N89-23895

THE DESIGN AND ANALYSIS OF A DOUBLE SWIVEL TOGGLE RELEASE MECHANISM FOR THE ORBITER STABILIZED PAYLOAD DEPLOYMENT SYSTEM

Guy L. King* and Ted Tsai*

ABSTRACT

A new NASA deployment system called the Stabilized Payload Deployment System (SPDS) will soon be operational. The lightweight and heavy-duty system rolls payloads over the orbiter's side rather than ejecting them upward. The system will enhance the orbiter capability of carrying larger and heavier payloads. This paper describes the design, function, and analysis of a new three-pin "double" swivel toggle release mechanism which is crucial to the successful development of the SPDS.

INTRODUCTION

The SPDS is being jointly developed by NASA/JSC and Rockwell International/Space Transportation Systems Division. SPDS will be used in the National Space Transportation System (NSTS) to permit on-orbit deployment of payloads independent of other payload handling equipment such as the Remote Manipulator System (RMS). The SPDS is a compact electromechanical system which attaches the payload to the orbiter through the payload retention structure (trunnions). The system is designed to rotate payloads out of the Orbiter payload bay at a predetermined angular position and effect a payload separation on command. This paper focuses on the two fault tolerant release mechanisms that play a key role in the critical payload separation. It describes in detail the design, function, and analysis performed on the two fault-tolerant double-swivel toggle mechanism that is held in place by a swivel ring and three pyro actuated retaining pins under a high elastic preload. Removal of one or more pins will instantly release the spring loading and subsequently cause the unstable swivel links to move away from the joint. Detailed design analogy of the mechanism is discussed along with the NASTRAN finite element and stress concentration analysis that was performed to investigate the hoop strength and the local yielding of the double swivel configuration. Structural stress contours (load paths) are presented and the overall description of SPDS is also included.

RELEASE MECHANISM DESIGN

The 82-kg (180-lb) SPDS (Fig. 1) is a bridge-mounted structure that can be positioned in any bay on the port or starboard longeron of the orbiter. The first application of SPDS is on the port side longeron replacing the RMS. The payload deployment and release sequence are shown in Figure 2. In the

*Mechanical Design and Analysis Branch, Structures and Mechanics Division, Lyndon B. Johnson Space Center, Houston, Texas.

PRECEDING PAGE BLANK NOT FILMED

SPDS deployment sequence, the first two steps are to insure that there is sufficient clearance for payload deployment. The motions of SPDS pedestals are controlled by actuators and drive motors. Once the payload reaches the final angle, the payload oscillations are damped through the spring damper (Fig. 1). After the payload is stabilized, it is released through a double swivel toggle release mechanism (Fig. 3) located within the release head. The major components of the release head consist of the swivel, the housing, and the pyrotechnic retractors. A circular plate which is connected to the payload interface plate is held to the housing by the toggle. The other end of the toggle is held in place by three pyrotechnic pins. On command, the three pins will retract and release the toggle. The final design configuration of the mechanism assures free swiveling at both ends of the toggle. A cross section view of the swivel toggle mechanism is shown in Figure 4. The double swivel toggle is made of three main components: the swivel bolt (with the top swivel ball), the swivel socket (with the lower swivel ball), and the swivel ring as shown in the figure. The bolt is fastened into the lower swivel socket. The inner surface of the ring is spherically contoured and fits on top of the lower ball. The pyrotechnic pins are 120 deg apart and the flat side of the pin tip rests on top of the swivel ring. The mechanism is held in tension by tightening the swivel bolt into the socket. The assembly is strain gauged to obtain an accurate reading on the 8007 Newton (1800 lb) design preload. When the pyrotechnic pins are fired, the pins retract and the circular payload plate is separated by an expulsion spring within the main housing. The elastic preload of the toggle provides the additional spring load for payload separation. The three-pin toggle release sequence is illustrated in Figure 5. During a nominal deployment, payload interface components including the swivel toggle stay with the payload.

The illustration in Figure 5 constitutes the heart of the design concept of the release mechanism. During the earlier development of the SPDS, the design was simply inadequate in the release mechanism. A three-pin toggle release mechanism design was then brought into the system. The one piece toggle had a single swivel on the upper end (Fig. 6). To assure the free motion of the toggle after the pyro initiation, another swivel was developed on the other end (the swivel ring) of the toggle. When the swivel toggle is properly preloaded, any moment or lateral loading from the payload interface plate will be directly transmitted through the housing of SPDS. The toggle itself will experience very minimum load variations. Before the pyro initiation the tension loaded toggle mechanism restrained by equally spaced pins constitutes a well balanced and stable loading pattern. The retraction of all pins will immediately release the toggle. If any one or two pins were to malfunction, the double swivel toggle would still permit clean separation because the swivels would rotate clear of the failed pin/pins. The toggle mechanism had been through several preliminary design configurations until the final double swivel was fully developed and chosen as the baseline. The original design had a one piece toggle with a sloped toggle/pin contact surface (Fig. 6). The pendulum type movement of the single upper swivel should have provided enough rotation to move clear the un-retracted pin/pins. The clearance created by the toggle swing motion will be the length of the toggle times the swing angle. The sloped toggle/pin contact surface was intended for easier toggle separation. However, when the mechanism was preloaded, the sloped surface of the pins brought on high local contact stresses. The sloped surface, which varied at times, also induced an undesirable axial pin reaction tending to retract the pins prematurely. To prevent the inadvertent retraction, a high shear retaining pin must be incorporated into each pyro pin. When tested, the configuration was locked up by the preload when only one pin was retracted. The lock-up was caused by high frictional forces in the single swivel ball and the tolerance-induced unequal pin load distribution. The design progressed to the double swivel but still keeping the sloped toggle/pin interface contact (Fig. 6). The configuration during the test did not hang-up but the force tending to retract the pins remained. Flat surfaces were finally incorporated into the toggle/pins interfaces (Fig. 6). The configuration eliminated the undesirable pin retraction forces and became the baseline. The new swivel at the lower end of the toggle was made possible by the creation of the swivel ring. The new design provides additional flexibility to the mechanism. The additional toggle movement with respect to the un-retracted pin/pins will be based on (1) the movement of the swivel ring about the swivel ball, (2) the geometrical outer contour of the ring, and (3) the relative pin/ring location. Most of all, the flexibility of the lower swivel will induce the swing motion of the toggle and eliminate the lock-up of the mechanism. Tests were performed with the absence of dry film lubricant to intentionally induce sticking surfaces around the swivels. No hang-up occurred. The mechanism with 8007 N (1800 lb) design swivel preload has also been successfully tested through the design thermal and vibration load environments. The 8007 N (1800 lb) preload proved to be adequate to keep the system joint intact with no premature joint separation. Design analysis was also performed to evaluate the strength of the new, small, and high performance mechanism. As a result, a minimum modification was applied to the bolt head. The modification was to assure that the toggle is elastically preloaded and no permanent local structure yielding would occur. The analysis in detail is discussed in the following section. The new two fault tolerant double swivel release mechanism became baseline for the deployment system and has been released for patent.

DESIGN ANALYSIS

A MSC/NASTRAN (The MacNeal-Schwendler Corporation/NAsa STRuctural ANalysis) finite element analysis is performed to evaluate the strength of the toggle elements. Structure stress contours are plotted. Structure load paths and strength are evaluated. The analysis reconfirms that the design concept of the mechanism is sound. A stress concentration analysis is also performed and a simple design modification is applied to the fillet of the swivel bolt head. The modification significantly increases the preload/fatigue strength and the reliability of the mechanism.

