MECHANICAL SUPPORT DEPARTMENT
ENGINEERING NOTE

NUMBER: MSD EN-1115 - KTEV
MSD EN-1.1.1.6-KTEV

DATE: 25 May 1995

TITLE: KTEV 1.8 meter Vacuum Window and Vacuum Window Test Fixture Engineering Computations

AUTHOR(s): Andrew Szymulanski

REVIEWED BY: KTEV Mechanical Safety Review Panel, Joel Misek, Chairman

DATE REVIEWED: January 19, 1993

KEY WORDS: KTEV Vacuum Window structural components, Stress Analysis

ABSTRACT/SUMMARY:
This documentation contains an enginering stress analysis of the window assembly and the window testing fixture. The result of this was a base for starting the vacuum, impact, hydrostatic and long term creep testing. The conclusion of these tests is an essential approval element of using the window ssembly in the KTEV experiment.

January 19, 1993

TO: R. Dixon - Research Division Head

FROM: J. Misek - KTEV Mechanical Safety Review Panel Chairman JM

SUBJECT: KTEV Mechanical Safety Review - Window Test Vessel

Pursuant to the Panel's charge given to us on 30 June 92 by Peter Garbincious to review, "all significant mechanical components for relevant safety standard compliance," the Panel has reviewed documentation on the KTEV 1.8 m. dia. window test vessel. While this vessel is exempt from any specific requirements of the Fermilab Safety Manual, the Panel has reviewed documentation submitted to us and has concluded that adequate engineering analysis has been done to ensure its safe operation. The Panel has no objections in allowing fabrication to proceed on the test vessel per the drawings submitted to us.

Testing of the full size window will take place when this test vessel is ready. The window design appears to be consistent with existing window design and the upcoming test program should provide assurances to its safety and design limitations. Unless you feel it is necessary, the Panel will not make any recommendations to you regarding the window testing program. The Panel will review this activity and respond directly to the project engineer. A formal recommendation will be made by the Panel to you before production window operation.

Other aspects of the KTEV review will require review activity by the Panel. The Panel will make recommendations to you as they are warranted. Please let me know if you require any additional response or review by the Panel.

cc:
R.Currier
J. Kilmer
A. Szymulanski

Panel Members:
R. Bossert
J. Haggard
P. Hurh

KTEV
E-832

1.8 m VACUUM WINDOW

AND

VACUUM WINDOW TEST FIXTURE

ENGINEERING COMPUTATIONS

CONTENTS

I. STRESSES TABULATION

E-832/KTEV VACUUM WINDOW TEST FIXTURE

POS.	SYMBOL / NAME	NUMERICAL VALUE psi	REFERENCE PAGE	REMARKS
1	2	3	4	5
VACUUM WINDOW TEST FIXTURE FLANGE, ITEM (4) REF. DWG. 9220.832.MD-285386				
1.	σ_b OPERATING BOLT STRESS	20,337.5	18	
2.	S_i DESIGN PRESTRESS IN BOLTS	20,262.2	21	
3.	S_R RADIAL FLANGE STRESS AT BOLT CIRCLE	2552.2	21	
4.	S_R RADIAL FLANGE STRESS AT INSIDE DIAMETER	23.6	22	
5.	S_T TANGENTIAL FLANGE STRESS AT INSIDE DIA.	89.5	22	
6.	S_H LONGITUDINAL HUB STRESS	403.8	23	
VACUUM WINDOW ASSEMBLY, ITEM (2) DOWNSTREAM FLANGE, REF. DWG. 9220.832.MD-285394				
7.	σ STRESS DUE TO M_{ra} MOMENT (USING ALL TABULATED FACTORS)	2536	36	
8.	σ STRESS DUE TO M_{ra} MOMENT (USING FORMULA FOR SPECIAL CASE) -CONSERVATIVE APPROACH	5237.4	36	

I. STRESSES TABULATION

E-832 / KTEV VACUUM WINDOW TEST FIXTURE

POS.	SYMBOL / NAME	NUMERICAL VALUE	REFERENCE PAGE	REMARKS
1	2	3	4	5
9.	$\tilde{\sigma}_V$ TOTAL EXTERNAL PRESSURE DERIVATIVE STRESS (HOOP STRESS) — VACUUM CASE.	2435	41	
10.	$\tilde{\sigma}_P$ THE RESULTANT PRESSURE DERIVATIVE STRESS (HOOP STRESS) — 45psi HYDROSTATIC PRESSURE	4311	47	
11.	σ_{PL} - STRESS IN THE BOTTOM PLATE	12 525,6	50	
12	SUMMATION OF STRESSES POS. 10 AND 8 IS LOWER THAN ALLOWABLE STRESS OF 12700 (ASME. SECT. VIII. DIV. I FOR SA·36)			
13.	THE CRYTICAL PRESSURE FOR WINDOW FLANGES ASSEMBLY (p.41) IS EQUAL TO 1027.5 psi, THE 1027.5 > 45 psi; THEREFORE THE BUCKLING OF THIS COMPONENTS IS PRACTICALLY NONEXISTENT			

II. FLANGE ANALYSIS

VACUUM WINDOW TEST VESSEL
1.8 m VACUUM WINDOW TEST FIXTURE
DWG. MD - 285386 ITEM No (4)

