Reduced Order	% Error	Full Order	Reduced Order	% Error
1	11.01 Hz	11.01 Hz	0.08%	9.94 Hz	10.17 Hz	2.32%
2	28.44 Hz	29.46 Hz	3.61%	11.53 Hz	11.55 Hz	0.12%
3	60.67 Hz	63.77 Hz	5.11%	12.01 Hz	12.38 Hz	3.10%
4	70.80 Hz	71.12 Hz	0.46%	16.39 Hz	15.88 Hz	3.16%
5	111.12 Hz	444.66 Hz	300.16%	16.41 Hz	16.31 Hz	0.61%
6	127.62 Hz	659.66 Hz	416.89%	16. 53 Hz	16.39 Hz	0.85%

Table 1. Model Error.

As shown in Table 1, Craig-Bampton matches lower order modes to within 4%, whereas higher order modes were in substantial error. Since excitation frequencies were low (most of the structural energy was below 1Hz), accuracy was needed only at these lower order modes.

3.2 Collector Dynamics

Two joint configurations were studied. The first configuration lets d_{max} go to infinity (see Figure 9). This represents a Support Shaft with No Stops. This was the joint configuration during the Lander's quake. The second configuration assumes that d_{max} =1". This represents a Support Shaft with Stops at 1". This is the remedy proposed and implemented by the DLC. It was found that this was a sufficient remedy to the joint separation problem.

3.2.1 Support Shaft with No Stops

The first configuration lets d_{max} go to infinity. This is a model of the joint as it existed during the Lander's quake. Figures 12 (a), (b), and (c) show collector response versus time for this joint configuration. Input excitation came from measured Barstow data.

In Figure 12(a), the distance between the center of the support shaft and the journal is shown. If the distance between the inner end of the journal and the outer end of the support shaft is less than 4", the journal and support shaft will separate. In this configuration, friction was not great enough to stop joint failure. Figure 12(c) shows relative displacement between mirror corners. This relative displacement was measured at the location of the largest relative displacement in the Figure 7 mode shape. To assure no mirror breakage, this displacement should be less than one inch. As shown in this plot, mirror breakage due to collision of mirrors will not occur.

3.2.2 Support Shaft with Stops at 1"

This joint configuration consists of a stop welded to the end of the support shaft which limits its travel to a maximum of one inch. Figures 13(a), (b), and (c) show responses for this configuration.

As shown in Figure 13(a), the 1" maximum displacement limit results in impulse loads occurring in the support shaft. These loadings represent the forces required to set the 1 ton trough assembly into motion. A stop placed 1" from the journal should have the ability to survive these high impact loads. As shown in Figure 13(b), the stop must be able to withstand at least a 1000 lb_f load. This is a very approximate answer considering that numerically it is difficult to produce an impulse function. Therefore, an adequate safety factor should be applied (~2-4).

As shown in Figure 13(c), even with impact loading, the mirrors will not break.