Design Preload

The double swivel mechanism is designed for the reliable payload deployment. A proper amount of preload applied to the swivel bolt is important. The abutment materials (housing and the attached plate) should always be in compression. In the meantime, the total bolt tension should not induce permanent deformations which would consequently release the elastic spring loading of the mechanism. For a normal operational environment, the load experienced by the toggle mechanism will mainly be the high preload. As long as the release head (preloaded joint) is in compression, the moment and lateral loads from the payload interface plate will be directly transmitted into the housing. There will be minimum tension loads applied to the release head. Based on the relative stiffness between the toggle and the abutment housing material, only a portion of the tension load will actually be transmitted into the preloaded toggle. Design analysis is performed to evaluate the load paths and the capability of the swivel mechanism based on the 8007 N (1800 lb) design preload.

Materials

The swivel bolt is made of MP35N (AMS 6884) bar. The swivel socket and the ring are Inconel 718 (AMS 5664) bar. In the range of the system operating temperature, which is from -73°C (-100°F) to 135°C (275°F), the high strength alloy have similar thermal expansion coefficients. At the highest 135°C (275°F) operating temperature, the MP35N will lose 10 percent of its room temperature strength and the Inconel 4 percent.

NASTRAN Finite Element Analysis

All three major components of the toggle mechanism (Fig. 4) are modeled (Fig. 7). All nominal dimensions are used. A cylindrical coordinate system (R, θ, and Z) was used for the model. The cylindrical geometry of the structure cannot be treated axisymmetrically because of the three localized pin reactions. However, with proper applications of mirror-imaged boundary conditions (constrained in the circumferential θ-direction at the RZ planes), only one-sixth (a 60-deg section) of the 360-deg circumference is needed to represent the entire swivel structure. The circular section is from the middle of a pin connection to half way to the next pin. NASTRAN three-dimensional solid elements (CHEXA/CPENTA) are used for the model. A simple inhouse preprocessor is developed to generate the math model. The model mesh size/density and element aspect ratios are arranged for proper model fidelity. The unconstrained model contains 7000 degrees-of-freedom (DOF). Boundary conditions are constrained in Z-direction at the lower pin/ring and upper bolt head/abutment interfaces (Fig. 7). The spherical surface contact between the ring and lower swivel ball are simulated with NASTRAN gap elements in the spherical radial direction. All gaps will be closed under the preload compression. It requires no iterative scheme to search for the load-induced gap contact or opening. A linear static analysis is performed. The results prove that all gap elements are in compression (closed). The bolt preload is simulated with fictitious thermal shrinkage of the material in the axial (Z) direction. The applied thermal load, which is a -111°C (-200°F) temperature differential, is randomly selected for the linear analysis. The computed total reactions at the boundary constraints will be the equivalent preload which produces the linear resultant stresses of the math model.

The computed results are processed to produce the structure deformed shape and stress contours (in forms of constant stress lines or color fringes) with the PDA Engineering/PATRAN (post-NASTRAN) processor. The exaggerated (dramatized) model deformed shape is shown in Figure 8. The total computed boundary reactions (equivalent swivel preload) is 7940 N (1785 lb) which is coincidentally very close to the actual design preload of 8007 N (1800 lb). For the linear static analysis, the displacements/stresses at the desired magnitude of applied loading are simply obtained by a linear ratio. The computed maximum tensile stress is 5.158×10^8 N/m^2 (74,806 lb/in.^{2}). The stress contours of the major principal stresses of the worst stressed (deformed) section are presented in Figure 9. High stresses and stress concentrations are clearly shown at the neck of the swivel bolt. The bolt neck has the minimum area for load paths with sharp change of the cross section from the bolt head. Stresses (mainly the hoop stresses) in the swivel ring are relatively low and uniformly distributed. The concern of the distortion and the separation of the swivel ring from the lower swivel is thus removed. Although the toggle is loaded at three localized pin/ring interfaces, the resultant loads (stresses) transmitted into the swivel, especially the bolt, are nearly axisymmetrical. The finite element analysis provides a good overall view of the structural load paths (stress patterns). The analysis reconfirms the overall design of the toggle mechanism. The critical link of the release mechanism is the swivel bolt which is subjected to the common stress concentration effect. The strength (preload allowable) of the mechanism will be based on the strength/shape of the bolt. A change to one or more of the geometrical/material parameters of the bolt (for instance, increasing the rounding radius at fillet or the radius of the neck) can rather easily increase the total strength of the entire mechanism. Analysis is now focused on the stress concentration of the swivel bolt.

Evaluation of Swivel Strength and Preload

To evaluate the highly localized and concentrated stresses, the finite element analysis will require additional local model refinement in a great detail at the fillet of the bolt. The complexity of the model will be further increased if the 20 node brick elements are to replace the current 8 node elements for better accuracy. The finite element solution is always considered an approximation to the usually unknown exact solution. Without the actual experimental data (say, photoelasticity), the accuracy level of detailed stress concentration analysis of the conventional h-version NASTRAN finite element method, which uses fixed low order polynomial element shape functions, will still be somewhat uncertain. No model refinement is performed. Experimental datum/formulations of stress concentration factors of available design configurations will be used to evaluate the bolt strength.

The diameter of the bolt shank is 5.03×10^{-3} m (0.198 in.). The cross section area A of the shank is 1.9864×10^{-5} m^2 (0.03079 in.^{2}). For a preload P of 7940 N (1785 lb), the average shank tensile stress σ_z (P/A) becomes 3.997×10^8 N/m^2 (57,972 psi). The NASTRAN computed maximum tensile stress is 5.158×10^8 N/m^2 (74,806 psi) which is at the neck of the bolt. If the maximum is divided by the average, the stress concentration factor K_t is estimated at

1.29. The term "stress concentration factor K_t" is loosely defined as the maximum local stress divided by the average stress in the bolt shank. The concentration factor of 1.29 is certainly unrealistic (and, in this case, low), because the math model does not include the necessary detailed refinement for the highly localized effect. Two configurations of the available textbook charts for stress concentration factors are selected for evaluation. The first is the "Round Shaft with Shoulder Fillet in Tension" and the second is the case of "T-head" as shown in Figure 10. The major difference between the two is the manner of loading. The loading of T-head will result in much higher concentrated stresses. As shown in Figure 10, the actual swivel bolt is compared to the two classical configurations. The configuration of the swivel bolt is less critical but closely resembles the T-head. The spherical contour of the bolt head will assist the line flows and reduce the local stress concentration. The stress concentration effect for the swivel bolt should be between the two configurations as the flow lines demonstrate in the figure. The existing NASTRAN swivel model is again utilized for the general comparison of three configurations. Because the load path into the bolt is nearly axisymmetrical, a simplified and axisymmetrical swivel model (a 10-deg section) is utilized (Fig. 11). First (as case 1), a uniform tension is applied to the top surface of the bolt head to simulate the round shaft with fillet in tension. Reactions at the pin-ring interfaces are computed. Next, for the case of the preloaded swivel, the spherical bolt head is constrained and the previously computed pin-ring interface reactions (from case 1) are re-applied at the same locations. Finally (case 3), the bolt head is constrained only at the flat shoulder surface of the bolt head as the worst case (proximity effect) of the T-head. The results (Fig. 11) clearly showed the highest stresses (stress concentration) for T-head and the lowest for the round shaft in tension. The design stress concentration factors for T-head will be conservatively used to evaluate the swivel strength. The design factors for T-head are available in R. E. Peterson's "Stress Concentration Design Factors." Although the T-head is of rectangular cross sections (with a constant thickness h), the design data is applicable to the head of a round bolt as discussed in the text. The major geometrical parameters for the T-head are the size (width D and depth m) of the head, the width of the shank (d), and the fillet radius (r). For a constant ratio of r/d, the stress concentration factors with respect to various D/d and m/d are provided. A total of four charts are available in the book for r/d = 0.05, 0.075, 0.10, and 0.20. Based on the geometry of the bolt head, the ratios of D/d (= 2.2), m/d (= 1.2), and r/d (=0.126) are defined (Fig. 12). By using the available datum of the four charts and the geometrical parameters of the bolt head (D/d = 2.2 and m/d = 1.2), a relation between the stress concentration factor (K_t) and the fillet rounding ratio (r/d) is developed in Figure 12. For the current swivel bolt configuration with the nominal ratio r/d of 0.126, the corresponding stress concentration factor becomes a high 3.5 based on the T-head configuration. Accordingly, for the 8007 N (1800 lb) design preload, the maximum stress will exceed the yielding allowable and the bolt head design is modified.

