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BOLT PRELOAD SELECTION FOR PULSE-LOADED VESSEL CLOSURES

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ABSTRACT

Bounding, closed-form solutions are developed for selecting the bolt preload for a *square*, flat plate closure subjected to a pressure pulse load. The solutions consider the limiting case in which preload is primarily dependent on closure bending response as well as the limiting case in which preload depends on elastic bolt response. The selection of bolt preload is illustrated. Also presented in the paper is a detailed finite element analysis of dynamically loaded, bolted *circular* closure. The responses of the structure, closure, and bolts are included, and results are obtained for various preloads. The analysis illustrates a method of bolt preload modeling for use in general finite element computer programs.

DISCLAIMER

1. INTRODUCTION

Methods for determination of bolt preload or pretorque are typically based on fatigue loading considerations as well as the prevention of bolt loosening [1,2]. While recommendations in the recent literature differ somewhat [3,4], classical design practice is based on a bolt preload in the range of one-half to three-quarters of the yield stress [1,2].

Little guidance is available on preload values for bolted joints subjected to single-pulse dynamic loading, such as found in explosion containment vessels and shipping containers undergoing impact. A design methodology is presented in [5] for dynamic loading of closure bolts, but no information is provided on the role of bolt preload on bolt or closure response. Finally, a procedure for determining the optimal bolt preload for a dynamically loaded circular closure is presented in [6], but the solution ignores the effect of closure bending response.

In this paper, bounding closed-form solutions are developed for selecting the bolt preload for a square, flat plate closure subjected to a pressure pulse load. The solutions consider the limiting case in which preload is primarily dependent on closure bending response as well as the limiting case in which preload depends on elastic bolt response. The selection of bolt preload is illustrated.

Also presented in the paper is a detailed finite element analysis of a dynamically loaded, bolted circular closure. The responses of the structure, closure, and bolts are included, and results are obtained for various preloads. Details of modeling and a description of the results are presented in this paper.

2. ANALYSIS OF SQUARE PLATE CLOSURE

Consider a square plate bolted closure, as shown in Fig. 1. The plate is fastened to a pressure vessel by n bolts or capscrews and is subjected to a single pressure pulse loading, as shown in the figure. The loading is assumed spatially uniform over the inner surface of the plate and is taken as initially peaked and exponentially decaying.

It is possible to isolate the closure response into three cases. Denoting ω_p as the fundamental plate bending natural frequency and ω_b as the bolting system axial natural frequency, then the three cases are:

Case I: $\omega_p/\omega_b << 1$. This is the case in which the bolting system appears effectively rigid in comparison to plate bending.

Case II: $\omega_p/\omega_b >> 1$. In this case, the plate appears effectively rigid in comparison to the bolting system, i.e., no plate bending occurs: The bolts undergo stretching and the plate is modeled as a rigid body.

Case III: $\omega_p/\omega_b \sim 1$. In this case, plate bending frequency and bolt axial frequency are of the same order. Bolt stretching and plate bending simultaneously occur, i.e., dynamic response is coupled.

Cases I and II are analyzed in the next two sections to determine bolt loading and the influence of bolt preload on maximum dynamic bolt stress and displacement. Case III is not considered further.

3. CASE I: PLATE BENDING ONLY

Simplified methods of modeling structural elements as equivalent single-degree-of-freedom spring-mass systems are presented in Reference 7. The procedure consists of developing an equivalent one-degree-of-freedom representation of the structure, as illustrated in Figure 2, where the square plate of Figure 1 loaded by a uniformly-distributed pressure pulse is used for illustrative purposes. The actual elastic plate is loaded by pressure pulse p(t), has total mass, M, and central displacement or response, y(t). The equivalent spring-mass system, also shown in Figure 2, has some effective mass, M_e, and effective stiffness, K_e. The equivalent system is subjected to some equivalent loading, $F_e(t)$. The procedure is particularly suited to simple structural elements, such as the square plate closure of interest in this paper.

Representation of the structure by an equivalent one-degree-of-freedom system is based upon the principle of dynamic similarity: The requirement that work done by external forces, strain energy stored, and kinetic energy of the equivalent system be identical to those respective quantities in the actual structure.

As discussed in Reference 7, the equivalent system is selected so that the deflection of the concentrated mass is the same as that for some significant point on the actual structure, i.e., the midspan of the plate shown in Figure 2. The displacement-time history of the equivalent system directly corresponds to that of the selected point of interest on the actual structure, although stresses and forces in the idealized system are not directly equivalent to the same quantities in the structure. Stress and force quantities can, however, be determined (time scales of loading and response between actual and idealized structure remain the same).

Mass, stiffness, and loading constants for the equivalent system are evaluated on the basis of an assumed shape of the actual deforming structure. The shape in the elastic range, $\phi(x,y)$, taken is that resulting from the static application of the dynamic loading. This in general differs from the first-mode shape.

The method permits determination of the deflection-time history of the square plate closure as well as the plate boundary reactions. For this case, the boundary reactions are used to determine the (uncoupled) bolt peak response and the influence of the bolt preload on bolt response.

Let y(t) denote the central displacement of the square plate, and M_t the actual mass of the plate. The effective mass of the plate is given by

$$M_e = \int_A m\phi^2(x,y)dxdy \tag{1}$$

where $\phi(x,y)$ is the assumed shape function corresponding to the static deflected shape due to a uniform applied pressure and m is mass per unit area of the plate. The integral is taken over the plate area, A. Similarly, F_t is the actual peak force acting on the plate and the equivalent loading is

$$F_e = \int_A p(x,y)\phi(x,y)dxdy \tag{2}$$

Then, based on Reference 7, the equation of motion of the single degree of freedom system is given by

$$K_{LM}M_t\frac{d^2y}{dt^2} + ky = F(t) \tag{3}$$

where $K_{LM} = K_M/K_L$

and $K_M = M_e/M_t$

$K_L = F_e/F_t$

Values of K_{LM} needed in Equation (3) for a square plate are tabulated in Reference 7, so that evaluation of Equations (1) and (2) do not have to be performed in this case. For a simply supported square plate in the elastic range, the value is $K_{LM} = 0.67$, and the tabulated dynamic boundary reaction along each edge of the plate (determined from dynamic equilibrium) is

$$V(t) = 0.07F + 0.18R \tag{4}$$

where F = F(t) is the applied force due to the pressure pulse and R is the resistance of the structure. In the elastic range R is

$$R(t) = ky(t) \tag{5}$$

where k is (Reference 7)

$$k = 21E\frac{t^3}{a^2}$$

Here, E denotes the elastic modulus, t is plate thickness, and a is the length of a side of the plate.

For an initially peaked, exponentially decaying pressure pulse,

$$F = pA = F_0 e^{-\alpha t} \tag{6}$$

The dynamic boundary reaction, Equation (4), is found using Equation (5) to be

$$V(t) = 0.07F_0e^{-\alpha t} + \frac{3.78Et^3}{a^2}y(t) \tag{7}$$

It remains to solve for dynamic central plate deflection, y(t). The displacement solution of Equation (1) for the exponential pressure pulse in Equation (6) is

$$y(t) = \frac{F_0}{K_{LM}M_t}\left[\frac{e^{-\alpha t}}{\alpha^2+\beta^2} + \frac{\sin(\beta - \tan^{-1}(\beta/\alpha))}{\beta\left(\alpha^2+\beta^2\right)^{1/2}}\right] \tag{8}$$

Now for $\omega_p << \omega_b$, bolt motion is small in comparison to plate deflections, i.e., plate boundary motion can be neglected. For a simply supported square plate, the average applied load per bolt is

$$P(t) = \frac{4V(t)}{N} \tag{9}$$

where N is the total number of bolts. This force is effectively applied statically relative to the bolting system. Tensile stress in each bolt is then (Reference 1)

$$\sigma_b = \left(\frac{K_b}{K_b+K_m}\right)\frac{P(t)}{A_t} + \sigma_{pr} \tag{10}$$

up to the point of closure "lift-off". Here, σ_{pr} denotes the initial bolt prestress, A_t is the bolt tensile stress area, and K_b and K_m denote, respectively, the spring constant of the bolt and bolted material. These parameters are defined in Reference 1 and evaluated in Reference 6.

