77D 26557

ENGINEERING ANALYSIS REPORT

of the

DESIGN OF THE F-144-1 SHIPPING CONTAINER

to

E. I. DU PONT DE NEMOURS & COMPANY

September 22, 1972

BATTELLE
Columbus Laboratories
505 King Avenue
Columbus, Ohio 43201

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Atomic Energy Commission, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

MASTER

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

DESIGN OF THE F-144-1 SHIPPING CONTAINER

by

Elmer C. Lusk and Richard J. Burian

September 22, 1972

INTRODUCTION

The objective of the program for the design of the F-144-1 shipping container was to develop a container for the shipment of FFTF fuel material and to verify the container design by a series of drop tests. The design constraints indicated by E. I. du Pont at the beginning of the program included the following:

- (1) Payload envelope. The container must be able to transport fuel pin bundles or fuel assemblies having a maximum envelope diameter of 5.94 inches and maximum length of 144 inches.
- (2) Container envelope. The width of the container should not exceed 42 inches. No constraints were imposed on the height or length.
- (3) Container weight. No constraints were imposed on container weight.
- (4) Transportation and normal handling impact. The container should be so designed that the canister which contains the payload would experience a maximum of 6 g impact force during normal transportation and handling. The worst credible "normal handling" incident was a free fall of the container from 48 inches onto a hard, unyielding surface. Moreover, the protection system must continue to operate under repeated incidents, i.e., plastically deformable impact absorbers are unsatisfactory.

- (5) Transportation vibration. The container should be so designed that the canister which contains the payload will not experience forces above 6 g due to normal transportation vibrations. The vibration spectrum was not defined.
- (6) AEC licensing requirements. The container design should comply with the licensing regulations of 10 CFR-Part 72 of the Code of Federal Regulations governing the design of packages for the transportation of fissile materials.

Description of the Container

The F-144-1 shipping container is described below. The various major components are noted on Figure 1 which is a photograph of one prototype unit fabricated at Battelle.

General

A general description of the container is as follows:

- (1) Shape - A regular six sided "box" shape with a large aspect ratio of length to height or width
- (2) Size - 188 inches long x 42 inches high x 42 inches wide with lead impact absorbers at each corner external to the container envelope
- (3) Construction - A welded frame made of steel angles and braces with a sheet metal cover on all six sides; fuel subassembly or bundled fuel pins transported in a canister made of nominal 6 inch, schedule 40 stainless steel pipe. The fuel element canister is suspended by 16 cables attached to springs and columb (friction) dampers on the frame.
- (4) Capacity - One FFTF or other LMFBF subassembly or bundle of up to 120 fuel pins up to 144 inches in total length and up to 5.94 inches in diameter.
- (5) Loaded shipping weight (calculated) - 3578 lb.

Legend:

1. Frame
2. Spring Tube
3. Steel Cables
4. Coulomb Damper
5. Canister
6. End Honeycomb impact absorber
7. Side Honeycomb impact absorber

FIGURE 1. PHOTOGRAPH OF F-144-1 SHIPPING CONTAINER

Frame

The frame of the F-144-1 shipping container can be described as follows:

- (1) Shape: Rectangular box with large length to width and length to height ratios.
- (2) Size: 188 inches long x 42 inches high x 42 inches wide.
- (3) External impact absorbers: Eight lead absorbers, one on each corner. At each corner, the absorber is shaped to fit "over" the corner so that a "pad" of lead 2 inches thick x 10 effective square inches in cross section protects each of the three sides meeting at the corner. (The external impact absorbers are not shown on the photograph in Figure 1.)
- (4) Structural construction: Basic box frame made of 4 x 4 x 1/4 steel angles welded together. Basic frame is braced by 2 x 1/4 steel straps welded to the frame.
- (5) Cover: The frame is covered on all six sides by 22 gauge steel sheet riveted and screwed to the frame. (The cover is not on the container shown in Figure 1.)
- (6) Suspension system: Located generally in the longitudinal corners of the frame are 16, 4-1/2 ID steel tubes (spring tubes), each housing two springs of different spring rate and preload force. A steel wire rope (cable) attached to the spring system passes over a pulley at the end of each spring tube and extends to the fuel element canister in which the subassembly or bundled fuel pins are contained during shipment.
- (7) Coulomb dampers: Attached to each pulley at the end of the spring tubes is an adjustable torque limiter for providing energy absorption and vibration damping.

Fuel Element Canister

- (1) Shape - cylindrical
- (2) Size - 6.02 inch ID x 6.625 inch OD x 145.75 inches long;
9.75 in. diameter x 1/2 inch thick flange on one end;
other end closed with flat welded bottom, 1/2 inch thick.
- (3) Closure - 9.75 inch diameter x 1/2 inch thick cover
secured to flange with six 1/2 - 13 x 1.0 inch ASTM-A193
bolts; seal accomplished with silicone rubber O-ring.
- (4) Impact absorber - The flange end of the fuel element canister
is protected by a honeycomb impact absorber on the end
and side of the canister at the top. The end absorber is
6 inches thick. The side absorber is 4 inches thick
radially and extends 9.5 inches down the canister below
the flange.
- (5) Materials of construction - The fuel element canister and
flange are made of type 304 stainless steel. The canister
impact absorber is a commercially available aluminum honeycomb.

Breakdown of Weight

A Breakdown of the weight of the components of the F-144-1 Shipping Container and contents are presented in Table 1.

TABLE 1. WEIGHT BREAKDOWN OF F-144-1 SHIPPING CONTAINER

Component	Weight, lb
Frame	
Structural Steel Members	780
Panels	300
Spring and Spring Tubes	1040
Torque Limiters (Friction Clutches)	240
Lead Energy Absorbers	180
Miscellaneous including cable	200
Subtotal, Frame	2740
Canister, Cover	365
Ballast	46
Fuel Elements, Spacers, etc.	375
Fuel Element Slide	52
Total	3578

DESIGN ANALYSES

The design analysis effort was divided into two major phases. During the first phase, Concept Evaluation, several concepts were developed and evaluated in cooperation with E. I. du Pont. The second phase was concerned with detailed development of the most promising concept and detailed thermal, structural, and nuclear analyses.