The swivel bolt is MP35N (AMS 5884) bar. In the range of the system operating temperatures (-73°C to 135°C), the minimum F_{ty} is 1.427 x 10^9 N/m^2

(207,000 psi) at 135°C (275°F): To consistently maintain the linear spring loading of the toggle, any localized yielding (permanent release of the spring load) is considered unacceptable. A 1.4 factor of safety is applied to the yielding (not the ultimate) allowable.

$$F_{ty}\text{, allowable} = 1.427 \times 10^9/1.4 \text{ N/m}^2$$
$$= 1.020 \times 10^9 \text{ N/m}^2 \text{ (= 147,857 psi)}$$

For the design preload of 8007 N (1800 lb), the average bolt shank stress is

$$\sigma_z\text{, average} = P/A$$
$$= (8007 \text{ N}) / (1.9864 \times 10^{-5} \text{ m}^2)$$
$$= 4.033 \times 10^8 \text{ N/m}^2 \text{ (= 58,495 psi)}$$

Based on the material yielding allowable with 1.4 factor of safety, the allowable stress concentration factor of the swivel bolt should not exceed

$$K_t = (F_{ty}\text{, allowable}) / (\sigma_z\text{, average})$$

$$= (1.020 \times 10^9 \text{ N/m}^2) / (4.033 \times 10^8 \text{ N/m}^2)$$
$$= 2.53$$

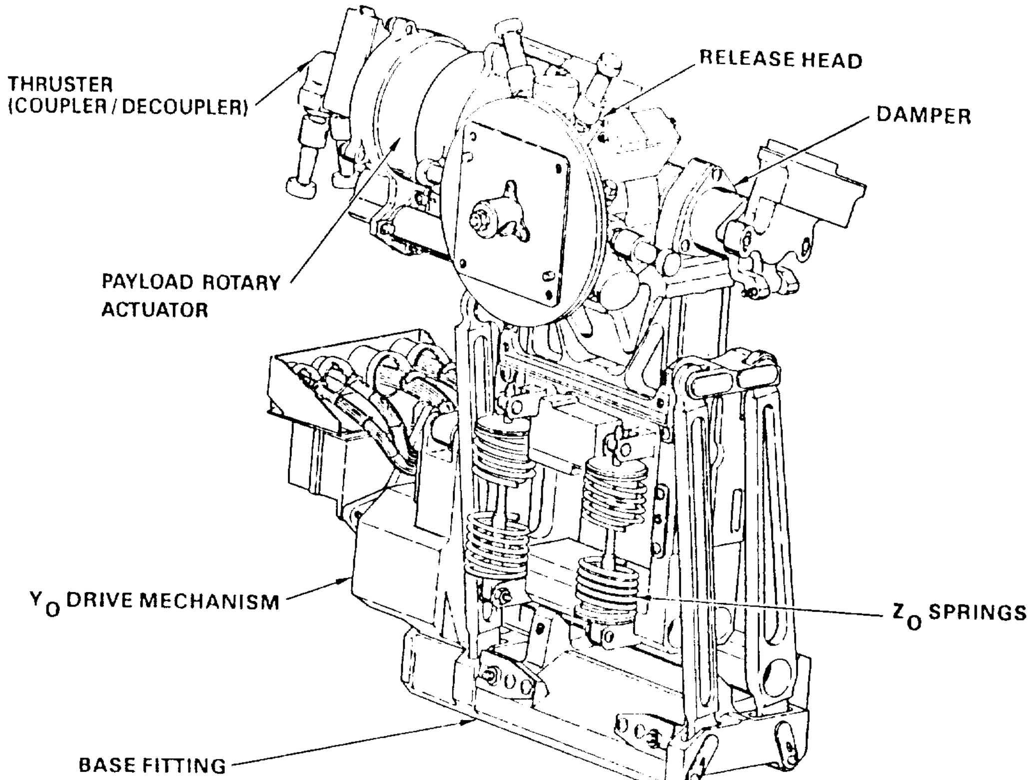
Going back to the K_t and r/d relationship in Figure 12, the corresponding r/d ratio for K_t equals to 2.53 is 0.26. In other words, if the shank radius d (5.0292×10^{-3} m) remains unchanged, the radius r (6.35×10^{-4} m) of the fillet rounding should be increased to avoid any local yielding of the bolt.

$$r = (d)(0.26)$$
$$= (5.0292 \times 10^{-3} \text{ m})(0.26)$$
$$= 1.31 \times 10^{-3} \text{ m (= 0.051 in.).}$$

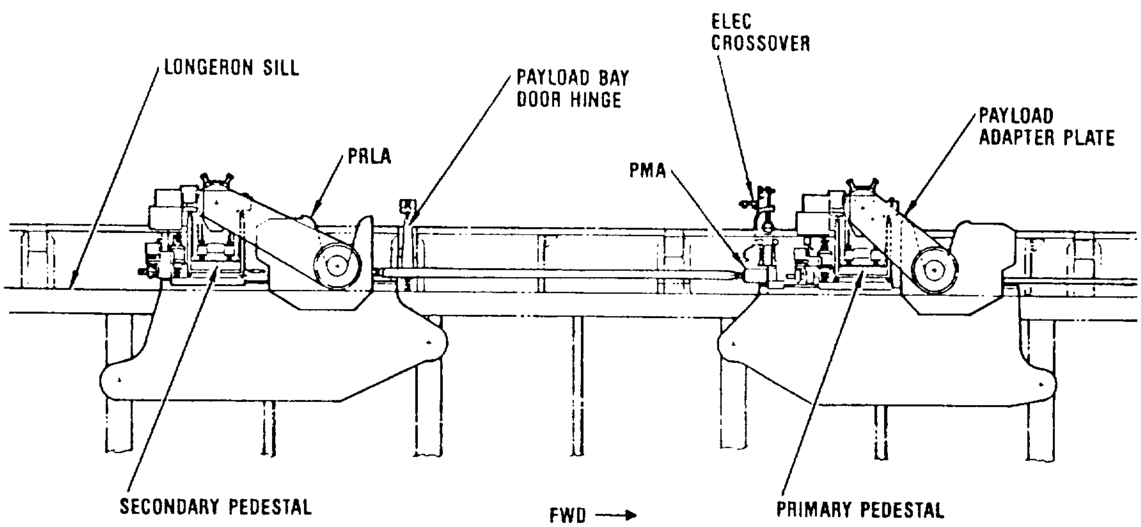
The 6.35×10^{-4} m (0.025 in.) fillet rounding radius of the original design was consequently changed to 1.31×10^{-3} m (0.051 in.). The minimum modification significantly increased the reliability and the static/fatigue strength of the mechanism.

CONCLUSION

A three-pin double swivel toggle release mechanism has been developed for the new SPDS. The two fault tolerant mechanism is small, lightweight, heavy-duty, and easy to assemble. The mechanism is innovative of its "double" swivel design. The design concept was verified by tests and analyses. The mechanism is designed for the long duration in space prior to the payload deployment. It is essential that the design be fully reliable. With the aid of the finite element analysis, the stresses in the mechanism were visualized. By simply modifying a geometrical parameter (the fillet rounding) of the swivel bolt, the toggle became insensitive to local stress concentrations and the system strength and reliability was significantly increased.