If the pressure-pulse loading is sufficient to drive the closure system temporarily beyond the point of lift-off, then bolt stress becomes simply

$$\sigma_b = \frac{P(t)}{A_t} \tag{11}$$

Peak dynamic bolt stress (and displacement) can then be readily determined by evaluating Equations (7)-(10) (and (11) if appropriate) up to the time of maximum bolt stress (or displacement). Results for one example closure are shown in Figure 3 ("Rigid Boundary") for the parameters indicated on the figure.

4. CASE 2: RIGID PLATE CLOSURE

In this case, the bending response of the plate closure occurs at sufficiently greater frequency than the bolting system ($\omega_p >> \omega_b$) that the plate bending response can be neglected. The bolt response can then be modeled as the single-degree-of-freedom system shown in Figure 4. The closure is assumed to respond as a rigid body of mass M, i.e., does not undergo bending. The static equilibrium position, x_e, of the bolt-closure interface is given by

$$x_e = \frac{K_m x_m + K_b x_b}{K_m + K_b} \tag{12}$$

where K_M is the stiffness of the bolted material and K_b is the stiffness of the closure bolts. x_m denotes the uncompressed position of the bolted material and x_b denotes the unstretched position of the closure bolts. Equation (12) is developed in Reference 6 based on well-known static bolting principles (Reference 1). It can be shown using Reference 6 that the equation of motion of the rigid mass subjected to an initially peaked, exponential pressure pulse acting on the inner surface of the closure is

$$M\frac{d^2x}{dt^2} + K_b(x - x_b) - K_m(x_m - x) = F_0 e^{-\alpha t} \tag{13}$$

With the initial conditions $x = x_e$ and $dx/dt = 0$ at $t = 0$. This equation is valid for $x \leq x_m$ (phase 1). For $x > x_m$ (phase 2), the closure plate lifts off and the equation of motion becomes

$$M\frac{d^2x}{dt^2} + K_b(x - x_b) = F_0 e^{-\alpha t} \qquad (14)$$

with appropriate "initial" conditions on displacement of $x = x_m$ and velocity equal to that at the end of phase 1. Solution to these equations are given in Reference 6 and are used here to determine maximum bolt displacement and stress. Resulting peak bolt stress and displacement are plotted in Figure 3 ("Rigid Plate") as a function of bolt prestress for the parameters indicated.

Comparisons of the "Rigid Plate" and "Rigid Boundary" solutions indicates similar qualitative behavior: Bolt prestress values that minimize peak dynamic bolt stress and displacement are seen in both cases to be substantially below classical recommended prestress values, at least for the numerical example shown in Figure 3.

5. FINITE ELEMENT ANALYSES OF CIRCULAR CLOSURES

The influence of bolt preload on the dynamic response of two circular closures was investigated using the finite element method. The analyses were performed on the spherical containment vessel shown in Figure 5. The vessel is 72 inches in diameter with nominally 2 inch thick walls of HSLA 100 steel. The top port is 22 inches in diameter and the four side ports are 16 inches in diameter. Each port consists of a thick nozzle welded to the vessel shell and a circular closure bolted to the nozzle using 64 high strength bolts in two bolt circles. The dynamic pressure loading on the inside surface of the vessel due to a 40 lb explosive charge is shown in Figure 6 (Reference 8). Because of the severity of the loading, the HSLA 100 steel nozzles and closures were modeled with elastic-plastic properties. The static yield for HSLA 100 was taken as 100,000 psi, but was increased to 124,000 psi in the dynamic analyses to account for strain rate effects. The bolts were given a dynamic yield of 170,000 psi. Figures 7 and 8 show three-dimensional analysis models of the 16 inch and 22 inch port closures, respectively. The 16 inch model consists of over 28,000 elements representing half of the closure, 32 bolts, and half of the nozzle. Symmetry boundary conditions simulate the other half of the port. The 22 inch port closure model has approximately 30,000 elements representing a quarter of the closure and nozzle, and 16 bolts. The X-Z and the Y-Z planes are planes of symmetry.

Bolt preload was included in the finite element analyses by utilizing an artificial temperature change to prestress the bolts. Each bolt was modeled explicitly, and a temperature dependent elastic-plastic material model was utilized for the bolt shanks. The initial preload was applied by decreasing the temperature of the bolt shanks while holding the temperature of the surrounding material constant. The bolts attempt to shorten due to cooling, causing compression of the closure between the bolt head and the nozzle. The single node at the center bottom of each bolt was constrained to have the same motion as the nozzle node at the same location. The remaining nodes representing the bottom of each bolt were allowed to move on the nozzle surface. This was accomplished using a sliding interface without gaps which constrains the nodes to stay on a surface. Thus, the bottoms of the bolts were constrained to move with the nozzle, but were able to radially contract, as they could realistically do. Sliding surfaces were defined between the nozzle and cover where they are in contact, and between the bolts and cover on the sides of the bolt holes and top of the cover where the bolt heads rest.

The bolt stresses are correctly calculated in this manner, but the strains are not. Figure 9 illustrates the differences between mechanically and thermally applied prestress. In this discussion, tension and extension are positive values for stress and strain. For the mechanically applied prestress, the bolt shank is stretched by extension of the bolt into the nozzle by tightening. First the assumption is made that the cover is rigid and does not compress as the bolt is tightened. This causes an axial bolt strain and a directly corresponding stress of

$$\varepsilon = \frac{x}{l} \tag{15}$$

and

$$\sigma = \frac{Ex}{l} \tag{16}$$

where ε and σ are strain and stress, x is the extension of the bolt, l is the original length of the bolt, and E is the elastic modulus of the material. The force in the bolt that must be resisted by the cover is

$$F = \sigma A = \frac{xEA}{l} \tag{17}$$

where A is the area of the bolt. If the closure is allowed to compress, the strain and stress become

$$\varepsilon = \frac{(x - c_1)}{l} \tag{18}$$

and

$$\sigma = \frac{E(x - c_1)}{l} \tag{19}$$

where c_1 is the cover compression.

In the case of the thermally applied preload, the bolt shank is cooled by some ΔT. Assuming that the bolt length is changed by a value -h due to the temperature change, the strain is

$$\varepsilon = \alpha \Delta T = \frac{-h}{l} \tag{20}$$

Now the bolt is pulled back to its original position and the rigid cover is inserted. Since the bolt is back to its original length, the total strain must equal zero and the stress is

$$\sigma = -E\alpha\Delta T = \frac{Eh}{l} \tag{21}$$

The stress in the bolt is positive (tensile). The force in the bolt is

$$F = \sigma A = \frac{hEA}{l} \tag{22}$$

The cover is allowed to compress by a value of c_2 and the bolt strain becomes

$$\varepsilon = \frac{-c_2}{l} \tag{23}$$

again a compressive strain, and stress is

$$\sigma = \frac{E(h - c_2)}{l} \tag{24}$$

If $\alpha\Delta T$ is chosen such that h = x, then the bolt force causing cover compression is equivalent giving $c_2 = c_1$. Then the stress in each case is the same but the strains are different by the amount of the thermal strain. That is,

$$\varepsilon_1 = \varepsilon_2 + \frac{h}{l} \tag{25}$$

where ε_1 is the "true" preload strain and ε_2 is the thermally applied preload strain. The analysis was run using DYNA3D [9] with temperatures input on the bolt shanks only. The temperature was ramped on before the pressure load was applied. Figure 10 shows stresses at the center of one of the bolt shanks as a function of time. The bolt was preloaded by uniformly decreasing the temperature of the bolt shank over the first 0.4 millisecond of the analysis. The temperature was then held constant for the remainder of the analysis to keep the preload on the bolt. The pressure load started at 0.5 millisecond. The temperature applied to the bolt shank in this example was -850 degrees. This temperature change would give a thermal stress of 165,000 psi if the bolt length were rigidly held, but it gives a prestress of approximately 60,000 psi due to closure compression. The strain due to a mechanically applied prestress of 60,000 psi would be approximately 2 millistrain (tension). However, the strain in the bolt is calculated to be about -3.5 millistrain (compression). The difference between these two numbers is the thermal strain which is 5.5 millistrain.

