

Technical Report on Capabilities of API Integral Flanges Under Combination of Loading—Phase II

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Contents

	Page
1 Scope	1
2 Introduction	2
3 Three-dimensional Finite Element Analysis	3
3.1 Finite Element Modeling	3
3.2 Finite Element Results	11
4 Two-dimensional Heat Transfer Analysis	18
4.1 Finite Element Modeling	18
4.2 Temperature Analysis Results	18
5 Limiting Criteria for Combined Loading	21
5.1 Introduction	21
5.2 Leakage Criterion	21
5.3 Stress Criterion	23
5.4 Development of Load Combination Charts	25
5.5 Results	26
6 Reanalysis of Failed Flanges	31
7 Effect of Lockdown Screw Holes on Strength	31
8 Conclusion and Recommendations	38
Annex A—Comparison Between Prac 86-21 and Prac 88-21 Results	40
Annex B—Load Combination Charts	45
Tables	
1 Nodal Temperatures for Typical Type 6BX Flanges	20
2 Adopted Stress Criteria	23
3 Flanges That Did Not Meet Makeup-only Criterion, Makeup = 52.5 ksi.	33
4 Flange Geometry Modifications to Meet ASME Criterion for 52.5 ksi Makeup	33
5 Lockdown Screw Holes	33
6 Effect of Lockdown Screw Holes on Flange Stiffnesses	34
7 Effect of Lockdown Screw Holes on Flange Combined Load Capability	34
8 Modification Coefficients for Bending Capacity at Zero Tension and Zero Pressure	34
A.1 Comparison for Type 6B Flanges	41
A.2 Comparison for Type 6BX Flanges	42
B.1 Makeup Values for Flanges with Stress Criterion Governing at Least for Part of the Range of the Load Combination	45
Figures	
1 A Typical Finite Element Model of Flange and Bolt Superelements	4
2 Unit Load Cases	5
3 Superelemental Hierarchy	7
4 Gasket Modeling	8
5 Typical Lotus 1-2-3 Worksheet for Calculating Equivalent Stresses Due to Applied Forces	9
6 Equivalent Stresses Due to Applied Bending Moment	10
7 Deflected Shape Due to Unit Makeup (1.0 ksi)	11
8 Deflected Shape Due to Unit Pressure	12
9 Deflected Shape Due to Unit Tension (1.0 kip)	12
10 Deflected Shape Due to Bending Moment (1.0 K.in.)	13

11	Maximum Shear Stress Contours Due to Unit Makeup	13
12	Maximum Shear Stress Contours Due to Unit Pressure	14
13	Maximum Shear Stress Contours Due to Unit Tension	14
14	Maximum Shear Stress Contours Due to Unit Bending Moment	15
15	Von Mises's Equivalent Stress Contours for 2 1/16 in. 3,000 psi Type 6B Flange Under 52.5 ksi Makeup and 6,000 psi Test Pressure	15
16	Bolt Stress (σ_z) Contours for Unit Makeup	16
17	Bolt Stress (σ_z) Contours for Unit Pressure	16
18	Bolt Stress (σ_z) Contours for Unit Tension	17
19	Bolt Stress (σ_z) Contours for Unit Bending Moment	17
20	Typical Heat Conduction 2-D Mesh for Type 6BX Flanges	19
21	Typical Leakage Load Combination Charts for 52.5 ksi and 40 ksi Makeup Loads	22
22	Stress Linearization	24
23	Critical Section Locations	25
24	Comparison Between 2-D and 3-D Results for Leakage Criteria for 2 1/16 in. 3,000 psi 6B Flange.	27
25	Comparison Between 2-D and 3-D Results for Leakage Criteria for 16 3/4 in. 3,000 psi 6B Flange	27
26	Stress Limiting Criteria for 2 1/16 in. 3,000 psi 6B Flange.	28
27	Stress Limiting Criteria for 16 3/4 in. 3,000 psi 6B Flange.	28
28	Leakage Criteria with Temperature Effect for 2 1/16 in. 3,000 psi 6B Flange	29
29	Leakage Criteria with Temperature Effect for 16 3/4 in. 3,000 psi 6B Flange	29
30	Stress Criterion with Temperature Effect for 2 1/16 in. 3,000 psi 6B Flange	30
31	Stress Criterion with Temperature Effect for 16 3/4 in. 3,000 psi 6B Flange	30
32	Bolt Stress Contours for 5 1/8 in. 10,000 psi Type 6BX Flange	32
33	Bolt Stress Contours for 11 in. 10,000 psi Type 6BX Flange	32
34	Finite Element Model of Flange with Lockdown Screw Holes	35
35	Effect of Lockdown Screw Holes on Leakage Charts	36
36	Effect of Lockdown Screw Holes on Stress Charts	37
A.1	Hopper Chart (from ASME <i>Boiler and Pressure Vessel Code</i> , Appendix 4)	44

Technical Report on Capabilities of API Integral Flanges Under Combination of Loading—Phase II

1 Scope

The evaluation of the load carrying capacity of API 6A integral flanges is the objective of this work. The applied loading includes the end tension and bending moment in addition to the conventional rated pressure and makeup forces. The effect of a temperature difference corresponding to 250 °F on the inside and 30 °F on the outside was also evaluated.

Three-dimensional finite element meshes were generated for the Type 6B, and Type 6BX flanges. The bending moment load case required a model of one quarter of the flange which was built up from the smaller segments and the half-bolt superelements. The computer program SESAM was used to obtain the stresses at selected critical flange and hub sections and to determine the gasket reaction due to each of the four unit load cases and the temperature difference load case. Leakage criterion was defined as the load combination which reduces the initial makeup compressive forces in the gasket to zero. The stresses in each defined section were linearized in accordance with the ASME *Section VIII, Division 2*, procedure to determine the membrane and membrane-plus-bending stress intensities. These stress intensities were checked against the allowables specified in API 6A, and the limiting loads were determined. A computer program LCCP was written to carry out this code check and a LOTUS 1-2-3 Release 3 worksheet was used to plot the load combination charts.

The results of the analysis carried out indicate that the leakage criterion governs the capacity of the smaller flanges in the Type 6B flanges. Leakage was governing for up to 9 in. size flanges in both the 52.5 ksi and 40 ksi makeups for the 2000 psi pressure. Leakage was governing the 5 1/8 in. for the higher pressures. Leakage was also found to be governing all Type 6BX flanges for working pressures of up to 5,000 psi. For the 10,000 psi and 15,000 psi flanges, leakage governed only in the larger size range greater than 2 9/16 in. Leakage was governing in all the 20,000 psi API 6BX flanges. The leakage model adopted in this study employs several approximations that have not yet been evaluated. Therefore, the actual leakage forces, i.e. load combinations leading to leakage, may be considerably higher than assumed herein. In reality, the gasket only leaks when its energized capacity is exceeded.

The state of stress at the stress governing hub section under the combined loading of makeup, pressure, tension and bending moment is considered to be "secondary." However, when pressure, tension, and bending moments are applied together with the necessary makeup to resist these actions without leakage, the state of stress is rendered "primary" and, therefore, the allowable stress intensities are halved. This does not seem to be consistent, and it may by far exceed the intention of the code. However, the oversight subcommittee preferred to adopt the conservative route, which may be overly conservative pending further evaluation. Therefore, it may be concluded that when the hub stresses are treated as primary, most flanges do not possess significant reserve strength beyond the leakage condition. In fact, if the leakage condition was somewhat conservative, the stress condition may become governing for most flanges.

The temperature difference of 250 °F internal and 30 °F external leads to increases in the load-carrying capacity of the flanges. This condition is caused by the compressive forces generated in the gasket due to this temperature difference, and the increase in the allowable stresses when the self-limiting temperature load condition is included. It is recommended that a 3-D finite element, nonlinear material and geometric models of approximately eight flanges be carried out to determine the actual failure mechanism that governs the behavior of these flanges. This includes the prediction of the response of the gasket under increasing load and a more accurate definition of the leakage mechanism. The elimination of the raised face does not significantly reduce the stresses in the hub which caused six Type 6B flanges to fail to meet the ASME criterion for makeup load only (52.5 ksi for 105 ksi bolting). The stress intensities were reduced only by about 5 % when the raised face was eliminated, increasing the thickness of the flange by about 10 %. The hub thickness for these flanges had to be increased by up to about 27 % of their existing thicknesses together with the elimination of the raised face.

The bolt stresses did not govern for any of the flanges analyzed. Bolt stresses are typically within approximately 67 % of their yield strength due to makeup, pressure, tension, and bending moment loads. The bolts are expected to be made up to half their yield. The stresses in the bolts due to temperature differences increase by about 5 ksi to 7 ksi, which is

about 6 % to 8 % of the bolt yield stress. The other load conditions (pressure, tension, and bending moments) increase bolt stress by twice the increase due to the temperature difference. Therefore, it is concluded that the bolts will not approach their limiting criterion under the investigated load conditions.

2 Introduction

The original design criteria, of which API flange dimensions are based, were developed in the late 1920s and 1930s by Waters prior to the advent of the recent powerful, computer technology and associated finite-element analysis programs. The original analysis methods were used successfully in defining the flange design and pressure rating but did not address external loading capabilities. The flange designs were based on surface stress calculations, which are appropriate for uniform-axis symmetric loading conditions as used for bolt loading and pressure. Determining the limiting conditions of flanges beyond that of pressure rating, where external loading is considered, requires knowledge of stresses as they are distributed through the thickness. Methods of calculating these stresses have been developed through the use of finite element programs, and have been applied in this technical report.

The criteria for design stress allowables of API 6A body, bonnet, inlet and outlet connections were established in the Fifteenth Edition. Conformance with the API 6A criteria requires stress linearization calculations based on the methodology established by ASME *Pressure Vessel Code, Section VIII, Division 2*. These stress allowables are defined in terms of stress distributions through the thickness and necessitate a more accurate means of calculating component stresses as provided by finite element analysis.

Therefore, it was the intent of PRAC 86-21 (API 6AF) and the present work to establish a better understanding of the stress distribution in these flanges under the effect of the combined actions of pressure, bending moment, tension, and the original bolt makeup of the flange. Thus, the flange capacities which meet the design stress allowables of API 6A are determined.

The pressure/bending moment/tension charts developed in this technical report provide the critical load combination limits for either ASME/API 6A stress criteria or to leakage. These charts are extremely useful for both designers and users of API flange connections where external load capacities are critical.

The results of the work conducted in this technical report indicate increased load-carrying capacity for several flanges based on separate stress-limiting criteria. These stress limiting charts were not provided separately in the original work of API 6AF, which combined them with leakage criteria. The three-dimensional model analyses of this study provide verification that axisymmetric finite elements results of flanges, as used in API 6AF, are conservative. Additionally, this study determined a few flanges to have less loading capacity than originally defined in API 6A for makeup loading, and thus, have been reduced to meet design requirements.

The specifications for flange designs of Type 6B and Type 6BX originated in API 6A but have extensive diversification into other areas, such as surface and subsea drilling equipment and subsea production equipment. The API specifications for these applications either duplicate the design or reference back to the API 6A document. Therefore, use of the charts provided within this technical report have a broad base application where external loading conditions are critical.

It should be noted that the work presented herein has several limitations due to the defined scope and the assumptions involved in the analysis: the effects of transverse shear and/or torsion were not considered; the results are for static loading only, no dynamic, fatigue, or fretting phenomenon were accounted for; no elevated temperature effects were considered in the thermal stress analysis performed, only steady-state heat conduction was applied; the results representing the onset of leakage are based on linear behavior assumptions and are therefore only applicable to the linear range of the gasket response; and the load-combination capacity charts given are not intended to replace critical evaluation of any particular connection in an application where the charts show the flange to be marginal. These charts are intended to be used only as general guidelines for design.

Subsequent to the completion of this work, the 5¹/₈ in. 15,000 psi flange was added to API 6A. This new flange was analyzed using the same basic methodology but with the ABAQUS general-purpose finite-element system, and using a

half-symmetry three-dimensional model instead of a quarter model in order to find the same data as shown for the other flanges in this technical report.

Details of the analysis of the 5/8 in. 15,000 psi flange can be found in the report, *Finite Element Analysis of 5 1/8 in. 15 ksi API 6BX Flange*, which was prepared for API under report, SES Report Number PN114809, dated February 2006.

3 Three-dimensional Finite Element Analysis

The main effort in this work was expanded in the three-dimensional finite element modeling of the Type 6B and API 6BX flanges of API 6A. The objective of this task is to determine the response of the flange to makeup/preload, pressure, tension, and bending moment forces. This response is then used to produce load-carrying capacity charts, which may be employed safely by designers to select the appropriate size flange that can resist the applied design forces. Originally, it was thought that only a few models would have to be generated due to the similarities of the geometry of the flanges. However, as the work progressed, it became evident that almost each flange had different geometry that necessitated a different finite element mesh model. Therefore, at least 45 different models had to be developed. In the following two subsections, the finite element modeling and the results thereof are described.

3.1 Finite Element Modeling

The finite element model developed for the Type 6B flanges is typically depicted in Figure 1. For each flange, follow the following steps:

- a) draw the flange to scale;
- b) select suitable mesh;
- c) calculate the dimensions necessary for input to the program creating the mesh (the preprocessor program PREFEM);
- d) specify material properties; and
- e) represent the four loading conditions by equivalent nodal forces or surface pressures. These loading conditions are makeup, pressure, tension, and bending moment.

3.1.1 Load Cases

Four unit load conditions were applied (see Figure 2).