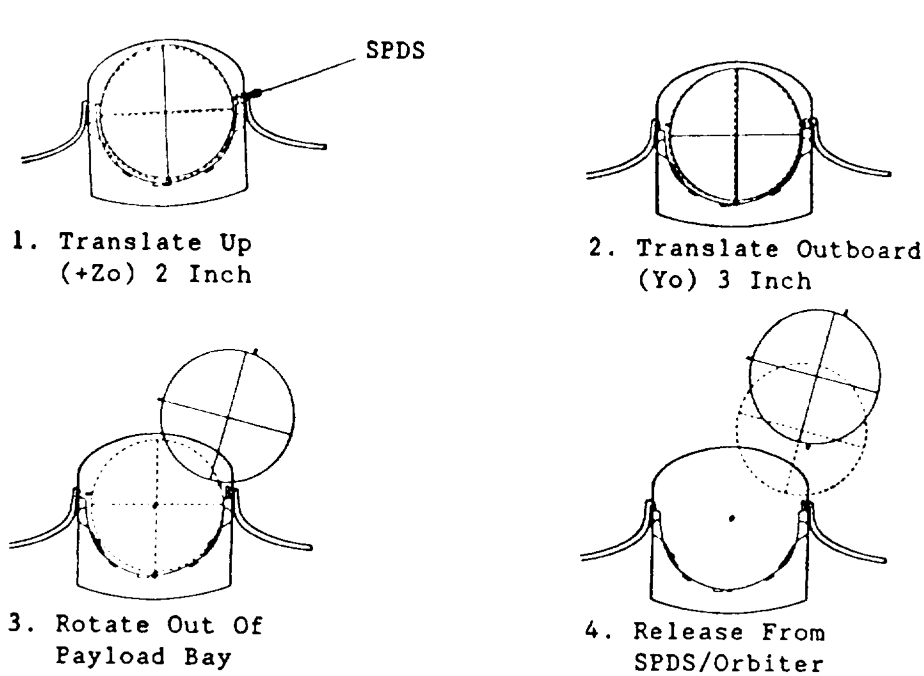
(SPDS)



ORBITER SPDS APPLICATION

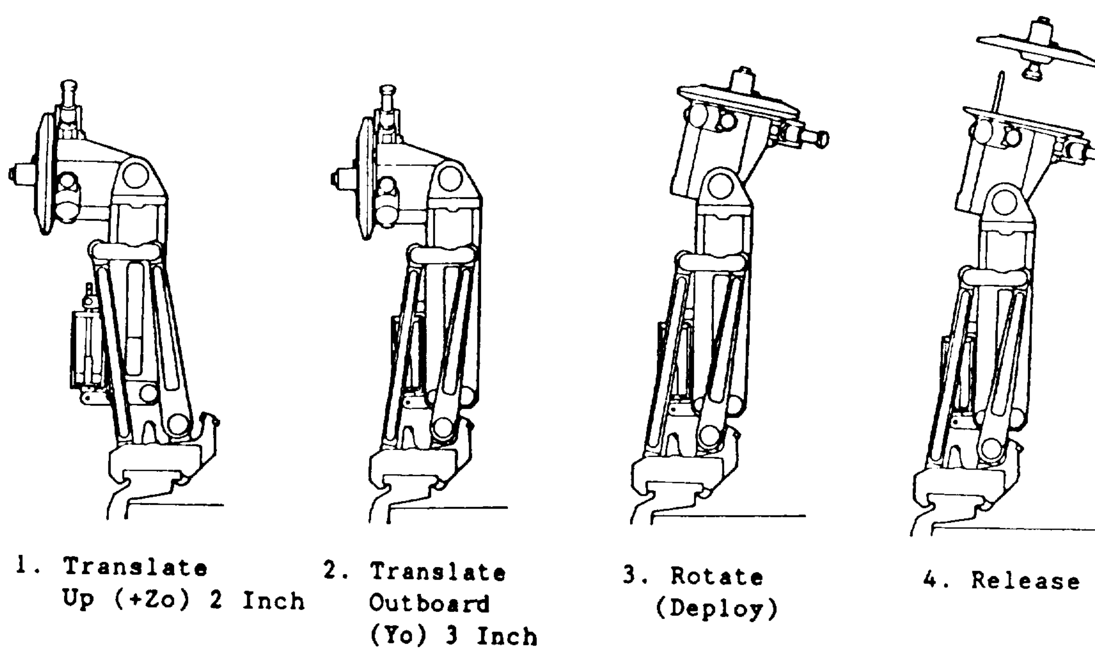
(PAYLOAD BAY VIEW LOOKING OUTBOARD PORTSIDE SHOWN)

Figure 1. Stabilized Payload Deployment System (SPDS).



PAYLOAD RELEASE SEQUENCE

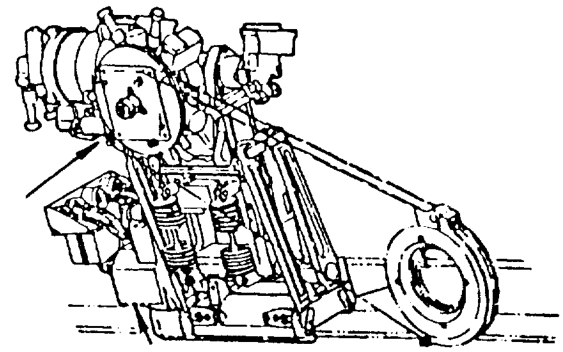
(ORBITER PAYLOAD BAY LOOKING AFT)



SPDS PAYLOAD RELEASE MOTION

(SPDS SIDE VIEW)

Figure 2. SPDS payload deployment and release sequence.



SPDS RELEASE HEAD

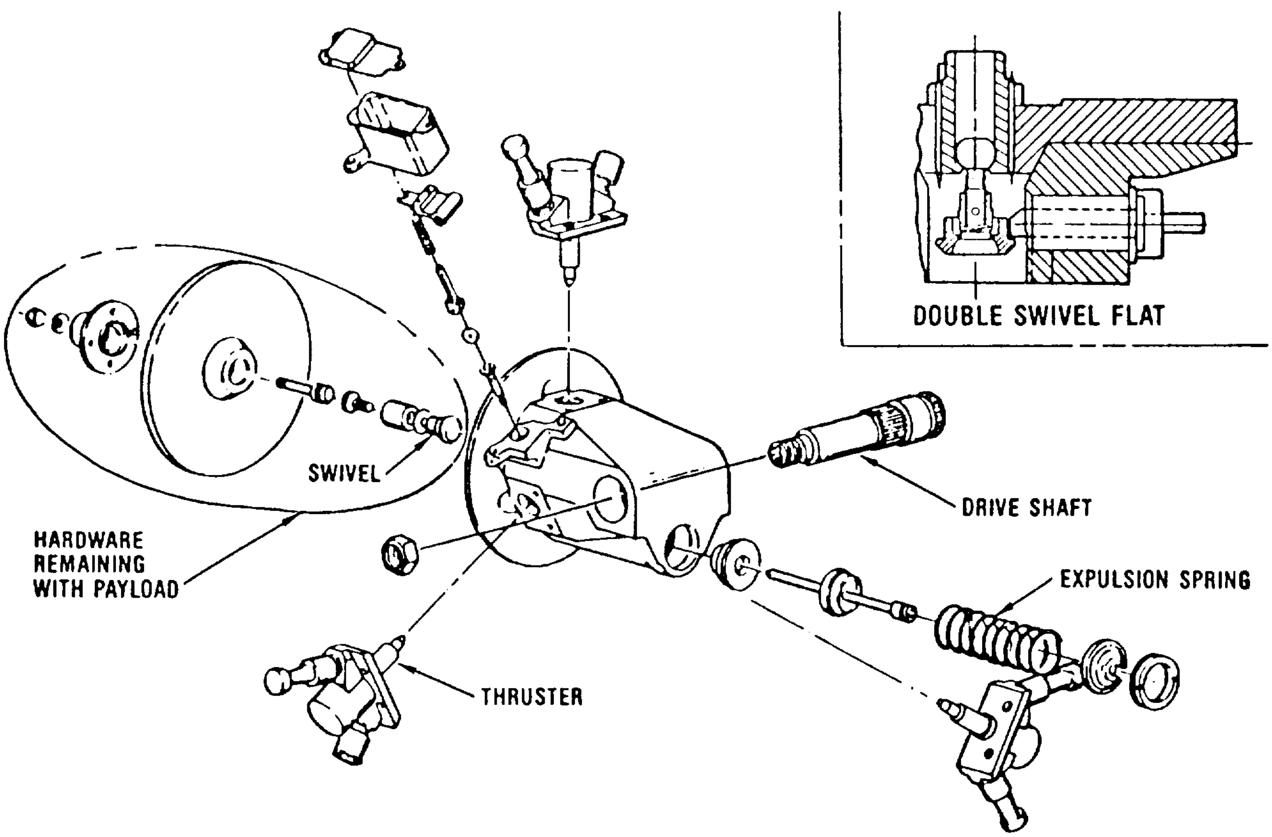


Figure 3. SPDS three-pin toggle payload release mechanism.