Preliminary Concept Evaluation

The most stringent design constraints were the ones limiting the impact force to 6 g for a 48-inch free fall incident within a 42-inch-wide container. In order to achieve the relatively low impact force, it is necessary to permit the payload considerable freedom of motion within the container envelope, restraining its motion only to the extent that all the kinetic energy is absorbed without exceeding the 6 g force restriction.

The basic container configuration developed early in this phase was that of a round, rigid canister (to hold the fuel materials) "lightly" suspended within a rigid frame.

In the ideal case, the restraining force on the payload is a constant 6 g over the entire impact period. If this condition could be achieved, the minimum displacement required by the payload is $x = 48 \text{ inches}/6g = 8 \text{ inches}$.

The force displacement (relative to the frame) curve for this ideal suspension system is shown in Figure 2a. The total kinetic energy which is absorbed is represented by the area under the curve. In actual practice, however, a "perfectly square" deceleration curve is difficult to achieve. This is especially true when the suspension system must be able to properly respond regardless of the impact orientation of the container and then return the canister to the null position without external assistance. It, therefore, was evident that the payload must be able to translate more than 8 inches in any direction while still meeting the 42-inch-width requirement of the frame. Meeting these two requirements simultaneously then became the most significant consideration in the program.

A system which would respond in the ideal manner illustrated by Figure 2a could be physically achieved only by the use of constant force devices such

(a) Ideal case, constant decelerating force

(b) Hydraulic and undamped spring system

(c) Damped, nonlinear spring system

FIGURE 2. FORCE-DISPLACEMENT CURVES FOR VARIOUS SUSPENSION SYSTEMS

as coulomb (friction) devices or by springs which continue to extend under a constant force, i.e., zero spring rate. Although such springs have been made and employed where small forces in the order of ounces are required, springs in the force range required in this design are not made commercially.

Perhaps the most common energy absorber is the hydraulic shock absorber. Unfortunately, however, the hydraulic energy absorbing devices do not produce a constant decelerating force but rather the resistance is proportional to the velocity. Thus, the decelerating force for a hydraulically damped (dash pot) system would decrease as the velocity decreased and the force-displacement curve might look like the indicated curve of Figure 2b. The area under this curve would be equal to the area under the curve of Figure 2a.

Figure 2b also illustrates the force-displacement curves which would be exhibited by pure spring suspension systems. A true linear spring system would have a triangular shaped force-displacement curve and thus the displacement would be twice that of the ideal system of Figure 2a (areas under the curve must be equal). A non-linear spring system where the spring rate increases with the displacement would exhibit a force-displacement curve as shown in the third curve of Figure 2b. In order to absorb all the kinetic energy in this case, displacements greater than twice the ideal case would be required.

Several combinations of the above devices were evaluated. A partial list includes

- (1) Spring suspension (coil and torsional).
- (2) Spring suspension rigidly constrained below 6 g by a break away tie-down system.
- (3) Spring - hydraulic (pneumatic) damper suspension.
- (4) Elastomeric shock cord suspension.
- (5) Elastomeric shear pad suspension.
- (6) Coulomb damped spring suspension.

After detailed evaluation, the coulomb damped spring suspension system was considered the most feasible from all considerations including both the space requirement and economic standpoints, as well as from simplicity of design, operation, and maintenance. Elastomeric shear pad suspension systems using commercially available devices appeared marginal and was assigned the role of an alternate system.

Designs using only externally undamped springs or elastomeric shock cords were discarded because preliminary analysis indicated that not only would extreme displacements as indicated in Figure 2b occur but the internal damping of the springs were so low as to permit prolonged oscillations. Moreover, the natural frequency of these systems was in the range of 1 to 4 Hz. As shown in Figure 3 this is in the critical range for normal truck transportation vibration and an inadequately damped system could experience a severe reinforced oscillation condition.

Hydraulically damped, spring return systems were examined in detail but were eventually eliminated from consideration because of the space requirements as well as the economics of such systems.

The force-displacement curve for a coulomb damped, spring suspension system is shown in Figure 2c. As shown, a minimum force, F , will always act to resist impending motion and as the displacement increases, the spring will also contribute to the deceleration. As shown in Figure 2c, such a system should be able to absorb all the kinetic energy with a total displacement between the ideal case, Figure 2a, and a pure spring system, Figure 2b.

Cursory evaluation of the other structural, thermal, and nuclear design aspects indicated that all the requirements could be met since considerable freedom was permitted in material selection and there was no weight restriction.

Detailed Concept Evaluation and Analysis

The detailed analysis effort was divided into four tasks, detailed suspension system design and analyses, structural analyses, thermal analyses, and nuclear analyses. The effort and results of the first task and the detailed design of the suspension system is discussed below. The detailed analysis and results of the structural, thermal, and nuclear analyses were included in the Safety Analysis Report (SAR) submitted to E. I. du Pont in May, 1972.⁽¹⁾

Suspension System Evaluation

As indicated above the suspension system selected as most feasible for use on the F-144-1 shipping container was one using coulomb damped springs.

FIGURE 3. REGIONS OF VIBRATION OCCURRENCE
IN VARIOUS TRANSPORTATION MEDIA

From Figure 2c it is seen that the ideal case of Figure 2a is approached as the coulomb force, F , increases and the spring rate decreases. There are, however, several practical constraints. First, in order for the suspension system to be able to perform under repeated impact conditions and, moreover, to respond to impact forces applied in any orientation, it is necessary for the canister to be returned to near the starting or null position after each impact occurrence. This requires that either (1) the coulomb damping device produces a frictional force for motions only in one direction while returning to null position with little returning force being required, or (2) the spring must be sufficiently extended in the null position so that the spring can overcome the frictional force and return the canister to the null position. The former design was considered more desirable but such a system would require the use of devices such as a rack (or pinion) and pawl, overriding clutches, etc. Several designs of this nature were investigated but their reliability under the impact conditions and at any orientation appeared questionable. Moreover, it was felt that these devices might require frequent maintenance and adjustment. The other alternative of spring and damper was, therefore, examined.