II. 1. CLASSIFICATION OF THE FLANGE / SHELL CONNECTION, ACCORDING TO ASME CODE SECTION VIII, DIV 1 APPENDIX Y.

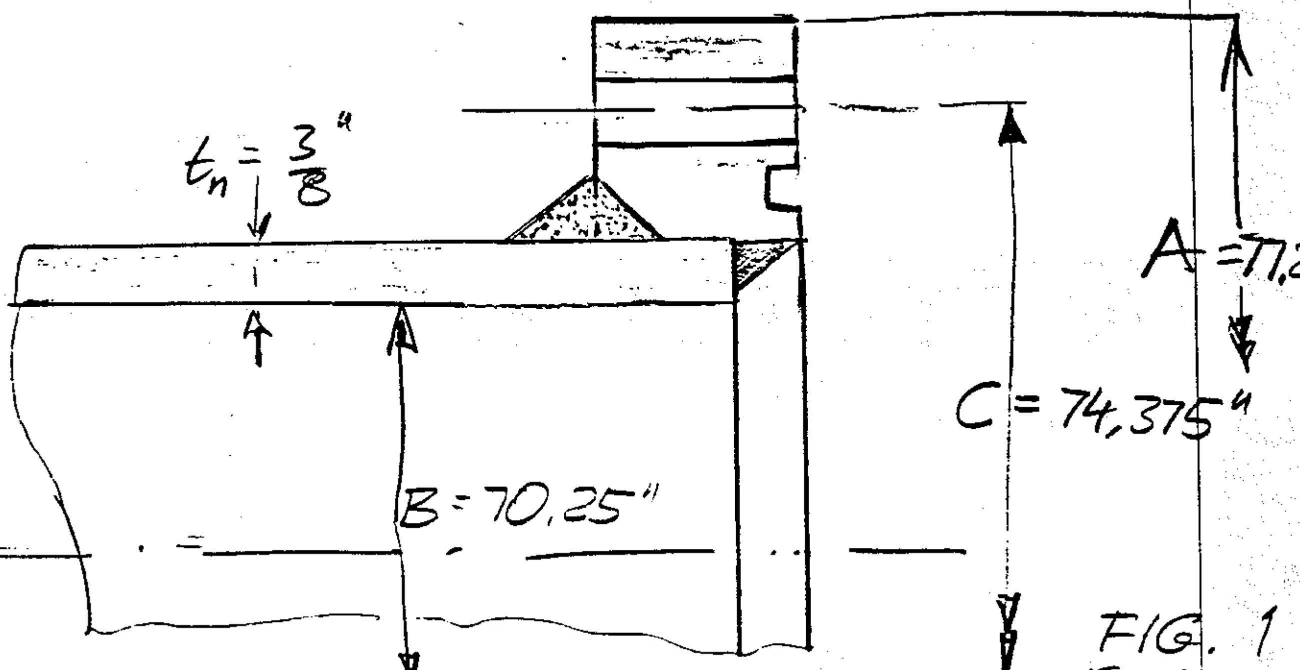


FIG. 1

FLANGE : OPTIONAL TYPE

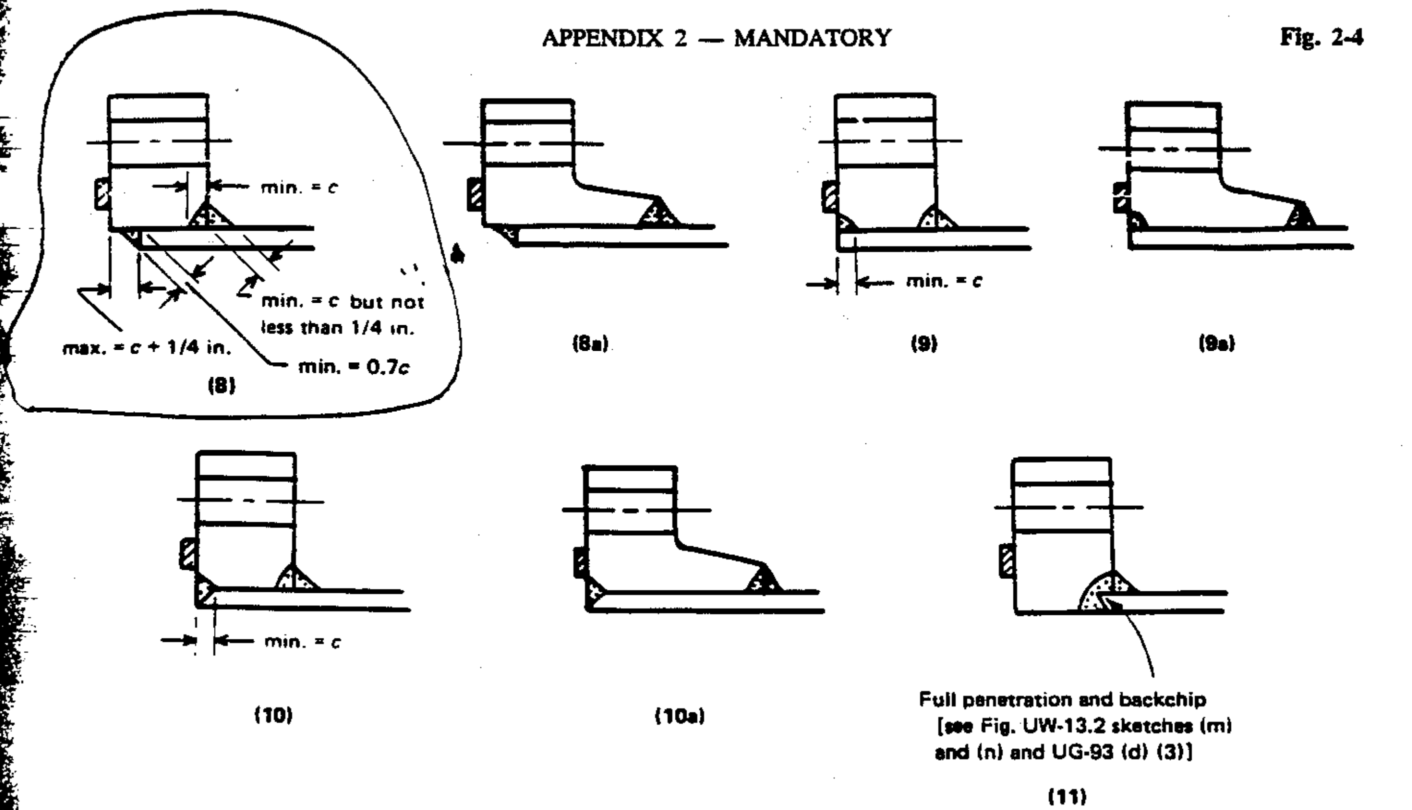
CLASS 1 ASSEMBLY

CATEGORY 1 FLANGE

CALCULATION : AS AN INTEGRAL TYPE

E-832/KTEV

6



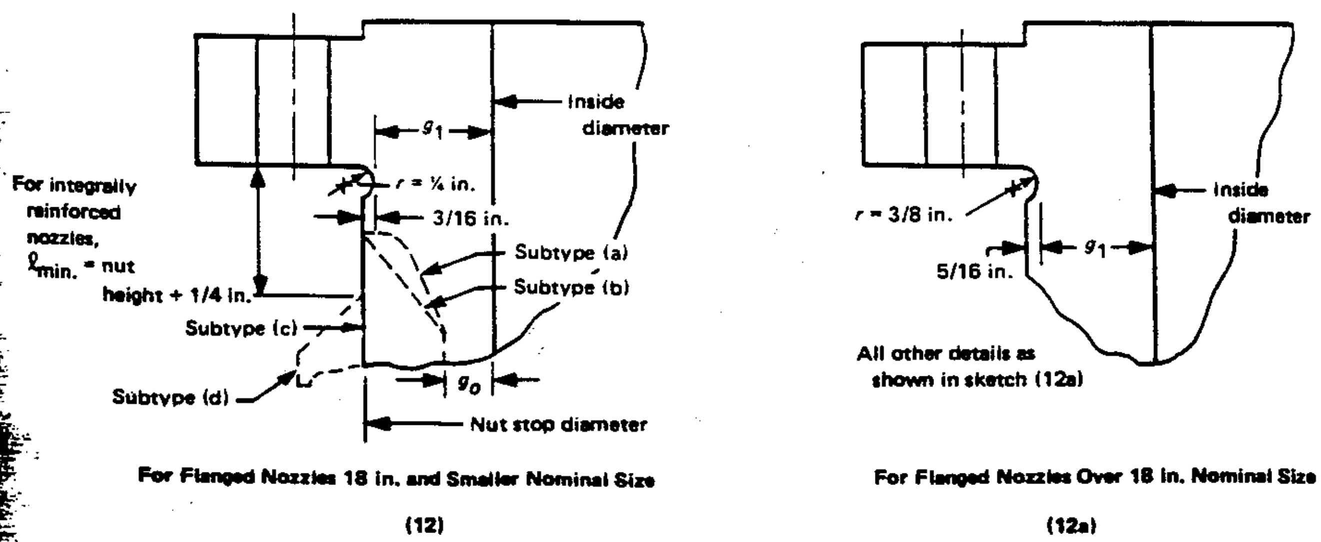
GENERAL NOTES (Optional Type Flanges):

(a) Optional type flanges may be calculated as either loose or integral type. See 2-4.

(b) Loadings and dimensions not shown in sketches (8), (8a), (9), (9a), (10), and (10a) are the same as shown in sketch (2) when the flange is calculated as a loose type flange and as shown in sketch (7) when the flange is calculated as an integral type flange.

(c) The groove and fillet welds between the flange back face and the shell given in sketch (8) also apply to sketches (8a), (9), (9a), (10), and (10a).

Optional Type Flanges



GENERAL NOTE (Flanges With Nut Stops):
For subtypes (a) and (b), g_0 is the thickness of the hub at the small end. For subtypes (c) and (d), $g_0 = g_1$.

Flanges With Nut Stops

FIG. 2-4 TYPES OF FLANGES (CONT'D)

II.2 DESIGN DATA:

SHELL THICKNESS $t_n = 0.375"$

INSIDE DIAMETER OF FLANGE $B = 70.25"$

APPENDIX Y — NONMANDATORY Y-6.1

TABLE Y-6.1
