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COMPENSATION FOR OPENINGS IN PRIMARY SHIP STRUCTURE

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When an opening penetrates the strength deck of a ship, the flow of stress in the deck is altered such that concentrations of stress form near the opening. Stress concentrations are of concern during both the preliminary and detail design and need to be addressed from the perspectives of strength and fatigue. The following paper illustrates the classification and analysis methods for openings in primary ship structures. An overview of this method and specific examples illustrate typical means of compensating and reinforcing deck openings.

The characteristics reviewed include the estimation of overall hull bending moment and longitudinal stresses for various opening configurations. Other features include the stiffness and strength characteristics due to openings and added stiffness in primary hull structures. This parametric assessment encompassed the qualitative evaluation of various configurations of holes and openings as well as their interaction with the members of the primary hull structure envelope.

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16. Abstract

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CONVERSION FACTORS
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To convert from	to	Function	Value
LENGTH			
inches	meters	divide	39.3701
inches	millimeters	multiply by	25.4000
feet	meters	divide by	3.2808
VOLUME			
cubic feet	cubic meters	divide by	35.3149
cubic inches	cubic meters	divide by	61,024
SECTION MODULUS			
inches^2 feet^2	centimeters^2 meters^2	multiply by	1.9665
inches^2 feet^2	centimeters^3	multiply by	196.6448
inches^4	centimeters^3	multiply by	16.3871
MOMENT OF INERTIA			
inches^2 feet^2	centimeters^2 meters	divide by	1.6684
inches^2 feet^2	centimeters^4	multiply by	5993.73
inches^4	centimeters^4	multiply by	41.623
FORCE OR MASS			
long tons	tonne	multiply by	1.0160
long tons	kilograms	multiply by	1016.047
pounds	tonnes	divide by	2204.62
pounds	kilograms	divide by	2.2046
pounds	Newtons	multiply by	4.4482
PRESSURE OR STRESS			
pounds/inch^2	Newtons/meter^2 (Pascals)	multiply by	6894.757
kilo pounds/inch^2	mega Newtons/meter^2 (mega Pascals)	multiply by	6.8947
BENDING OR TORQUE			
foot tons	meter tons	divide by	3.2291
foot pounds	kilogram meters	divide by	7.23285
foot pounds	Newton meters	multiply by	1.35582
ENERGY			
foot pounds	Joules	multiply by	1.355826
STRESS INTENSITY			
kilo pound/inch^2 inch$^{1/2}$(ksi$\sqrt{\text{in}}$)	mega Newton MNm$^{3/2}$	multiply by	1.0998
J-INTEGRAL			
kilo pound/inch	Joules/mm^2	multiply by	0.1753
kilo pound/inch	kilo Joules/m^2	multiply by	175.3

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1: INTRODUCTION

When an opening penetrates the strength deck of a ship, the flow of stress in the deck is altered such that concentrations of stress form near the opening. Stress concentrations are of concern during both the preliminary and detail design and need to be addressed from the perspectives of strength and fatigue. The following paper illustrates the classification and analysis methods for openings in primary ship structure. An overview of this method and also some specific examples to illustrate typical means of compensating and reinforcing deck openings will be provided.

The characteristics reviewed included the estimation of overall hull bending moment and longitudinal stresses for various configurations of openings. Other features include the stiffness and strength characteristics due to openings and added stiffness in primary hull structures. This parametric assessment encompassed the qualitative evaluation of various configurations of holes and openings as well as their interaction with the members of the primary hull structure envelope.

Under Task 2.0 a parametric assessment of characteristics and specifications of existing vessels was conducted. The parametric evaluation was performed across ship type and ship size. Among the ship types reviewed were bulk carriers, container ships, general cargo ships, roll on/roll off ships, tanker ships and naval combatants. Ship sizes evaluated were from 10,000 DWT to 150,000 DWT.

Task 3.0 and Task 4.0 were executed concurrently. The results of the parametric evaluation done in Task 2.0 were the building blocks of these two tasks. Using a qualitative assessment and the characterization of the types and sizes of openings, the penetration classification was developed. The classification of openings and holes will be under 3 broad categories, namely:

Small Holes (pipes & wire-ways)
Medium Holes (vents and access)
Large Holes (machinery space and hatch openings)

Methods were established to evaluate attributes and corresponding load assessments of openings. Criteria have been developed for openings and compensation. Engineering analysis methodologies were derived based on the qualitative evaluation of various configurations of openings and their interaction with the primary hull structural members. This was done for both uncompensated and compensated openings.

2: LONGITUDINAL STRENGTH AND HULL INTEGRITY

The three types of loading on a vessel are primary, secondary, and tertiary loads. These loads are additive and determine stresses within the ship structure. The structure must be designed to handle this loading. Understanding such loading is key to understanding the effect of discontinuities from openings in ship structure.

Primary loading is caused by global ship bending loads. To understand this it is useful to think of a ship as a simply supported beam subjected loads from waves. Primary loading usually results from two different circumstances. The first is hogging, where a substantial portion of the center of the vessel (amidships) is supported by the buoyant force of a wave while the rest of the vessel is relatively unsupported. This tends to produce tensile stresses in the strength deck and compressive stresses in the hull bottom. The second condition is called sagging and occurs when the vessel is supported at either end by a wave while the center of the vessel is relatively unsupported. This results in compressive stresses in the strength deck and tensile stresses in the hull bottom. Figure 1 below shows the vessel in the different global loading conditions.

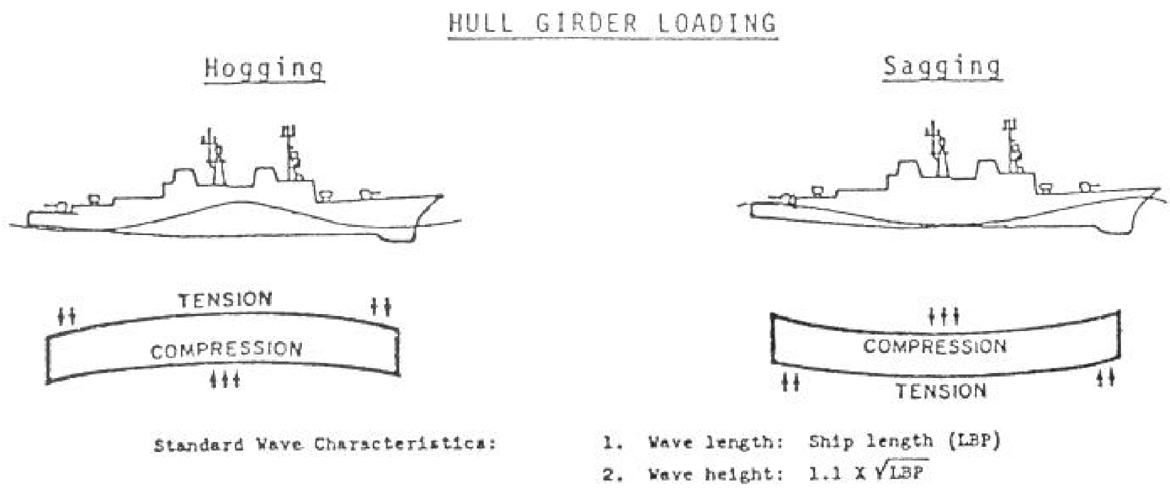


Figure 1: Vessel in Hogging and Sagging Conditions.

Figure 2 below shows the stresses from primary loading on a vessel. These stresses result from the conditions depicted in Figure 1 above.

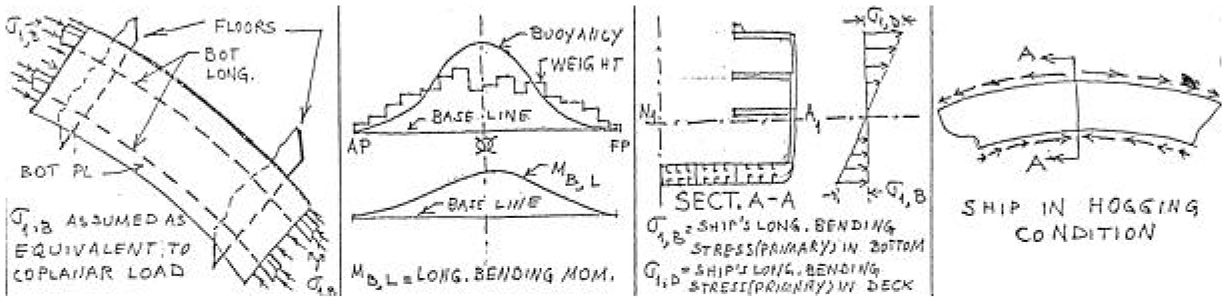


Figure 2: Primary Stresses Induced by Global Bending Moments.

Illustrated from left to right in Figure 2 is the bottom structure cutaway, loading/bending moment diagram, stress distribution, and direction of stresses.

This leads to an assumed primary stress distribution through the cross section of a ship's hull. Such a distribution is shown in Figure 3 below.

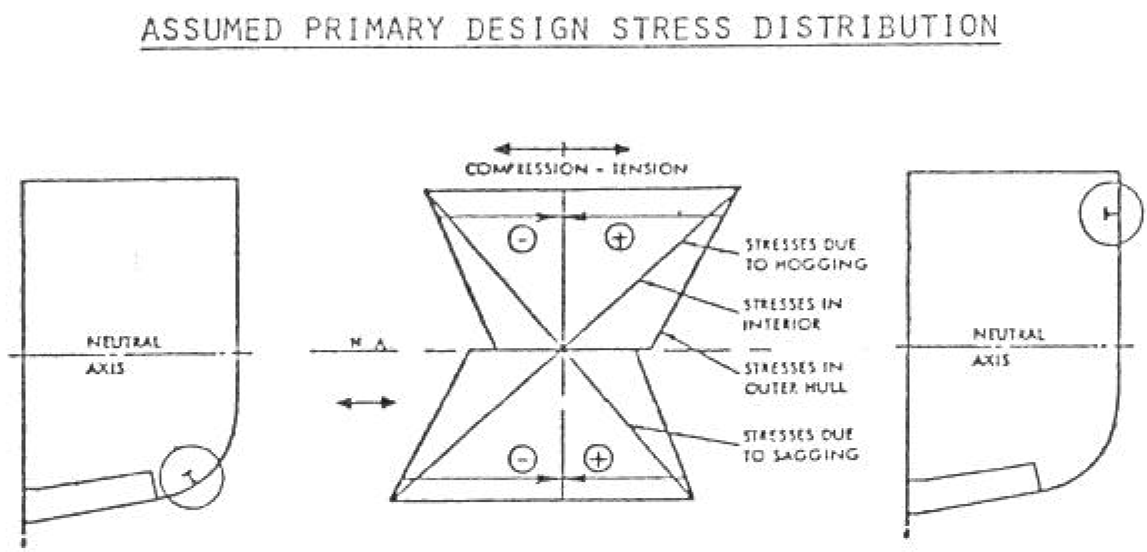


Figure 3: Design Stress Distribution through the Hull cross-section.

Note that the design stresses are not assumed to be zero at the neutral axis. In a beam, it is well known that the stresses at the neutral axis are zero. When designing a ship's hull, it is not appropriate to assume that zero stresses occur anywhere in the vessel. Designing to some primary stress here will help to account for double bending, rolling, and surge loads.

Secondary loads are induced in plating/girder combinations by local pressure between a pair of major athwartship supporting structural members. The local pressure causes bending in individual plates and stiffeners in the vessel. This bending varies according to the magnitude of the applied pressure. Examples of such pressure are hydrostatic pressure on the bottom of the vessel induced by slamming, pressure on decks from cargo, wind loading on the side or front of the vessel, and wave slap on side of vessel. Secondary stresses at any location in the plate-stiffener combination are typically assumed to vary linearly with the distance to the neutral axis. Secondary stresses will add to primary stresses to either increase or decrease overall stress depending on the manner in which the considered scantling is loaded locally and what part of said scantling is being considered. Figure 4 below depicts secondary loading.

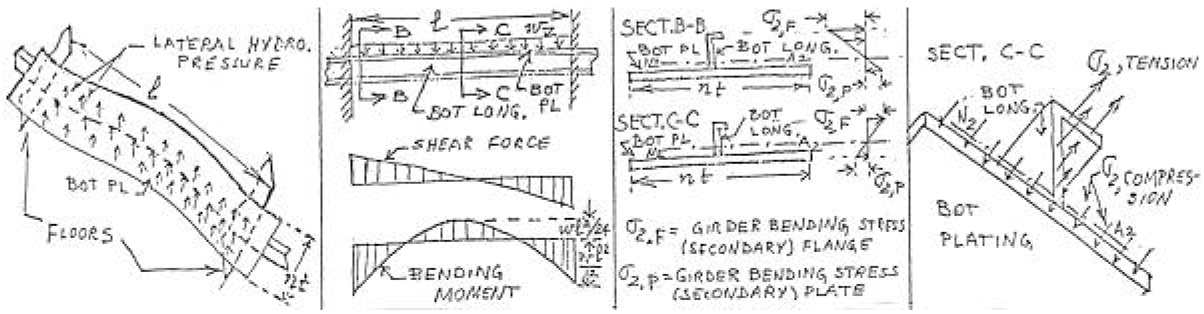


Figure 4: Secondary Stresses Induced by Local Pressure.

Illustrated from left to right in Figure 4 is a bottom structure cutaway, loading/bending moment diagram, stress distribution, and direction of stresses.

Tertiary stresses are bending stresses that occur in a panel of plating bounded and supported by two pairs of orthogonal stiffeners. These stresses are caused by lateral loading (hydrostatic pressure, deck loads, wind loads, wave slap, etc.) and only occur within the plate. The stress will vary linearly with the distance from the neutral axis, which is located halfway through the thickness of the plate. These bending stresses will add to the primary and secondary stresses to either increase or decrease the overall stress within the plate depending on the location, loading, and the area of the plate being considered. Figure 5 depicts tertiary stress.

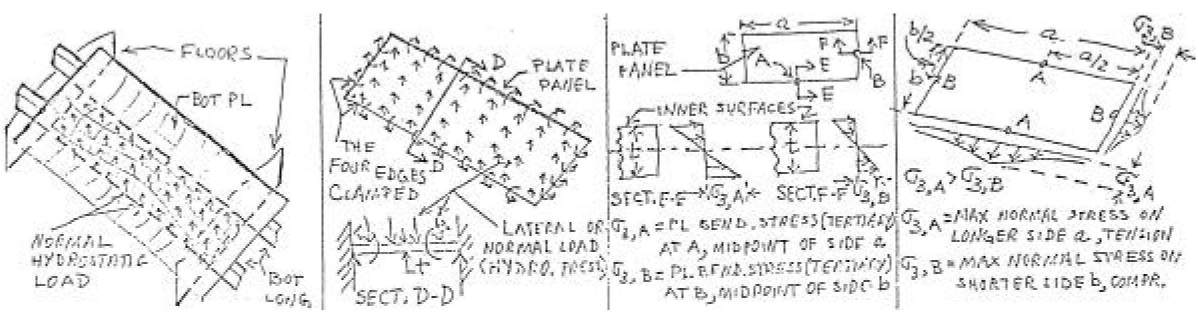


Figure 5: Tertiary Stress Induced by Local Pressure on Plate.

Illustrated from left to right in Figure 5 is bottom structure cutaway, loading/bending moment diagram, stress distribution, and direction of stresses.

The loads described above constitute the design criteria for any vessel. Such loads are typically determined with the aid of class society rules such as ABS (American Bureau of Shipping), Lloyds of London, DNV (Det Norske Veritas), etc. or in accordance with military guidance for openings in ships. The ship structure must then be designed to take these loads with the appropriate factors of safety. Typically, in preliminary design, the structure will be designed without too much consideration being given to local discontinuities, such as openings. However, as the design evolves, careful consideration must be given to such areas. Examples of discontinuities that need be considered are hatches in decks, piping openings in decks and bulkheads, ventilation openings in structure, openings for through hull fittings, etc. Such discontinuities will compromise the structure of a vessel.

The concern of this paper is with openings placed in primary structure. Primary structure is structure subjected to primary loads as defined above and where said structure is important and effective to the global integrity of the vessel. Primary loads include wave induced bending moments, shear forces, and torsion moments acting on the ship. These loads can be determined by many different methods. Examples include first principles calculations based on statistical data on waves and main vessel parameters or rule calculations from class societies (ABS, Lloyds, DNV, Navy methods, etc.).

The first method for computing primary loads on a ship is based on statistical data on waves. Approaches to global hull strength and fatigue evaluations encompass direct methods that calculate pseudo-static hull bending moments and shearing forces by integration. Also encompassed is the use of probabilistic statistics for wave height, bending moment and stress in a reliability-based approach to ship hull design. This cyclical pattern of wave loading is characterized to address fatigue performance as a key aspect of hull structural design. The development of global bending moments in association with an equivalent fatigue stress range can be used to establish a fatigue design methodology. Such a methodology would be applied to openings in primary ship structure.

The notion of cumulative damage to the ship's structure demands statistical definition of principal stress distributions with time. It has been found from full-scale ship measurements at sea that the distribution of stress, bending moments and wave amplitudes agrees well with the "Rayleigh function" over short periods of time.

The Rayleigh function defines the distribution of the wave heights, and the corresponding stress levels in equation 1.

$$q(x) = \left[x/m_o \right] \exp\left[-x^2/2m_o \right] \qquad (1)$$

Thus it may be confidently used as the short-term distribution. The summation of a number of short-term distributions results in a long-term distribution. The probability of exceeding this is defined in equation 2.

$$Q(x \rangle x_1) = \int_{x_1}^{\infty} q(x)dx = \exp\left[-x_1^2 \big/ 2m_o \right] \qquad (2)$$

The value m_o is the statistical variance of the random process, equal to the zero-th moment or the area under the spectrum of the process and $\sqrt{m_o}$ is the rms value. It may be convenient for the designer to work with equation 3, where the value Y is the maximum expected value in a given sample. [2]

$$\bar{Y} = C_n \sqrt{m_o} \qquad (3)$$

Both the “normal” and Weibull distributions fit measured full-scale stress and bending moment data well and yield similar results.

From a long-term wave distribution a wave bending moment in average irregular seas can be calculated readily in the conventional way from the estimated “significant wave height” and the average apparent wave period.

Statistical Wave theory does not define an upper bound. That is, any wave can be said to have a probability of exceeding the significant wave height of 13.53% (The significant wave height is defined as the average of the 1/3 highest of a group of waves).

The next method for determining primary loads on a ship is to use well accepted class society rules. Class societies such as ABS (American Bureau of Shipping), DNV (Det Norske Veritas), Lloyds of London, Bureau Veritas, etc. have established rules based on statistical data and well established theories over a long period of time. These rules establish a very straightforward means of computing primary loads. These are based on main vessel parameters and design standards that are set based upon the function and intended range of use of the vessel.

The ship structural designer is faced with the need to minimize the risk of failure. The wave stress and corresponding wave bending moment are used to determine the design bending stress. This stress is governed by the significant wave theory or class society rules. Thus, the section modulus for ships subjected to wave bending moments should be the basis for design that will result in a stress that is considered safe. It is also standard practice to ensure that the stress level assumed for preliminary design is a certain percentage less than allowable stress to accommodate future vessel growth and modifications.

Important to the process of achieving our goal is the ability to differentiate between primary and secondary hull structure. Also important is the need to determine the ship's primary hull structural envelope for the purpose of establishing penetration exclusion zones. Identification of applicable strength and fatigue criterion is critical since the evolvement and implementation of compensation and reinforcement methodology must be predicated to satisfy these criteria. The primary hull structural envelope and strength criterion is shown in Figure 6 for some military vessels.

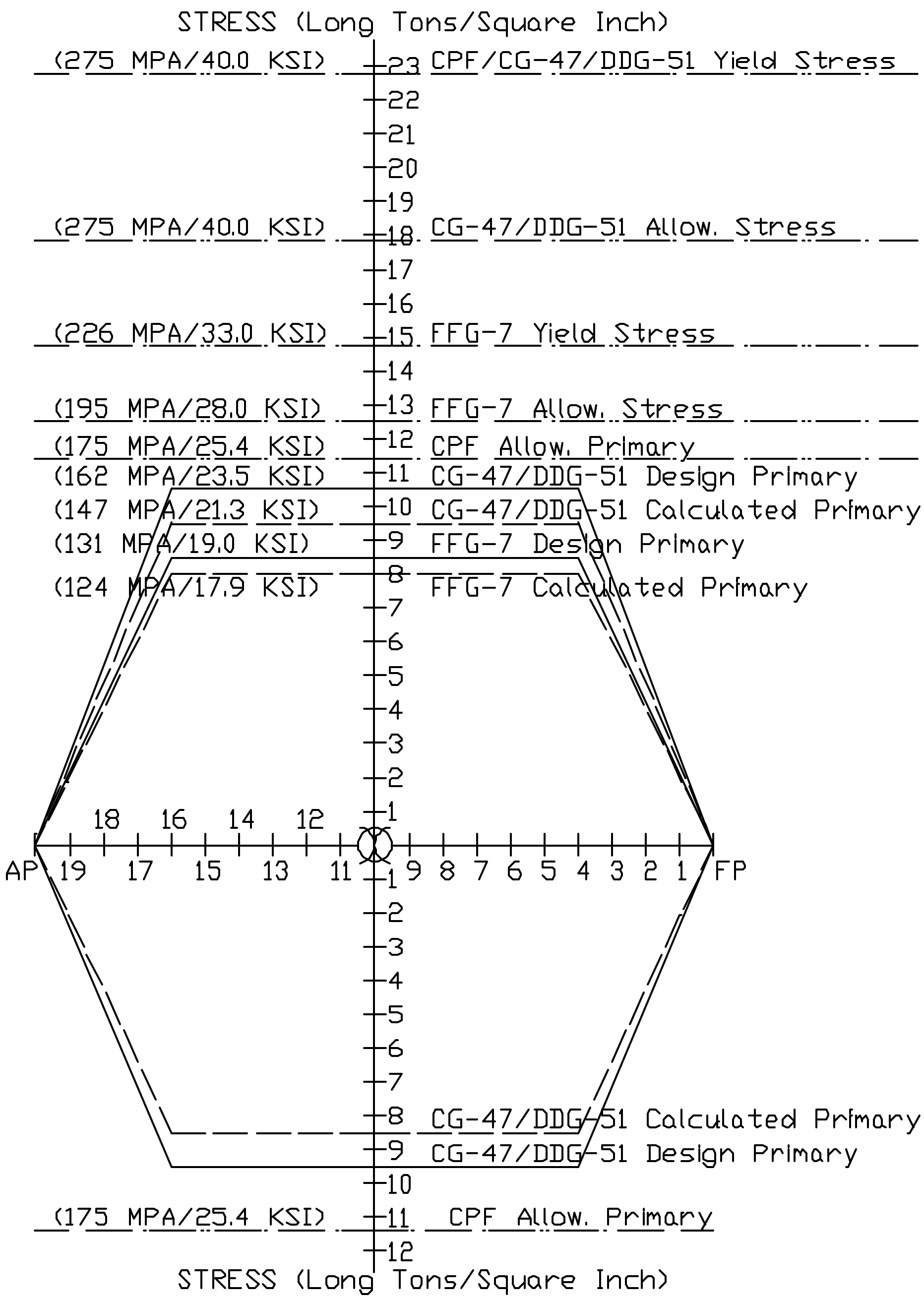


Figure 6: Ship Primary Hull Structural Envelope and Strength Criterion.

Figure 6 shows the hull structural envelope and strength criterion for three different military vessels, the FFG-7, CG-47, and DDG-51. Allowable and yield stresses are also shown for a fourth vessel, the Canadian Patrol Frigate (CPF). This figure shows different design criterion based on different stations (1 through 20) on the horizontal axis. Station 0 corresponds to the forward perpendicular (FP) and station 20 corresponds to the aft perpendicular (AP). The other stations occur at evenly spaced points along the length of the vessel. The origin of this axis corresponds to amidships. The vertical axis is stress in long tons per square inch. It is important to note that there are 2240 pounds in a long ton. The trapezoidal shapes above the horizontal axis represent stress in the strength deck along the length of the different vessels. The trapezoidal shapes below the horizontal axis represent stress in the keel along the length of the different vessels. Together, these form the hull structural envelope and strength criterion for the CG-47, DDG-51, and FFG-7. Note that for each ship, there is a calculated stress (dashed lines) and a design stress (solid lines). The calculated stress is the stress that was estimated in the final design. The design stress is the stress that is allowed. The difference between these is a reserve that allows for future growth in the vessel. For example, the calculated stress in the DDG-51 and CG-47 is 9.5 TSI. The design stress is 10.5 TSI. This allows some reserve such that a growth in stress of 1 TSI is allowed to occur over the life of the vessel. This might occur if additional weight is added to the vessel or modifications are made to the hull/deck that will decrease the vessel's section modulus.

The shape of the hull structural envelopes as shown in Figure 6 is critical. Observe that the stress tapers to zero in the forward fifth and aft fifth of the vessel. The main area of concern is therefore the center three fifths of the vessel. This is where the primary loads are greatest. It is within this area that designers must maintain the integrity of the primary structure. The effect of openings on structure within this area should be carefully considered.

Primary structure will only fall within amidships 3/5 the length of the vessel (60%), with the most critical area being amidships 2/5 the length (40%). The forward fifth and aft fifth of the vessel need not be considered during preliminary design as primary structure because these represent areas where primary loads "taper" to zero. Secondary loads tend to govern the design here. Within amidships 3/5 the length of the vessel, the following structure is defined as primary structure: The strength deck, the vessel bottom shell, the vessel side shell, continuous longitudinal bulkheads, and continuous internal decks.

The strength deck and vessel bottom plates are subjected to stresses resulting from the primary bending moments imposed on the vessel. These are perhaps the two most important primary structural elements in a ship as they are exposed to the maximum primary stresses as is shown in Figure 3 and Figure 6. Of specific interest here is the strength deck. This is due to the simple fact that most openings will be placed in this deck.

The hull sides are subjected to primary shear loads imposed on the vessel from the various conditions of wave support (i.e. hogging and sagging as shown in Figure 1). The hull sides are also subjected to varying stresses resulting from the vertical and, less predominant, lateral bending moments. However, these stresses are not as pronounced in the vessel's sides as they are in the vessel's strength deck and bottom shell. In addition, openings in the side shell in way of shear loads do not result in as much of a stress raiser as they do in areas of tension/compression loading.

Primary loading is applicable to longitudinal bulkheads that are continuous through 3/5 the length (60%) of the vessel amidships. This loading will be very similar to that described for the hull side shell. Longitudinal bulkheads that are not continuous are not subjected to primary loads.

Interior decks are the final consideration. Similar to the strength deck, interior decks will be subjected to tensile and compressive stresses resulting from primary loading. Conversely, the stresses will not be as predominant here as these decks are closer to the neutral axis of the ship. Bending stresses increase linearly with the distance from the neutral axis of the ship. The stresses will therefore be greatest in the strength deck and hull bottom, as these are at the greatest distance from the neutral axis. If the interior deck is not continuous through 2/5 the length of the vessel amidships (40%), it will not be subjected to primary loading and reinforcement or compensation of openings in this deck for primary stresses is not necessary.

Given all of these considerations, the most important primary structural element for consideration is the strength deck (weather deck) of a vessel. This is the most highly loaded primary structural element and will have a substantial amount of openings. Guidance will be developed for placing openings in this part of the primary structure for both preliminary and detailed design. This guidance will also be applicable to the other primary structural elements described above.

3: CHARACTERIZATION OF OPENINGS

Typically during the first few turns of the design spiral (preliminary design) hull structure is always analyzed from an idealized continuous perspective. That is, without openings. Unfortunately ships require openings in the structure in order to satisfy their function. Openings in the hull structure constitute a discontinuity that can result in fatigue cracking. The cyclic loading incurred from the hogging and sagging of the ship as it encounters waves causes this fatigue cracking. To compound the problem, once a fatigue crack has formed the tip of the crack stores an enormous amount of energy. This results in the propagation of the crack at an accelerated rate.

It is common practice in preliminary design to place openings wherever they are necessary to fulfill the functional requirements of the vessel. It is generally assumed that these openings can be compensated and or reinforced to maintain the structural integrity of the vessel. During the last few turns of the design spiral (detailed design) the appropriate means of accomplishing the reinforcement and or compensation of these openings is decided upon.

Despite the fact that little consideration is given to the effect of openings in the preliminary design phase, it is important that one have a basic understanding of the effects in way of different openings. Such effects are well understood and are detailed in this paper. These effects include stress concentration factors and stress patterns typical to different types and sizes of openings. It is important to understand the overall effect of openings on primary structure.

Openings are comprised of many shapes and are often arrayed in many configurations. They may be referred to as simple or complex openings. Openings where stress gradients are isolated from the superimposing effects of other openings or discontinuities and that have a basic geometry are considered simple. Areas of the ship that have complex stress fields, a complicated geometry and multiple closely spaced openings or combinations of these effects are considered complex. Analysis by the finite element method shall be required for complex openings. The loading on finite element models shall be derived from the Total Seaway Design Bending Moments. The method of applying the loads may vary as well as the complexity of the model. For example, applying the calculated axial stress for a given deck may be adequate to determine the stresses in way of a complex set of deck openings. However, a more complicated model comprising several decks and longitudinal bulkheads may be required to accurately assess the effects of multiple complex superimposed openings. This model may require that a bending moment be applied at the boundaries to assess the interaction between the longitudinal bulkhead and the decks. The size of the model is determined by assessing the boundary conditions.

The development of the hull structural design is based on preliminary longitudinal strength requirements. The development of the primary hull scantlings is based on satisfying section modulus strength requirements and stress range limits prescribed by ship specifications. Openings in ship's structure shall be configured to maintain structural integrity for the design service life and shall be compensated or reinforced to satisfy stress range criteria selected to provide adequate strength and fatigue performance over the life of the vessel.

Throughout the design evolution process, the ever-changing arrangements and their effect on distributive system runs as well as their structural openings will significantly affect the design effort. The designer must accommodate strengthening provisions by employing compensations and reinforcement. The identification of major distributive systems runs and their prospective locations for openings in primary hull structure should be part of a design integration process. Arrangements and structure are developed concurrently to effect an optimized solution to the iterative design cycle.

Openings (both major and minor) on primary ship structure has a significant effect on the development of deck plating arrangements and hull straking schemes. Employing compensation and reinforcement methods to support the preliminary hull structural design process will minimize this effect. Compensation methods requiring significant area will drastically increase the thickness of the plating in way of openings. Gradual tapering of deck plating and shell strakes forward and aft of the area being compensated is necessary to avoid excessive stress concentrations in way of thickness changes. Clever arrangement of the plating is important to minimize the number of plates with different thickness requirements to support practical shipyard stock ordering methods. The development of compensation schemes during the preliminary design phase should also address the provision of strength margin requirements in the selection of plating.

Weight growth appears to be a chronic problem in combatant design, even though considerable effort is expended to institute weight control during the preliminary and contract design phases. Very often during the detail design phase, problems not accounted for in earlier design phases result in necessity to substantially increase the section modulus of the vessel through the critical 3/5ths midship length. These requirements will significantly affect the plating thickness in way of compensation areas around major openings. Providing margin in the design may preclude a cascade of changes that result due to insufficient compensation. The development of adequate hull strength and compensation with sufficient margin has implications for weight estimating, ship stability and other naval architectural requirements. These requirements must necessarily be addressed during the preliminary design stage and carried forward into the detail design phase.

Other major considerations during both the preliminary design and detail design effort should be the effect of superstructure and longitudinal bulkhead interaction with the hull. The effect of this interaction on compensation and reinforcement strategy is important. Some considerations include:

- Development and effective integration of superstructure (both structure that is considered effective and that which is traditionally considered not effective for longitudinal strength) and hull longitudinal bulkheads with the overall ship structure.

- Account should be taken of asymmetrically stiff port and starboard longitudinal bulkhead structure since differences in stiffness affect the longitudinal stress distribution within the hull.

- Special care should be taken with openings in the longitudinal bulkheads that are placed to facilitate passageways and or distributive system penetrations and that may be adjacent to openings in deck and hull primary ship structure. The proximity of these openings may cause significant stress redistribution as well as excessively high stresses.

Since these effects may bear on the strength and fatigue capability of the hull, it may be necessary to evaluate these areas using finite element analysis. This analysis is performed to develop successful design accommodations to openings made in zones that are traditionally taboo.

Unfortunately, designers like to route distributive systems overhead in passageways, especially in ships with restrictive headroom clearances. Consequently, distributive system penetrations of longitudinal bulkheads occur often above penetrations made for doorway access through the bulkheads. These adjacent penetrations are often not envisioned during the preliminary design stage since the system runs and their requirements are not yet identified.

When considering the hull form it is most convenient to characterize it as a girder without any openings. The strength deck is represented as the top flange of the girder and the side shells behave like the webs. The bottom of the vessel imitates the behavior of the bottom flange of a girder. When the representative box girder is subjected to hogging the weather deck is subjected to tension. Likewise, when the ship is in a sagging mode, the deck is subject to compression. This sets up a cyclic loading pattern that imparts fatigue loading in the deck.

If one considers the strength deck of the vessel, one can determine the stress in the deck as though there were no openings. The stress inherent in this situation would be the force divided by the cross sectional area of the deck plate. The cross sectional area is represented in Figure 7 as A and is being subjected to a force P. Thus we have a stress of P/A in the large plate at the top of Figure 7.

An opening of any kind in the deck plating will disrupt this flow of stress and lead to a stress concentration in way of the opening. The stress is now forced to flow through the material remaining around the opening. Such a situation is shown in the bottom two cases of Figure 7. It is well known that the stress in the bottom two plates that have penetrations is no longer equal to P/A or even P/2a, but varies widely depending on the location in question. The areas closest to the hole will be affected by the discontinuity more than areas some distance away. This occurs due to the abrupt change in the geometry. This variation is referred to as a stress concentration. These stress concentrations have been calculated theoretically for various configurations of openings and experimental tests have confirmed the theory. It has also been determined that rounding the corners of an opening as shown in Figure 7 reduces this effect.

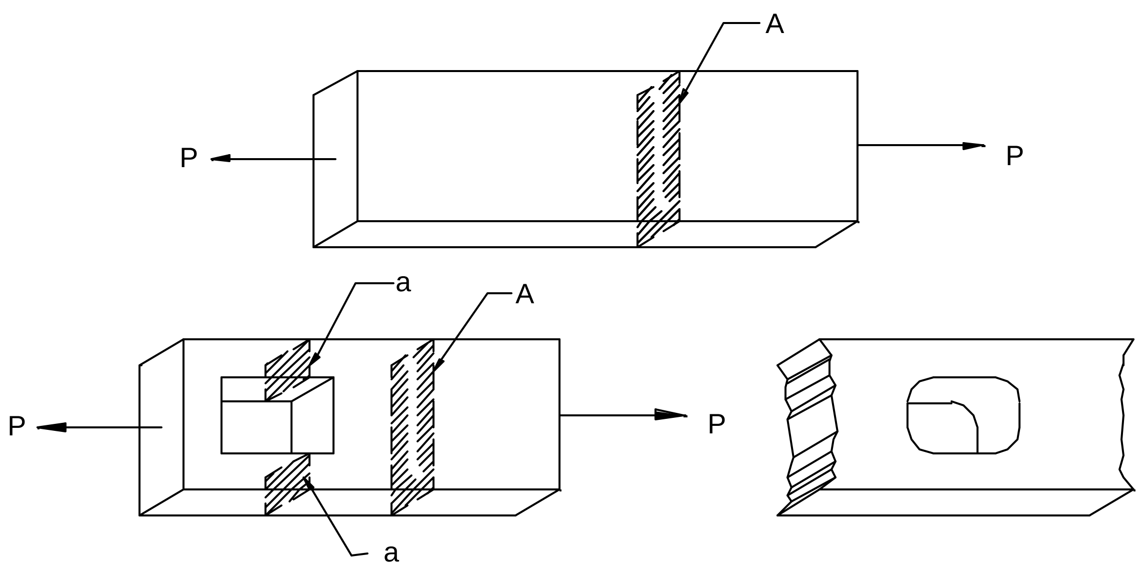


Figure 7: Plates of Uniform Thickness under Stress.

Through theory and experimentation, it has been shown that some of the stress concentration areas will have a stress value several times that of P/A or P/2a. The ratio of these increased stress values as compared to the P/2a value is termed the stress concentration factor.

$$K = \frac{\text{Local Stress}}{P/2a}$$

Given a plate under tensile stress, as in Figure 7, a K value greater than one signifies that the local stress exceeds the average stress in the smaller cross sectional area, 2a. A K value less than one but greater than zero indicates that the local stresses are reduced. K values of less than zero mean that the member is in compression.

It may be convenient to express the stress concentration factor, as it would relate to the average stress across the entire plate in the absence of openings. This would correspond to the cross sectional area A in Figure 7. This is termed the stress concentration factor k.

$$k = \frac{\text{Local Stress}}{P/A}$$

These two factors relate to each other in the following manner:

$$K\frac{P}{2a} = k\frac{P}{A} \qquad \text{or} \qquad K = k\frac{2a}{A}$$

For a plate with a finite width, 2a/A will always be less than one. Therefore the K value will always be smaller than the k value for unreinforced and uncompensated plates of uniform thickness. This will be true unless the plate is characterized as having infinite width, in which case the two values will be equal. To avoid confusion, K will be referred to as the local stress concentration factor and k will be referred to as the basic or far-field stress concentration factor.

Of particular interest to designers are the maximum values of the local and far-field stress concentration factors (K and k, respectively). The location and magnitude of this maximum will vary depending on a number of factors. These factors include the size of the opening, the shape of the opening, the number of openings, the location of the opening, and the relative size of the opening when compared with the structural member that it is in. In general, these values can be as high as 3 or even higher in the immediate area of the geometrical discontinuity. It is desirable to apply techniques to reduce these factors as much as is practicable.

As previously mentioned, the local and far-field stress concentration factors have been determined theoretically for a number of different opening configurations. There are several ratios that are related to the geometry of an opening in a plate. The stress concentration factors are dependent on these ratios. Figure 8 below shows a typical parameterized opening in a plate.

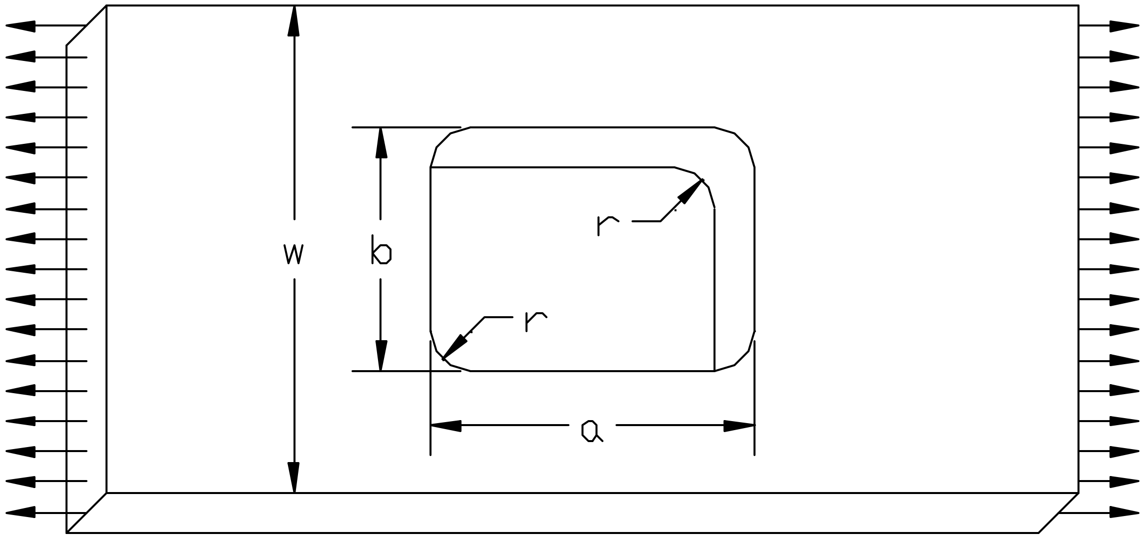


Figure 8: Parameterized opening in a uniformly loaded plate in tension.

In Figure 8, a is the opening's length (parallel to the plate loads), b is the openings width (normal to the plate loads), r is the radius of the opening's corners, and w is the width of the plate (normal to the loading). There are three ratios that can be derived from these parameters that are of significant bearing on the stress concentration factor. The ratio w/b is the plate to opening width ratio and is sometimes referred to as the relative size of the opening. The ratio r/b is the corner radius to opening width ratio. Finally, the ratio b/a is the opening width to length ratio. The ratio that influences K more than the others is typically r/b.

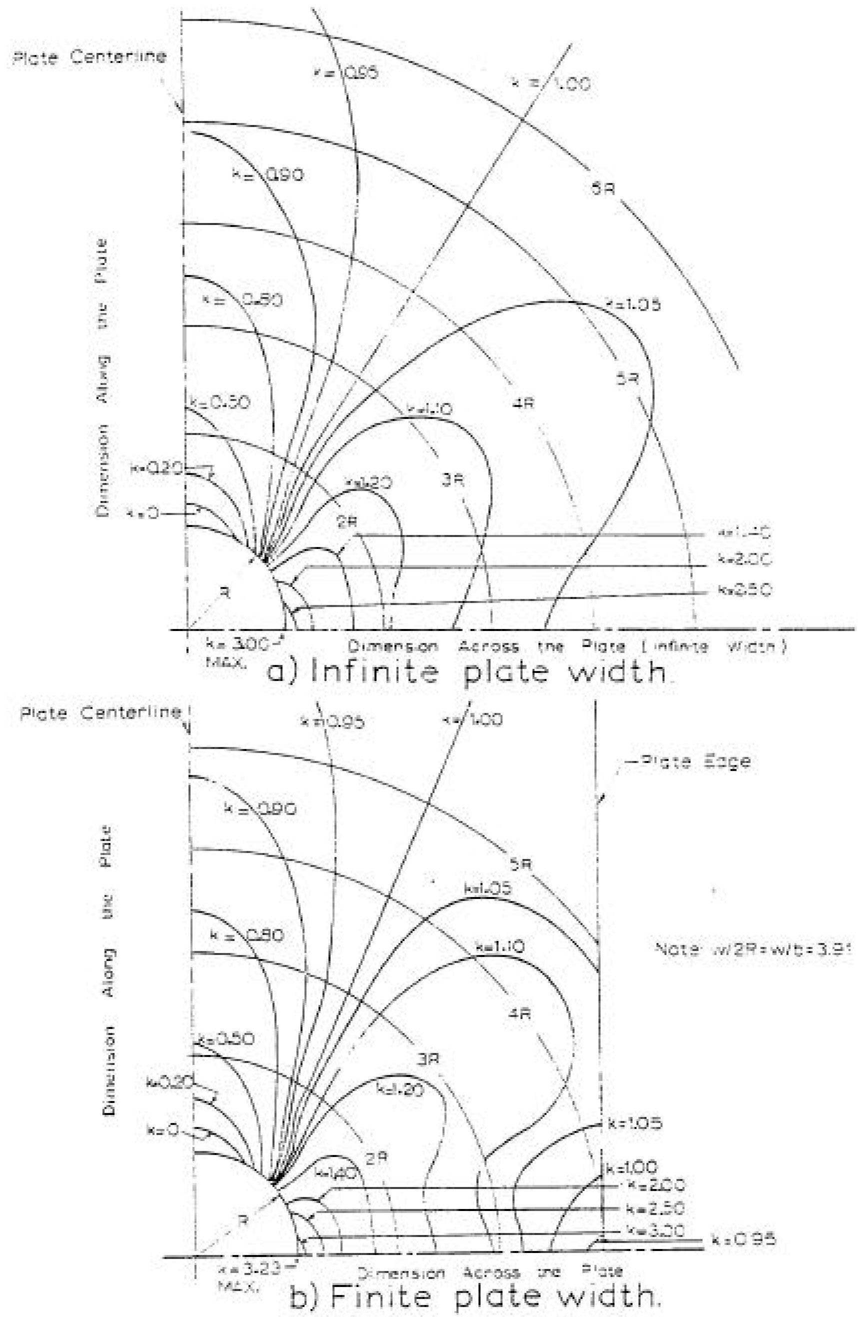


Figure 9: Local Stress Concentration around a Circular Hole in Plates.

Figure 9 shows how the local values of the stress concentration factor vary through a finite and infinite plate with a circular hole of radius r. This is useful to understanding how stress will redistribute in way of an opening such as a pipe penetration through a longitudinal bulkhead. The data shown is symmetric for each quarter of the plate. A good point to note is that if the plate width is at least four times the diameter of the hole, one may use the solution for an infinite plate. This is particularly useful for pipe penetrations through decks and bulkheads. A closer view of what happens in way of the hole for this case is illustrated in Figure 10 below [3].

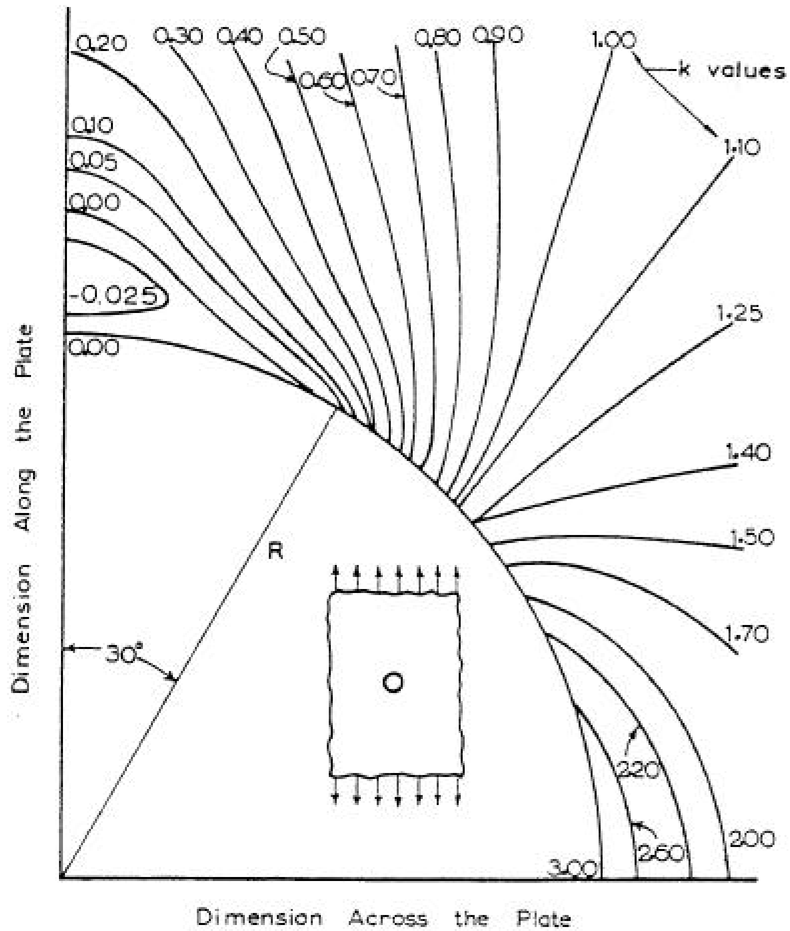


Figure 10: Local Stress Concentration for a Circular Hole in an Infinite Plate.

Should such a circular hole penetrate a smaller primary structural element, such as the web of a continuous longitudinal girder, it may be necessary to refer to Figure 11 below [3].

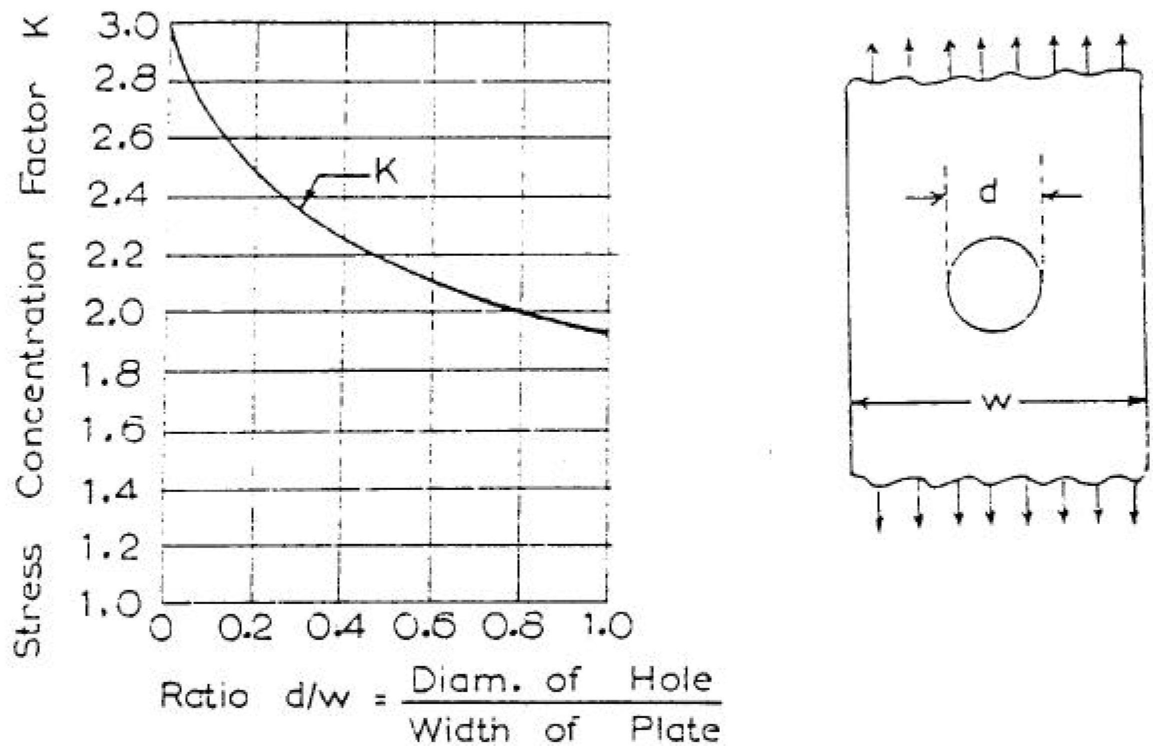


Figure 11: Stress Concentration Factor for Holes of Significant Size.

