Design Optimization of a Vibration Exciter Head Expander

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Recent advances in computer technology make it possible to utilize finite element techniques in the design process to optimize the dynamic response of a structure. Further, the analytical model and its assumptions can be validated by correlation of the dynamic characteristics with the modal parameters obtained from testing a prototype. This article discusses the application of structural optimization in the design of a frequency constrained structure. The structure under consideration is the head expander component of an electrodynamic exciter used for vibration tests. The head expander is a welded magnesium plate structure. The constraint is that flexural or bending modes of vibration occur at frequencies greater than 2100 lb when the head expander is connected to the armature of the shaker. Included are the results of the finite element design process, resulting head expander design configuration and a correlation of the analytical prediction of the dynamic normal modes response to the experimentally determined modal parameters obtained from an existing head expander design.

High performance shakers have relatively small armatures. This characteristic allows very high vibration levels to be easily produced on small payloads. However many people have multipurpose requirements, i.e., they have to test large payloads to low vibration levels as well as small payloads to high vibration levels. This type of situation requires a multipurpose system and cannot be addressed by shakers designed specifically to test large payloads at low vibration levels (e.g., environmental stress screening vibration). One approach to this multipurpose requirement is to use a “head expander.” Figure 1 shows a 24 × 24 in. head expander mounted on a shaker having an armature with a 12 in. outer bolt circle. The item on top of the head expander is a fixture used for holding a large test article (not shown). Historically, head expanders have not been “high tech” components. Most suppliers use magnesium for head expanders rated out to 2000 Hz, but the design techniques have been mostly “cut and try” or, in a few cases, using standard FE analysis. This area was therefore ideal for structural optimization.

Modes of vibration within the usable bandwidth of a vibration exciter mounting table will result in significant variations in vibration level between various points on the table This is obviously very undesirable and it applies to diaphragming and bending modes of shaker armatures as well as to head expanders. On the other hand, head expanders having more dead weight than necessary will reduce the usable payload by the amount of unnecessary weight.

The object of this research was therefore to design an optimized 24.0 in. (61.0 cm) square head expander having minimum weight and having all troublesome modes of vibration at frequencies greater than 2100 Hz when the head expander is attached to a shaker armature having a 12 in. outer bolt circle. To achieve this objective, the head expander was considered both analytically and experimentally, employing two boundary conditions: free-free and operational. In summary, this article details the steps involved in the baseline analysis, experimentation, and correlation involved with the original design of a head expander. The design steps and experimental results associated with the first generation optimized head expander are discussed.

Finally, the second generation redesign analysis of the optimized head expander which resulted from knowledge gained in the first generation design effort is presented. The first generation analysis resulted in a head expander design that satisfied the frequency constraint, but with an increase in mass of approximately 23% over the baseline design. The second generation head expander design satisfied the frequency constraint with a safety margin of 200 Hz while at the same time weighing less than the baseline bead expander design.

It should be noted that SDRC I-DEAS Design Engineering Analysis Optimization Software was employed in the redesign of the head expander. Specifically, I-DEAS level 4.0 was used for the first generation redesign and I-DEAS level 5.0 was used for the second generation redesign. The addition of shape optimization capability in I-DEAS level 5.0 software resulted in increased efficiency in the design process over the I-DEAS level 4.0 size optimization. Shape optimization allowed the movement of node groups as optimization variables. Size optimization considered plate element thickness as the optimization variable.

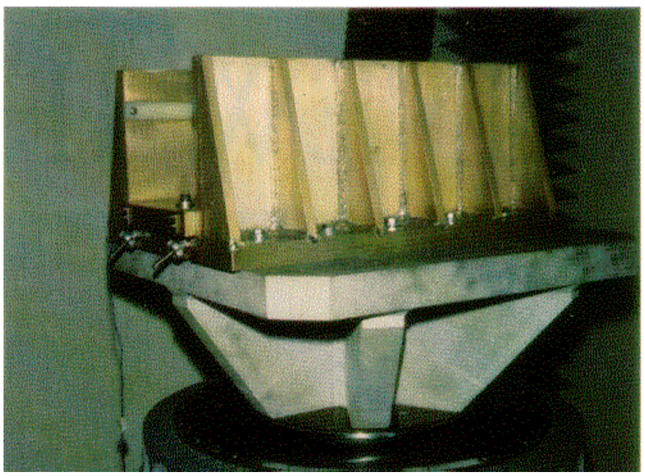

Figure 1 Baseline head expander design.