- a) *Load Condition 1 (LC1)*—A unit pressure of 1.0 ksi in the bolts was applied to represent the makeup or preload condition.
- b) *Load Condition 2 (LC2)*—A unit pressure of 1.0 ksi was applied on the inside surface of the hub and to the front face of the flange outward to gasket support location.
- c) *Load Condition 3 (LC3)*—A unit tension of 1.0 kip was applied at the hub end section.
- d) *Load Condition 4 (LC4)*—A unit bending moment of 1.0 kip-in. was applied to the hub end section using the corresponding stresses given by:

$$\sigma_m = x/I_h$$

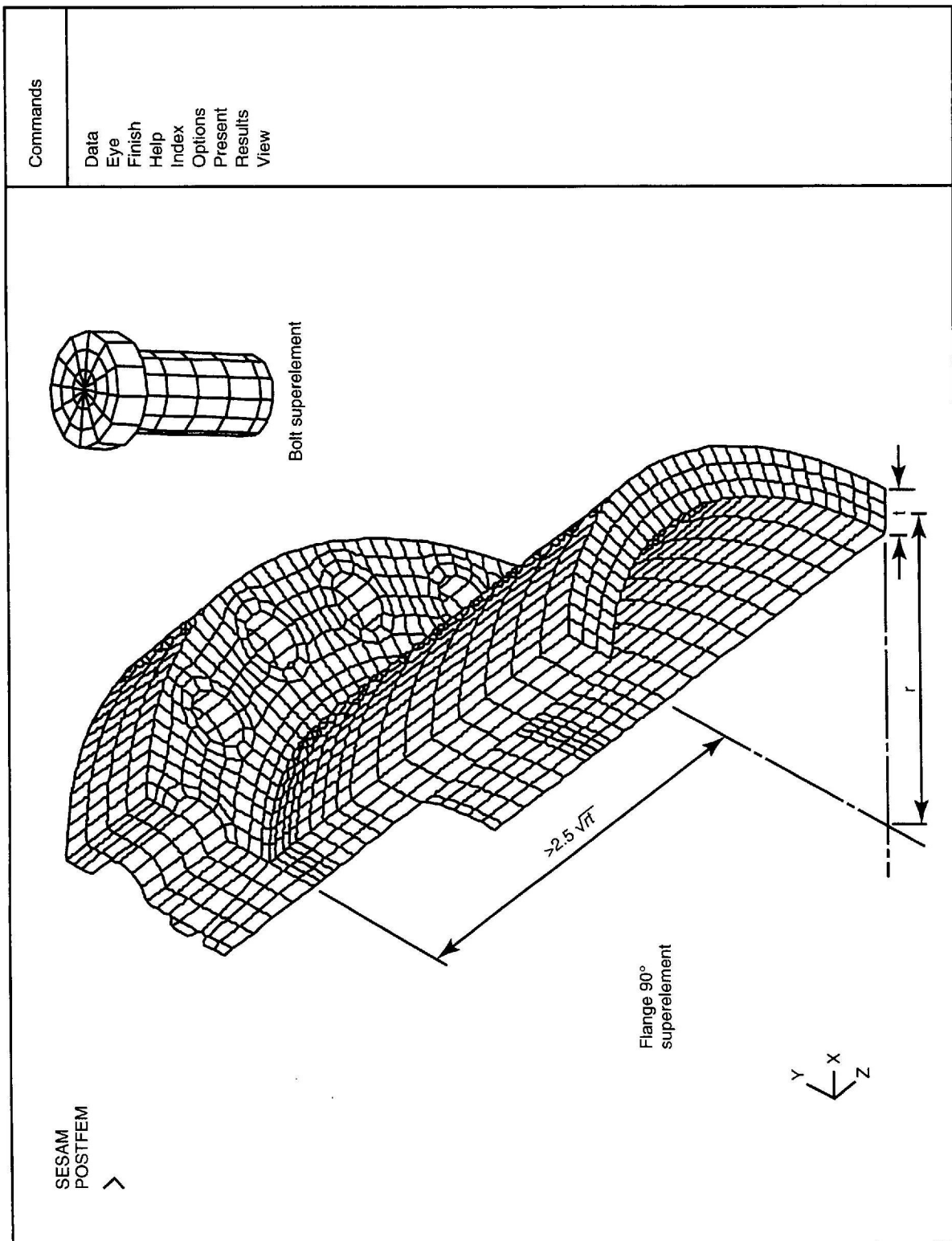


Figure 1—A Typical Finite Element Model of Flange and Bolt Superelements

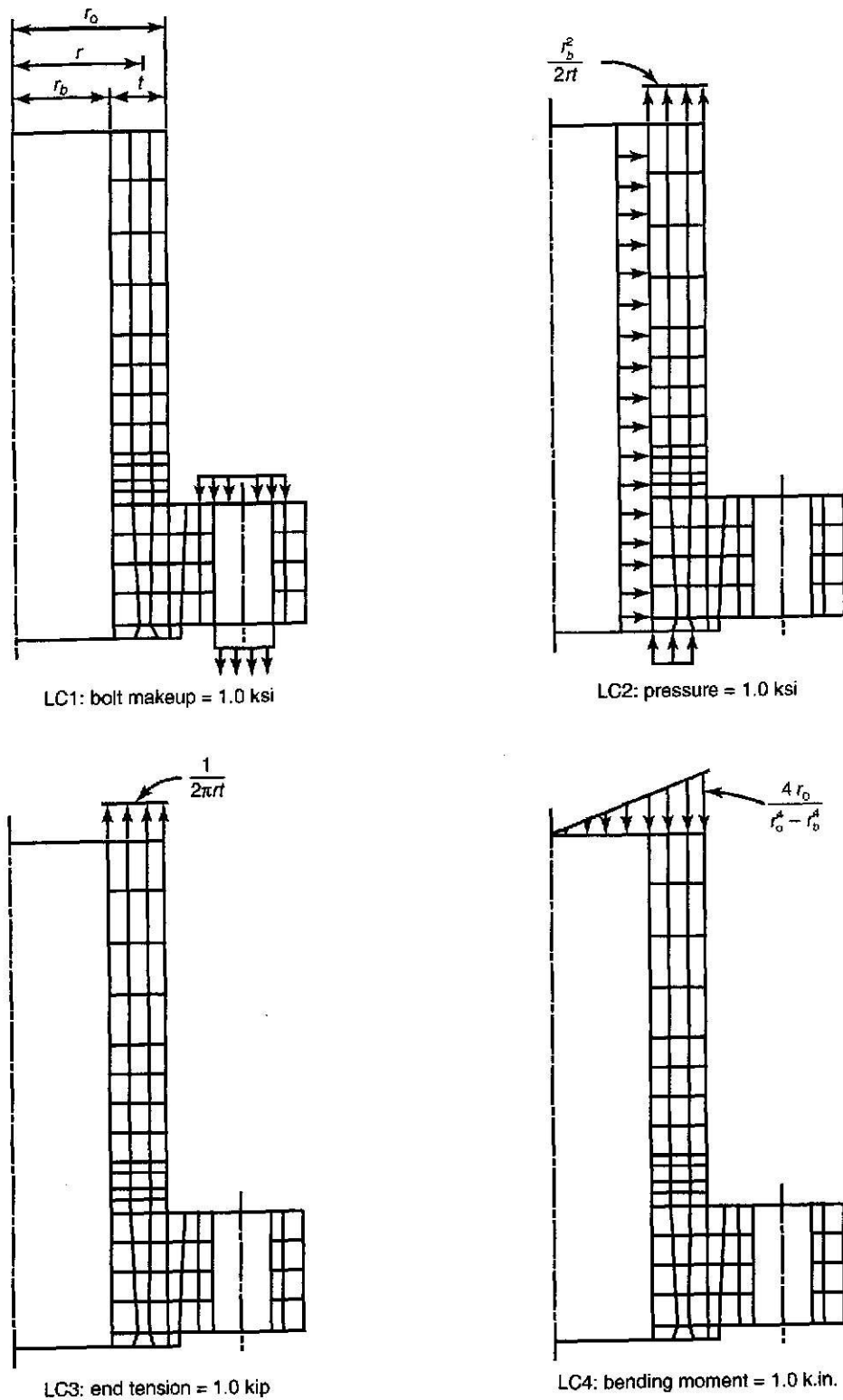


Figure 2—Unit Load Cases

where

x is the x-coordinate of any point on the hub end section; and

I_h is the moment of inertia of that section.

An additional load case representing the temperature difference between external and internal surfaces was considered. The inside surface temperature was 250 °F with the outside surfaces at 30 °F.

As described in 3.1.2, all axisymmetric loading conditions required the modeling of only one segment containing one half of a bolt. The bending moment, however, required a model of at least 90° (one quarter) of the flange.

3.1.2 Mesh Geometry

After plotting each flange to scale, a reasonable size mesh is selected. The mesh varied according to the radial location of the gasket and the raised face, and according to the location of the bolt hole relative to the raised face. It also varied with the number of bolt holes and therefore the size of the minimum segment that needed to be modeled.

Ideally, three super-elements would be required to model the flange, bolt, and gasket. However, it was decided to substitute the gasket super-element with a circumferential roller support to be able to employ the linear behavior theory to define the leakage condition which is reached when the compressive preload in the gasket is offset by the tension forces resulting from the pressure, tension, and bending moment actions.

All basic finite elements used to model the flange and the bolt were 8-node and 6-node solid elements as shown in Figure 1. The computer program SESAM was employed in the analysis. This program has been widely used in the design and reanalysis of many offshore structures. The program has pre- and post-processor routines that facilitated the analysis of the API integral flanges.

As noted in Figure 1, the model incorporated a distance of at least $2.5\sqrt{rt}$, where r is the mean radius of the hub and t is its thickness, to prevent the end boundary conditions from affecting the results in the flange. The gasket ring groove was modeled, and the gasket support point was selected to represent the Type 6B and Type 6BX flanges with appropriate extension pressure to the surfaces on which the internal pressure is applied.

Because of symmetry conditions, only a segment of the flange is required to be modeled. For a flange with n_b bolts, only $1/2n_b$ segment of the flange needs to be analyzed for the first three load conditions; namely, the makeup, pressure and tension. However, one quarter of the flange is required for the bending moment load condition. The computer program SESAM had a superelement facility which was utilized to build up one quarter of the flange from the $1/2n_b$ segments generated. Figure 3 shows an example of a tree structure of the superelement hierarchy. The assembly of the quarter of the flange is carried out by properly rotating and mirror imaging the lower level superelements. In addition, the loads can only be given on the lowest level superelement and were defined such that the actual bending stresses are obtained when the highest level superelement is created.

The advantages of using three-dimensional finite element modeling compared to two-dimensional include the following:

- a) more accurate representation of the bolts and bolt holes;
- b) more accurate representation of the applied loading;
- c) prediction of the effects of bolt holes on hoop and other stresses; and
- d) representation of lockdown screw holes as necessary for predicting stresses in the flange.

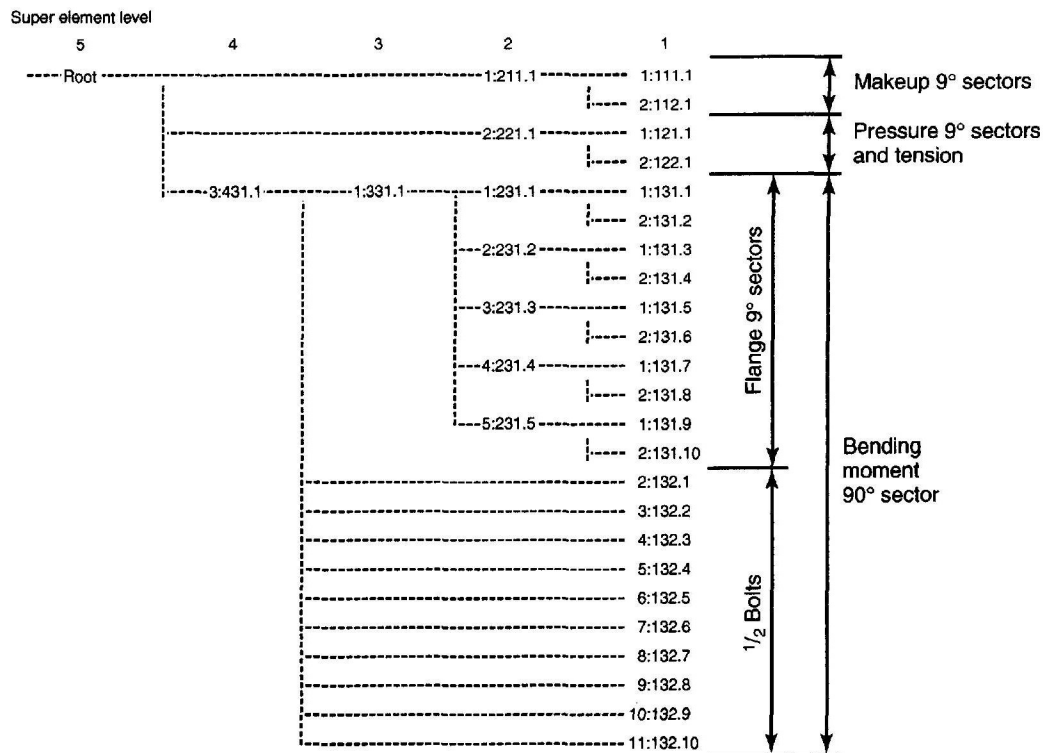


Figure 3—Superelemental Hierarchy

Only for the purpose of the discussions given in this report the flange will be assumed to lie such that the axis of the pipe to which it is connected is vertical.

3.1.3 Material Properties

All material properties were considered to be representatives of steel. A Young's modulus of 29,000,000 psi and a Poisson's ratio of 0.3 were employed. A coefficient of thermal expansion of $6 \times 10^{-6} / ^\circ\text{F}$ and a material density of $7.35 \times 10^{-7} \text{ kip/in.}^3$ were also employed.

3.1.4 Assumptions and Approximations

The following assumptions and approximations were made in the modeling of the flanges.

- Mesh density is only adequate for predicting global stresses (stress concentrations). No attempt is made here to determine which would be required for a fatigue-type analysis.
- All flanges used the maximum raised face thickness as specified in API 6A.
- The gasket is represented as one line circular roller support on which the flange reacts when subjected to makeup forces.
- When pressure, tension, bending, and temperature load cases are applied, the flange is supported by both the gasket and the bolts.
- The makeup load was assumed to be 50 % of the bolt yielding load. This amounts to 40 ksi and 52.5 ksi for 80 ksi and 105 ksi bolting, respectively.

- f) The roller support representing the gasket was located at second outermost radius as shown in Figure 4 for both the Type 6B and Type 6BX flanges.
- g) The pressure was extended to the location of the gasket line for the Type 6B flanges. For the Type 6BX flanges, the pressure was assumed to extend all the way to the outermost radius of the gasket groove (see Figure 4).
- h) The nominal flange dimensions given in the tables of API 6A were employed. No account of the tolerances given was included in the analysis. The effect of tolerances is assumed to be small enough to be neglected.
- i) In calculating the makeup loading, a basic minor diameter of the bolt was used. *Mark's Mechanical Engineer Handbook* was used to determine this dimension.
- j) The thickness of the nut or bolthead was determined using the same reference handbook shown in item i.
- k) The bending moment is applied as external normal stresses applied at the hub boundary section.
- l) The hub boundary section was taken at a minimum distance of $2.5\sqrt{rt}$.
- m) The flange is assumed to mate into an identical flange or a stiffer structure capable of withstanding the applied forces.
- n) The bolt superelements interface with the flange only at the bolthead contact surface with the flange. The bolt threads are not modeled.
- o) Only force boundary conditions were applied at the hub boundary surface. The boundary conditions were changed from free to fixed, and no significant effect on the stresses in the neighborhood of the flange was found. This means that the length of the hub used in the analysis was adequate.

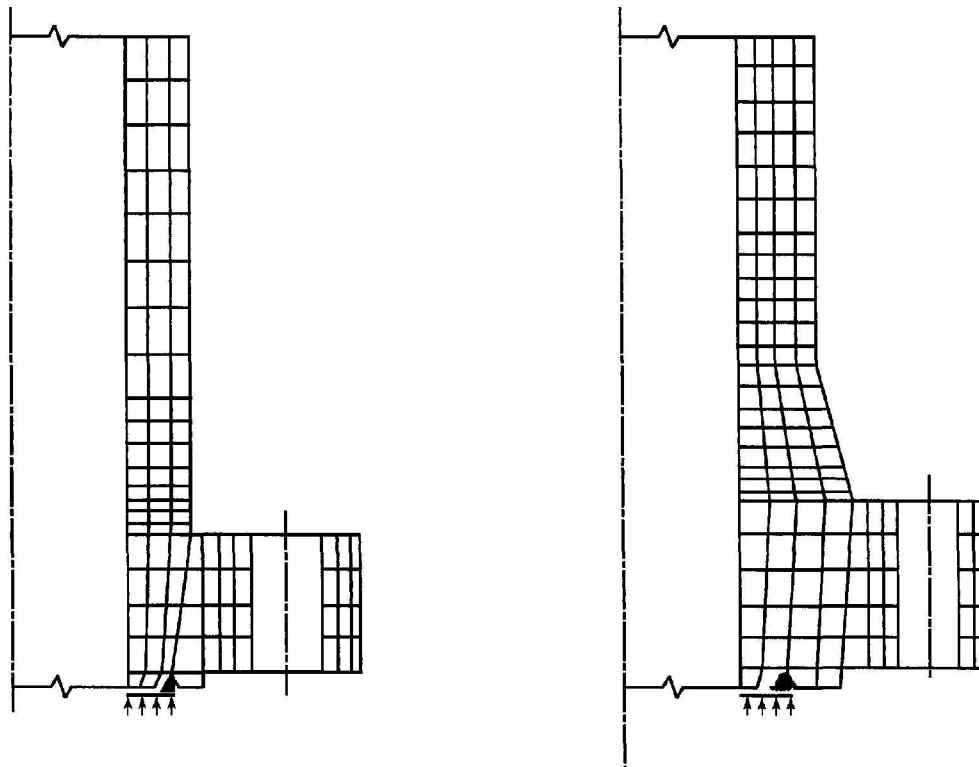


Figure 4—Gasket Modeling

Figure 5 shows a typical Lotus 1-2-3 worksheet used to calculate the unit stresses, which were input to the finite element model. The particular worksheet shown was developed for the 16 ³/₄ in., 2,000 psi Type 6B flange which has 20 bolts. The axisymmetric load cases used a 9° segment model with symmetry boundary conditions applied at 0° and 9° surfaces. A 90° model is built out of the 9° segment using mirroring and rotating techniques of the superelement program, PRESEL. The equivalent load representation for the bending moment load case is shown in Figure 6. Only loads on the lower level superelement, element No. 1, were permitted in the program; therefore, the input loads were such that the correct bending stresses are produced when the element is mirrored or rotated to build the 90° sector. Antisymmetric boundary conditions are used at the plane about which the bending moment is applied. Figure 6 shows four load cases defined for a flange with eight bolts on a 22.5° segment. When the 90° sector is created, these loads give the correct triangular stress distribution.