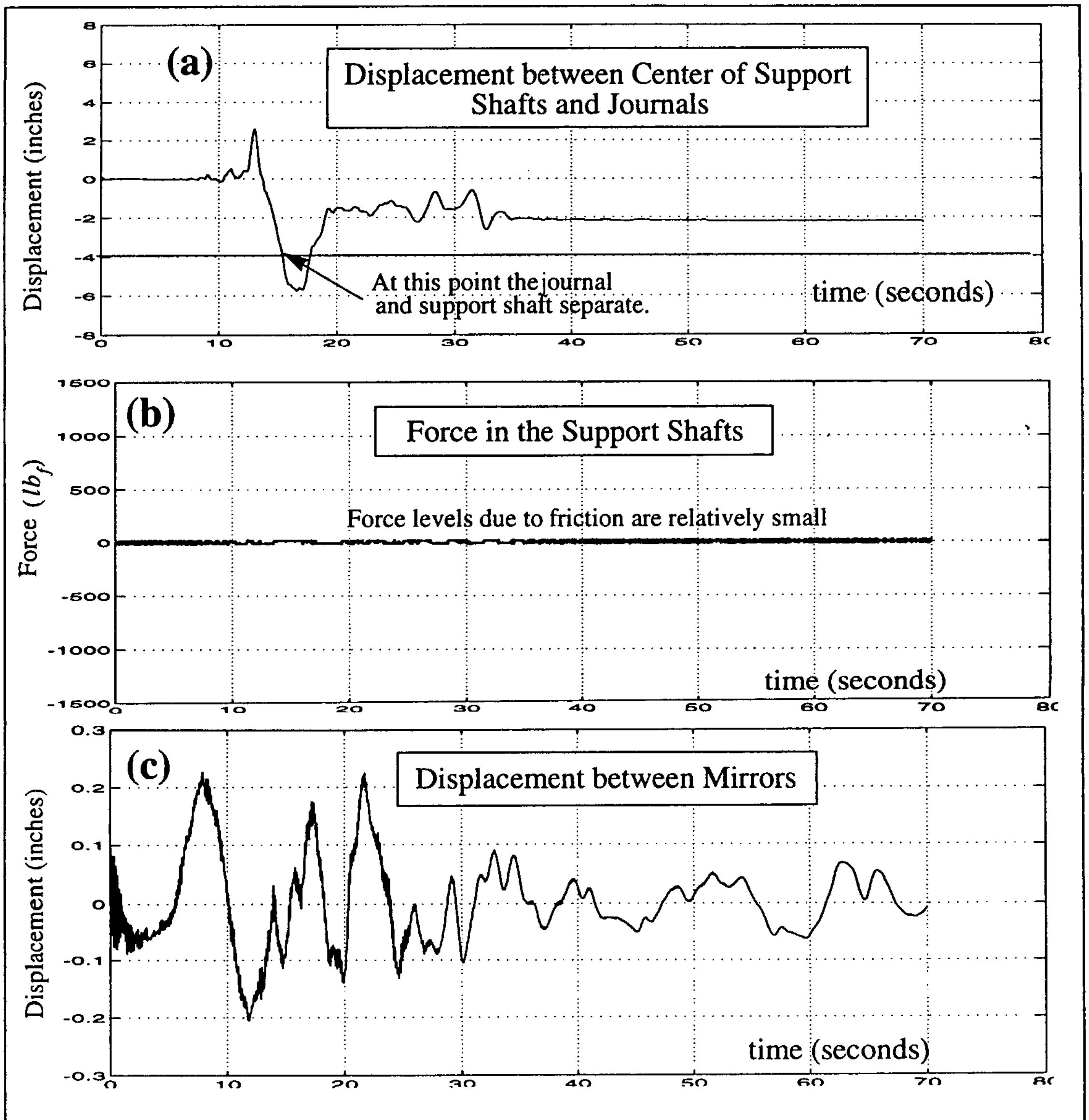

Figure 12. Collector Dynamics with No Stops.