SUMMARY OF APPLICABLE FORMULAS FOR DIFFERENT CLASSES OF ASSEMBLIES AND DIFFERENT CATEGORIES OF FLANGES

Class	Category [Note (1)]	Applicable Formulas
1	1	(5a), (7)–(13), (14a), (15a), (16a)
1	2	(5b), (7)–(13), (14b), (15b), (16b)
1	3	(5c), (7)–(13), (14c), (15c), (16c)
2	All	See Y-6.2
3	1	(1)–(4), (6a), (17)–(31), (32a), (33a), (34a), (35)–(38)
3	2	(1)–(4), (6b), (17)–(31), (32b), (33b), (34b), (35)–(38)
3	3	(1)–(4), (6c), (17)–(31), (32c), (33c), (34c), (35)–(38)

NOTE:
(1) Of the nonreducing flange in a Class 2 or Class 3 assembly.

OUTSIDE DIA OF FLANGE $A = 77.25"$

FLANGE THICKNESS $t = 2"$

HYDROSTATIC PRESSURE $P = 45$ psi

II.3. ANALYSIS

II.3.1

Flange Moment due to Flange-Hub Interaction

$$\boxed{M_S = -\frac{J_P F' M_P}{t^3 + J_S F'}}$$

$$J_P = \frac{1}{B_1}\left[\frac{h_D}{\beta} + \frac{h_C}{a}\right] + \pi r_B$$

$$B_1 = B + g_o$$

0.25g_o but not less than 1/4 in., t
for either leg. This weld may be
to a corner radius as permitted i
sketch (5) in which case $g_1 = g_o$

GENERAL NO
(a) Fillet radius
(b) Facing thicl
in excess of
equal to ·

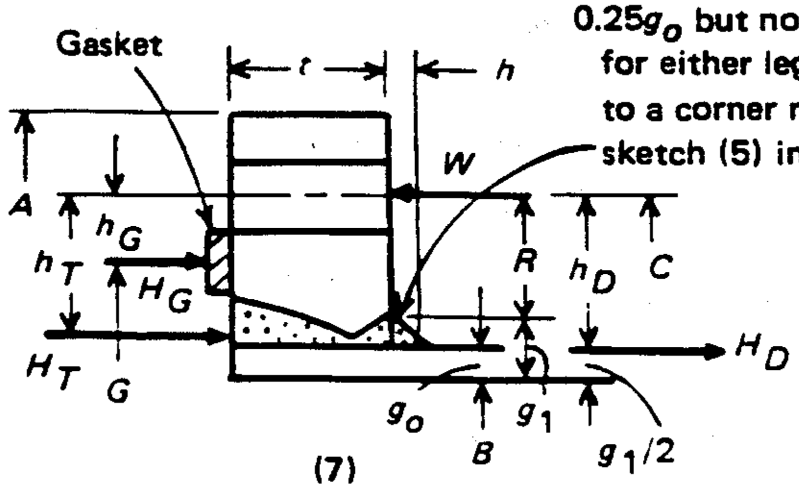


FIG. 2

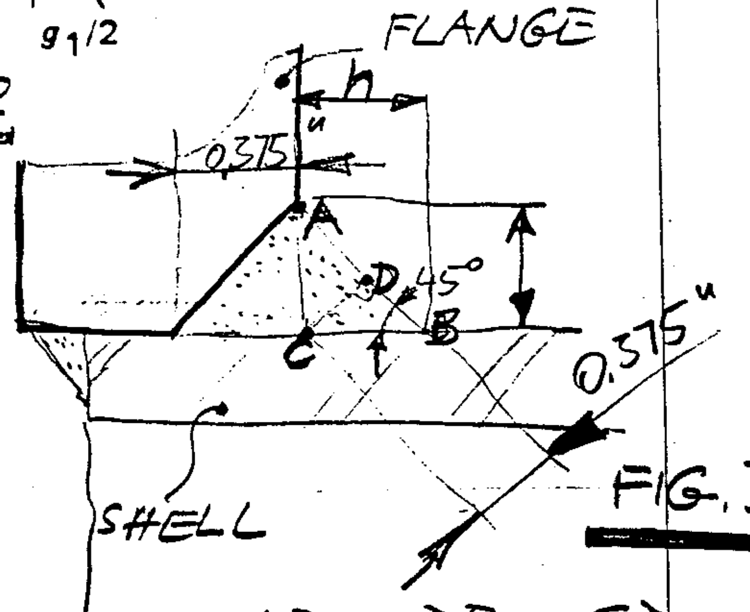


FIG. 3

$$g_o = \underline{0.375''}$$

$$g_1 = 0.375 + AC$$

$$g_1 = 0.375 + 0.53$$

$$g_1 = \underline{0.9}$$

$$\frac{g_1}{g_o} = 2.4$$

AD = DB = CD = 0.375

AB = 0.75"