The approach taken can be best described by first considering a single linear spring with a low spring rate which has a force-displacement curve as shown in Figure 2a. If two identical linear springs with a spring rate shown in Figure 4a are joined and extended so they are under load at the null position, they will have the force-displacement curve shown in Figure 4b. If stops are introduced so that neither spring can contract but only extend from the null position, the force displacement curve will be as shown in Figure 4c. Then if a coulomb force device is introduced to each spring, the force-displacement curve of Figure 5a would be produced. If the springs were nonlinear, the force-displacement curve would be as shown in Figure 5b.

In the suspension system employed in the F-144-1 shipping container design, the springs themselves were linear. However, since they acted at an angle to the canister and the angle changed with displacement, the effective force on the canister was nonlinear. In addition, in the actual design the stops (contraction restraints) were adjusted to permit approximately ± 1 inch of travel from the null position before the stops were contacted. Moreover, to reduce the harshness of the blow as the stop was encountered by the spring,

$$\text{spring rate} = k = y/x$$

(a) Single linear spring

$$\text{effective spring rate} = k_e = 2y/x$$

(b) Two preloaded, opposed linear springs

$$\text{effective spring rate} = k = y/x$$

(c) Two preloaded, opposed linear springs with contraction restraints

FIGURE 4. SCHEMATIC REPRESENTATION OF UNDAMPED SPRING SYSTEMS

(a) Two preloaded, opposed, coulomb damped, linear springs with contraction restraints

(b) Two preloaded, opposed, coulomb damped nonlinear springs with contraction restraint

FIGURE 5. SCHEMATIC REPRESENTATION OF DAMPED SPRING SYSTEMS

a cushioning spring with about 1 inch of travel was added to take up the blow. Thus, the force displacement curve for each pair of springs was more nearly as shown in Figure 6.

The analysis of the suspension system was performed by examining the performance of the damped-spring system in absorbing the kinetic energy of the canister. The nonlinear spring force curve was determined from geometric considerations of the spring cable at various displacements. Analyses were performed for displacements in the longitudinal and transverse directions. The spring pairs used for these analyses are shown in Figure 7. Initial studies were performed using the SHOCK⁽²⁾ computer program.

The SHOCK program is a generalized program able to analyze the dynamic response of a lumped mass system connected by springs and frictional devices when subjected to a variety of external forces including gravity, shock, and vibration. The model used is shown in Figure 8. As shown, the frame was divided into 6 lumped mass nodes connected by linear springs. The spring properties were calculated based on the elastic properties of the frame structure. The canister was divided into four nodes also connected by linear springs which represent the elastic properties of the canister. The end nodes of the canister were connected to the frame nodes by nonlinear springs and friction dampers. For purposes of analysis, it was assumed that the static friction of the dampers is equal to the dynamic or sliding friction. Although this is not true, there was no way of accurately estimating the static friction. The static friction force theoretically does not produce any energy absorption since no motion occurs. Thus, neglecting the static friction force would only affect the impact force at $T \cong 0$. It was estimated that in the worst case, the static friction would be no greater than twice the dynamic friction. Since the preliminary analyses indicated the initial forces to be in the range of 1 to 1-1/2 g (assuming only dynamic friction) it was considered an acceptable simplification to the analysis to not consider the effect of the static friction.

Preliminary analysis indicated that an external "cushion" was desirable to absorb some of the kinetic energy of the frame and reduce the "bounce" upon impact. Consequently, lead pads were added to the corners of the frame. These are represented by the compression-only, hysteresis spring couplings shown between the frame and the fixed masses in Figure 8.

FIGURE 6. SCHEMATIC REPRESENTATION OF DAMPED SUSPENSION SYSTEM USED ON THE F-144-1 SHIPPING CONTAINER

(a) longitudinal motion

(b) transverse motion

FIGURE 7. SCHEMATIC ILLUSTRATION SHOWING SPRING PAIR COMBINATIONS CONSIDERED FOR LONGITUDINAL AND TRANSVERSE MOTION

Legend:

Canister Nodes

Frame Nodes

Linear Springs

Nonlinear Springs

Coulomb (friction) Dampers

Compression only spring with hysteresis

FIGURE 8. MODEL OF F-144-1 SHIPPING CONTAINER USED FOR INITIAL STUDIES OF TRANSPORTATION VIBRATION AND 48-INCH FALL ANALYSES

The model was analyzed for impact in both the axial and side orientations and for the most critical vibration frequency in the side orientation. For the impact condition, the F-144-1 shipping container was assumed to fall 48 inches to a hard unyielding surface as defined in the work statement of the program. For the vibration condition, the random vibration limits for leaf springs from Reference 3 were examined at frequencies from 1 to 30 Hz, Figure 9. The maximum transmissibility for the coulomb damped spring system was found to occur at a frequency of about 5.3 Hz.

The results of the analyses indicated that the 48-inch free fall incident was a more severe condition than vibration. It was, therefore, possible to use a second program, LSPG, which was specially written to calculate the response of the canister under the 48-inch free fall condition. This program only calculated the first shock peak and did not analyze for the subsequent lesser rebound shocks. Comparison of the results of the LSPG program with those of SHOCK indicated agreement to within 5 percent. Therefore, after the basic configuration and component parameters were established using SHOCK, refinements were made using the LSPG program which was more efficient for this problem and required much less running time on the computer.

The curves presented in Figures 10 through 16 are typical of the response of the F-144-1 container to fall and vibration as calculated by the SHOCK program. It should be noted that these results were obtained from the SHOCK program prior to final refinement of the design. Therefore, they are approximate only but are considered of sufficient accuracy and interest to be included in this report.

Table 2 presents the properties of the suspension system assumed for the analytical results presented in Figures 10 through 16.

TABLE 2. ASSUMED PROPERTIES OF SUSPENSION SYSTEM COMPONENTS
FOR ANALYTICAL RESULTS SHOWN IN FIGURES 10 THROUGH 16.