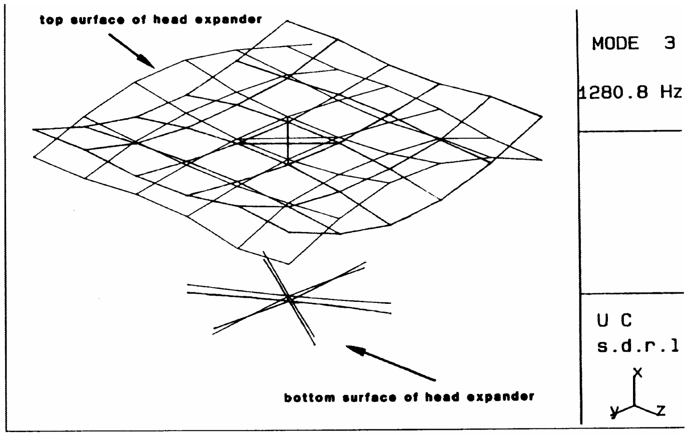

Figure 2 Baseline head expander design — weak bending mode experimental modal analysis.

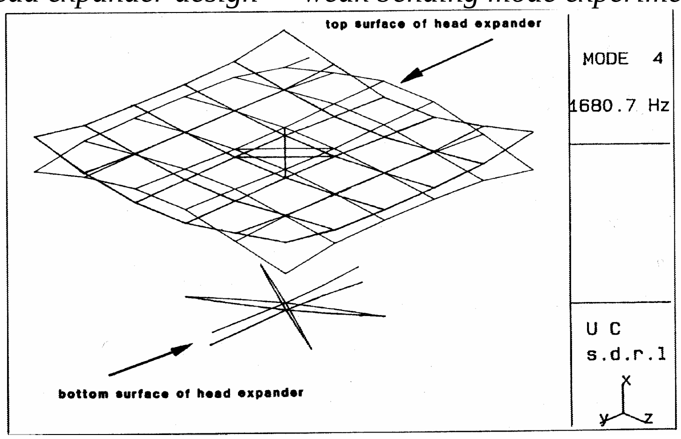

Figure 3 Baseline head expander design — strong bending mode experimental modal analysis.

Baseline Experimentation and Analysis

The baseline head expander was found to have five fundamental modes of vibration (see Table 1) when impact tested as a “free-free” structure. Two of these modes were bending modes at 1280 Hz and 1680 Hz (see Figures 2 and 3). The bending modes occurred at different frequencies because of the stiffness bias associated with the geometry of the six support ribs.

It should be noted that the baseline head expander design did have other fundamental modes of vibration less than 2000 Hz (see Table 1). These modes were antisymmetric torsional modes of the head expander surface and were judged not to be excited by uniform base input excitation of the armature of the shaker. This was verified by an operational test of the baseline head expander. It was also assumed that a test payload mounted over the center of mass of the head expander would not significantly excite torsional modes. Consequently antisymmetric torsional modes of vibration were not considered important modes for optimization in this application, but were identified and quantified in the analysis (see Table 1).

The baseline head expander was found to have experimentally determined “operational” bending modes of 1420 Hz and 1800 Hz (see Table 2). These operational bending modes were the modes that were targeted for optimization. It was also experimentally demonstrated that operational conditions could be experimentally verified by performing a “free-free” impact test of the baseline head expander fastened to an armature “simulator.” The armature simulator consisted of an aluminum cylinder having a diameter of 13.0 in, (33.0 cm), a height of 3.0 in. (7.6cm). and a weight of 39.0 lb (17.7 kg).The bending modes of the head expander/armature simulator assembly were experimentally determined to be 1893 Hz and 1837 Hz (see Figure 4). These modes correlated to the operational bending modes measured at 1420 Hz and 1800 Hz (see Table 1). It was then concluded that the armature simulator cylinder was an appropriate boundary condition to simulate operational conditions. Incorporation of the armature simulator in the finite element analysis to duplicate operational conditions was essential in the redesign of the head expander.

Table 1: Fundamental deformation modes in Hz of original design head expander, free-free boundary conditions without armature simulator.

Deformation Mode	Experimental Impact Test	Finite Element Model		
		Linear Thick Shell	Parabolic Thick Shell	Parabolic Rigid Offset
First Torsion	1024	1060	950	—
Second Torsion	1255	1350	1200	—
Weak Bending	1280	1300	1180	1260
Strong Bending	1680	1730	1530	1670
Third Torsion	1774	1820	1660	—

Table 2: Experimental deformation modes in Hz of original design head expander in various configurations.

Deformation Mode	Operational Test	Impact Test Head Expander-Armature Simulator Assembly	Impact Test Without Armature Simulator
Weak Bending	1420	1393	1280
Strong Bending	1800	1837	1680

Before the baseline head expander was redesigned, finite element models were built and normal modes analyses of the “free-free” head expander were performed using SDRC I-DEAS Simultaneous Vector Iteration (SVI) solution method. This permitted a baseline correlation of the experimentally and analytically determined modal parameters of the “free-free” baseline head expander design (see Table 1*)*. The baseline head expander (see Figure 5) was modeled using three different mesh configurations. All three models incorporated plate elements because of the proposed optimization solution method. The first model was a linear plate element mode. Figures 6 and 7 depict the first two bending modes solved for this model. The second model was a parabolic plate element model. The final model was also a parabolic plate element model, but included rigid offsets such that the mass was exact and not overstated by the plate element overlap at the intersection nodes of adjacent elements. It was concluded that the parabolic plate element model with the rigid offsets was the most accurate model. But since the rigid offset length was a function of the parameter to be optimized, the rigid offset method was not used for the optimization redesign model.