Worksheet to Calculate Flange Properties and Loads:

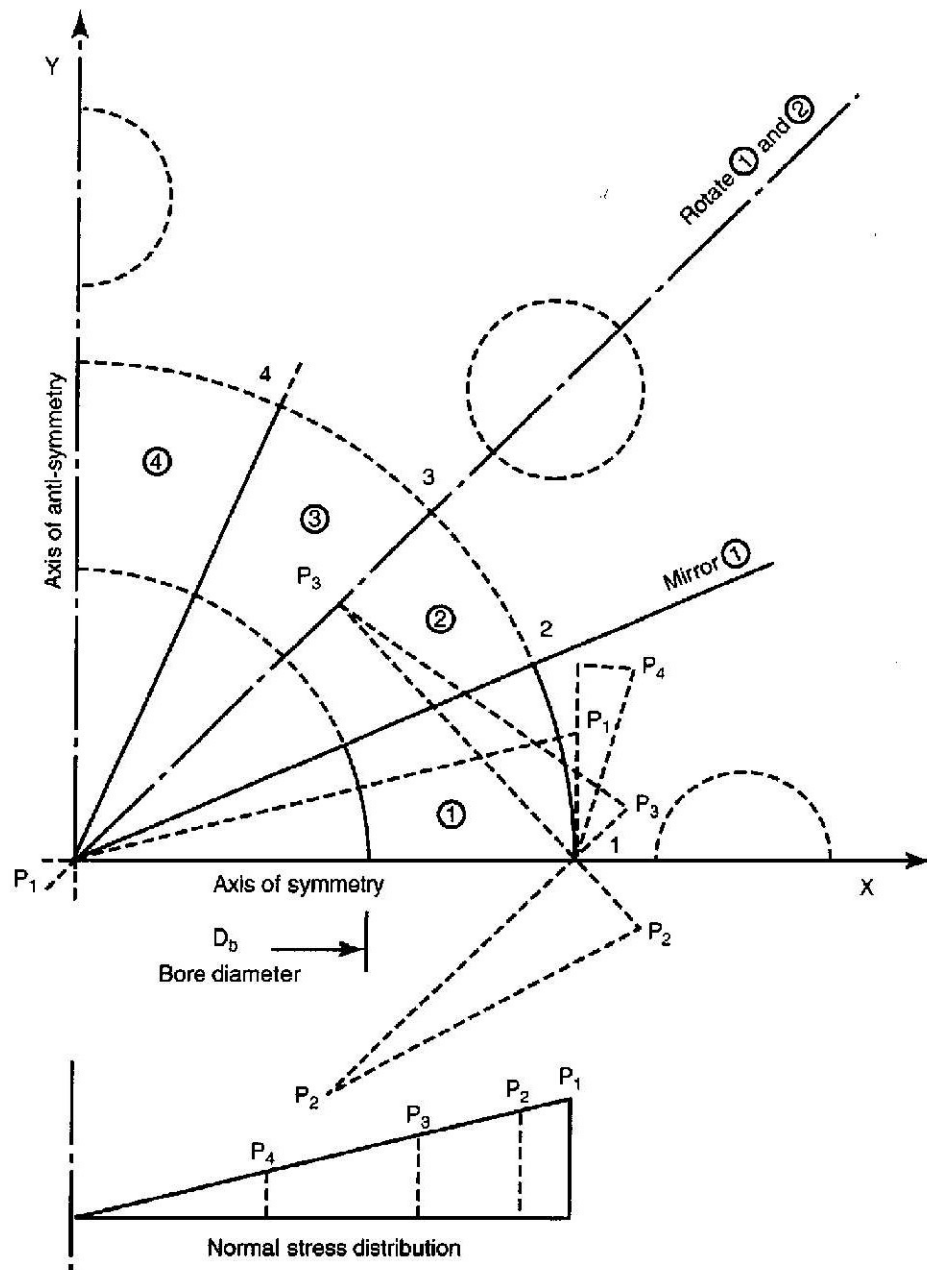
Parameters:	Flange No.	6BR65
Hub Outer Radius	Rho	9.75 in.
Hub Inside Radius	Rhi	8.39 in.
Hub Sec Area	Ahub	77.504347401 in. ²
Stress for $T = 1$ k	SigT	0.012902502 ksi
Stress for $p = 1$ ksi	SigTp	2.8533019327 ksi
Moment of Inertia	I	3205.8626994 in. ⁴
Max BM stress	$p1$	3.041303E-03 ksi
No. of Bolts	N_b	20
	n	10
	Theta	9 Degrees

Bending Moment Representation:

Point Where	$p = 0$		$p1$	Node where $p = p1$
No.	X (in.)	Y (in.)	(ksi)	
1	0.00000	0.00000	3.04130E-03	1
2	0.93104	-2.86545	2.89245E-03	1
3	0.93104	2.86545	2.89245E-03	1
4	3.36854	-4.63640	2.46047E-03	1
5	3.36854	4.63640	2.46047E-03	1
6	6.38146	-4.63640	1.78763E-03	1
7	6.38146	4.63640	1.78763E-03	1
8	8.81896	-2.86545	9.39814E-04	1
9	8.81896	2.86545	9.39814E-04	1
10 Node		1	4.75765E-04	9.75 1.525236

Figure 5—Typical Lotus 1-2-3 Worksheet for Calculating Equivalent Stresses Due to Applied Forces

It should be noted that on a VAX 11/750 computer, each flange analysis takes approximately 6 CPU hours for a flange with 8 bolt holes and 18 CPU hours for a flange with 24 bolts. The mesh for the 24 bolt flange was designed with smaller number of nodes and elements in the basic 7.5° sector compared to the 22.5° sector of the 8 bolt flange.



Note: Dotted line triangles represent the loading applied with respect to the only segment modeled No. 1. These loads are mirrored and rotated to produce the correct stress distribution due to the bending moment.

Figure 6—Equivalent Stresses Due to Applied Bending Moment

In the unit makeup loadcase, the unit stress of 1.0 ksi is applied on the bolt gross cross section in the X-Y plane at $Z = 0$ (see coordinate system in Figure 1). In the postprocessing, the stresses due to makeup were scaled down with the ratio between the basic minor area and the nominal area of the bolt section.

3.2 Finite Element Results

The SESAM preprocessor routine PREFEM was employed to create the finite element mesh and input the material and load definitions for the smallest segment that had to be analyzed. The superelement program PRESEL was then employed to build the bolt-flange assembly and to assemble the 90° quadrant required to analyze the bending-moment load case. The stress analysis was then performed using the program SESTR. The output results consisted of all the nodal displacements and element stresses at the Gaussian points. These are the traditional output results for the 8-node and 6-node solid isoparametric elements used in the analysis. The program POSTFEM was employed to plot the stress contours and deflected shapes to ensure that the analysis was reasonable. This program was also employed to create a listing of the Gaussian stresses at the selected critical sections where the maximum stresses occur. These results were then used by the program LCCP (Load Combination Charts Program) which checks the criteria under all possible load combinations and produces data necessary for developing the charts describing the load carrying capacity of the flange. This program is discussed in detail in Section 4.

Figure 7 through Figure 10 show typical deflected shape output results for a flange under the four unit load cases described in 3.1.1. The corresponding maximum shear stress contours are given in Figure 11 through Figure 14. The von Mises equivalent stress contours for a makeup of 52.5 ksi and test pressure of 6,000 psi for this 2 1/16 in. 3,000 psi Type 6B flange are shown in Figure 15. The last graph was produced for 56 flanges to aid in the specification of material requirements carried out by another API committee.

Example stress contours for the bolts due to the above unit load cases were found to be mainly axial (σ_z). Figure 16 through Figure 19 show the stress contours for a bolt under the effect of the four unit load cases. The results shown are given for the Type 6BX flange 5 1/8 in. 10,000 psi.

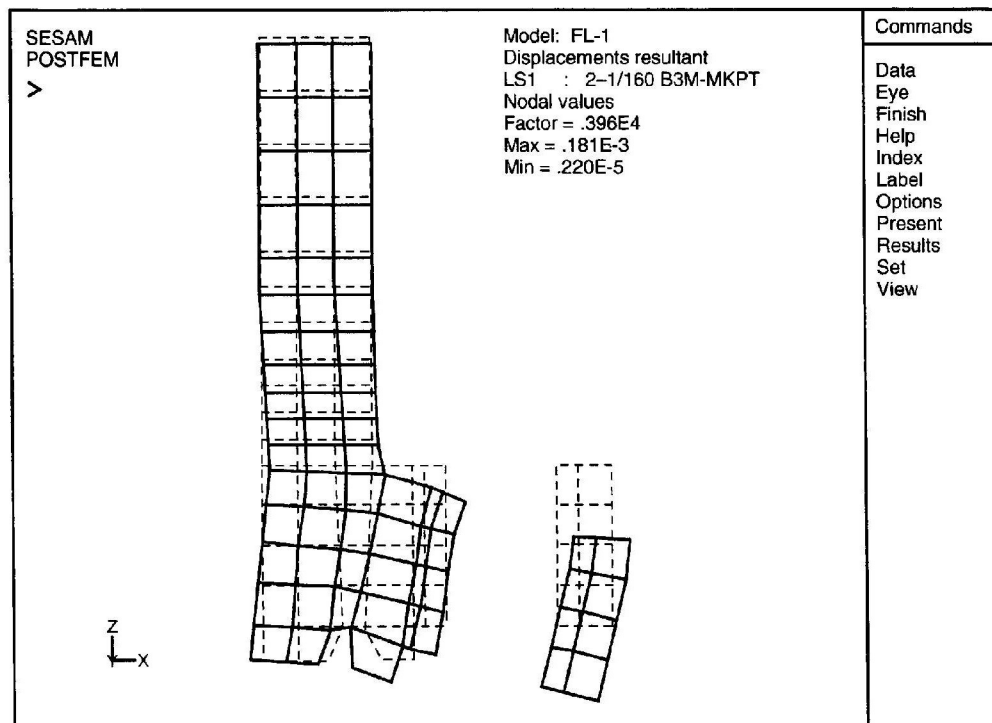


Figure 7—Deflected Shape Due to Unit Makeup (1.0 ksi)

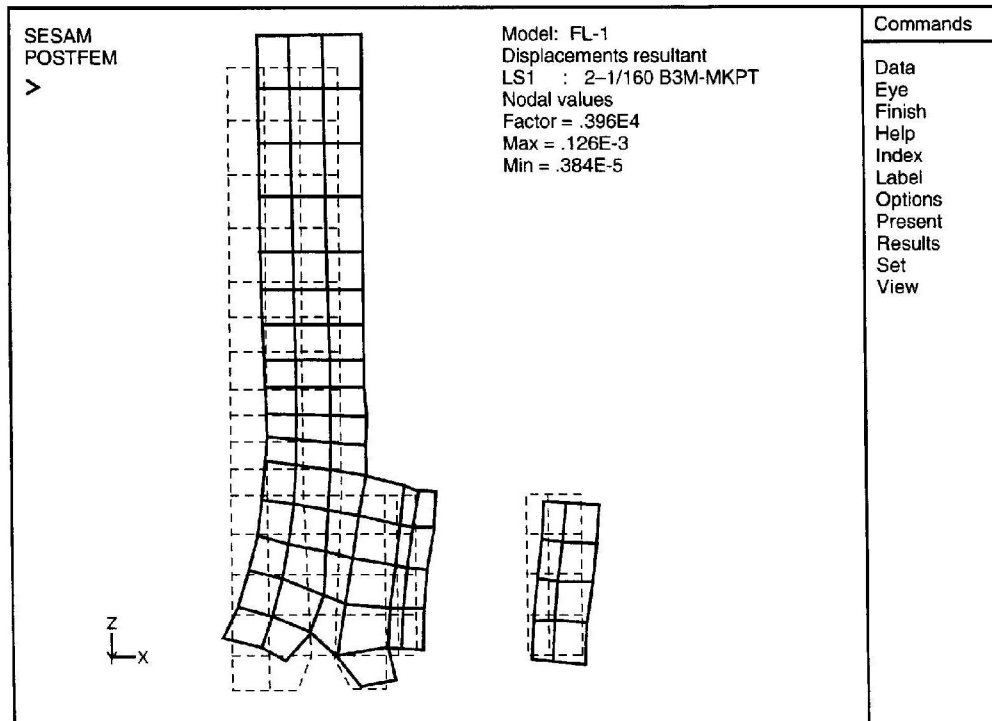


Figure 8—Deflected Shape Due to Unit Pressure

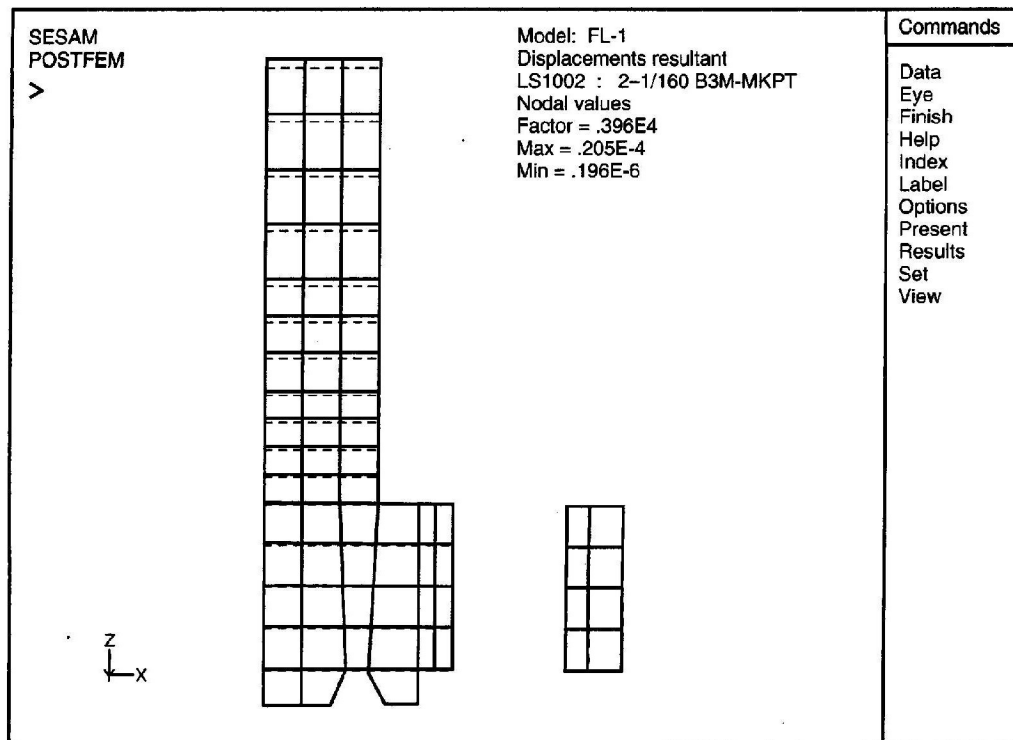


Figure 9—Deflected Shape Due to Unit Tension (1.0 kip)