Although each model consisted of only the closure and nozzle, the response of the vessel was included using a substructuring technique in DYNA3D. The vessel response was included as displacement and velocity boundary conditions on the boundary of each port model. These conditions were saved from an earlier analysis of the confinement vessel [10, 11]. Since the pressures applied to the port models were delayed to get the preload on, the pressures in the overall model also had to be delayed in order that the pressure load applied to the port was synchronized with the displacements applied to represent the overall structure response. The pressure load was applied to all inside surfaces of the nozzle and closure for each port.

The preloads examined for the side port were zero, 40,000 and 60,000 psi. The principal response of the 16 in. closure was outward bulging. The bolts on the inner bolt circle sustained some yielding when they had no preload, a smaller amount of yielding at 40,000 psi preload, and no yielding with 60,000 psi preload. The outer bolts did not yield in any case. The maximum stresses in the closure were at the corner where the bolt flange connects to the thick part of the closure and yielding was predicted at this location for all cases, but the amount of yielding decreased with increasing preload.

The 22 in. port model was analyzed for 0, 60,000 and 100,000 psi preload. The corner of the closure where the bolt flange is attached showed no yielding for zero preload and a small amount of yielding for 60,000 and 100,000 psi preloads. All cases examined for the 22 in. port predicted yielding of the inner bolts. The 60,000 and 100,000 psi preload cases predicted yielding of the outer bolts as well. All bolts are predicted to have strains from 6 to 10 percent by 6.5 milliseconds. These strains are considerably larger than the initial preload strain, indicating not just loss of preload, but looseness of the cover. Due to the severity of the pressure pulse loading and resulting yielding, peak bolt stress is not used as a measure of the influence of bolt prestress. Rather, the influence of the bolt preload on closure response is isolated in terms of closure lift-off, as shown in

Figure 11. The response, in terms of minimizing lift-off and cover yielding, improved with increasing preload on the 16 in. port. But for the 22 in. port, little difference closure lift-off is predicted in the range between 0 and 60,000 psi, although results in Figure 11 suggest that there appears to be an optimal value of bolt preload well below classical preload values and on the order of that found in the previous section for the square plate closure. Beyond 60,000 psi preload, the cover lift-off increases as the bolts yield more.

One final analysis was done to show the effect of the vessel response. The analysis was of the larger port with 60,000 psi preload and with a fixed boundary in place of the substructure interface. That is, the response of the vessel was not included. All bolts again yielded, but lift-off was substantially reduced, as shown in Figure 11. It is clear that the influence of vessel dynamic response is very significant on the response of the closure and bolting system.

6. CONCLUSIONS

Two types of pulse-loaded vessel closures are examined to determine the influence of closure bolt preload on the peak response of the closure/bolting system. For the square-plate closure, the two bounding solutions were found to indicate similar qualitative behavior: Bolt prestress values minimizing peak bolt response are found to lie well below classical recommended prestress values, at least for the numerical example evaluated.

For the circular closure, a procedure for inclusion of bolt preload in finite element analysis is presented. Cover liftoff behavior was found to be sensitive to bolt preload as well. However, no general trend could be isolated for the two closures examined.

Confounding issues in this case are the presence of significant plasticity in some bolts for certain preload values and pronounced influence of pressure vessel vibratory response.

In both cases it is clear that bolt preload is a major design consideration and that classical bolt preload guidelines may be inappropriate for pulse-loaded structures.

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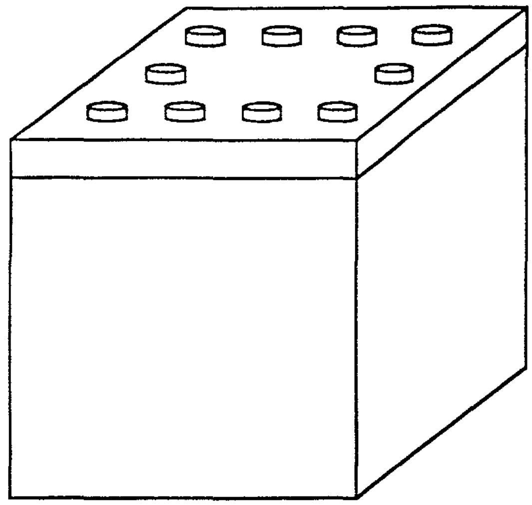
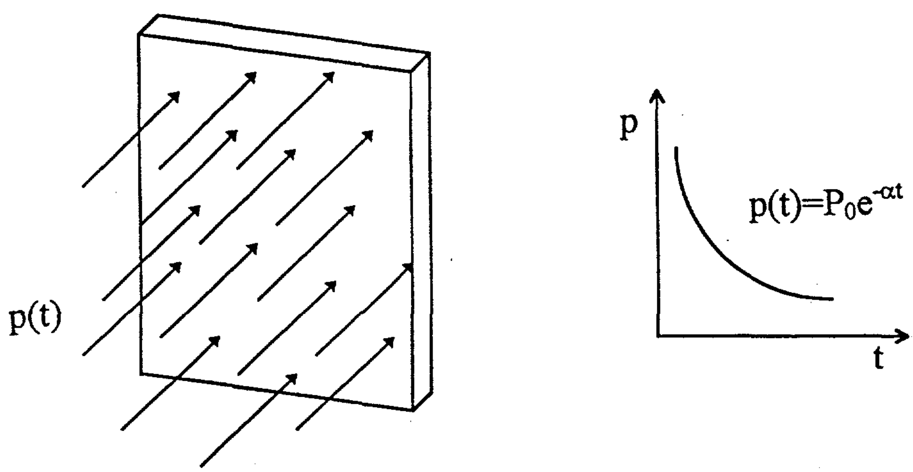


a. Bolted Closure for Pressure Vessel



b. Pressure-Pulse Loaded Plate Closure

FIGURE 1. SQUARE PLATE BOLTED CLOSURE SUBJECTED TO INTERNAL PRESSURE PULSE LOADING.