Component	Property
Suspension Spring	18 lb per inch spring rate 432 lb preload (in null position)
Cushion Spring	432 lb per inch spring rate 0 lb preload 1 inch maximum travel
Coulomb Damper	60 lb "drag" in cable

FIGURE 9. MAXIMUM SHOCK AND VIBRATION ENVELOPES FOR VERTICAL AXIS AT CARGO FOR THE TWO BASIC TRUCK SUSPENSION SYSTEMS

FIGURE 10. LINEAR ACCELERATION OF CANISTER DURING 48 INCH FALL OF THE F-144-1 SHIPPING CONTAINER ON THE END

FIGURE 11. LINEAR VELOCITY OF CANISTER RELATIVE TO THE UNYIELDING SURFACE DURING 48 INCH FALL OF THE F-144-1 SHIPPING CONTAINER ON THE END

FIGURE 12. RELATIVE LINEAR DISPLACEMENT BETWEEN CANISTER AND FRAME
DURING 48 INCH FALL OF THE F-144-1 SHIPPING CONTAINER ON THE END

FIGURE 13. LINEAR ACCELERATION OF THE CANISTER DURING 48 INCH FALL OF THE F-144-1 SHIPPING CONTAINER ON THE SIDE

FIGURE 14. LINEAR VELOCITY OF THE CANISTER RELATIVE TO THE UNYIELDING SURFACE DURING 48 INCH FALL OF THE F-144-1 SHIPPING CONTAINER ON THE SIDE

FIGURE 15. RELATIVE LINEAR DISPLACEMENT BETWEEN CANISTER AND FRAME
DURING 48 INCH FALL OF THE F-144-1 SHIPPING CONTAINER ON THE SIDE

FIGURE 16. LINEAR ACCELERATION OF THE CANISTER DURING TRANSPORTATION
VIBRATION OF THE F-144-1 SHIPPING CONTAINER. FRAME VIBRATING AT 5.3 HZ
WITH MAXIMUM APPLITUDE OF APPROXIMATELY + 0.65 G to - 0.93 G.

All of the fuel-element-canister acceleration curves, Figures 10, 13, and 16, include the acceleration due to gravity. In the case of the velocity and acceleration curves, positive is the direction away from the unyielding surface. In the case of the canister displacement curves, positive is measured from the null position toward the unyielding surface.

Figure 10 shows that after the initial impact the force on the canister reaches a peak of about +6.2 g after about 0.09 seconds, at which time the velocity of the canister is reversed, Figure 11. An oscillation then becomes apparent which reaches a peak at about 0.13 seconds after impact. At this point the force varies between +0.75 g and +9.6 g. It is then damped out so that at about 0.21 seconds, the oscillation has stopped and the impact force is about 2.5 g. In reviewing the computer printout, it was noted that up to the onset of this extreme oscillation, the impact loads vary by as much as 4.5 g between adjacent nodes as the shock wave travels from one end of the 6-inch diameter pipe to the other and returns.

However, during this period the average impact force for the four nodes is below 6 g and often below 5 g. As an example, at about 0.13 seconds when the amplitude of the acceleration of the canister node plotted in Figure 10 changes rapidly from +0.74 g to +9.4 g, the average acceleration for all four fuel element canister nodes is +5.03 g.

The extreme oscillation is believed to be a "ringing" of the 6-inch diameter pipe at its natural frequency. This cannot be definitely confirmed, however, since the data printed out and plotted by the computer are for the conditions existing at finite time intervals. The time intervals used are too coarse relative to the natural period to assure that each vibration cycle of the "ringing" can be discerned. Figure 10 suggests that the ringing is initiated immediately upon impact but is damped out before the peak force of 6.2 g is reached at 0.09 seconds. At this point a second more violent oscillation occurs. By close examination of the curve in Figure 10 the "ringing" frequency as plotted is measured to be about 400 to 450 Hz. It should be noted, however, that due to the coarseness of the plot this is probably low and the true frequency is likely greater. The calculated natural frequency of the 6-inch diameter pipe neglecting the effect of the damped suspension springs is about 540 Hz. This is considered

relatively good agreement with the value measured from Figure 10 in view of the difficulty in measuring the frequency from this curve and since the calculation of 540 Hz is simplified and neglects the interaction of one node on another as well as the effect of the suspension system.

After the "ringing" is damped out at about 0.21 seconds the acceleration then rapidly reverses sense as the canister moves past the zero position relative to the frame, Figure 12. The maximum acceleration in the reverse direction (downward) is approximately 6.7 g after 0.276 seconds at which time the ringing phenomenon again occurs. It is less severe, however, than the "ringing" after the initial impact and is damped out more rapidly. It should be noted that the 6.7 g includes the normal 1 g gravitational force.

The rate at which the velocity of the canister is reduced by the coulomb damped spring suspension system is illustrated in Figure 11. Figure 12 indicates that the maximum relative displacement between the canister and the structural frame for the axial impact orientation is about 12 inches. On the first rebound the amplitude was reduced to -6 inches. The irregularity of the relative displacement curve of Figure 12 at about 0.12 seconds is attributed to a second contact of the frame with the unyielding surface after the first "bounce". The computer analyses indicated that the frame "bounced" approximately 1-1/2 inches off the unyielding surface after the initial impact.

Figure 13 presents the deceleration loads on the canister for impact on the side of the frame. The acceleration pattern of the canister for this orientation is much different from that for the axial impact orientation. Although the canister again enters a "ringing" vibration mode immediately upon impact its vibration period is much greater than for axial impact since the canister is effectively a beam simply supported at its ends and uniformly loaded along its length. The ringing amplitude varies considerably at first but is significantly reduced by the time the peak acceleration of +5.7 g is reached after about 0.108 seconds. A second phase of extreme ringing then begins as the velocity is reversed as shown in Figure 14 with large amplitudes again being experienced although not as great as after the initial impact. The second phase of extreme ringing is again reduced until, at about 0.34 seconds, a third phase is again experienced which is, as expected, less extreme than at the onset of the second phase. As noted for the case of the axial impact, although the curve

of Figure 13 (showing the acceleration for one of the four nodes used to model the canister) indicates a maximum acceleration of about +7.3 g at the onset of the second phase of extreme ringing, the average for the four nodes used in the model was less than +6 g at all times. The "ringing" frequency shown in the curve of Figure 13 was measured to be about 82 Hz. The natural frequency of the canister was calculated to be 58 Hz. This is considered reasonable agreement since the effect of the interaction of adjacent nodes and the suspension system was neglected as before.