The information above should be of interested to the structural designers faced with pipe or wire penetrations in primary structure. It allows the designer to get a feel for what will happen to the stresses in way of this penetration. Discussions on how to design for this will be reserved for later. Also of interest are openings that are of a rectangular shape. Figure 12 below can be contrasted with Figure 9 and Figure 10. [3]

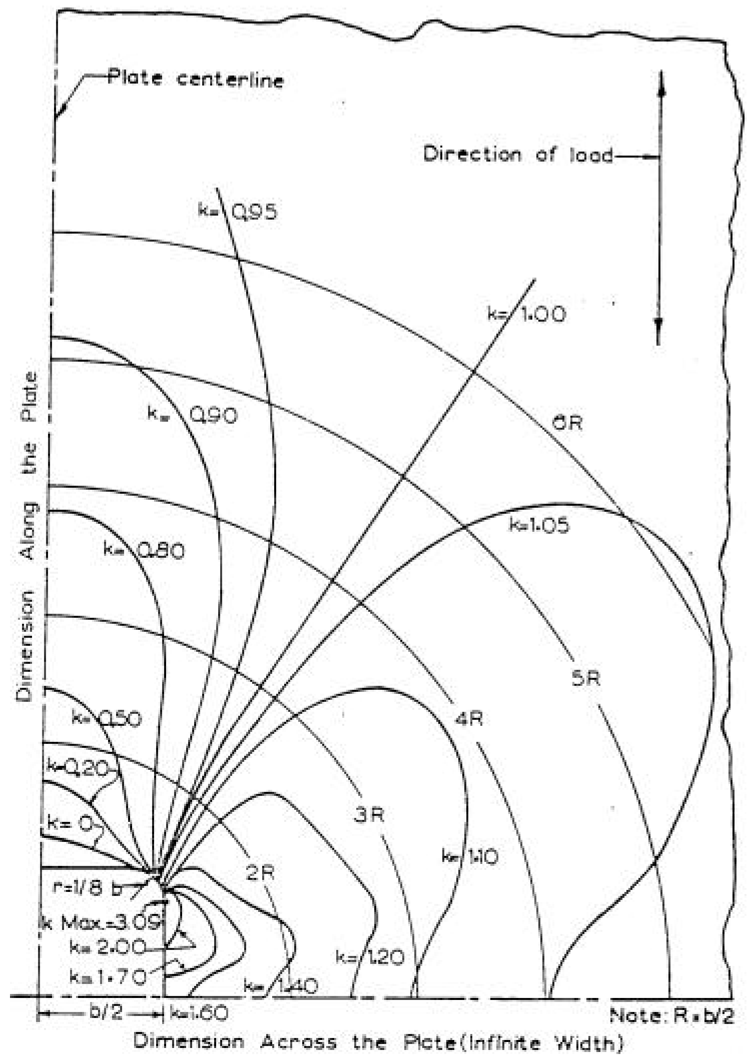


Figure 12: Stress Concentration Factor for a Square Hole in an Infinite Plate.

Note that the maximum value of the stress concentration factor in Figure 12 is 3.09. For a circular hole as depicted in Figure 9 and Figure 10, it is 3.00 for an infinite plate and 3.23 for a finite plate. The values for a square hole and a circular hole in an infinite plate are very similar and very close to 3. Another important parameter is the effect of the r/b ratio on the maximum stress concentration factor for rectangular openings. This is shown in Figure 13 below. [3]

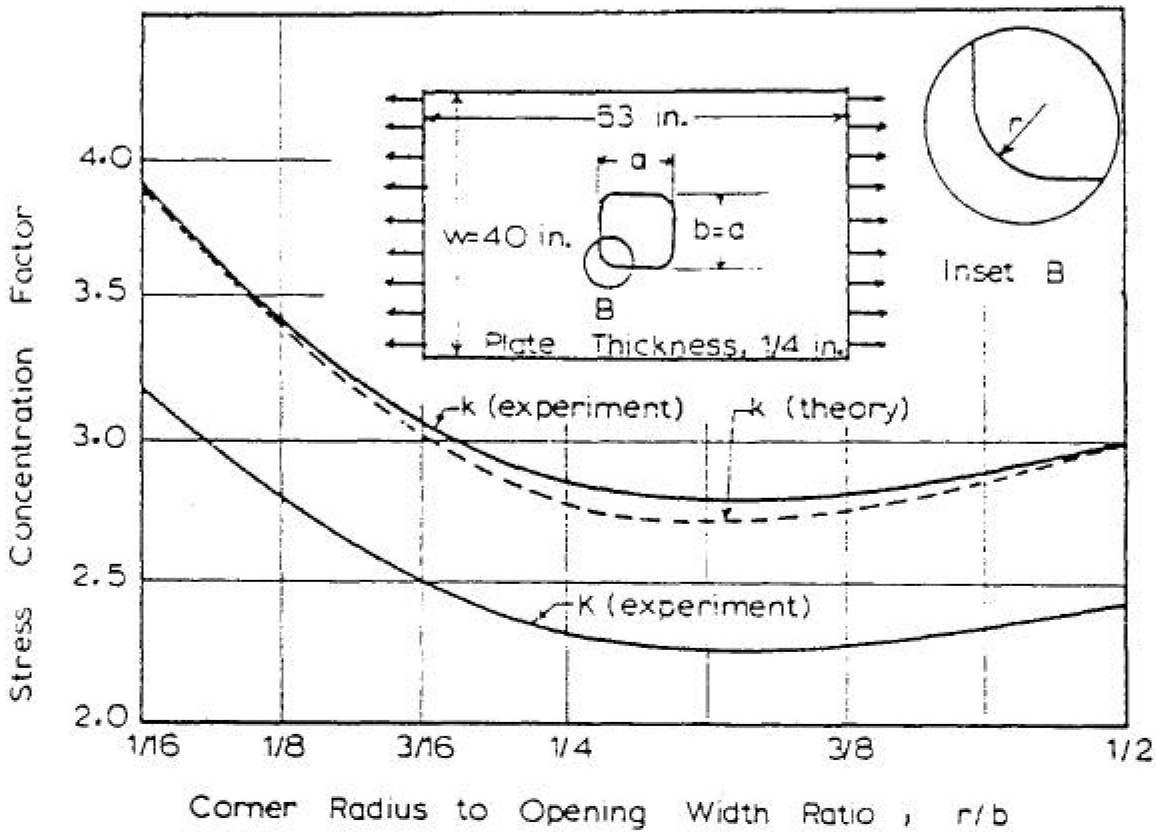


Figure 13: Variation of Stress Concentration Factor with Corner Radius Size.

From Figure 13 one can see that the worst stress concentrations occur as the radius becomes very small relative to the width of the opening. Also, as the radius approaches ½ the width of the opening such that one side of the opening is a half circle, the stress concentration factors approach that of a circular opening. The minimum seems to occur where r/b is equal to 5/16. Most guidelines/regulations require that an r/b value of at least 1/8 be used and a value of ¼ is typically recommended.

Figure 13 is applicable to minor and medium sized openings. Sensitivity to the relative size of the radius with respect to the opening decreases, as the size of the opening becomes significant (the curves in Figure 13 would tend to shift left as well as change shape). By significant it is to be meant that the w/b ratio becomes small (approaching 1). This is true of many major openings such as a cargo hatch in a container ship. In this situation it has been determined through experience that a radius that is at least 1/20 the opening width seems to provide acceptable results.

Another very useful guideline is that rectangular openings oriented such that their long dimensions are parallel to the direction of loads result in lower stress concentration factors than if they were oriented normal to the direction of loading. This is particularly important when placing openings that are allowed to vary in shape and orientation substantially, such as ventilation openings. This effect is shown through the b/a ratio (hole width normal to loading/hole length parallel to loading).

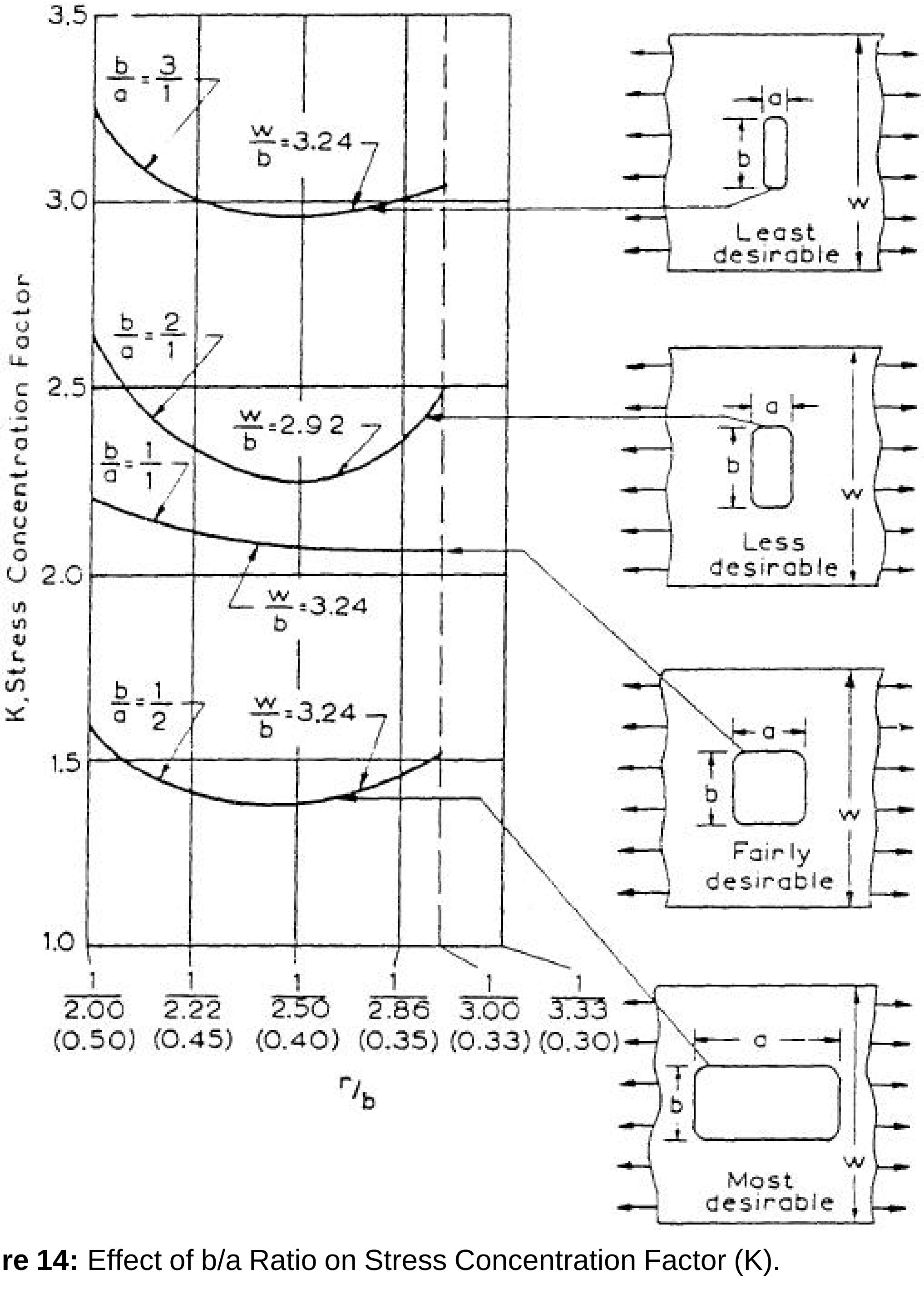


Figure 14: Effect of b/a Ratio on Stress Concentration Factor (K).

It should be apparent from Figure 14 [3] that if one were to take an opening with a b/a ratio of 3:1 and rotate it 90 degrees such that its b/a ratio becomes 1:3 a significant advantage is gained. The maximum stress concentration factor will drop to roughly half its original value. This illustrates the importance of orientation of openings. Another case where orientation is important occurs for a square opening with rounded corners. It can be shown that the opening should be oriented with one of its dimensions parallel to the direction of loading, rather than orienting it such that one of its diagonals is parallel to the direction of loading. The magnitude of this difference is dependent on the radius of the corners of the opening. For small radii (r/b=1:32), the stress concentration factor will double for the latter case. For larger radii (3:8), the stress concentration factor is roughly 20% greater for the latter case.

Perhaps the most important point of reference is presented in Figure 15 below. This shows the cumulative effect of aspect ratio and radius size for minor and medium sized openings [5].

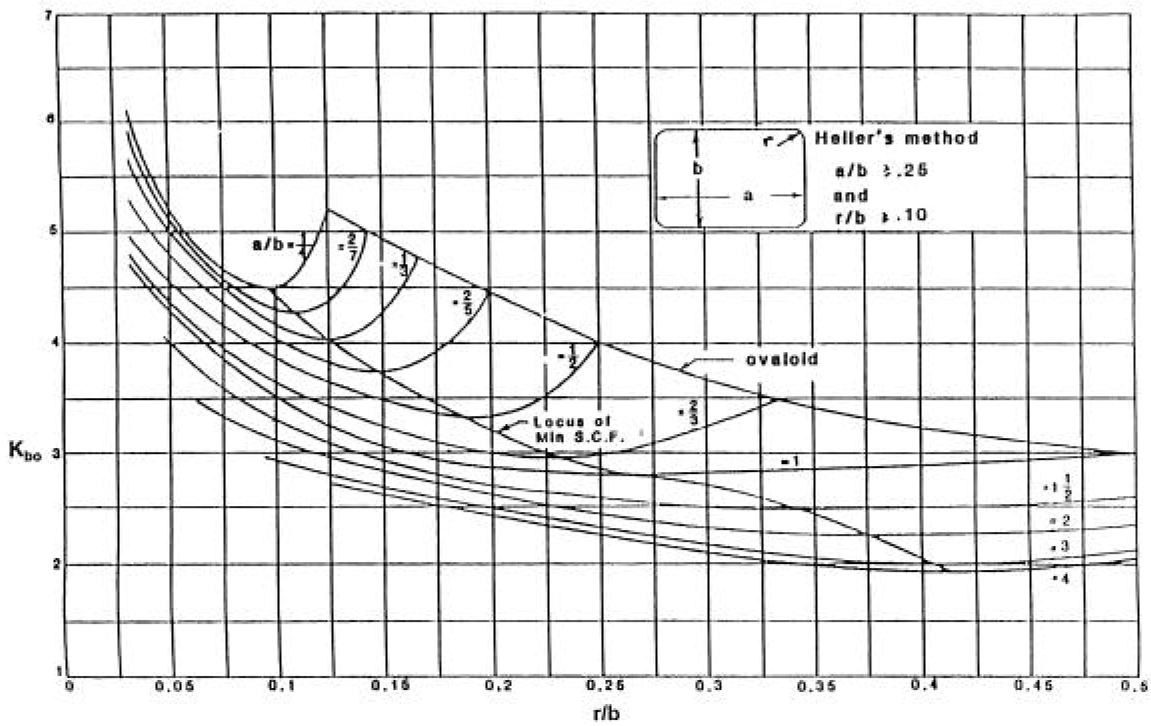


Figure 15: Far-Field SCF for Rectangular Openings.

Please note that K_{bo} in Figure 15 refers to either the local or far-field stress concentration factors (K or k) since they are the same for an infinite plate. Figure 15 is the most comprehensive source for stress concentration factors of minor and medium sized isolated openings. By isolated, it is assumed that a distance of at least four times the width of the smaller opening separates the openings. Figure 15 should be of interest to engineers working on distributive systems as well as engineers working on access hatchways.

Certain phenomena have been observed when considering a rectangular opening in a plate under tensile stress. It has been found that there are areas of reduced stress in the plate bordering on the edges of the hole in the direction of the stress. The corners of the hole do not exhibit this property. These areas are illustrated in Figure 16 below.

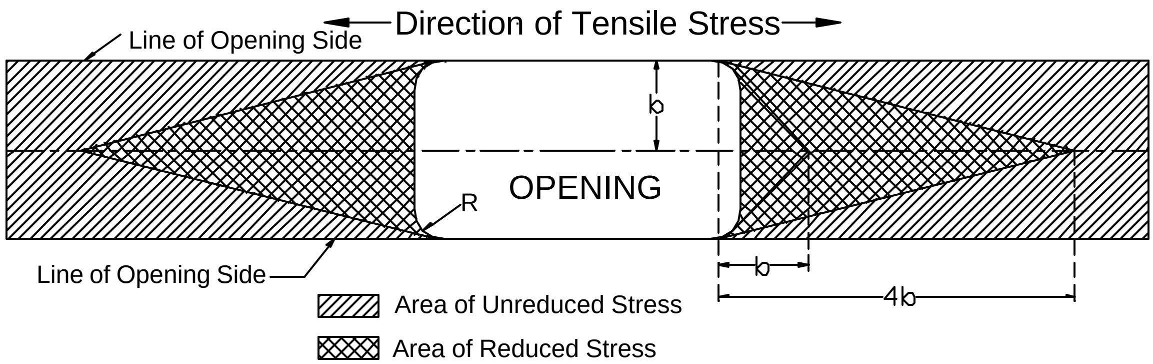


Figure 16: Rectangular Opening in Primary Structure.

A triangular shape that has amplitude four times half the width of the opening defines the area shaded as reduced stress. It is also known that the inner triangle, which has amplitude equal to half of the width of the opening, will exhibit stress levels of roughly one third of the stress levels present in the plate in the vicinity of this opening. Due to these stress levels, small openings should be placed in the reduced stress level areas whenever possible. Any holes that are outside of the reduced stress level area will need to be analyzed as a primary hole. Secondary openings that are between the smaller and larger triangle need to be examined in relation to the stress level at that point.

The triangular area of reduced stress is usually considered to be ineffective for the longitudinal strength of the vessel. The designer must carefully consider these areas and be certain that they are not included in calculations of the section modulus of the ship. This may become a serious concern if additional openings are placed fore or aft of the considered opening. If these additional openings are transversely adjacent and fall outside of the ineffective area, then the ineffective area plus the area of the openings must be deducted from the section modulus. Such a situation is illustrated in Figure 17 below.

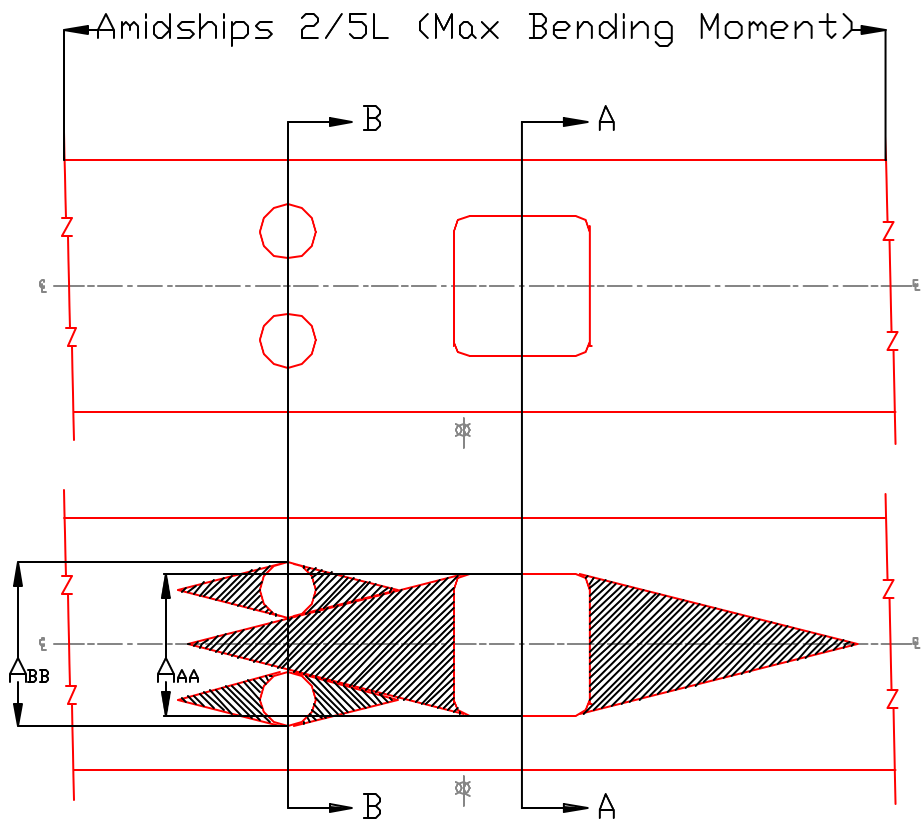


Figure 17: Ineffective Areas from Openings in Strength Deck.

Figure 17 is an excellent illustration of the problems a designer will face with ineffective areas of openings. The top of Figure 17 illustrates three openings in the strength deck of a ship. One is a large opening and the other two are transversely adjacent smaller circular openings. Both of these openings fall within amidships 2/5 the length. This means that the openings are all in primary structure and subjected to the varying bending moments in the ship. These sections must be designed for the same allowable stress levels. When a designer computes section modulus of the ship to determine worst stresses, they must compute it at the worst section (section that will yield the least section modulus). In Figure 17, it is apparent that there are two candidate sections at which the least section modulus will occur, section AA and section BB.

Looking at the top of Figure 17, it may seem intuitively obvious that section AA is the worst section. If the designer designs the vessel considering AA to be the worst section and the ship is eventually constructed based on this design, major hull failure may occur relatively early in the service life of the vessel. Such a failure will begin in the form of cracks in the outboard quadrants of the smaller circular openings. Eventually these cracks will propagate to the outboard sides of the deck and then continue down the sides of the vessel with disastrous consequences. Such instances have been known to occur in practice.

This type of failure would not have occurred if the designer had considered the stress flow around the openings. This flow leads to the ineffective shadow areas described above and illustrated in the bottom of Figure 17. Considering these areas, it is apparent that the ineffective area at section BB (A_{BB}) is larger than the ineffective area at section AA (A_{AA}). This means that section BB is the worst section and the section modulus must be computed through this section. The ship will be designed from this information. Additional considered models similar to Figure 17 are found in APPENDIX A.

It is a common functional concern that multiple openings be placed in line with the stress field. These situations are depicted in Figure 18, where the configuration a) would correspond to the openings in a destroyer deck, b) the hatches in a general cargo vessel and c) depicts the closely spaced openings typical of an ore carrier. The designer can utilize the superposition of these stress fields and the reduced stress areas. As the distances between the holes change the areas of reduced stress also change. Again it is advisable to include smaller holes within the crosshatched areas of reduced stress. The guidelines presented in reference to Figure 16 should be followed whenever possible.

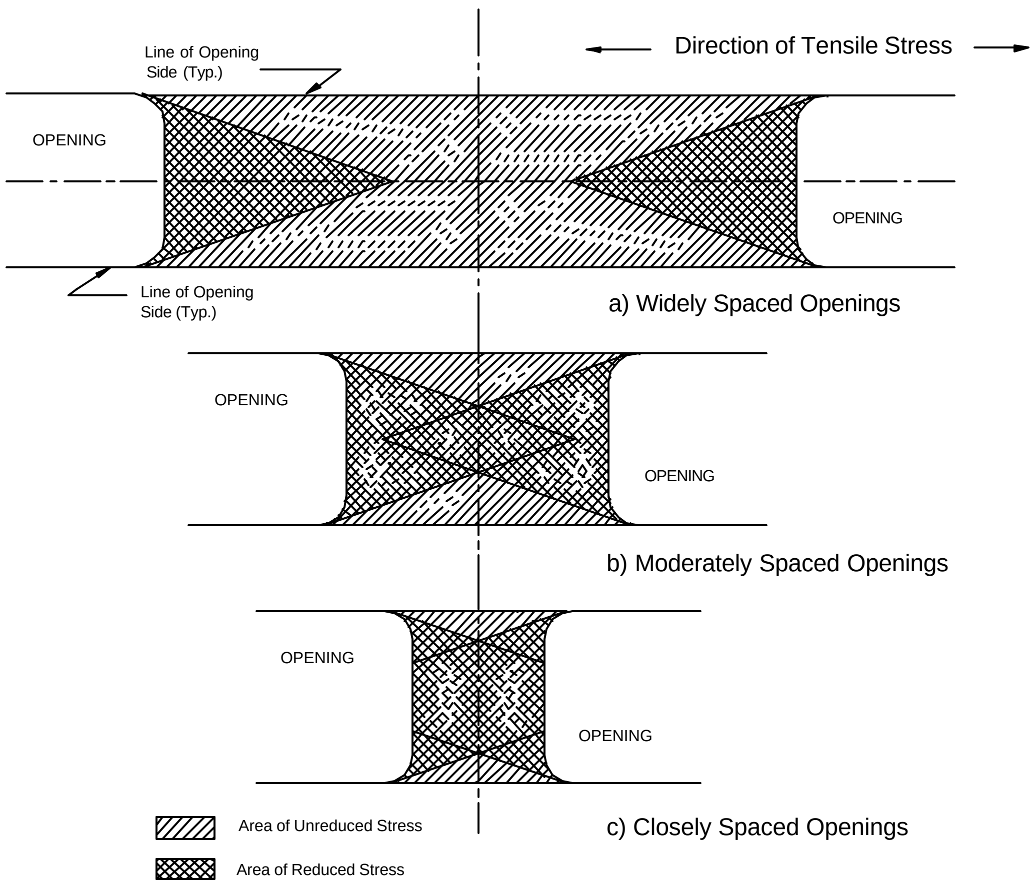


Figure 18: Multiple Rectangular Openings in Primary Structure.

It should be noted that openings in primary ship structure are not limited to deck openings. Thus far, the discussion has been limited in scope to members under unidirectional loading (tension or compression). Openings in a deck, such as the weather deck (strength deck) are generally very important as they relate directly to the section modulus requirements for global ship bending moments. Global bending moments are an integral portion of any preliminary design. A large hatchway can significantly reduce the amount of material available to resist the bending moment. These holes can then also have a large effect on other primary structure such as longitudinal box beams on cargo ships. When discussing openings in primary structure one must also include details regarding internal bulkheads. Longitudinal bulkheads may have a large or small influence on section modulus, but per the rules previously stated, they are typically included in the definition of primary ship structure.

In way of longitudinal bulkheads, openings present a different problem than they generally do when present in decks. A deck opening is generally analyzed in terms of the amount of longitudinal material that is absent and the stress concentration effect present at its edges due to corner shape and curvature. A longitudinal bulkhead however, generally has much more complex loading patterns due to its interactions with other pieces of primary and secondary ship structure.

A longitudinal bulkhead can be indirectly connected to the ship's hull by way of transverse bulkheads supporting it along its length. Since the transverse bulkheads are connected to the ship's hull, the longitudinal bulkhead is included in the section modulus calculations for global ship bending moment. The bulkhead therefore exhibits loading due to the global bending moment. In general, any longitudinal structure attached to the main hull will exhibit loading due to the global ship bending moment. In this case the bulkhead behaves as a beam in the longitudinal direction of the ship. The bulkhead will then exhibit bending stresses in tension or compression along the longitudinal direction. These forces can be represented as axial loads on the bulkhead, either compression or tension. This type of loading will tend to stretch or contract the bulkhead [3].

The longitudinal bulkhead is also subjected to vertical loads acting on it due to deck loading. The decks transfer vertical forces into the bulkhead which cause the longitudinal bulkhead to act as a large beam subjected to the vertical deck loads and supported again by the transverse bulkheads. The transverse bulkheads then distribute these deck loads into the hull of the ship. While the global bending moment stress load on the longitudinal bulkhead generally can be modeled as a set of forces acting axially on the bulkhead, the deck loading causes the bulkhead to exhibit both compression and tension forces varied along its height. Portions of the decks themselves will contribute to the section modulus of the bulkhead beam.

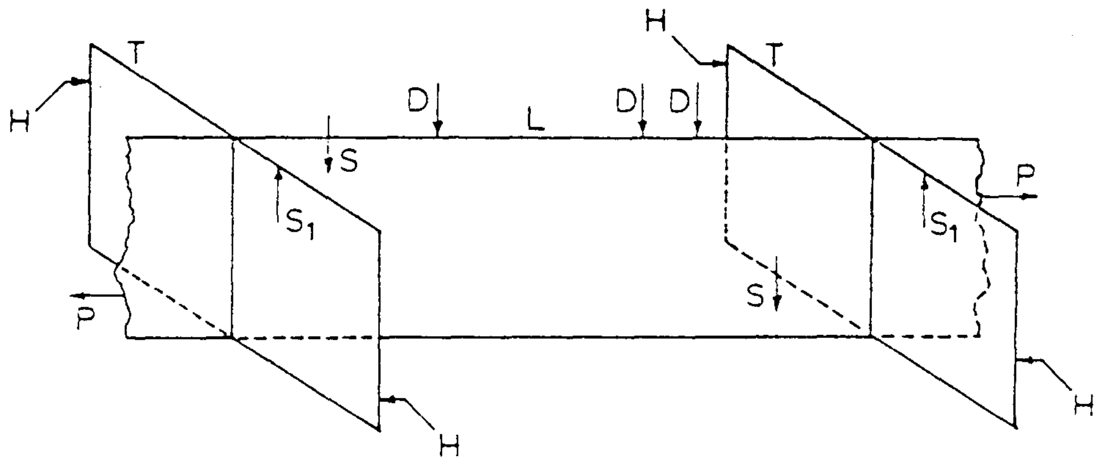


Figure 19: Load Bearing Intersecting Ship Structures.

D'Arcangelo presents a very concise description of the interactions between decks, longitudinal bulkhead, and transverse bulkheads in Figure 19 above [3]. In this figure, the longitudinal bulkhead is subjected to an axial bending load P due to the global ship bending moment. The bulkhead is further subjected to deck loads D acting vertically. These deck loads are then transferred into the transverse bulkheads by forces S. The transfer of vertical forces from one member to another in this way is described as shear. Thus the combined deck loading and reaction forces due to the transverse bulkheads cause both beam bending moment stresses as well as shear stresses in the longitudinal bulkhead. This combined state of stress for the longitudinal bulkhead is comprised of a global bending load P as well as vertical deck loading D and shear loading S. The shear loading is lowest at points far from the supports of a beam and increase closer to the supports. It is this shear loading that differentiates the analysis of longitudinal bulkhead openings from that of deck openings.

When performing a preliminary design of a ship, bending is generally considered to be a much more important issue than internal shear forces. This is most likely due to the required simplicity in a preliminary design. It is often much easier to size a midship section from a global bending moment than to perform shear calculations on individual pieces of primary structure along the length of the ship. Due to recent developments and realizations in the field of plate penetrations, shear has become an increasing concern, as will be explained in further detail below. There are two particular instances where shear loading is often an important factor. These locations are at support points, as between a longitudinal bulkhead and a transverse bulkhead, and at the ship's neutral axis in the vicinity of one quarter of the length from either end, typically referred to as the ship's quarter points [3].

It is a common misconception that the excess stresses often found in corners and intersections are due to tensile loads alone. Shear forces can be shown to contribute in a small or sometimes large proportion to these structurally significant areas. The above demonstration of a typical load distribution through ship structure will be useful in describing the design considerations of openings in longitudinal bulkheads. In this case an opening will be cut in the longitudinal bulkhead discussed above.

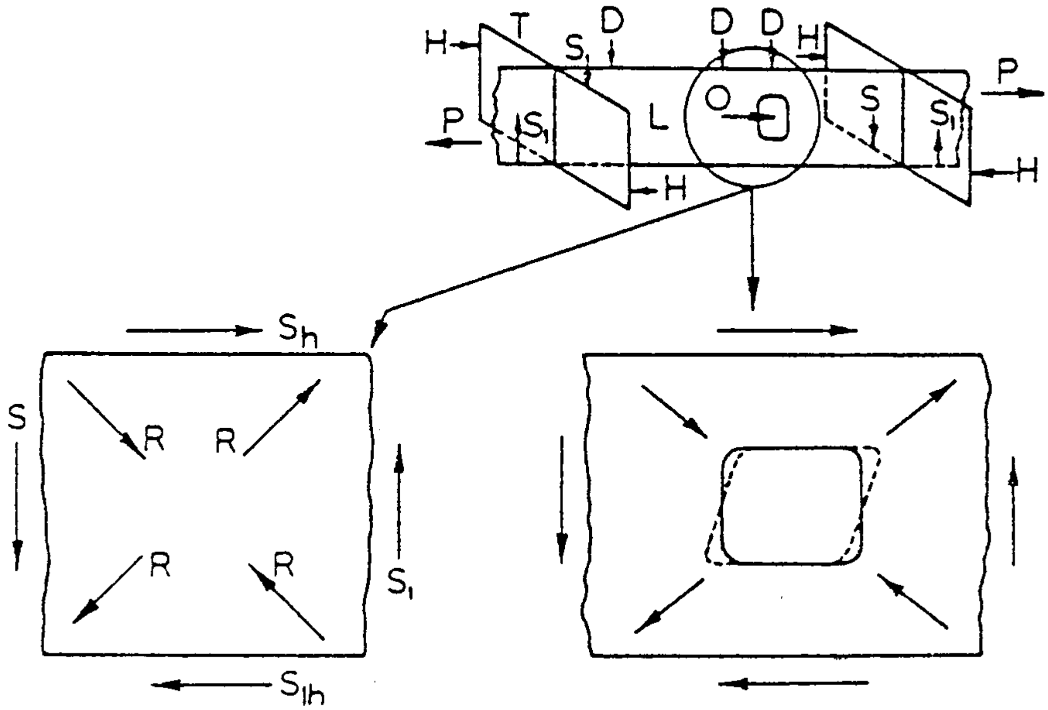


Figure 20: Shear Force Distribution in a Penetrated Longitudinal Bulkhead.

Figure 20 above shows the shear distribution in a system comprised of a deck, a longitudinal bulkhead, and a transverse bulkhead. The shear forces that exist at the intersection of the transverse and longitudinal bulkhead act on elements of the longitudinal bulkhead near the supports. A small section is represented in Figure 20 at the bottom left. These shear forces are called S and S_1. In order to have a proper free-body diagram of this element, these shear forces would have to be balanced by equal and opposite shear forces elsewhere on the section so as to prevent free motion in a theoretically static object. These reactionary shear forces, S_h and S_{1h}, are caused by the intersection of the longitudinal bulkhead and the deck. The combined shear loading is shown above. For representational purposes, the shear loading can be thought of more simply as a combination of tensile and compressive forces on the element. When two forces are of equal magnitude and act at 90 degrees from each other, a single force R acting at 45 degrees between the pair can replace them. One can surmise that the above shear forces can also be represented by the collection of tensile and compressive forces R shown in the figure. By utilizing this force substitution, the effects of shear on the element are easier to visualize. The forces R in the left bottom of the element and right top of the element will serve to elongate the element along their common axis. The forces R in the upper left area and bottom right will serve to compress the elements along their axis [3].

An opening O is introduced at the top of Figure 20. This hole is typical of a medium sized hatchway that would be found in a longitudinal bulkhead. Figure 20 at right demonstrates the results that the shear forces mentioned above would have on this opening. The effect on the opening is similar to what the effect would be on an element itself. The opening is compressed along an axis from one of its corners to the opposite one, and elongated along the axis containing the other 2 corners. While the effect on a representative element may commonly be disregarded since the deformation is often hard to notice, the deformation of the opening may cause problems in the structural analysis of the ship. Openings in longitudinal bulkheads should be located far from areas where high shear forces may be present, such as load-carrying members [3].

3.1: An example of the finite element method, Example A

Figure 21 represents an example of the train of thought that goes into the preliminary design of a ship's structure. This particular example was taken from the calculations and analysis of a Canadian Patrol Frigate. This vessel was constructed at the shipyard of St. John's Shipbuilding. This section also presents the calculations performed to determine the stress safety factor for each location where the stress level is significant.

The Bonjean curves determine the volume of the vessel. Given the volume of the vessel, specifically the volume of each section of the vessel, the bouyancy can be determined. The bouyancy and the weight of the vessel dictate the shear and bending moment of the vessel. The bending moment and the shear diagrams are integral to the design of openings and the reinforcement of openings.

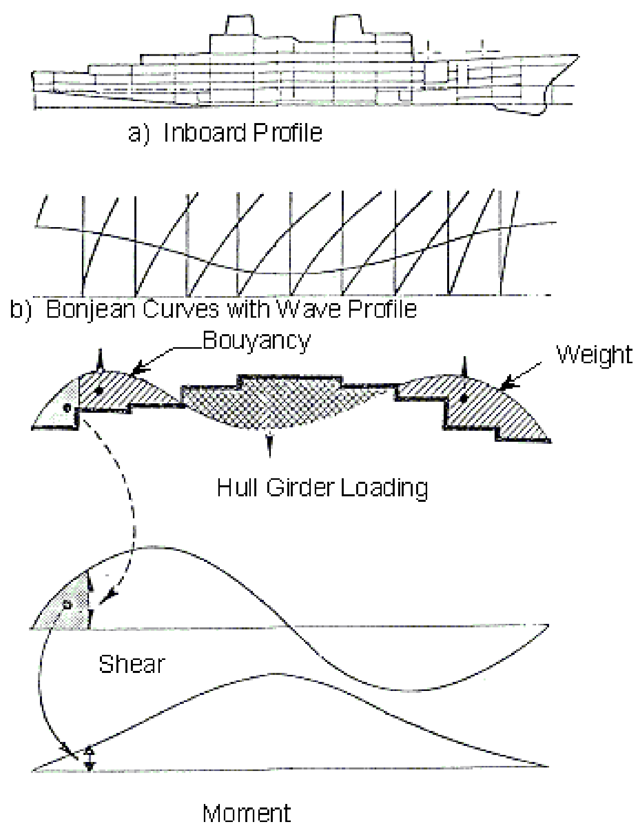


Figure 21: A representation of Buoyancy, Bending Moment and Shear.

Once a designer has determined the over all bending stresses, the local stresses can be determined for a specific opening. The stresses inherent in penetrating the deck are represented in Figure 22 below. Figure 22 is a depiction of the stress fields in the deck of the Canadian Patrol Frigate using a finite element analysis program. Throughout this section it will be illustrated that the results determined from the finite element analysis program concur with the results of a cruder and more arduous process within a reasonable percentage difference.

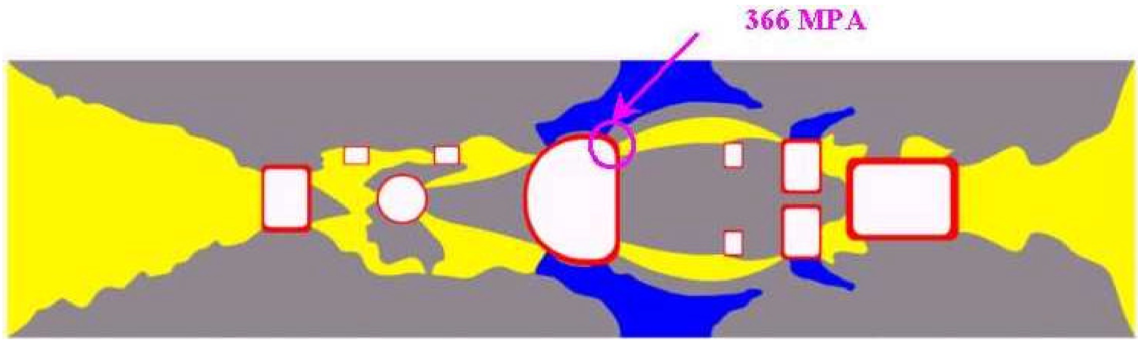
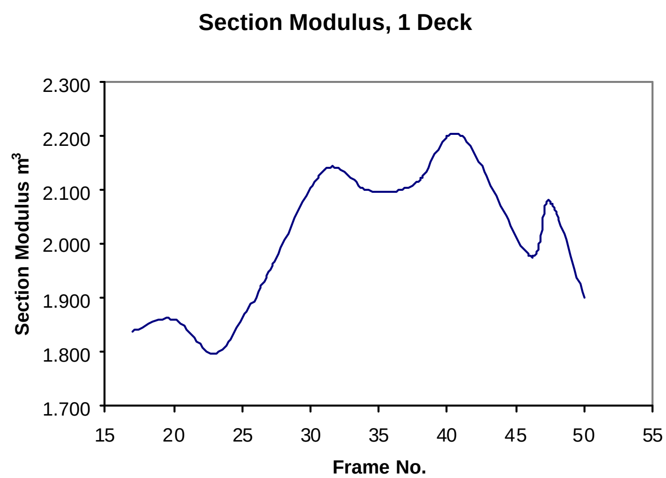


Figure 22: A Representation of Stress Fields.

Using the model depicted in Figure 22 the section modulus, bending moment, and stress were calculated at different locations in the vessels. These values are presented in Figure 23.

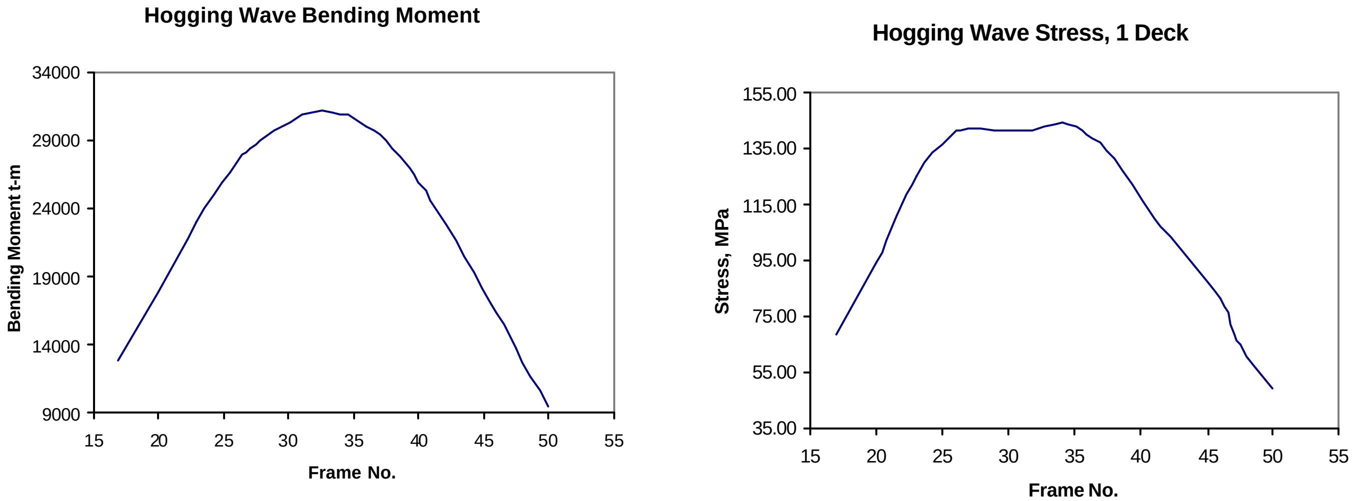
Frame #	Section Modulus m^3	Bending Moment (t-m)	Hogging Wave Stress, Mpa
17	1.839	12894	68.79
20	1.860	17897	94.37
23	1.797	22969	125.41
25.5	1.880	26613	138.85
27	1.949	28303	142.43
31	2.136	30785	141.40
34	2.101	30939	144.46
36	2.097	30028	140.51
38	2.119	28450	131.70
41	2.201	24685	110.03
46	1.978	16387	81.25
47.5	2.079	13678	64.55
50	1.900	9532	49.22

a) The values calculated for the deck of the Canadian Patrol Frigate



b) The Section Modulus vs. the longitudinal position on the ship

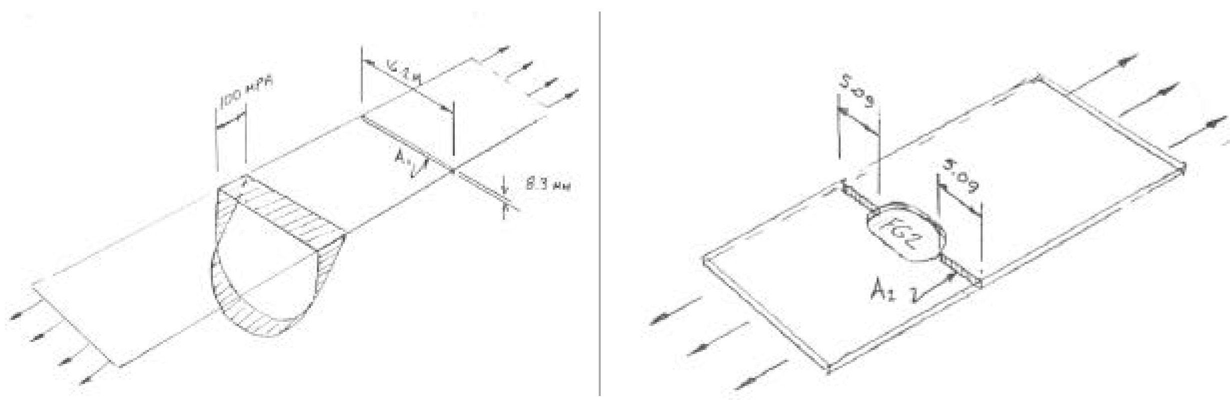
Figure 23: Strength and Section Modulus Data.



a) The Hogging Wave Moment vs. the longitudinal position on the ship.

b) The Stress vs. the longitudinal position on the ship

Figure 24: Bending Moment and Stress Distributions.



a) A Detailed Drawing of the Dimensions and Stresses.

b) The Cross Sectional Area of a plate with opening FG2.

Figure 25: Plate under Axial Stress from Global Bending Moments.

The unit load applied to the deck plating for these calculations was given to result in a stress of 100 Mpa. The cross sectional area was calculated from the measurements given in Figure 25, 16.2m wide by 8.3 mm thick. This yields a cross sectional area of 1344.6 cm^2.

The cross-sectional area of the plate that has been penetrated by an opening FG2 is calculated to be 845 cm^2.

$$A_2 = 1018\text{cm} \times 0.83\text{cm} = 845\ \text{cm}^2$$

To calculate the field stress in the way of the hole it is necessary to follow these steps.

$$s_{A_2} = \frac{s_{\infty} \times A_1}{A_2} = \frac{100\text{MPa} \times 1344.6\,\text{cm}^2}{845\,\text{cm}^2} = 159.12\,\text{MPa}$$

F_C = Stress Concentration = $\dfrac{s_{LFE}}{A_2} = \dfrac{366}{159.2} = 2.3$

σ_L = Local Stress = $\sigma_P \times F_C$ = 140 Mpa $\times$ 2.3 = 322 MPa

Conventional wisdom dictates that the stress can be calculated with a few simple steps. First it is necessary to develop a relationship between the stress in the cross sectional area A_1 and A_2.

$$A_2 / A_1 = \frac{1344.6\ \text{cm}^2}{845\,\text{cm}^2} = 0.6284$$

$$\text{Assuming}\quad s_{A_1} = 100\text{MPa} \quad \therefore \quad s_{A_2} = \frac{100\text{MPa}}{0.6284} = 159\,\text{MPa}$$

The Primary Stress, σ_P, is determined from Figure 6. This value is determined to be 140 Mpa. Thus the σ_P at A_2 is determined by:

$$s_P \times 0.6284 = 87.98\,\text{MPa}$$

For the opening FG2 at Frame 30 more calculation was done.

There is a relationship to determine the far field stress, σ_∞.

$$s_{\infty} = \frac{(A_j)(s_{A_j})}{B_{\infty} t} = \frac{845 \times 140}{1620 \times 0.83} = 87.98\ \text{MPa}$$

$$s_L = \text{Local Peak Stress}$$

σ_{LFE} = finite Element Analysis Local Peak Stress = 366 Mpa

σ_∞ = Stress at infinity (boundary of model) = 88 MPA

$$\therefore \quad s_L = \frac{366\,\text{MPa}}{100}(88) = 322\text{MPa}$$

Actual local peak concentration factor = $\frac{322}{140} = 2.3$

Allowable Stress (700 Mpa) = σ_{AL}= 0.5×700 = 350 Mpa

Safety Factor = $\frac{350}{322} = 1.09$

Reinforcement of this design example will be discussed in 6.3: Analysis of Reinforcement Requirements.

3.2: Complex Stress Patterns in Bulkhead-Deck, Example A

Presented in this section will be an alternative approach to design of that presented in 3.1: An example of the finite element method, Example A. That section presents a simplified method of approaching the problem. This section presents a finite element approach.

Complex stress fields in decks and longitudinal bulkheads result from primary stress, shear, and vertical forces induced by the interaction of the bulkheads and superstructure. Although the overall loads are simply primary loads, the three-dimensional stress flow that results can create high stress concentrations that are not readily apparent [LPD 17]. The finite element method should be used to assess the combined loading since a simple deck model with an applied axial stress would not reflect the complex stress field. Figure 26 below depicts the results of a finite element model that was created to illustrate a complex set of openings in a typical combatant. The model contained sections of two decks with a longitudinal bulkhead between them. The openings depicted in Figure 26 are in close proximity resulting in overlapping stress fields.