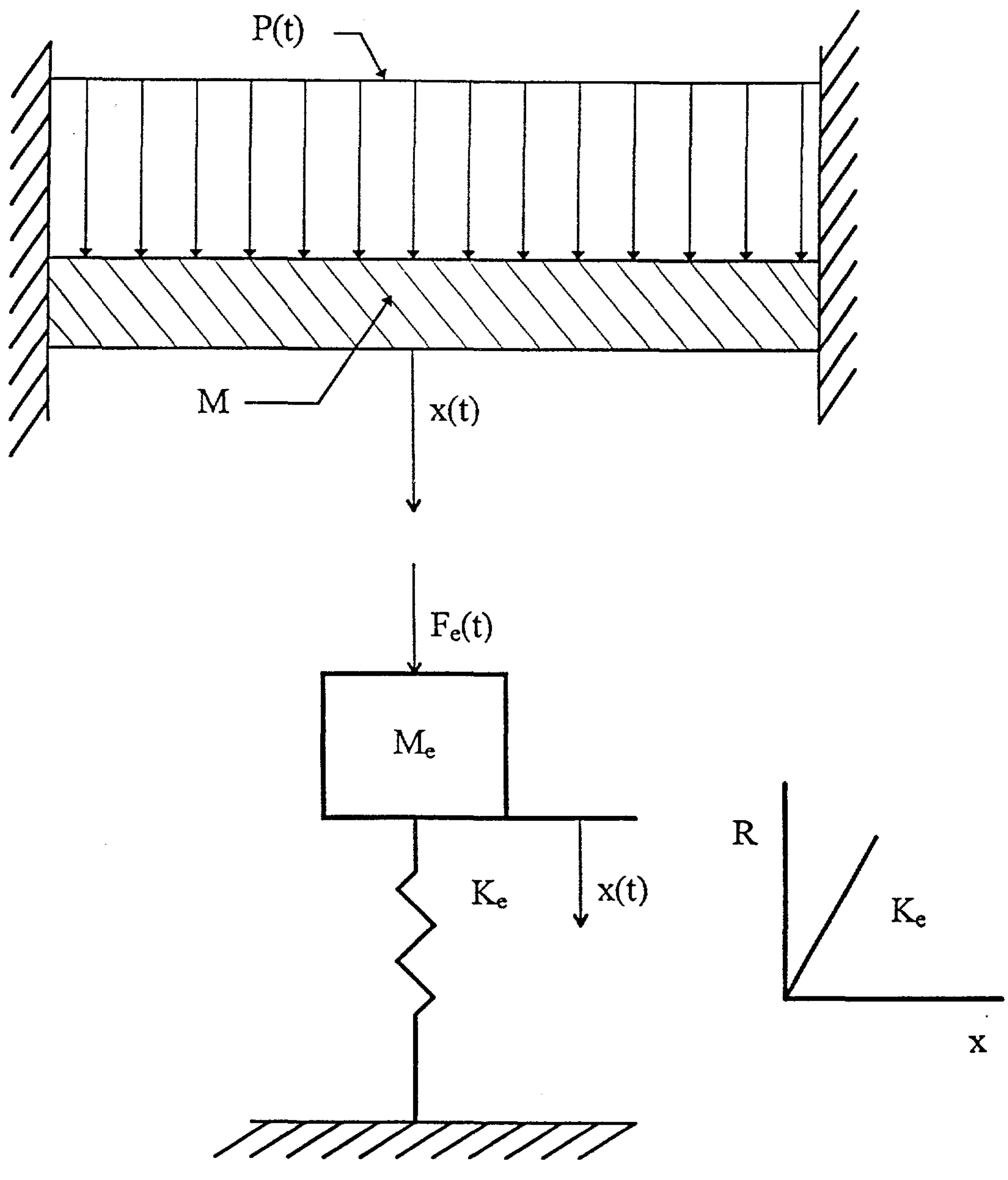


FIGURE 2. CLAMPED SQUARE ELASTIC PLATE CROSS SECTION WITH EQUIVALENT 1-DOF SPRING-MASS MODEL.