CB = 0.53

AC = 0.53

$B_1 = 70.25 + 0.375$

$B_1 = \underline{70.625}$

$C = \underline{74.375''}$

$\frac{g_1}{2} = \frac{0.9}{2}$

$\frac{g_1}{2} = \underline{0.45}$

$h = 0.53$ $\qquad \frac{h}{h_o} = \frac{h}{\sqrt{B g_o}}$

$\frac{h}{h_o} = \frac{0.53}{\sqrt{(71)(0.375)}}$

$\frac{h}{h_o} = \underline{0.102}$

$h_o = \sqrt{B g_o} = \sqrt{70.25(0.375)}$

$h_o = \underline{5.13}$

h_D FROM TABLE 2.6

$h_D = R + 0.5 g_1$

FOR INTEGRAL AND HUB FLANGES

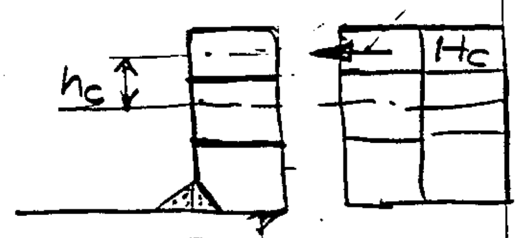
$R = \frac{C - B}{2} - g_1$

$R = \frac{74.375 - 70.25}{2} - 0.9$

$R = \underline{1.162}$

$h_D = 1.162 + 0.5(0.9)$

$h_D = \underline{1.61}$



$h_c = \frac{0.875}{2} + 0.063$

$h_c \ min = \frac{0.875}{2}$

FIG. 4

$h_c \ max = \frac{77.25 - 74.375}{2}$

h_c (MEAN VALUE) $\quad h_c = \underline{0.93}$

β - SPACE FACTOR

$$\beta = \left(\frac{C + B_1}{2B_1}\right) = \frac{74.375 + 70.625}{2(70.625)}$$

$$\beta = \underline{1.026}$$

a = shape factor

$$a = \frac{A + C}{2\ B_1}$$

$$a = \frac{77.25 + 74.375}{2(70.625)}$$

$$a = \underline{1.07}$$

$$n = 60$$

$$r_B = \frac{1}{60}\left[\frac{4}{\sqrt{1-\overline{AR}^2}}\tan^{-1}\sqrt{\frac{1+\overline{AR}}{1-\overline{AR}}} - \pi - 2\overline{AR}\right]$$

$$\overline{AR} = \frac{n \cdot D}{\pi \cdot C} = \frac{60(0.937)}{\pi(74.375)}$$

$$\overline{AR} = \underline{0.2406}$$

$$r_B = \frac{1}{60}\left[\frac{4}{\sqrt{1-(0.2406)^2}}\tan^{-1}\sqrt{\frac{1+0.2406}{1-0.2406}} - \pi - 2(0.2406)\right]$$

$$r_B = \frac{1}{60}\left[4.1210\left(\underset{\text{DEGREES}}{51.96097777}\right) - \pi - 2(0.2406)\right]$$

$$\frac{360° - 2\pi}{51.9609 - x} \quad x = 0.906890$$

$$r_B = \frac{1}{60}\left[4.1210\left(0.906890\right) - \pi - 2\left(0.2406\right)\right]$$

$$r_B = \frac{1}{60}(0.1145)$$

$$r_B = \underline{0.0019}$$

$$n\, r_B = 60(0.0019)$$

$$n\, r_B = \underline{0.1145}$$

$$J_P = \frac{1}{70.625}\left[\frac{1.61}{1.026} + \frac{0.93}{1.07}\right] + \pi(0.0019)$$

$$J_P = \frac{1}{70.625}(2.4383) + \pi(0.0019)$$

$$J_P = \underline{0.0404}$$

$F' = ---$

CATEGORY I CLASS 1

$F' = g_0^2 (h_0 + Ft)/V$

$$F' = \frac{(0.375)^2 (5.13 + Ft)}{V}$$

F - FROM (CODE) FIG. 2-7.2

FOR $\frac{g_1}{g_0} = 2.4$ AND $\frac{h}{h_0} = 0.1$

$F = \underline{0.9}$

$V = ----$

V FROM (CODE) FIG. 2-7.3

FOR $\frac{h}{h_0} = 0.1$

$\frac{g_1}{g_0} = 2.4$

$V = \underline{0.43}$

$$F' = \frac{(0.375)^2 [5.13 + 0.9(2)]}{0.43}$$

$F' = \underline{2.26}$

NUMERICAL DATA TABULATION

$t_n = 0.375$
$A = 77.25$
$B = \cancel{70}.25$
$B_1 = 70.625$
$C = 74.375$
$t = 2''$
$g_o = 0.375$
$g_1 = 0.9$
$\frac{g_1}{g_o} = 2.4$
$h = 0.53$
$\frac{h}{h_o} = 0.102$
$h_o = 5.13$
$R = 1.162$
$h_D = 1.61$
$h_C = 0.93$
$\beta = 1.026$
$a = 1.07$
$r_B = 0.0019$
$n r_B = 0.1145$
$J_P = 0.0404$

$F = 0.9$
$V = 0.43$
$F' = 2.26$
$h_G = 0.842$
$h_T = 1.452$
$P = 45$
$M_P = 298562.3$
$J_S = 0.0627$
$M_S = 3348.19$
$E\theta_B = 2666.01$
$H_C = 324634.93$
$H = 186651.51$
$H_T = 12320.51$
$H_D = 174331$

$M_P = H_D h_D + H_T h_T + H_G h_G$

$h_D =$ ---- (SEE FIG. 2)