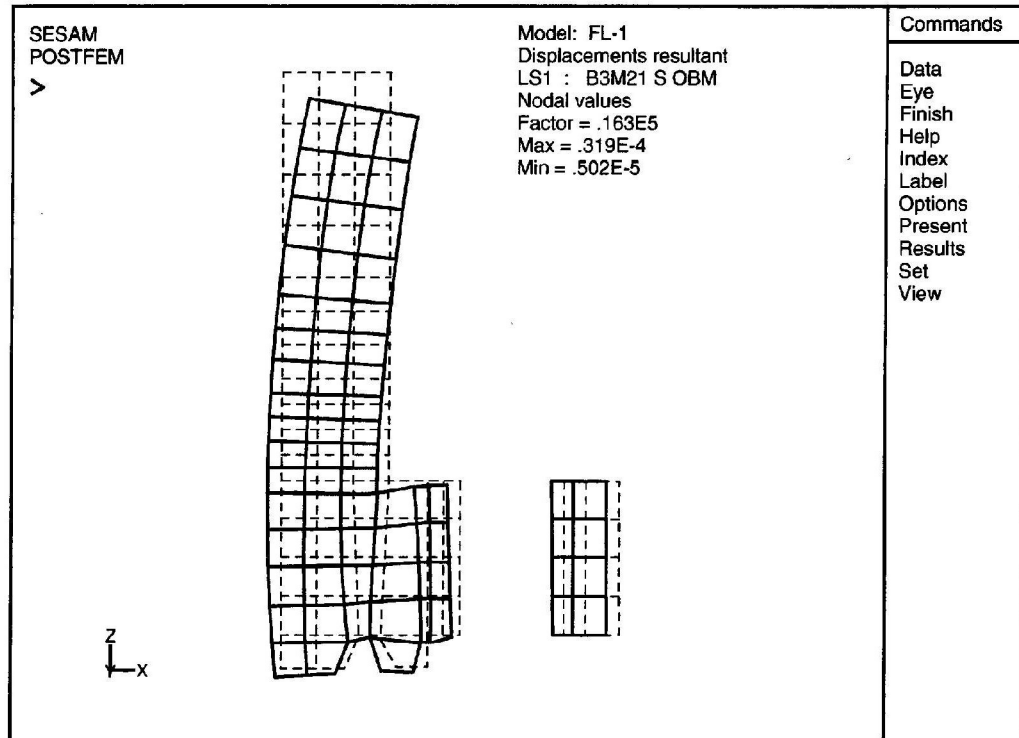


Figure 10—Deflected Shape Due to Bending Moment (1.0 K.in.)