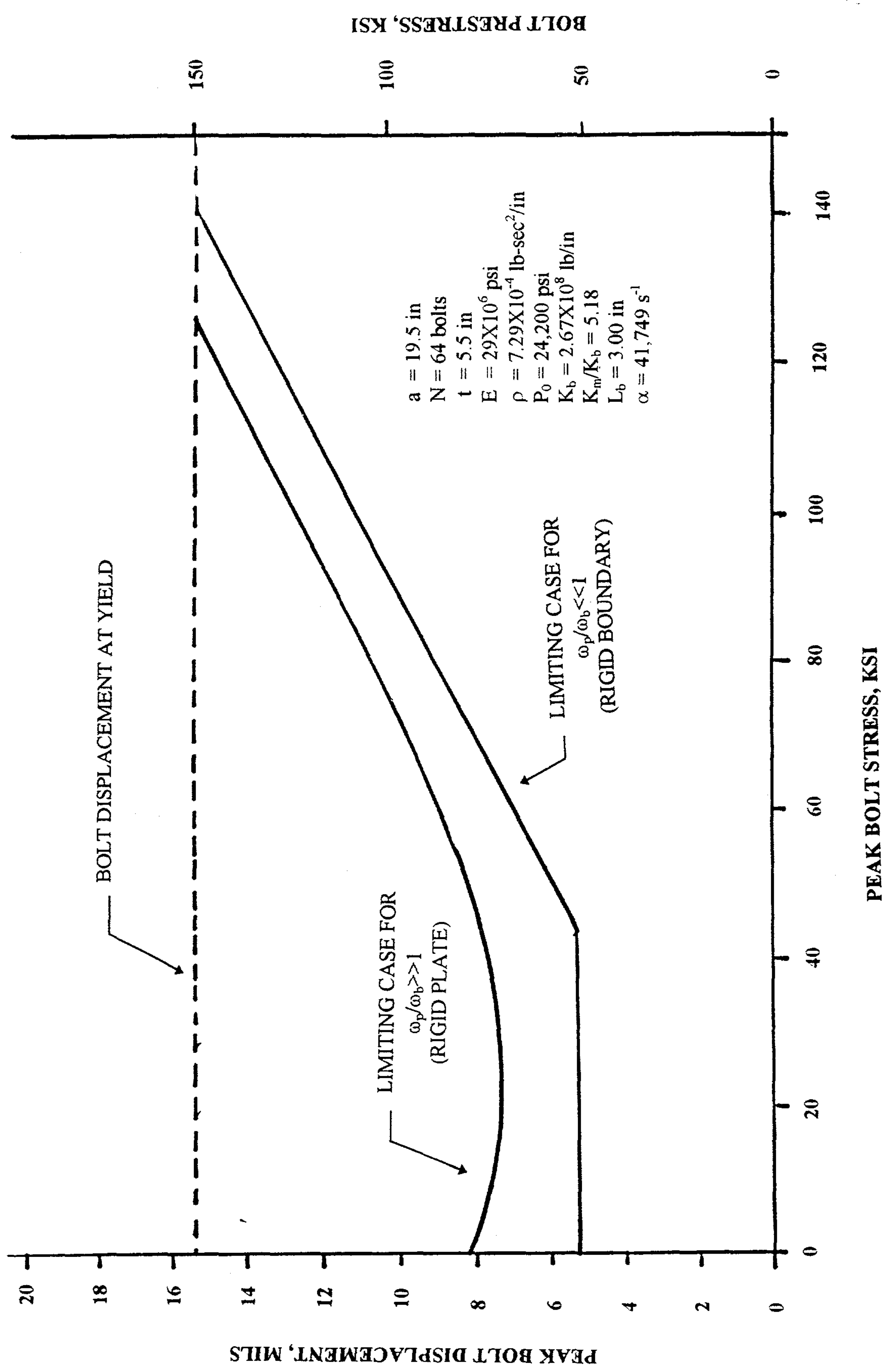
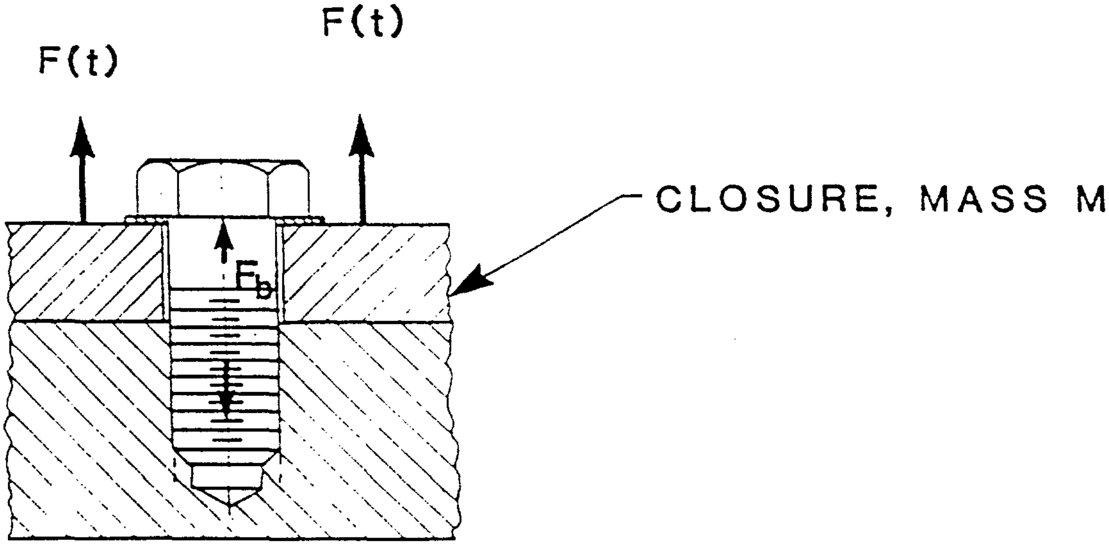
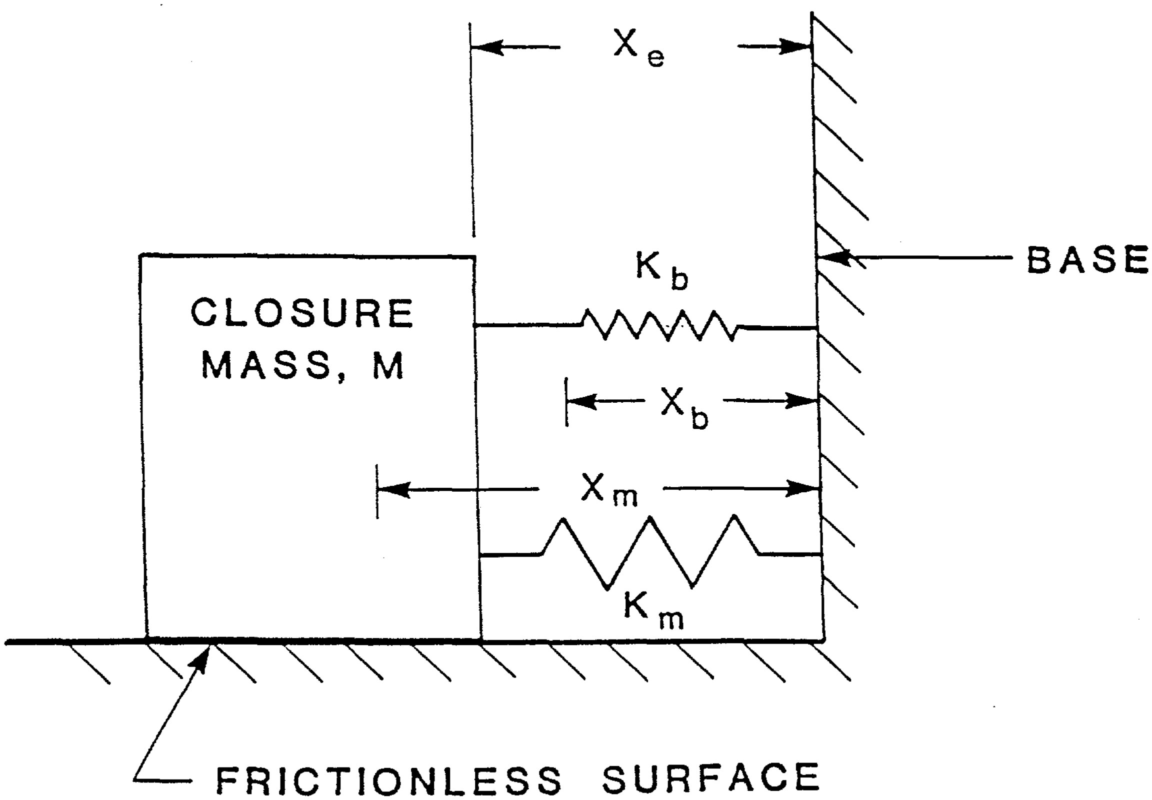