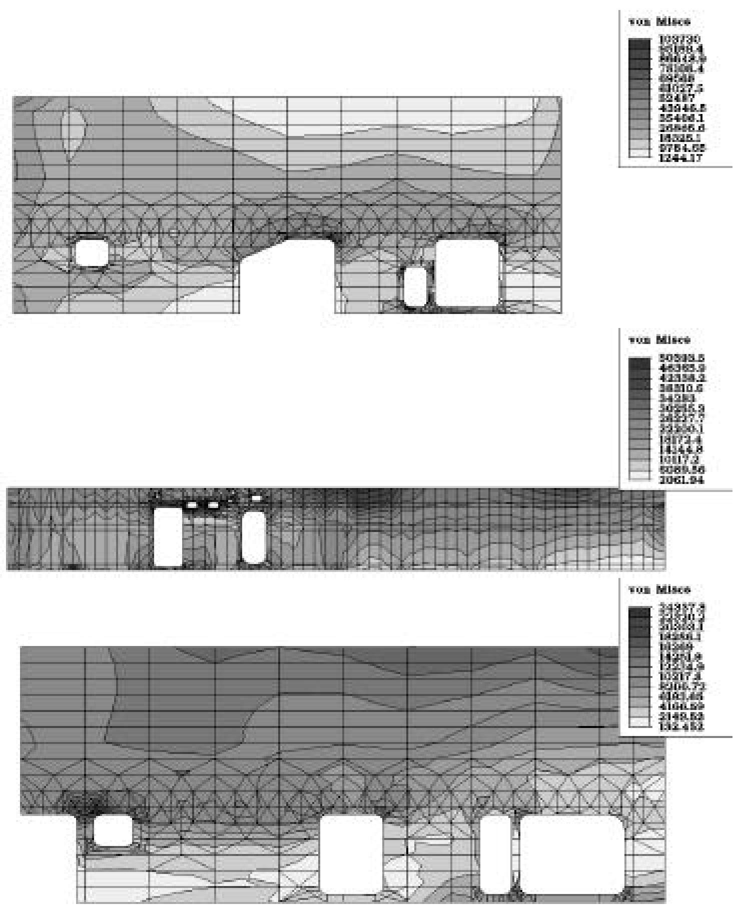


Figure 26: Stress Fields in Deck 1, Longitudinal Bulkhead, and Deck 2.

The following are excerpts from the Navy LPD 17 Section 100 Addendum 3 design specifications regarding finite element modeling technique in way of holes and basic stress concentration factor.

> *" **Loading** - The loading on the finite element model shall be derived from the total seaway design bending moments. The method of applying the load may vary depending on the situation. Applying the calculated axial stress for a given deck may be adequate in some cases. For a larger model, such as a ship module, an applied bending moment may be required to accurately model the loading. The size of the model is determined by assessing the boundary conditions. Some iteration during development of the model may be required to ensure that geometry is sufficient to produce the proper load paths. "*
>
> ***Boundary conditions and meshing** - When a finite element model is required it shall be constructed to reflect the primary structure contributing to stress flow so that the boundary conditions reflecting connections to the support structure do not affect the stress patterns in the areas to be considered, and the meshing is fine enough to provide accurate results.*
>
> *Finite element models shall ensure a satisfactory level of precision. In areas where the stress values are to be extracted, proper element configuration must be maintained. Quadrilateral shell elements must have levels of warp, taper, skew, and aspect ratio that are acceptable for the analysis package being used. Triangular elements should be avoided unless their performance relative to quadrilateral elements can be verified. Structurally important details shall be modeled with a mesh fine enough to produce accurate stress values at the areas of interest. In some cases, it may be necessary to prove that stress levels are accurate by performing sensitivity studies or supplementing the finite element analysis with manual calculations. Non-conventional modeling techniques shall be verified through the use of example finite element models.*
>
> *The precision of the model shall be tested by a means appropriate to the analysis package and shall indicate a difference in values at adjacent nodes of less than ten percent.*
>
> ***Stress results** - Analysis of complex arrangements will typically result in complex stress fields. The maximum principle stress shall be used to determine the peak stress. This is consistent with the theoretical development of stress concentration factors where a stress tangent to the radius is compared to the far-field stress.*
>
> *The far-field stress from the finite element model is that stress where the change or slope of the stress gradient is negligible. As one moves from the opening the rate of change in the stress gradient decreases and eventually the change levels off. This is the start of the far-field stress region. The basic stress concentration factor is calculated from the results of the finite element model by calculating the ratio of the peak stress to the far-field stress. Basic stress concentration factor for the finite element model would then be equal to the maximum principal stress divided by the far-field stress."* [5]

Compensation and reinforcement of complex openings is discussed in 6: COMPENSATION AND REINFORCEMENT METHODS.

4: PRELIMINARY DESIGN

As discussed previously, during the first few turns of the design spiral, little consideration is given to the effect of openings on ship structure. Openings are typically placed, as needed, to fulfill the functional requirements of the vessel. Such requirements typically include minor openings, medium openings, and major openings. Minor openings typically include penetrations necessary for basic distributive systems. Medium sized openings include access hatches that are required for egress and also those that are required to meet regulations. Ventilation openings also typically are classified as medium sized openings. Major openings include such things as cargo hold hatches, aircraft elevator openings and any other opening that is relatively large in comparison with the primary structural element being penetrated. Some guidance for placing and sizing openings during the preliminary design phase is as follows:

1) General Guidance for Openings in Primary Structure

    1.1) Penetrations should only be used when they are absolutely necessary.

    1.2) Openings should be sized according to what is required, and no more.

    1.3) Openings should be placed in the structure to minimize the need for compensation.

    1.4) Careful consideration must be given to placement of multiple openings in the same vicinity.

    1.5) The number of openings should be minimized where multiple openings are required.

    1.6) Consideration must be given to closely spaced openings whose adjacency is normal to the direction of primary loading. It is sometimes advantageous to replace these with a single, larger opening. This is especially true for many distributive systems.

    1.7) Good communication between systems design groups, penetration control, structural engineers, and production engineering and planning is vital.

    1.8) Standards should be established for cutting and welding penetrations.

2) Location of Openings

    2.1) Openings in structure should be spaced, when possible, such that they are near the centerline of the structure that is parallel to the direction of primary loading. This ensures that stress can flow evenly around either side of the opening.

2.2) 2.1 is especially applicable to the strength deck, where the shear lag effect can be taken advantage of by placing openings in the center 2/3 width of the deck where stress is lower.

2.3) Multiple openings that must be located adjacent to each other should be placed in line, such that their line of adjacency is parallel to the direction of primary loading. This allows one to take advantage of their mutual stress-relieving characteristics as illustrated in Figure 18 in 3: CHARACTERIZATION OF .

2.4) Openings should not, if possible, be placed in a line normal to the direction of loading as more structure will become ineffective than necessary

2.5) A minimum distance of 4 times the width of the smallest opening should separate openings. If this is not possible, 1.6 should be considered.

2.6) If small openings must be placed adjacent to larger openings, they should be located in the area of reduced stress (ineffective area) shown in Figure 16 in 3: CHARACTERIZATION OF .

3) Shape and Size of Openings.

3.1) Sharp corners should never be used in openings in primary structure. All sharp corners must be rounded.

3.2) Openings should be as small as practicable.

3.3) For openings that are not circular, the minor dimension of the hole should always be placed normal to the direction of primary loading. The major dimension should be placed parallel to the direction of primary loading. For square shaped openings, a minor dimension should be placed normal to the direction of loading.

3.4) Radii should be selected for corners of small and medium sized openings to minimize the stress concentration factor. This can be accomplished using Figure 13 in 3: CHARACTERIZATION OF . A minimum standard radius size of 1/8 the width of the opening should be employed. A size of ¼ the width of the opening is recommended.

3.5) Radii for corners of major openings can be relatively smaller in size. It has been determined that a minimum size of 1/20 the opening width seems to work well in practice [3].

3.6) Larger radii are desirable from a fatigue standpoint as well as from a static stress standpoint.

4) Surrounding Structure.

4.1) Plating in way of major openings should be compensated (thickened) to make up for lost area due to the opening.

4.2) At the preliminary stage, the plating need only make up for lost areas. Details on straking need not be considered yet.

5: REPRESENTATIVE MODELS

Original models of the deck with representative patterns of openings were to be created with longitudinal and transverse stiffeners supporting the deck. The two methods that were to be used for modeling the stiffeners were to model them as plate elements connected to deck plating or to model them as beam elements superimposed along the deck plating. Both of these methods were disregarded using the assumption that the stiffeners do not affect the shear-flow appreciably. Load forces were applied to the models in order to create a nominal stress field environment. It is apparent therefore that disregarding the stiffeners will not affect the overall stress levels in the plating. If the stiffeners had been modeled, loads would have had to be applied to the stiffeners until the average nominal stress level in them matched the plating stress. With this in mind, it was decided to model the deck as plate elements and to apply loads at the ends of the overall plate.

The deck is a representative deck that has a beam of 61 feet. The frame spacing is 8 feet, and the longitudinal spacing is 620mm (24.41in). The plating thickness is 0.8125 in order to have a plate-buckling factor that is less than or equal to 1.25 in accordance with the navy's DDS 100-4 - Strength of Structural Members. This plate-buckling factor is desired as it determines the ratio of allowable compressive design stress to yield strength of the material. If the buckling factor is kept less than 1.25, than the ratio of design stress to yield stress is constant at 1. If the buckling factor had been allowed to be greater than 1.25, than the compressive yield strength of the material would have been reduced by some factor. Having the ratio above be equal to 1 means that compressive yielding can be assumed to occur at the same stress level at which buckling would occur. By designing the plating minimum thickness to this value for the set stiffener spacing, buckling therefore does not need to be considered for any actual panels in the deck. This frees the designer/FEA modeler to focus on SCF's (Stress Concentration Factors) and also to fully utilize the yield stress level of the plating.

The stress that was applied to the deck (10.5 Tsi, 23.5 Ksi) corresponds to the design primary stress for high strength steel towards the end of the vessel's service life. In order to load the deck to achieve this stress, forces were applied to the right edge or far-field areas of the models. Typically, the total force to be applied was obtained by multiplying the total cross-sectional area of plate (this is equal to the product of the deck width 732.3 inches and the nominal plate thickness 0.8125 inches) by the desired nominal stress level. This total force was then divided by the number of plate elements along the edge of the model which will be the same as n-1, where n is the number of nodes along the edge. A resultant smaller force equal to this value was then applied to each node along the right edge of the model except for the nodes at Y=366.

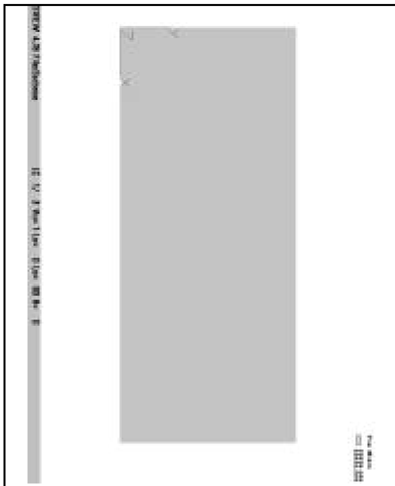
The first modeling attempts were fraught with problems. The initial choices for boundary conditions were to constrain all nodes on the left edge (X=0) of the deck from translating in both the “X” and “Y” directions. The nodes of the other end of the plate were loaded with forces. This combination of loading and constraints resulted in a stress gradient from one edge to the other and an asymmetrical stress pattern about the transverse section at the longitudinal midpoint of the plate. This was not the desired result of applying a uniform line load across one edge of the plate. Two causes for this anomaly were found.

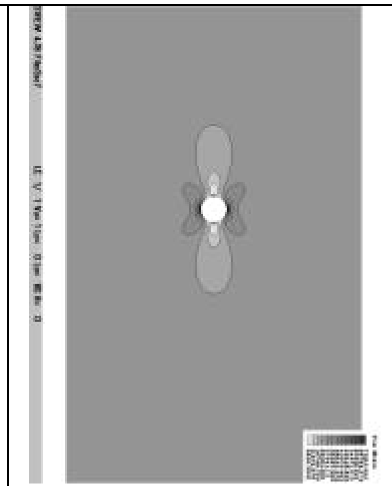
The first problem was that the boundary conditions did not allow for the Poisson effect. By having an entire edge of the plate constrained from translation in the “Y” direction, the area near that end of the plate was not allowed to contract transversely as it was stretched longitudinally. This effect results in “artificial stresses” near the edge, which are in reality relieved through transverse strain resulting from the Poisson effect. This error would produce a non-uniform stress pattern with the highest stresses typically in way of the “Y” direction constraints. This error was corrected by changing the left edge constraints of the plate. Every node except for the transversely centralized one was constrained from translation in the “X” direction, but not the “Y”. This allows the plate to freely contract transversely when it is stretched longitudinally (Poisson effect). The center node (X=0”, Y=366.15”) was constrained in both the “X” and “Y” directions to properly fix the model in space and assure better symmetry about the longitudinal centerline.

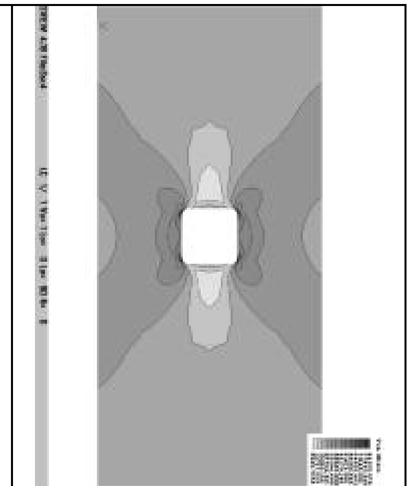
The second problem was that the elements were modeled as plate elements. In the finite element software, sections of plating could be modeled either as plate elements or membrane elements. The distinction between the two is that plate elements incorporate a bending stiffness and can be stressed both through inplane tension/compression and also through bending. Membrane elements only incorporate stresses due to inplane tension and compression. By using plate elements, the elements therefore can exhibit tension stresses on one side of the plate and compression stresses on the other side. Since we are idealizing the plate as only being subjected to inplane loads, membrane elements are the proper choice. All FEA programs exhibit some relatively small errors in calculations due to rounding and truncation of decimals. The bending stresses allowable with plate elements were shown to cause a change of less than 1% in the overall maximum stress for some preliminary models, which is relatively small but avoidable. Once the elements were modeled as membrane elements and the above error was corrected, the plate exhibited a uniform stress gradient throughout.

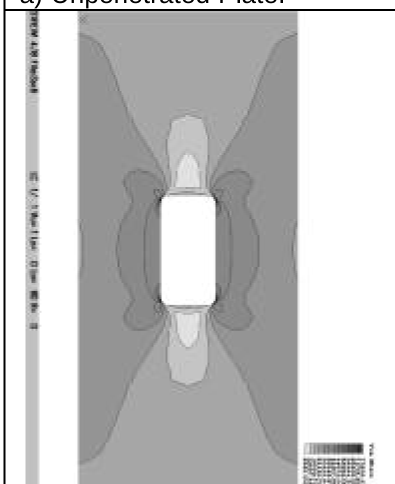
Other problems discovered and accounted for during the modeling process included the use of triangular elements as well as quadrilateral elements with inappropriate aspect ratios. Using either of these categories of elements can produce significant errors in stress distribution and maximum stress levels. In order to avoid this, the finite element meshes were created to use quadrilateral elements with aspect ratios less than 4:1 whenever possible. There were areas where triangular elements had to be used in order to assure continuity in the meshes, but these elements were kept as small as possible in size and number in to alleviate any errors or uncertainties they may cause. Rounded corners in openings were converted into 8 or 16 line segments for every 90 degrees of rounding where possible. Circular openings were converted into either 32 or 64 line segments when possible. It is important in modeling to have at least 8 elements per 90 degrees of curvature to avoid errors and adequately display the stress levels around corners and circular openings. The use of 16 elements can give more precise answers for stress levels, but generally requires more powerful computers to complete the modeling in adequate time. The same can be said for the use of a small or large mesh in the areas far from openings. A larger mesh will typically show an adequate stress distribution in this area (since the stress field is usually uniform transversely far from openings), and a smaller mesh can be used provided that sufficient computing power is available. The use of smaller elements also allows for more varied insert and compensation plate sizes to be used in later models. Typically, the ideal quadrilateral element dimensions in an area distant from an opening would be equal to stiffener spacing in both directions. One should try to at least achieve a rectangular element with the transverse dimension equal to the longitudinal stiffener spacing and the longitudinal dimension equal to some fraction of the frame spacing. This was the case for the models created in this project.

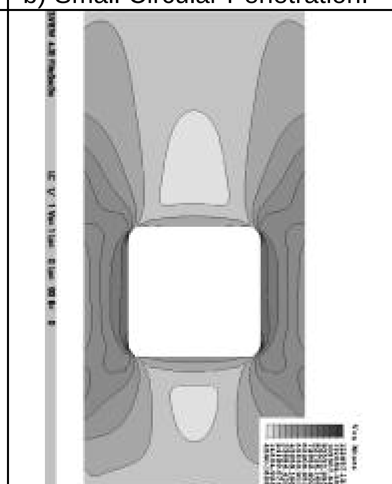
As with any FEA model, the mesh of the model becomes finer near expected stress concentrations. This is done to accurately map the complex stress patterns that takes place in way of such a discontinuity. A finer mesh also yields more accurate results, and is thus used around points of interest. Since the stress patterns of the far edges of the plate are not of interest, a large mesh is used. The large mesh portrays the proper stiffness of the plate so that the in-plane forces are distributed properly. Use of a large mesh in these areas does not affect the accuracy of the stress patterns near the fine mesh.

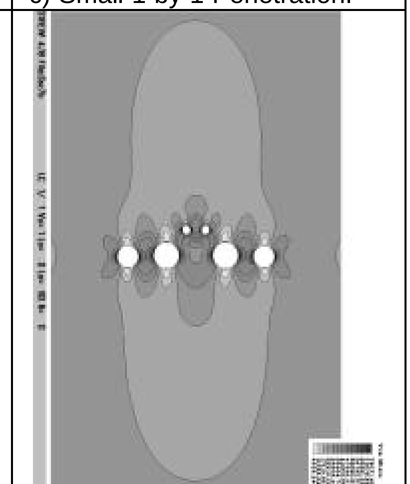

a) Unpenetrated Plate.

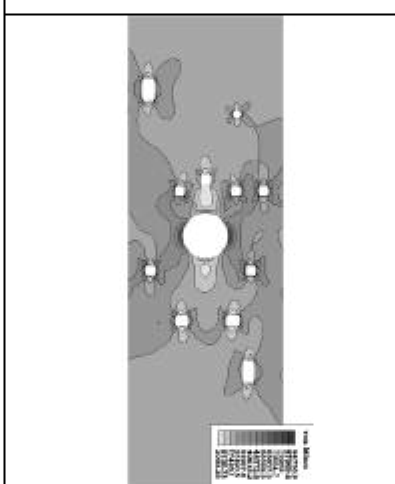

b) Small Circular Penetration.

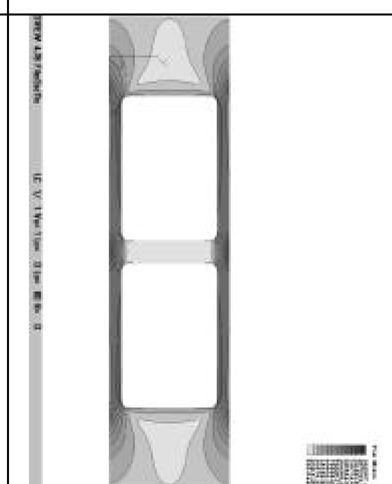

c) Small 1 by 1 Penetration.


d) Small 1 by 2 Penetration.


e) Large 1 by 1 Penetration.


f) Zipper Effect with Small Circular Penetrations.


g) Array of Small Penetrations and Large Circular One.


h) Two Large Sequential Adjacent Hatches.

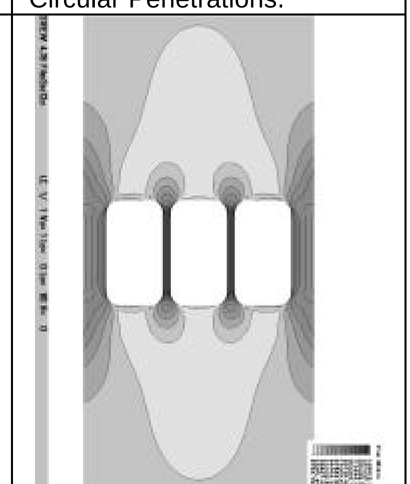

i) Three Large Side-by-Side Adjacent Hatches.

Figure 27: Some representative Finite Element Models.

Figure 27 above is a collection of various finite element models created to aid in the analysis of ship penetrations and reinforcement/compensations methods. This collection represents a characterization of holes and penetrations that are representative of a variety of ship types, including: Bulk Carriers; Container Ships; General Cargo Ships; Roll On/Roll Off Ships; Tanker Ships; & Naval Combatants. Figure 27 a) above (model SSC2) is the baseline model created to determine the basic modeling guidelines presented above. This model has loads applied to one end and boundary conditions on the other. The desired nominal stress level in the plate was 23.5 Ksi. As can be seen in a), the finished model exhibits a uniform stress of approximately 23.5 Ksi (10.5 Tsi) throughout. This model and its loading and boundary conditions were used as guidelines in creating the subsequent models b) through i) contained in Figure 27. A collection of results for all models contained in Figure 27 above are included in APPENDIX B. All referenced stress concentration factors can be found in APPENDIX B as well as views of the models and their results. The model filenames included below (ex. SSC2) will indicate the section of APPENDIX B containing the appropriate model discussed. This collection of results includes magnified pictures of the stress gradients in the models showing values of the basic stress concentration factor calculated at various points in the models. Also included are the values for the maximum basic stress concentration factor (denoted as BSCF) and maximum local stress concentration factor (denoted as FWSCF). The locations of the maximum basic and local stress concentration factors can generally be taken from these magnified images.

Figure 27 b) (model SSC7) demonstrates the results of a finite element model that was created of a deck with a small circular centrally located penetrations. The results of this model indicated a maximum basic stress concentration factor k of 3.03. This value compares well with the results of Figure 9 that indicate a stress concentration of 3.00 for a hole in an infinite plate. The maximum local stress concentration factor K is equal to 2.76. This value was computed by comparing the maximum stress to the theoretical average stress found outboard of the hole. This theoretical stress for an uncompensated, unreinforced plate (a plate that has uniform thickness) is equal to the nominal stress multiplied by the nominal cross-sectional area divided by the reduced cross-sectional area in way of the penetration. These same definitions for basic and local stress concentration factors will be used in describing the models in Figure 27 c) through i). In this model, the diameter of the central hole is approx. 64.6” and the beam of the deck is the same as in all models at 732.3”. Figure 11 can then be used to calculate the theoretical local stress concentration factor for a circular hole in a finite plate. The figure shows a value of approx. 2.73. Thus the finite element model value of 2.76 agrees within 1% with the aforementioned theories. In terms of stress flow, this model represents one of the simplest patterns. Two apparent shadow zones are displayed as areas of low stress fore and aft of the penetration. These zones appear to extend to distances of approximately 4r fore and aft from the center of the circular penetration, where r is the radius of the circle. The size and shape of these zones seems to further enforce the notion of shadow zones described in 3: CHARACTERIZATION OF . These zones are considered ineffective to the longitudinal strength of the vessel. The maximum stresses are located at the furthest outboards of the penetration, which agrees with Figure 10. The stress contours shown could be used to aid in placement of further penetrations in the plate. Also, it is interesting to note that the stress outboard of the penetration is nearly uniform transversely.

Figure 27 c) (model SSC4) demonstrates the results of a finite element model that was created of a deck with a small 1 by 1 centrally located penetration. The results of this model indicated a maximum basic stress concentration factor k of 3.60. The maximum local stress concentration factor K is 2.70. Both stress concentration factors are maximum on the outboard edges of the penetration along its corners. The radius of the fillet on the central hole is 22.9" and the width and length of the hole are both 183.1", so the radius to width factor r/b is .125. Using Figure 13 for a square hole with filleted corners in a finite plate, this would result in a theoretical local stress concentration factor of approximately 2.78. Thus the model value of 2.70 agrees within 3% with the aforementioned theories. The stress contours for this model are very different than those for the above model with a small circular penetration. The contours outboard of the penetration in this model show that stress seems to increase as the distance from the centerline of the deck increases. At a distance away from the centerline, the stresses then begin to decrease until the edge of the deck is reached. Areas of lower stress resembling apparent shadow zones are present fore and aft of the penetration. This correlates with theory. The shape of these regions seems more rectangular than triangular in shape, although this could be an artifact of using a stepped value for stress contour levels, rather than a smooth transition or the meshing technique in the finite element program.

Figure 27 d) (model SSC6) demonstrates the results of a finite element model that was created for a deck with a small 1 by 2 centrally located penetration. The results of this model indicated a maximum basic stress concentration factor k of 3.33. The maximum local stress concentration factor K is 2.50. Both stress concentration factors are maximum on the outboard edges of the penetration along its corners. Comparing these results to the ones for Figure 27 c) seems to indicate that for two centrally located penetrations of equal width and corner radius, the longer penetration longitudinally will have the lower basic and local stress concentration factors. In this case it would seem that the larger a/b aspect ratio of allows stresses to better distribute in way of the corners of the penetrations, thus reducing stress concentrations. The stress contours in this model are similar to those for the small 1 by 1 penetration. The notable exception to this is in the regions outboard of the penetration along its edges. The stress contours along these edges seem to better distribute the concentrating effect caused at the model corners. The stress contour shapes tend to extend farther outboard and are longer longitudinally due to the shape of the hole. This wider area of similar stress usually serves to alleviate local stresses. The areas of lower stress found fore and aft of the 1 by 2 penetration are very similar in size and shape to those of the 1 by 1 penetration. The model exhibits the same decreasing stress pattern at the extreme outboard edges of the deck in way of the penetration. Generally, the stress contours in this model show a much more uniform stress distribution than that in the model for the 1 by 1 small penetration.

Figure 27 e) (model SSC5a) demonstrates the results of a finite element model that was created for a deck with a large 1 by 1 penetration located along the centerline of the deck. The results of this model indicated a maximum basic stress concentration factor k of 5.19. To reduce the stress to acceptable levels, it is logical to increase the thickness of the plating. The maximum local stress concentration factor K is 2.07. Both stress concentration factors are maximum on the outboard edges of the penetration along its corners. As stated previously, a generally accepted practice is to use at least 1/20 as a ratio for r/b on very large hatches and penetrations. The ratio r/b is actually less for very large penetrations than the .125 to .25 commonly used for small penetrations. In this model, an arbitrary value of .1 was used for r/b. The maximum local stress concentration factor in this model is approximately 25.5% lower than the theoretical answer from that of the small 1:1 penetration model above. This seems to illustrate that larger openings in fact do require smaller ratios for r/b than small openings while achieving the same or a lower local stress concentration factor. This is true since it has been earlier shown in Figure 13 that local stress concentration factor values decrease as r/b increases in the r/b range below 0.25. The stress contours in this model show large areas of reduced stress fore and aft of the penetration. Due to the fact that the penetration is not spaced centrally in the longitudinal direction, there is some asymmetry to stress fields fore and aft of the penetration. The stress contour shapes are similar to those from the previous 1:1 and 1:2 opening models. There are more contours present in this model than the others are. This is most likely due to the high level of stress outboard of the penetration and relatively low local stress concentration factor. In terms of modeling with penetrations along the deck centerline, a low local stress concentration factor generally results in less stress contours near a stress concentration and more evenly spaced contours outboard in way of the penetration.

Figure 27 f) (model SSC7b) demonstrates the results of a finite element model that was created for a deck with a collection of small circular penetrations arranged to show what is known as the ‘zipper effect’. These holes are arranged in 2 groups. Each of these groups contains holes that are located adjacent to each other along the transverse axis of the deck model. The results of this model indicated a maximum basic stress concentration factor k of 3.36. The maximum local stress concentration factor K is 3.26. The value for local stress concentration factor is higher than typically found for small round openings in plating. This is most likely due to interactions in stress fields between the penetrations in the plate. This phenomenon is precisely what the “zipper effect” model was created to show. Another interesting effect displayed in the results of this model is that the maximum stress does not occur in an area of least cross-sectional area. The same is true of the basic and local stress concentration factors. Due to interactions between stress fields and shadow area effects, areas of highest stress can be formed through the addition of stress concentrations on multiple openings. In models such as this, the stress flow (shown as areas of similar stress levels in grayscale) is very important in determining where the highest stresses occur. It should also be noted that the two smaller penetrations in the model would have much lower stress levels associated with them had they each been inserted fore or aft of one of the larger penetrations. If this were the case, then the small penetrations could have been inserted into a shadow region of the larger penetrations and allowed the stress pattern to flow more smoothly around the smaller ones. A last interesting note on this model is the large quasi-shadow zone that developed in way of the 'zipper' pattern itself. Although stress concentration factors in this region were not as low as commonly found in true shadow regions, the effect of the 'zipper' did serve to lessen stresses in a semi-triangular pattern fore and aft of the array.

Figure 27 g) (model SSC9newa) demonstrates the results of a finite element model that was created of a deck with a collection of small circular, 1:1, 1:2, and 2:1 penetrations. Centrally located on this deck is a large circular penetration. The results of this model indicated a maximum basic stress concentration factor k of 4.03. The maximum local stress concentration factor K is 3.32. It is interesting to note that the maximum basic stress concentration factor is found at approx. X=1007.3" and Y=484.6", but the maximum local stress concentration factor is at approx. X=1347.3" and Y=81.6". This model is a good example of the differences between local and basic stress concentration factors. The maximum local stress concentration factor was found in a region of higher cross-sectional area than where the maximum basic stress concentration factor was found. Additionally, both maximum stress concentration factors were found along the edges of 1:1 openings with the same corner radii. In terms of shape and r/b ratios alone, this generally results in almost identical local stress concentration factors. In this model the effects of stress-flow, shadow areas, and to a smaller effect cross-sectional area have served to greatly vary the value for local stress concentration factor along the boundaries of these openings. This model shows the importance not only of choosing appropriate r/b ratios and geometric shapes for penetrations, but also that location of a penetration in a deck with multiple penetrations can have a great influence on the local stress concentration factor. In order to minimize stresses more effectively, this model could be modified to place holes in an arrangement allowing for a number of advantages. These include more consistent cross-sectional area in way of penetrations, better utilization and control of stress flow and better consciousness of shadow zones. In addition, the holes should be aligned such that their longitudinal length is greater than their transverse width and the penetrations should be placed as close to the centerline of the deck as possible.

Figure 27 h) (model SSC11a) demonstrates the results of a finite element model that was created for a deck with two large longitudinally adjacent hatches. The results of this model indicated a maximum basic stress concentration factor k of 7.75. The maximum local stress concentration factor K is 1.55. The very high maximum basic stress concentration factor is obviously due to the very small amount of cross-sectional area in the deck in way of the holes. The very low value for maximum local stress concentration factor is most likely due to the same reasons as in Figure 27 e). For very large penetrations, the local stress concentration factor is typically smaller for the same r/b ratio as in a small penetration. In this model, r/b is 1/12. This decreasing r/b ratio necessary is likely due to the small amount of outboard cross-sectional area present with very large penetrations. In terms of stress flow, this abrupt change in transverse cross-sectional area at the edge of either hatch serves to prevent distribution of stress. Generally, stress concentrations of any kind exist because stress is allowed to nonuniformly spread around a penetration. Areas of stress higher than nominal and areas of stress lower than nominal are both generally present near penetrations. This fact can be proven with a simple free body diagram of the plating in question. When there are restrictions in the stress flow, these high and low values are generally much closer to the nominal stress value than they would be in an infinite plate.

Figure 27 i) (model SSC12a) demonstrates the results of a finite element model that was created for a deck with three large transversely adjacent hatches. The results of this model indicated a maximum basic stress concentration factor k of 6.32. The maximum local stress concentration factor K is 1.66. The high maximum basic stress concentration factor in this model is obviously due to the small cross-sectional area in way of the three openings. This model also has a relatively low local stress concentration factor as it is a large opening and requires a small r/b ratio to achieve the same local stress concentration factor that a smaller penetration with a larger r/b ratio would exhibit. The stress fields show triangular areas of reduced stress fore and aft of the hatches. The strips of material between the hatches show extremely high stresses and the points of maximum basic stress concentration factor can be found here. These areas seem to exhibit such high levels of stress because of the abrupt change in cross-sectional area. There appears to be a buildup of stress going from the reduced stress regions through to these strips fore and aft of the hatches. The internal strips seem to represent a similar restriction of the stress flow as discussed in the model with two large hatches. In this case, the cross-sectional area of the strips between hatches is much smaller than the cross-sectional area outboard of the hatches and thus the stresses in the strips can be expected to be somewhat higher. Stress concentrations can also be seen on the outboard edges of the hatches furthest port and starboard. Outboard of the hatches, the stresses are generally higher than nominal, but much lower than in the strips between hatches. This model exhibits similar stress flow behavior in the region outboard of the hatches as does the model with two hatches.

6: COMPENSATION AND REINFORCEMENT METHODS

After preliminary design is completed, there will be a number of openings in the primary structure of the ship. Assuming the preliminary design guidance was carefully followed the stress concentration factors should all be as minimal as possible before the detail design stage. Now a designer must account for the increased stress concentrations that are incurred from these openings. These concentrations can never be entirely eliminated but can be reduced to acceptable levels such that the hull integrity is not unduly compromised. Techniques such as compensation and or reinforcement will help minimize the effect of stress concentrations and are at the disposal of the designer. To what extent these methods will be required is highly dependent on the stress levels in the primary structural members under consideration and also on the value of the stress concentration factor. The latter is the reason it is so important for designer to follow preliminary design guidance as openings that are poorly designed at the preliminary stage may require compensation and or reinforcement to such extents that manufacturing the opening becomes impractical. Compensation and reinforcement will be described in detail in this section.

The first method for dealing with the problem of an opening is known as compensation. This is typically applied to major openings in primary structure. These openings are often the ones that are predominant in preliminary design. This results from the fact that the necessity of these openings for the function of the vessel is realized quite early in the design spiral. Compensation involves specification of thicker plate in the area of the opening to make up for material lost due to the location of the opening. The plating adjacent to the hole is increased in thickness to provide a cross sectional area equivalent to the area of the plating removed as seen in Figure 28.

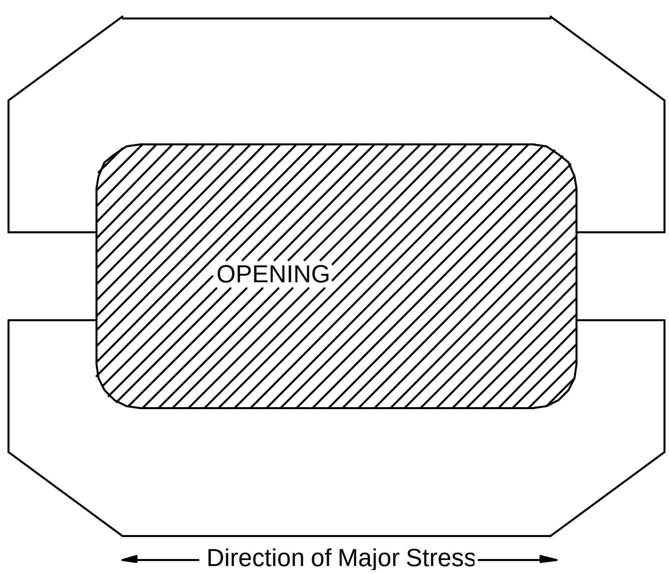


Figure 28: Typical Compensating Insert of a Larger Hole.

The stress concentrations can be further mitigated by a further increase in the thickness of the plate and/or the shaping of the edge of the hole to reduce the effects of the stress concentrations. Plating thickness forward and aft of the hole are graduated to preclude stress raisers that may occur due to discontinuity in the thickness of the plate. Thickness gradations should be no more than ¼ inch. As an alternative, plating may be scarfed at the boundaries to effect a change in thickness and avoid discontinuities. Such techniques are typically applied to the plate straking. Strakes of different thickness are chosen rather than attempting to vary the thickness across an individual strake. This makes fabrication much easier and is shown in Figure 29 below.

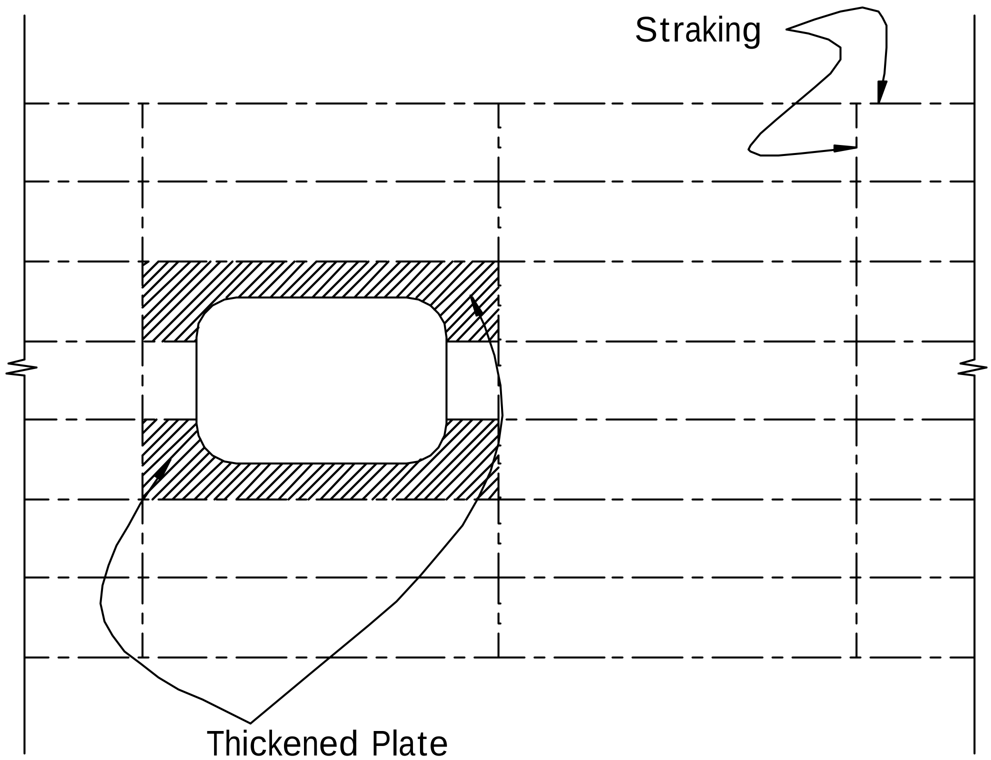


Figure 29: Compensation Plate Thickening and Plate Straking.

It is critically important to ensure that stress concentrations are not created by the choice of materials in the compensation effort. This can occur if the thickness of the strake plate chosen is drastically different than the thickness of the strakes around it. These changes in thickness should be gradual. If at all possible, the designer should dictate that the thickness of the plate vary a maximum of one eigth of an inch (3 millimeters) per plate. The corners of the larger plate should also be chamfered or filleted.

Reinforcement is the other method of further reducing local and far-field stress concentration factors present around openings in primary structure.

Reinforcement is typically applied to minor or medium sized openings. The necessity of such openings is not always realized in preliminary design, so reinforcement is not usually considered until the detailed design stages. This technique generally involves modification to an opening in the primary structure after the opening has been cut. In other words, no modification is made to the primary structural scantlings (strakes) that will be penetrated with the exception of making the opening. Reinforcement is then applied locally around the opening boundaries. Figure 30 is one example of reinforcement techniques.

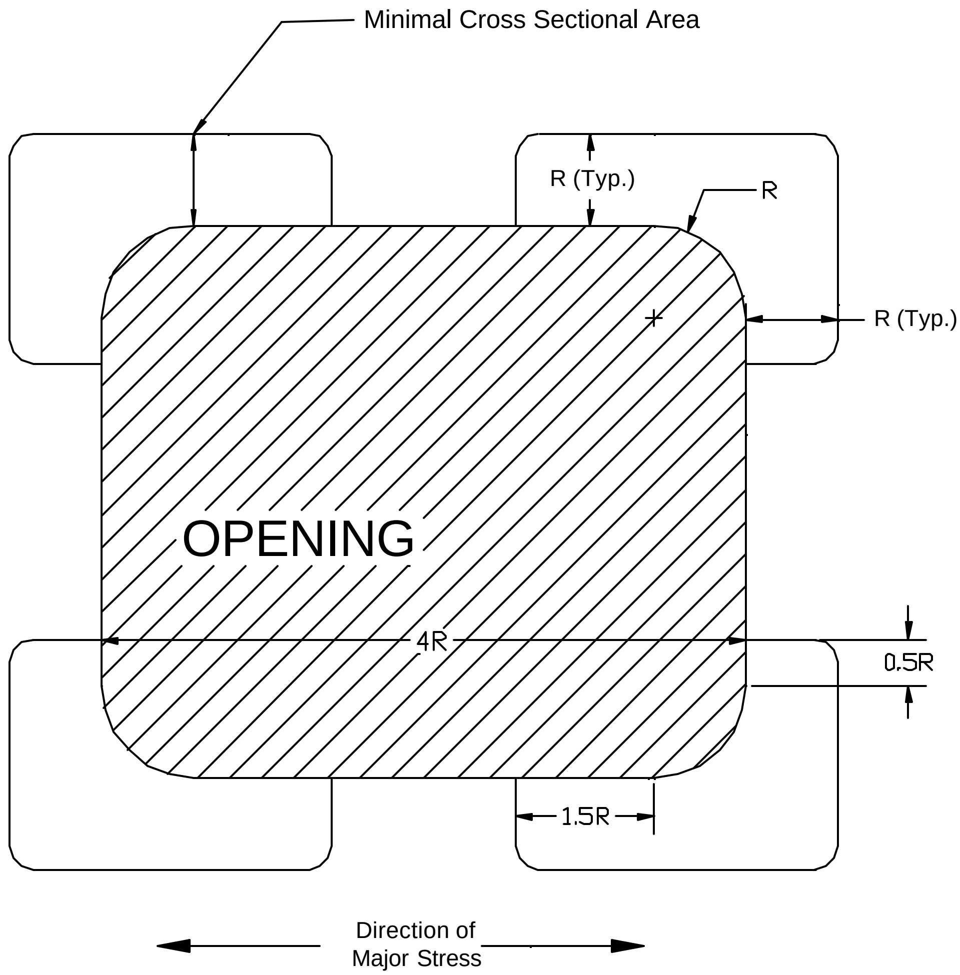


Figure 30: A Large Opening that has been Reinforced at the Corners.

Figure 30 depicts a square opening in primary structure that has been reinforced. As described in 3: CHARACTERIZATION OF , the maximum stress concentration occurs around the corners of the opening. The opening has been reinforced accordingly in way of these corners. The reinforcement depicted in Figure 30 can be welded to the primary structure in way of the corners after the opening is cut. A more advantageous way of accomplishing this is to cut the opening so that the thicker reinforcements can be inserted into the opening and welded at their boundaries. This type of reinforcement is typically applied to major openings [4] where compensation did not reduce the stress concentration factor adequately. For minor and medium sized openings, the more common way of reinforcing the opening is to weld a flat bar around the opening or a reinforcing ring to the top and or bottom of the opening as shown in Figure 31 [3].

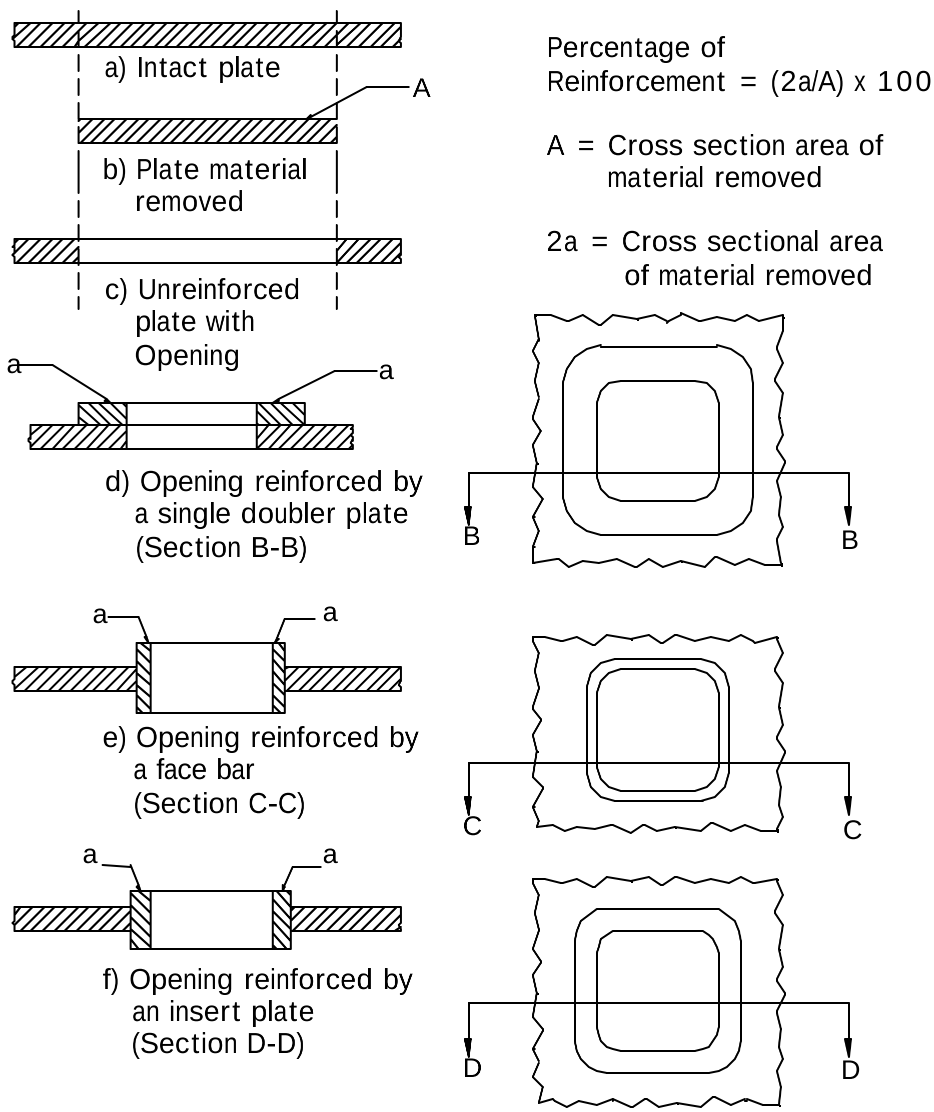


Figure 31: Typical Types of Reinforcements.

Figure 31 illustrates how stress concentrations that are caused by minor and medium sized openings can often be partially alleviated by replacing the material that was removed. Careful consideration of how to do this will allow the designer to gain an equivalent stiffness in the material such that the primary structure will respond to loads as though the opening were not present. Despite this, the local mismatch between stiffness of the reinforcement and the plate constituting the primary structural element will cause stress concentrations. The goal is to ensure that these stress concentrations are substantially lower than the stress concentrations present before reinforcement. Another problem occurs when the depth of the reinforcement is large compared with the surrounding plate. This is often the case for reinforcements in the form of flat bars welded around the boundary of the hole. If the flat bar is too deep, the material farthest from the surrounding plate will not be effective in alleviating stress.

Using the formula for percentage reinforcement from Figure 31, it is theoretically possible to reinforce a hole to 100% of the cross sectional area by the methods described. It has been determined through experimentation, however that the optimum amount of reinforcement for the typical methods described in Figure 31 can be considerably less than 100%. These values are presented in Figure 32.

Type of Reinforcement	Optimum Percentage of Reinforcement
Single Doubler Plate, Figure 31 (d)	90% - 100%
Face Bar, Figure 31 (e)	34% - 40%
Insert Plate, Figure 31 (f)	30% - 60%

Figure 32: Optimum Percentage of Reinforcement.

The method of reinforcement depends on several factors. Amongst these factors are the type of construction used, degree of workmanship required, cost and weight. Figure 32 illustrates the relative effectiveness of the different types of reinforcement. To achieve adequate reinforcement using a single doubler plate will require 90 – 100% of the sectional area of the removed material. The other two methods require substantially less material. The single doubler plate method is usually only employed where the cost of material and labor exceeds the need for weight savings.

Given these two methods for lowering the stress concentration factors from openings, it is of primary interest to any designer to determine how much the stress consentration factors have been lowered after application of the methods. There are limited theoretical means of determining this. The best means of determining the improvement is to perform a finite element analysis for the reinforced and or compensated opening. Despite this, a theoretical approach will be presented here for rectangular openings with rounded corners in an infinite plate [5]. This is especially applicable to minor and medium sized openings.

6.1: A Theoretical Approach

If we assume the parameterized dimensions given in Figure 8 with w approaching infinity, then we can determine the uncompensated stress concentration factor from Figure 15. This stress concentration factor can be referred to as either the local or far-field stress concentration factor as these are the same for an infinite plate. To prevent confusion, it will simply be referred to as K. Given K, we will now assume that some kind of reinforcement and or compensation is applied to the opening under consideration in order to reduce K to some acceptable value. Acceptable values for K are presented in Figure 33 below.

FATIGUE DETAIL CATEGORY (Note 5)	MAXIMUM PERMISSABLE SCF (K_{all}) (Notes 3 and 4)		TYPICAL APPLICATION (Note 5)
	FOR THE HULL GIRDER ENVELOPE (Note 2)	FOR INTERIOR STRUCTURE AT THE NEUTRAL AXIS	
A	2.87	5.74	A machine ground flame cut edge with ANSI smoothness of 25 μm or less, totally isolated from welded attachments, butt welds and other details.
B	2.24	4.48	A longitudinal fillet weld , such as where an opening is reinforced with a ring and has a longitudinal fillet weld at the area of peak stress.
C	1.60	3.20	An unreinforced opening or an opening with an insert plate that has a flame cut edge at the location of the peak stress. Chocks or vertical stiffeners, shorter than 50 mm, attached to the deck by a fillet weld. Full penetration butt welds, such as formed when reinforcing rings are fabricated from several pieces of flat bar and are butt welded with full penetration welds to form a ring.
D	1.27	2.54	Non-load carrying attachment from 50 mm to 100 mm long.
E	1.00	2.00	Non-load carrying attachment longer than 100 mm and < 25 mm thick, load carrying attachment < 25 mm thick.

NOTES:

1. The ranking of details for reinforcement assumes fabrication performed as provided in Sect. 074.
2. The hull girder envelope consists of the shell, innerbottom, double bottom plate longitudinals, and uppermost Strength Deck as defined in Sect. 100.
3. The maximum permissible SCF shall be interpolated linearly for interior structure between those values provided for the extreme fiber of the hull girder and the neutral axis.
4. A reduction in primary stresses may be considered for openings fore and aft of the middle 3/5 length. The reduction would result in an increase in K_{all} by the ratio of the maximum permissible stress range (267 N/mm^2) to the calculated stress range at the extreme fiber using the wave induced plus whipping components (only) of the Design Seaway Bending Moment.