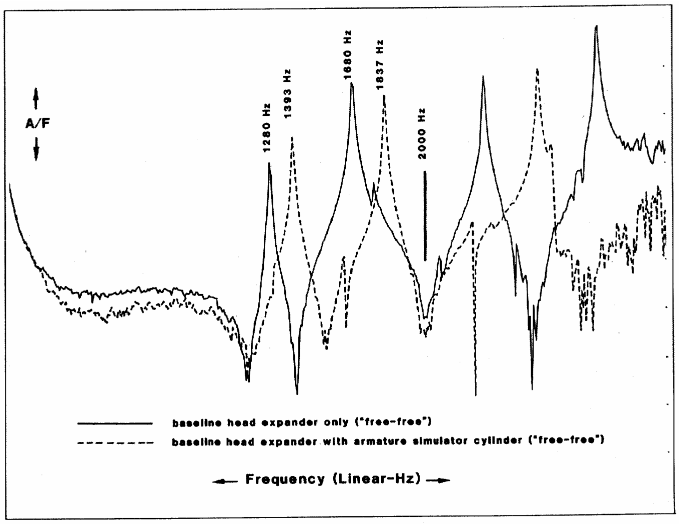

Figure 4 Experimental results of baseline head expander.

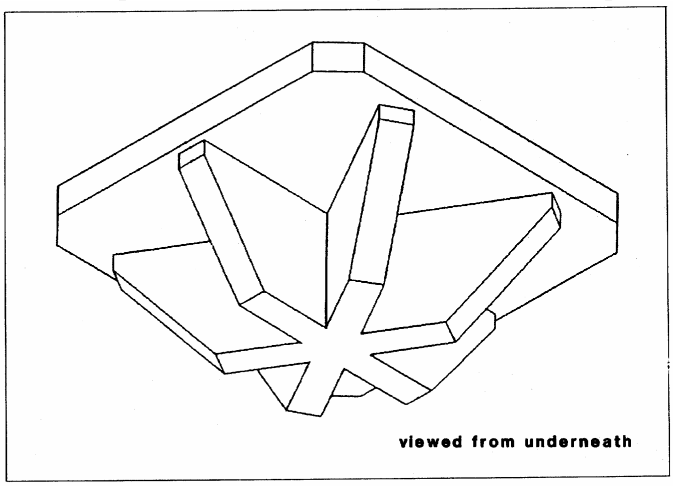

Figure 5. Baseline head expander design.

First Generation Redesign

In the initial redesign of the head expander, the fixed design criteria were as follows:

- Minimum frequency constraint of 2100 Hz for bending modes when the head expander/armature simulator assembly was analyzed as a "free-free" structure.
- Platform table surface dimensions remained the same as baseline design at 24.0 in. (61.0 cm) square sides with 2.0 in. (5.0 cm) truncated corners.
- Mounting bolt pattern for attachment to armature remained constant.

The following redesign variables were such that the 2100 Hz minimum frequency constraint was satisfied for a head expander having minimum weight:

- Thicknesses of the head expander platform table and support ribs were optional.
- Shape and height of the support ribs were optional.
- Number and location of the support ribs were optional, but rib configuration must accommodate the given mounting bolt configuration.
- Head expander material was optional.

The baseline normal modes analysis targeted the two bending modes that were to be strengthened in order to satisfy the minimum 2100 Hz frequency constraint. Strain energy contour plots of these modes identified specific areas of critical deformation that were to be stiffened to increase the natural frequencies of the bending modes.

From the deformed geometry plots of the two fundamental bending modes (see Figures 6 and 7), it was determined that support ribs should be added at the approximate locations of the node lines of both bending modes to strengthen these modes. For example, support ribs added at the node lines of the strong bending mode strengthened the weak bending mode. Stated otherwise, the weak bending mode was stiffened while minimally mass loading the strong bending mode. Conversely, support ribs added at the node lines of the weak bending mode strengthened the strong bending mode.

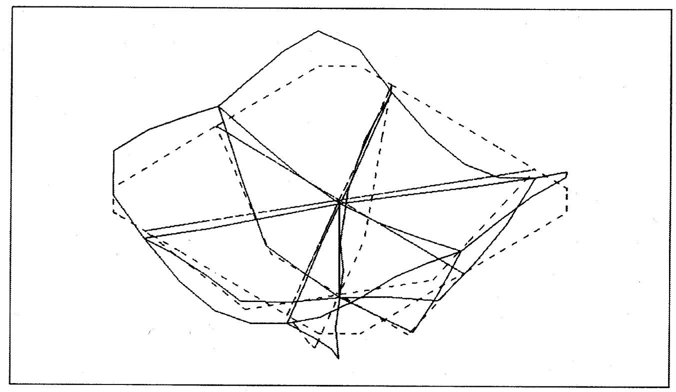

Figure 6. Baseline head expander design–
weak bending model finite element normal modes analysis.