$h_D = \underline{1.61}$

$h_T =$ ----

$h_T = \dfrac{R + g_1 + h_G}{2}$ (CODE) TABLE 2-6

$h_G = \left(\dfrac{C - G}{2}\right)$

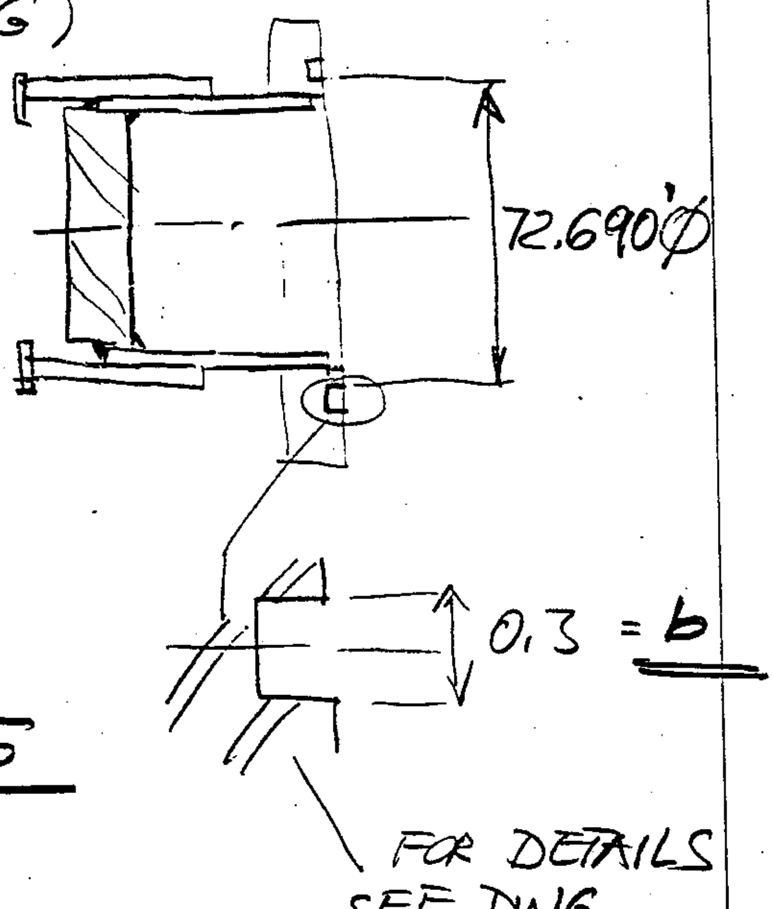


FIG. 5

$G = \dfrac{72.690"}{b_0 > \frac{1}{4}"}$

$$h_G = \frac{74.375 - 72.690}{2}$$

$$h_G = \underline{0.842}$$

$$h_T = \frac{1.162 + 0.9 + 0.842}{2}$$

$$h_T = \underline{1.452}$$

$$H_D = 0.785\, B^2\, P$$

$$H_D = 0.785\,(70.25)^2\,(45)$$

$$H_D = \underline{174331}$$

$$H_T = H - H_D$$

$$H = 0.785\, G^2\, P$$

$$H = 0.785\,(72.690)^2\,(45)$$

$$H = \underline{186651.51}$$

$$H_T = 186651.51 - 174331$$

$$H_T = \underline{12320.51}$$

H_G = ... GASKET LOAD DUE TO SEATING PRESSURE —

IN THIS CASE THE GASKET IS VERY SOFT.

WE CAN NEGLECT THE REACTION FORCE OF THE GASKET

$$M_p = 174331(1.61) + 12320.51(1.452)$$

$$M_p = \underline{298562.3}$$

$$J_s = \frac{1}{B_1}\left[\frac{2h_D}{\beta} + \frac{h_c}{a}\right] + \pi r_B$$

$$J_s = \frac{1}{70.625}\left[\frac{2(1.61)}{1.026} + \frac{0.93}{1.07}\right] + \pi(0.0019)$$

$$J_s = \underline{0.062713}$$

$$M_s = \frac{0.0404(2.26)(298562.3)}{(2)^3 + 0.0627(2.26)}$$

$$M_s = \underline{3348.19}$$

II.3.2. Slope of Flange at Inside Diameter Times E

$$E\theta_B = \frac{5.46}{\pi t^3}(J_S M_S + J_P M_P)$$

$$E\theta_B = \frac{5.46}{\pi(2)^3}\left[(0.0627)(3348.19) + 0.0404(298562.3)\right]$$

$$E\theta_B = \underline{2666.016}$$

$$\theta = 0.005^\circ$$

II.3.3 Contact Force Between Flanges at h_C

$$H_C = (M_P + M_S)/h_C$$

$$H_c = (298562.3 + 3348.19)/0.93$$

$$H_c = \underline{324634.93}$$

II.3.4 Bolt Load at Operating Conditions

$$W_{m1} = H + H_G + H_C$$

$$W_{m1} = 186651.51 + 0 + 324654.93$$

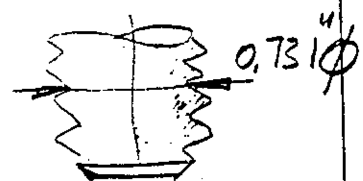
$$W_{m1} = \underline{511286.44}$$

II. 3. 5 **Operating Bolt Stress**

$$\sigma_b = W_{m1}/A_b$$

$$\sigma_b = \frac{511286.44}{A_b}$$

$$A_b = \frac{\pi(0.731)^2}{4}(n)$$



$$A_b = 0.419 \ in^2 (61$$

$$\sigma_b = \underline{20,337.56}$$

II.3.6

Design Prestress in Bolts

$$S_i = \sigma_b - \frac{1.159 h_C^2 (M_P + M_S)}{a t^3 l r_E B_1}$$

$l = 2t + t_s + \frac{1}{2} d_b$

CLASS 1 ASSEMBLY

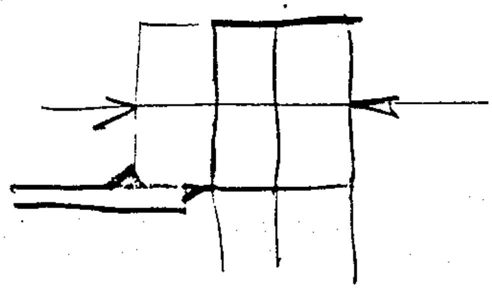
$t = 2$

$t_s = 0.21$ (WASHER)

$d_b = 0.875$

IN OUR CASE

$2t = 3t$



(3t)

FIG. 6

$l = 6 + 0.21 + \frac{1}{2}(0.875)$

$l = 6.64$

$r_E = \frac{29.5}{29.3}$

$r_E = 1.006$

FLANGE MATERIAL

$E = 29.5 \times 10^6$ FROM: UF-27 70°F AND C < 0.3%

BOLTS: ASTM A-574 C = 0.33%

Designation: A 574 – 87

FERRY CAP SCREW 20
MATERIAL!

Standard Specification for Alloy Steel Socket-Head Cap Screws[1]

This standard is issued under the fixed designation A 574; the number immediately following the designation indicates the year of original adoption or, in the case of revision, the year of last revision. A number in parentheses indicates the year of last reapproval. A superscript epsilon (ε) indicates an editorial change since the last revision or reapproval.