FIGURE 3. PEAK BOLT STRESS AND DISPLACEMENT AS A FUNCTION OF BOLT PRESTRESS.



a. BOLTED JOINT



b. STATIC EQUILIBRIUM POSITION OF MODEL

FIGURE 4. MODEL FOR ANALYSIS OF CLOSURE SYSTEM.

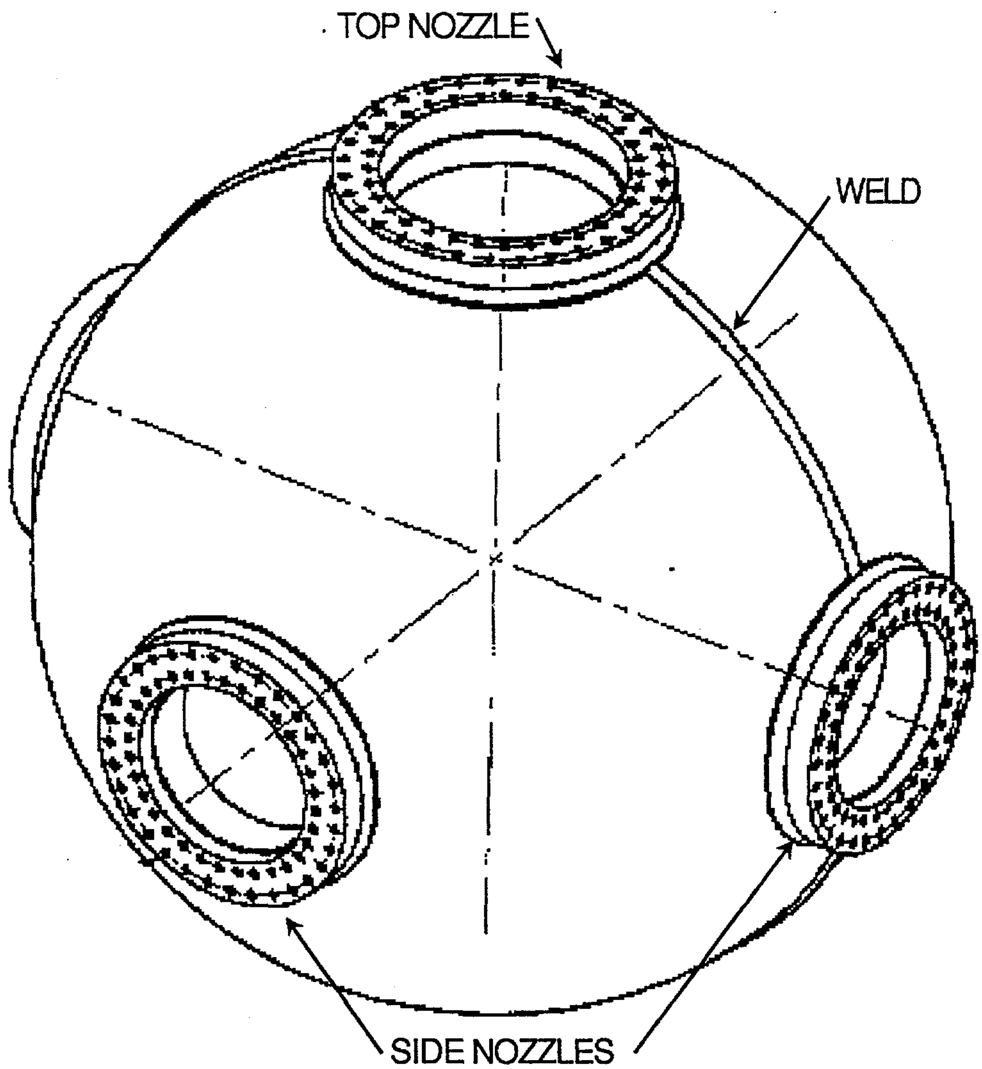


FIGURE 5. SPHERICAL CONTAINMENT VESSEL

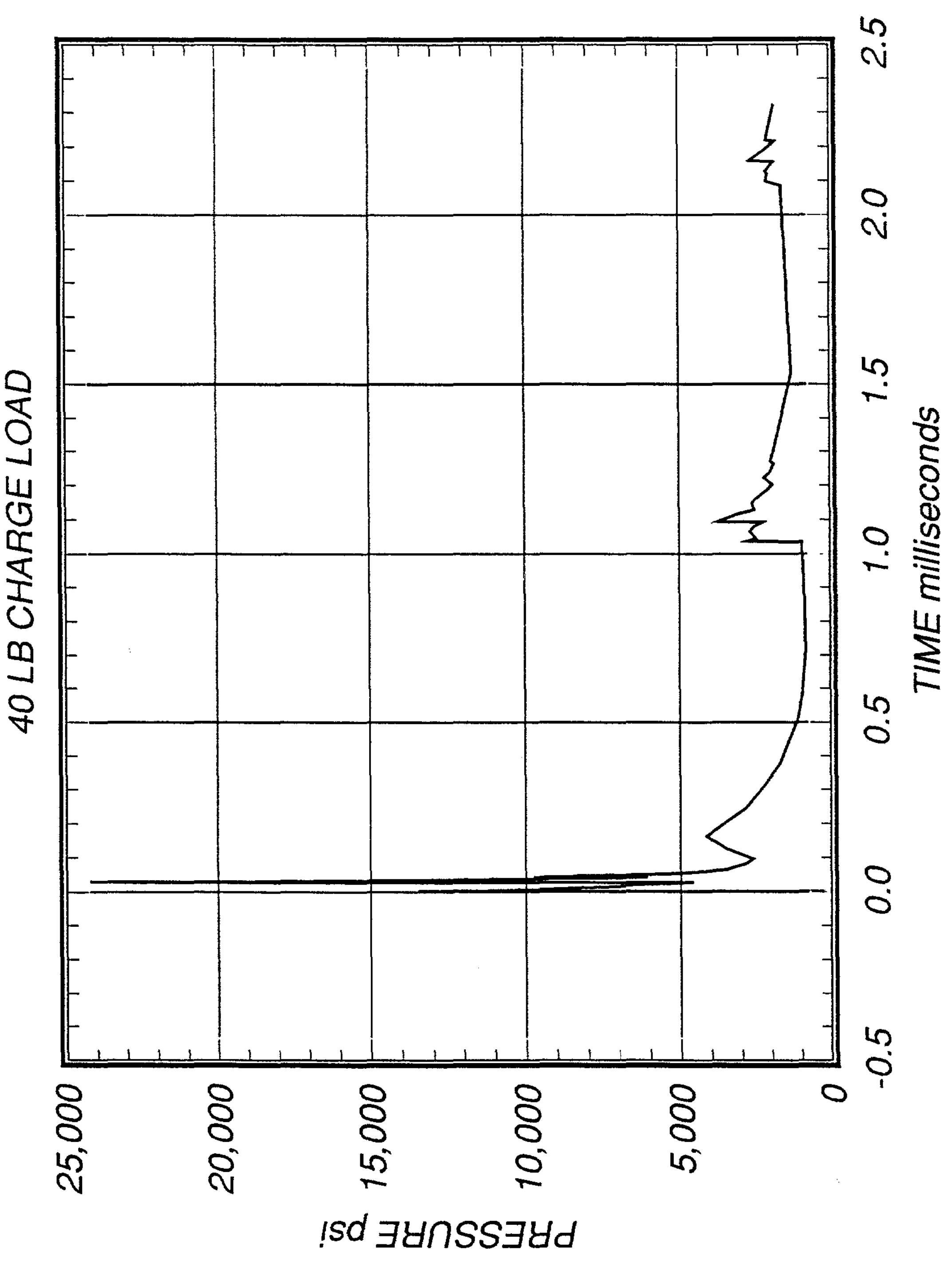


FIGURE 6. DYNAMIC PRESSURE LOADING ON INSIDE SURFACE OF VESSEL.

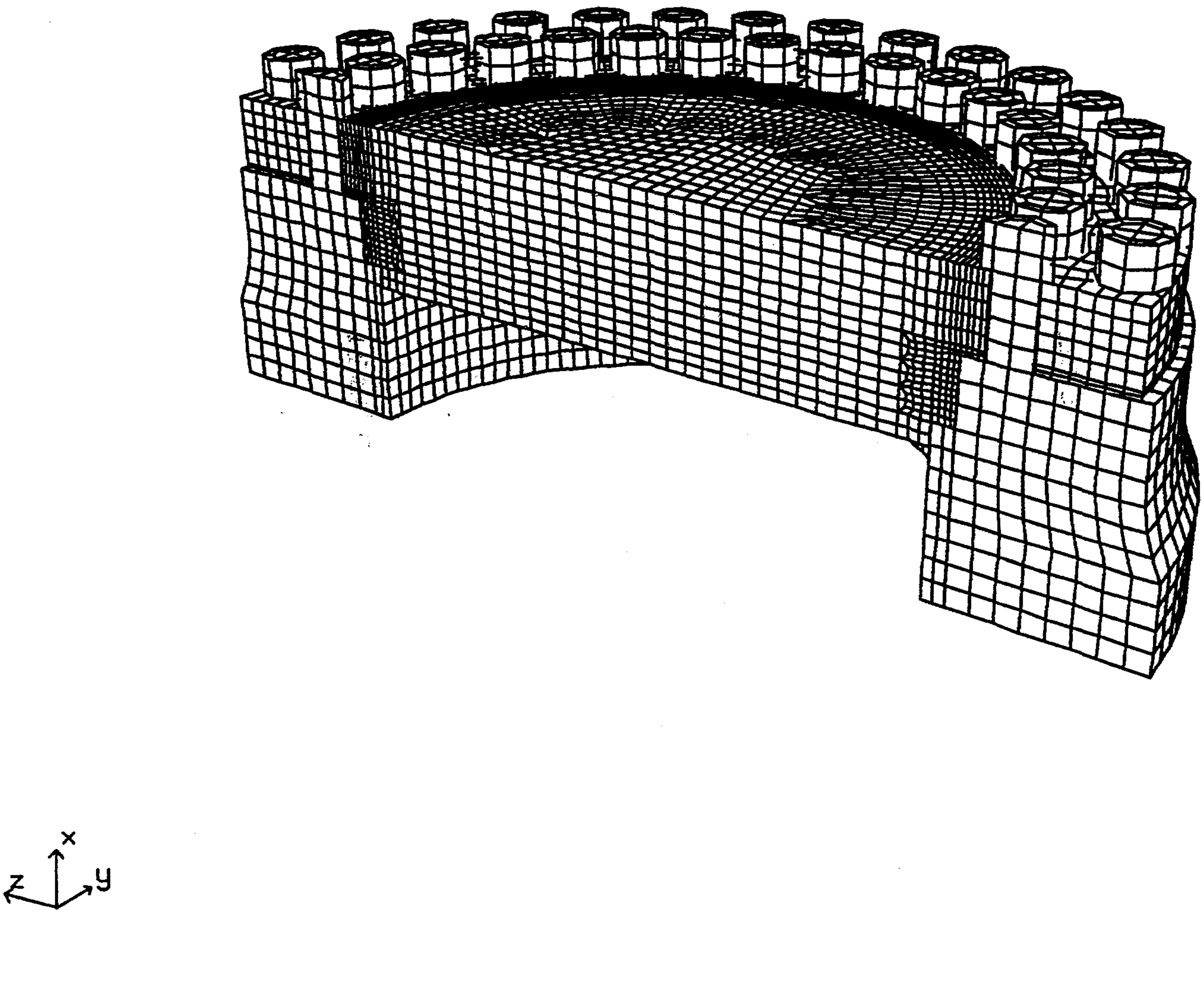


FIGURE 7. FINITE ELEMENT MODEL OF 16-INCH PORT CLOSURE.