Figure 14 shows the rate at which the velocity of the canister is reduced. The amplitude of the velocity curve is about the same as for the case of the axial impact orientation although the period is 15 percent higher.

The relative displacement between the fuel-element canister and the frame is presented in Figure 15. The maximum displacement after the initial impact is about 13.6 inches and the maximum rebound displacement is about 7.0 inches. The slight irregularities noted in the displacement curve at about 0.065, 0.115, 0.16, and 0.20 seconds are again due to "bouncing" of the frame on the unyielding surface. The maximum rebound height of the frame is about 1.25 inches.

The curve of Figure 16 is the response of fuel-element canister to normal truck-transportation vibration. The vibration input to the frame is presented in Figure 17. The SHOCK computer program did not lend itself to simple insertion of the 0.45 g vibration acceleration desired on the frame. Consequently, the desired force amplitude input to the program had to be estimated by successive trials. As shown in Figure 17, a maximum acceleration of +0.65 g to -0.93 g was achieved on the last trial attempted. Although this is greater than the requirement of 0.45 g, the curve of Figure 16 shows that the acceleration on the fuel-element canister varied between +2.2 and -3.4 g. Since this is less than the 6 g design criterion and significantly less than the loads indicated by the analyses for the 48 inch fall condition, further analyses were not warranted. Although the acceleration of gravity was considered acting on the fuel element canister, it was not considered to act on the frame since the frame is assumed firmly attached to the vehicle bed. Thus although a difference of about 1 g is evidence between the absolute values of the maximum and minimum acceleration on the canister, the absolute difference in the acceleration of the frame, Figure 17, is only about 0.3 g.

FIGURE 17. LINEAR ACCELERATION OF FRAME OF THE F-144-1 SHIPPING CONTAINER FOR NORMAL TRANSPORTATION CONDITION

Subsequent to the above analyses, further refinements were made using the LSPG program. From these analyses the final component parameters as shown in Table 3 were determined. The calculated impact forces and displacements experienced by the canister with those components is presented in Table 4.

TABLE 3. PROPERTIES OF SUSPENSION SYSTEM COMPONENTS FOR
FINAL DESIGN OF F-144-1 SHIPPING CONTAINER

Component	Property
Suspension Spring	21.5 lb per inch spring rate 430 lb preload (in null position)
Cushion Spring	430 lb per inch spring rate 0 lb preload 1 in maximum travel
Coulomb Damper	246 lb "drag" in cable

TABLE 4. CALCULATED RESPONSE OF FINAL DESIGN OF F-144-1
SHIPPING CONTAINER SUSPENSION SYSTEM TO 48-INCH
FALL INCIDENT

Impact Orientation	Fuel Element Canister Response
End	5.8 G impact force 10.9 inch displacement
Side	5.1 impact force 13.2 inch displacement

Structural Analyses

In order for a shipping container to be licensed by the Department of Transportation, compliance with the requirements of the U. S. Code of Federal Regulations 10 CFR-Part 71 must be demonstrated. These regulations include a comprehensive list of structural design requirements. The analyses which were performed to show compliance of the F-144-1 shipping container were included in the Safety Analysis Report for this container. The results of those analyses are summarized in Table 5.

Thermal Analyses

As in the case of the structural analyses, 10-CFR-Part 71 of the U. S. Code of Federal Regulations specifies the thermal environment which a shipping container must withstand in order to be licensable. A summary of those analyses performed for the Safety Analysis Report is presented in Table 6.

Nuclear Analyses

The U. S. Code of Federal Regulations, 10-CFR-Part 71 stipulates the nuclear requirements which a shipping container must meet in order to be licensed by the Department of Transportation. They include both nuclear shielding and criticality requirements. Since the nuclear material proposed for shipment in the F-144-1 shipping container is unirradiated, virtually no biological shielding is required. Thus no shielding calculations were necessary. However, it was necessary to demonstrate the criticality condition for not only a single container but an array of similar containers. The results of these analyses performed for the Safety Analysis Report are summarized in Table 7.

TABLE 5. RESULTS OF THE STRUCTURAL INTEGRITY ANALYSIS

Requirement	Results	Margin of Safety (if applicable)
<u>General Standards</u>		
No internal reaction	Materials of construction do not react	
Positive closure	Bolts provide closure, closure under accident conditions is most severe requirement and is shown below	
<u>Lifting Device</u>		
Support three times loaded weight		
Lifting in horizontal position	3,800 lb per bolt	0.58
Raising from horizontal position	2,685 lb per bolt	1.23
Lifting in vertical position	5,370 lb per bolt	0.12
Support three times lid weight	Manual handling, no lifting, attachments provided	
Failure of lifting device would impair containment	Less severe than accident condition	
<u>Tiedown</u>		
No yield with 10, 5, and 2G forces	Frame to be held to vehicle by straps, no tiedown devices provided	
Failure of tiedown device would not impair other requirements	Less severe than accident conditions	
<u>Container Body</u>		
Load resistance (as a simple beam)		
container frame	2,860 psi	large
canister	8,990 psi	2.34

TABLE 5. (Continued)

Requirement	Results	Margin of Safety (if applicable)
External pressure of 25 psi		
Stress in end plates, bottom, edge	694 psi	large
Center	1,790 psi	large
Top	Less severe than bottom	
Stress in side shell	270 psi	large
<u>Normal Transportation Environment</u>		
Pressure of 0.5 times standard atmosphere	Less severe than 25 psi requirement of 71.32-b	
Vibration	0.85G Max. (less severe than free drop from 4 ft)	
Water spray	Corrosion resistant metals used in components; others to be painted with corrosion resisting paint	
Free drop from 4 ft		
Side orientation	5.1G	
Bolt shear	Less severe than for 30-ft free fall	
Simple beam	9,179 psi	2.27
Axial orientation	5.8G	
Bolt tension	490 psi	large
Cover plate fiber	2,475 psi	large
Flange weld shear	655 psi	large
Bottom plate weld	2,470 psi	large
Penetration of 13-lb bar, 1-1/4 diam. dropping from 4 ft	Energy which can be absorbed without penetration = 12,300 in-lb; bar has 624 in-lb of energy	large
Compression of five times loaded weight of 2 psi	Less severe than 25 psi re- quirement of 71.32-b	