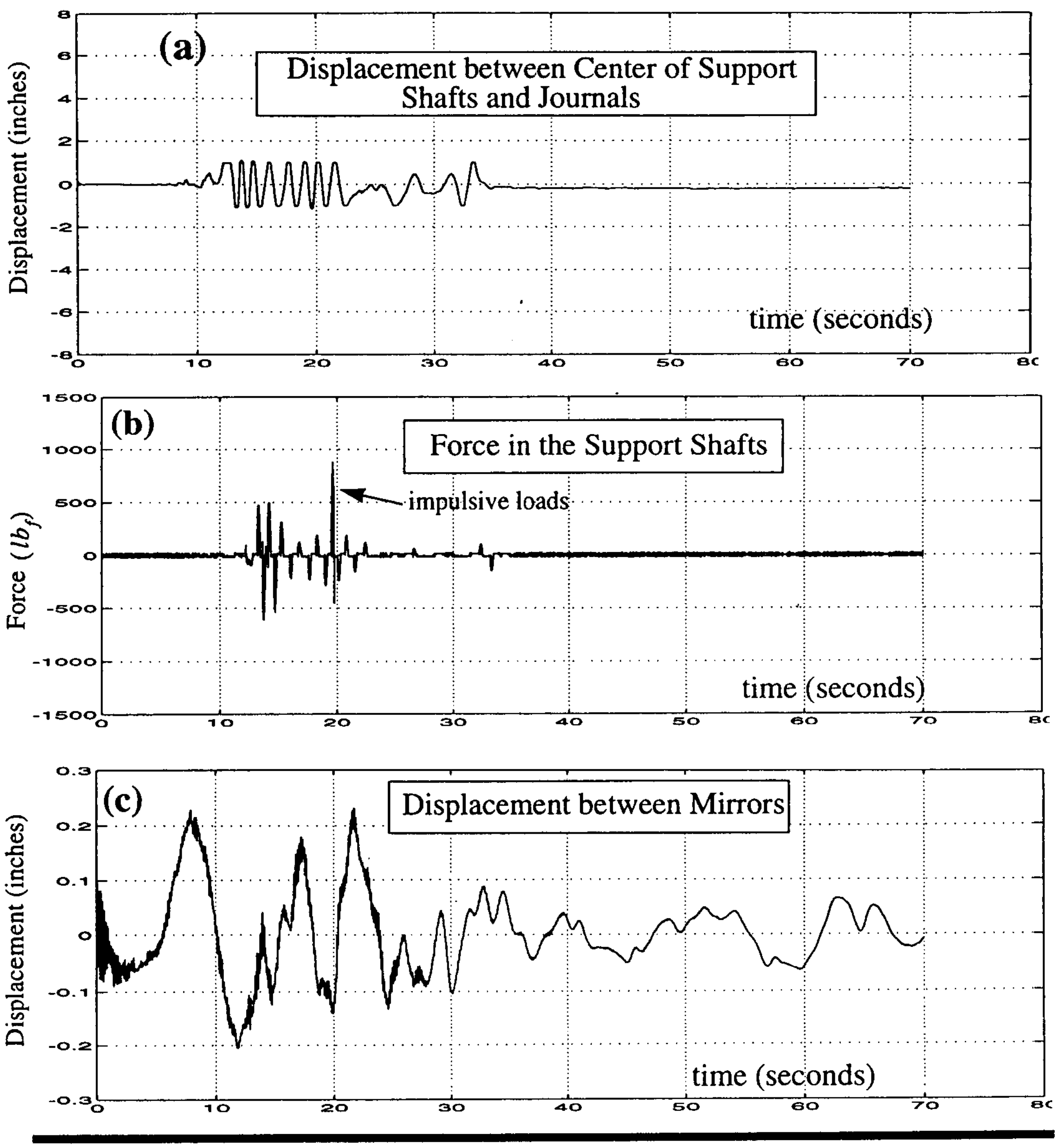

Figure 13. Collector Dynamics with Stops.

4. CONCLUSIONS

The purpose of this project was to redesign collector joints so that joint separation and costly breakage would not occur. Since a remedy to the joint separation problem had already been implemented, it was necessary to determine the sufficiency of this remedy before further redesign. For structural integrity purposes, this remedy is sufficient.

Assuming the validity of assumptions made within this report, the implementation of metal stops welded to the support shaft is a suitable solution to the joint separation problem when the stops are 1" from the journal. Stops must withstand loads of over $1000 lb_f$.

Another concern in this analysis was the breakage of the mirrors due to enhanced vibration in the trough assembly. It was found that the addition of stops produced little change in mirror dynamics. Mirror breakage is unlikely.