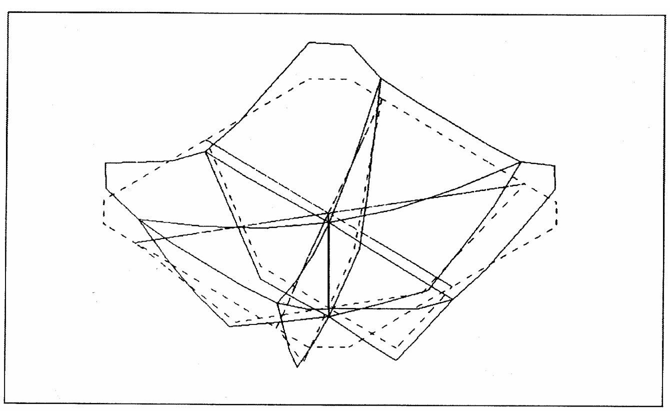

Figure 7 Baseline head expander design–
strong bending mode finite element normal modes analysis.

A first generation redesigned head expander finite element model was constructed by locating support ribs at the approximate node lines as described above (see Figure 8). Also, central ribs were maintained to accommodate the existing mounting bolt pattern.

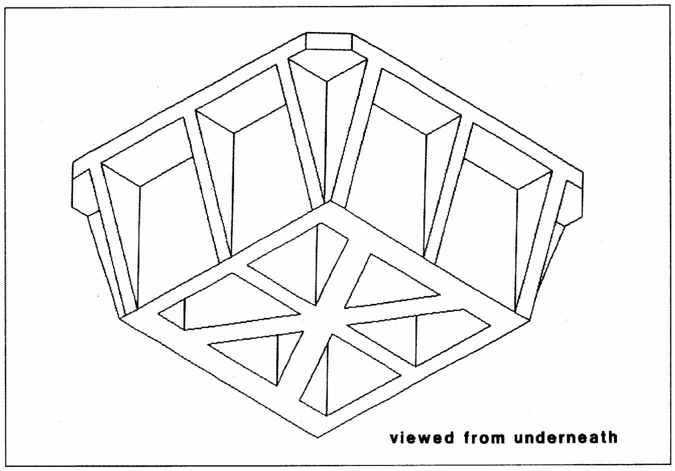

Figure 8. First generation redesigned head expander

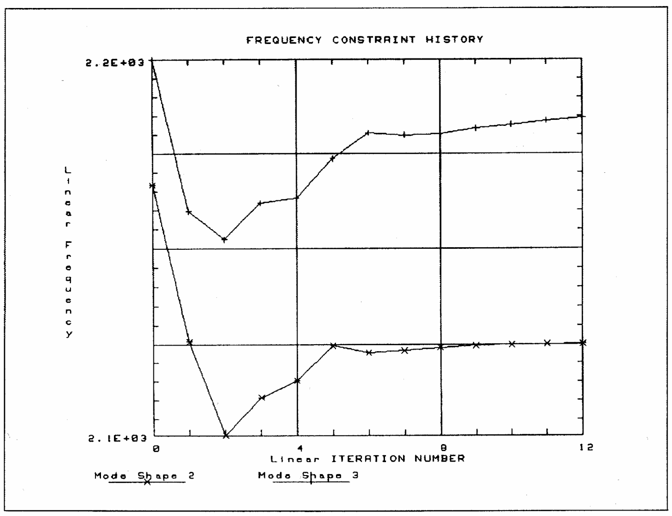

Figure 9. First generation redesigned head expander optimization iteration frequency constraint history.

Magnesium was chosen for the redesigned head expander material — unchanged from the original head expander design. Magnesium exhibited a greater bending stiffness and a comparable axial stiffness when compared to other material sections having the same weight. Fabrication consisted of continuous welding of the support ribs to the surface plate.

The Optimization module of the SDRC I-DEAS Engineering analysis family was used to redesign the head expander for minimum weight given the minimum 2100 Hz frequency constraint. Since bending modes were the modes to be considered by Optimization, a quarter model using symmetric boundary conditions along the intersecting planes of symmetry was employed to solve only for the bending modes and isolate the inactive torsional modes. It should be noted that since the orthogonal support rib was located along one of the planes of symmetry, only one half of this rib thickness was used in the Optimization redesign. This was allowed because the fundamental bending modes subject this rib to axial deformation only, not bending deformation where reduced section thickness would change the rib inertia and the quarter model solution would be invalid. Seven optimization groups of elements were chosen for the quarter model optimization redesign solution. It should be noted that the armature simulator was also incorporated into the Optimization redesign to simulate operational conditions. The finite elements that defined the armature simulator were frozen with their original properties and not resized by optimization.