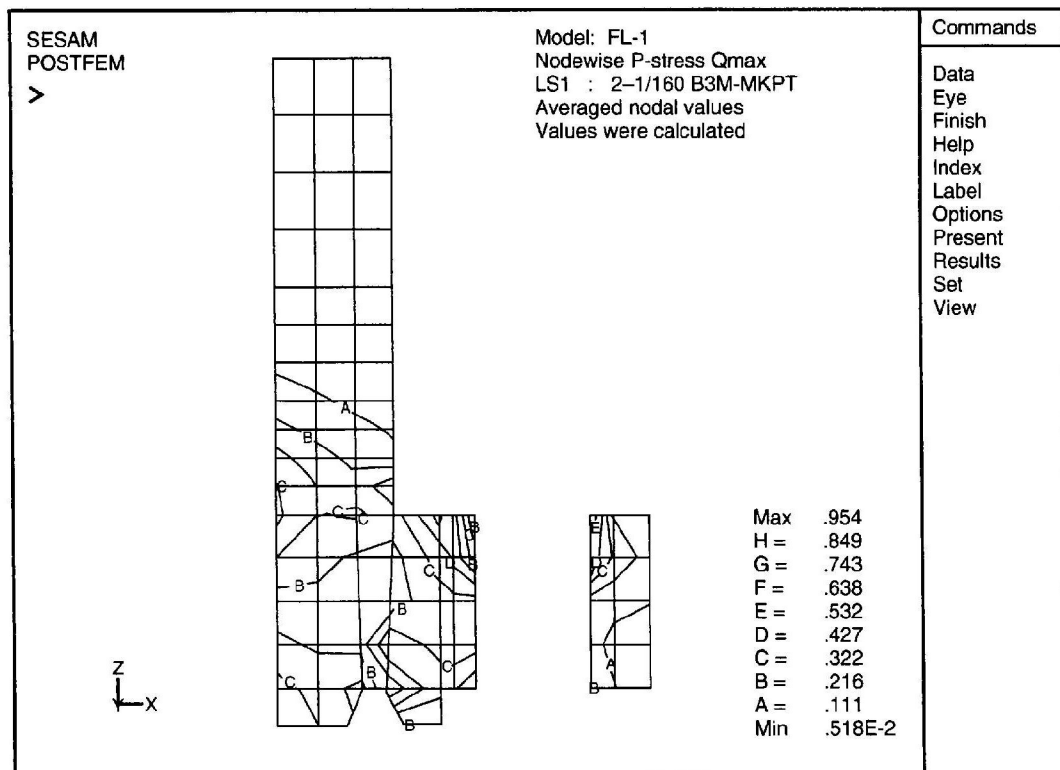


Figure 11—Maximum Shear Stress Contours Due to Unit Makeup

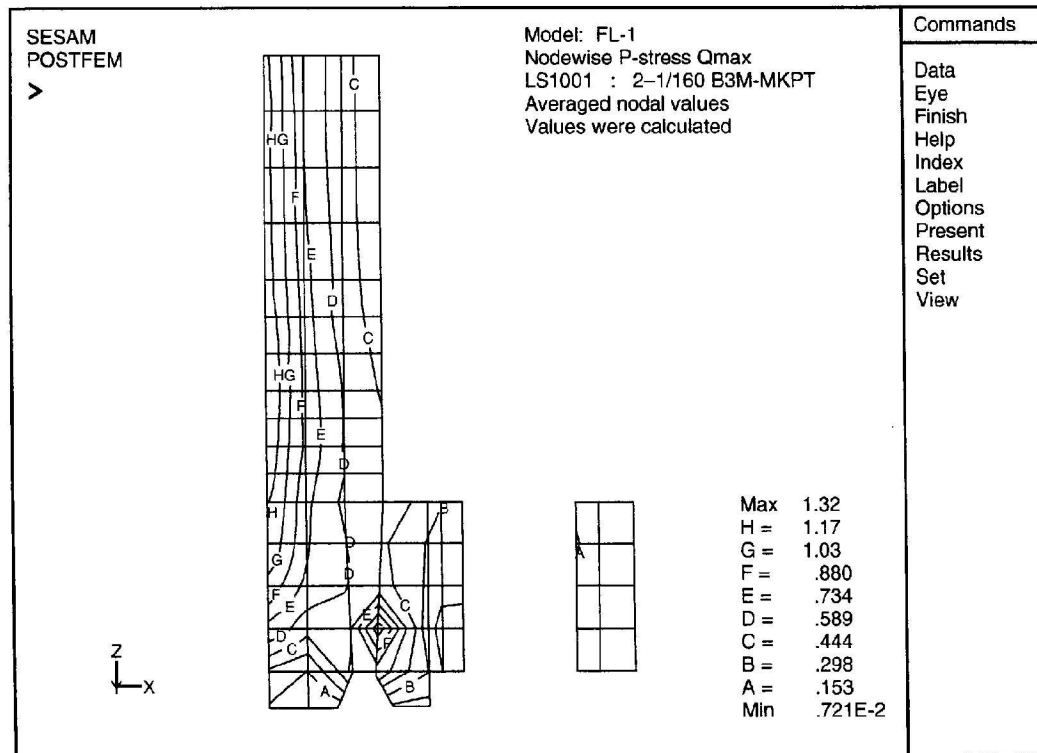


Figure 12—Maximum Shear Stress Contours Due to Unit Pressure

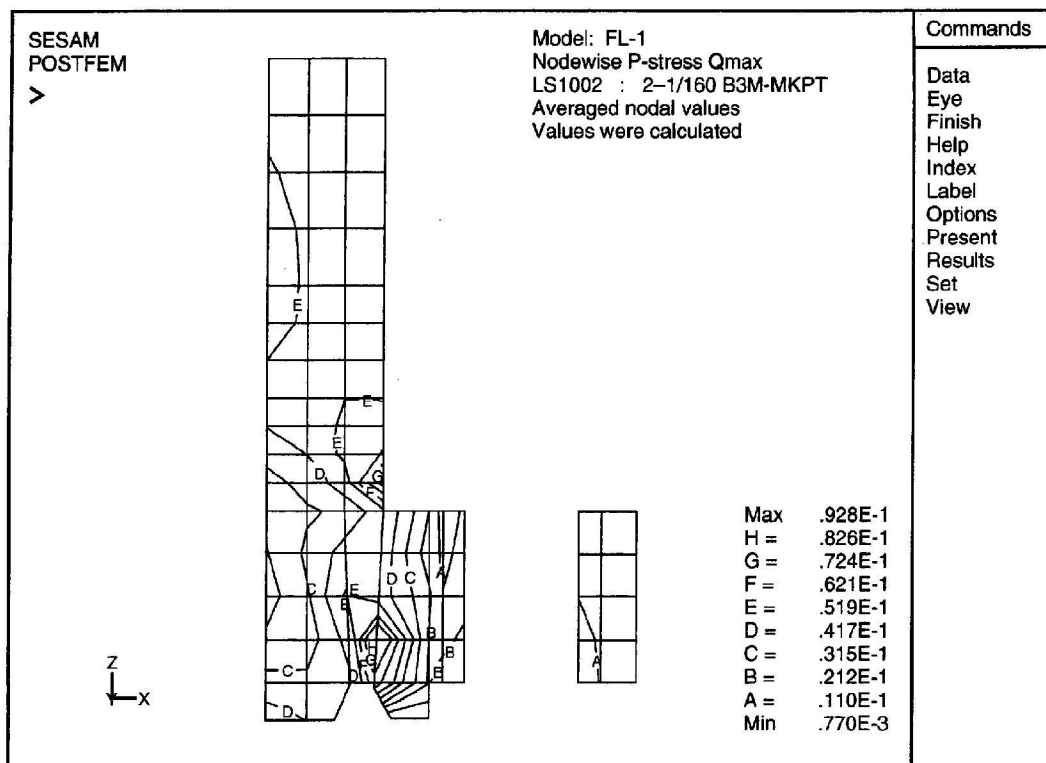


Figure 13—Maximum Shear Stress Contours Due to Unit Tension

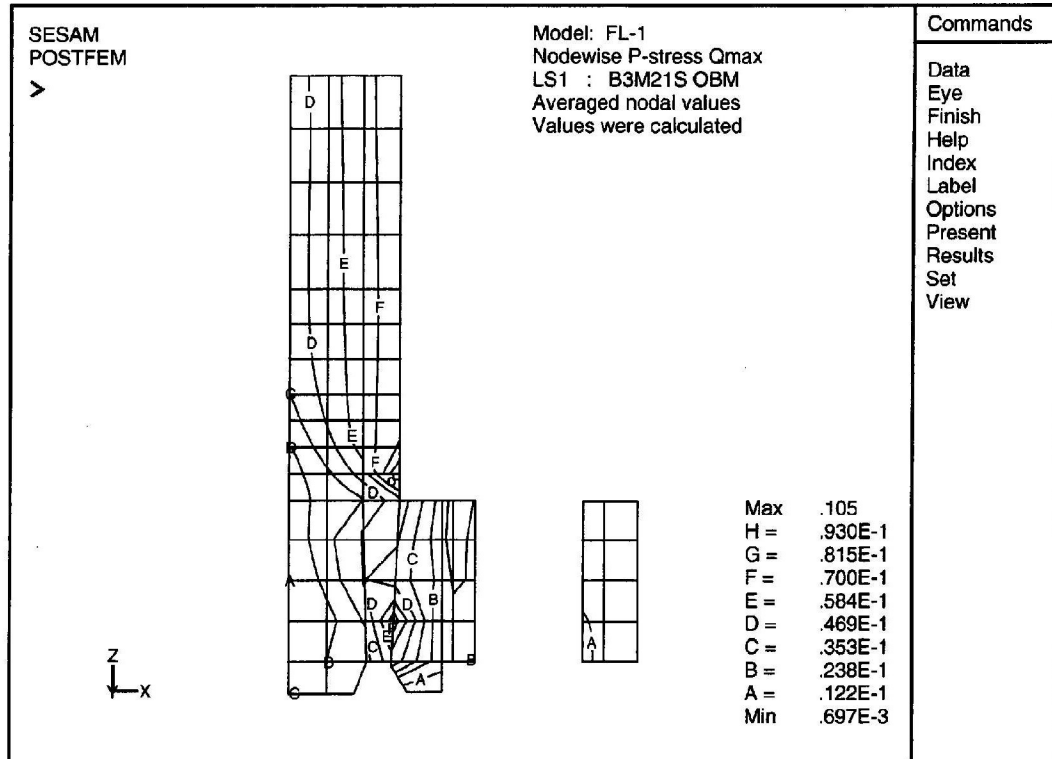
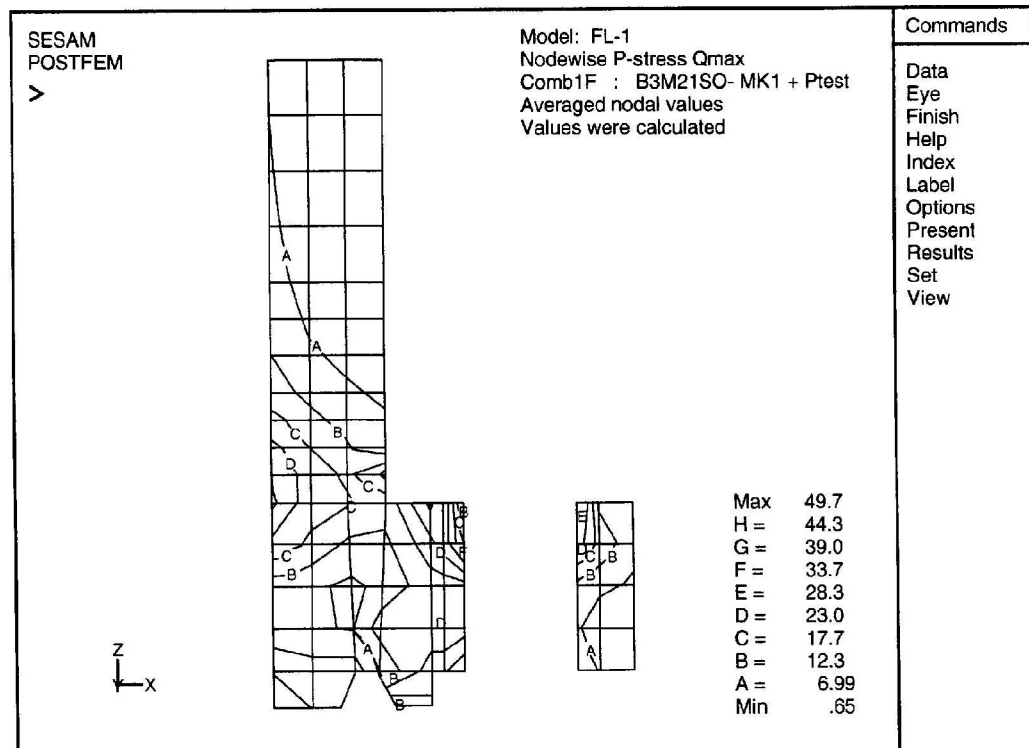
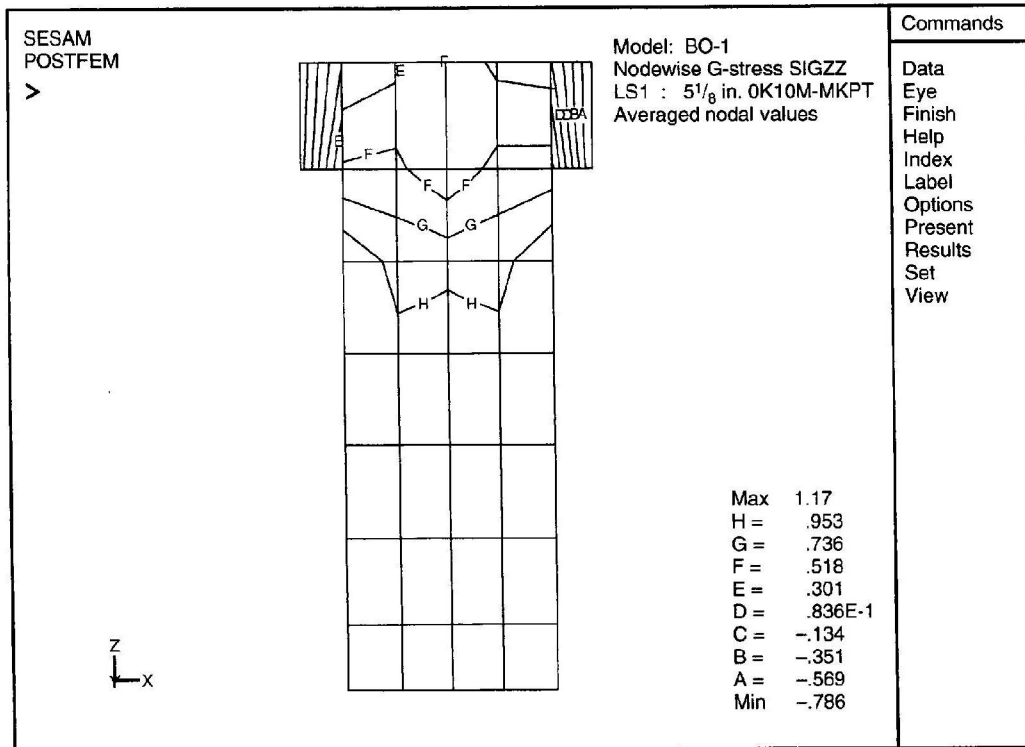
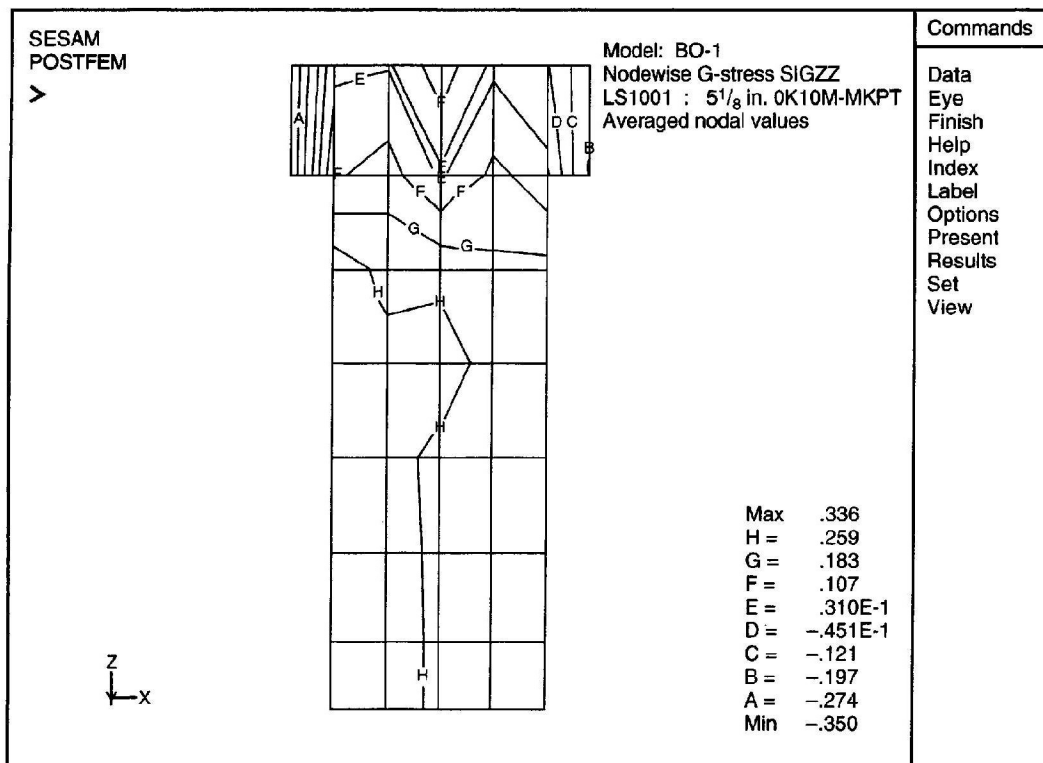
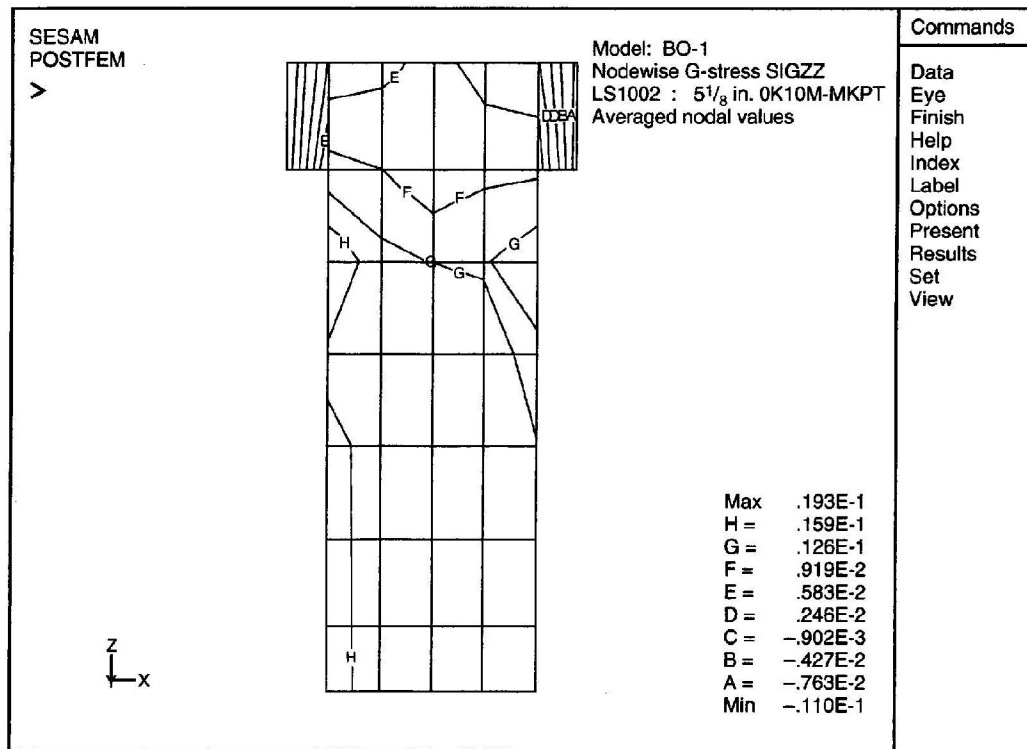
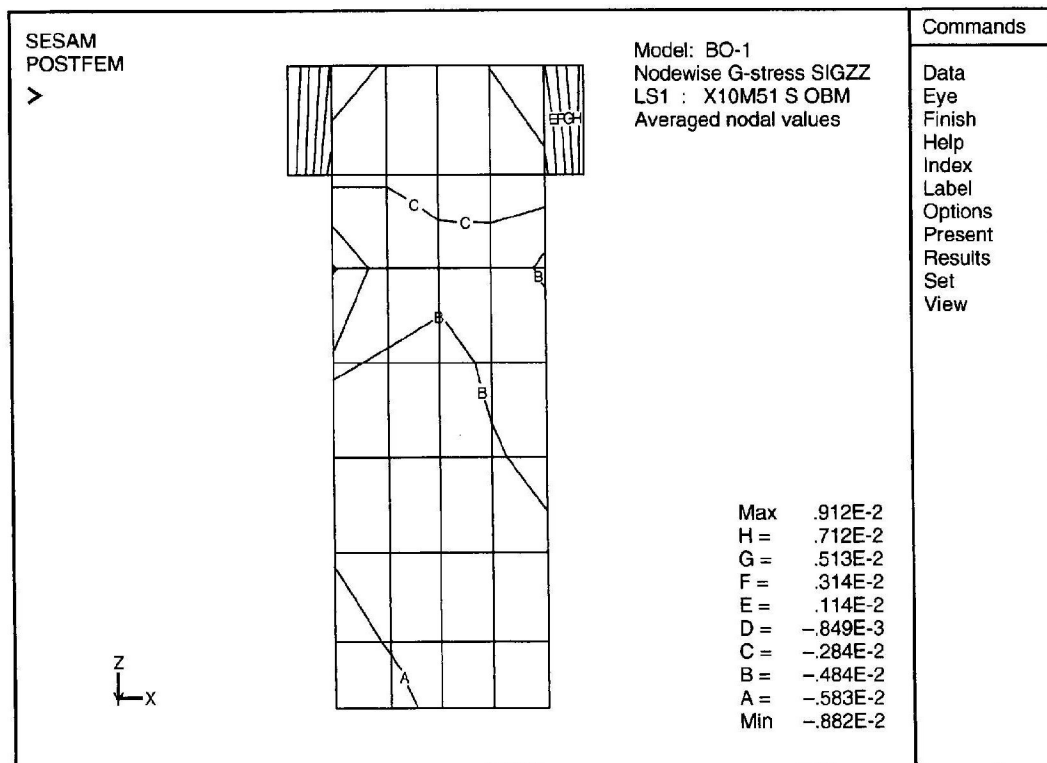


Figure 14—Maximum Shear Stress Contours Due to Unit Bending Moment


 Figure 15—Von Mises's Equivalent Stress Contours for 2 ¹/₁₆ in. 3,000 psi Type 6B Flange Under 52.5 ksi Makeup and 6,000 psi Test Pressure

Figure 16—Bolt Stress (σ_z) Contours for Unit MakeupFigure 17—Bolt Stress (σ_z) Contours for Unit Pressure


 Figure 18—Bolt Stress (σ_z) Contours for Unit Tension

 Figure 19—Bolt Stress (σ_z) Contours for Unit Bending Moment

4 Two-dimensional Heat Transfer Analysis

In addition to the four unit load conditions described in the previous section, the effect of temperature difference on the stresses was considered. A heat conduction analysis was carried out to determine the temperature distribution throughout the flange. A two-dimensional model was adequate since this load condition is fully axisymmetric. Four flanges were analyzed to determine the temperature distribution in a small and large bore flange in the Type 6B and Type 6BX series. The temperature distribution in the other flanges was interpolated using a computer program that was specifically written for this purpose. The program TEMP converts the temperature distribution into nodal temperatures at the stress analysis mesh nodes.

4.1 Finite Element Modeling

The finite element mesh used for the heat conduction analysis was similar to that used for the stress analysis with the exception of using more elements in the radial direction to more accurately determine the temperature distribution. Figure 20 shows typical meshes generated for Type 6BX flanges. The program used for the heat conduction analyzed is called FETAB and was developed at the University of Calgary. The models shown in Figure 20 had 85 and 136 nodes for the small, 2 ¹/₁₆ and the large, 16 ³/₄ Type 6BX flanges, respectively.

The following properties were used in the analysis:

Convection coefficient $k_{xx} =$	k_{yy}	$= 2.027$
Capacity	c	$= 0.1102$
Density	ρ	$= 0.284 \text{ lb/in.}^3$
Coefficient of thermal expansion	α	$= 6.6 \times 10^{-4} / ^\circ\text{F}$
Young's modulus	E	$= 29 \times 10^6 \text{ psi}$

It should be mentioned that the temperature boundary conditions were selected as follows:

- a) all external surfaces were set to be at 30°F;
- b) all internal surfaces up to the gasket line were assumed to have the internal temperature of 250°F;
- c) the interface surface and the hub end sections were assumed to be fully insulated.

4.2 Temperature Analysis Results

The temperatures at the nodal points were output by FETAB in tabular form as shown in Table 1, which gives the results for both flanges shown in Figure 20. Similar results were obtained for the Type 6BX flanges.