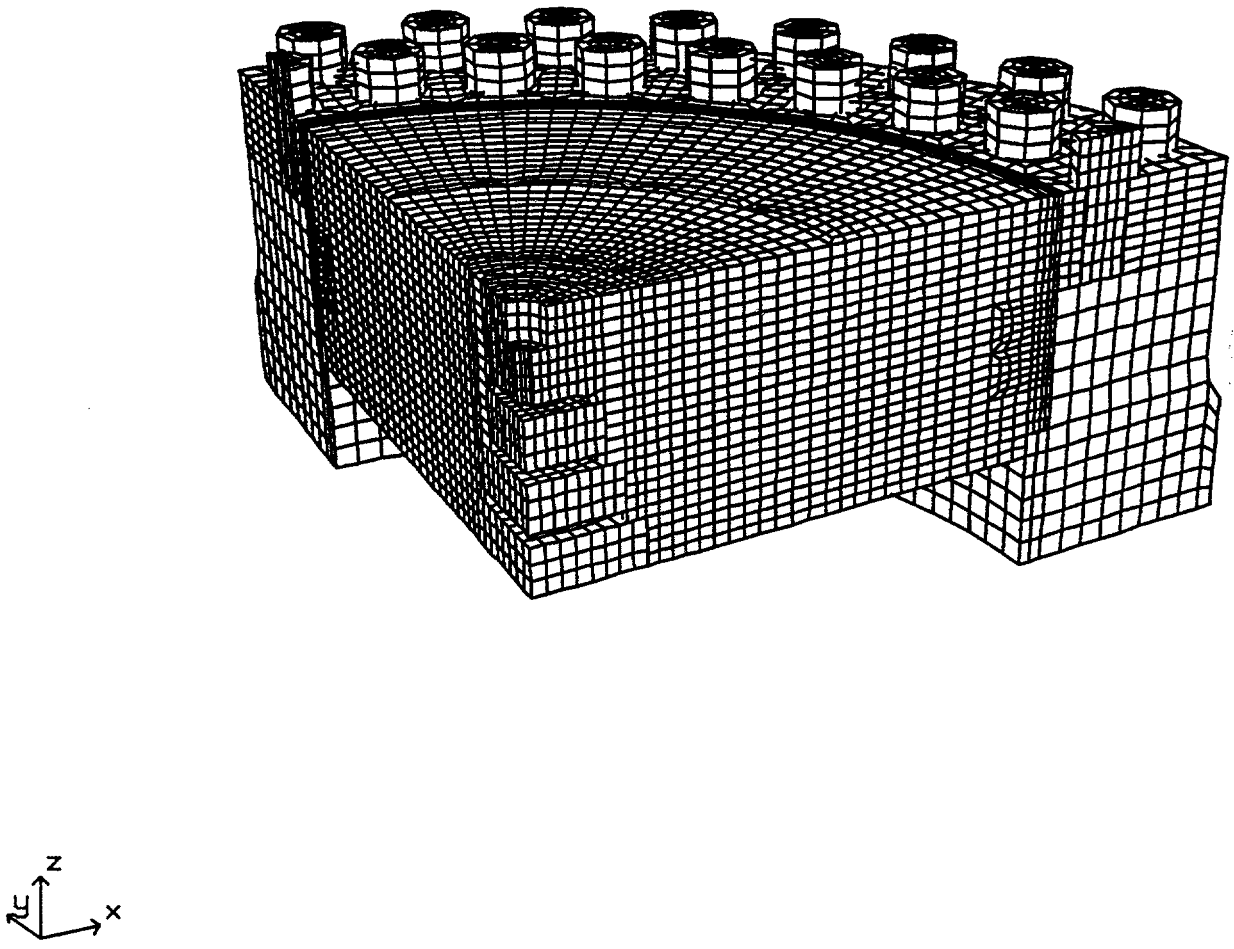


FIGURE 8. FINITE ELEMENT MODEL OF 22-INCH PORT CLOSURE.

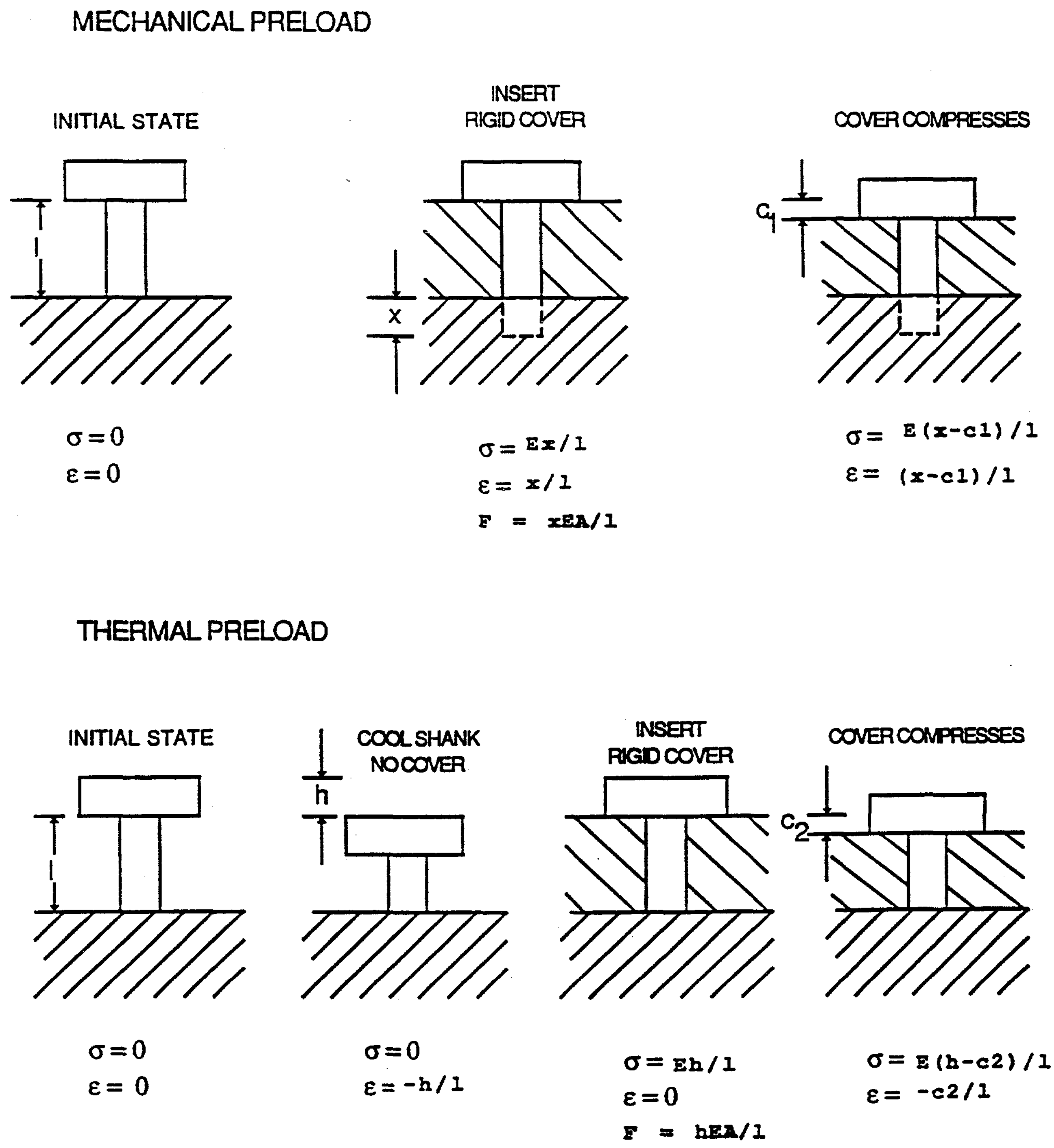


FIGURE 9. COMPARISON OF MECHANICALLY - AND THERMALLY - APPLIED STRAIN.