TABLE 5. (Continued)

Requirement	Results	Margin of Safety (if applicable)
<u>Hypothetical Accident Conditions</u>		
30-ft free fall onto flat surface		
Top end impact	Impact limiter prevents deformation of container, g - 74G, seal remains intact	
Bottom end impact	Approx. 0.53 in. of deformation to canister, ~9% strain, welds remain intact	4.6
Side impact	229G at top end	
Load on cover bolts	5020 psi per bolt	7.4
Load on canister	Welds remain intact	
Top and bottom oblique impacts	Less severe than end or side impacts	
Side oblique impact	Less severe than side impact	
Puncture by 40-in. fall on 6-in. diam. bar		
Side fall orientation	Direct puncturing blow to canister prevented by frame structure and suspension mode of canister	
End fall, top	Energy absorbed by impact absorber	0.95
End fall, bottom	Kinetic energy absorbed with 0.0418 inch of penetration. Plate is 0.5 in. thick.	large
Containment for 30-ft free fall condition	Canister integrity maintained, analysis of fuel element integrity not required	

TABLE 6. RESULTS OF THE THERMAL ANALYSES

Analysis Conditions	Results	Evaluation
130 F ambient, direct sunlight, still air	Contents at ambient temperature	No damage to fuel assemblies
-40 F ambient, direct sunlight, still air	Contents at ambient temperature	No damage to fuel assemblies
Heat flux from 1475 F fire with 0.9 emissivity for 30 min., no artificial cooling after "fire" exposure until temperatures at center of package begin to fall; cask surface with 0.8 emissivity	Max. fuel temp: Approx. 250 F with side panels on frame; approx. 210 F with side panels removed.	No damage to fuel assemblies; allowable fuel element temp. = 1250 F; No gas generation from plastic wrapping; Integrity of seal maintained.

TABLE 7. RESULTS OF CRITICALITY ANALYSIS

Analysis Conditions	Results	Evaluation
Arrays of undamaged packages in most reactive credible configurations. Minimum number of packages which must be subcritical is 25.	Critical number of packages is 51	Shipment of 4 packages meets requirements for normal conditions
Array of 8 packages under hypothetical accident conditions having close reflection and optimum interspersed moderation.	K_{eff} for array of 8 packages is $0.70 \pm .03$	Shipment of 4 packages meets requirements for hypothetical accident conditions

PROTOTYPE EVALUATION PROGRAM

In order to verify the design developed during the detailed analysis phase of the development program, a subsequent program was undertaken to fabricate and drop test a prototype model of the F-144-1 Shipping Container. Also included was the fabrication of two other units not to be tested but which included any configuration or component modifications indicated by the drop tests.

Prototype Fabrication

A complete set of detailed fabrication drawings were prepared and submitted to the Machine Shop of Battelle's Columbus Laboratories for fabrication. A rigorous quality control program was followed to assure complete compliance with all material and fabrication specifications and to insure a high quality of workmanship. The history of the fabrication has previously been submitted to E. I. duPont in the quality assurance report⁽⁴⁾.

Drop Test Program

In collaboration with representatives of E. I. du Pont a drop test program was developed. It included a series of drop tests from various heights and orientations as indicated in Table 8. High and medium speed photography as well as accelerometer instrumentation were included to permit evaluation of the container performance.

TABLE 8. SCHEDULE OF DROP TESTS OF THE
F-144-1 SHIPPING CONTAINER

Test No.	Type	Similar Test Specification	Test Description
1	Slant, cal	-	One end of container on ground, other end dropped from 12 in. (Used for verification of accelerometer operation.)
2	Side	-	Container dropped in horizontal orientation from 48 in.
3	Edgewise	MIL-STD-810A Procedure III, Step B	One end of container on 5 to 6 in. sill, other end dropped from 24 in.
4	Cornerwise	MIL-STD-810A Procedure III, Step C	At one end, one corner on 5 to 6 in. sill, other corner on 12 in. sill; other end (lower corner) dropped from 24 in.
5	End	-	Container dropped in vertical orientation from 48 in.
6	Slant, 4 ft	-	One end of container on ground, other end dropped from 48 in.
7	Slant, 6 ft	-	One end of container on ground, other end dropped from 72 in.

In order to obtain as nearly as possible a hard, unyielding surface on which to drop the container a drop pad was prepared on a several year old asphalt drive which has had limited use. The base under the asphalt is hard pack clay and gravel. On top of the asphalt, 1-inch-thick steel plates were layed with damp sand used as a grout between the plates and the asphalt.

The container was lifted for the various drops by a mobile crane. Previous experience indicated that for objects of this weight a quick release device such as an explosive bolt is not necessary. Negligible error is introduced by instantaneously releasing the brake on the cable drum of the crane and allowing it to "free wheel". In this way, the cable remains attached to the test object and the crane operator can regain control of the object if it should begin to fall or roll after the initial impact. (Examination of slow motion films of the drop tests of the F-144-1 container indicated that the actual terminal velocity of the container was equal to the theoretical terminal velocity, thereby verifying the acceptability of the release method.)

Drop Test Performance

The drop tests were performed as shown in Table 8 with the exception that Test No. 2, the side drop test, was repeated because of difficulty with the accelerometer read-out equipment on the first test. In general the container responded as expected. Upon impact the container was displaced within the frame returning immediately and smoothly to near the null position. The canister appeared to reach the null position and come to a halt. (Subsequent examination of the motion pictures indicated that for the tests from the higher drop heights the canister actually returned past the null position a short way, before reversing itself again and coming to rest near the null position.) The approximate displacement of the canister relative to the frame was estimated for each test by noting the amount of slack produced in a string tied to the canister and passing under a piece of tape on the frame.