1. Scope

1.1 This specification covers the requirements for quenched and tempered alloy steel hexagon socket-head cap screws, 0.060 through 4 in. in diameter where high strength is required.

NOTE 1—A complete metric companion to Specification A 574 has been developed—A 574M; therefore no metric equivalents are presented in this specification.

2. Referenced Documents

2.1 *ASTM Standards:*

A 370 Methods and Definitions for Mechanical Testing of Steel Products[2]

A 574 Specification for Alloy Steel Socket-Head Cap Screws[3]

A 751 Methods, Practices, and Definitions for Chemical Analysis of Steel Products[2]

E 3 Methods of Preparation of Metallographic Specimens[4]

F 606 Test Methods for Determining the Mechanical Properties of Externally and Internally Threaded Fasteners, Washers, and Rivets[3]

F 788 Specification for Surface Discontinuities of Bolts, Screws, and Studs, Inch and Metric Series[3]

2.2 *ANSI Standards:*[5]

ANSI B1.1 Unified Screw Threads

ANSI B18.3 Socket Cap, Shoulder, and Set Screws

2.3 *Federal Standard:*[6]

Fed. Std. H28

3. Definitions

3.1 Definitions of discontinuities covered by 8.2 follow:

3.1.1 *crack*—a clean crystalline break passing through the grain or grain boundary without inclusion of foreign elements.

3.1.2 *seam or lap*—a noncrystalline break through the metal which is inherently in the raw material.

3.1.3 *inclusions*—particles of nonmetallic impurities, usually oxides, sulfides, silicates, and such, which are mechanically held in the steel during solidification.

3.1.4 *nicks or pits*—depressions or indentations in the surface of the metal.

4. Ordering Information

4.1 Orders for socket head cap screws under this specification shall include the following:

4.1.1 ASTM designation and year of issue,

4.1.2 Quantities (number of pieces by size),

4.1.3 Size and length,

4.1.4 Specify if inspection at point of manufacture is required,

4.1.5 Specify if certified test reports are required (12.2), and

4.1.6 Specify additional testing (12.3) or special requirements.

5. Material and Manufacture

5.1 The screws shall be fabricated from a steel which has been made by the open-hearth, basic-oxygen, or electric-furnace process.

5.2 Unless otherwise specified, the heads of screws through 1.500-in. diameter shall be fabricated by hot or cold forging. Over 1.500-in. diameter, the heads may be fabricated by hot or cold forging or by machining. Sockets may be forged or machined.

5.3 Unless otherwise specified, threads of screws shall be rolled for diameters through 0.625 in. and for screw lengths through 4 in. For diameters and lengths other than this, threads may be rolled, cut, or ground.

5.4 The screws shall be heat treated by oil quenching from above the transformation temperature and then tempering at a temperature not lower than 650°F.

6. Chemical Composition

6.1 The heat analysis of the screw material shall conform to the chemical composition specified in Table 1.

6.2 Product analyses may be made by the purchaser from finished material representing each lot. The chemical composition, thus determined, shall conform to the requirements prescribed for product analysis in Table 1.

6.3 One or more of the following alloying elements: chromium, nickel, molybdenum, or vanadium shall be

TABLE 1 Chemical Requirements

Element	Composition, %	
	Heat Analysis	Product Analysis
Carbon, min	0.33	0.31
Phosphorus, max	0.040	0.045
Sulfur, max	0.040	0.045

[1] This specification is under the jurisdiction of ASTM Committee F-16 on Fasteners and is the direct responsibility of Subcommittee F16.02 on Steel Bolts, Nuts, Rivets, and Washers.

Current edition approved Aug. 28, 1987. Published October 1987. Originally published as A 574 – 67. Last previous edition A 574 – 83.

[2] *Annual Book of ASTM Standards*, Vols 01.01 to 01.05.

[3] *Annual Book of ASTM Standards*, Vol 15.08.

[4] *Annual Book of ASTM Standards*, Vol 03.01.

[5] Available from American National Standards Institute, 1430 Broadway, New York, NY 10018.

[6] Available from General Services Administration, Specification and Consumer Information Distribution Branch, Bldg. 197, Washington Navy Yard, Washington, DC 20407.

$$S_i = 20337.56 - \frac{1.159(0.93)^2(298562.3 + 3348.19)}{1.07(2)^3(6.64)(1.006)(70.625)}$$

$$S_i = 20337.56 - 75.34$$

$$S_i = \underline{20262.2} \quad [psi]$$

II.3.7

Radial Flange Stress at Bolt Circle

$$S_R = \frac{6(M_P + M_S)}{t^2(\pi C - nD)}$$

$$S_R = \frac{6\ (298562.3 + 3348.19)}{(2)^2[\pi(74.375) - 60 \times 0.937]}$$

$$S_R = \underline{2552.27} \quad (psi)$$

II.3.8

Radial Flange Stress at Inside Diameter

$$S_R = -\left(\frac{2Ft}{h_0 + Ft} + 6\right)\frac{M_S}{\pi B_1 t^2}$$

$$S_R = -\left(\frac{2 \times 0.9}{5.13 + 0.9 \times 2} + 6\right)\frac{3348.19}{\pi \times 70.625 \times 2^2}$$

$S_R = \underline{23.6}$ psi

Tangential Flange Stress at Inside Diameter

II 3.9

$$S_T = \frac{tE\theta_B}{B_1} + \left(\frac{2FtZ}{h_0 + Ft} - 1.8\right)\frac{M_S}{\pi B_1 t^2}$$

Z . . . FROM ; FIG. 2.7.1

FOR $K = \frac{A}{B} = \frac{77.25}{70.25} = 1.099$

$$Z = \frac{K^2+1}{K^2-1} = \frac{(1.099)^2+1}{(1.099)^2-1}$$

$$Z = \underline{10.6245}$$

$$S_T = \frac{2 \times 2666.016}{70.625} + \left[\frac{2(0.9)(2)(10.6245}{5.13 + 0.9 \times 2} - 1.8\right]\frac{3348.19}{\pi \times 70.625 \times 2^2}$$

$$S_T = \underline{89.5}$$

II. 3.10

Longitudinal Hub Stress

$$S_H = \frac{h_0 E \theta_B f}{0.91\,(g_1/g_0)^2 B_1 V}$$

f ----

FROM FIG. 2-7.6

$$\frac{h}{h_0} = 0.102$$

$$\frac{g_1}{g_0} = 2.4$$

$$f = 4.7$$

V -----

FROM FIG. 2-7.3

$$V = 0.43$$

$$S_H = \frac{5.13 \times 2666.01 \times 4.7}{0.91\,(2.4)^2\ 70.625 \times 0.43}$$