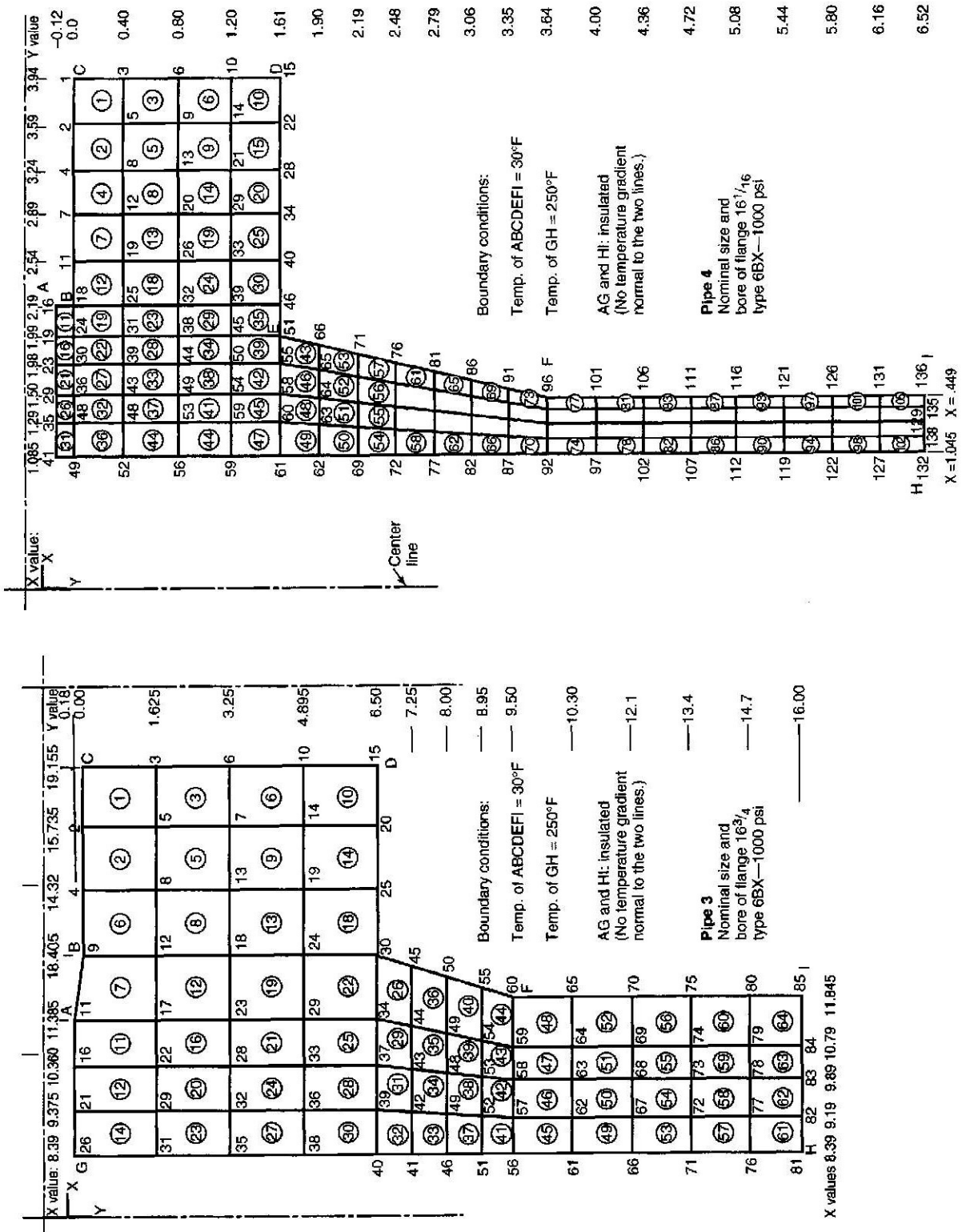


Figure 20—Typical Heat Conduction 2-D Mesh for Type 6BX Flanges

Table 1—Nodal Temperatures for Typical Type 6BX Flanges

16 3/4 in. 10,000 psi 6BX											
Node	Temp	Node	Temp	Node	Temp	Node	Temp	Node	Temp	Node	Temp
1	30.000	16	161.395	31	250.000	46	250.000	61	250.000	76	250.000
2	45.227	17	126.935	32	196.549	47	185.057	62	185.450	77	184.383
3	30.000	18	81.348	33	144.725	48	131.075	63	130.795	78	129.239
4	64.637	19	46.765	34	89.779	49	85.080	64	83.323	79	82.241
5	44.418	20	30.000	35	250.000	50	30.000	65	30.000	80	30.000
6	30.000	21	200.701	36	191.553	51	250.000	66	250.000	81	250.000
7	91.204	22	159.363	37	132.618	52	185.900	67	184.669	82	184.371
8	63.040	23	120.294	38	250.000	53	131.919	68	129.613	83	129.223
9	41.711	24	67.809	39	185.082	54	84.918	69	82.539	84	82.229
10	30.000	25	30.000	40	250.000	55	30.000	70	30.000	85	30.000
11	130.346	26	250.000	41	250.000	56	250.000	71	250.000		
12	88.250	27	199.595	42	184.859	57	187.384	72	184.441		
13	57.834	28	153.931	43	131.459	58	134.175	73	129.322		
14	36.732	29	106.895	44	86.598	59	87.787	74	82.302		
15	30.000	30	30.000	45	30.000	60	30.000	75	30.000		
2 1/16 in. 10,000 psi 6BX											
Node	Temp	Node	Temp	Node	Temp	Node	Temp	Node	Temp	Node	Temp
1	30.000	24	111.724	47	250.000	70	71.096	93	181.872	116	30.000
2	36.071	25	91.494	48	197.675	71	30.000	94	124.191	117	250.000
3	30.000	26	66.692	49	162.360	72	250.000	95	74.733	118	186.592
4	44.069	27	42.189	50	149.700	73	172.559	96	30.000	119	125.231
5	35.684	28	30.000	51	250.000	74	113.549	97	250.000	120	73.772
6	30.000	29	160.048	52	250.000	75	67.478	98	186.579	121	30.000
7	55.196	30	132.516	53	199.540	76	30.000	99	125.156	122	250.000
8	43.271	31	108.395	54	170.117	77	250.000	100	73.589	123	186.591
9	34.496	32	87.874	55	153.808	78	172.783	101	30.000	124	125.231
10	30.000	33	54.778	56	250.000	79	113.664	102	250.000	125	73.773
11	71.173	34	30.000	57	201.808	80	67.341	103	186.581	126	30.000
12	54.172	35	197.521	58	160.787	81	30.000	104	125.224	127	250.000
13	40.691	36	160.082	59	250.000	82	250.000	105	73.798	128	186.591
14	32.536	37	132.171	60	194.730	83	174.757	106	30.000	129	125.231
15	30.000	38	110.392	61	250.000	84	115.343	107	250.000	130	73.772
16	101.612	39	97.289	62	250.000	85	68.241	108	186.594	131	30.000
17	111.243	40	30.000	63	184.979	86	30.000	109	125.233	132	250.000
18	96.513	41	250.000	64	138.787	87	250.000	110	73.768	133	186.592
19	69.822	42	197.498	65	102.067	88	176.967	111	30.000	134	125.231
20	50.293	43	159.847	66	30.000	89	117.884	112	250.000	135	73.772
21	36.139	44	131.085	67	250.000	90	69.680	113	186.591	136	30.000
22	30.000	45	97.937	68	175.980	91	30.000	114	125.230		
23	132.770	46	30.000	69	119.697	92	250.000	115	73.773		

5 Limiting Criteria for Combined Loading

5.1 Introduction

To calculate the capacity for each flange for carrying pressure, tension and bending loads at a specific initial makeup, a failure criterion must be adopted. For the flanges analyzed herein, these failures are divided into two main categories:

- a) leakage; and
- b) stress exceedance.

Both failure categories are discussed in more detail in 5.2.

It must be recognized that the analysis presented herein allows only an approximate evaluation of the first type of failure; namely, the leakage. This is due to the approximation inherent in modeling the gasket as a line support for the flange, but beyond the resources available for the present work. The behavior of the gasket is, in reality, nonlinear; and therefore a nonlinear model would have been more appropriate.

With regard to the stress failure criterion, we believe that the main unknown is not the stress field, but rather, the definition of the criteria against which these stresses are to be compared. Similar to the previous project PRAC 86-21, it has been agreed to adopt the ASME *Section VIII, Division 2 Pressure Code* criteria and the API material yield and stress allowables. Although this is quite reasonable, as these codes have been based on extensive experience over the past three or four decades, it should be recognized that when it comes to applying the code, the analyst/designer is faced with the question of how to categorize the stresses in the form of primary, secondary, local, self-limiting, etc. Although in some cases these decisions are obvious, situations exist where the division lines are not quite so obvious. Therefore, as will be shown herein, whenever such a condition was faced, the committee has always selected the more conservative option, even if it meant being more conservative than that of the code.

5.2 Leakage Criterion

The leakage failure is assumed to occur when the gasket reaction reaches a value of zero. This indicates that the makeup compressive forces in the gasket have all been exhausted at a point on the gasket, signaling that lift-off, and consequently, leakage is incipient.

It should be emphasized that the assumption of a rigid point support gasket may be significantly conservative. A nonlinear procedure in which the gasket is modeled would be more realistic. The contact between the gasket and the front face of the flange in such a model would be monitored, and failure would occur when contact is lost at any radial location.

The gasket reaction was calculated at a discrete point as:

$$R_g = R_{mu} \sigma_{mk} + R_{pu} P + R_{tu} T + R_{bu} M \quad (2)$$

where

R_{mu} is the unit reactions for makeup;

R_{pu} is the unit reactions for pressure;

R_{tu} is the unit reactions for tension;

R_{bu} is the unit reactions for bending moment;

σ_{mk} is the magnitude of the makeup;

P is the magnitude of the pressure;

T is the magnitude of the tension;

M is the magnitude of the bending moment.

Leakage is assumed to occur when R_g vanishes, i.e.:

$$P = -(R_{mu} \sigma_{mk} + R_{tu} T + R_{bu} M) / R_{pu} \quad (3)$$

which is a linear relationship between p and M for given values of makeup and end tension. The API End Connection Task Group agreed to the presentation of the load combination charts in the format shown in Figure 21 for two makeup conditions of 52.5 and 40 ksi, which correspond to bolt material yield strength of 105 ksi and 80 ksi, respectively.

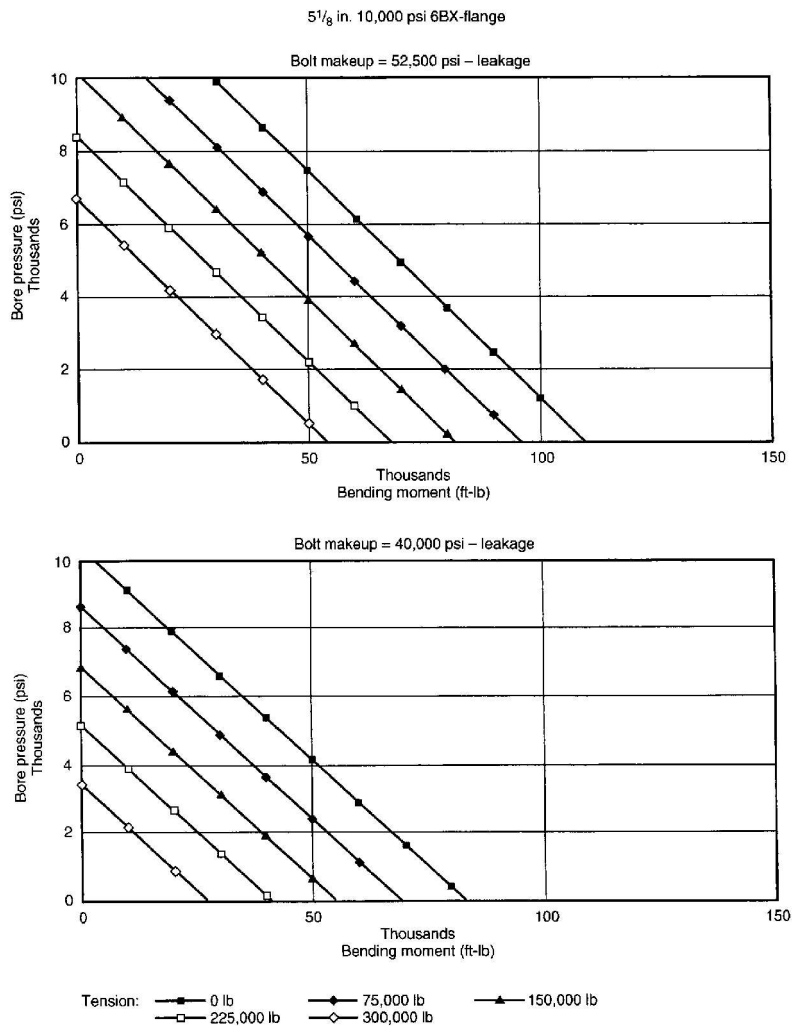


Figure 21—Typical Leakage Load Combination Charts for 52.5 ksi and 40 ksi Makeup Loads^a

a. Users of these graphs should not rely exclusively on the information contained therein. Sound business, scientific, engineering, and safety judgement should be used in employing this information.

5.3 Stress Criterion

The stresses are checked in hub and flange sections according to the ASME *Section VIII, Division 2 Pressure Vessel* Code criterion, and the API 6A material yield and stress allowables. Six loading conditions were identified and are summarized in Table 2. As noted, the ASME criterion employs the concept of stress intensity, which is defined as twice the maximum shear stress; which in turn is equal to the algebraic difference between the maximum and the minimum principal stresses:

$$SI = 2.\tau_{\max} = \sigma_1 - \sigma_3 \quad (4)$$

Table 2—Adopted Stress Criteria

			Allowable Stress Intensity			
Load Condition			Membrane Component SI_m		Membrane and Bending SI_{mb}	
No.	Description	ASME Classification	Flange Sections	Hub Sections	Flange Sections	Hub Sections
1	Makeup only	Primary	S_m^a	S_m	$1.5 S_m$	$1.5 S_m$
2	Test pressure only ^d	Primary	S_T^c	S_T	$1.35 S_y^b$	$1.35 S_y$
3	Makeup	Primary	S_T	—	$1.35 S_y$	—
	plus test pressure	Secondary	—	$1.5 S_m$	—	$3.0 S_m$
4	Work pressure					
	plus tension plus bending moment	Primary	S_m	S_m	$1.5 S_m$	$1.5 S_m$
5.	Makeup					
	plus work pressure plus tension	Primary	S_m	—	$1.5 S_m$	—
	plus bending moment	Secondary	—	$1.5 S_m$	—	$3.0 S_m$
6	makeup					
	plus work pressure plus tension plus bending moment plus temperature	Secondary	$1.5 S_m$	$1.5 S_m$	$3.0 S_m$	$3.0 S_m$

^a S_m = Allowable membrane stress intensity, = $2/3 S_y$.
^b S_y = Yield strength = 60 ksi for up to 10,000 psi work pressure, or = 75 ksi for 15,000 psi and 20,000 psi work pressure.
^c S_T = Allowable testing stress intensity, = $0.83 S_y$.
^d Test pressure = $2 \times$ work pressure for < 14 in. flanges or < 10,000 psi working pressure, or = $1.5 \times$ work pressure for > 14 in. flanges or $\geq 10,000$ psi working pressure.

In addition, the criterion employs two characteristic values of stress intensities representing a particular section of unit width through the material. These are:

- a) SI_m = membrane stress intensity defined as the average SI across the section.
- b) SI_{mb} = membrane plus bending stress intensity which is defined as $SI_m + SI_b$ where SI_b is the bending component calculated as the linear value that gives the same bending moment as that corresponding to the actual stress distribution across the section.

The procedure employed in calculating the above stress intensities is summarized as follows.

- a) At each of the sections checked, the distribution of the six stress components due to the four unit loading conditions are extracted from the finite element results file.
- b) The total stresses due to the load combination under consideration is then obtained:

$$\sigma = \sigma_{mu} \sigma_{mk} + \sigma_{pu} P + \sigma_{tu} T + \sigma_{bu} M \quad (5)$$

The stresses are obtained at the Gauss points and are assumed to be linearly distributed between each pair of adjacent Gauss points.

- c) Each component of the stress vector, s is then linearized across the section as shown in Figure 22. The membrane component σ_m is calculated as the average stress and the bending component σ_b as the edge stress corresponding to a linear stress distribution which has the same resultant bending moment as the actual distribution:

$$\sigma_m = \frac{1}{t} \int \sigma_x dx \quad (6)$$

$$\sigma_b = \frac{6}{t^2} \int x \sigma_x dx \quad (7)$$

- d) The resulting linearized stress vectors (σ_m) At the midpoint of the section and (σ_{mb}) at the near and far edge points are then employed to calculate the principal stresses and the corresponding stress intensities.

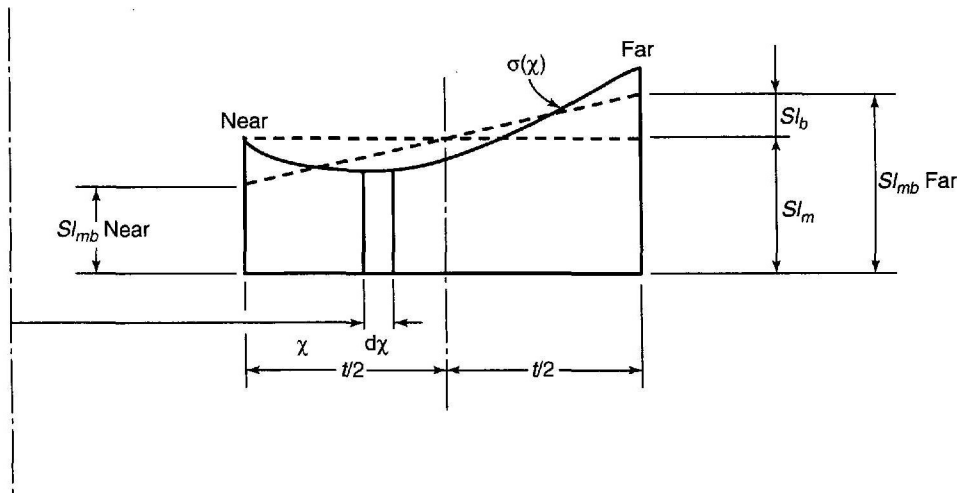


Figure 22—Stress Linearization

In addition to the criteria depicted in Table 2, the triaxial stress condition—that the algebraic sum of the principal stress should be smaller than $4 S_m$ for all primary loading conditions—was also checked.