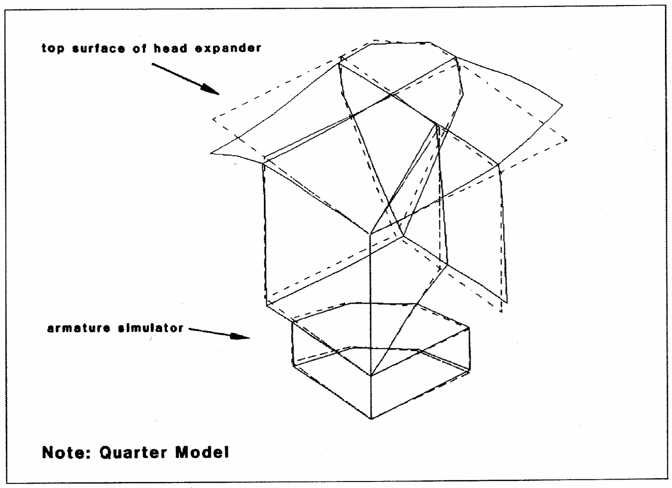

Figure 10. First generation redesigned head expander optimization first bending mode.

The choice of modeling one quarter of the head expander (with symmetric boundary conditions) greatly increased the computational efficiency of the redesign solution The quarter model solution also considered the fundamental bending modes targeted in the Optimization redesign and isolated the torsional modes not energized by a uniform base input excitation.

A plot of the frequency constraint history (see Figure 9) shows the frequencies of the two fundamental bending modes considered by Optimization as a function of iteration number. The deformed geometries of these modes are shown in Figures 10 and Il. The final weight of the first generation redesigned head expander was 165 lb (75 kg). The baseline head expander weighed 134 lb (61 kg). The 2100 Hz minimum frequency constraint was achieved with an increase in weight of 31 lb (14kg) or approximately 23%.

A first generation redesigned head expander was fabricated for test. Experimental verification of the redesigned head expander showed good correlation of the two bending modes of the finite element optimization process. However, operational testing revealed the existence of a mode with slightly higher damping than the two bending modes and at a frequency of just less than 2000 Hz (see Figure 12). This mode proved to be a system mode of the head expander/armature assembly.

This system mode was not identified during the finite element Optimization procedure because of the inappropriate assumption of the armature simulator boundary condition. While the armature simulator proved to be a suitable boundary condition for the baseline head expander, the stiffness properties of the armature simulator rendered it inappropriate for use as a boundary condition for the redesigned head expander.

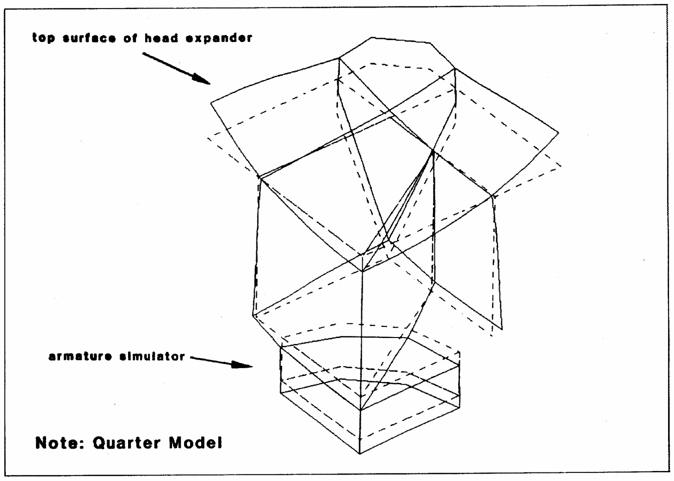

Figure 11 First generation redesigned head expander optimization second bending mode.

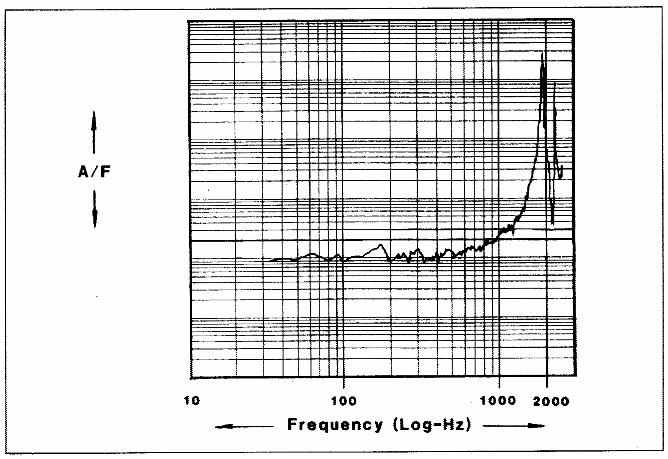

Figure 12. First generation redesigned head expander experimental response spectrum of head expander surface.

Experimental and finite element analyses revealed the system mode to be an out of phase axial (vertical) mode between the head expander and the armature assembly. further analysis showed that the addition of a baseplate welded to the lower surface of the head expander would add stiffness to the system mode. The baseplate modification was made to the first generation head-expander (see Figure 13). An operational lest of the modified first generation head expander verified the-stiffened system mode at approximately 2060 Hz, but with an increase in weight of 8 lbs.