Figure 33: Maximum Allowable K for Fatigue.

The maximum allowable K will heretofore be referred to as K_A. The reduction in the stress concentration factor can be computed from the following equation:

$$K_R = K_b \cdot ß \cdot ? \tag{1}$$

where K_b is the finite width stress concentration factor, β is the coaming factor, and ξ is the insert factor. K_b is found from the following equation:

$$K_b = a \cdot \left(1 + \frac{b}{S_1}\right)^{\frac{1}{2}} \frac{\left(S_2 + 0.5 \cdot b\right)}{\left(S_2 - S_1 + \left(S_1^{\ 2} + S_1 \cdot b\right)^{\frac{1}{2}}\right)} \tag{2}$$

where α is found from the following equation:

$$a = K \frac{2 \cdot S_1}{\left(2 \cdot S_1 + b\right)} + \frac{b}{\left(2 \cdot S_1 + b\right)} \tag{3}$$

In (2) and (3), S_1 is the smallest transverse distance between the structural boundary and the edge of the opening in millimeters. S2 is the largest transverse distance from the structural boundary to the edge of the opening in millimeters. Finally, b is previously defined as the transverse dimension of the opening and is in millimeters.

The remaining two factors in (1) constitute the reduction in K_b resulting from reinforcement and or compensation. If reinforcement is applied in the form of a ring, the factor β (coaming factor) will change and cause a reduction in K_b from the reinforcement. If insert plates are used to compensate for the material lost in the opening, the factor ξ will change and also cause a reduction in K_b from the compensation. Each of these will be discussed individually.

Reinforcement of an opening is the first means by which the stress concentration factors can be reduced. This theoretical approach assumes a reinforcing ring is employed. A number of important considerations must be made as given below.

- The thickness of the reinforcing ring (t_c) must be between the values of 1.0 and 1.2 times the thickness of the parent plating (t_p) or the insert plating (t_i), where applicable.

- The depth of a symmetrical reinforcing ring (h) must not be less than 8.0 $\times$ (t_c). Exceeding this value does not reduce the stress in the member. It is recommended that a multiplier value of 10.0 is specified so that a ring will not develop a stress concentration in the event that the ring sustain nicks in the course of its service life.

- An unsymmetrical reinforcing ring requires the β value obtained from Figure 34 shall be multiplied by 5/4. The total depth of the ring shall meet the same requirements as the standards for the symmetrical ring.

Given the above information, one can use Figure 34 below to determine the coaming factor.

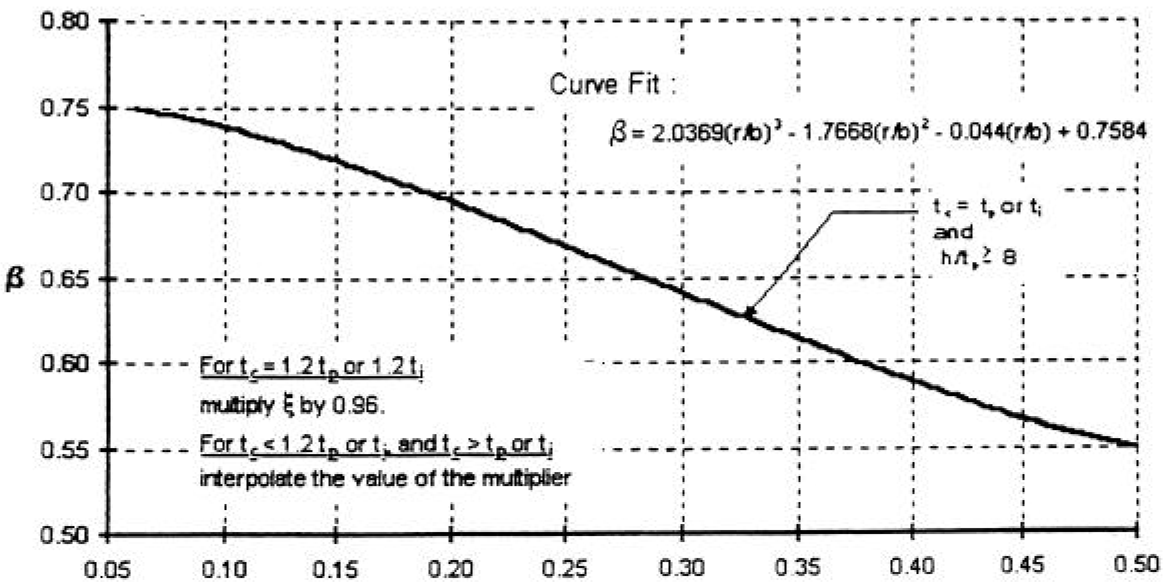


Figure 34: Coaming Factor to Help Determine Reduction in K.

When considering insert plates, any further reduction in K_b can be achieved based on reduction factors (ξ) which are presented in Figure 35. All insert plates should be specified with welding in the areas of minimum stress. This is due to the fact that welds themselves will create residual stresses. This is well documented.

$x = 1 - \frac{(t_i/t_p - 1)}{3}$										
$\frac{t_i}{t_p}$	1.20	1.30	1.40	1.50	1.60	1.70	1.80	1.90	2.00	2.10
ξ	0.933	0.900	0.867	0.833	0.800	0.767	0.733	0.700	0.667	0.633
$\frac{t_i}{t_p}$	2.10	2.20	2.30	2.40	2.50	2.60	2.70	2.80	2.90	3.00
ξ	0.633	0.600	0.567	0.533	0.500	0.467	0.433	0.400	0.367	0.333

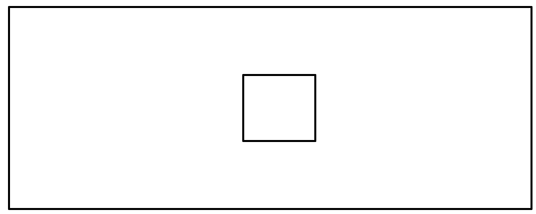
Figure 35: Insert Factor to Determine Reduction in K.

An opening that has a complex geometry or has complex stress fields must meet a separate set of criteria. Analysis by the finite element method is required under these circumstances.

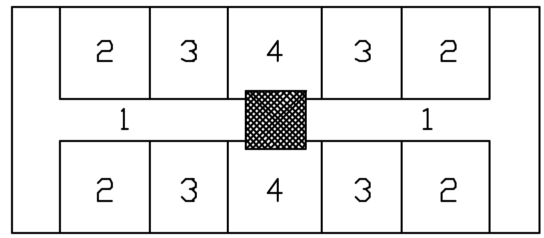
6.2: A Brief Study of Straking and Load Attraction

Nearly every ship is designed in terms of both strength and weight. When one compensates a deck for lost area by thickening plating, the ship's weight will also increase. By looking at the stress results of a finite element model for a penetrated deck, it is apparent that stress patterns outboard of the penetration are not uniform either transversely or longitudinally. Since stress is sensitive to thickness of plating, it therefore would not make sense to utilize a constant thickness for the entire area outboard of the penetration. As a simple guideline, if weight is going to be added to a ship in terms of compensation (or reinforcement for that matter), than the additional weight should be used to lower your maximum stresses as much as possible. Typically, strake plates should be thickened by utilizing the stress flow pattern for the deck. Also, one can alter the stressflow in the deck by thickening plating of the deck.

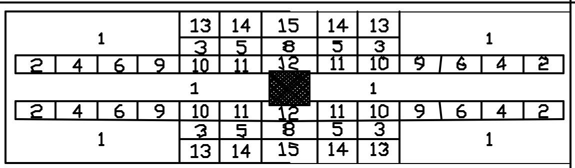
In a cross-section of a deck comprised of different thickness of plating, the system can be compared to a system comprised of springs. Each strake plate can be thought of as having a stiffness in the longitudinal direction of the deck, which is the direction of loading for the above deck models. When a force is applied to a system of spring connected to operate as one spring, the actual force distribution, and therefore stress distribution) is not even amongst the springs. Forces are typically higher in the stiffer springs and lower in the springs with the lower stiffness. In a similar way, plates of different thickness can be thought of as having a different stiffness, even though the material may have the same modulus of elasticity for each plate. If a force is applied to each of two plates of the same material and width, one which has a thicker cross-section, then they both will be subjected to a stress based on the force applied. This stress will then cause a strain in each plate based on their individual stresses. The plate with the thicker section will have a smaller stress since stress is equal to force divided by area. This smaller stress will then produce a smaller strain and displacement than there will be in the other plate. In terms of a spring, stiffness is the relationship between displacement and force applied to the spring. Using this analogy, stiffness for a piece of plate can be though of as a ratio of the strain in the plate to the force applied. For the two plates mentioned, the thicker plate can then be thought of as being 'stiffer' than the thinner plate since the thicker plate exhibits a smaller strain to force ratio. Thus, in a system comprised of the two plates connected transversely to each other, forces in the overall plating tend to be higher section comprised of the thicker plate. Due to this phenomenon, it is actually possible to increase stress in a region of plating by thickening it, since the thicker region may become much 'stiffer' than the plating around it in the cross-section of the deck. This result can actually be utilized to achieve a more uniform stress gradient throughout the deck, even by actually thickening the plate in areas of lower stress rather than the highest stress points.



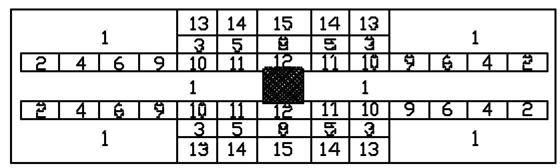
a) SSC4- 1X1 Penetration



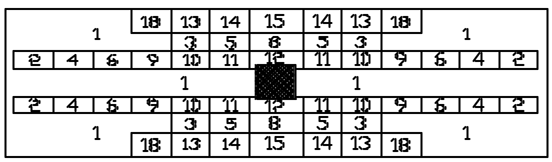
b) SSC4b- 1X1 Area-Compensated Penetration



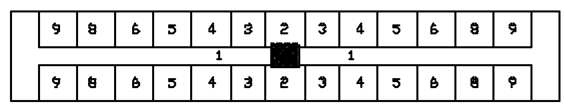
c) SSC4c- 1X1 Plate-Reinforced Penetration, Thicker Stringer Plating



d) SSC4d- 1X1 Plate-Reinforced Penetration, Thicker Stringer Plating



e) SSC4e- 1X1 Plate-Reinforced Penetration, 1.25" Thick Stringer Plating



f) SSC4f- 1X1 Plate-Reinforced Penetration, Transversely Uniform Plate Thickness

Figure 36: Illustrations of Compensation for a 1 by 1 Deck Penetration.

	Thickness in inches for the patterns in FIGURE #					
Thickness Number	a) SSC4	b) SSC4b	c) SSC4c	d) SSC4d	e) SSC4e	f) SSC4f
1	0.8125	0.8125	0.8125	0.8125	0.8125	0.8125
2		0.8330	0.8750	0.8750	0.8750	1.5625
3		0.9580	0.9375	0.9375	0.9375	1.4375
4		1.0830	1.0000	1.0000	1.0000	1.3125
5			1.0625	1.0625	1.0625	1.1875
6			1.1250	1.1250	1.1250	1.0625
8			1.1875	1.1875	1.1875	0.9375
9			1.2500	1.2500	1.2500	0.8125
10			1.3750	1.3750	1.3750	
11			1.5000	1.5000	1.5000	
12			1.6250	1.6250	1.6250	
13			0.8125	0.8750	1.0000	
14			0.8750	1.0000	1.1250	
15			1.0000	1.0830	1.2500	
18					0.8750	

Figure 37: Plate Thickness for 1 by 1 Deck Compensation Techniques.

Figure 36 above shows an overview of the compensation techniques used to create the series of finite element models where the deck is characterized by a small 1X1 penetration centrally located. Figure 37 is a table of the various plate thickness corresponding to the labels in Figure 36. The base deck model is listed as Figure 36 a) above (model SSC4). In this model, the thickness of plating is held constant at .8125" throughout. The stress gradients for this model can be seen in Figure 27 c). The other patterns presented in Figure 36 above demonstrate compensation techniques that would be utilized for the plate with the 1X1 penetration. These variations in thickness presented are typically called straking, since the plate of a deck is composed of smaller sections called strakes. A strake is a section of plate bounded by stiffeners in a ship. When compensating for lost area in way of a penetration, theory would indicate that one could simply thicken the entire area outboard of the penetration. In reality, this area would be composed of smaller strake plates. The size of these plates is governed by availability and manufacturing ability utilized by the builders and designers of the ship. While having to use small strake plates at first may seem like a complication, it actually is a benefit in terms of compensation techniques.

The above illustrations have been created as finite element models. Analyses have been run on these models to determine stresses. The stress was recorded at the midpoint of the outboard side of the penetration. Stresses were then recorded at distances further outboard of this point. Stresses for the same points were recorded for each of the other various compensation illustrations above. The basic stress concentration factor at each point can be computed directly as the stress level divided by the far-field stress of 23.5 Ksi. In a typical design, compensation for lost area would generally be performed before reinforcement. Thus, overall stress levels and basic stress concentration factors are more important in the study of compensation techniques than local stress concentration factors. The local stress concentration factors are generally computed after compensation and reinforcement would be used to reduce them. The results of these models have been included in Figure 38 and will be discussed below. Figure 38 shows large steps in stress levels for model SSC4 near the edge of the hole and a somewhat linear decreasing stress pattern far outboard of the edge. There is a local maximum on the plot at approximately X=530". Additionally, it is apparent that the stress level near the edge is much higher than that far outboard. This demonstrates the typical effect of stress concentration on the stress field.

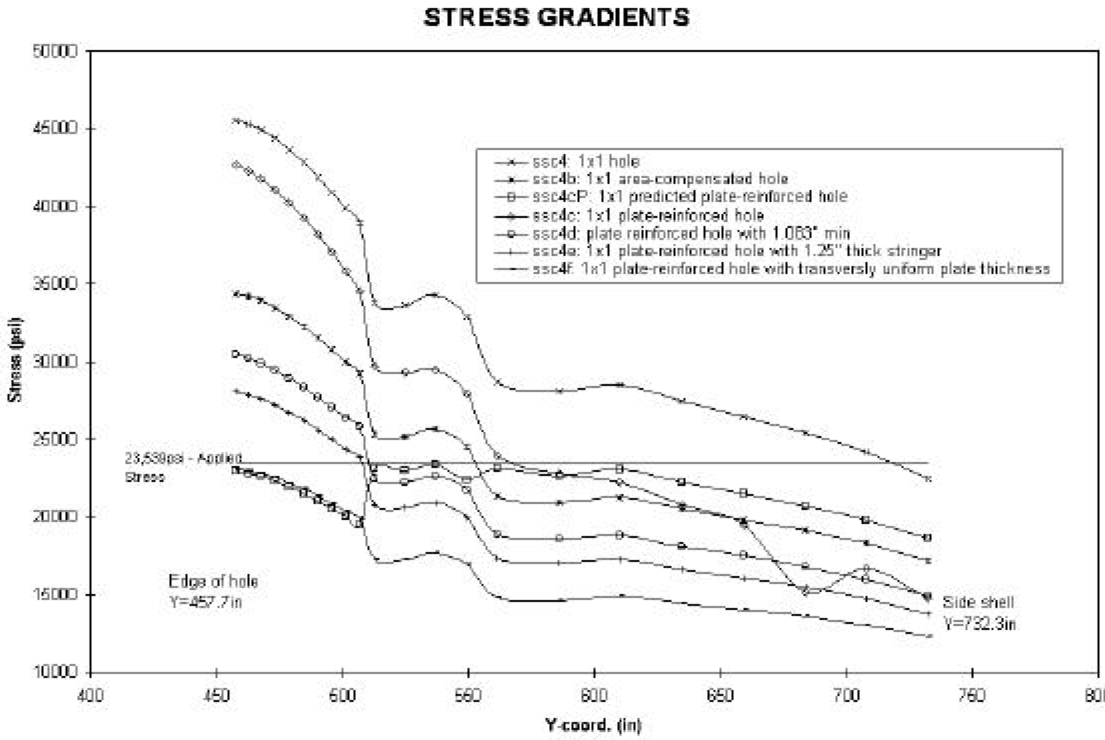


Figure 38: Stress Levels Outboard for Compensation Variations of a 1 by 1.

Figure 36 b) above (model SSC4b) shows a simple compensation for the 1 by 1 penetration. In this illustration, large sections of plate outboard of the penetration are thickened to compensate for the lost area in way of the penetration. The area along the centerline of the deck is typically not thickened, as stresses are generally much lower in this area than they are outboard in way of the hole and fore and aft of the hole. Also, this allows the thickened plate sections to be smaller than the width of the deck, which reduces weight and increases producibility. Thus it makes sense to increase the plate thickness in way of these high stress areas to utilize the increased cross-sectional area. Since plate thickness can only be increased in steps of .125" for every 20 feet of deck longitudinally, it was necessary to include a few different thickness of plate fore and aft of the thickest plate. These stepped thickness plates allow stress to flow into the thickest section of compensated plate while avoiding any stress concentrations that may be caused by having too thick of a plate stepping over an insufficient distance. These larger plates are actually composed of smaller strake plates which have a common thickness. In reality, compensation Plating can only be made by assembling smaller strake plates of the desired thickness into the larger plates shown. Figure 38 shows that stresses for this model throughout the width have been decreased. The largest ratio of reduction though is in the area far outboard of the penetration. Since these stresses were much lower than those at the edge of the hole (and in fact very similar to the far-field stress), this compensation technique was not very efficient in terms of reducing the maximum stress levels with the minimum added plate weight. This method does however accomplish a goal of reducing the maximum stress, but when designing, one must consider both weight and strength.

Figure 36 c) (model SSC4c) demonstrates a more complex straking technique that could be used for compensation. The technique used here can actually accomplish much the same goals are reinforcement methods and is sometimes known as plate-reinforcing. This illustration shows the same cross-sectional area in way of the hole as the technique in b). The difference is that strakes of different thickness were used transversely across the deck. In order to accommodate the rule of increasing plate thickness .125" for every 20 feet longitudinally of deck, strake plates had to be inserted for and aft of the ones in way of the hole. These additional plates are used to step the thickness of plate from the nominal value of .8125" to the thickness in way of the hole listed in Figure 37. This straking/compensation technique basically replaces the area lost due to the penetration. Instead of replacing the area by thickening uniformly across the deck as in b), the area is replaced unevenly with the thickest strakes being located along the outboard edges of the hole. This technique was developed by using the stress gradient of the uncompensated finite element model for the 1X1 penetration.

Since the largest stresses in the uncompensated plate are located on the outboard edges of the penetration, thickening the plates along this edge can provide the same effect as reinforcing the corners of the penetration. Theoretically, stress is usually equal to the force applied divided by the cross-sectional area of a plate. So the method of c) seeks to decrease the stresses along the penetration due to the internal forces in the plate by increasing the area in these regions. Since areas further outboard of the hole exhibit less stress in the uncompensated model, these areas are compensated using thinner strakes than along the edge of the penetration. This thinking has both merit and problems. This methodology corrects the flawed assumption made earlier for compensation that the stress gradient is constant over the transverse distance outboard of the hole.

The methodology now recognizes that the stress is not uniform in the area outboard of the penetration. The theory is flawed though as increasing the plating thickness relative to the stress in the uncompensated model does not in reality achieve a uniform stress gradient outboard of the hole. In Figure 38 , a plot of predicted stress levels assuming that stresses will be affected linearly with thickness of plate is included as SSC4cP. SSC4c in Figure 38 demonstrates the actual results of the finite element model. The results show that stresses near the edge of the penetration are in fact very high when compared to the predicted values. Far outboard of the penetration, the stress levels were actually less than predicted values. These results indicate that stress in the model was indeed attracted to the thick plate at the edge of the penetration. This is a result of internal forces being higher near the edge of the hole than outboard. Much the same as the spring analogy earlier, the internal forces were higher in the stiffer section of plate than in the less stiff outer strakes.

Figure 36 d) (model SSC4d) is very similar to c). In this illustration the strakes are organized in the same fashion as c). The thickness of the furthest outboard strakes in way of the penetration have been changed so as not to be less than the original plate thickness specified in b) for areas outboard of the hole. Theoretically, this should ensure that stresses at any distance outboard of the penetration are the same or less than those from b), since the plating thickness at any point outboard of the penetration should be at least what it was in b). The theory is once again that a thicker plate will have the same internal forces in it and therefore exhibit less stress than a thinner plate with the same internal forces. In reality internal forces change nonlinearly as the thickness of plating changes. This is because a thicker plate has a greater stiffness than a thinner plate, as described above, and therefore attracts more internal load forces than the thinner plate. Figure 38 displays the results of the changes in this technique. When compared to the results of the previous technique, stresses at most are generally lower than before. This is most likely because no strakes have in this technique are thinner than the previous one, but some are actually thicker which increase cross-sectional area. Far outboard, stresses at some points were in fact raised. This demonstrates an attraction of stresses to the thicker strakes far outboard. In addition, the stress field far outboard shows a much smoother and more constant slope than before. The largest percent decrease in stress occurred near the edge of the hole and is very pronounced.

Figure 36 e) (model SSC4e) recognizes the effect of thicker plates attracting load and actually attempts to utilize the fact to achieve a more uniform stressflow and a lower stress at the edge of the penetration. In this illustration, the most outboard strake plates in way of the penetration (corresponding to the plating that would be the upper flange of a box girder/stringer) have been increased to 1.25". The strake plates fore and aft of these have been thickened to accommodate the rule about thickness stepping longitudinally. This rule has in fact required that one additional plate fore and aft of the outer collection of strake plates be thickened from the nominal value of .8125". In this case, it is hoped that the thicker outer plates will attract some greater portion of the load away from the penetration area than the previous thickness plate furthest outboard had attracted. The effects of this technique are displayed in Figure 38. The stresses near the edge have been further reduced through the further thickening of the outboard strakes/stringer. Stresses throughout the distance are less than in the previous technique and in fact appear to be approximately some ratio of the previous stress levels. The thicker outer strakes do not appear to cause any significant stress concentrations outboard as the stress curve is very smooth far outboard. It is also noteworthy that the stress levels throughout the width are lower than the far-field stress in the model

Figure 36 f) (model SSC4f) is an illustration of overcompensating. This technique involves thickening the deck with large sections of uniform thickness plate. These large sections are once again comprised of smaller strake plates of uniform thickness. This technique attempts to both compensate for area lost due to the penetration and also to reinforce the model in way of the hole. The results of this theoretically should be a similar or lower stress at all points outboard of the penetration when compared to Figure 36 a) through e). By examining Figure 38 it is apparent that this is in fact the case. Stress levels near the edge are similar or lower than for any of the other illustrations and additionally the stress gradient has been smoothed along the entire transverse distance outboard of the hole. Thus, the use of the thick plate with uniform thickness serves to both reinforce the edge of the hole and also compensate for lost area. By keeping the thickness constant, the model avoids attracting stress toward the edge of the hole. This is because the entire width of the deck should have similar plate stiffness. As in e), the stresses throughout the width are lower than the far-field stress applied.

The conclusions drawn from these compensation techniques are that thicker plate does generally attract load patterns in a deck. When compensating, one should use this attraction to decrease stress levels near penetrations while at the same time compensating for lost area in way of the penetration. At first, the laod attractions can be a design problem. Recognized and utilized effectively in design, the load attractions can help reduce the overall weight of compensation plate necessary.

Reinforcement inserts behave similarly to compensation strakes in terms of attracting loads. As the length of an insert plate (such as in a ring reinforcement technique) is increased, stiffness of the model in way of the reinforced corner is increased relative to the plating around the corner of the penetration. Stresses in the plate can then actually increase as one attempts to decrease then through reinforcement. In order to show this effect, finite element models of uniform nominal plate thickness .8125" were created as series SSC4g.These models were subjected to the same loading and boundary conditions as the models in Figure 36. Figure 39 is an illustration of the various corner insert lengths used in the models. The first length is computed as the distance along the edge of the penetration from the right edge of the black region to the left edge. All inserts used begin at the right edge of the black region. Each successive shade along the edge of the penetration denotes the extent of an insert used in a model. The results of this study are included in Figure 40.

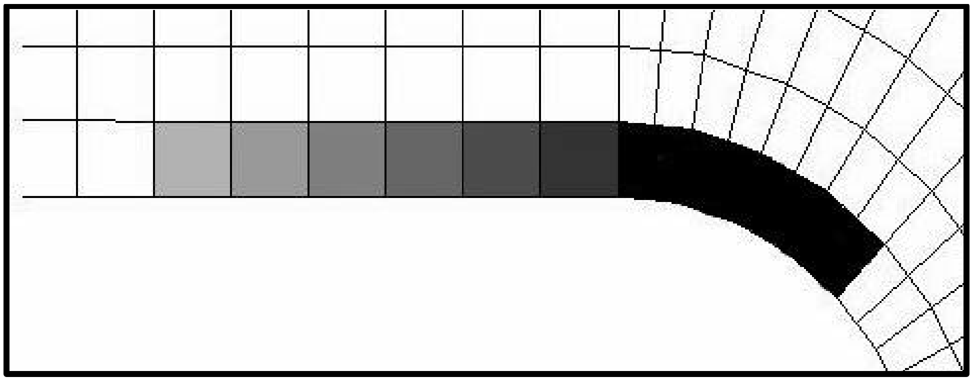


Figure 39: Illustration of Corner Reinforcement Insert Lengths.

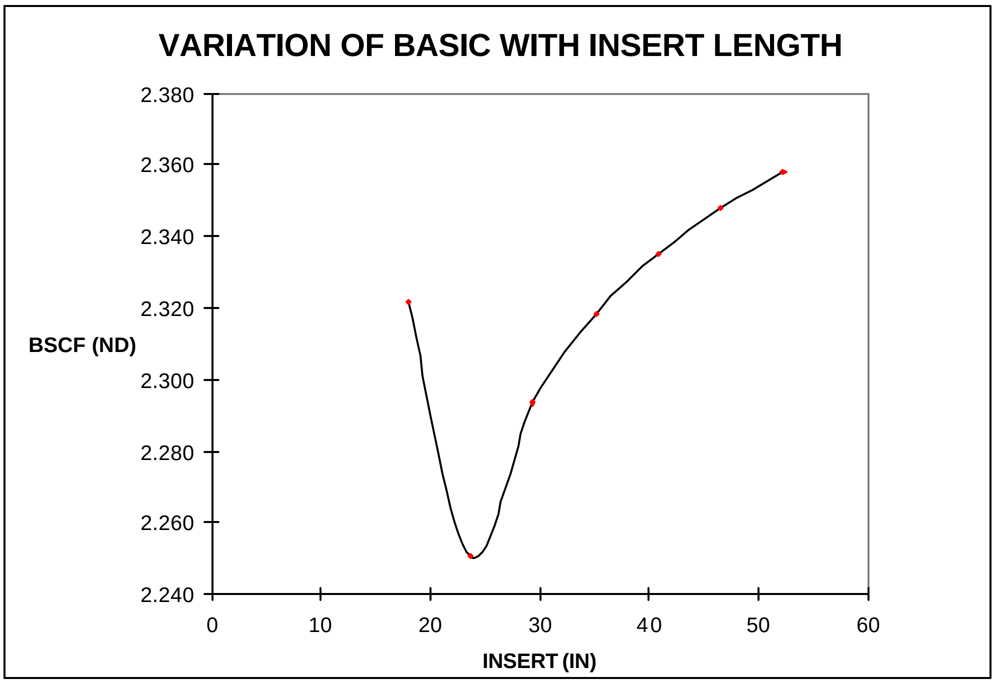


Figure 40: Variation of SCF with Corner Insert Length.

In Figure 40, the values of basic stress concentration factor for various corner insert lengths were plotted and curve-fitted against the lengths of the insert used for a small 1 by 1 penetration. It is clear that there is a minimum value that can be achieved for basic stress concentration factor with the inserts and also that this minimum does not necessarily occur with the longest insert. The minimum value for basic stress concentration factor was achieved with the model having an insert length of 29.407”.

6.3: Analysis of Reinforcement Requirements

This section presents the calculations performed to determine the stress safety factor for each location where the stress level is significant for the model described in 3.1: An example of the finite element method.

The safety factor used here is defined as:

$$SF = \frac{\sigma_{AL}}{\sigma_L}$$

σ_{AL} = allowable local peak stress

σ_L = computed local peak stress

For 700 MPa. yield steel σ_{AL} = (0.5)(700 Mpa)
= 350 Mpa.

For 350 Mpa. yield steel σ_{AL} = (0.6)(350 Mpa)
= 210 Mpa.

Note that the safety factor here is the ratio of allowable stress to computed stress and thus it is a safety factor on top of a safety factor since the allowable stress already contains a safety margin.

A very important point relates to how the 1 Deck stress provided by the hull girder bending analysis is converted into the local peak stresses using the FE results of Figure 22. The methodology and rational is described below.

a) The 100 Mpa. axial load applied at “infinity” in the finite element analysis was an arbitrary number chosen to facilitate the reading of a stress concentration factor directly from the program-plotted output. Thus the FE contour plots indicate only the geometric interaction effect and the resulting stress concentration factor but do not indicate the actual stress level in the deck.

b) Consider a specific corner of an opening. There is a total force that is carried axially by the deck (defined by the hull girder bending moment curve) at the section of the ship coinciding with this specific corner. Thus for the purpose of evaluating local stresses at this ship section, the stress to be applied at infinity will in general be different from that of other sections of the ship. That is, the total axial force carried by the deck varies along the ship length. The hull girder bending moment curve defines this variation. Note that the FE model, as a defined problem, assumes a constant axial force as a function of the longitudinal position.

c) Given the hull girder bending stress in the deck, for a particular section of the ship, one computes the corresponding total axial force in the deck by multiplying the deck effective area by this stress. This total force must be considered to act on the FE model at "infinity" when determining local stresses at this particular reference section.

d) Thus in order to determine the actual peak stress level at a specific corner of an opening one must scale the FE results by a different factor for different longitudinal locations along the deck.

$$\sigma_L = \left(\frac{s_{LFE}}{100}\right)(Error) \times s_{\infty}$$

Where:

σ_L = local peak stress

σ_{LFE} = FE analysis local peak stress

Error = Factor ranging from 1.0 to 1.1 to allow for estimated error in local peak stress.

7: DETAILED DESIGN GUIDANCE

Following the preliminary design stage, it is time for the designer to carefully consider the effect of openings. Their effect should be as minimal as possible if Preliminary Design Guidance was carefully followed. Guidance for detailed design is presented here:

1) Determine if it is a Simple or Complex Opening [5].

    1.1) A simple opening must be isolated from other openings or discontinuities. This means that a minimum distance of at least four times the width of the smaller opening must separate the opening. [5]

    1.2) A simple opening uses standard radiuses to round corners. [5]

    1.3) A simple opening must have a corner radius to opening width ratio (r/b) as defined in 3: CHARACTERIZATION OF OPENINGS that falls between values of 0.10 and 0.50 inclusive. [5]

    1.4) A simple opening must have an a/b ratio as defined in Figure 8 that falls between values of 0.25 and 4.00 inclusive. [5]

    1.5) A simple opening is stressed uniaxially. [5]

    1.6) A simple opening has a basic geometry (circular or rectangular with rounded corners). [5]

    1.7) If requirements 1.1) to 1.6) are not fully met, the opening is to be considered complex.

2) Simple Opening Design Methodology.

    2.1) Simple openings may be analyzed theoretically. Finite element methods can be employed but are not considered necessary. If finite element methods are desired, refer to 4) below for guidance.

    2.2) Determine the Stress Concentration Factor from the opening's geometry. Use Figure 8, Figure 9, Figure 10, Figure 11, Figure 12, Figure 13, Figure 14, and or Figure 15 as necessary. [5]

    2.3) Compare this stress concentration factor with the allowable stress concentration factor from Figure 33 [5]. If the determined value is less than the allowable, no additional design is required. If the determined stress concentration factor is greater than the allowable, reinforcement or compensation of the opening is required.

3) Reinforcement and Compensation of Simple Openings.

3.1) Reinforce the opening with a ring and or compensate the opening with insert plates.

3.2) Application of the design methodology described in 6.1: A Theoretical Approach, equations (1) to (3) should be applied to determine the stress concentration factor as it has been reduced from reinforcement and or compensation of the opening. [5]

3.3) For reinforcement, the reinforcing ring must have a thickness between 1 and 1.2 times that of the parent or insert plating. [5]

3.4) For reinforcement, the depth of a symmetrical reinforcing ring is not to be less than 8 times the thickness of the ring. Any greater depths are not effective in reducing stress. It is recommended that 10 times the thickness of the ring be used to allow for service nicks and gouges. [5]

3.5) For reinforcement, the factor β (coaming factor) as determined from Figure 34 should be multiplied by 1.25 for unsymmetrical reinforcing rings. Depth requirements are as given in 3.4). [5]

3.6) For reinforcement, the information necessary for calculation of the reduced stress concentration factor in 6.1: A Theoretical Approach, equation (1) can be determined from Figure 34. If no reinforcement is applied, the coaming factor (β) should be set equal to 1. [5]

3.7) For compensation with insert plates, welds of the plate for corner inserts or full compensating inserts should be placed in areas of minimum stress. [5]

3.8) For compensation with insert plates, the information necessary for calculation of the reduced stress concentration factor in 6.1: A Theoretical Approach, equation (1) can be determined from Figure 35. If no compensation is applied, the insert factor (ξ) should be set equal to 1. [5]

3.9) Determine reduced stress concentration factor from 6.1: A Theoretical Approach, equation (1). Compare this with the allowable stress concentration factor as determined from Figure 33. [5]

3.10) If the reduced stress concentration factor is less than the allowable, no further design is required. If the reduced stress concentration factor is greater than the allowable, additional reinforcement and or compensation is required. In this case, repeat steps 3.1) to 3.9).

4) Complex Opening Design Methodology.

4.1) Complex openings may not be analyzed theoretically. Finite element methods are required to design complex openings.

4.2) Determine the Stress Concentration Factor from the opening's geometry. Use finite element modeling. Guidance on modeling is presented below.

4.3) For finite element modeling, loads applied to the model are to be derived from the ship design bending moments. For simple models involving a plate with the opening(s), application of axial stress computed from the bending moment and section modulus of the ship to the end of the plate is sufficient. For more complex models that encompass a larger portion of the ship, it may be necessary to actually apply the bending moments to the model. [5]

4.4) For finite element modeling, the model should be created such that it reflects primary structure contributing to stress flow in way of the opening. [5]

4.5) For finite element modeling, the boundary conditions should reflect connections to support structures so as not to disturb the stress patterns in the area of concern. [5]

4.6) For finite element modeling, the mesh should be constructed of a refinement suitable to obtain accurate results and satisfactory precision. [5]

4.7) For finite element modeling, proper element configuration must be maintained in areas where stress values are desired. [5]

4.8) For finite element modeling, quadrilateral elements should have acceptable aspect ratio and shape as prescribed for the FEA tool. [5]

4.9) For finite element modeling, triangular elements should be avoided. [5]

4.10) For finite element modeling, areas of high detail require a fine mesh. [5]

4.11) For finite element modeling, a difference in values of less than 10% is required at adjacent nodes. [5]

4.12) Calculate the far-field stress concentration factor from the FEA model for the unreinforced, uncompensated opening(s).

4.13) Compare the stress concentration factor to the allowable obtained from Figure 33.

4.14) If the calculated value is less than the allowable, further design of the opening is not required.

4.15) If the calculated value is greater than the allowable, reinforcement and or compensation of the opening is required. This reinforcement/compensation must be applied to the FEA model and it must be reanalyzed. Consideration of reinforcement and or compensation strategies prior to initial modeling will allow the designer to produce a mesh conducive to this situation. This will substantially speed the process.

4.16) Repeat finite element analysis on reinforced model until adequate stress concentration factors are obtained.

APPENDIX A: Representative AutoCad Models for Openings

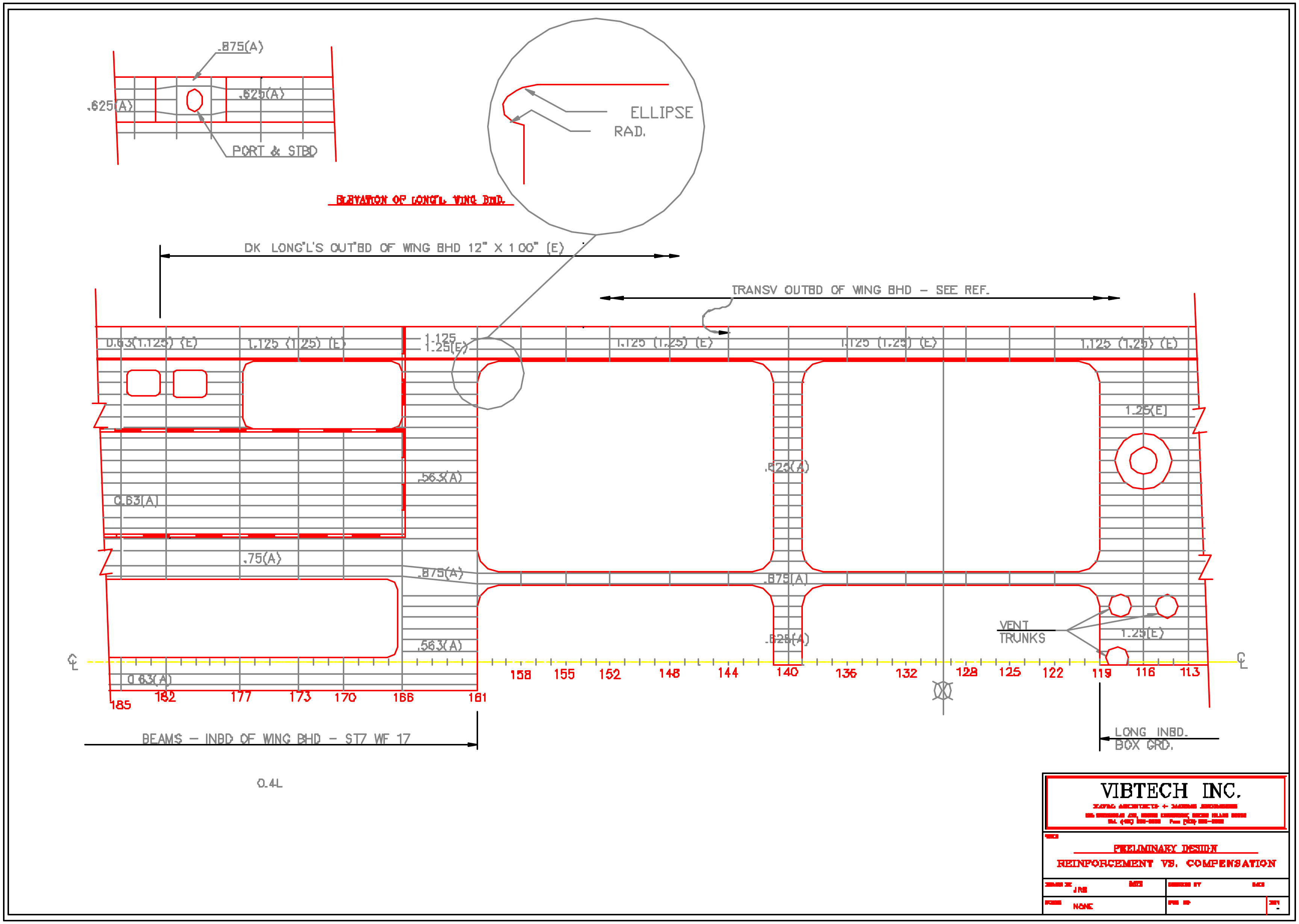


Figure 41: Cargo Ship Hatch Openings in Strength Deck.

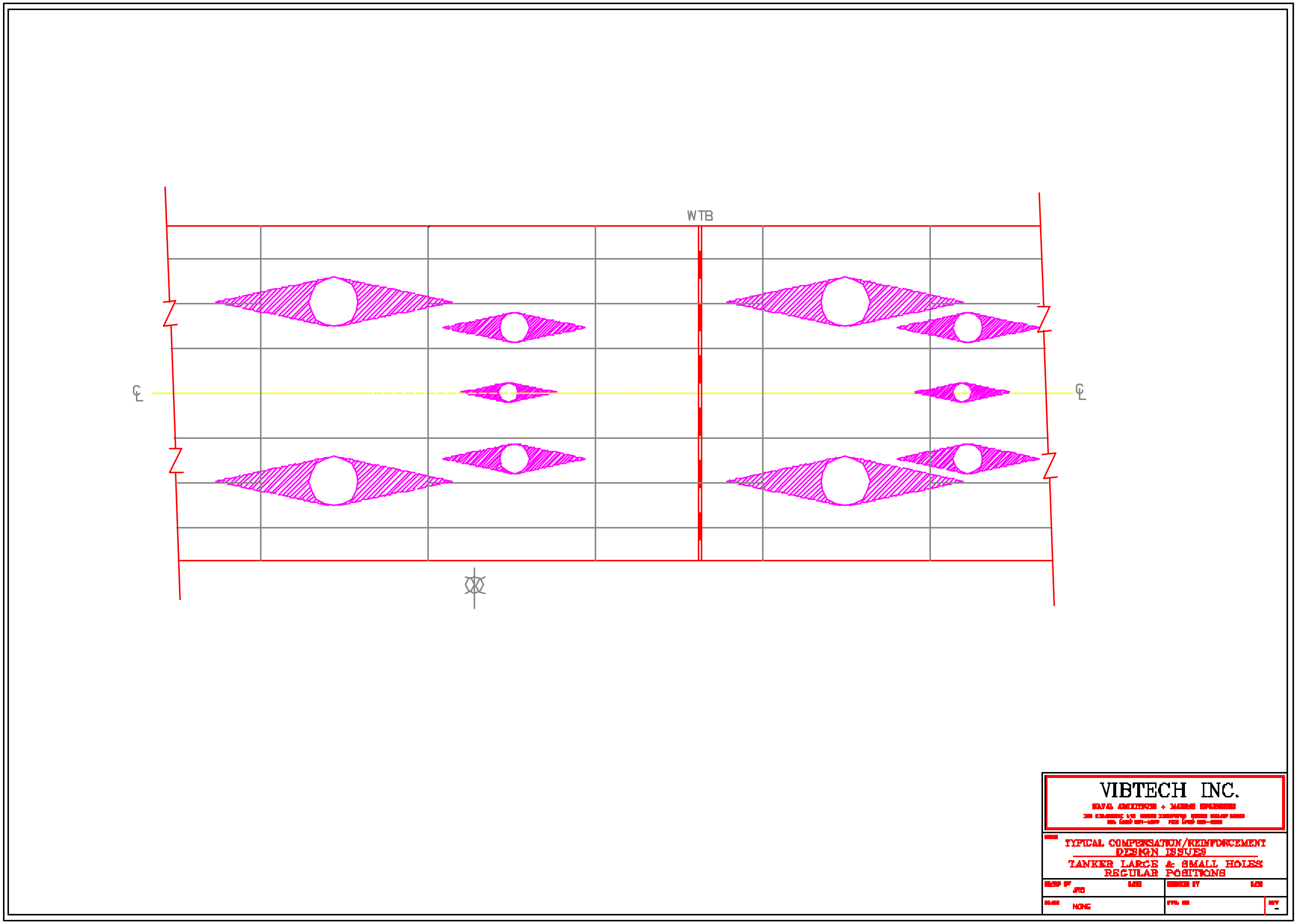


Figure 42: Circular Penetrations Showing Structurally Ineffective Areas.

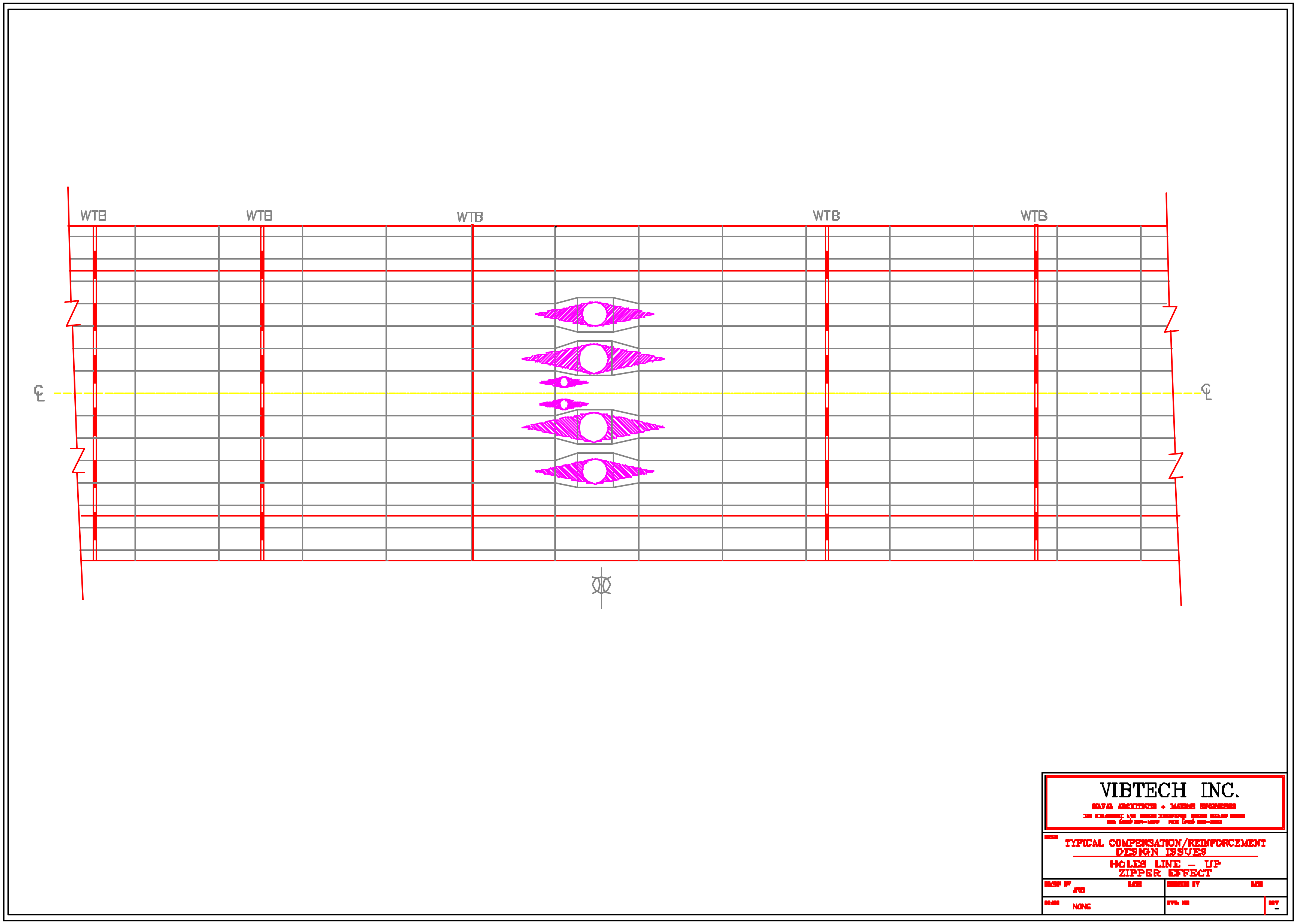


Figure 43: Circular Penetrations Arrayed in Line Creating a Zipper Effect.

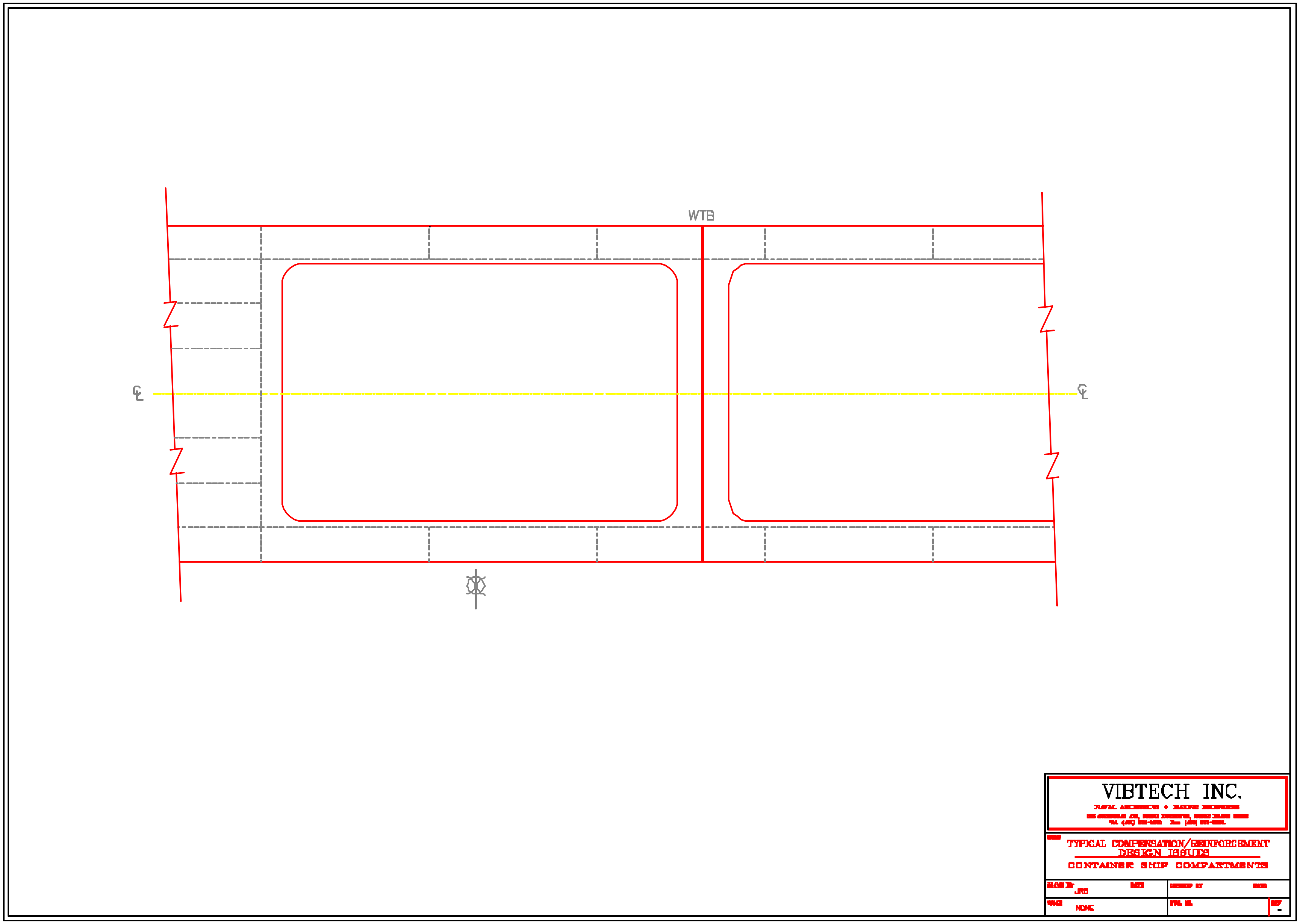


Figure 44: View Depicting Openings in Deck in way of Container Holds.