It should be noted that in this report, criterion No. 4 in Table 2 was interpreted as representing a load combination composed of pressure, tension, and bending moment, in addition to an initial makeup value calculated to balance these forces without leakage. In a previous project (API 6AF), this condition was interpreted differently. It assumed that these three actions existed without any makeup, i.e. the criterion is applicable only to the stress components caused by these three forces. As a result of this difference in interpretation, the load combination capabilities of a number of flanges were found to be in disagreement. The results of the present work reflect the latest thinking of the

API oversight committee and are, therefore, recommended. These results are more conservative than previously published results. A more detailed discussion of the differences and the specific flanges where these differences were found to be significant are given in Annex A.

5.4 Development of Load Combination Charts

5.4.1 The load combination charts program LCCP was written to postprocess the finite element stresses at the critical sections considered. Figure 23 shows the location of these sections for a Type 6B and a Type 6BX flange. The criterion given in Table 2 is then checked and a report is given on the result of each check carried out. The program employs either the hub or flange allowables, depending on a section type code input before the criteria checking is carried out.

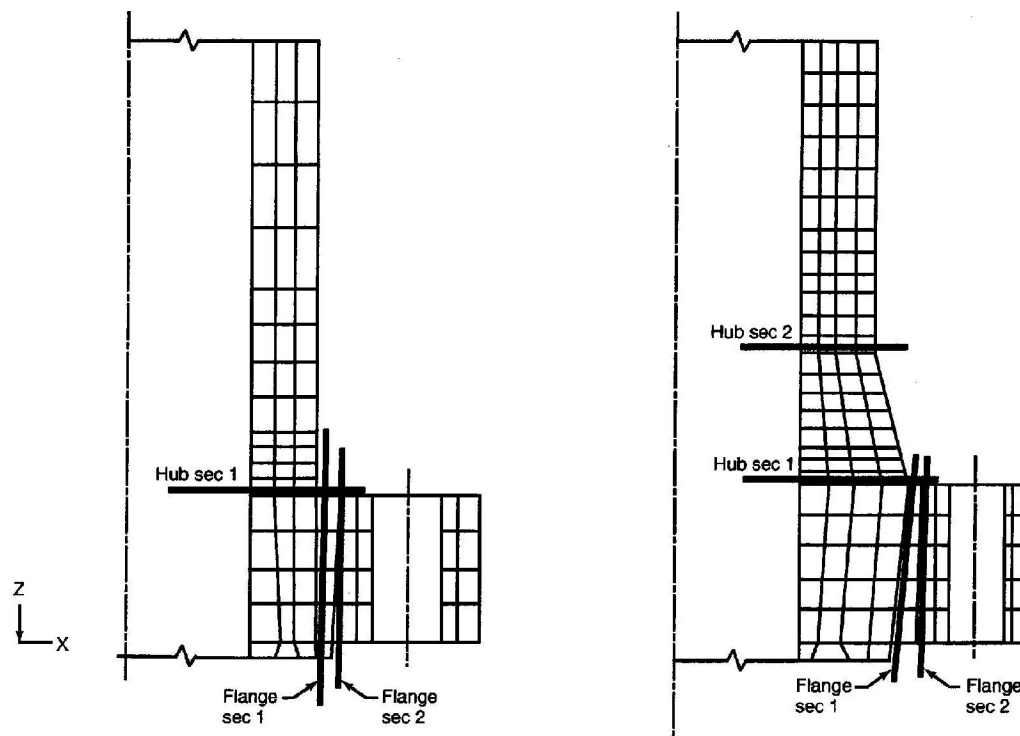


Figure 23—Critical Section Locations

The procedure programmed in FORTRAN in LCCP is summarized in the following steps.

- Read in the bore diameter; the working pressure; the makeup values (two values); the pressure, tension, and bending moment ranges and stepping increments; the number of section to be checked out; the type of each section; the number of elements across each section; and the criterion for controlling the convergence of the root-searching algorithm that iterates to determine the first principal stress.
- Read in the unit stresses (at the Gauss points in each element) for the four unit load cases and the temperature load case, if so required.
- The program determines the correct material yield and test pressure depending on the specified working pressure and bore dimension for the flange under consideration.
- Check the criteria for makeup only; test pressure only; and makeup plus test pressure; and report the results.

- e) Calculate the load combination charts for each section and each value of makeup. This is carried out by selecting a value for the tension and a value for the bending moment within the specified range and iterating over the pressure from the minimum value with the specified increment until one of the criteria is exceeded. The program interpolates to determine the pressure value within the increment at which the criterion was exceeded. The procedure described in 5.3 is used to determine the membrane and membrane plus bending stress intensities against which the criteria are checked.
- f) Print out the results. Two types of listings are produced by LCCP. The first is a long listing giving all the points on the load combination charts with specific indicators as to which criterion was exceeded and the corresponding values of the stress intensities and the value of the makeup in case criterion No. 4 of Table 2 was governing. The second printout gives only the points on the curve in the form of a matrix with its first column as the bending moment values (x-axis values on the charts), and the other columns giving the corresponding limiting pressures for each tension value.

The results from the LCCP are input into a Lotus 1-2-3 worksheet which is also used to check and plot the linear leakage criterion.

5.5 Results

Figure 24 and Figure 25 show the leakage load combination charts for the $2\frac{1}{16}$ in. 3,000 psi Type 6B flange and the $16\frac{3}{4}$ in. 3,000 Type 6B flange.

Comparison of the results indicate that the agreement between the 2-D and 3-D analysis is better for larger flanges that have an increased number of bolts, since the 2-D assumption of continuous bolt holes becomes more reasonable than for flanges with fewer bolt holes. The figure also indicates that the 3-D model is more conservative.

Figure 26 and Figure 27 give the load combination charts for the same flanges as limited by the stress criterion. For each flange, two charts are produced; one for leakage and the other for stress criterion. The designer must take the minimum of the two conditions. The governing criteria for the stress condition was the hub section closest to the flange and applies the primary stress condition No. 4 of Table 2.

It should be noted that the value of the makeup required for the $2\frac{1}{16}$ 3,000 psi Type 6B flange of Figure 26 ranged between 66 ksi and 75 ksi; and for $16\frac{3}{4}$ in. 3,000 psi Type 6B flange the makeup range was 42 ksi to 45 ksi.

It should be noted that because criterion No. 4 of Table 2 is almost always governing, the stress criterion curves are duplicated so it appears in the same format as the leakage charts. The value of the makeup used for each point on the curve was calculated using the unit load gasket reactions such that the applied loading can be resisted without leakage. The effect of including the temperature stresses on the leakage and stress charts is shown in Figure 28 and Figure 31. In Figure 30 and Figure 31, the effect of the temperature is shown as the second half of the chart, again because the minimum makeup condition was the governing case.

The reason the minimum makeup condition was almost always found to govern is that for the hub sections, the stresses due to pressure, tension and bending moments were considered to be primary, while those with makeup specified at 40 or 52.5 ksi were treated as secondary.

It should also be noted that the temperature effect has always been favorable; i.e. it increases the capacity of the flanges to resist leakage or stresses. The reason it improves the leakage situation is that temperature gives a compressive reaction on the gasket. The stress criterion is increased because of the increase in the allowable stress limits when the temperature is included. The temperature gradient, however, increased the stresses in the bolts by about 7 % to 9 %. Therefore, it is concluded that the bolt stresses would not be governed by the temperature effect. The contours for the axial stress in the bolt due to the temperature gradient are shown in Figure 32 and Figure 33 for the $5\frac{1}{8}$ in. 10,000 psi Type 6BX and the 11 in. 10,000 psi Type 6BX flanges, respectively.

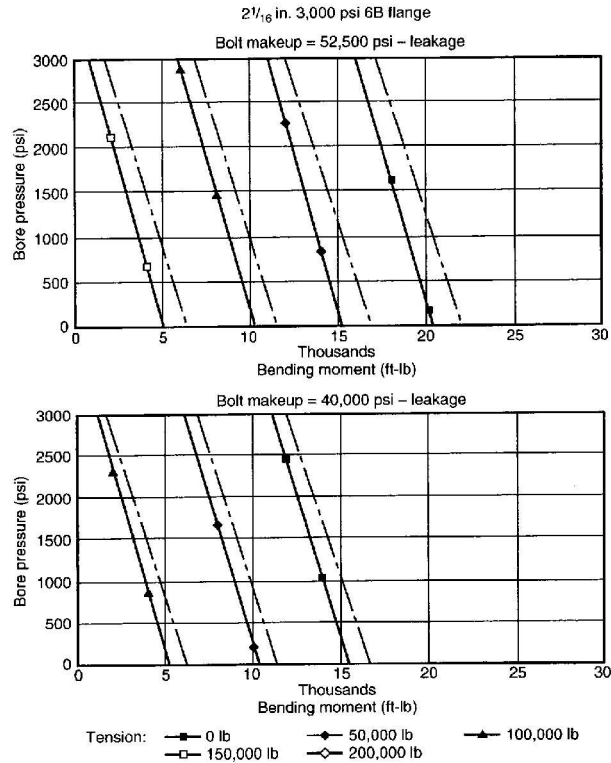


Figure 24—Comparison Between 2-D and 3-D Results for Leakage Criteria for 2 1/16 in. 3,000 psi 6B Flange^a

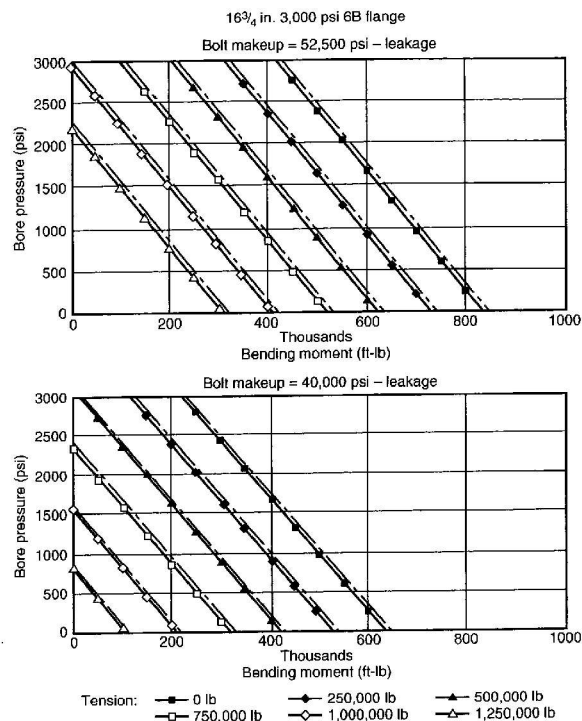


Figure 25—Comparison Between 2-D and 3-D Results for Leakage Criteria for 16 3/4 in. 3,000 psi 6B Flange^a

a. Users of these graphs should not rely exclusively on the information contained therein. Sound business, scientific, engineering, and safety judgement should be used in employing this information.

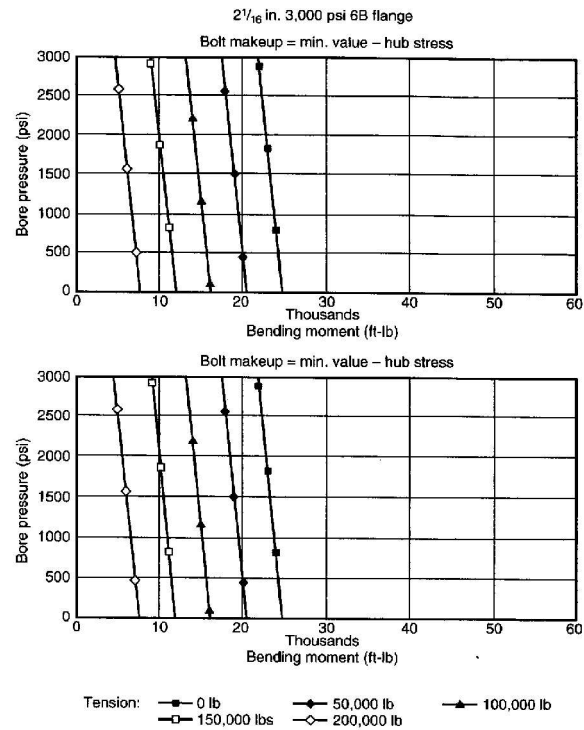


Figure 26—Stress Limiting Criteria for 2 1/16 in. 3,000 psi 6B Flange^a

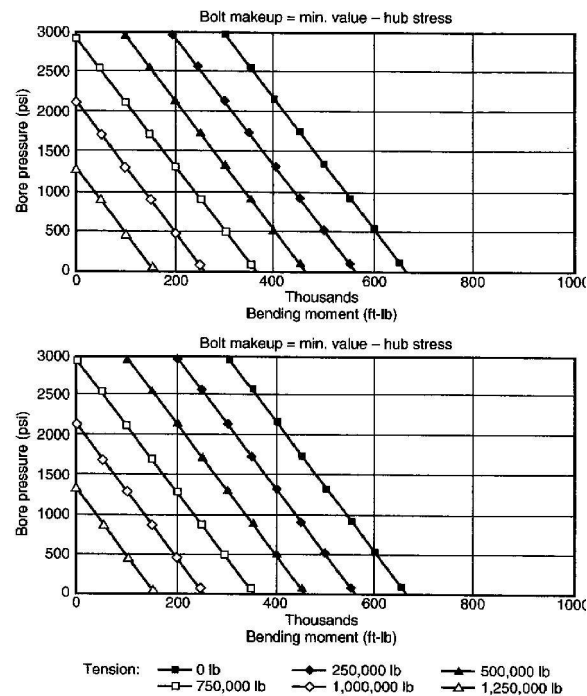


Figure 27—Stress Limiting Criteria for 16 3/4 in. 3,000 psi 6B Flange^a

a. Users of these graphs should not rely exclusively on the information contained therein. Sound business, scientific, engineering, and safety judgement should be used in employing this information.

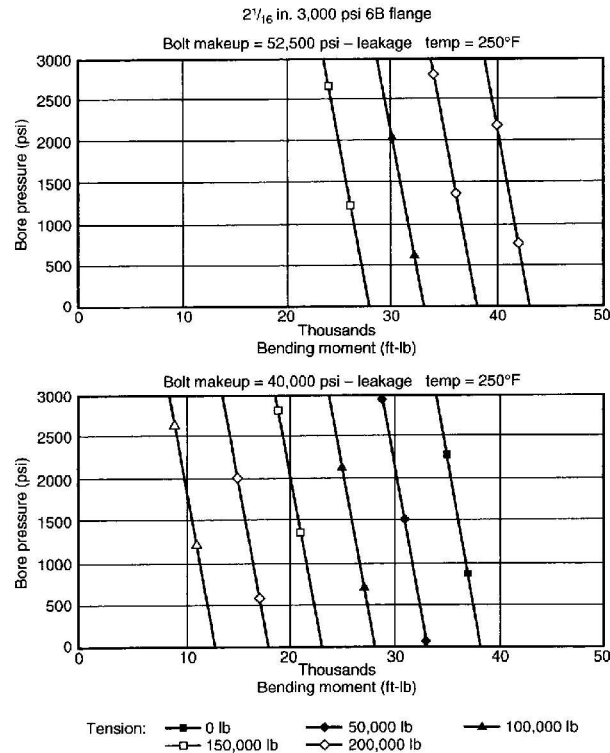


Figure 28—Leakage Criteria with Temperature Effect for 2 1/16 in. 3,000 psi 6B Flange^a

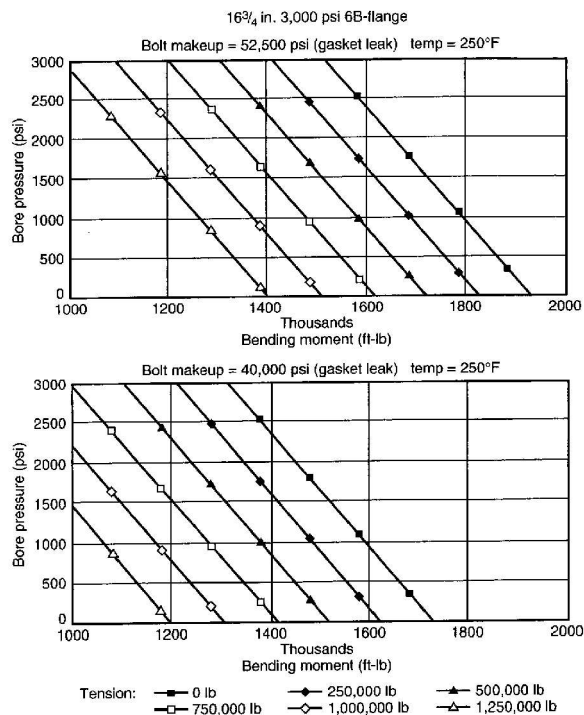


Figure 29—Leakage Criteria with Temperature Effect for 16 3/4 in. 3,000 psi 6B Flange^a

- a. Users of these graphs should not rely exclusively on the information contained therein. Sound business, scientific, engineering, and safety judgement should be used in employing this information.