During the first side drop test, Test No. 2a, four cables came off the pulleys on the torque limiters (damping devices). It was apparent that the retaining bars designed to prevent this would have to be moved on eight of the sixteen pulley brackets. The cables apparently came off when those cables became slack as the canister was displaced from the null position. In an effort

to prevent this from happening without suspending the testing until the retaining bars could be moved, screen door springs were attached to each cable to help keep it taut on the pulley regardless of the canister displacement. The effect of these springs was considered negligible compared to the massiveness and force developed by the suspension springs. This remedy worked fairly well; only one cable came off in subsequent tests.

The visual observations of each test are presented in Table 9.

TABLE 9. VISUAL OBSERVATIONS OF DROP TESTS

Test No.	Type	Observations
1	Slant, Cal.	No apparent damage.
2a	Side	9-3/8 displacement, 2 cracked welds between bracket assembly and spring tubes, flat straps forming struts on vertical sides bent ~1/2 to 1 in. out of line, 4 cables "popped" off torque limiter pulleys, lead feet deformed about 1/8 in.
2b	Side	Two more cracked welds between bracket assembly and spring tubes, slightly greater bending of straps forming struts on the vertical side, flat straps on top side to which spring tubes were welded bowed inward about 1/2 to 1 in., one cable popped off, lead feet on bottom side loose but securely in place.
3	Edgewise	2-1/2 displacement, no apparent additional damage.
4	Cornerwise	3-1/4 in. displacement, lead pads on impacting edge looser but still securely attached, no other additional damage.
5	End	Approximately 10-inch displacement, no additional damage.
6	Slant, 4 ft	No additional damage.
7	Slant, 6 ft	No additional damage.

Test Data Evaluation

Motion Picture Examination

Review of the motion pictures indicated that in general the container performed as anticipated. Upon impact of the frame, the canister continued to move and was slowed down in a much gentler manner. It was stopped before it touched the frame and was then returned to the null position by the spring suspension system.

It appeared from visual observations during the tests that the canister returned to the null position and stopped. The motion pictures indicated that for the drops from 48 inches the canister in returning after the initial displacement carried past the null position perhaps an inch or two before returning to the null position and stopping. This is in general agreement with the curves of Figures 12 and 15. Those curves indicate as much as 7 inches of travel past the null position after the return from the initial downward displacement. It should be noted, however, that those calculated curves were for a preliminary design in which the coulomb damping force was only 25 percent the force to which the damping devices were adjusted for this test. A lesser "overshoot" on the return thus would be expected.

As originally designed, the spring tubes were attached to the top and bottom cross struts of the frame with the pulleys on the torque limiters located in the side corners. The motion pictures indicated that the inertia force of the spring tubes caused severe deflection of the struts. This action was considered to be directly responsible for the deformation observed in both the top and side struts as well as contributory to the cracked welds on the bracket assemblies.

It had been planned to use the high speed photographic film to obtain time-displacement curves for the various tests and thus, by differentiation, obtain the deceleration of the canister. However, due to poor weather conditions on the day of the test, the high speed films were of a poor quality and time-displacement data could not be obtained. Medium speed cameras were also used and, although they were not as strategically located as the high speed camera,

it was possible to obtain a reasonable time-displacement curve for one test, the end drop test, Test No. 5. This curve is presented in Figure 18.

The maximum displacement as indicated by the curve of Figure 18 is 1.02 ft or 12.2 inches. The LSPG program predicted 10.9 inches. Regression analysis was used to fit the curve of Figure 18 to a polynomial equation of the sixth degree. The equation was then differentiated twice to obtain the equation for the acceleration. This is plotted in Figure 19. This curve indicates that the maximum deceleration force is 6.62 g. The LSPG program predicted 5.8 g.

It should be noted that polynomial equations up to the eighth degree were examined in fitting an equation to the curve of Figure 18. Upon differentiating these equations twice to obtain the acceleration curve, it was discovered that the initial and final portions of the resultant acceleration curves are highly sensitive to the changes in polynomial coefficients for different degree equations. The center portions of the acceleration curves for the higher degree equations were almost identical, however. The sixth degree equation was selected based on engineering judgement as to the most likely shape of the acceleration curve at the "ends".

The shape of the curve of Figure 19 is somewhat similar to the calculated curve of Figure 10 although some dissimilar features exist and are worthy of note. The curve of Figure 10 was calculated for a model in which the spring constant was only 84 percent of the spring constant on the prototype and the damping force was only 25 percent the force on the prototype. This could easily account for the shift in the time scale. In addition, the analytical model assumed lumped masses connected by springs. Moreover, the model was unidirectional (all spring properties and forces acted along a vertical line of action) since the computer program did not permit allowance to be made for the influence of lateral deflection of the frame and spring tubes.

In the case of the test results, it should be noted that they, of course, are subject to the test conditions and condition of the test object. The end drop test was the fifth test performed and, as noted above, several welds were cracked earlier in the test series. Moreover, the spring tube supports were bent. These factors undoubtedly had some influence on the test results although from a detailed review of the motion pictures no anomalies were noted in the end drop test.

FIGURE 18. TIME-DISPLACEMENT CURVE FOR END DROP TEST OF F-144-1 SHIPPING CONTAINER

FIGURE 19. ACCELERATION-TIME CURVE FOR END DROP TEST OF F-144-1 CONTAINER

The curve of Figure 19 indicates that the initial force at $T = 0$ was about 3 to 4 g decreasing to about 1.5 g before increasing then to a maximum of 6.62 g at $T = 0.074$ seconds. The high initial force which decreased during the first 0.018 seconds is most likely the result of the sensitivity of the method of the double differentiation of the time displacement equation. However, a spike in the acceleration curve would be expected at the beginning of the displacement since the static friction of the damping devices would have to be overcome. It would normally be expected that the maximum force would occur at the same time as the maximum displacement. However, the curve of Figure 19 indicates that the maximum force of 6.62 g occurred about 0.036 seconds before the maximum displacement. The apparent reason for this phenomena is the flexibility of the support system.