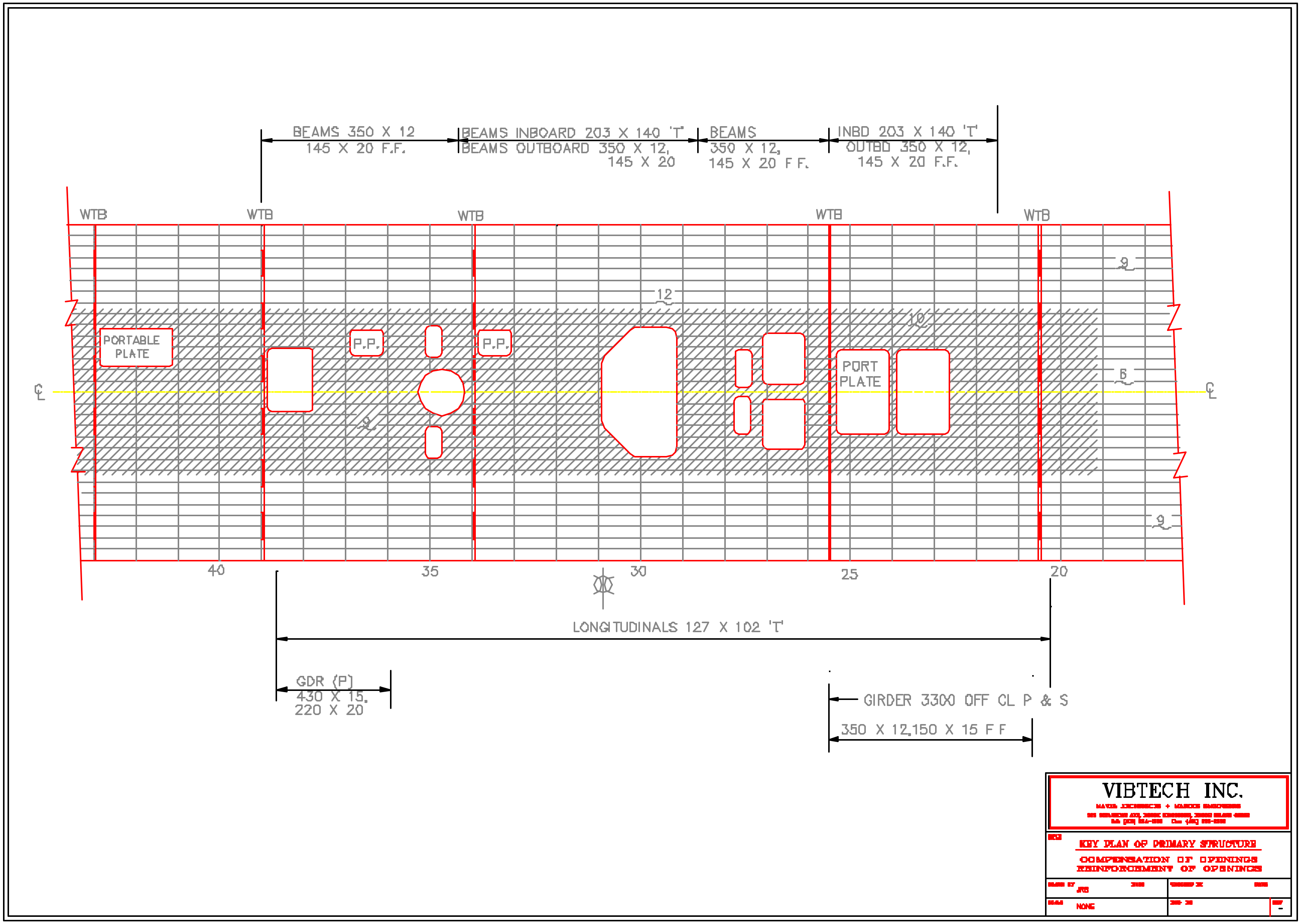


Figure 45: Example of Compensation of Deck Openings in a Combatant.

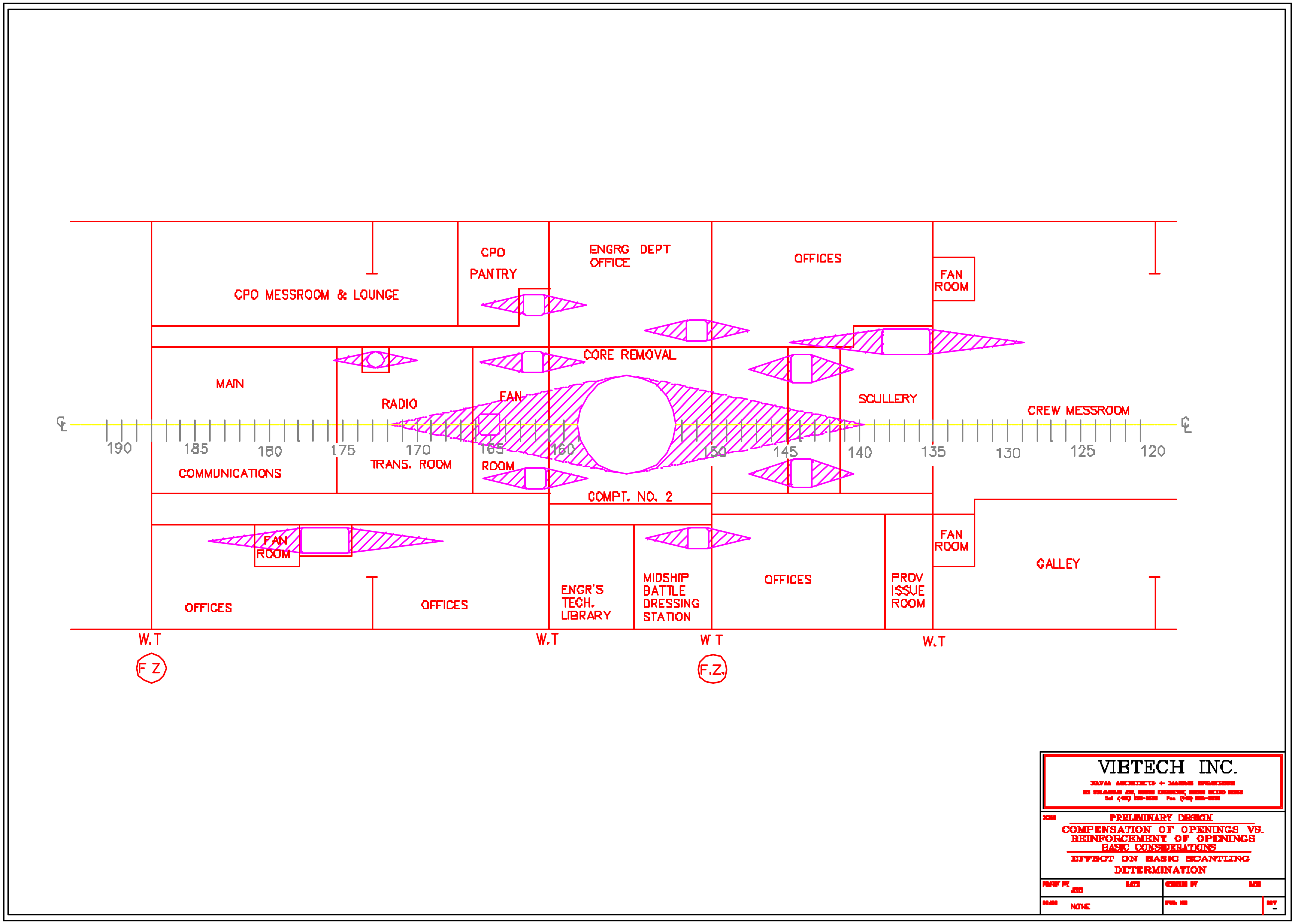


Figure 46: Arrangements Showing Penetrations for Distributive Systems.

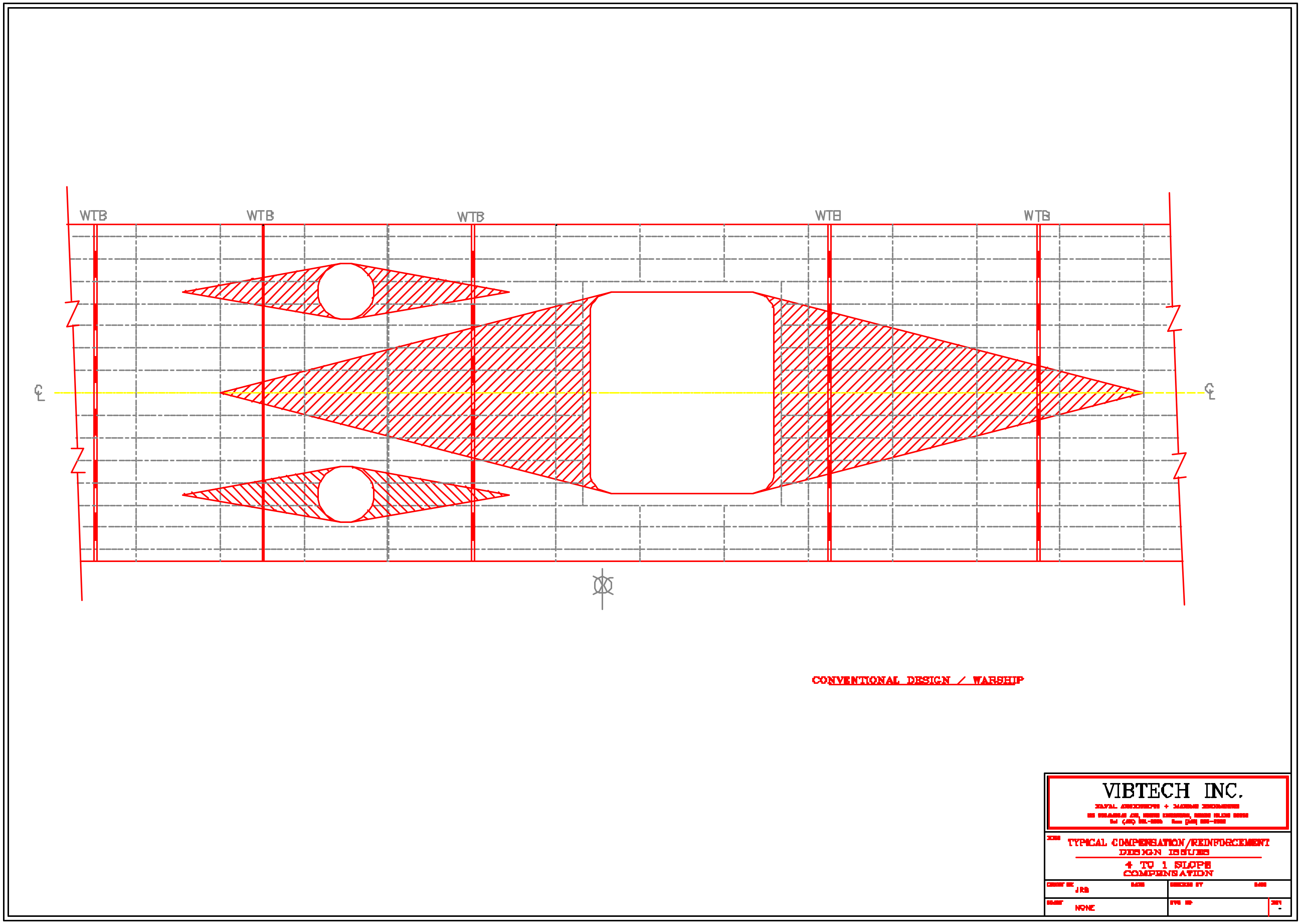


Figure 47: Large and Small Openings Showing Overlap of Ineffective Areas.

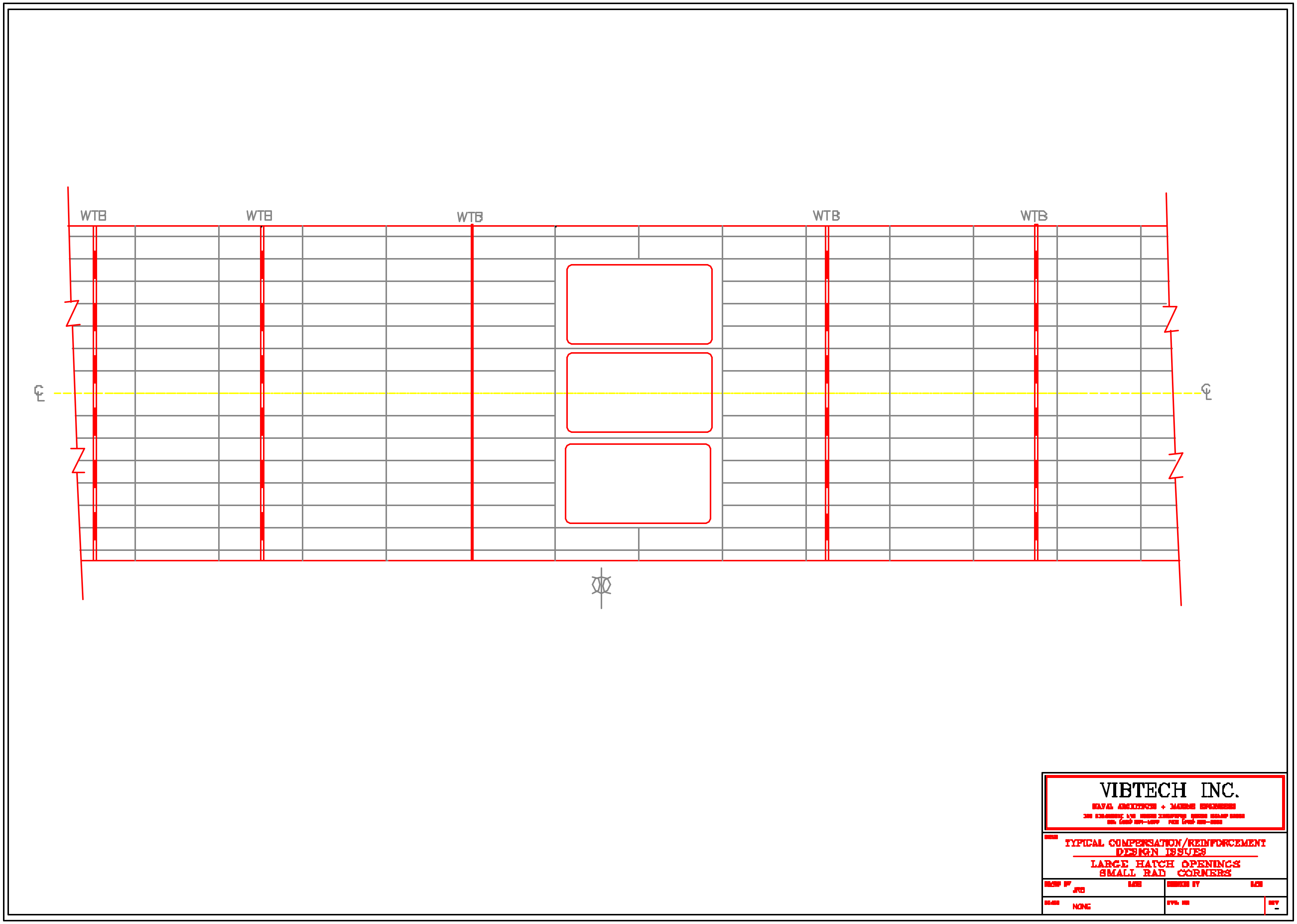


Figure 48: View of Deck with Cargo Hatch Openings.

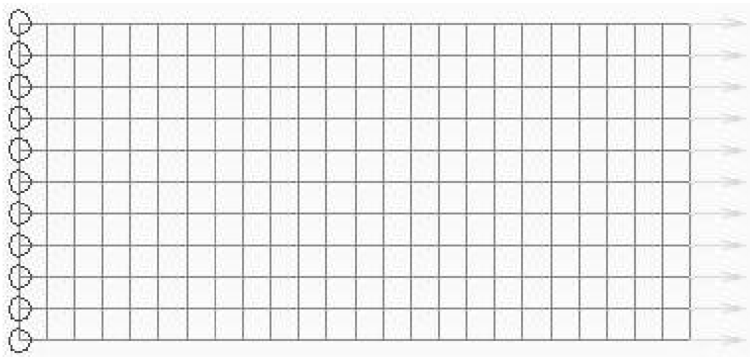
APPENDIX B: Representative Openings Finite Element Model Results

In performing this study, various finite element models were created to illustrate stress patterns and stress concentrations. These models are included in this appendix for further reference. The illustrations contained herein have stress include values of the maximum basic and local stress concentration factors for each model. In addition, enlarged views of important regions in the models have been presented with values for basic stress concentration factor labeled for important locations. In some cases, absolute stress levels may be presented. Following most models is a diagram with the various plating thickness used labeled. The nominal plate thickness in each case was .8125”. The far-field stress for each model was 23.5 Ksi. Updates to a few models have been made in order to achieve continuity in the work effort. Generally, the first model presented for each hole configuration is a model of an uncompensated, unreinforced plate of the appropriate geometry. Subsequent models for each configuration include compensation and reinforcement techniques that have been discussed. Please refer to the bibliography for further reference on compensation and reinforcement techniques that can be used to achieve a desired stress level in these models.

The following table lists the models contained herein and a short description of the model contents.

FILENAME	DESCRIPTION
SSC4	1x1 hole uncompensated, unreinforced
SSC4b	1x1 area-compensated hole
SSC4c	1x1 plate-reinforced hole, tOut=1, tMid=1.1875”, tIn=1.625”
SSC4d	1x1 plate-reinforced hole, thicker stringer, tOut=1.083”, tMid=1.1875”, tIn=1.625”
SSC4e	1x1 plate-reinforced hole with 1.25” stringer (tOut=1.25”, tMid=1.1875”, tIn=1.625”)
SSC4f	1x1 plate-reinforced deck plate with transversely uniform plate thickness (tmax=1.5625)
SSC4gb	1x1 base with insert (inside length of insert = 29.407”)
SSC5	1 big hole, partially finished
SSC5a	1 big hole, uncompensated, unreinforced
SSC5b	1 big hole, attempt at corner insert reinforcement
SSC5c	1 big hole, corner inserts and partially compensated
SSC6	2x1, base
SSC6b	2x1, area compensated
SSC6c	2x1, plate reinforced with straking
SSC7	One, cicular hole in the center of the deck
SSC7b	Zipper, 6 holes, base
SSC7c	Zipper, 6 holes, ring reinforced
SSC7d	Zipper, 6 holes, plate reinforced
SSC9newa	Collection of small holes with large circular hole uncompensated, unreinforced
SSC11	2 Large hatches uncompensated, unreinforced
SSC11a	Revised 2 Large hatches uncompensated, unreinforced
SSC12	3 Adjacent hatches, uncompensated, unreinforced
SSC12a	Revised 3 Adjacent hatches, uncompensated, unreinforced

Filename:	SSC2	Applied Stress:	23.5 ksi
Minimum Stress:	23.5 ksi	Maximum Stress:	23.5 ksi
BSCF (k_{bo})	1.000	FWSCF (k_{b1})	1.000



NOTES:

At all points on this plate have the same stress value of 23,539 psi (23.5 ksi). The lack of discontinuities in this model result in no stress concentration effect. This is to say that the value of the Basic Stress Concentration Factor (BSCF) is one at all points.

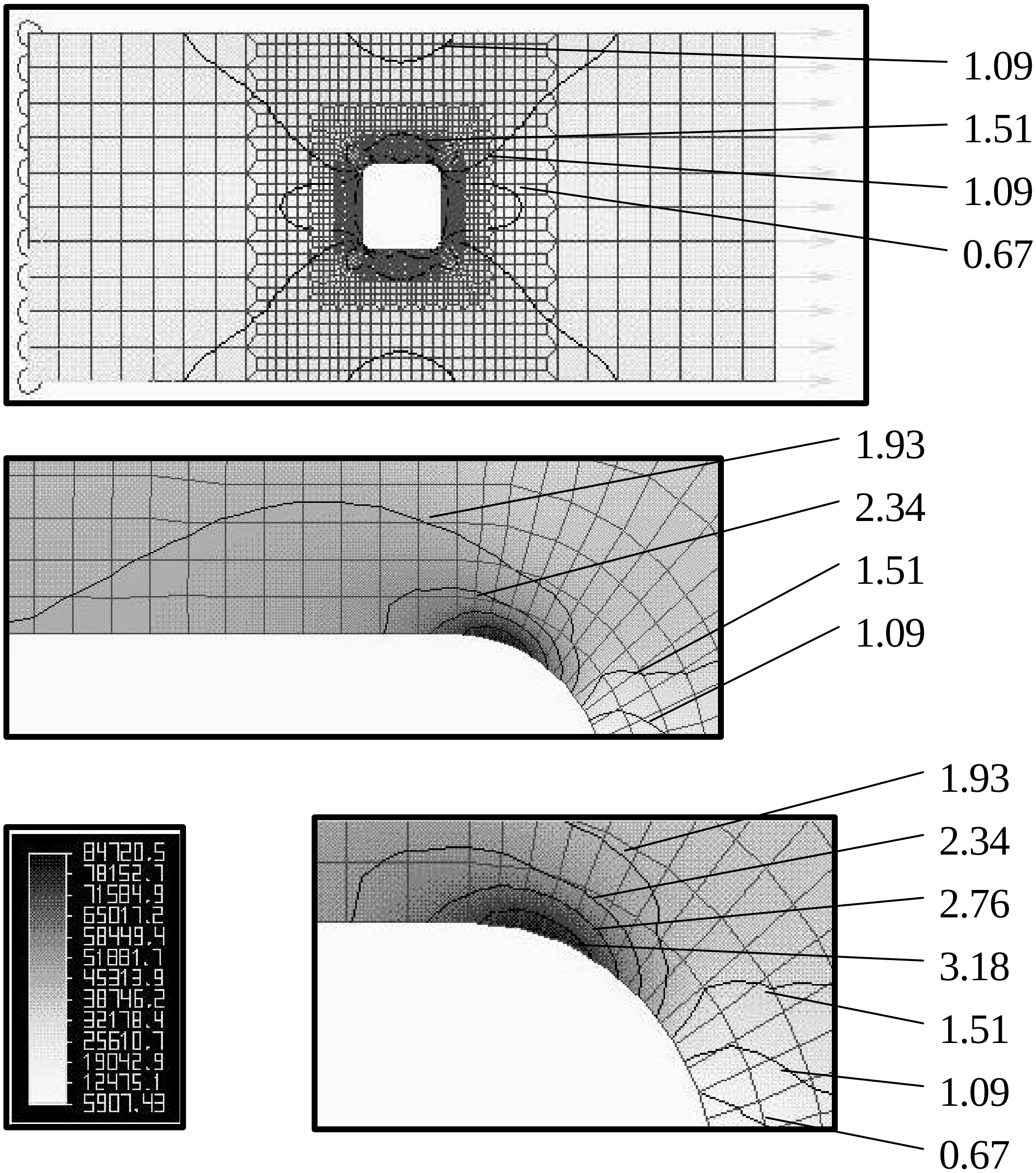
The sum of the forces applied to the right-hand side model divided by the cross-sectional area of the plate is equal to 23,539 psi. The forces applied at the corner nodes of the right-hand side of the plate are half the magnitude of the forces applied to the nodes in the middle. This was done assuming that the end nodes act on half the area that the center nodes do.

The plate is constrained along the left edge (X = 0). All nodes are constrained from translation in the “X” direction, and the middle node is also constrained from translation in the “Y” direction. These boundary conditions were put in place to allow for the Poisson effect, which causes the plate to contract along the left-hand side as it is pulled.

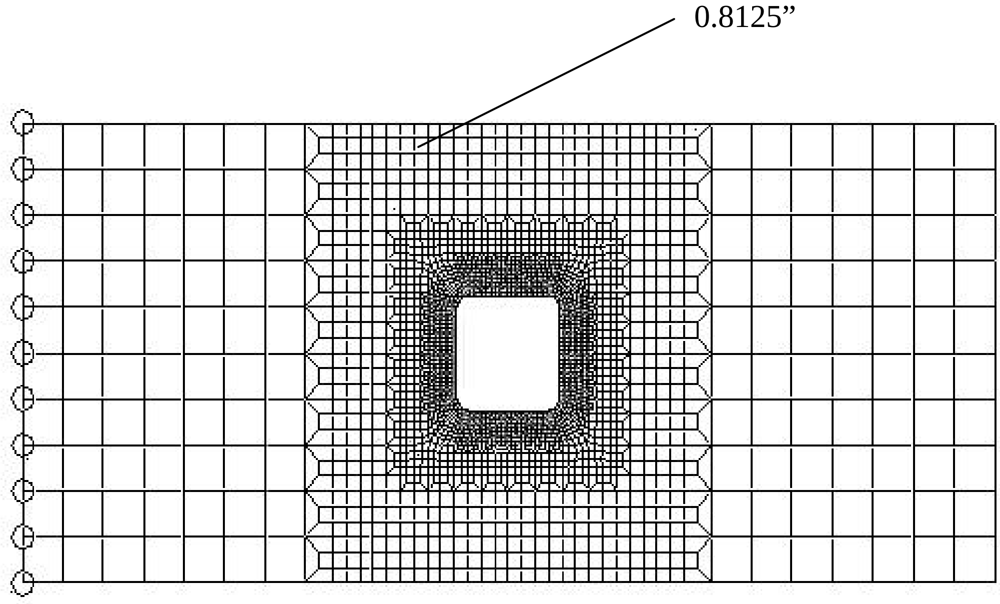
The plate is 1728” x 732.3” x 0.8125” (144’ x 61.025’x 0.0677’). Subsequent models vary in length to allow the stresses to be smoothly distributed.

CASE 2: 1X1 Central Hole

Filename:	SSC4	Applied Stress:	23.5 ksi
Minimum Stress:	5.91 ksi	Maximum Stress:	84.7 ksi
BSCF (k_{bo})	3.599	FWSCF (k_{b1})	2.699

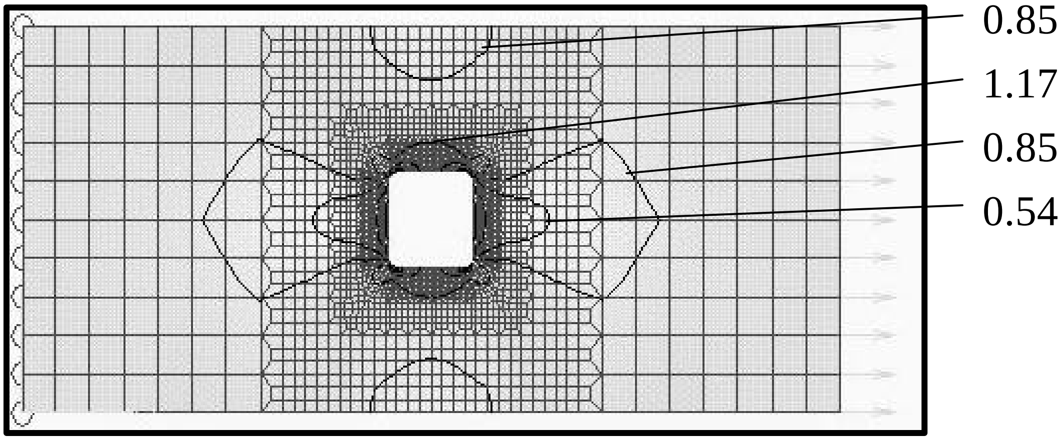


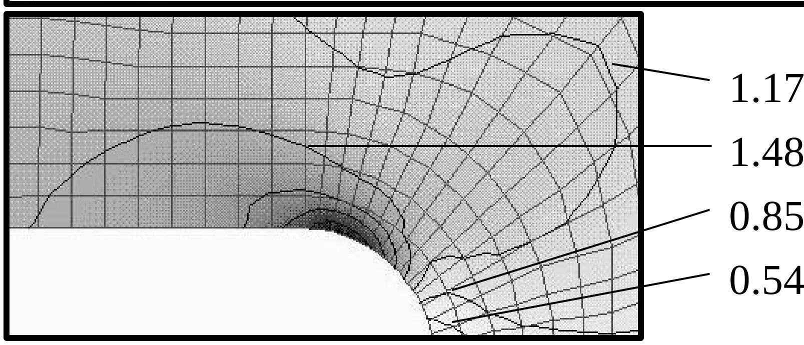
CASE 2: 1X1 Central Hole



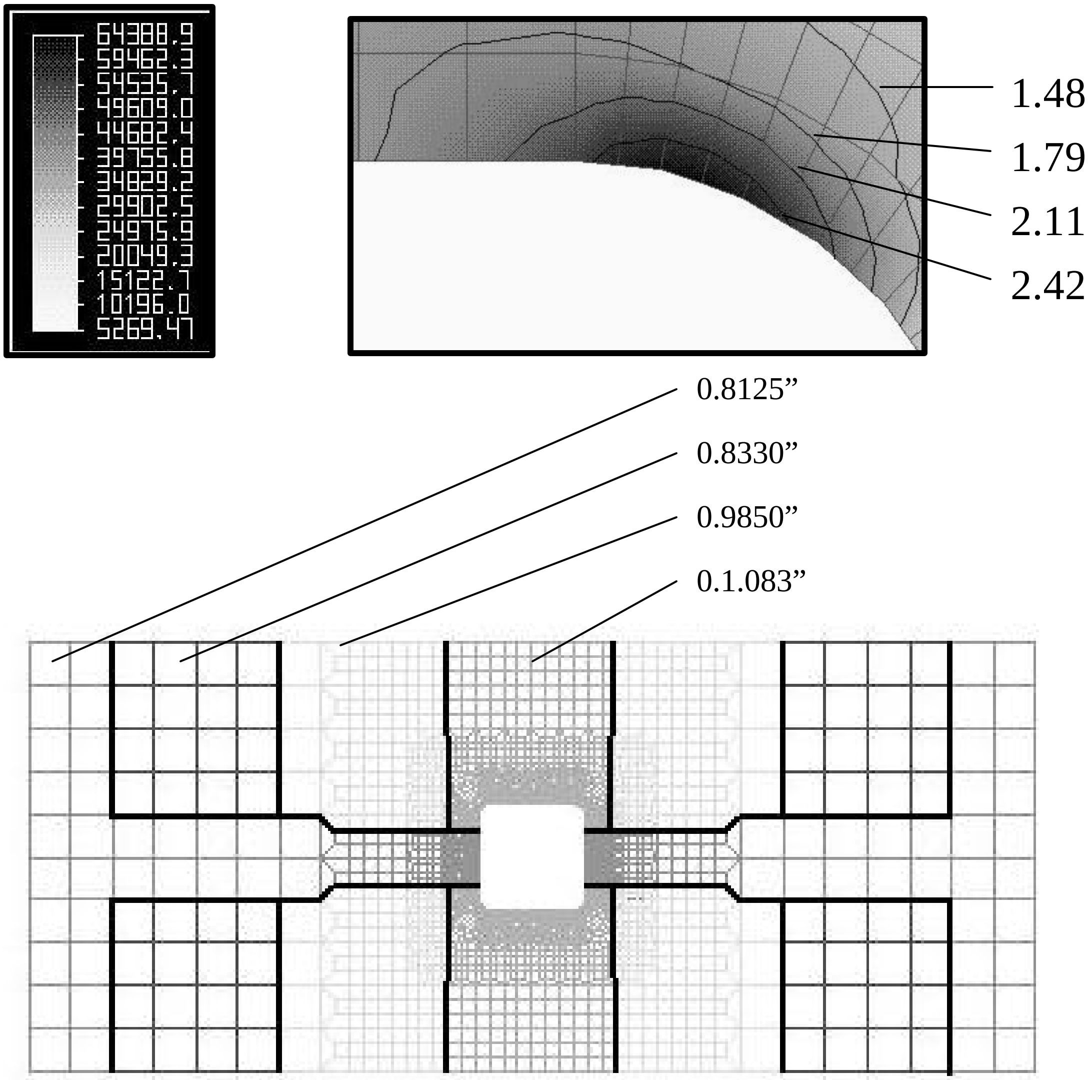
CASE 2: 1X1 Central Hole

Filename:	SSC4b	Applied Stress:	23.5 ksi
Minimum Stress:	5.27 ksi	Maximum Stress:	64.4 ksi
BSCF (k_{bo})	2.735	FWSCF (k_{b1})	2.734



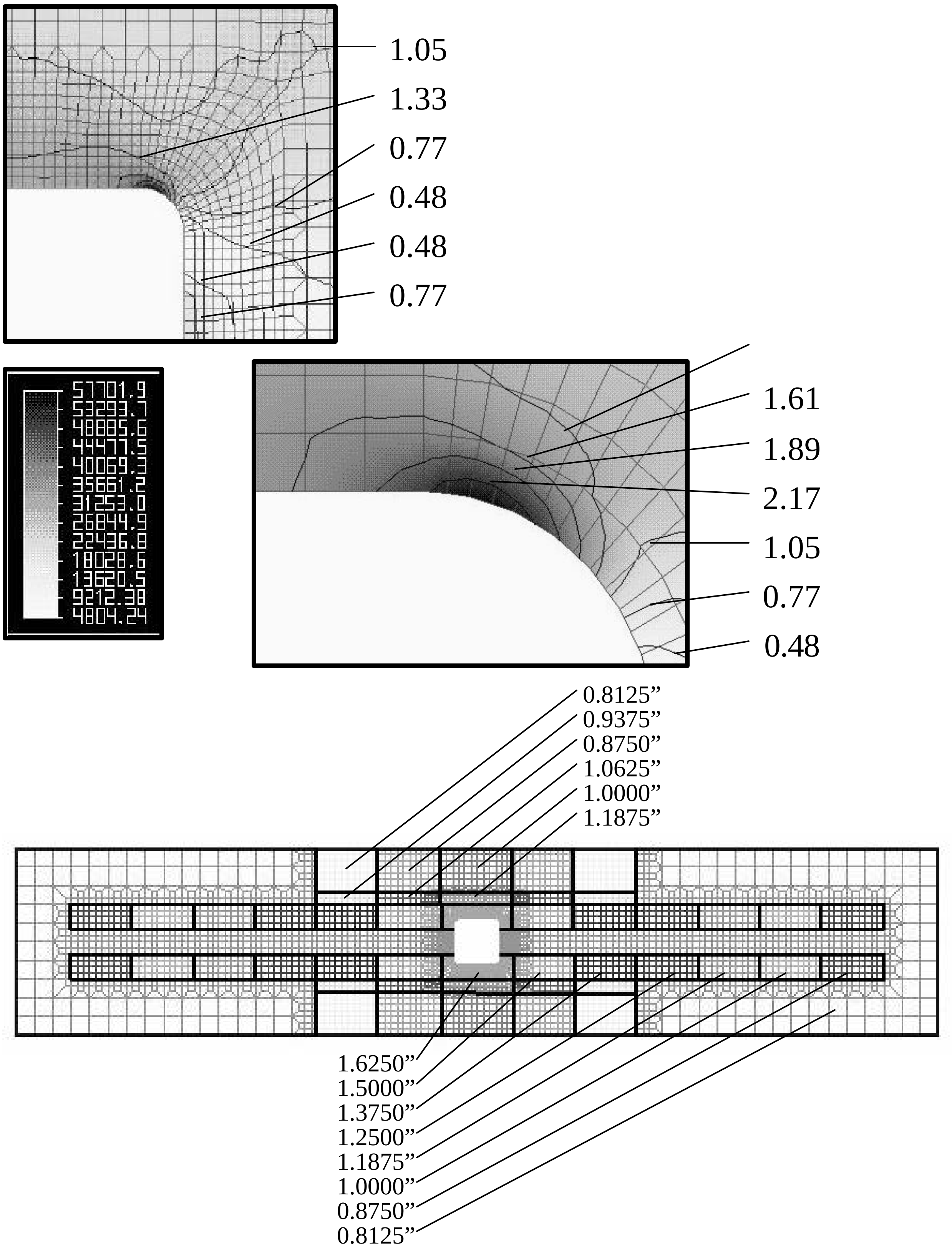


CASE 2: 1X1 Central Hole



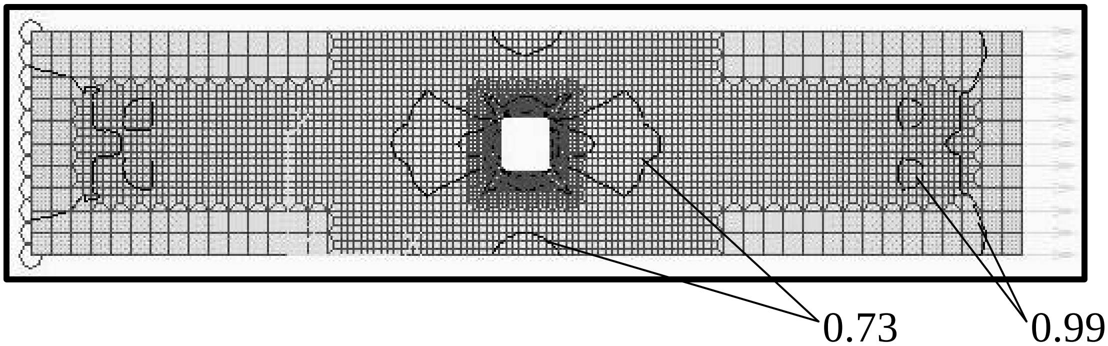
CASE 2: 1X1 Central Hole

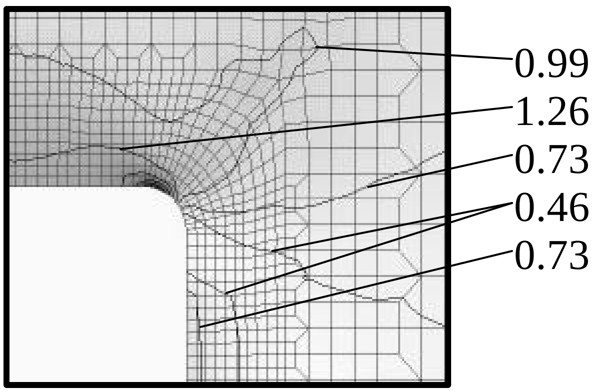
Filename:	SSC4c	Applied Stress:	23.5 ksi
Minimum Stress:	4.80 ksi	Maximum Stress:	57.7 ksi
BSCF (k_{bo})	2.451	FWSCF (k_{b1})	2.582



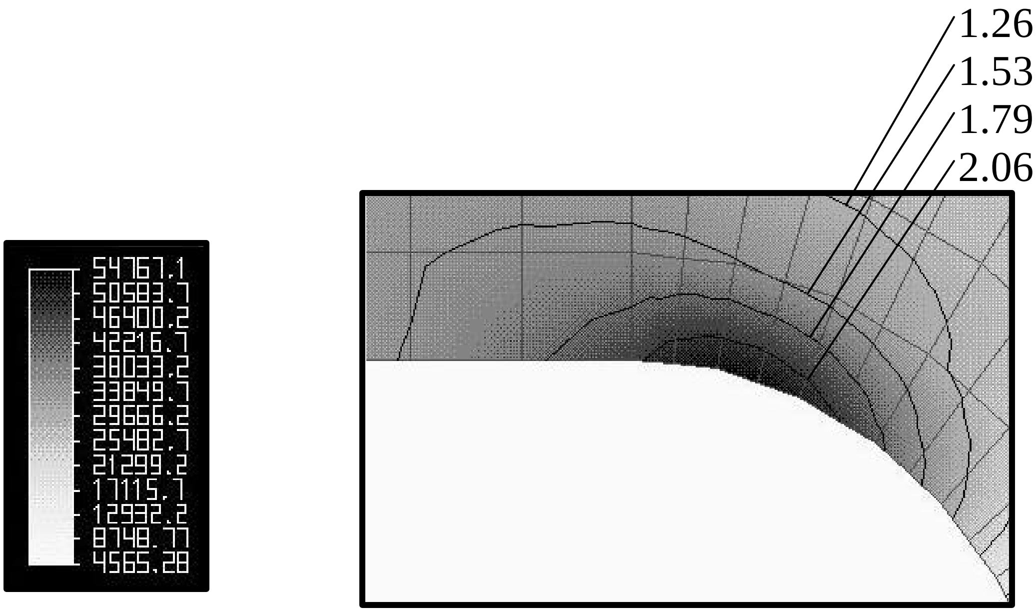
CASE 2: 1X1 Central Hole

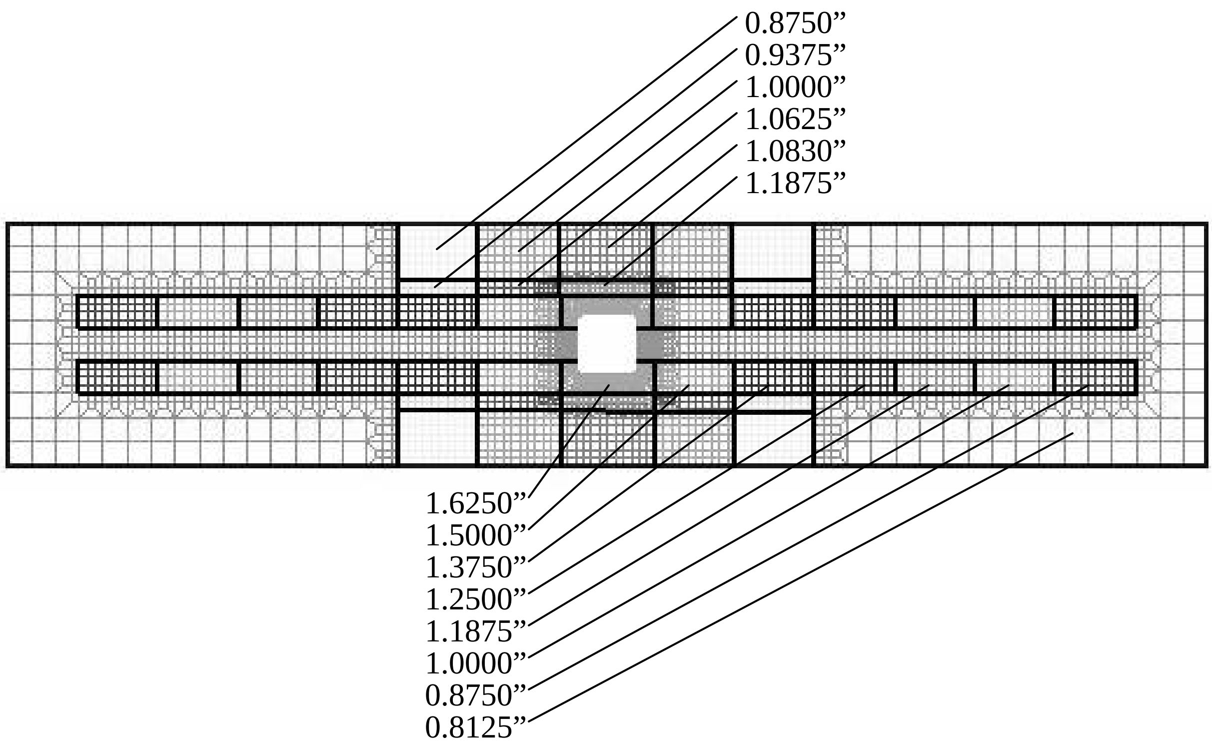
Filename:	SSC4d	Applied Stress:	23.5 ksi
Minimum Stress:	4.57 ksi	Maximum Stress:	54.8 ksi
BSCF (k_{bo})	2.327	FWSCF (k_{b1})	2.569





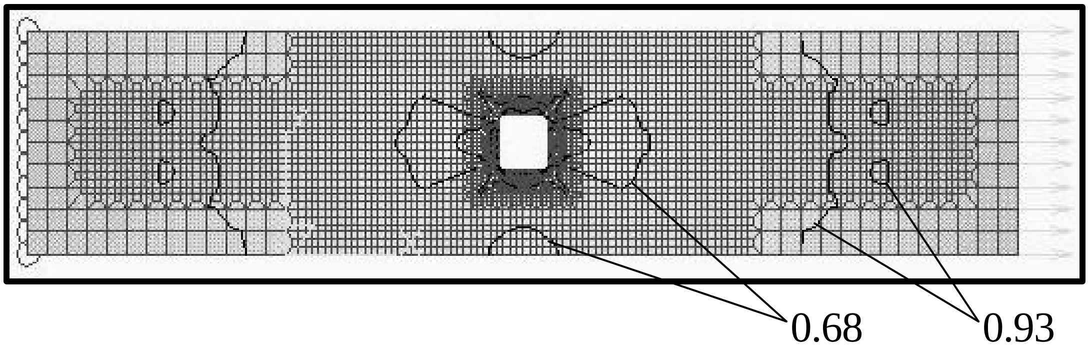
CASE 2: 1X1 Central Hole

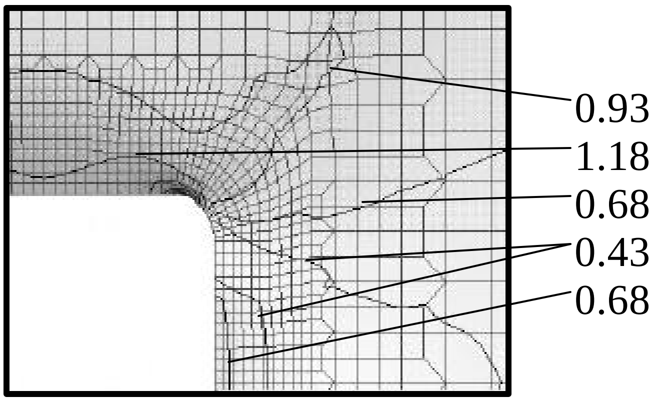




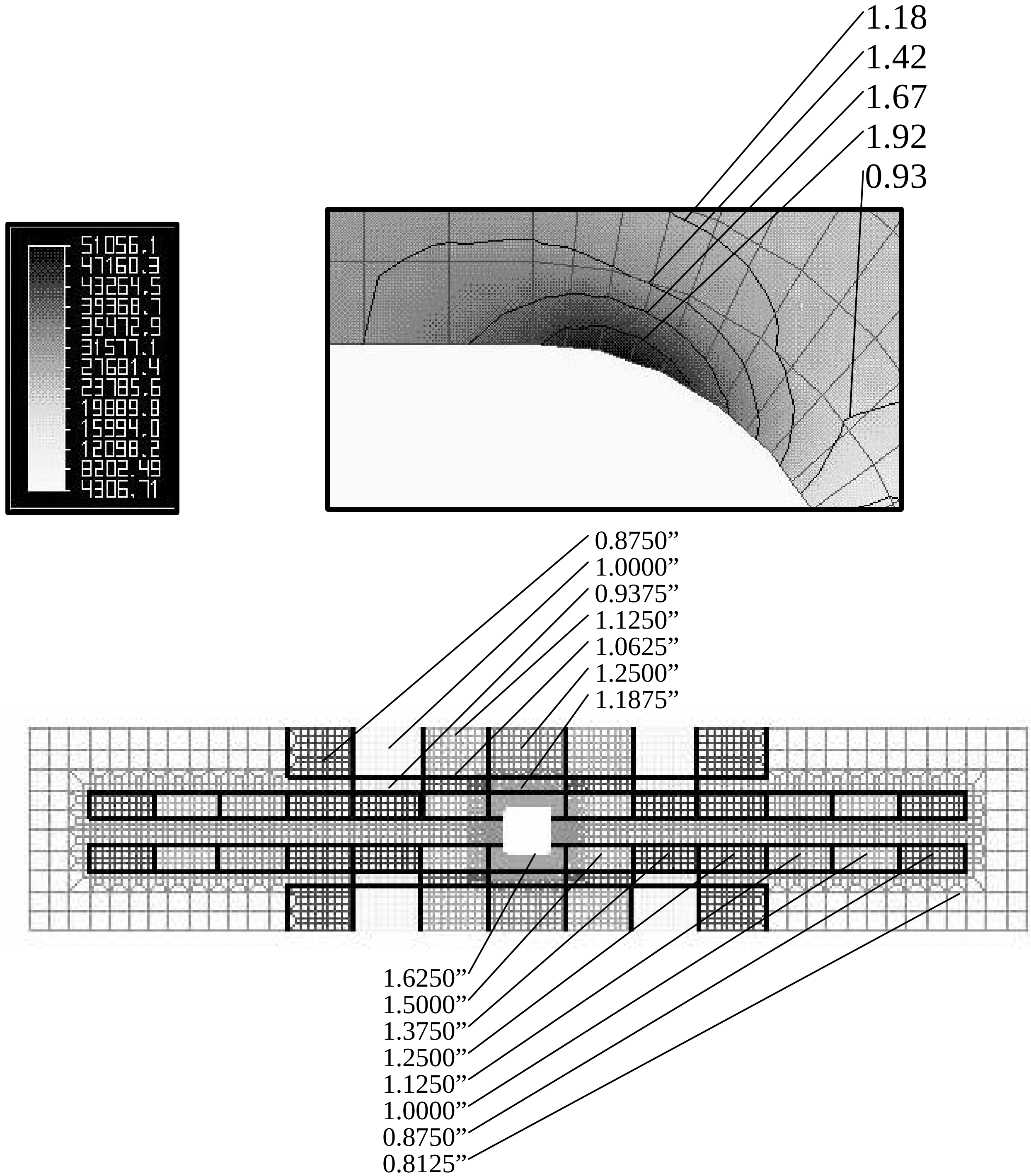
CASE 2: 1X1 Central Hole

Filename:	SSC4e	Applied Stress:	23.5 ksi
Minimum Stress:	4.31 ksi	Maximum Stress:	51.1
BSCF (k_{bo})	2.169 ksi	FWSCF (k_{b1})	2.618 ksi