APPENDIX A - Model Reduction using Craig-Bampton

When performing model reduction, some DOF are eliminated from the FE model, while others are preserved. If $\vec{u}_p \in R^{N_p x1}$ contains all DOF to be preserved while $\vec{u}_{\bar{p}} \in R^{N_{\bar{p}} x1}$ contains all other DOF, then the DOF of the FE model are given by $\left[\vec{u}_P^T, \vec{u}_{\bar{P}}^T\right]^T$. In Craig-Bampton, a transformation matrix, T, is produced such that

$$\begin{bmatrix} \vec{u}_P \\ \vec{u}_{\bar{P}} \end{bmatrix} = T \begin{bmatrix} \vec{u}_P \\ \vec{u}_g \end{bmatrix}$$

where $\bar{u}_g$ are a set of ordered generalized coordinates whose truncation will reduce model order.

The generalized coordinates, $\bar{u}_g$, are determined by constraining $\bar{u}_P = 0$ and transforming resulting system dynamics into modal coordinates. That is, if the FE model is given by

$$\begin{bmatrix} M_{11} & M_{12} \\ M_{21} & M_{22} \end{bmatrix} \begin{bmatrix} \vec{u}_P \\ \vec{u}_{\bar{P}} \end{bmatrix} + \begin{bmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{bmatrix} \begin{bmatrix} \vec{u}_P \\ \vec{u}_{\bar{P}} \end{bmatrix} = \begin{bmatrix} \vec{F}_P \\ 0 \end{bmatrix}$$

then, $\vec{u}_g = \phi^T u_{\bar{P}}$ where $\{K_{22} - \omega_n^2 M_{22}\} \vec{\phi}_n = 0$, and $\phi = \left[\vec{\phi}_1, \vec{\phi}_2, \vec{\phi}_3 \ldots \vec{\phi}_{N_{\bar{P}}}\right] \in R^{N_{\bar{P}} x N_{\bar{P}}}$ for $\omega_1 \le \omega_2 \le \omega_3 \ldots \le \omega_{N_{\bar{P}}}$. The vectors ϕ_1 to $\phi_{N_{\bar{P}}}$ are called the component modes.

These modes along with constraint modes, Ψ_n, will form a complete collection of basis functions for the FE model.[8] Constraint modes are determined by solving the n static problems given by the solution to the following problem statement

IF $u_P(n)$ is the n^{th} element of $\bar{u}_P$ then for $u_P(n) = 1$, $u_p(m \ne n) = 0$,

FIND $\bar{u}_{\bar{P}} = \Psi_n$ for $n, m = 1, 2, 3, \ldots N_P$ in the static case.

These N_P solutions are given by $\Psi = -K_{11}^{-1} K_{12}$ where $\Psi = \left[\Psi_1 \; \Psi_2 \; \ldots \; \Psi_{N_P}\right]$.

Therefore the transformation matrix, T, is given by

$$T = \begin{bmatrix} I & 0 \\ \Psi & \phi \end{bmatrix}$$

where $0 \in R^{N_P x N_{\bar{P}}}$ is a zero matrix, and $I \in R^{N_P x N_P}$ is an identity matrix. Since $\bar{u}_g$ is in modal coordinates, model reduction may be performed by simply eliminating the highest modes

8. This means that any response can be represented by the time weighted sum of the basis functions.

of $\bar{u}_g$. Therefore, for the truncated matrix $\hat{u}_g \in R^{N_g x 1}$, and $T \in R^{(N_P + N_{\bar{P}}) x (N_g + N_P)}$ where $N_g < N_{\bar{P}}$, the reduced order model is given by

$$m_r \begin{bmatrix} u_P \\ u_g \end{bmatrix} + k_r \begin{bmatrix} u_P \\ u_g \end{bmatrix} = \begin{bmatrix} \hat{F}_P \\ 0 \end{bmatrix}$$

where $m_r = T^T \begin{bmatrix} M_{11} & M_{12} \\ M_{21} & M_{22} \end{bmatrix} T$, $k_r = T^T \begin{bmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{bmatrix} T$, $m_r, k_r \in R^{(N_g + N_P) x (N_g + N_P)}$.

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