Accelerometer Data Evaluation

Three strain gage accelerometers were mounted on the canister to monitor acceleration levels during the drop tests. A consolidated Electro-dynamics (CEC) signal conditioning and recording system was used, consisting of strain gage carrier amplifiers and a light-beam galvanometer recorder, to provide a frequency response flat to about 300 Hz. A diagram of this system is shown in Figure 20. Accelerometers were mounted with sensitive axes vertical and longitudinal at the dropped (flange) end of the canister, and vertical at the midpoint of the canister.

Results of the acceleration measurements were inconclusive, to some extent due to the method of recording the data. Predominant frequencies of 38 Hz (apparently the first bending mode of the canister), 110-150 Hz, and 450-500 Hz were found to be excited by impact. Lower-frequency components were completely masked by these higher-frequency, high-amplitude accelerations. Typical response was recorded for the 48-inch side drop (No. 2). After the free-fall (zero g) period, a maximum vertical acceleration peak of 10.3 g at 38 Hz occurred at the flanged end at impact. This was followed by a 10 milli-second burst of higher-frequency components of up to 19 g superimposed on the 38 Hz oscillation. After several cycles of 38 Hz oscillation, a second burst of

FIGURE 20. INSTRUMENTATION SYSTEM FOR DROP TEST ACCELERATION MEASUREMENT AND RECORDING

higher frequencies up to 17 g in amplitude was noted. These acceleration bursts were probably associated with the action of the friction devices. Similar, but lower-amplitude accelerations were recorded vertically at the center of the canister, while higher-frequency accelerations on the order of 5 g were recorded longitudinally.

The edgewise drop (No. 3) also produced a peak of 10.3 g at 38 Hz, although somewhat lower amplitudes at high frequencies, which indicates that the peak acceleration at this frequency was more a function of the friction device setting (breakaway friction) than the retarding acceleration level. Displacements of the canister relative to the frame were approximately 9.4 inches for the 48-inch side drop, compared with 2.5 inches for the edgewise drop orientation as noted above.

The cornerwise drop test (No. 4) resulted in a less pronounced 38 Hz component, reaching a maximum of 7.1 g at impact, but again with high-amplitude, high frequency components. For the case of the end drop (No. 5) practically no 38 Hz component was noted, confirming that this component was indeed the bending mode of the canister. A strong 110-150 Hz component (up to 16.5 g) was recorded, as well as 400-500 Hz oscillations.

High acceleration levels recorded at frequencies above 100 Hz are of little consequence in the shipment of these fuel element containers since fuel elements usually have much lower natural frequencies. Accelerations at the bending frequency of the canister (38 Hz), however, might warrant further attention in the context of canister bending stresses. As the results indicate, the acceleration peaks in the range of 10.3 g on impact seem to be associated with friction breakaway force levels, which may be adjusted or may decrease with friction material wear.

CONCLUSIONS

The following conclusions are made from the results of the analytical and experimental phases of the program:

(1) A design of a shipping container for FFTF-type fuel was developed which met the size and environmental response conditions set forth by the program. This shipping container design extended the state of the art for shipping fragile fuel systems.

(2) An ideal container which meets the high cushioning requirements needs to provide 8 inches of travel for the fuel pin canister. In a practical working container design in which no external power assistance is provided, the canister will need to move about 10 to 12 inches.

(3) A practical working container was designed using commercially available components, i.e., component development was not required.

(4) Evaluation of drop tests of a prototype device indicated the following.

- (a) The prototype performed during the drop tests nearly as predicted.
- (b) Some high stress areas were noted which require moderate reinforcing.
- (c) Orientation of the container such that the spring tubes are on the vertical sides of the frame is preferred to an orientation with the spring tubes on the bottom and top sides.
- (d) Two locations of cable retainers are required depending on the angle at which the cable leaves the pulley of the torque limiter.
- (e) A reduction in the torque setting of the coulomb damping devices (torque limiters) was indicated in order to reduce the impact force of the canister (especially the force associated with the "breakaway" force of the torque limiters) to within the design criteria.
- (f) Although impact forces several times higher than the design force of 6 g were indicated by the accelerometers during the impact tests, these forces are associated with high frequency vibrations having

relatively low energy levels. Since these frequencies are well beyond the range of natural frequencies for fuel elements, the results of these forces are considered inconsequential.

DESIGN MODIFICATIONS INDICATED BY THE
RESULTS OF THE DROP TESTING PROGRAM

As a result of the evaluation of the test results the following design modifications were made:

- (1) The normal transit orientation of the frame was rotated 90 degrees so that the spring tubes are attached to the vertical sides of the frame rather than the top and bottom. Additional bracing was not added, however, because the flexibility of the spring tube supports is considered an asset.
- (2) The welds which cracked were short (approximately 1/2 inch long), welds at the juncture of the spring tube to the horseshoe shaped bracket in which the friction device is mounted. A plate was welded between the spring tube and the exterior side of the bracket to provide more strength in the joint between these two components.
- (3) In order to reduce the impact force below 6 g the damping devices were adjusted so the frictional force was halved. It was calculated that the slight increase in displacement could be accommodated.
- (4) The lead pads were redesigned to reduce the tendency for loosening upon repeated blows.

REFERENCES

- (1) Safety Analysis Report for the Shipment of FFTF Fuel in the F-144-1 Shipping Container from Battelle Columbus Laboratories, revised May 25, 1972.
- (2) Gabrielson, V. K., and Reese, R. J., Shock Code User's Manual, A Computer Code to Solve the Dynamic Response of Lumped-Mass Systems, SCL-DR-69-98, Sandia Laboratories, Livermore, California, November 1969.
- (3) Ahlbeck, D. R., Summary Report on "Study to Define the Shock and Vibration Environment During Truck Transport of Shipping Containers", to E. I. du Pont de Nemours and Company, Inc., December 22, 1971.
- (4) Quality Assurance Report for Fabrication of F-144-1 Shipping Container, June, 1972.