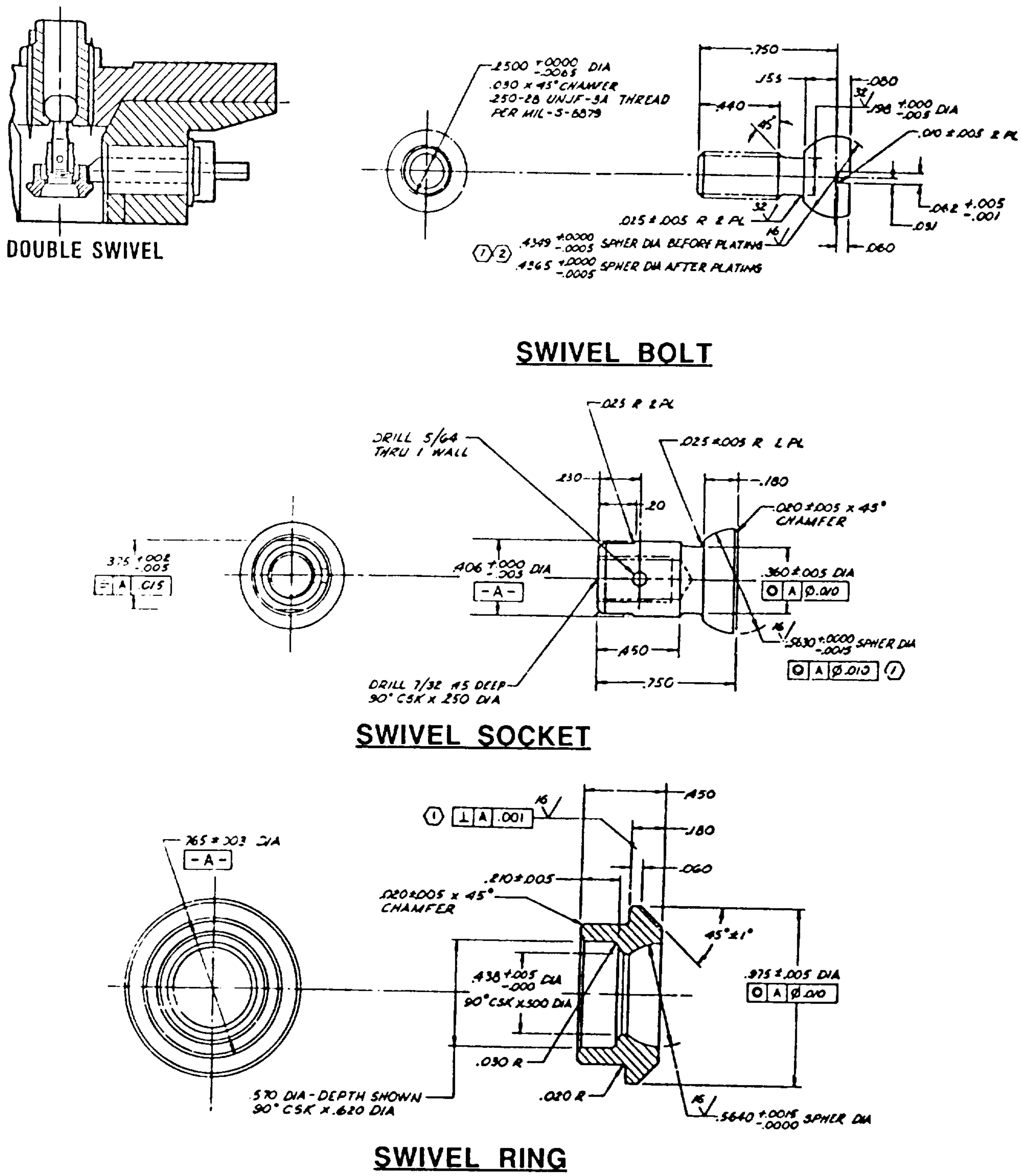
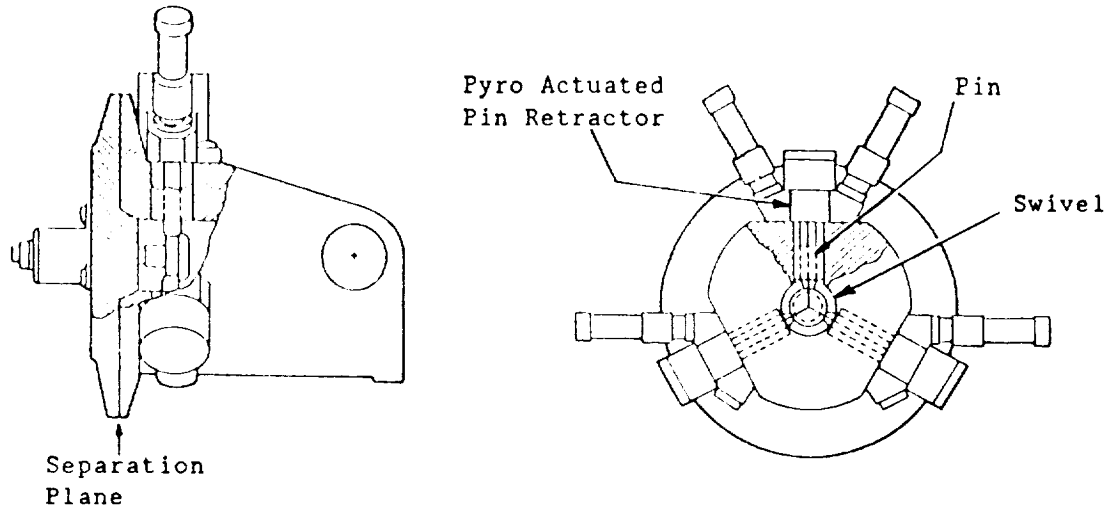
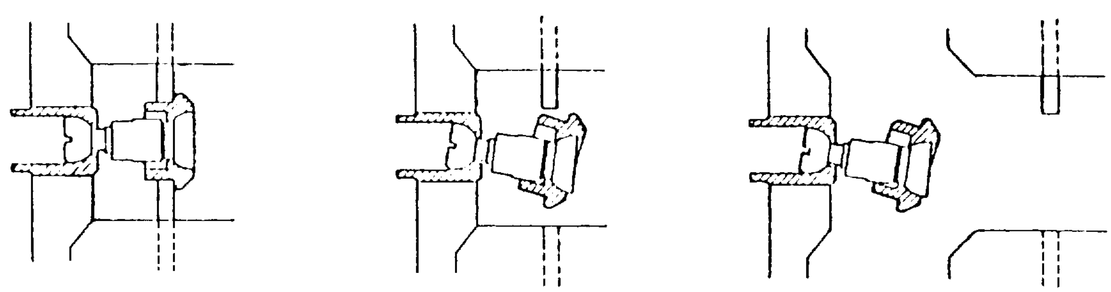


Figure 4. Double swivel toggle components.