From the first generation head expander design effort, it was concluded that a second generation head expander design process must consider the dynamics of the armature Also it was concluded that the out of phase axial system mode of the head expander/armature assembly was very sensitive to the increase in weight of the system.

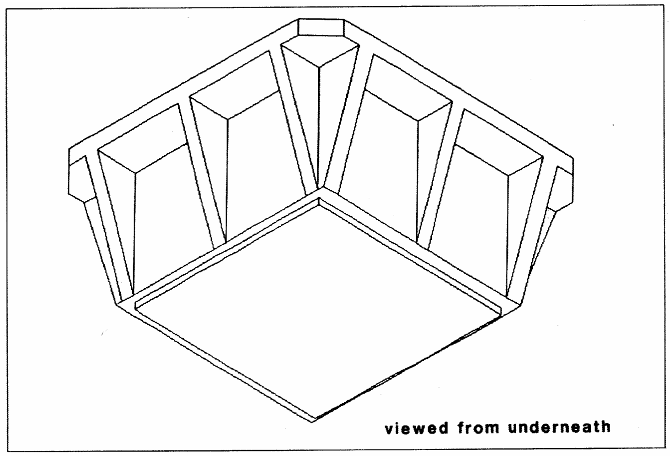

Figure 13. First generation redesigned head expander baseplate modification

Second Generation Head Expander

In order to consider the dynamics of the head expander/armature system, the dynamic properties of the armature were determined. The armature consisted of a magnesium upper casting which supported a lower aluminum driver coil. The armature was found analytically and experimentally to have an axial mode of the upper surface of the casting out of phase with the driver coil at approximately 2600 Hz (see Figure 14). This mode contributed to the out of phase system mode previously identified.

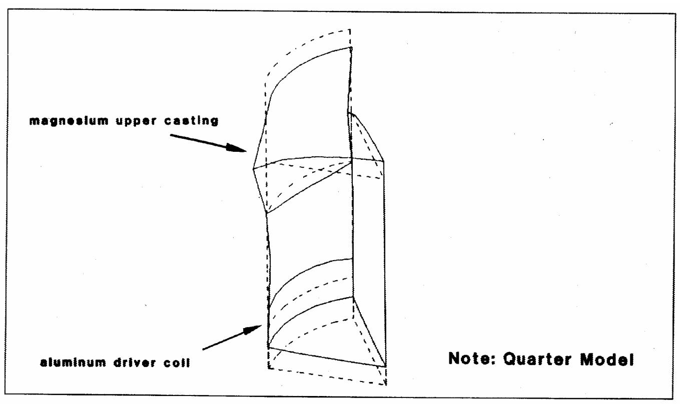

Figure 14 Armature component out of phase axial mode.

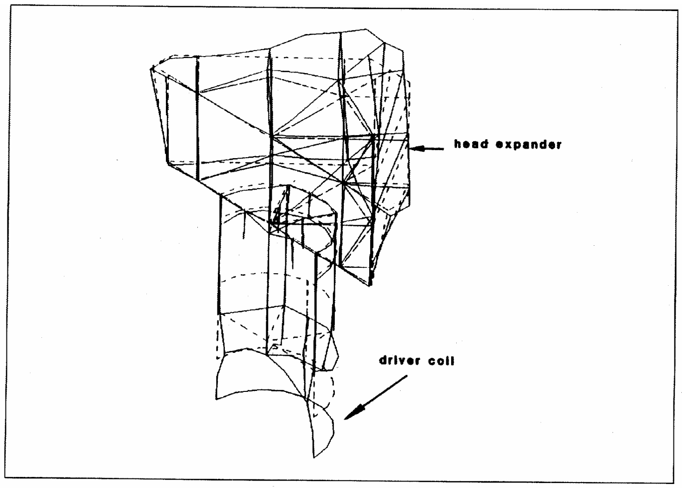

Figure 15 Proposed second generation head expander system first saddle mode.

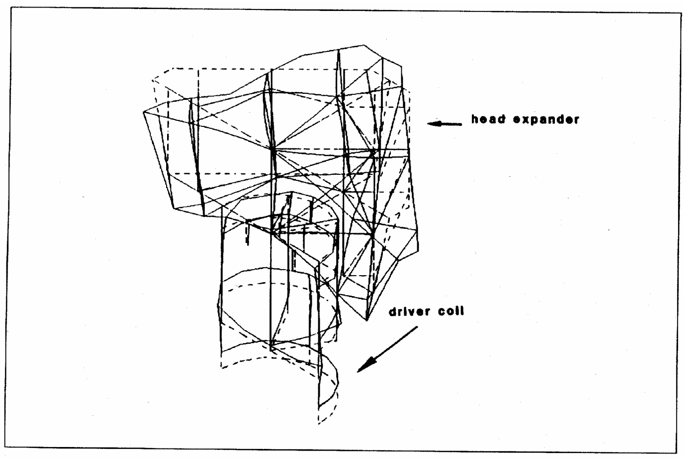

Figure 16 Proposed second generat ion head expander system second saddle mode.