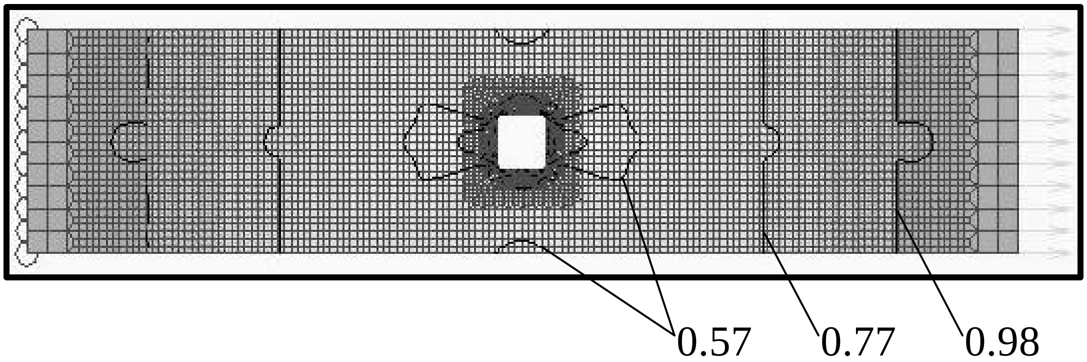


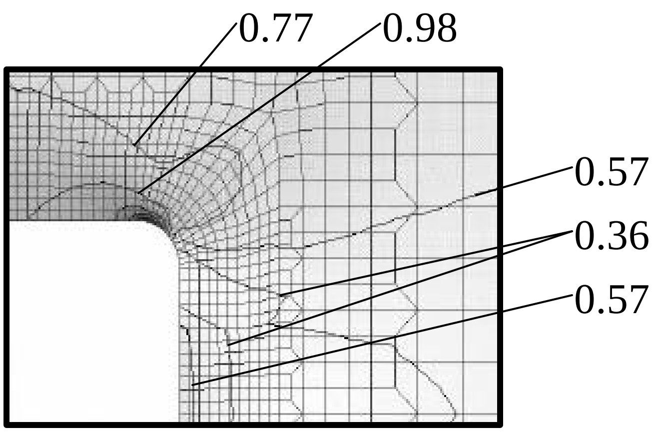
CASE 2: 1X1 Central Hole



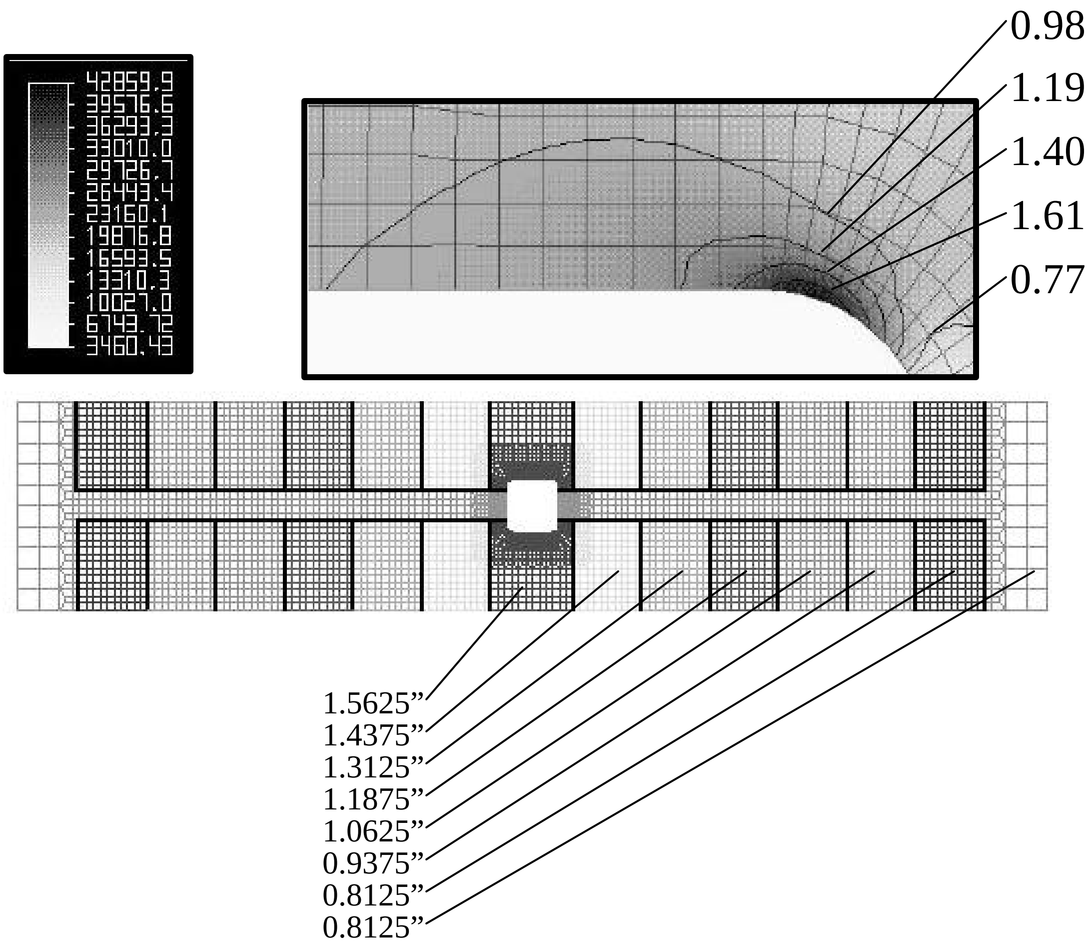
CASE 2: 1X1 Central Hole

Filename:	SSC4f	Applied Stress:	23.5 ksi
Minimum Stress:	3.46 ksi	Maximum Stress:	42.9 ksi
BSCF (k_{bo})	1.821	FWSCF (k_{b1})	2.626



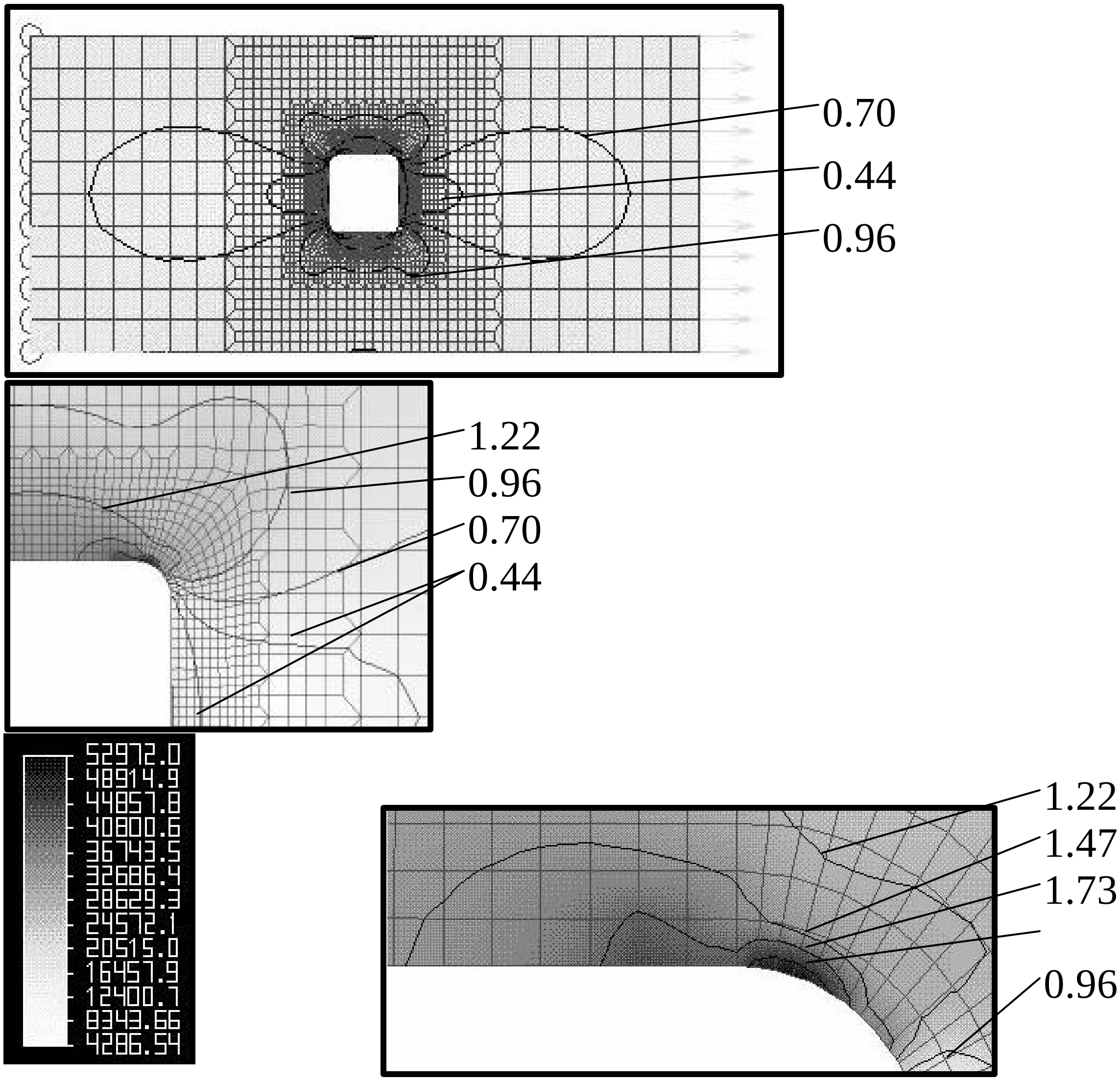


CASE 2: 1X1 Central Hole

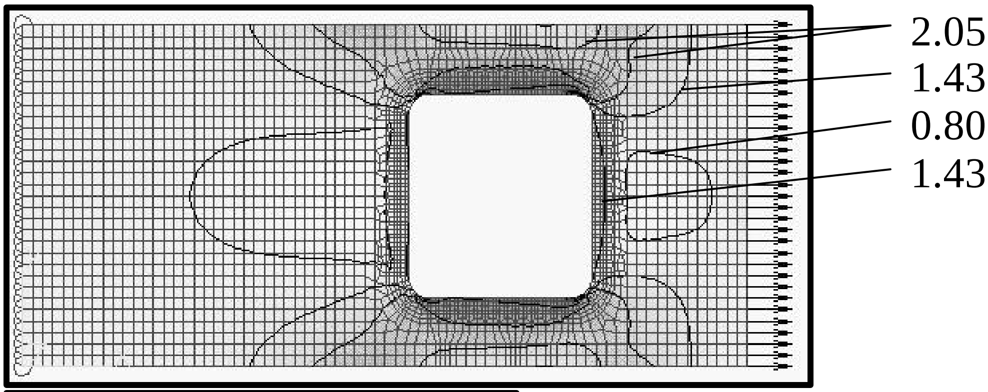


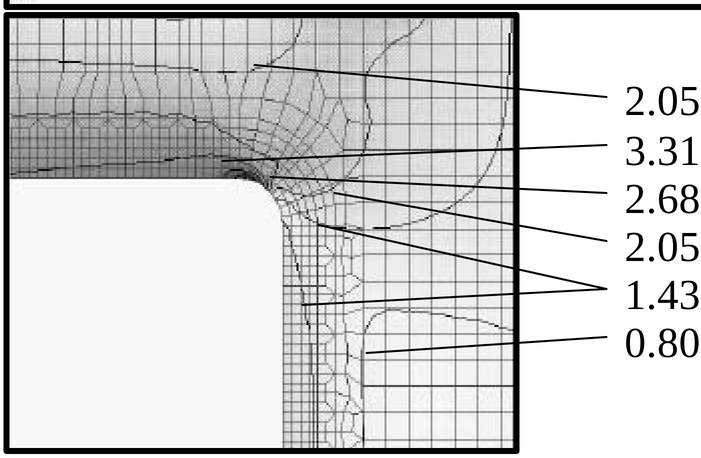
INSERT WITH THE LOWEST BSCF

Filename:	SSC4gb	Applied Stress:	23.5 ksi
Minimum Stress:	4.29 ksi	Maximum Stress:	53.0 ksi
BSCF (k_{bo})	2.250	FWSCF (k_{b1})	2.337

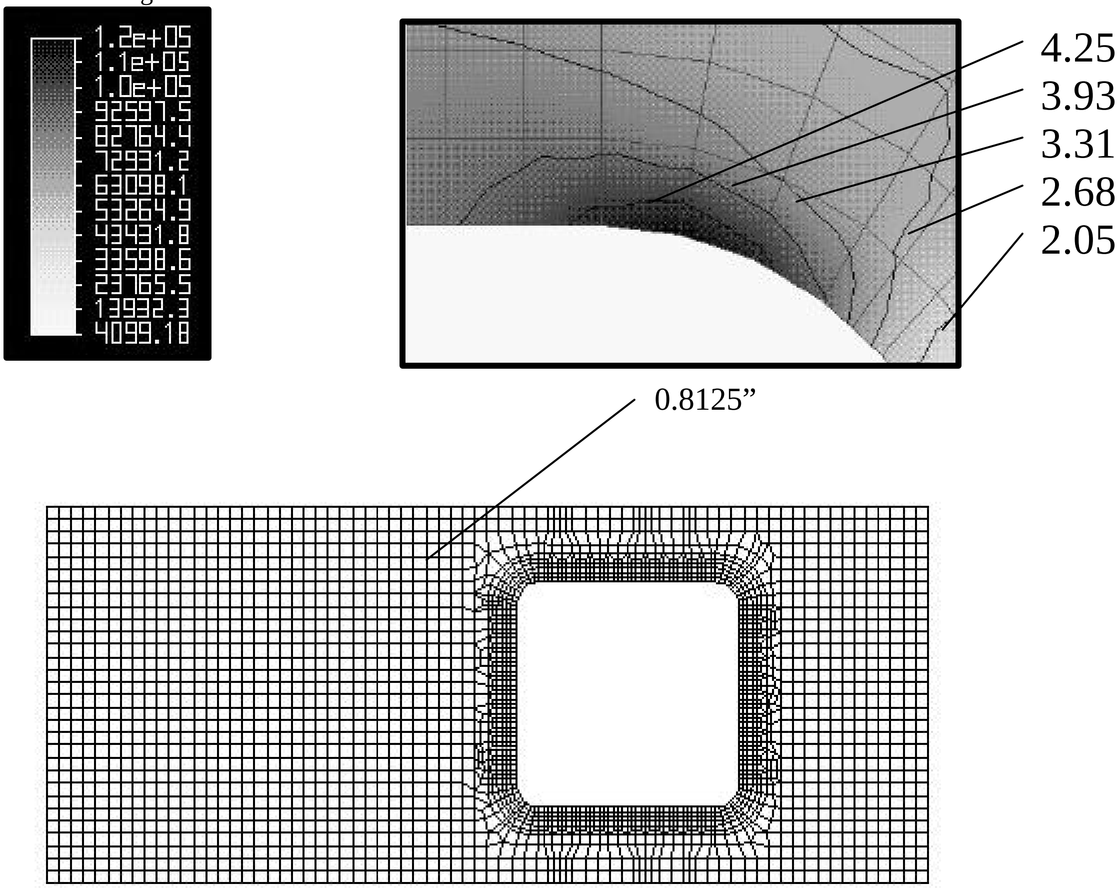


Filename:	SSC5	Applied Stress:	23.5 ksi
Minimum Stress:	4.10 ksi	Maximum Stress:	122 ksi
BSCF (k_{bo})	5.187	FWSCF (k_{b1})	2.092



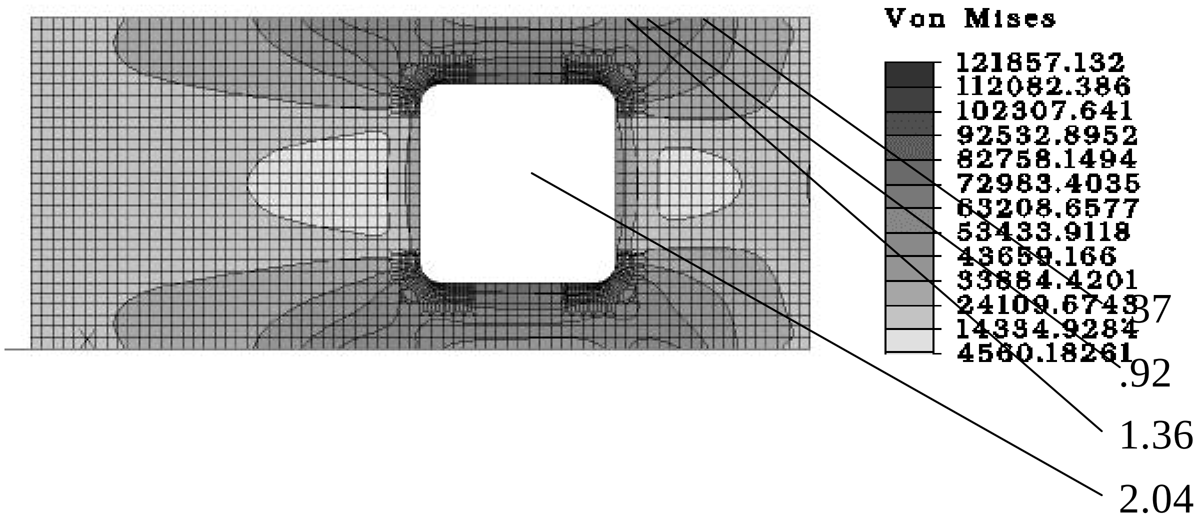


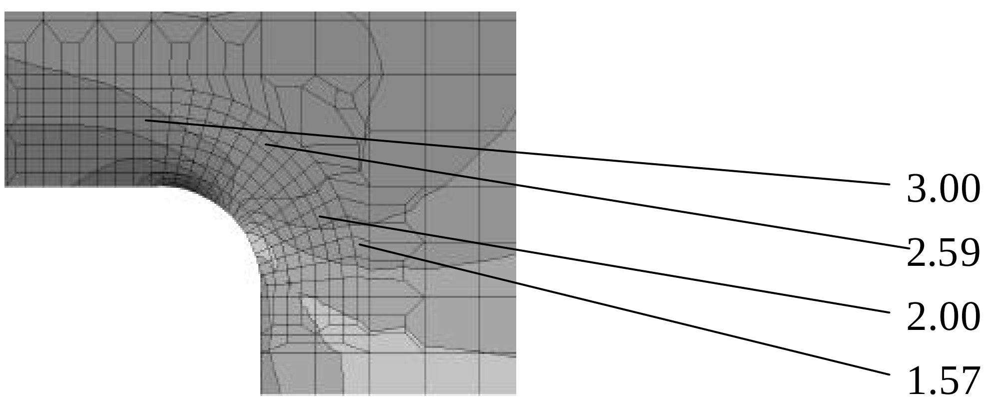
CASE 3: Large Central Hole

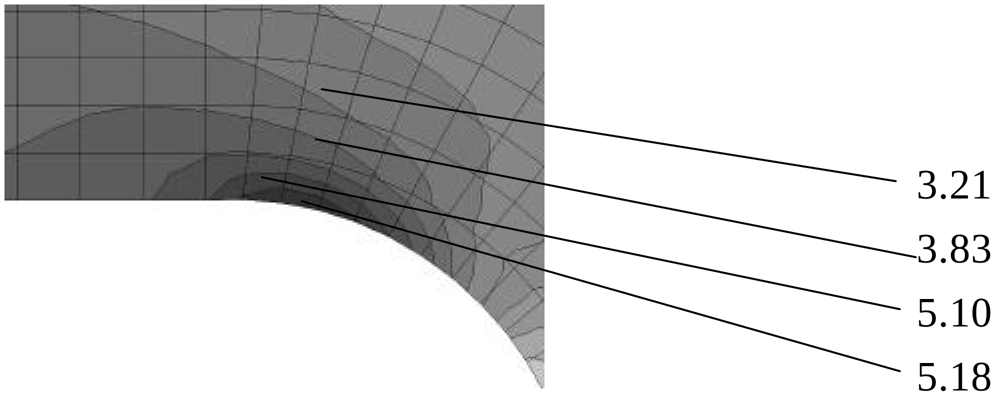


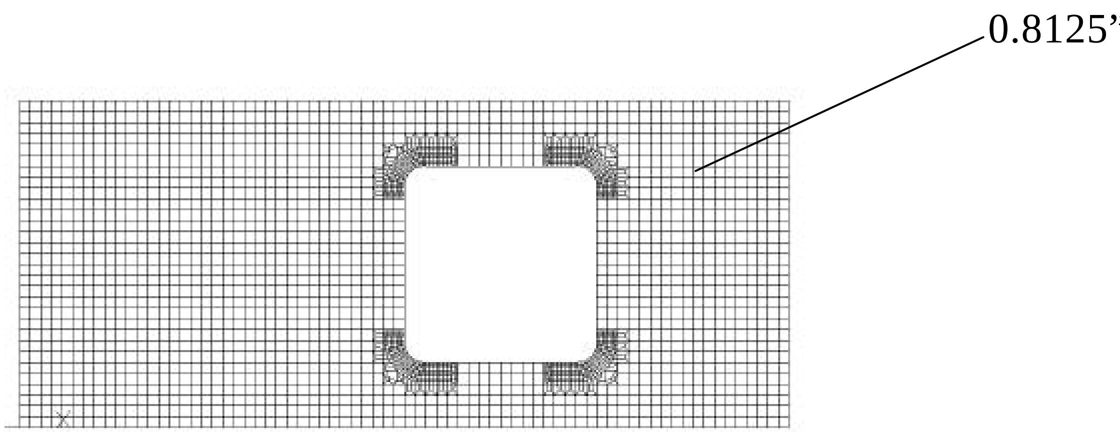
CASE 3: Large Central Hole

Filename:	SSC5a	Applied Stress:	23.5 ksi
Minimum Stress:	4.6 ksi	Maximum Stress:	121.9 ksi
BSCF (k_{bo})	5.19	FWSCF (k_{b1})	2.07



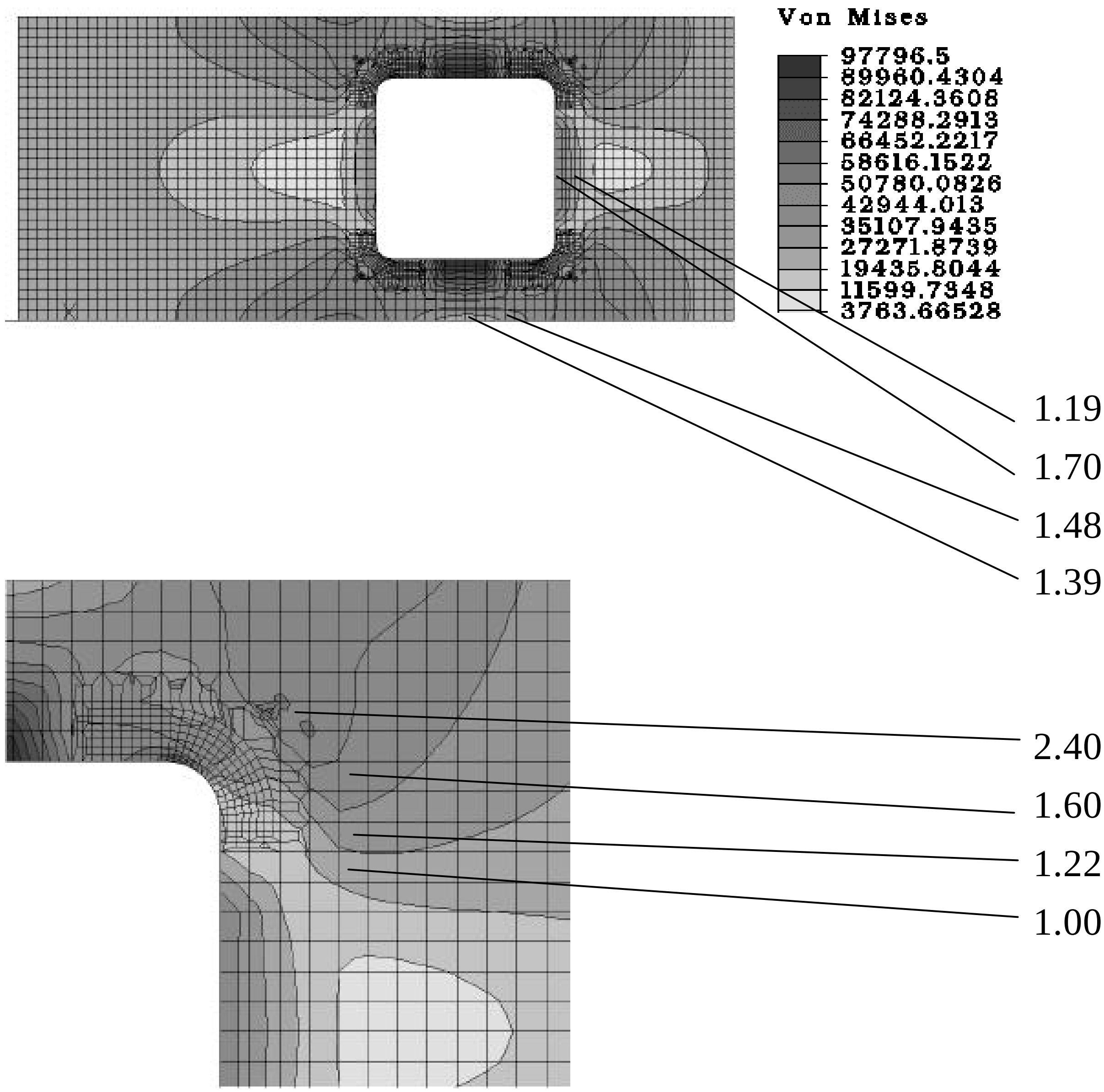


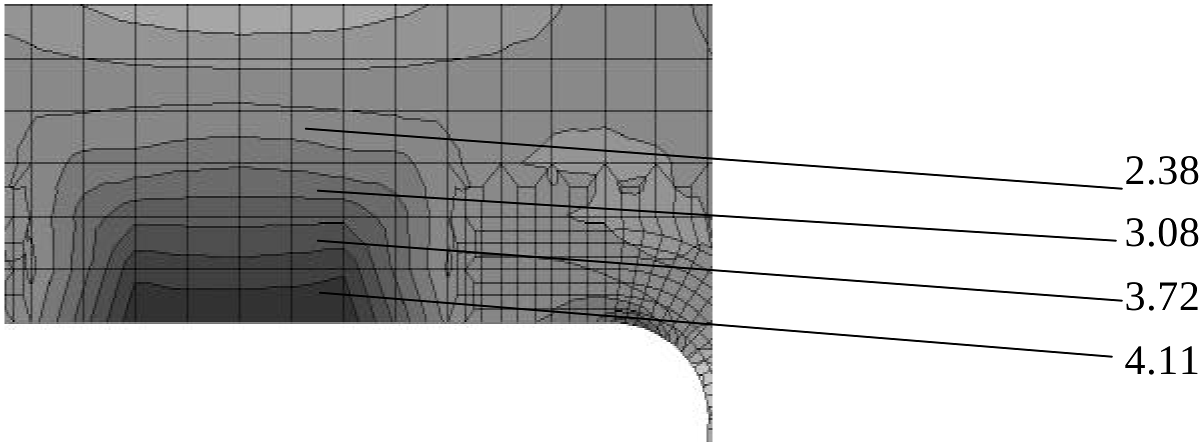

3.21
3.83
5.10
5.18

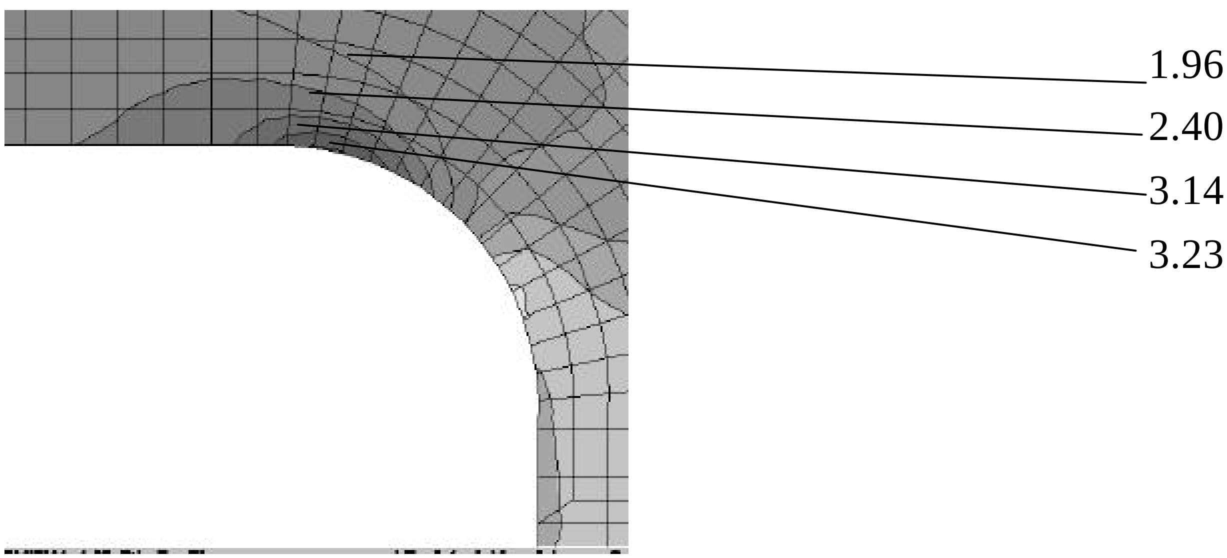

0.8125”

CASE 3: Large Central Hole

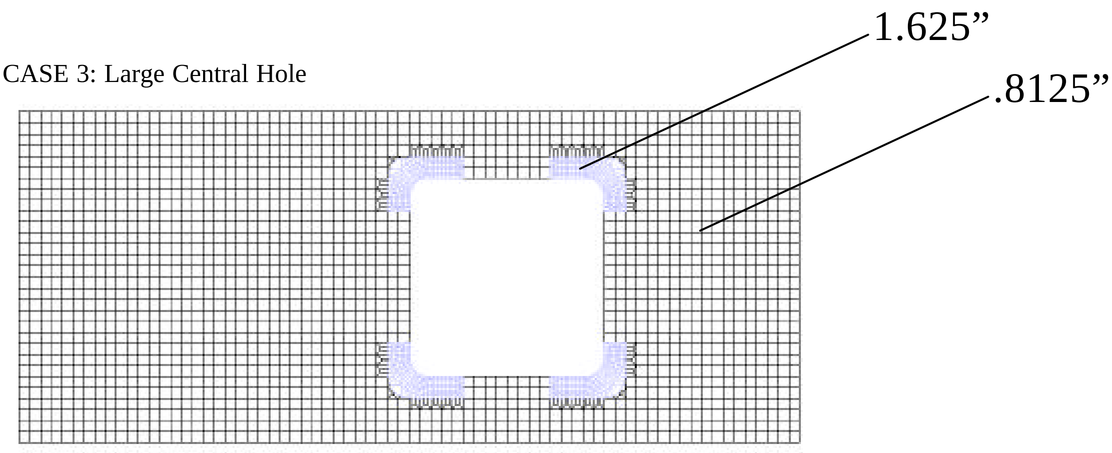
Filename:	SSC5b	Applied Stress:	23.5 ksi
Minimum Stress:	3.7 ksi	Maximum Stress:	97.8 ksi
BSCF (k_{bo})	4.16	FWSCF (k_{b1})	2.22




2.38
3.08
3.72
4.11

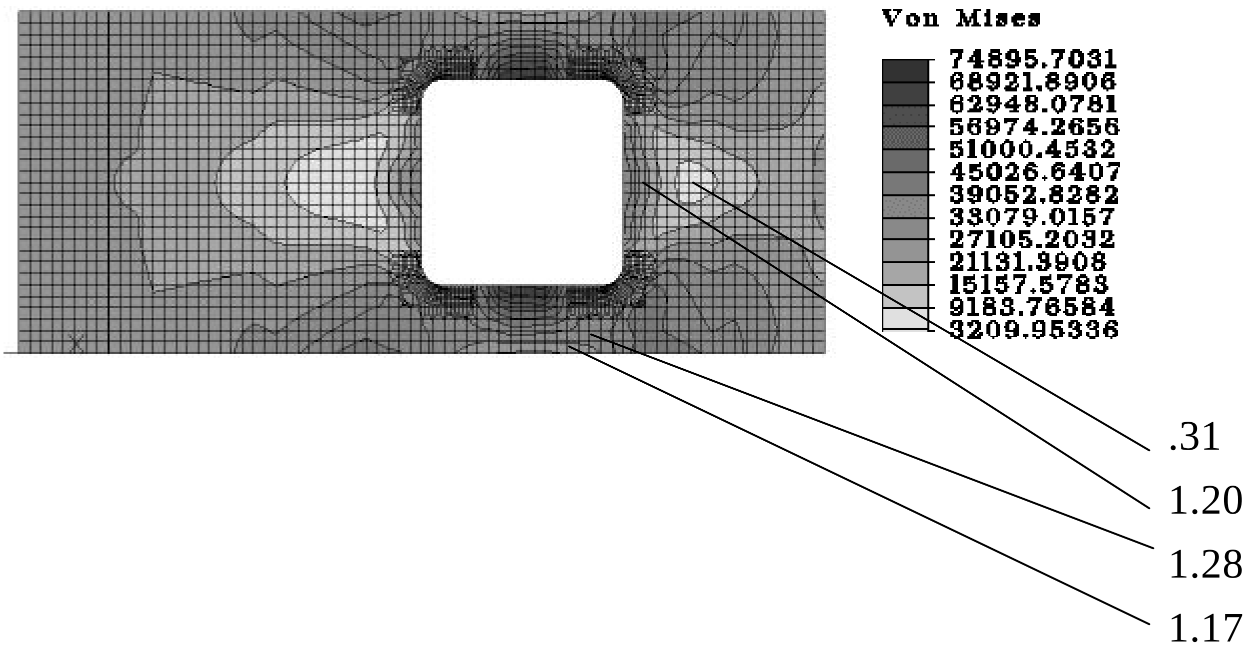

1.96
2.40
3.14
3.23

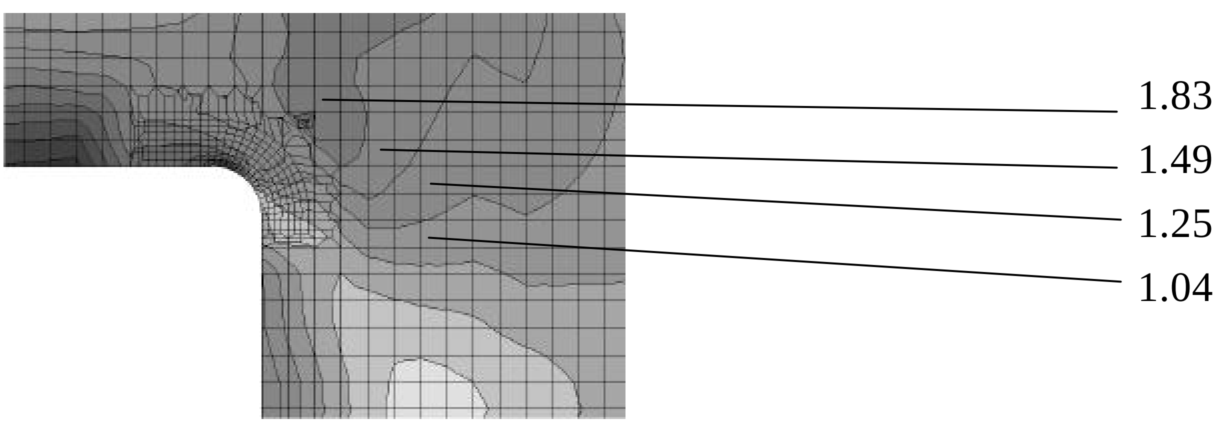
CASE 3: Large Central Hole

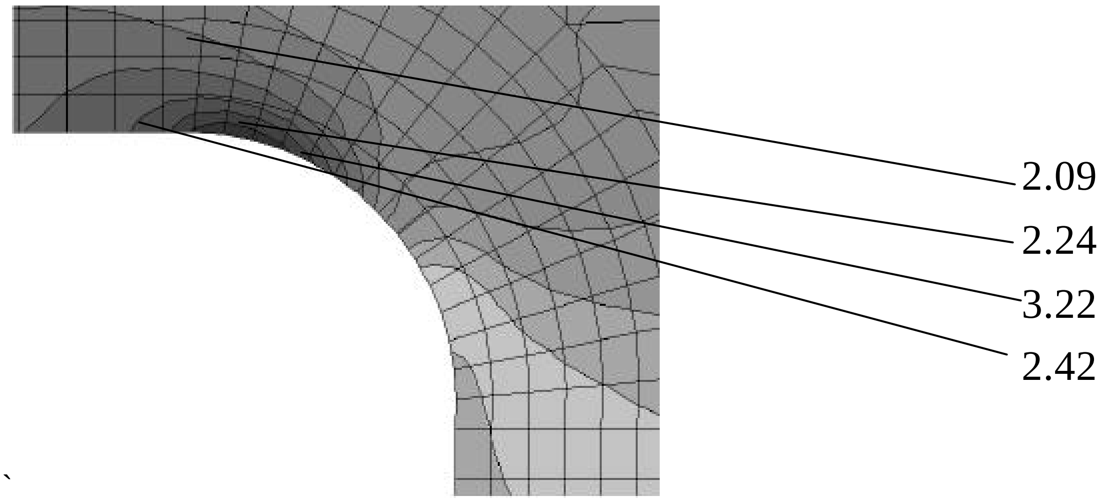


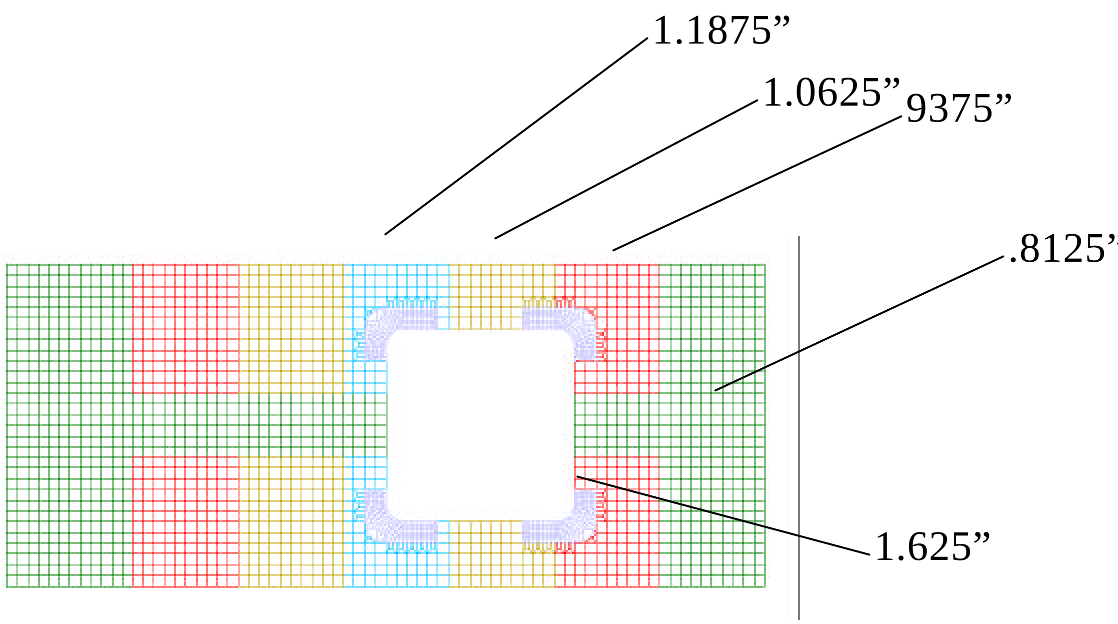
CASE 3: Large Central Hole

Filename:	SSC5c	Applied Stress:	23.5 ksi
Minimum Stress:	3.2 ksi	Maximum Stress:	74.9 ksi
BSCF (k_{bo})	3.18	FWSCF (k_{b1})	1.49



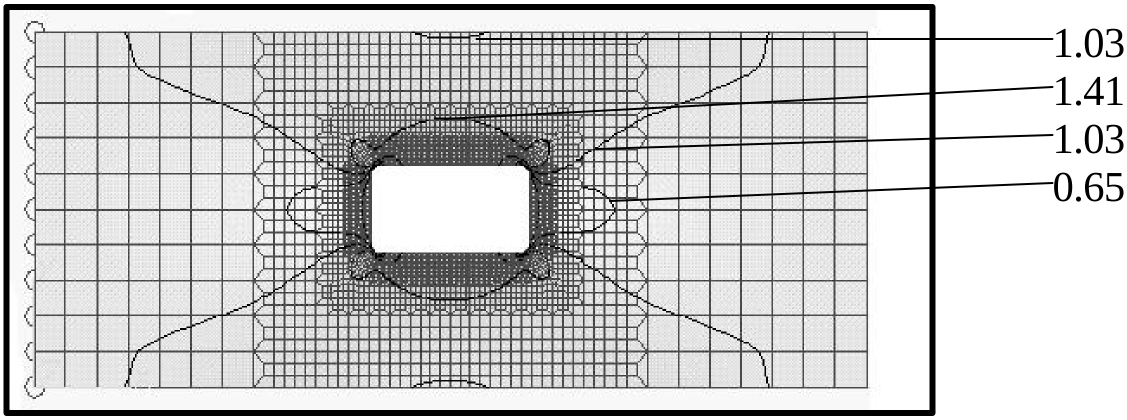


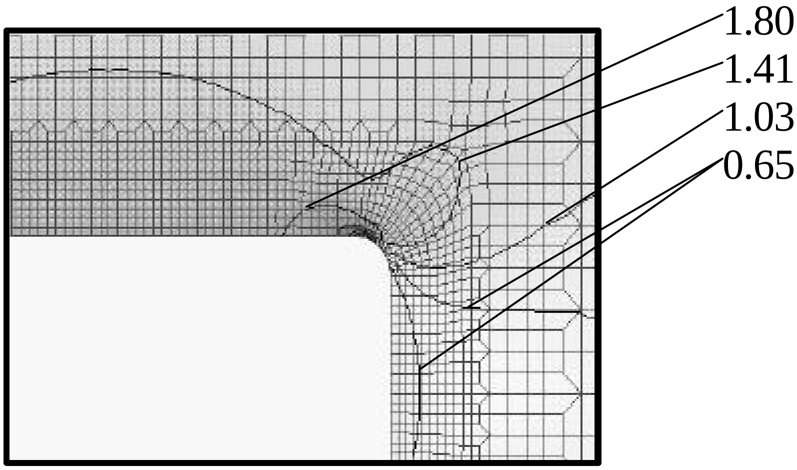

2.09
2.24
3.22
2.42


1.1875”
1.0625”
9375”
.8125”
1.625”

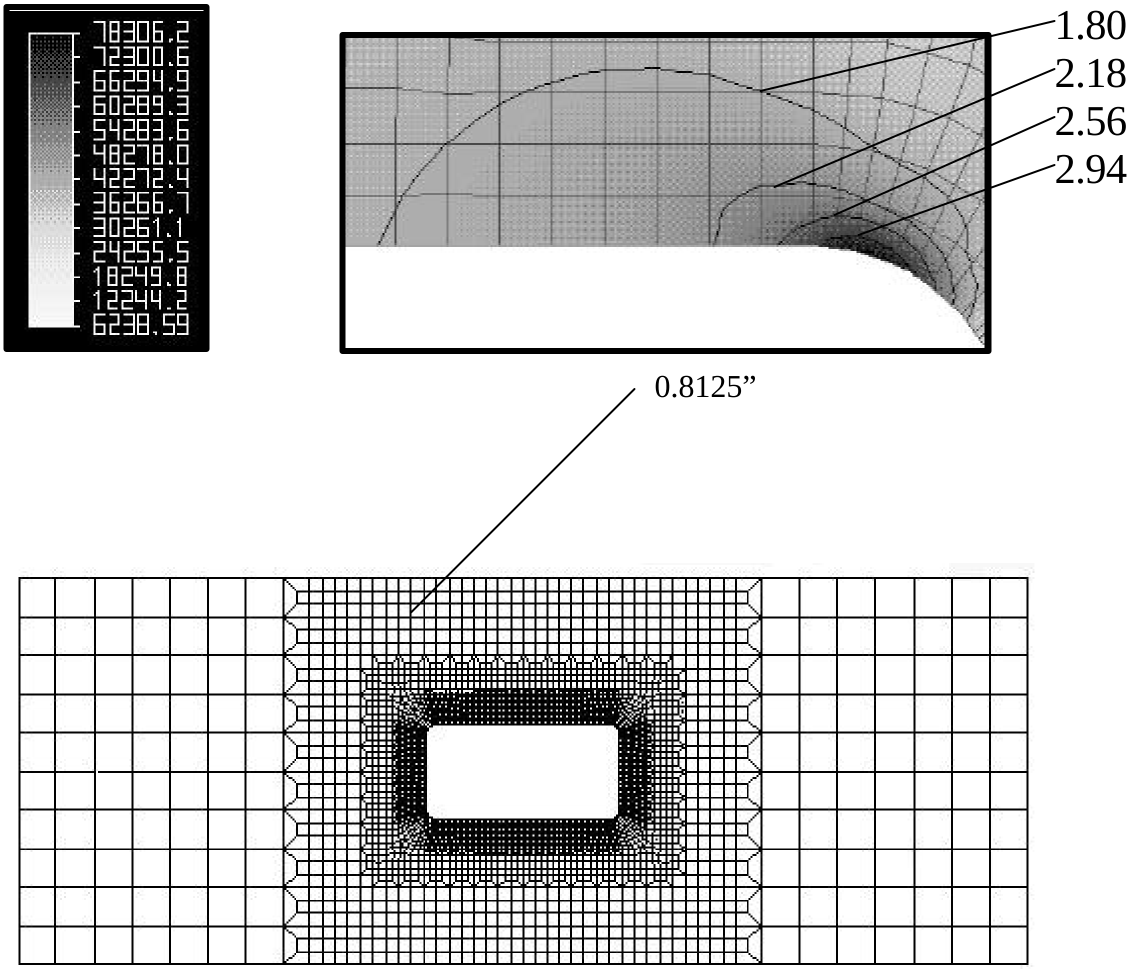
CASE 4: 1X2 Central Hole

Filename:	SSC6	Applied Stress:	23.5 ksi
Minimum Stress:	6.24 ksi	Maximum Stress:	78.3 ksi
BSCF (k_{bo})	3.327	FWSCF (k_{b1})	2.495