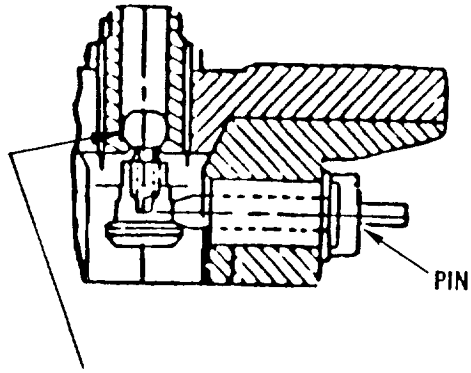
3-PIN TOGGLE RELEASE MECHANISM



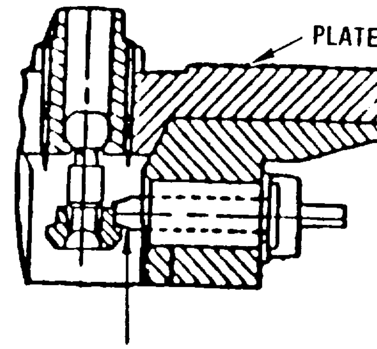
1. 3 pins lock swivel and payload in place
2. Removal of one or more pins causes the unstable swivel to move away from any remaining pins
3. Swivel and payload separation

RELEASE ANOLOGY

Figure 5. Two fault tolerant three-pin toggle release sequence.

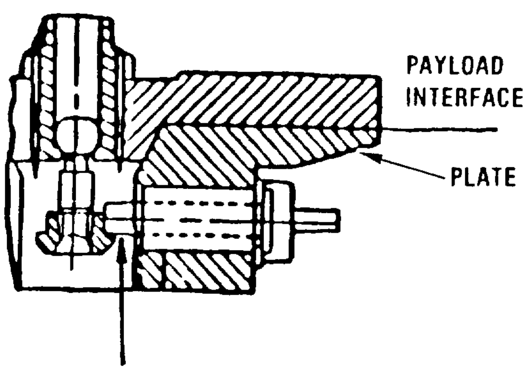


SINGLE SWIVEL



(SLOPED PIN/RING SURFACE CONTACT)

DOUBLE SWIVEL



(FLAT PIN/RING SURFACE CONTACT)

DOUBLE SWIVE BASELINE CONFIGURATION

Figure 6. Three-pin toggle release concept development.

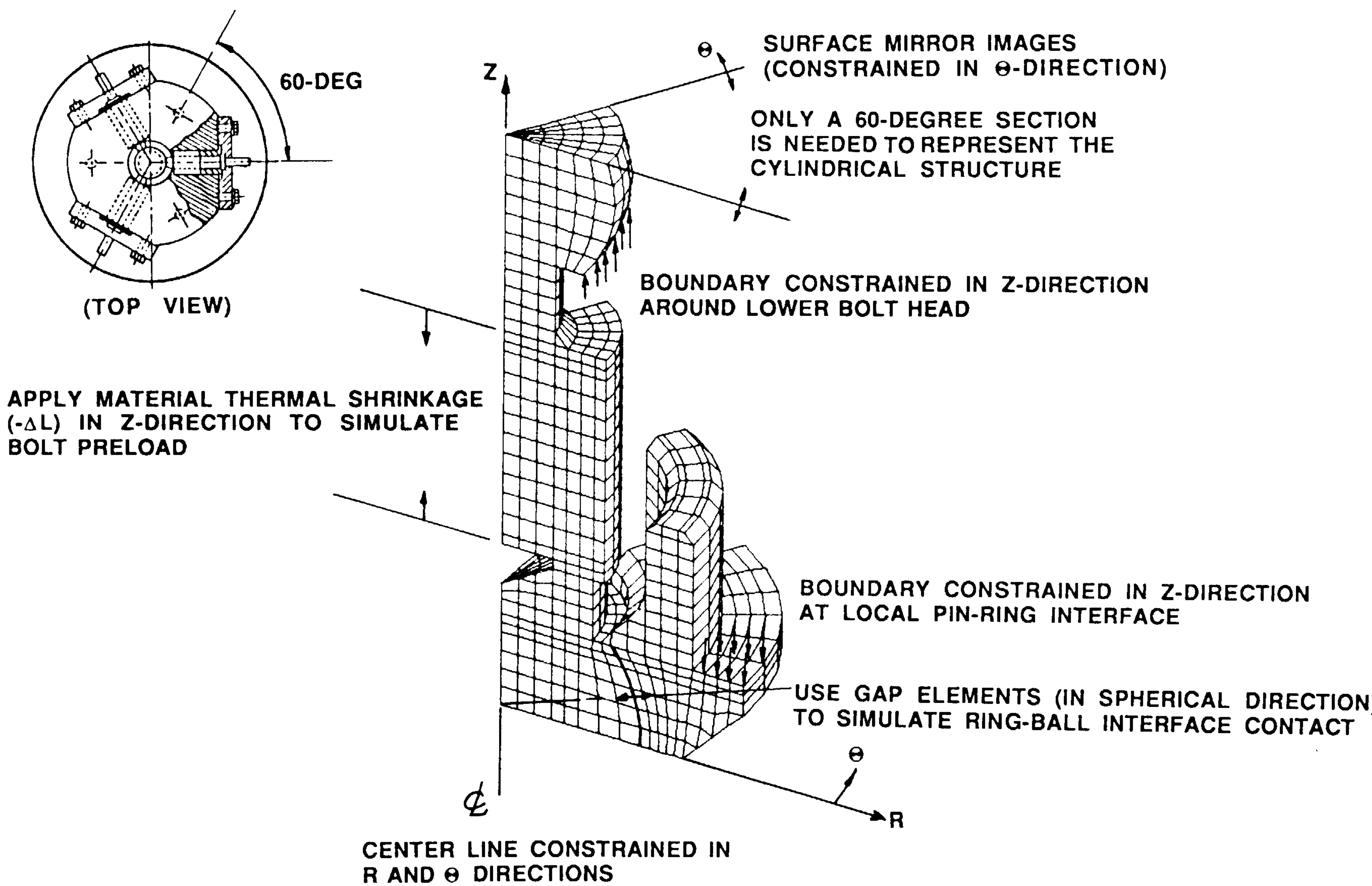


Figure 7. Double swivel toggle NASTRAN finite element idealization.

TOTAL PRE-LOAD = 1785 LB

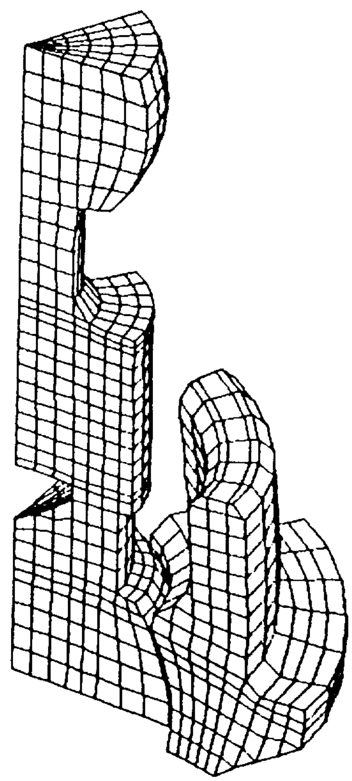


Figure 8. Exaggerated finite element model deformed shape.