$$S_H = \underline{403.8} \quad psi$$

✓

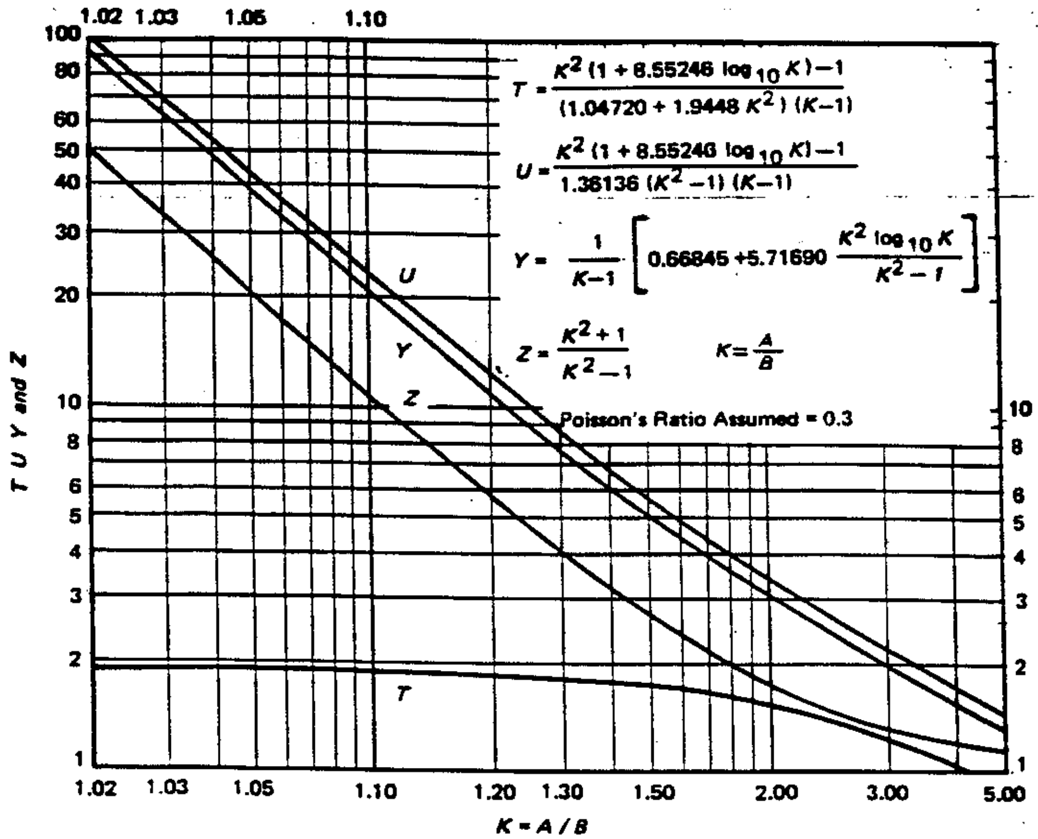


FIG. 2-7.1 VALUES OF T, U, Y, AND Z
(Terms Involving K)

$$W = W_{m1} \tag{3}$$

For gasket seating,

$$W = \frac{(A_m + A_b)S_a}{2} \tag{4}$$

S_a used in Formula (4) shall be not less than that tabulated in the stress tables (see UG-23). In addition to the minimum requirements for safety, Formula (4) provides a margin against abuse of the flange from overbolting. Since the margin against such abuse is needed primarily for the initial, bolting-up operation which is done at atmospheric temperature and before application of internal pressure, the flange design is required to satisfy this loading only under such conditions (see Note 2).

NOTE 2: Where additional safety against abuse is desired, or where it is necessary that the flange be suitable to withstand the full available bolt load $A_b S_a$, the flange may be designed on the basis of this latter quantity.

2-6 FLANGE MOMENTS

In the calculation of flange stress, the moment of a load acting on the flange is the product of the load and its moment arm. The moment arm is determined by the relative position of the bolt circle with respect to that of the load producing the moment (see Fig. 2-4). No consideration shall be given to any possible reduction in moment arm due to cupping of the flanges or due to inward shifting of the line of action of the bolts as a result thereof.

For the operating conditions, the total flange moment M_0 is the sum of the three individual moments M_D, M_T, and M_G, as defined in 2-3 and based on the flange design load of Formula (3) with moment arms as give in Table 2-6.

For gasket seating, the total flange moment M_0 is

E - 832/KTEV

25

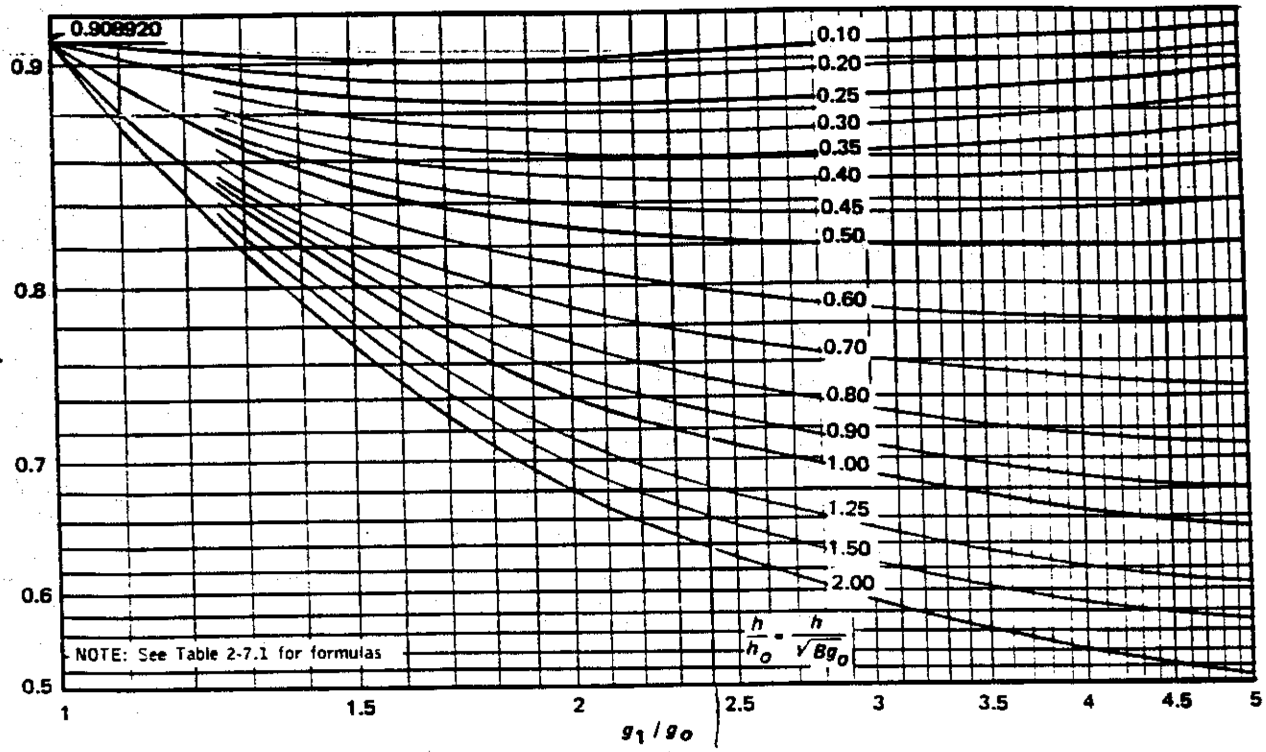


FIG. 2-7.2 VALUES OF *F*
(Integral Flange Factors)