The analytical integrity of the armature finite element component was verified by alternately attaching masses of 40 lb (18 kg) and 113 lb (51 kg) to the armature and determining the change in the 2600 Hz out of phase axial mode. These masses were 13.0 in. (33.0 cm) diameter cylinders having a height of 3.0 in. (7.6 cm) and consisting of aluminum and steel, respectively. When the 40 lb. aluminum cylindrical mass was attached to the armature, the out of phase axial mode dropped in frequency to 2400 Hz. When the 113 lb steel mass was attached the out of phase natural frequency became 2120 Hz. Both of the experimentally determined boundary conditions were verified and correlated with the armature finite element component.

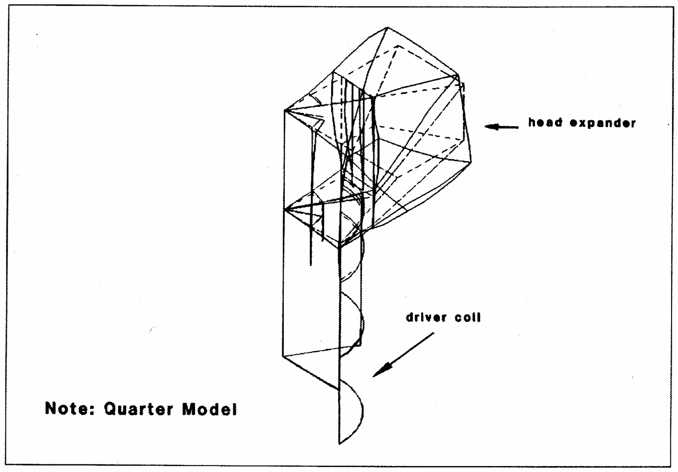

Figure 17 Final second generation head expander system first saddle mode

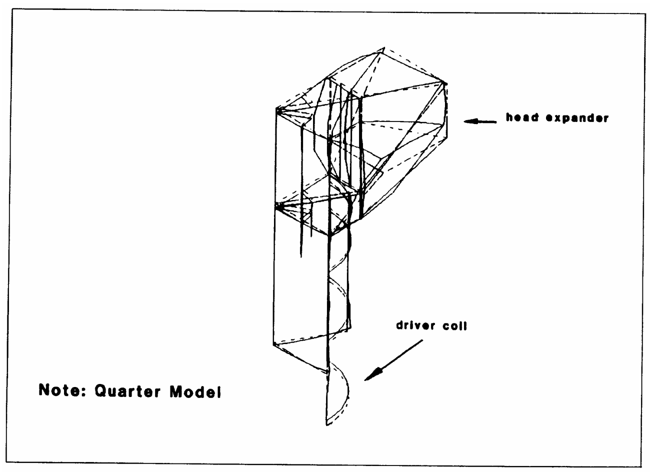

Figure 18 Final second generation head expander system second saddle mode

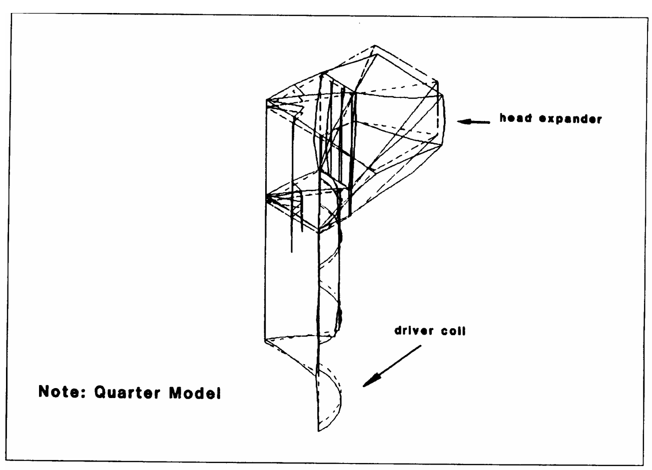

Figure 19 Final second generation head expander system "oilcan" mode

Before redesigning the second generation head expander, the design effort of the first generation head expander was reviewed. In reviewing the design procedure of the first generation head expander, it was noticed that as the Optimization process sized the support ribs, the two distinct bending modes converged to saddle modes and an “oil can” mode (see Figures 9 and 10). The node lines of these modes are not “parallel lines” typical of bending modes. Therefore, to achieve the maximum stiffness with minimum mass loading, the support ribs were located at the node lines of the saddle modes and the “oil can” mode for the second generation head expander. This resulted in a design that contained four major ribs that passed through the center of the head expander at the node lines of the saddle modes separated by a ring of support ribs at the approximate node line of the oil can” mode. This head expander component was then optimized for minimum mass subject to a more conservatively revised 2200 Hz frequency constraint.