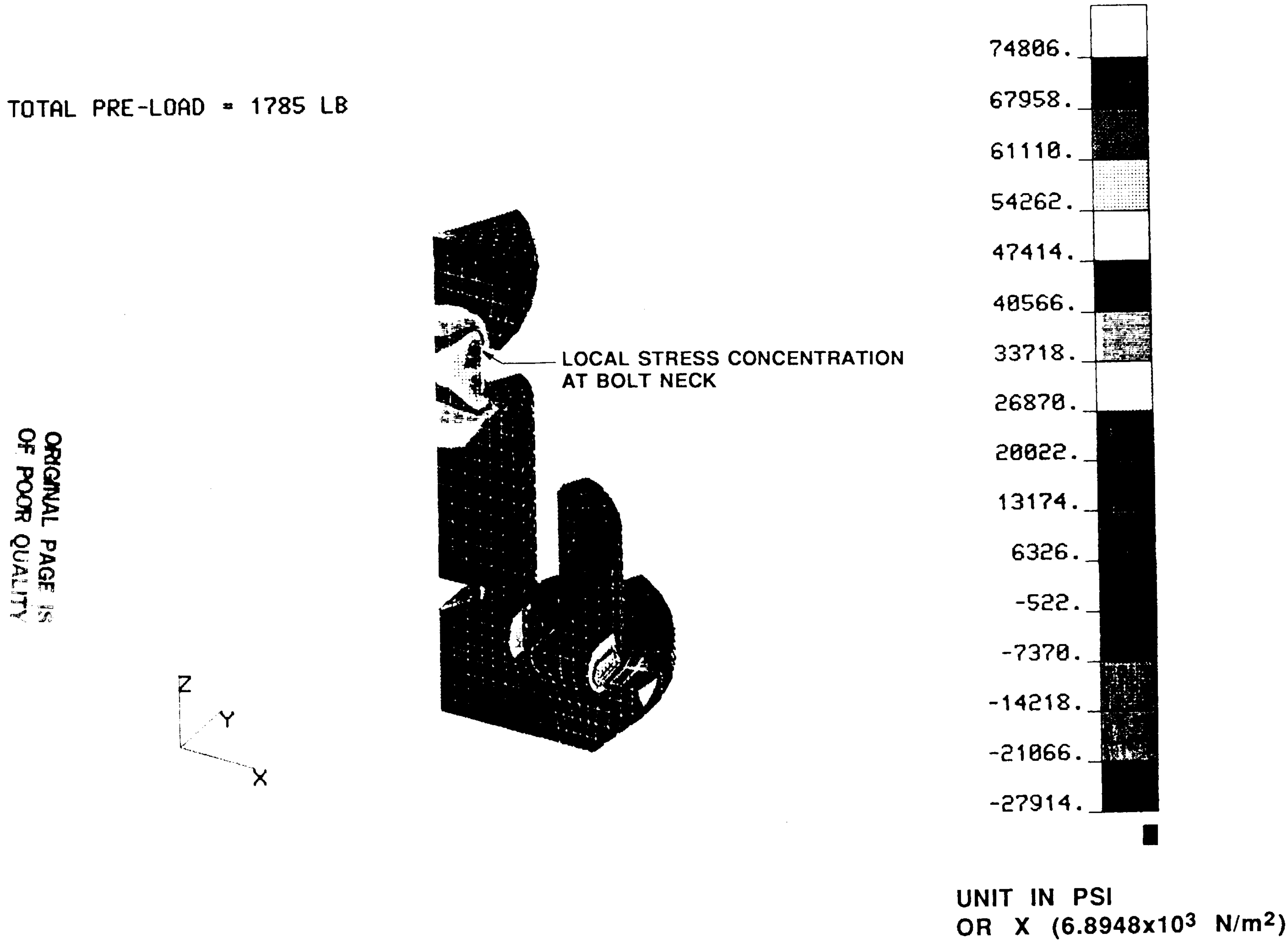
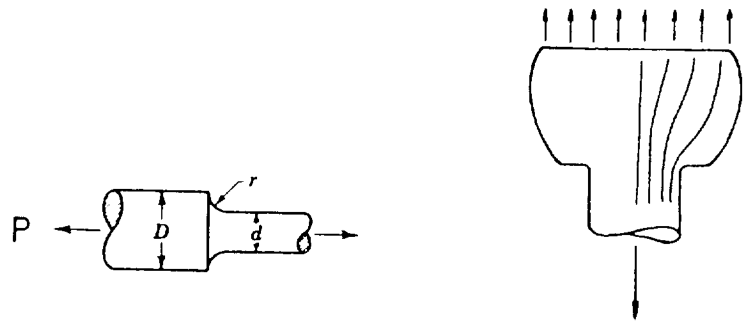
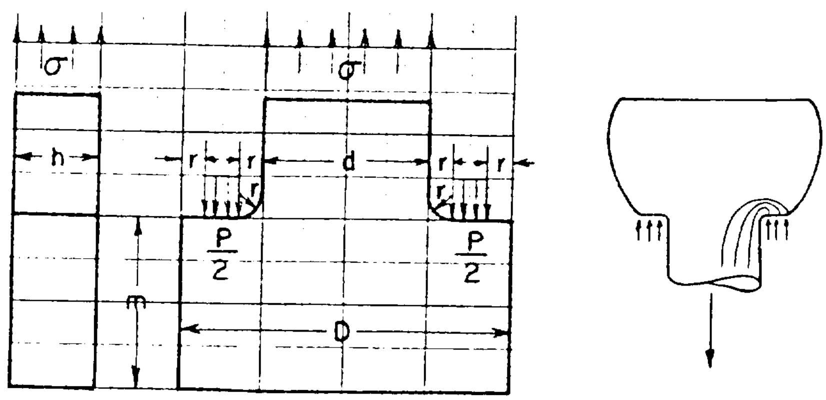


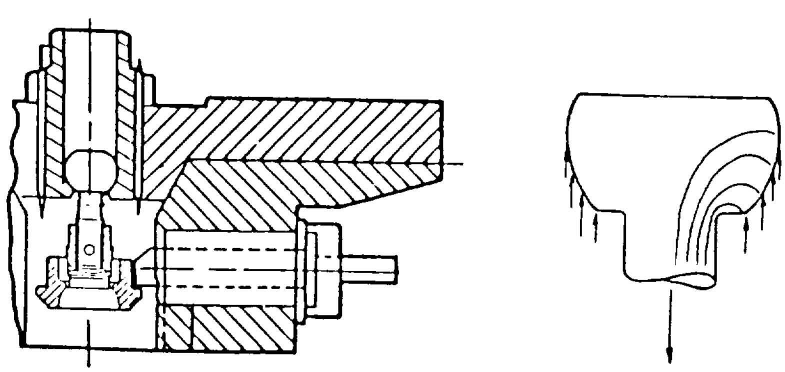
Figure 9. First principal stresses, pre-loaded three-pin toggle release swivel.



ROUND SHAFT WITH SHOULDER FILLET IN TENSION



T-HEAD



SWIVEL BOLT HEAD

Figure 10. Configurations of stress concentration effect.