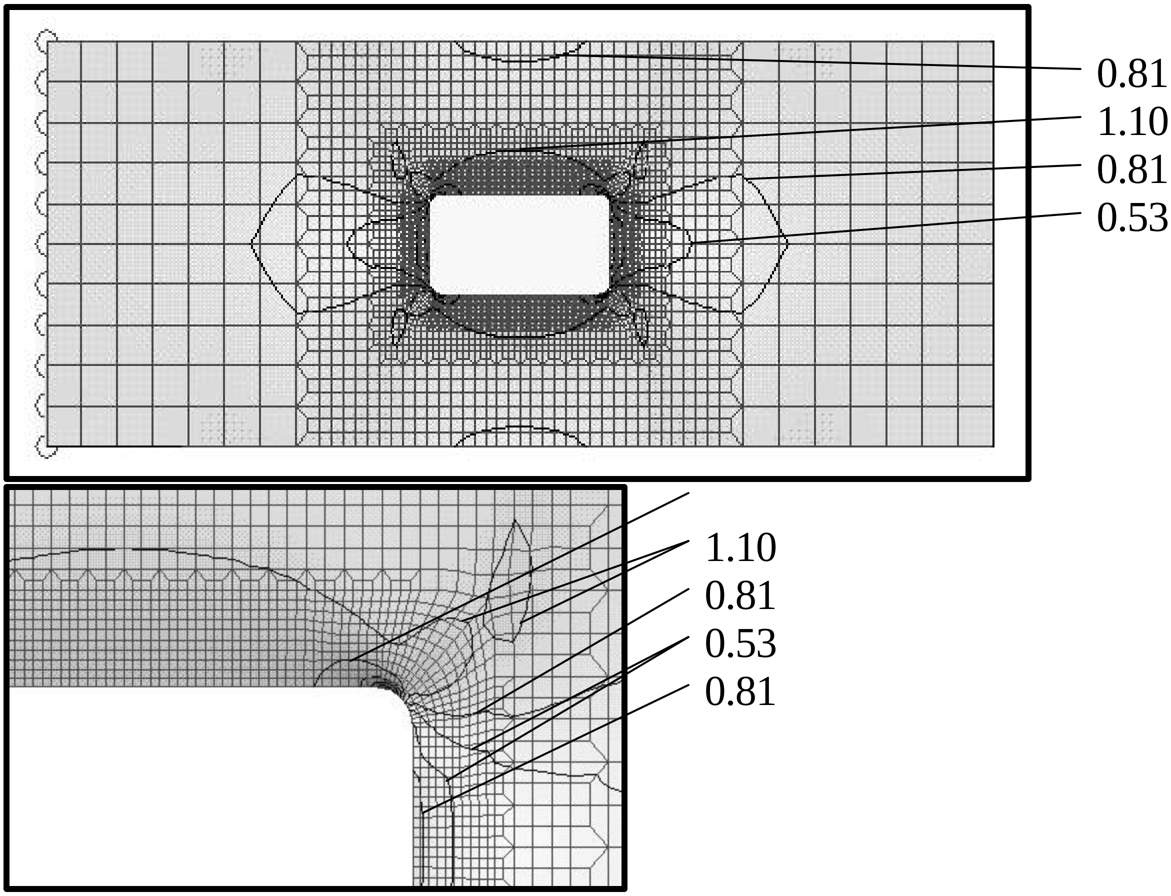


CASE 4: 1X2 Central Hole

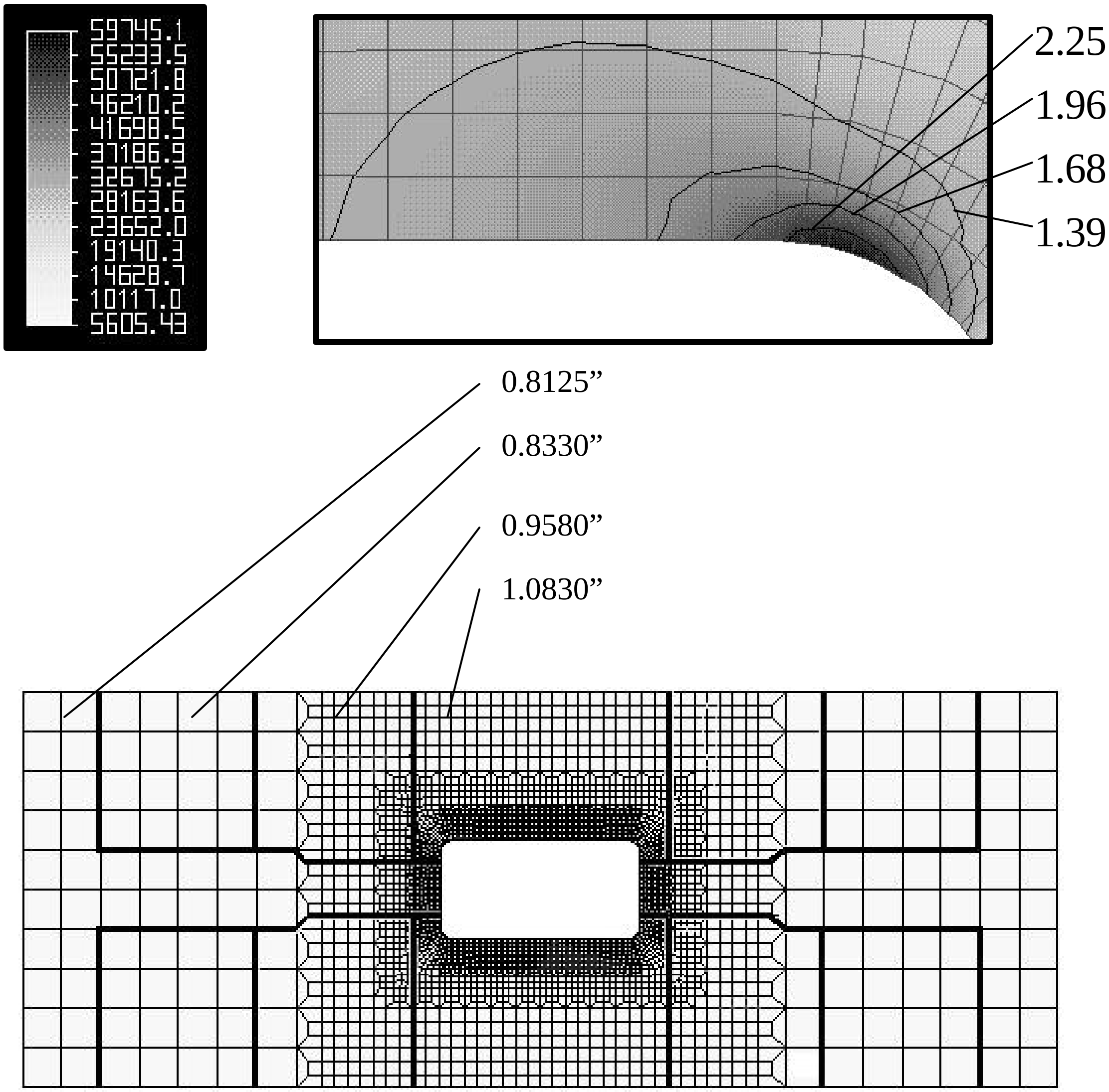


CASE 4: 1X2 Central Hole

Filename:	SSC6b	Applied Stress:	23.5 ksi
Minimum Stress:	5.61 ksi	Maximum Stress:	59.7 ksi
BSCF (k_{bo})	2.538	FWSCF (k_{b1})	2.537

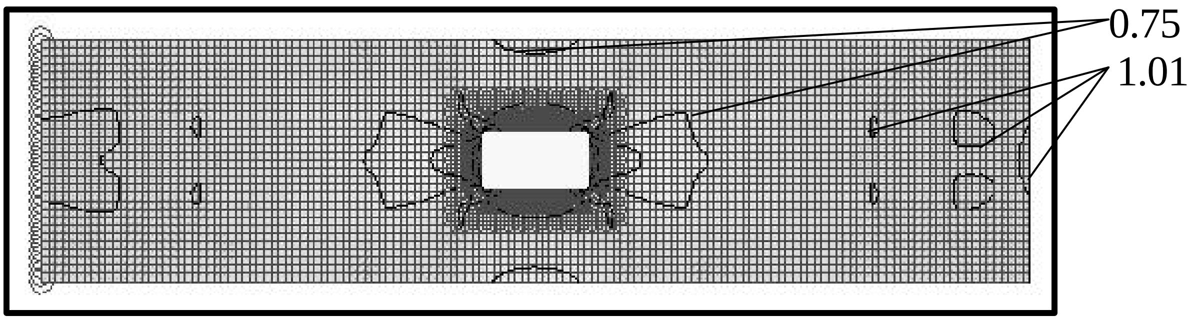


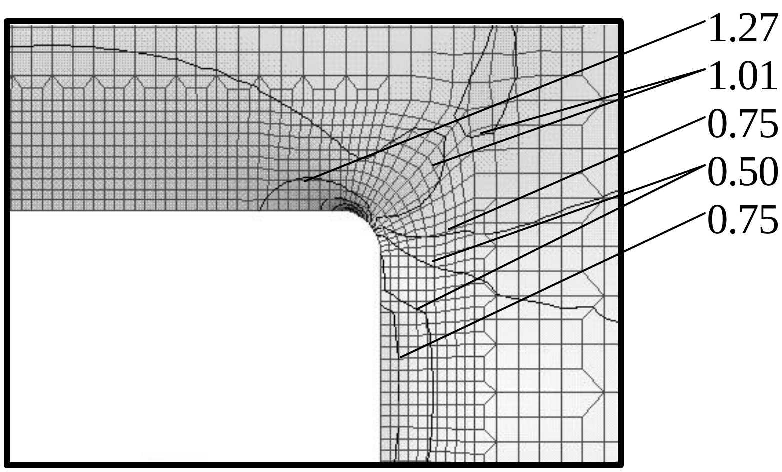
CASE 4: 1X2 Central Hole



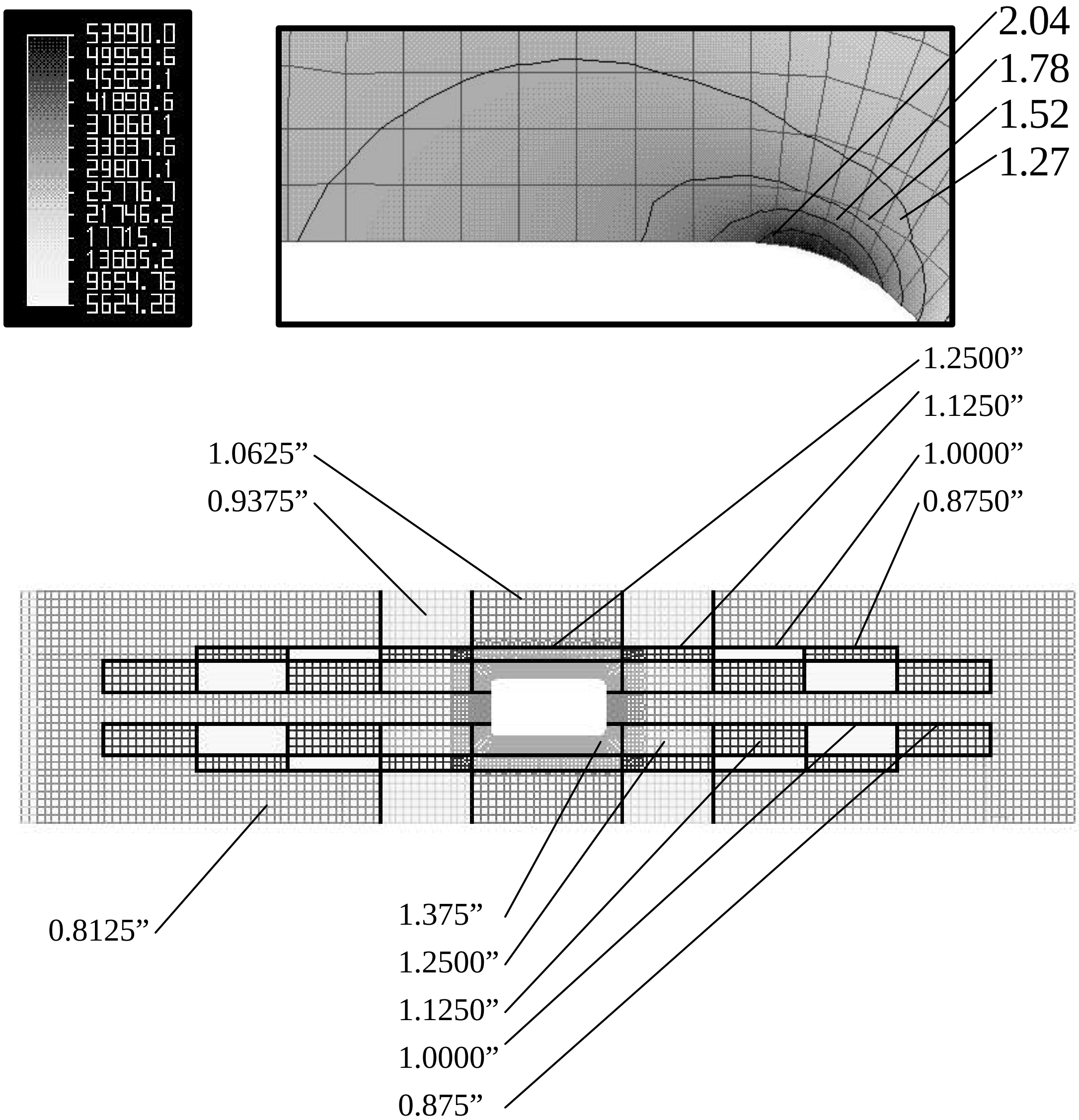
CASE 4: 1X2 Central Hole

Filename:	SSC6c	Applied Stress:	23.5 ksi
Minimum Stress:	5.62 ksi	Maximum Stress:	54.0 ksi
BSCF (k_{bo})	2.294	FWSCF (k_{b1})	2.452



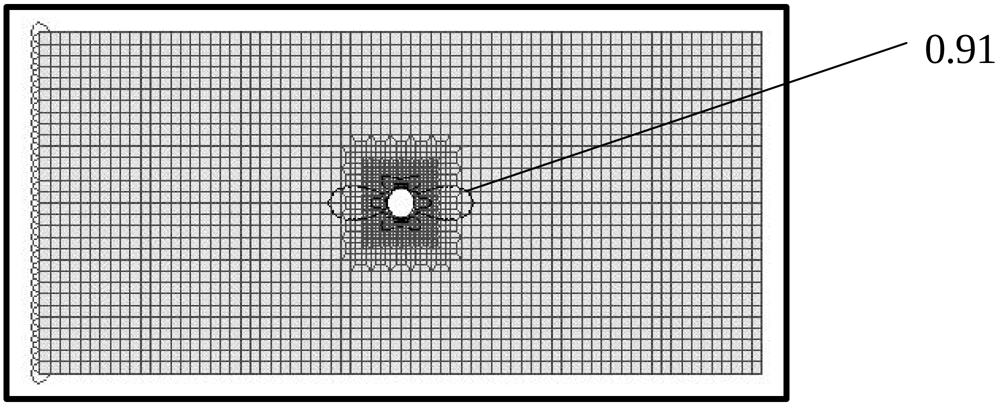


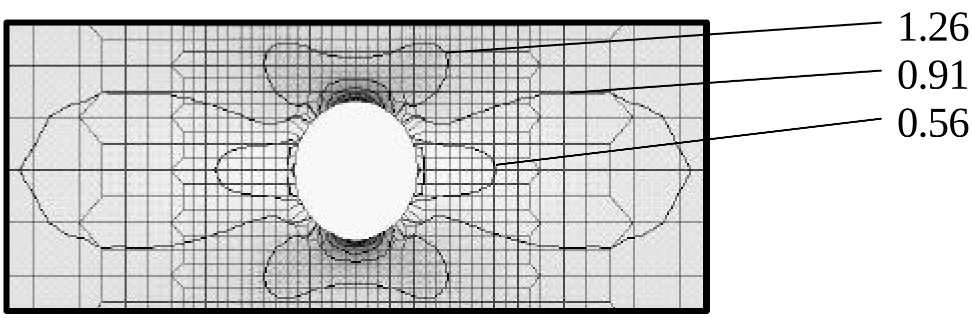
CASE 4: 1X2 Central Hole



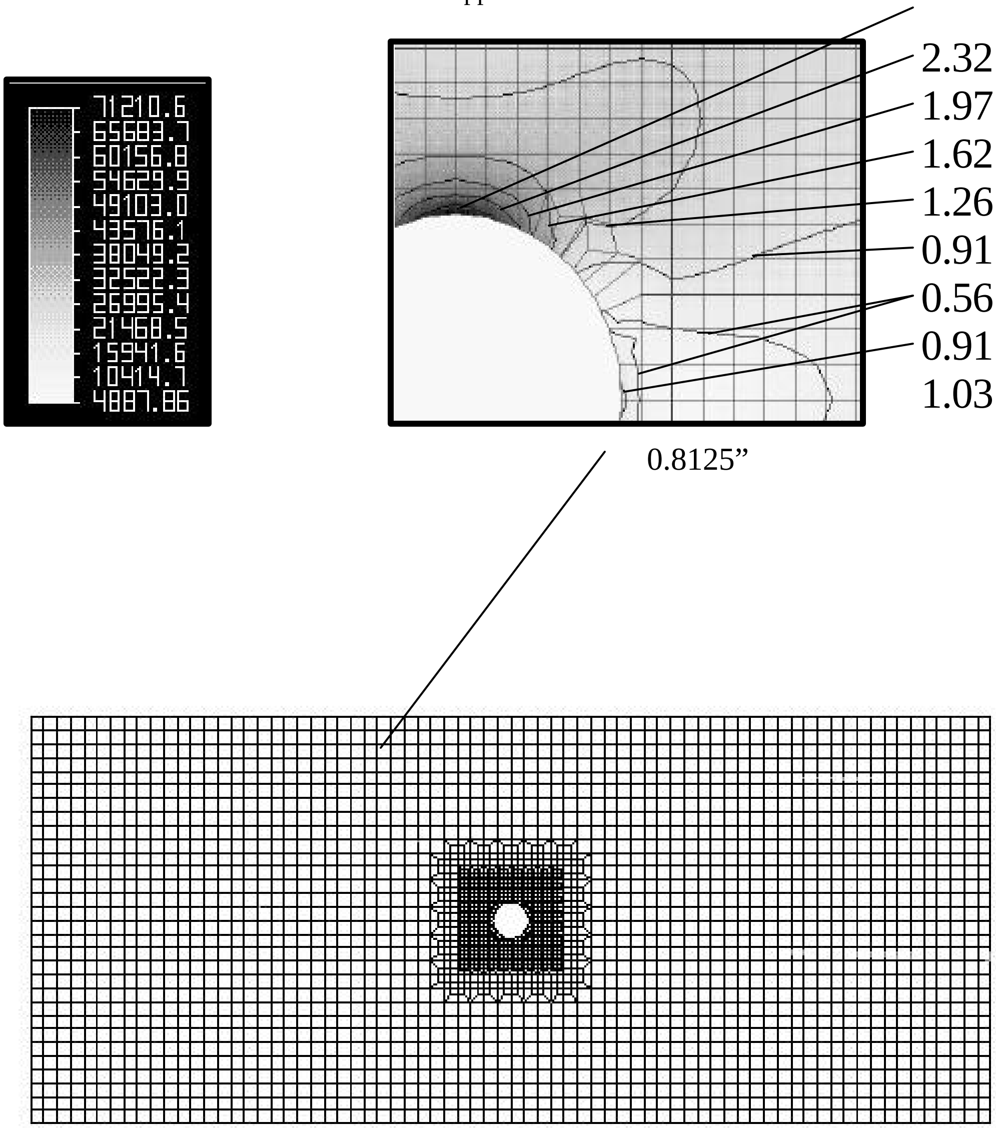
CASE 5: Small Round Central Hole and Zipper Effect

Filename:	SSC7	Applied Stress:	23.5 ksi
Minimum Stress:	4.89 ksi	Maximum Stress:	71.2 ksi
BSCF (k_{bo})	3.025	FWSCF (k_{b1})	2.758



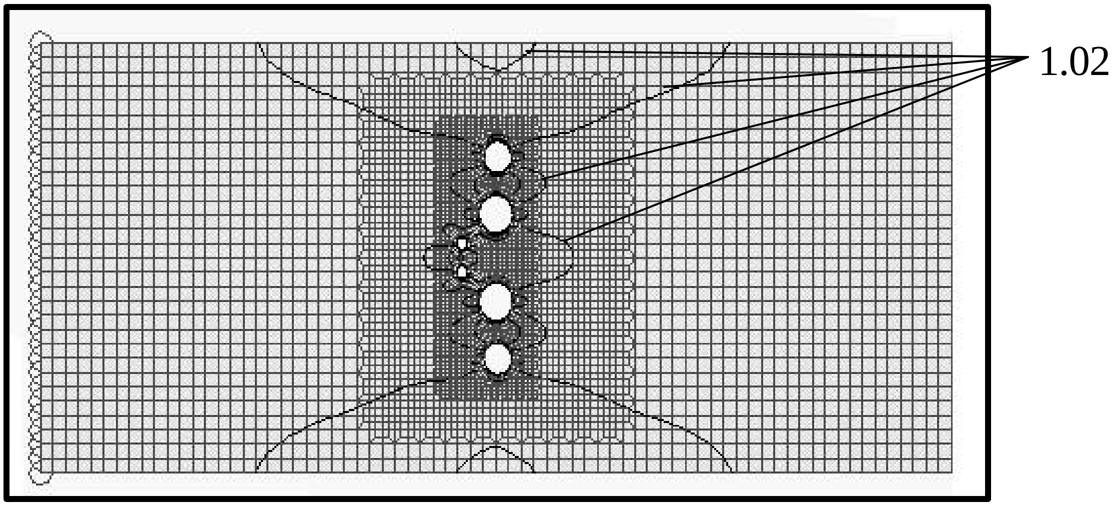


CASE 5: Small Round Central Hole and Zipper Effect

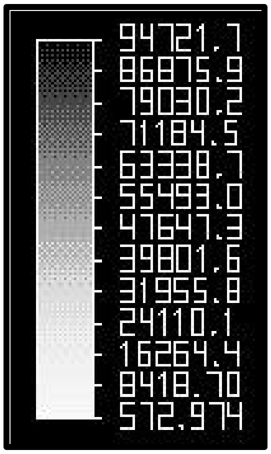


CASE 5: Small Round Central Hole and Zipper Effect

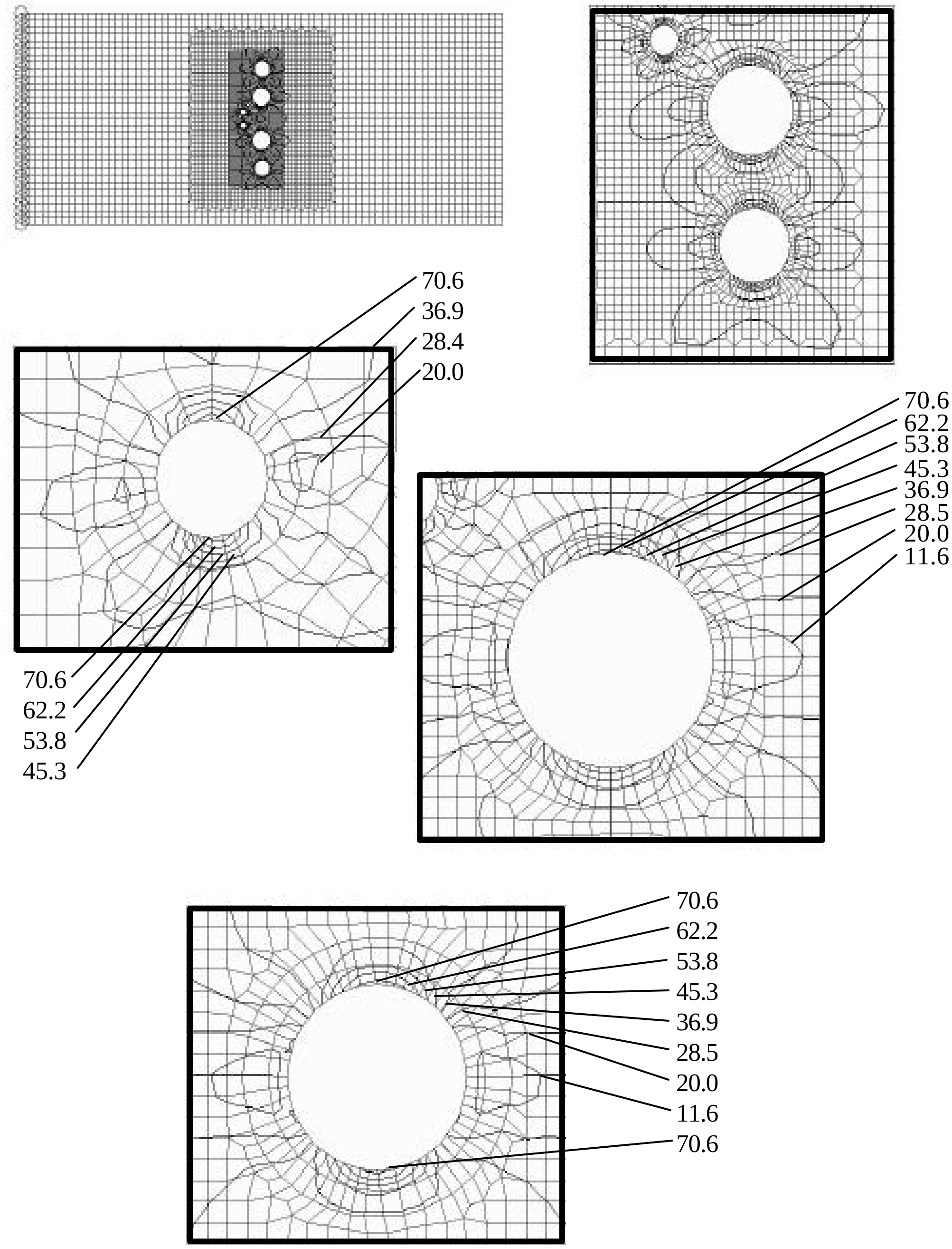
Filename:	SSC7b	Applied Stress:	23.5 ksi
Minimum Stress:	4.9 ksi	Maximum Stress:	79.0 ksi
BSCF (k_{bo})	3.36	FWSCF (k_{b1})	3.26



CASE 5: Small Round Central Hole and Zipper Effect

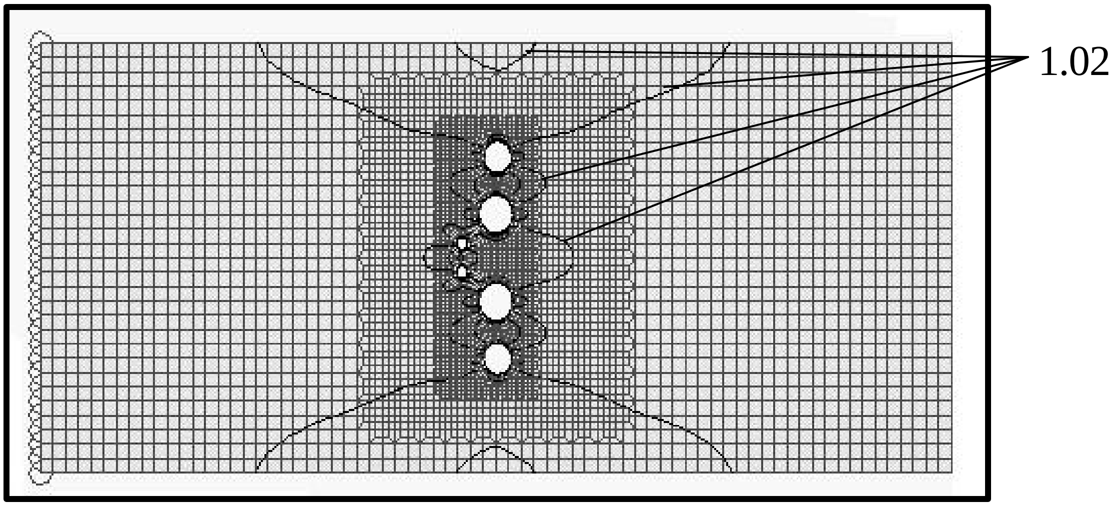


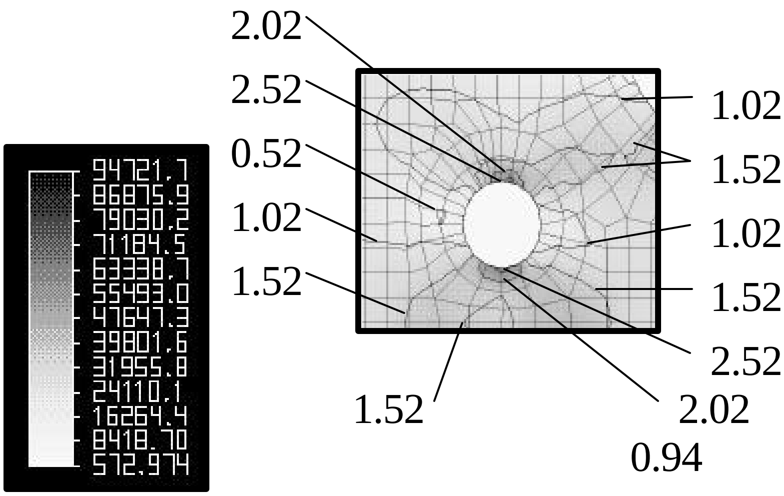
CASE 5: Small Round Central Hole and Zipper Effect

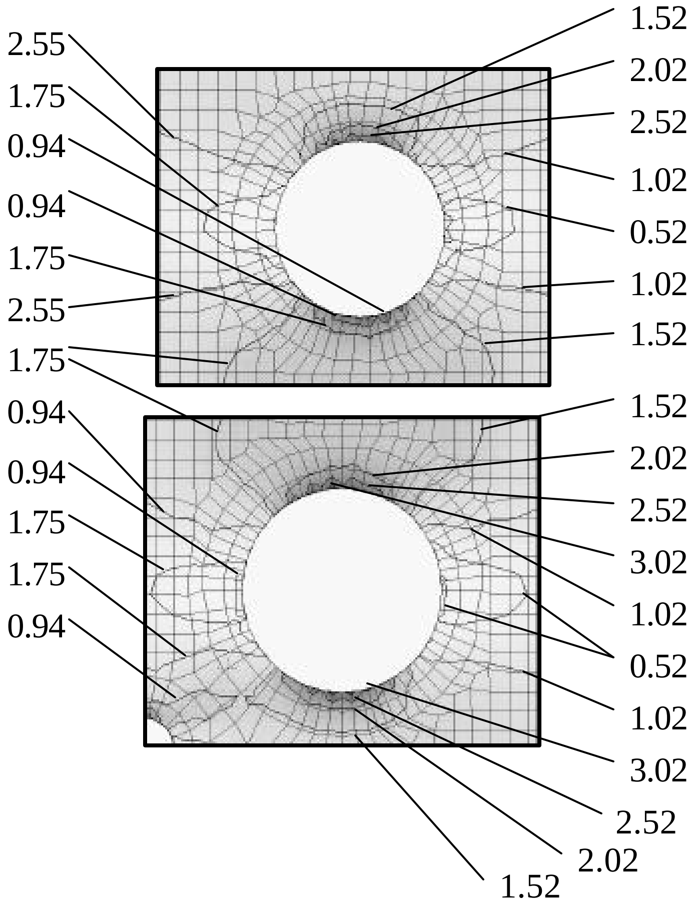


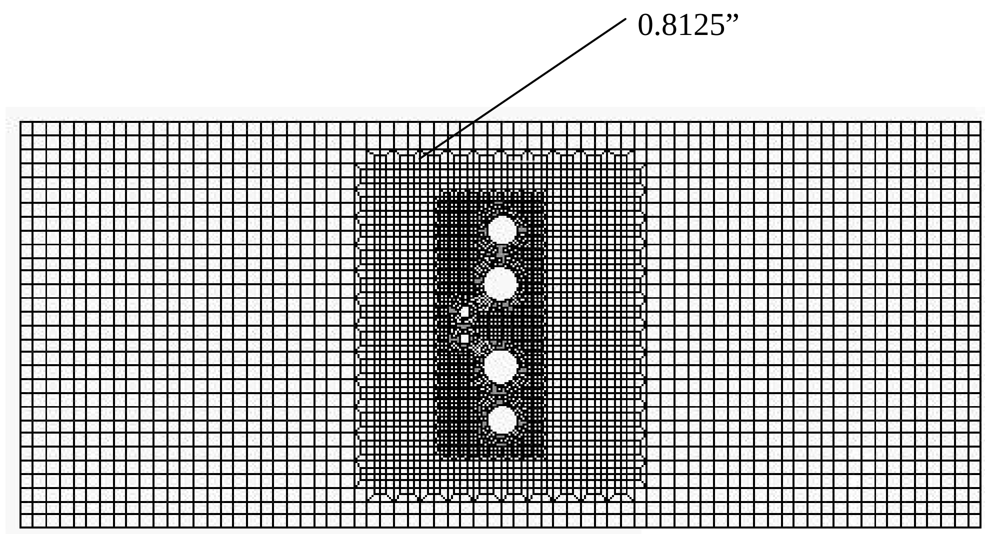
CASE 5: Small Round Central Hole and Zipper Effect

Filename:	SSC7c	Applied Stress:	23.5 ksi
Minimum Stress:	0.573 ksi	Maximum Stress:	94.7 ksi
BSCF (k_{bo})	4.024	FWSCF (k_{b1})	2.726

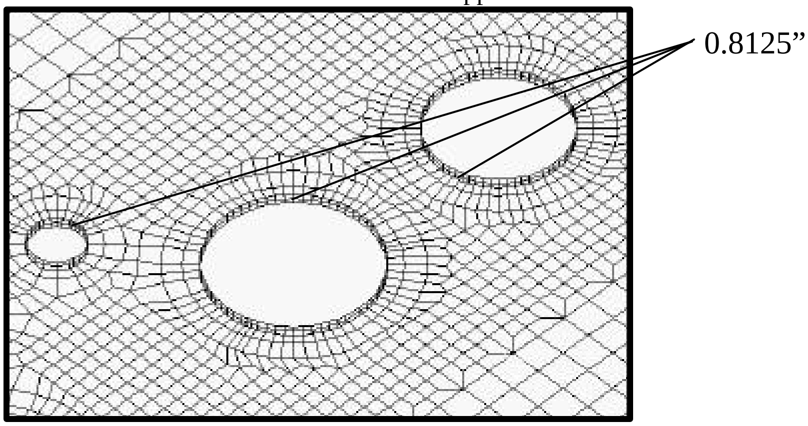





2.55
1.75
0.94
0.94
1.75
2.55
1.75
1.52
2.02
2.52
1.02
0.52
1.02
1.52
0.94
0.94
1.75
1.75
0.94
1.52
2.02
2.52
3.02
1.02
0.52
1.02
3.02
2.52
2.02
1.52

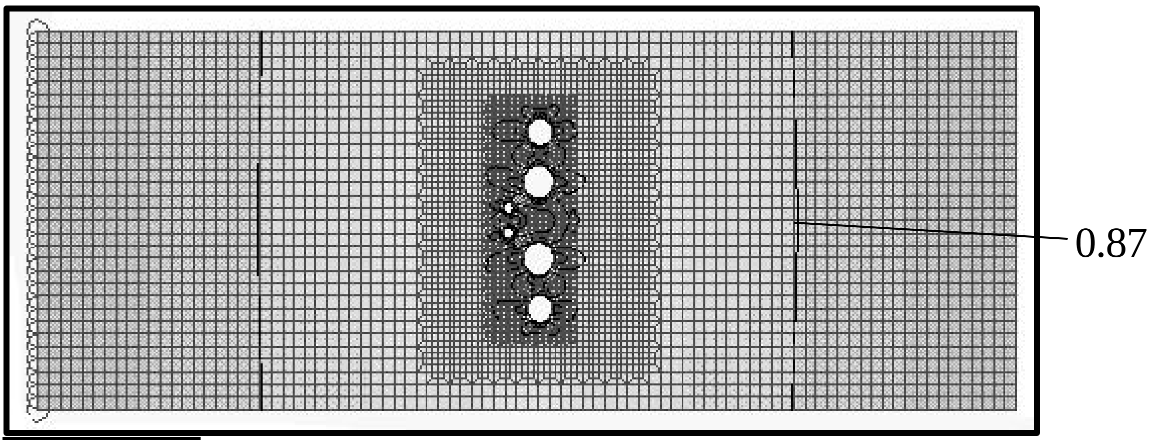

0.8125”

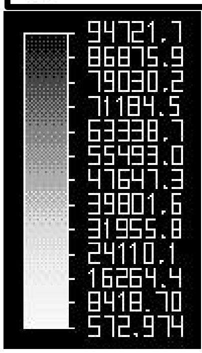
CASE 5: Small Round Central Hole and Zipper Effect

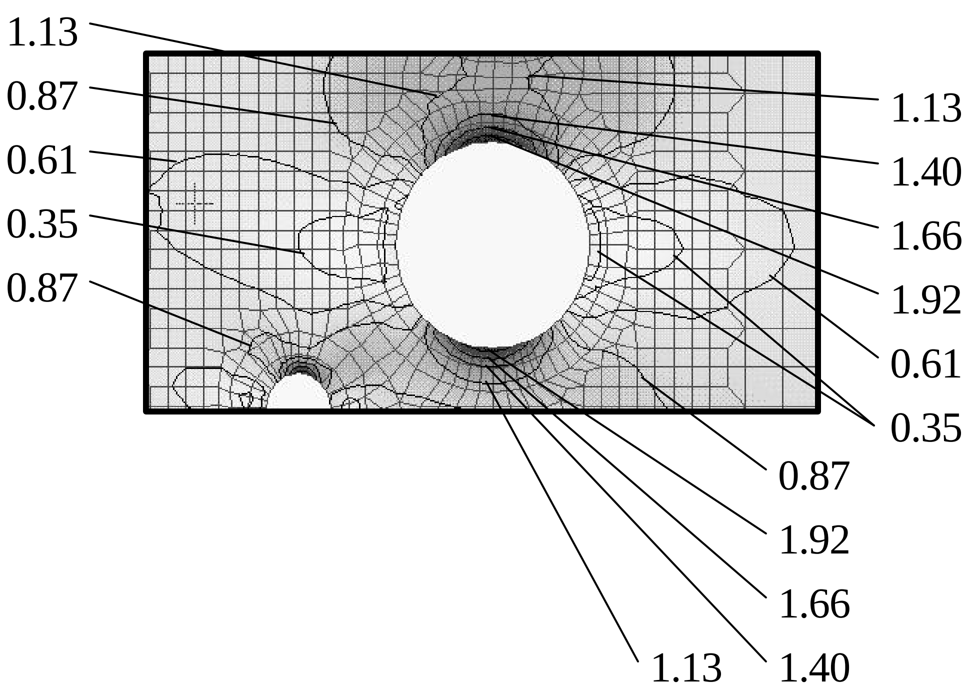


CASE 5: Small Round Central Hole and Zipper Effect

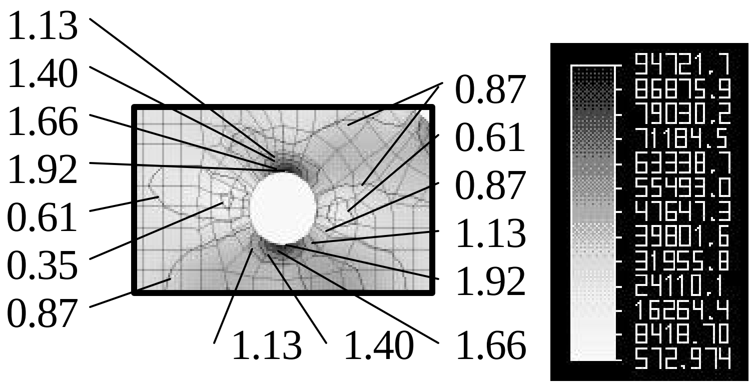
Filename:	SSC7d	Applied Stress:	23.5 ksi
Minimum Stress:	2.08ksi	Maximum Stress:	51.3 ksi
BSCF (k_{bo})	2.180	FWSCF (k_{b1})	2.271

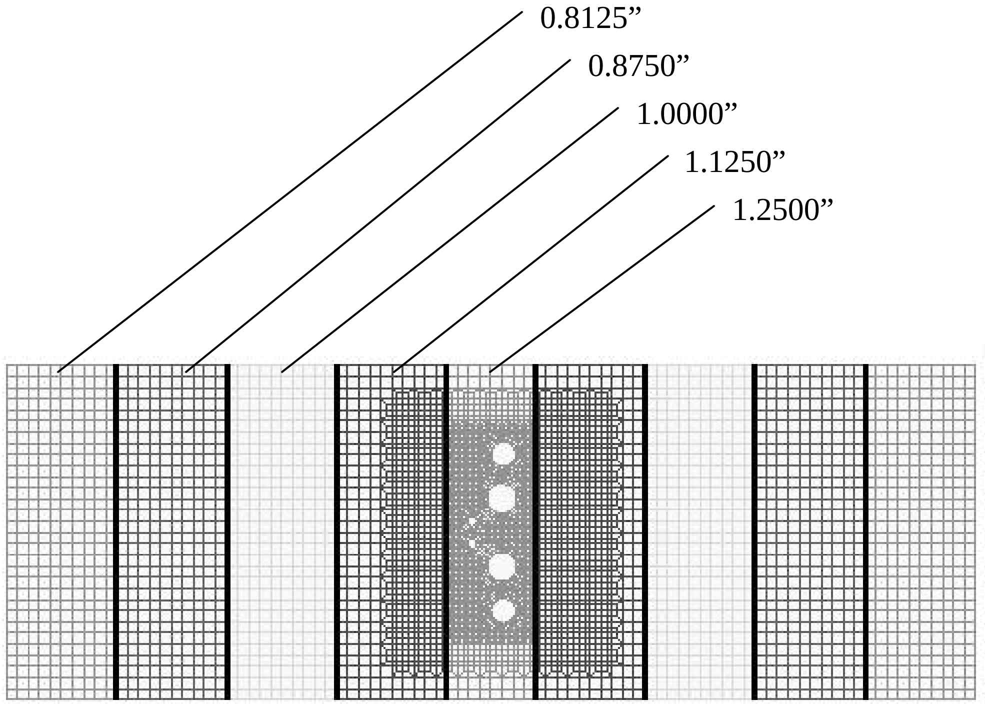






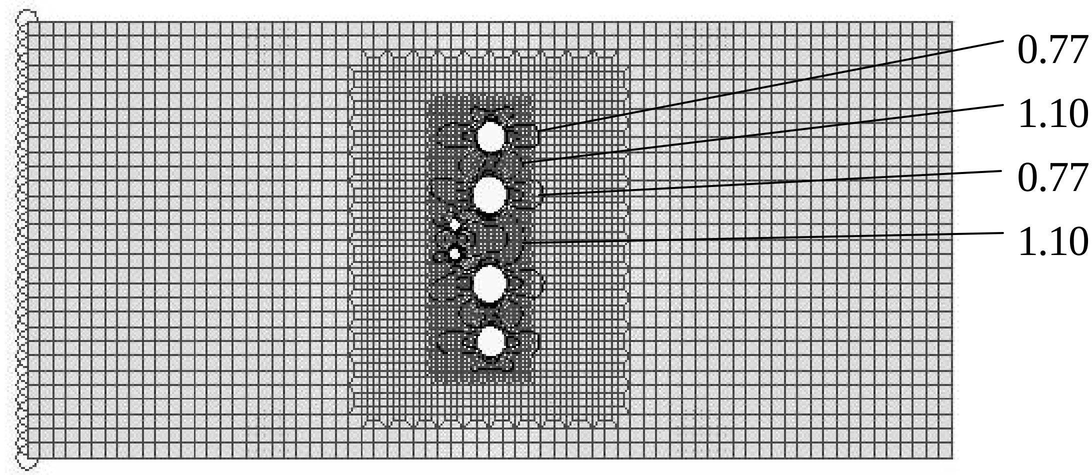
CASE 5: Small Round Central Hole and Zipper Effect

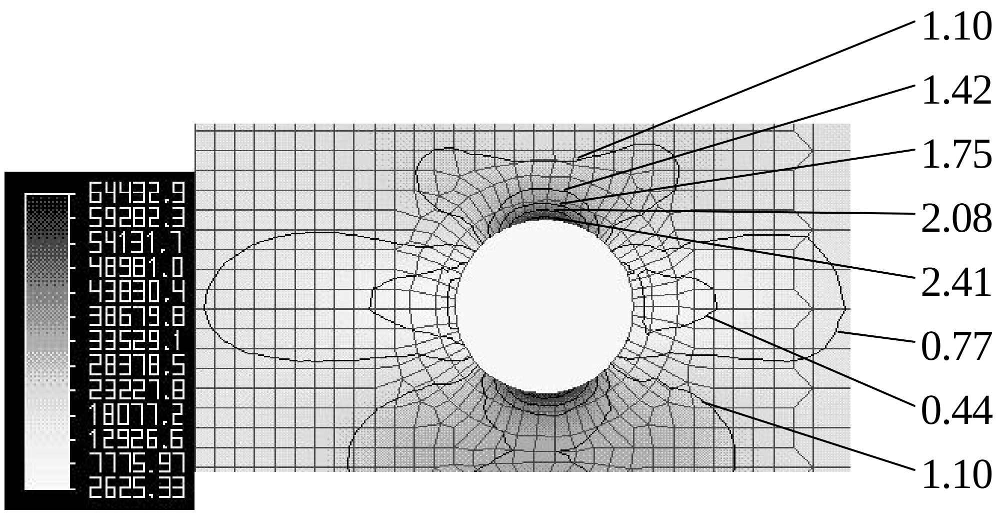




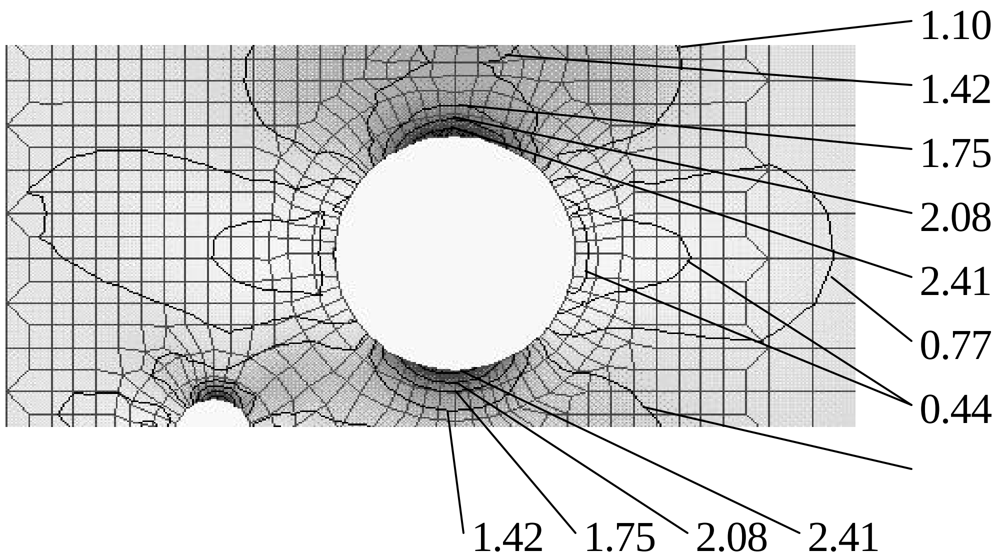
CASE 5: Small Round Central Hole and Zipper Effect

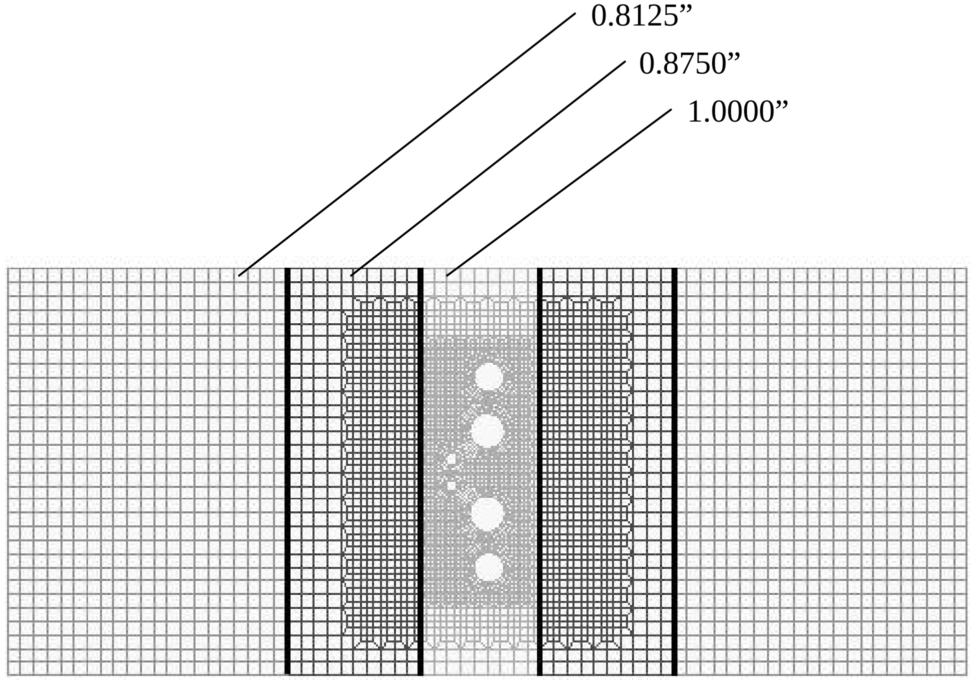
Filename:	SSC7e	Applied Stress:	23.5 ksi
Minimum Stress:	2.63 ksi	Maximum Stress:	64.4 ksi
BSCF (k_{bo})	2.737	FWSCF (k_{b1})	2.282





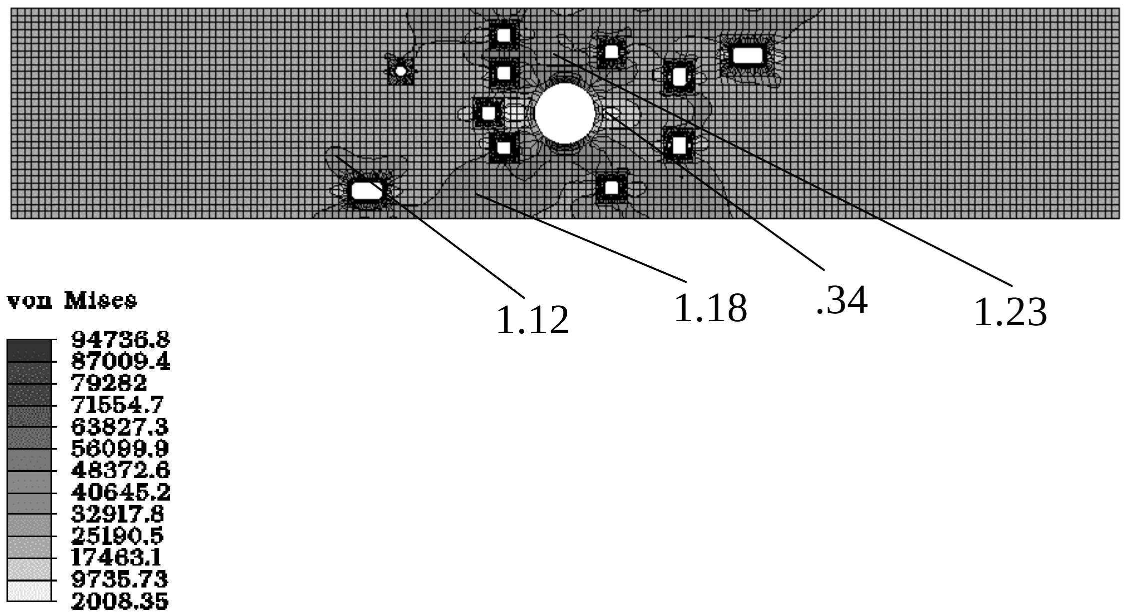
CASE 5: Small Round Central Hole and Zipper Effect