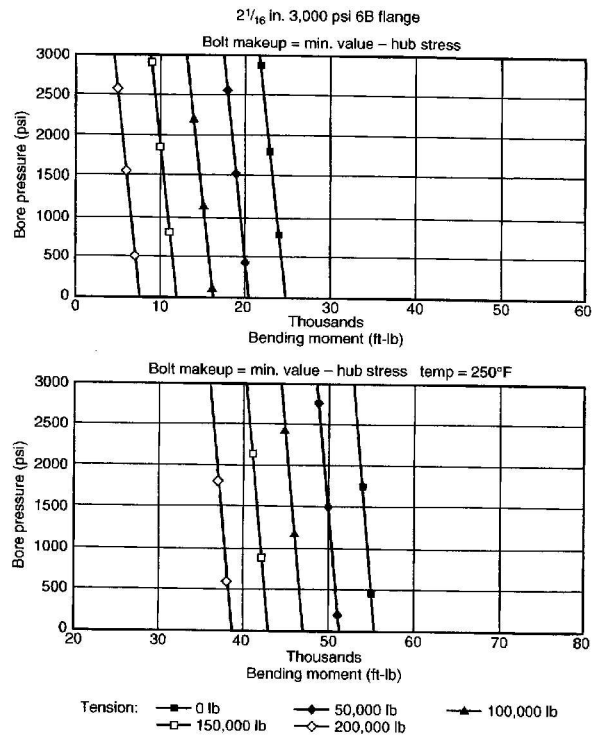


Figure 30—Stress Criterion with Temperature Effect for $2\frac{1}{16}$ in. 3,000 psi 6B Flange^a

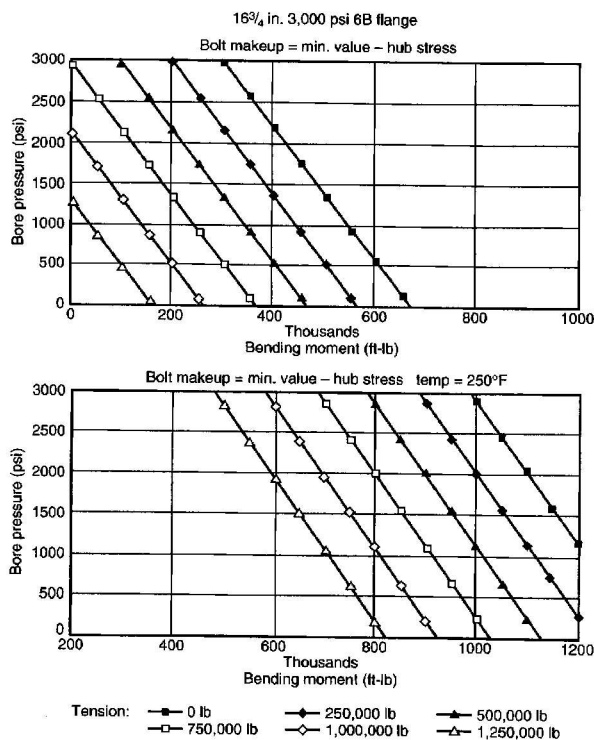


Figure 31—Stress Criterion with Temperature Effect for $16\frac{3}{4}$ in. 3,000 psi 6B Flange^a

a. Users of these graphs should not rely exclusively on the information contained therein. Sound business, scientific, engineering, and safety judgement should be used in employing this information.

For investigated flanges, the bolt stresses due to pressure, tension, and bending moment increased the makeup value of 52.5 ksi to a value from 77 ksi to 82 ksi, which is 88 % to 94 % of the allowable $0.83 S_y$. The temperature added another 10 ksi to this stress bringing the total to 85 ksi to 92 ksi, which is about 80 % to 88 % of the yield.

6 Reanalysis of Failed Flanges

It should also be pointed out that the four flanges that failed the makeup-only criterion (No. 1 of Table 2) also failed in the present analysis. However, two additional flanges did not meet the makeup-only condition in this project. A list of these flanges is shown in Table 3.

To bring the flanges that did not meet the ASME criterion for the makeup-only condition, several flanges were reanalyzed with the elimination of the raised face. However, since the governing section was in the hub, this modification resulted in a reduction in the stress intensities of approximately 5 %. Therefore, such an elimination of the raised face would be adequate for the 16 ³/₄ in. 3,000 psi 6B flange. In addition to this modification, however, it would be necessary to increase the thickness of the hub to the values given in Table 4 to reduce the stress intensities to the allowable value of 60 ksi for the 52.5 ksi makeup load. As noted, the hub thickness increased by anywhere from zero to 27 % with the elimination of the raised face; and from zero to 33.5 % with the raised face unchanged.

7 Effect of Lockdown Screw Holes on Strength

Due to the fact that many flanges are manufactured with lockdown screw holes, and that these holes can be of fairly large size compared to the flange dimensions, an analysis of this effect on the capability of the flanges was carried out.

Four flanges were analyzed with radial holes midway between each two-bolt holes. The diameter of radial holes was given by the relevant task group to be as shown in Table 5. Three positions were considered for each case:

- a) at 0.2 times the thickness above the centerline,
- b) at centerline, and
- c) at 0.2 times the flange thickness below the centerline.

Figure 34 shows a view of the finite element mesh with lockdown screw holes present. The effect of the lockdown screw holes on the combined load capacity of the 7 ¹/₁₆ in. 5,000 psi Type 6B flange is shown in Figure 35 and Figure 36 for the leakage and stress criterion, respectively. As shown, the capability to resist leakage is increased due to the fact that flange stiffness is reduced by the presence of these holes; therefore, less makeup load is transferred through the flange to the hub and higher gasket reaction occurs. This revised load path, however, did not change the location of the most critical section with respect to the stress criterion. This section is the hub section at the center line of the bolt holes. The combined load-carrying capability with regard to stresses is reduced due to the effect of the holes in reducing the strength of the flange.

The leakage criterion results shown in Figure 35 indicate that, on the average, the values of the maximum bending moment that can be resisted at a given end tension is increased by 6,250 ft-lb, which is only 3.4 % of the maximum bending moment capacity at zero tension and zero pressure.

The decrease in the stress carrying capability, which is the governing condition for this flange as depicted in Figure 36, is somewhat more significant. The decrease in the maximum bending moment that can be resisted is approximately 10,400 ft-lb, or 9.3 % of the maximum bending capacity at zero moment and zero pressure.

An analytical evaluation of the effect of the location of the holes with respect to the center of the flange was carried out. Table 6 shows the effect of the presence of the lockdown screw holes on the bending stiffness of the flange. The values shown are for minimum sections at the bolt circle for a 30° sector with a full lockdown screw hole.

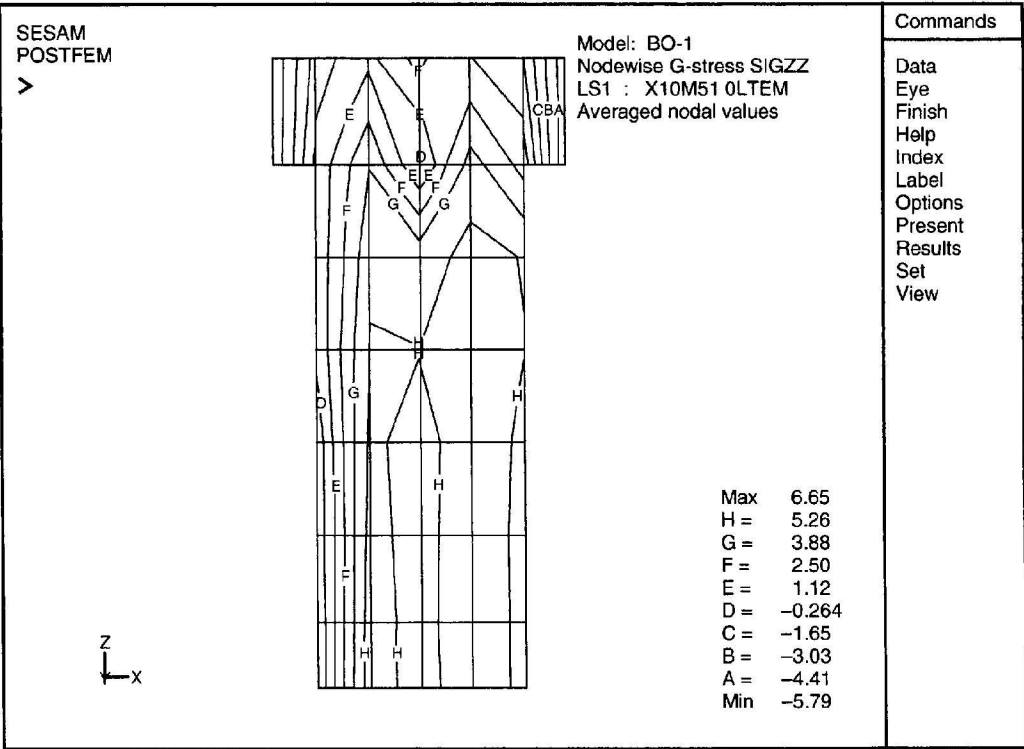


Figure 32—Bolt Stress Contours for 5 1/8 in. 10,000 psi Type 6BX Flange

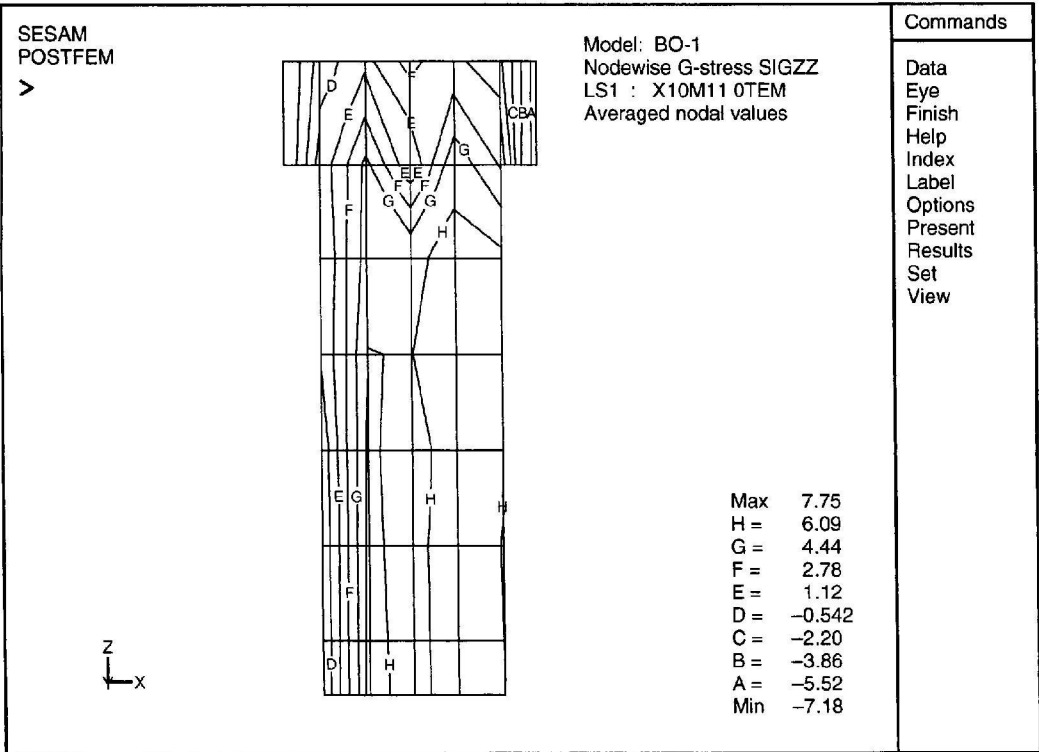


Figure 33—Bolt Stress Contours for 11 in. 10,000 psi Type 6BX Flange

Table 3—Flanges That Did Not Meet Makeup-only Criterion, Makeup = 52.5 ksi

No.	Flange Specifications	Location	3DFEM SI (ksi)	2DFEM SI (ksi)
8	13 5/8 in. 2,000 psi 6B	ID	70.35	65.91
		OD	51.73	61.85
9	16 3/4 in. 2,000 psi 6B	ID	72.02	67.06
		OD	54.86	65.03
10	21 1/4 in. 2,000 psi 6B	ID	82.40	62.82
		OD	62.53	60.99
18	13 5/8 in. 3,000 psi 6B	ID	66.66	65.37
		OD	49.15	60.07
19	16 3/4 in. 3,000 psi	ID	62.21	<60
		OD	46.53	<60
20	20 3/4 in. 3,000 psi	ID	65.51	<60
		OD	48.02	<60

Table 4—Flange Geometry Modifications to Meet ASME Criterion for 52.5 ksi Makeup

No.	Flange Specifications	Original Hub Thickness (in.)	Revised Hub Thickness (in.)	
			With No Raised Face	With Raised Face
8	13 5/8 in. 2,000 psi 6B	1.045	1.15	1.21
9	16 3/4 in. 2,000 psi 6B	1.360	1.50	1.59
10	21 1/4 in. 2,000 psi 6B	1.360	1.73	1.82
18	13 5/8 in. 3,000 psi 6B	1.420	1.49	1.57
19	16 3/4 in. 3,000 psi 6B	1.610	1.59	1.66
20	20 3/4 in. 3,000 psi 6B	1.860	1.92	2.01
Alternatively, it is recommended that the initial makeup for these flanges should be limited to the values listed below. These revised makeup values should be used whenever the wrench clearances become unacceptable due to the hub thickness increase of Table 4.				
No.	Flange Specification	Revised Maximum Initial Makeup (ksi)		
8	13 5/8 in. 2,000 psi 6B	44.5		
9	16 3/4 in. 2,000 psi 6B	43.5		
10	21 1/4 in. 2,000 psi 6B	38.0		
18	13 5/8 in. 3,000 psi 6B	47.0		
19	16 3/4 in. 3,000 psi 6B	50.5		
20	20 3/4 in. 3,000 psi 6B	48.0		

Table 5—Lockdown Screw Holes

No.	Flange Specification	Hole Diameter (in.)
18	13 5/8 in. 3,000 psi 6B	1.642
25	7 1/16 in. 5,000 psi 6B	1.505
40	7 1/16 in. 10,000 psi 6BX	1.620
55	13 5/8 in. 15,000 psi 6BX	1.642