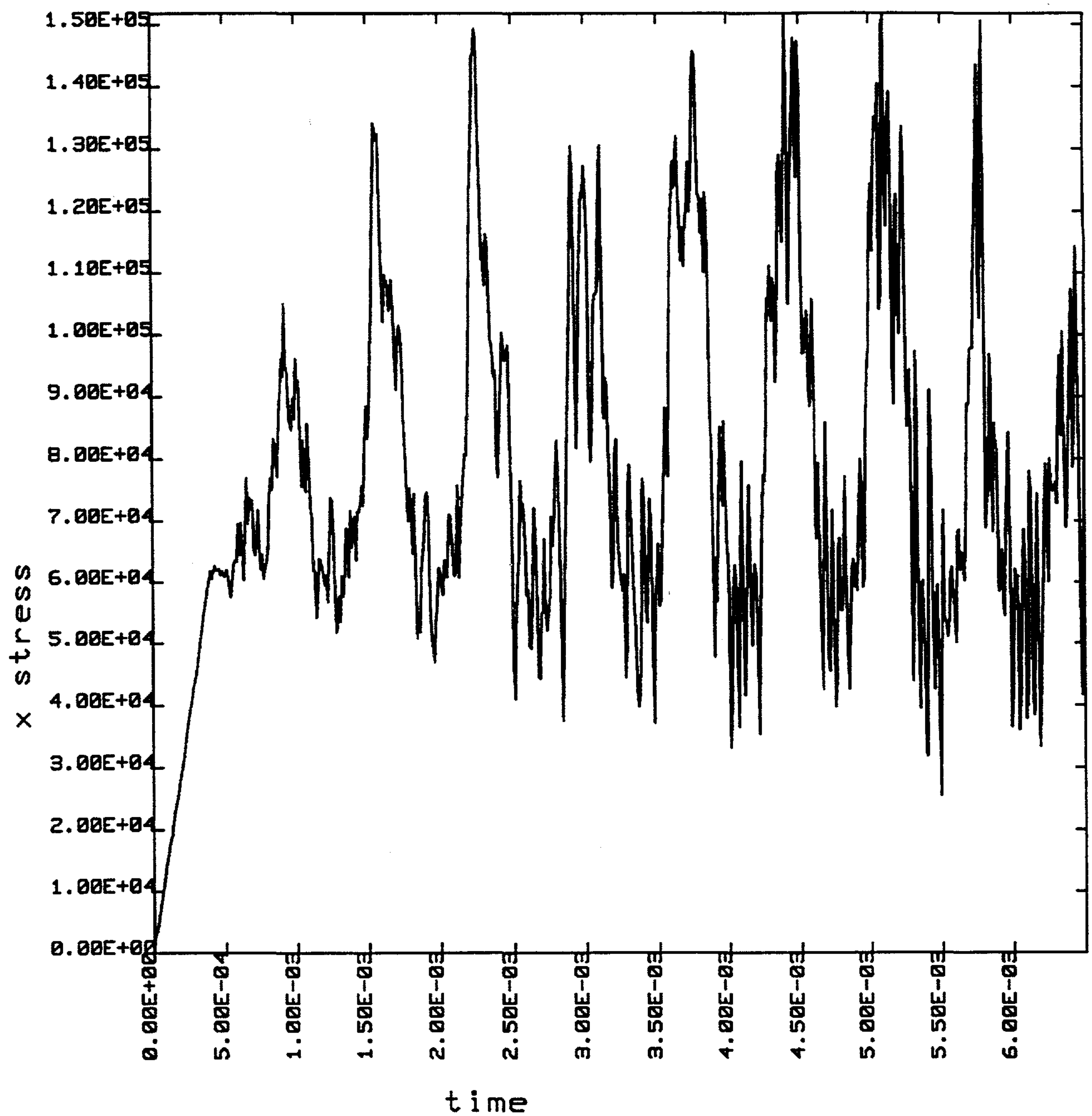


FIGURE 10. AXIAL STRESS - TIME HISTORY IN ONE BOLT.

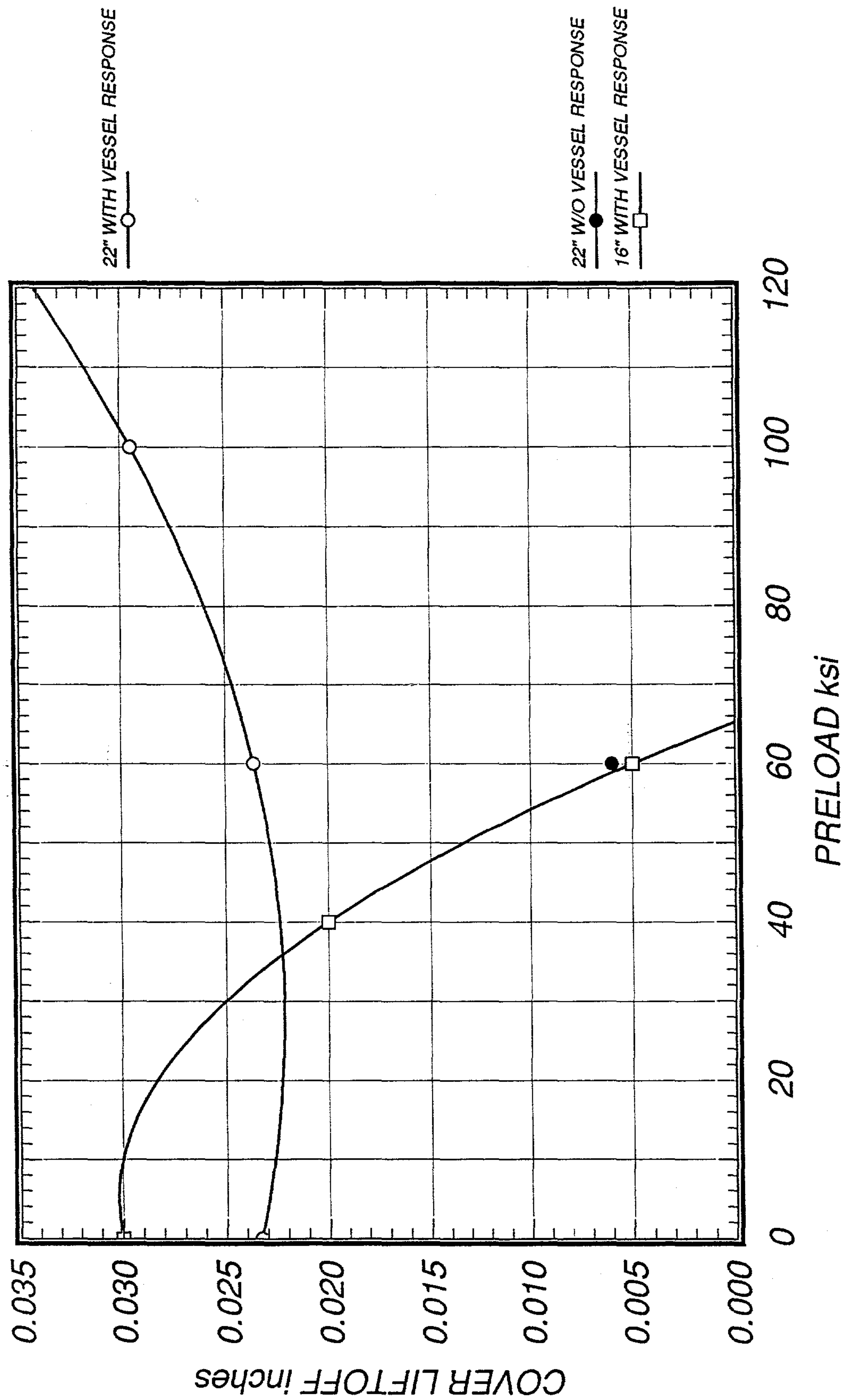


FIGURE 11. CLOSURE LIFT-OFF AS A FUNCTION OF BOLT PRELOAD.