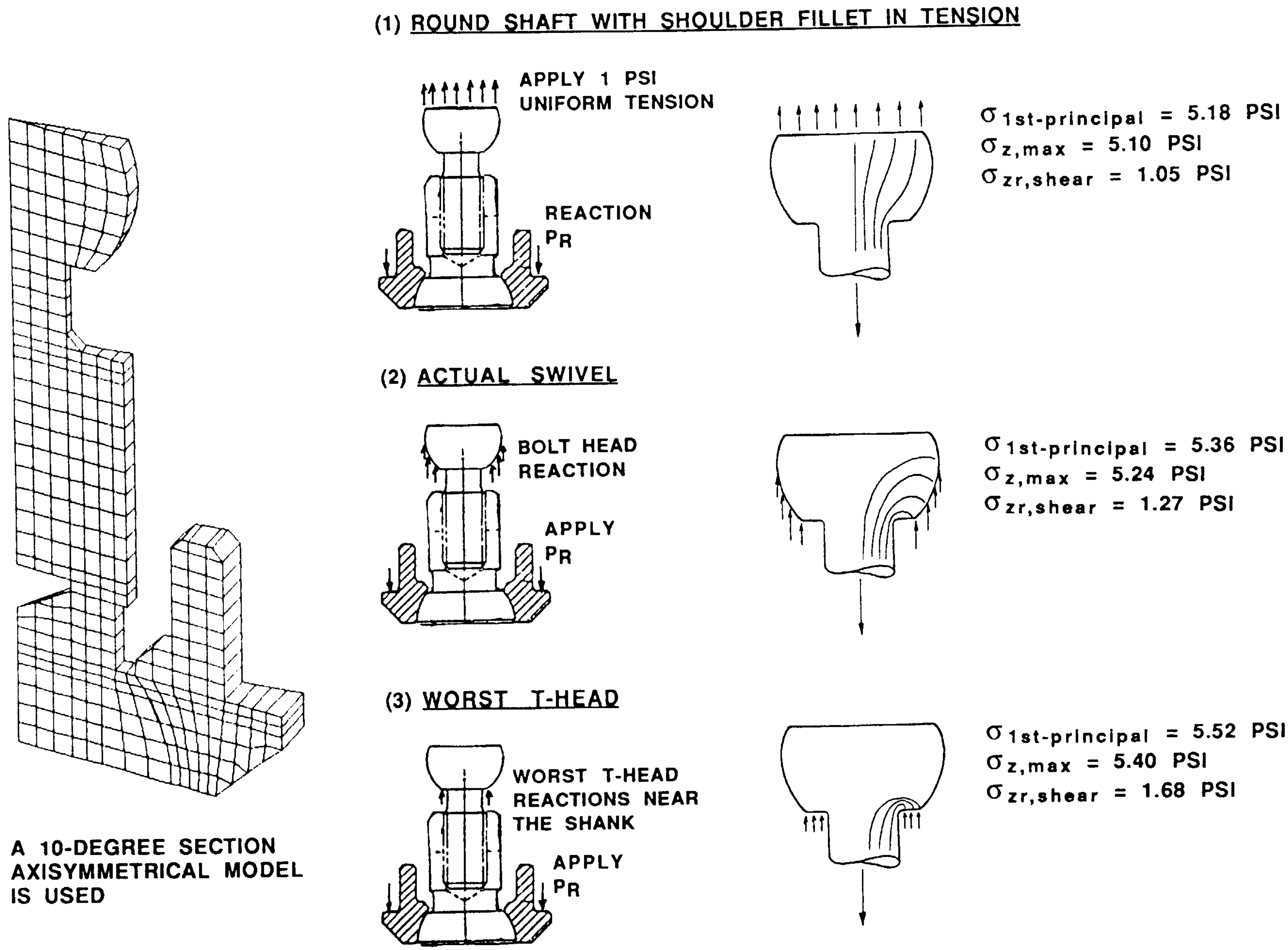
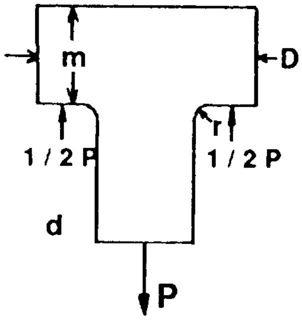


Figure 11. Comparison of tension-loaded, preloaded, and T-head swivel using NASTRAN.

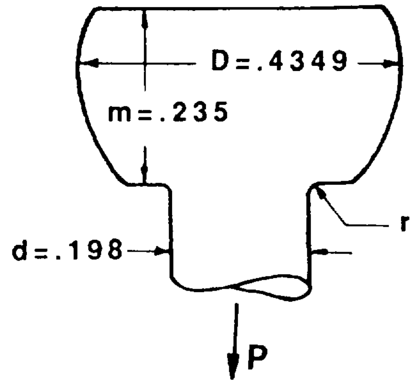
REF.R.E.PETERSON, "STRESS CONCENTRATION DESIGN FACTORS", JOHN WILEY & SONS, INC.



m/d=1.2 & D/d=2.2
for the swivel bolt

K_t=5.6 (for r/d=.05)
K_t=4.45 (for r/d=.075)
K_t=3.8 (for r/d=.10)
K_t=2.9 (for r/d=.20)

SWIVEL BOLT GEOMETRICAL CONFIGURATION



m/d=(.235)/(.198)=1.2
D/d=(.4349)/(.198)=2.2

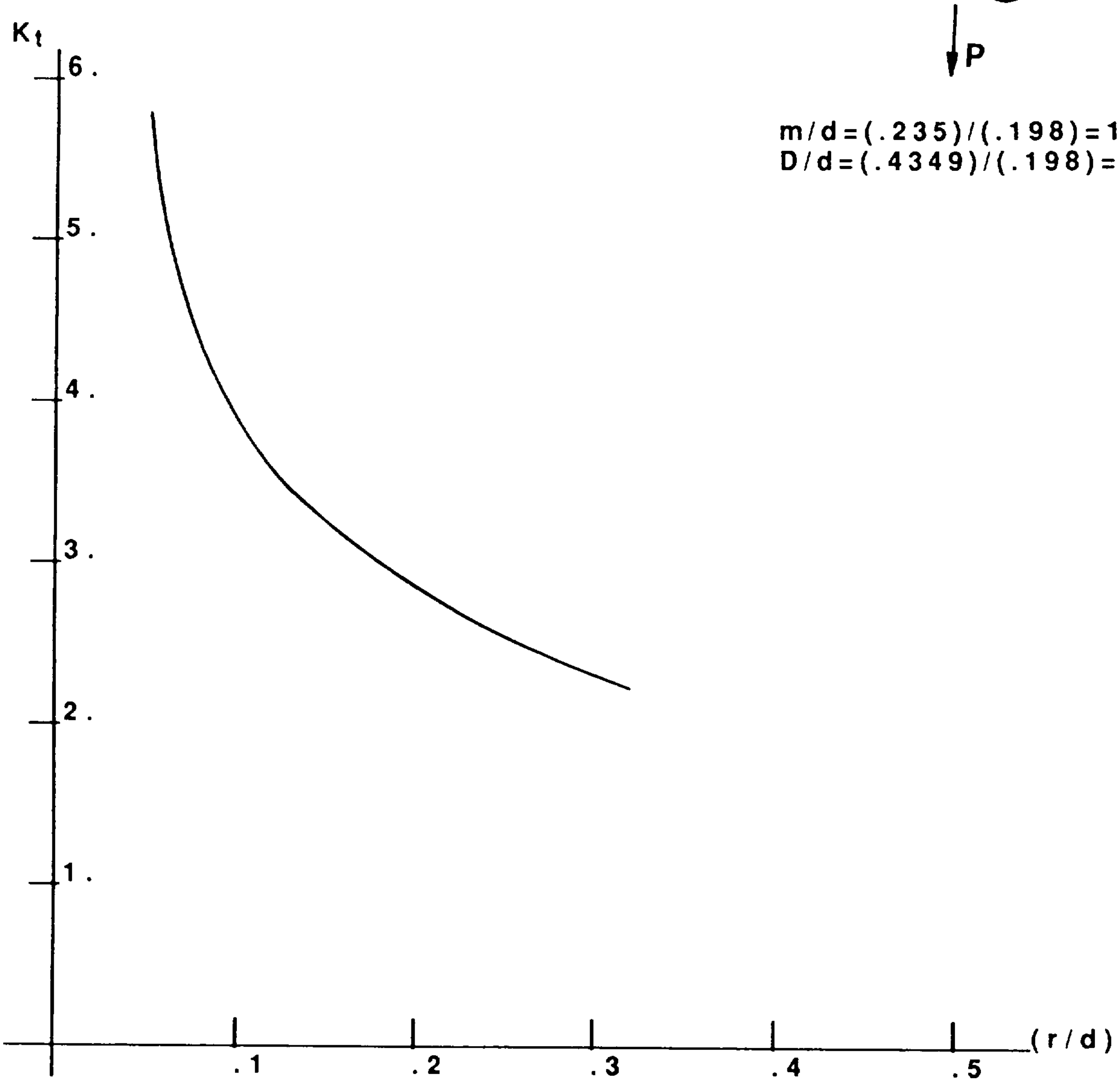


Figure 12. Stress concentration factors (K_t) for a T-head with variable fillet radius (r).