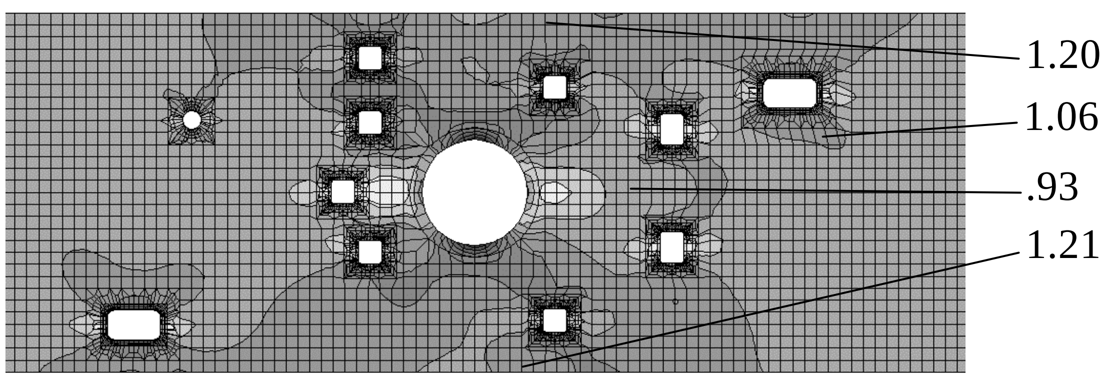




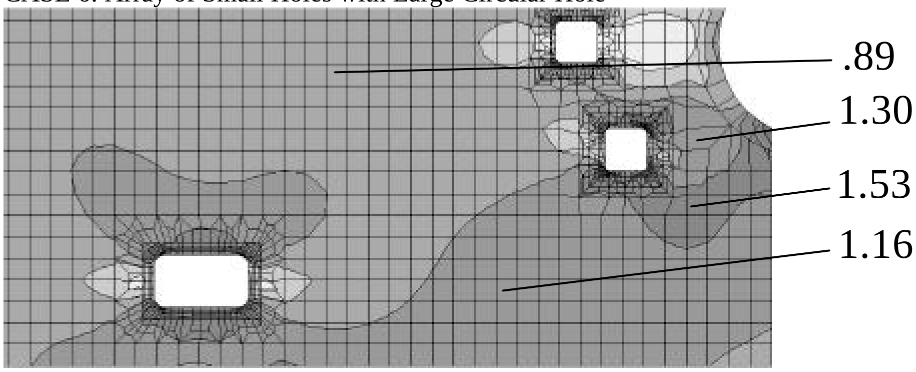
CASE 6: Array of Small Holes with Large Circular Hole

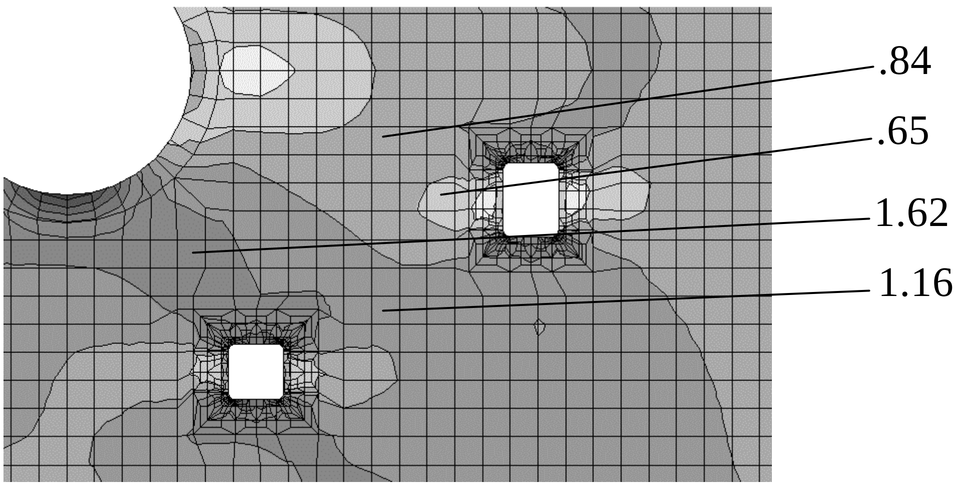
Filename:	SSC9a	Applied Stress:	23.5 ksi
Minimum Stress:	2.0 ksi	Maximum Stress:	94.7 ksi
BSCF (k_{bo})	4.03	FWSCF (k_{b1})	3.32

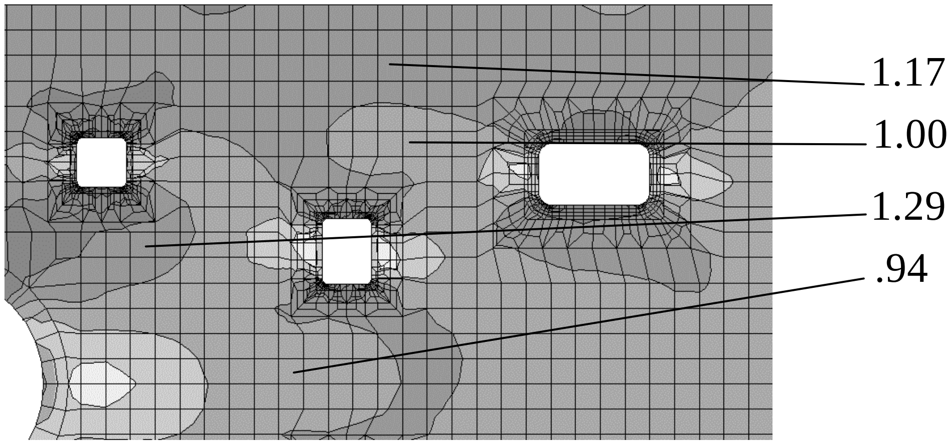


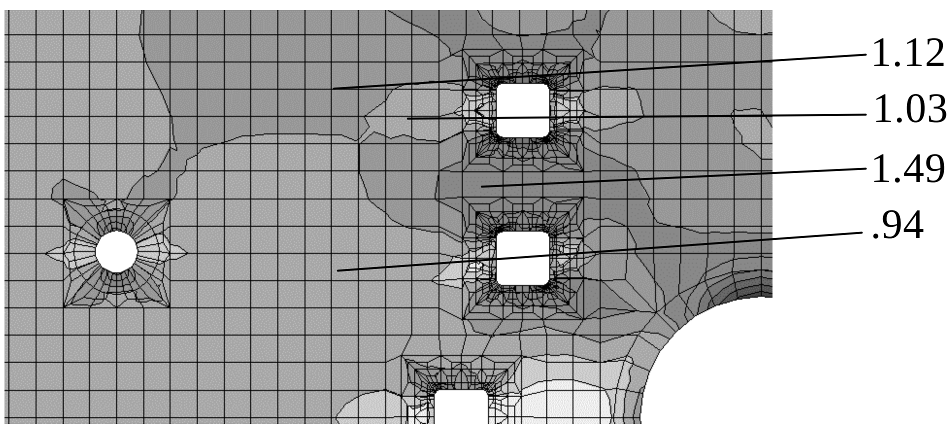


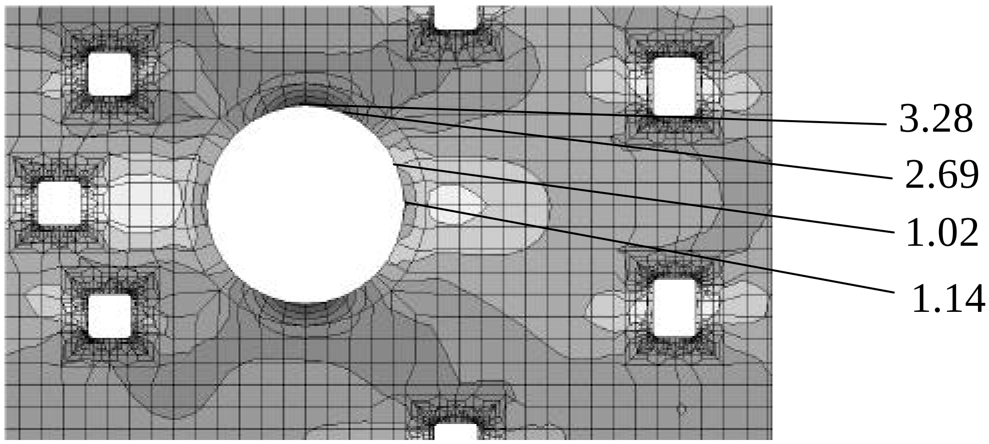
CASE 6: Array of Small Holes with Large Circular Hole

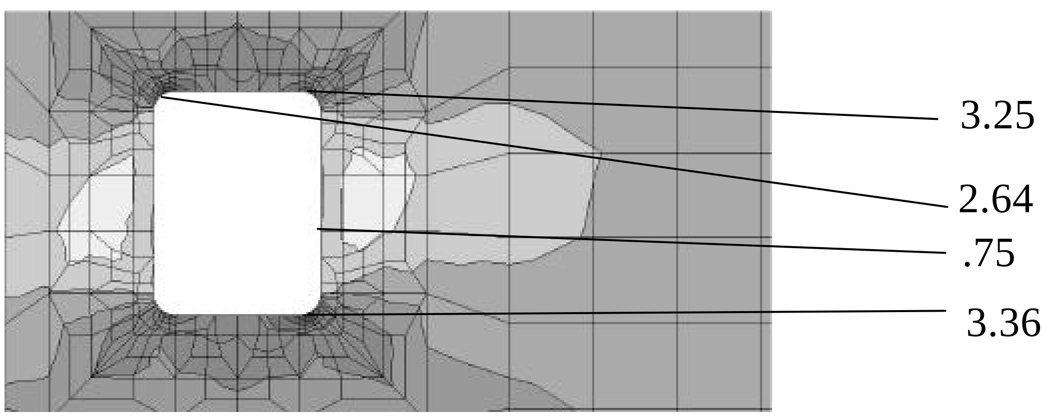


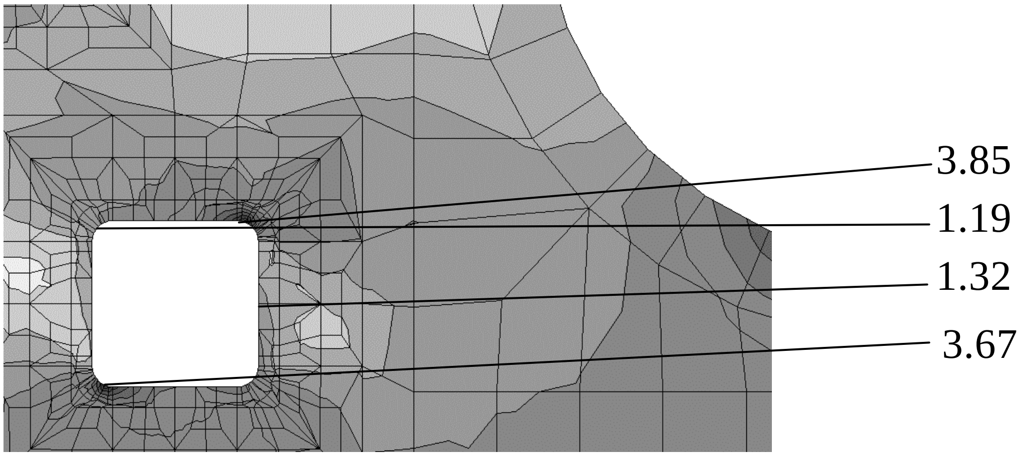


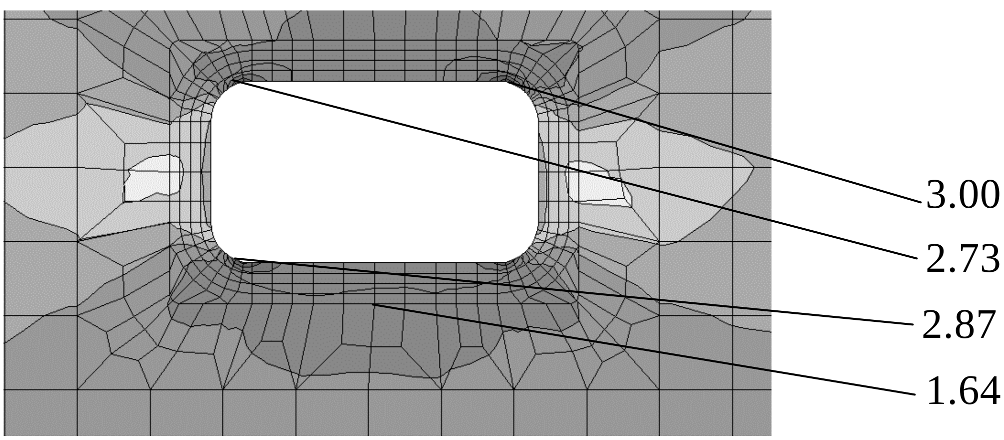

1.17
1.00
1.29
.94


1.12
1.03
1.49
.94

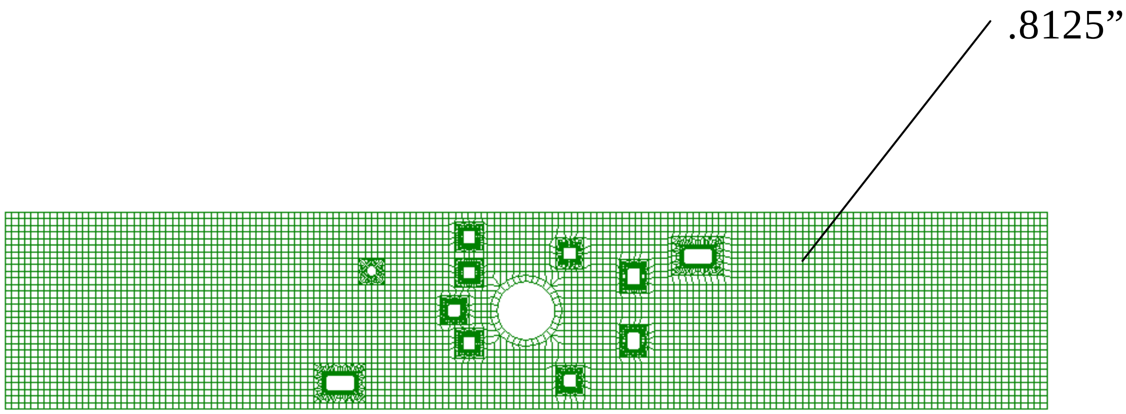

3.28
2.69
1.02
1.14


3.25
2.64
.75
3.36


3.85
1.19
1.32
3.67

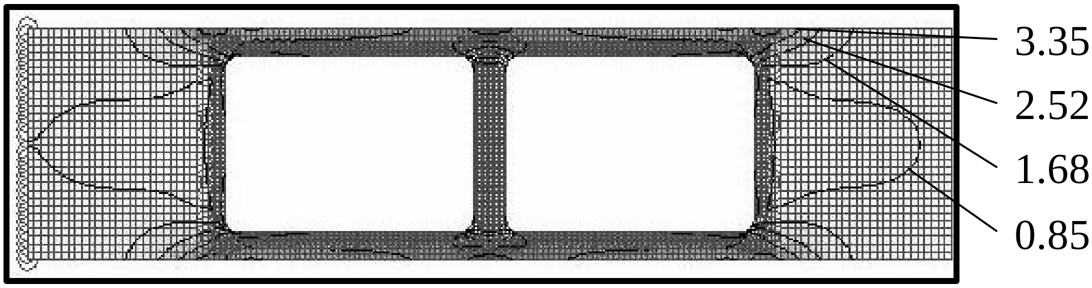

3.00
2.73
2.87
1.64

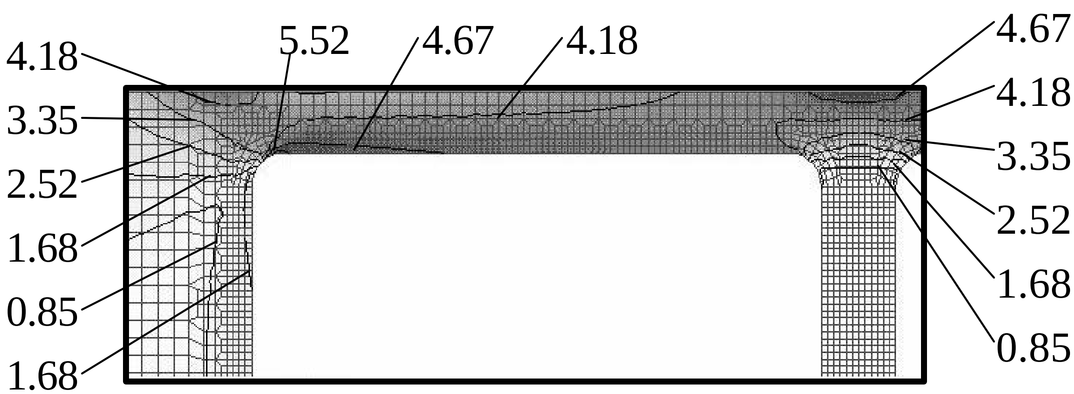
CASE 6: Array of Small Holes with Large Circular Hole

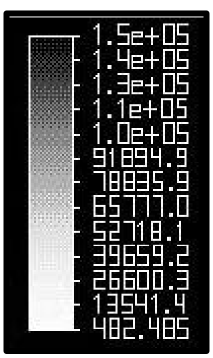


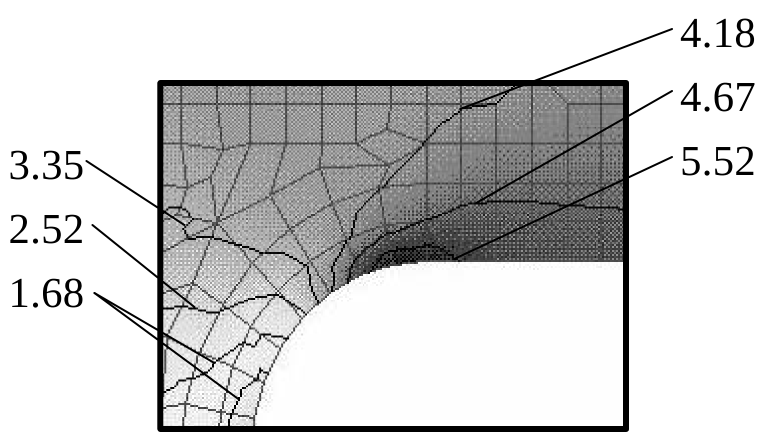
CASE 7: 2 Large Bays

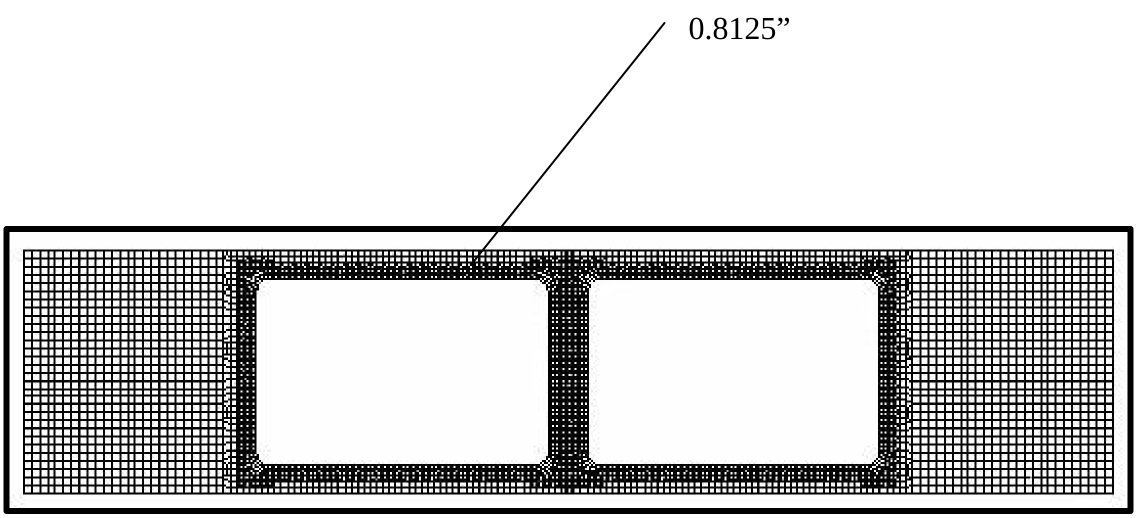
Filename:	SSC11	Applied Stress:	23.5 ksi
Minimum Stress:	0.483 ksi	Maximum Stress:	150 ksi
BSCF (k_{bo})	6.372	FWSCF (k_{b1})	1.478





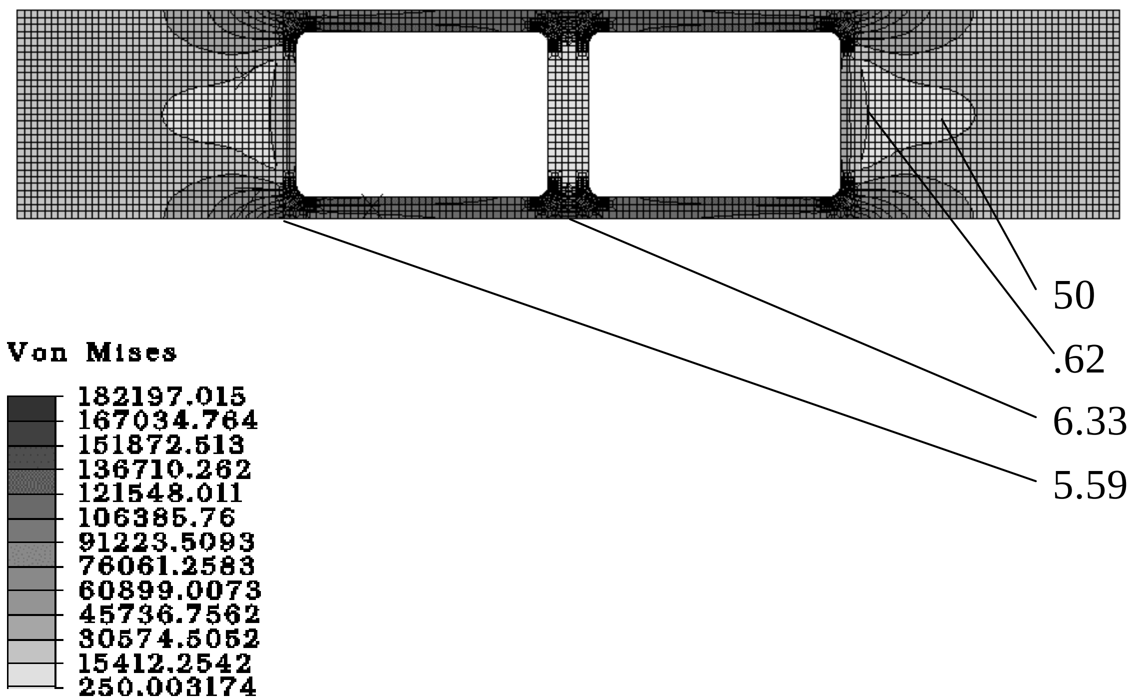




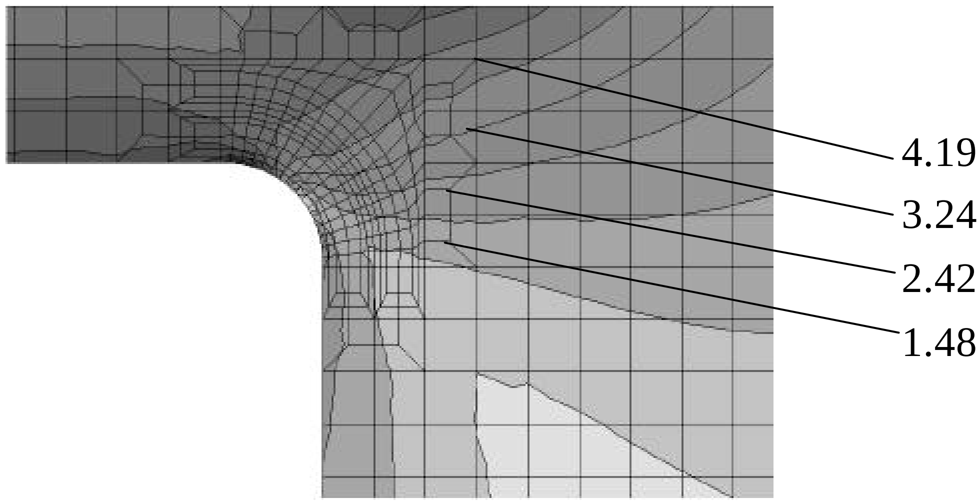


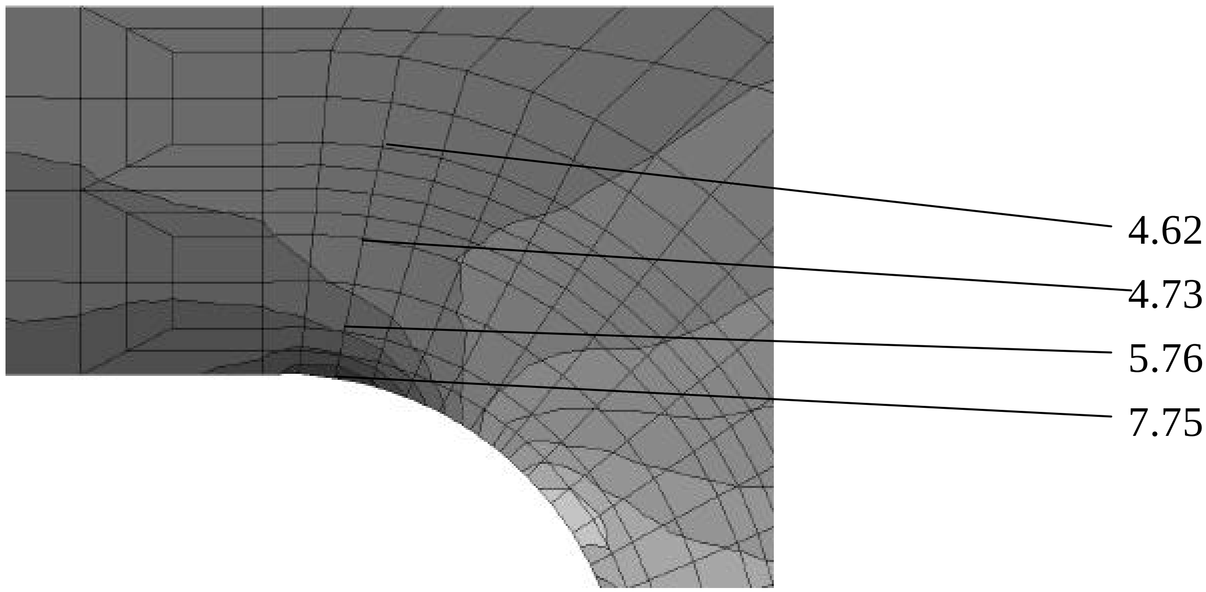
CASE 7: 2 Large Bays

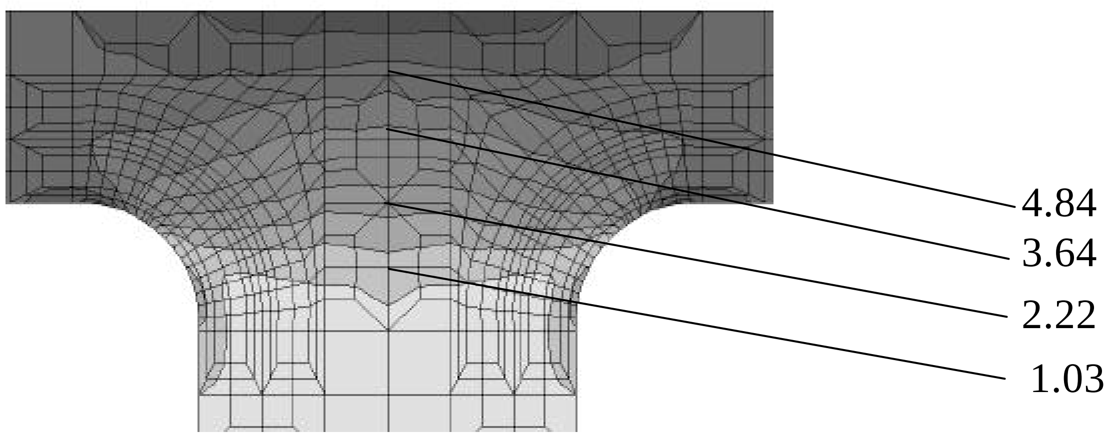
Filename:	SSC11a	Applied Stress:	23.5 ksi
Minimum Stress:	.3 ksi	Maximum Stress:	182.2 ksi
BSCF (k_{bo})	7.75	FWSCF (k_{b1})	1.55

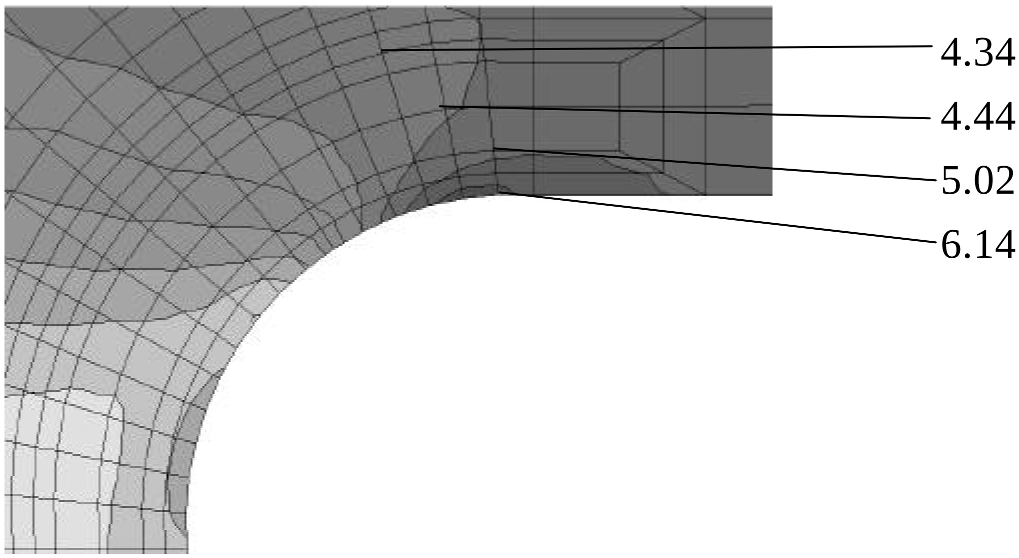


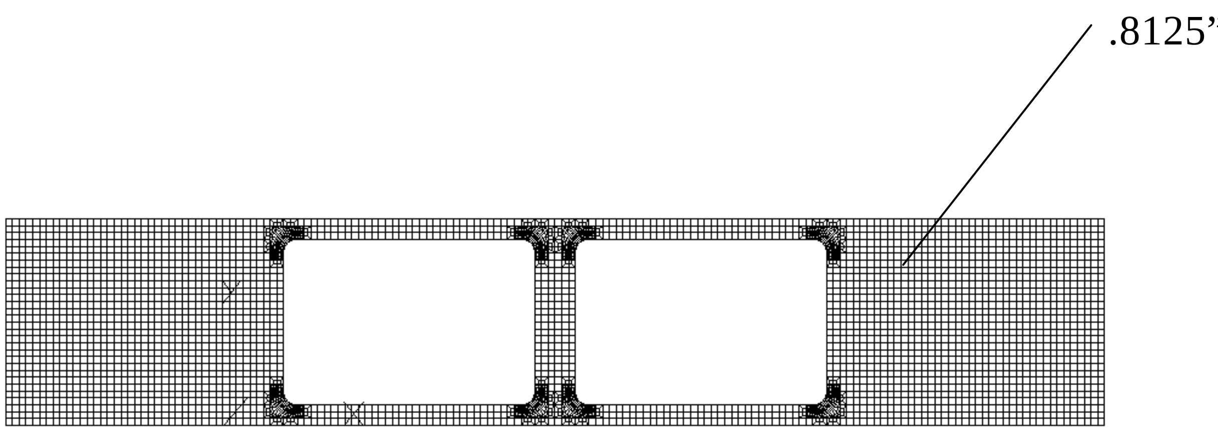
CASE 7: 2 Large Bays



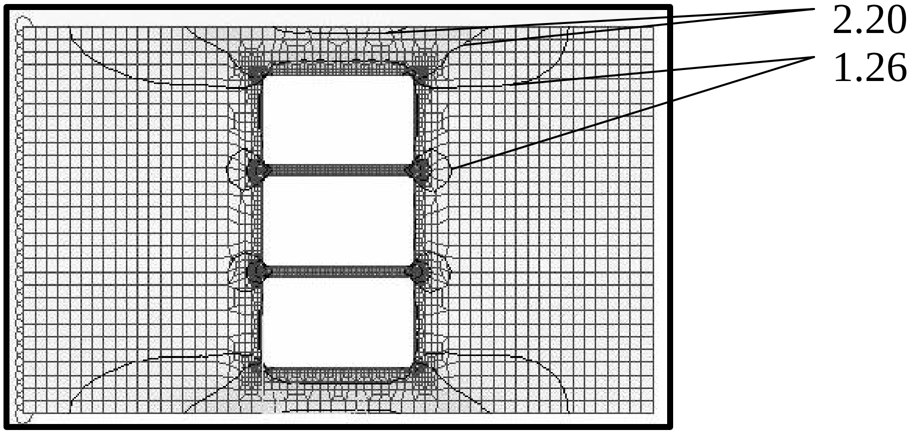

4.62
4.73
5.76
7.75

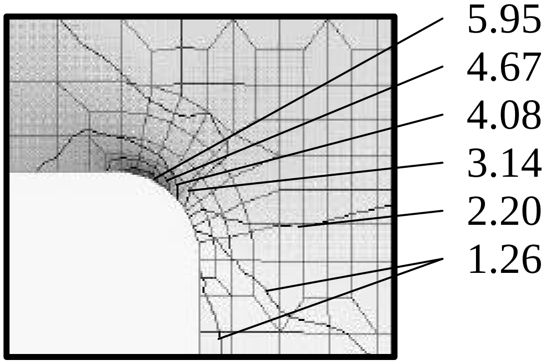

4.84
3.64
2.22
1.03


4.34
4.44
5.02
6.14

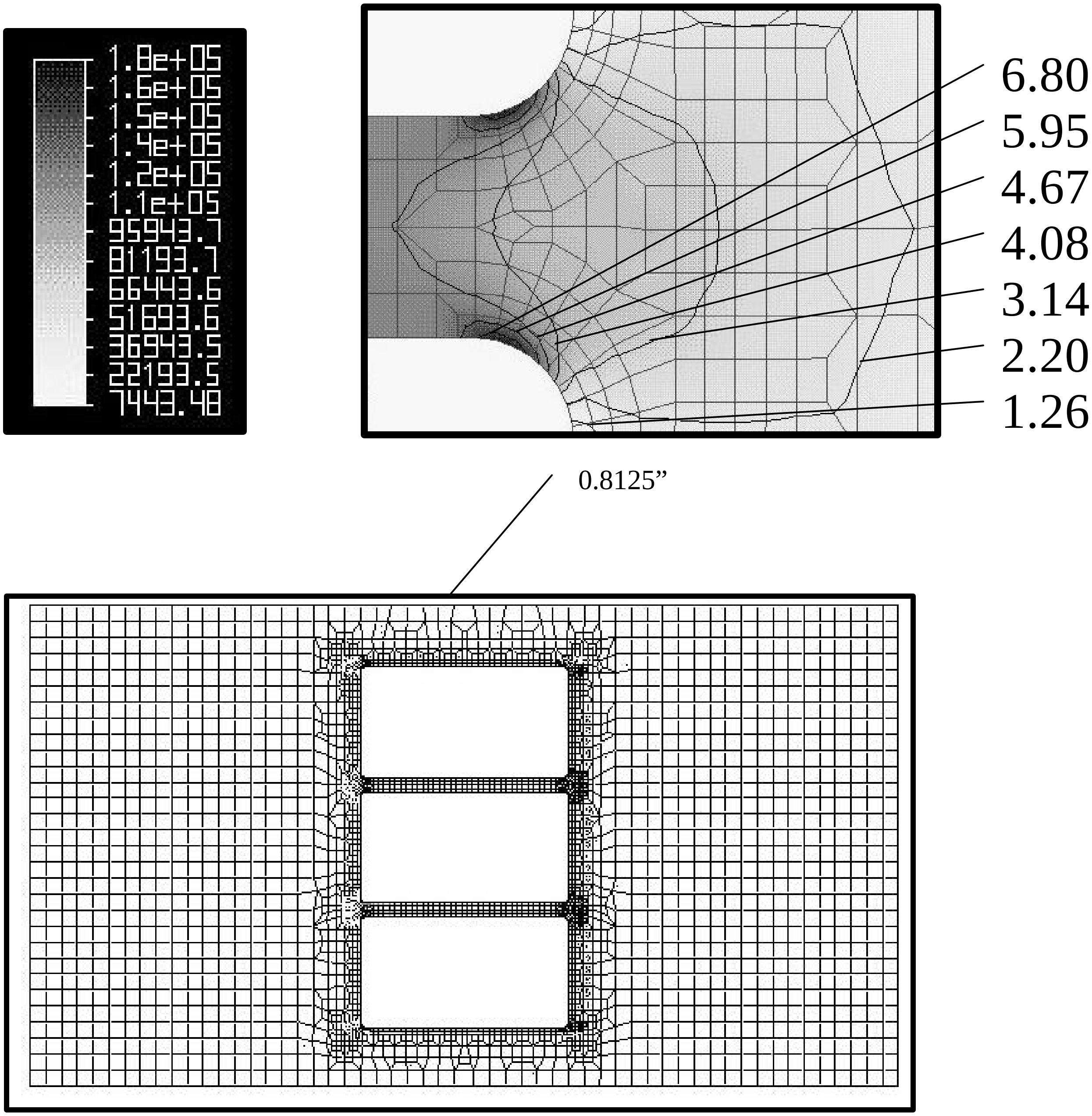

.8125”

Filename:	SSC12	Applied Stress:	23.5 ksi
Minimum Stress:	7.44 ksi	Maximum Stress:	180 ksi
BSCF (k_{bo})	7.647	FWSCF (k_{b1})	2.328

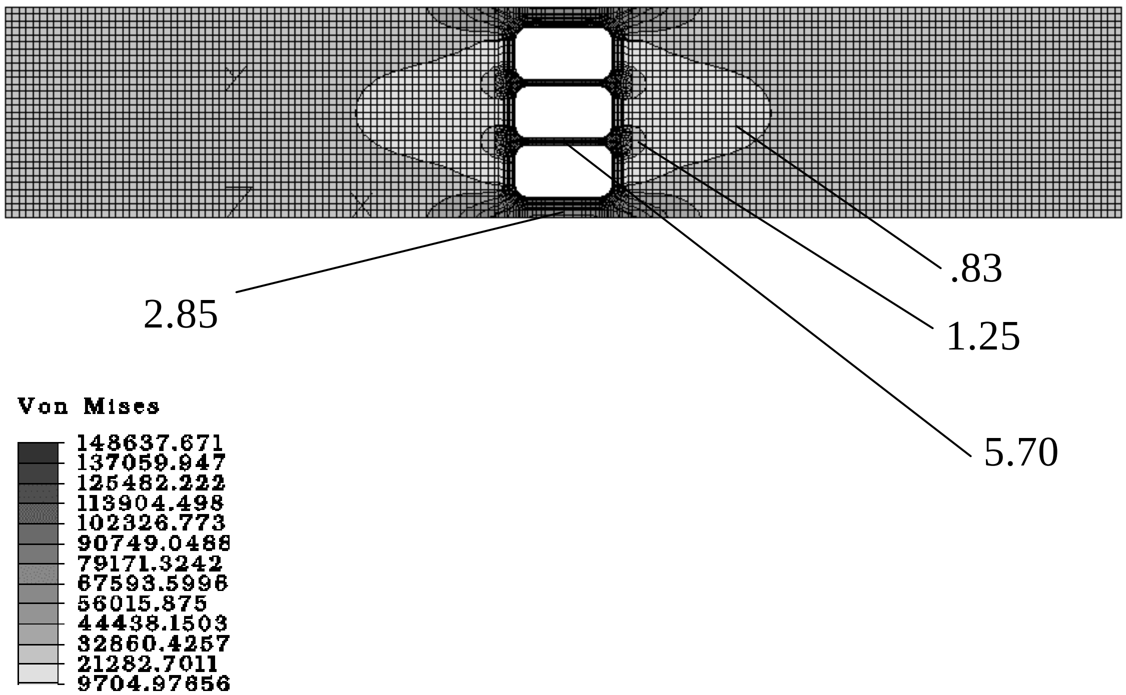




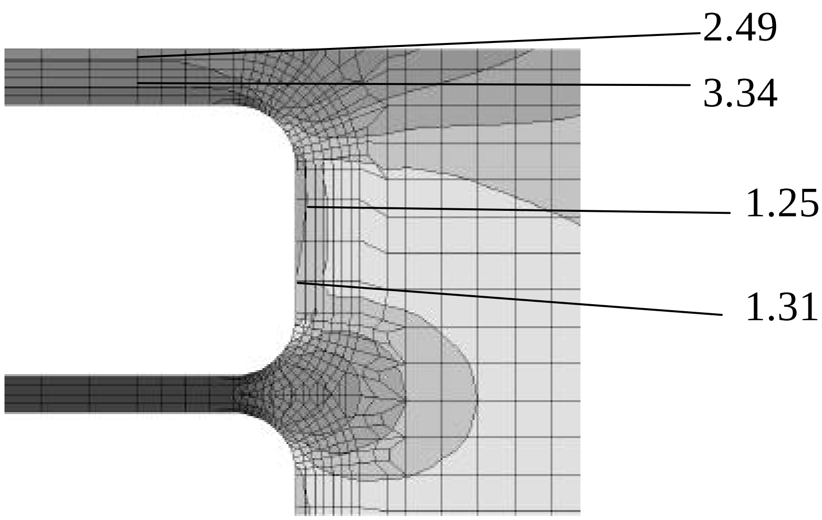
CASE 7: 2 Large Bays

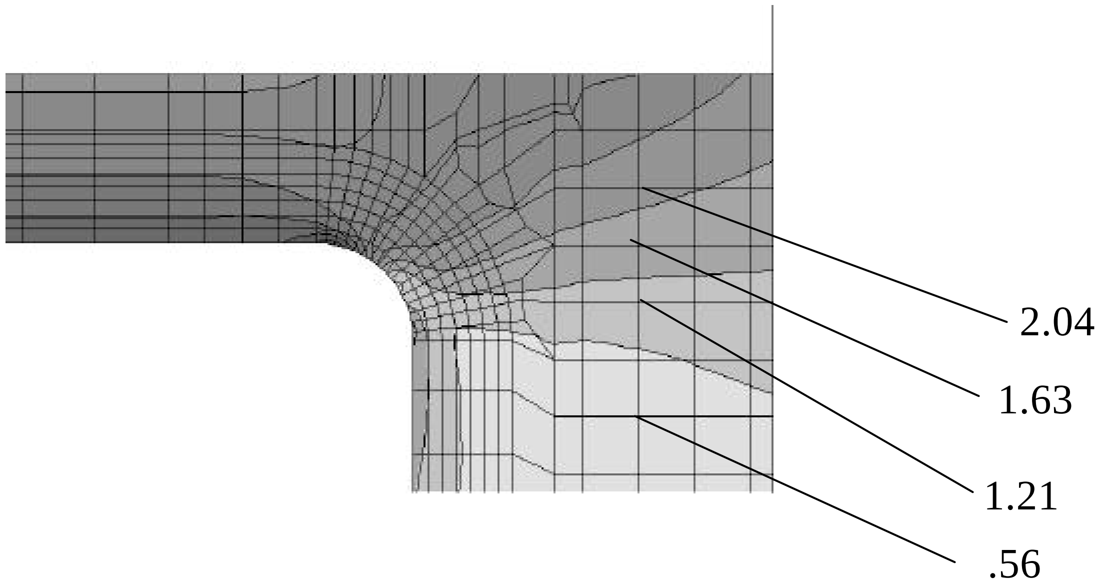


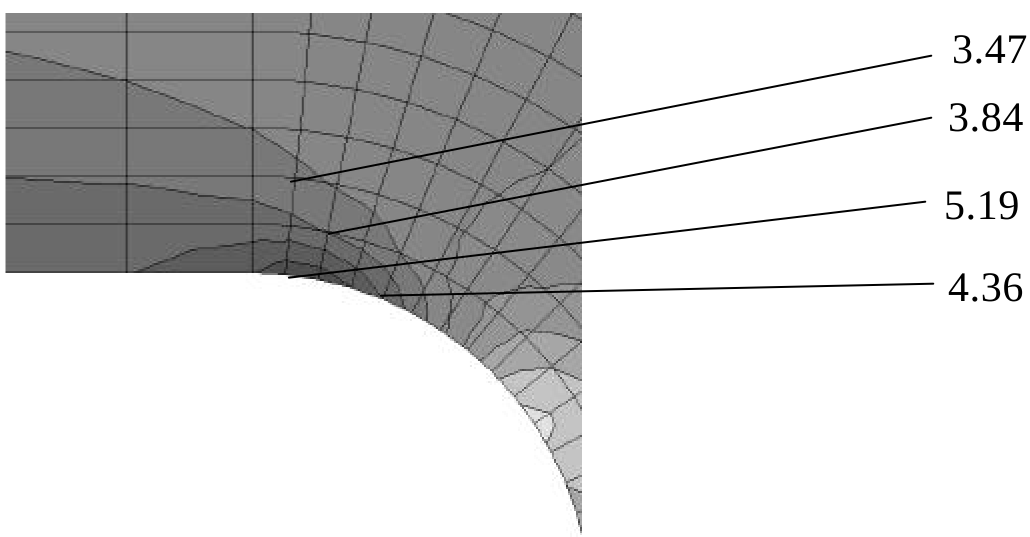
Filename:	SSC12a	Applied Stress:	23.5 ksi
Minimum Stress:	9.7 ksi	Maximum Stress:	148.6 ksi
BSCF (k_{bo})	6.32	FWSCF (k_{b1})	1.66



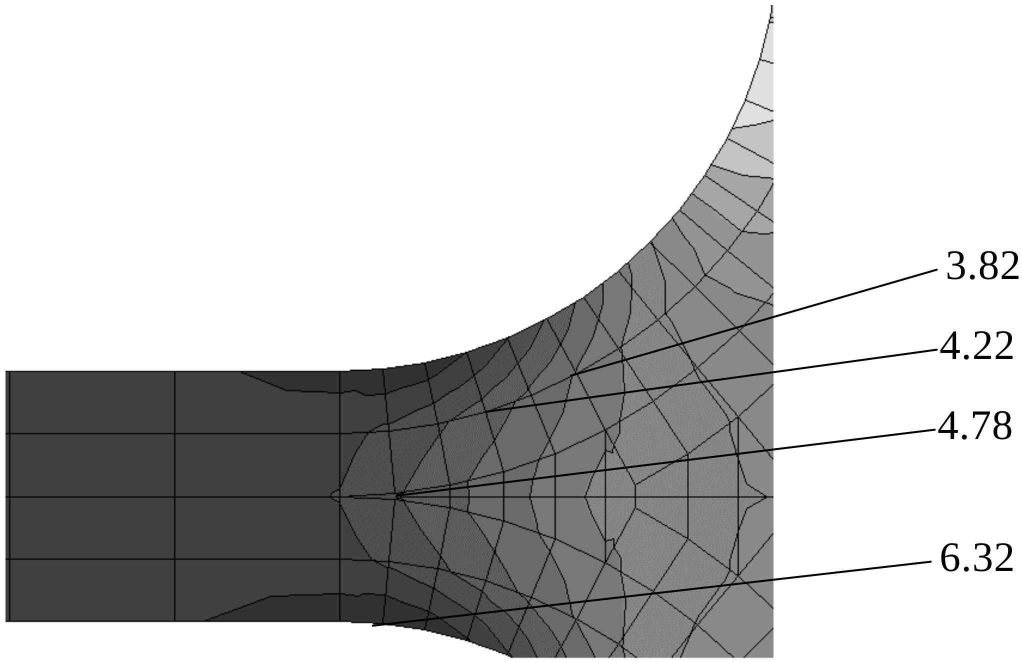
CASE 7: 2 Large Bays

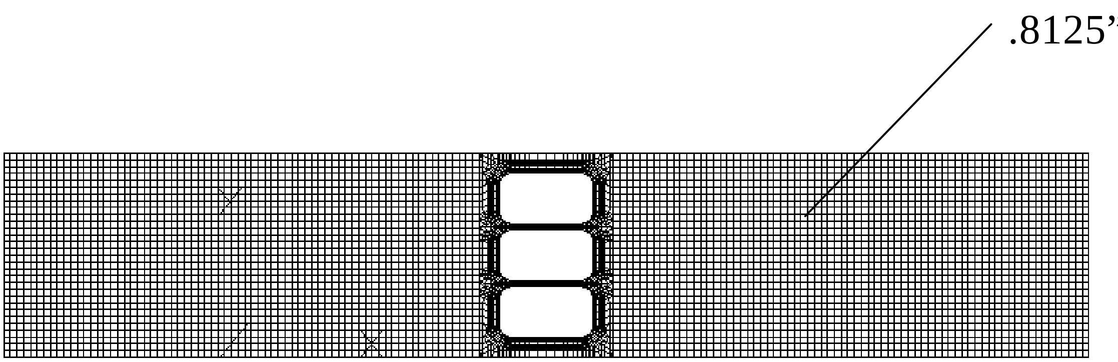



2.04
1.63
1.21
.56


3.47
3.84
5.19
4.36

CASE 7: 2 Large Bays





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The following persons were members of the committee that represented the Ship Structure Committee to the Contractor as resident subject matter experts. As such they performed technical review of the initial proposals to select the contractor, advised the contractor in cognizant matters pertaining to the contract of which the agencies were aware, performed technical review of the work in progress and edited the final report.

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SSC Report Number	Report Bibliography
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SSC 416	**Risk Based Life Cycle Management of Ship Structure** Dr. B.M. Ayyub, U.O. Akpan, G.F. DeSouza, T.S. Koko, X. Luo 2001
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SSC 406	**Sea Operational Profile for Structural Reliability Assessments** I.F. Glenn, R.B. Paterson, L. Luznik, 1999
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