Analysis revealed the head expander component had saddle modes at 2168 Hz and 2330 Hz, an “oil can” mode at 2507 Hz, and weighed 130 lb. This weight was less than the baseline head expander of 134 lb.

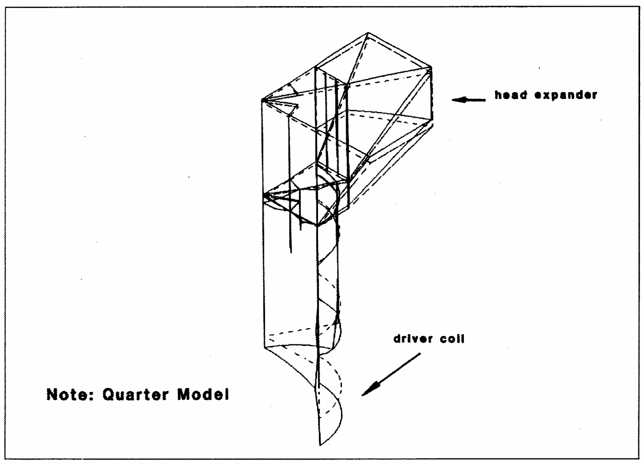

Figure 20 Final second generation head expander system out of phase system mode.

However, what was thought to be a satisfactory second generation head expander component design proved to be a marginal system design when connected to the armature of the vibration exciter. The head expander/armature assembly saddle modes occurred at 1476 Hz (see Figure 15) and 1523Hz (see Figure 16). Clearly, the second generation head expander component design was unsatisfactory when analyzed as a system. The correct procedure involved the optimization of the head expander/armature as a system not as individual components. The head expander was redesigned to take advantage of the stiffness of the armature by incorporating six support ribs that passed through the center of the head expander, while at the same time locating the ribs at the approximate node lines of the saddle modes. The ring of support ribs was relocated at the same diameter as the armature and inclined panels were added between the vertical support ribs. The head expander/armature assembly was then optimized as a system to achieve the frequency constraint of 2200 Hz with minimum mass.

The physical properties of the armature finite elements remained fixed in the analysis. Results showed the saddle modes to be at 2266 Hz (see Figure 17) and 2842 Hz (see Figure 8) and the “oil can” mode at 2658 Hz (see Figure 19). The out of phase system mode-occurred at 2043 Hz and consisted of a rigid body vertical translation of the head expander surface which moved out of phase with the driver coil (see Figure 20). Since the top surface of the head expander remains flat, this mode at 2043 Hz will not be significant to the user as long as it is controllable.

Controllability is not expected to be a problem since there is significant damping in this type of mode. The closed-loop controller that is always used for this kind of testing will be able to reduce the current to the driver coil at the frequency of this out of phase system mode, thus resulting in the requested vibration level. The final second generation optimized head expander design is predicted to weigh 125 lb (57 kg). The final design configuration of the second generation head expander component satisfied the 2200 Hz frequency constraint when connected to the armature and analyzed as a system.

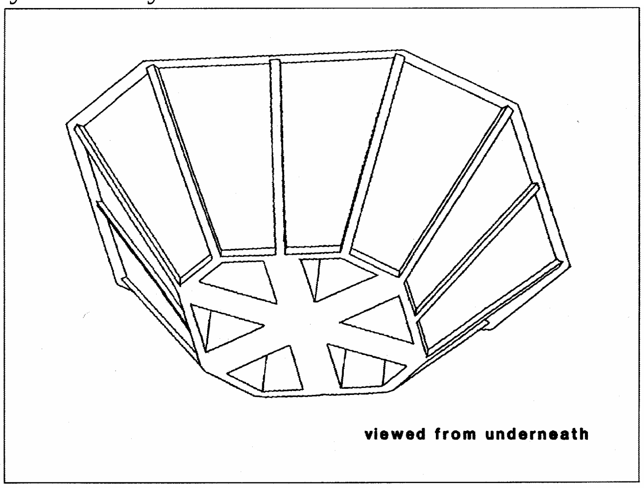

Figure 21. Final second generation head expander.

Conclusions

This research established a procedure for criticizing the dynamic response of the head expander/armature assembly of a vibration exciter to achieve a specified frequency constraint with a minimum mass. Specifically, the baseline head expander design weighed approximately 134 lb (61 kg) and had a troublesome first bending mode of 1280 Hz when attached to a shaker armature. The first generation redesigned head expander weighed 165 lb (75 kg) and analytically satisfied the frequency constraint of 2100 Hz. However, it was noted that the baseline head-expander bending modes converge to saddle modes and an "oil can" mode when optimized. Further, it was discovered experimentally that a system mode had been inadvertently ignored due to an inappropriate boundary condition for the analytical model. It was also analytically demonstrated that the correct boundary conditions must be included and the system, rather than the component, be optimized to achieve the optimum dynamic performance for minimum weight. This resulted in a head expander/armature assembly having analytically determined system modes of greater than 2200 Hz and the head expander component weighing only 125 lb (57 kg) (see Figure 21).

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