based on the flange design bolt load of Formula (4), which is opposed only by the gasket load, in which case

$$M_0 = W\frac{(C - G)}{2} \tag{5}$$

2-7 CALCULATION OF FLANGE STRESSES

The stresses in the flange shall be determined for both the operating conditions and gasket seating condition, whichever controls, in accordance with the following formulas:

(a) for integral type flanges [Fig. 2-4 sketches (5), (6), (6a), (6b), and (7)]; for optional type flanges calculated as integral type [Fig. 2-4 sketches (8), (8a), (9), (9a), (10), (10a), and (11)]; and for loose type flanges with a hub which is considered [Fig. 2-4 sketches (1), (1a), (2), (2a), (3), (3a), (4), and (4a)]:

Longitudinal hub stress

$$S_H = \frac{fM_o}{Lg_1^2 B} \tag{6}$$

Radial flange stress

$$S_R = \frac{(1.33te + 1)M_o}{Lt^2 B} \tag{7}$$

Tangential flange stress

$$S_T = \frac{YM_o}{t^2 B} - ZS_R \tag{8}$$

(b) for loose type flanges without hubs and loose type flanges with hubs which the designer chooses to calculate without considering the hub [Fig. 2-4 sketches (1), (1a), (2), (2a), (3), (3a), (4), and (4a)] and optional type flanges calculated as loose type [Fig. 2-4 sketches (8), (8a), (9), (9a), (10), (10a), and (11)]:

E-832/KTEV

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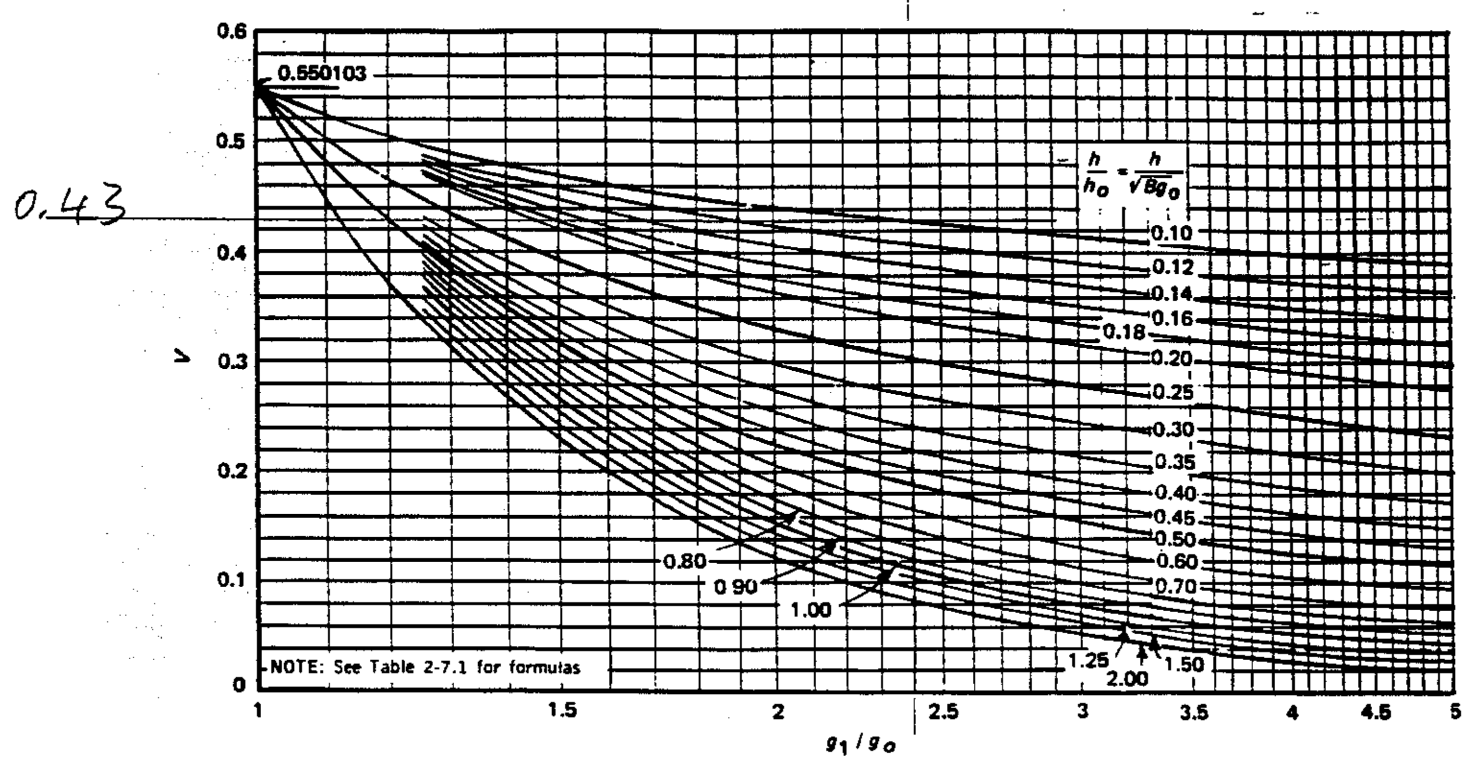


FIG. 2-7.3 VALUES OF V
(Integral Flange Factors)

$$S_T = \frac{YM_o}{t^2B} \qquad (9)$$

$$S_R = 0 \qquad S_H = 0$$

2-8 ALLOWABLE FLANGE DESIGN STRESSES

(a) The flanges stresses calculated by the formulas in 2-7 shall not exceed the following values:

(1) longitudinal hub stress S_H not greater than S_f for cast iron[1] and, except as otherwise limited by (1)(a) and (1)(b) below, not greater than 1.5 S_f for materials other than cast iron:

(a) longitudinal hub stress S_H not greater than the smaller of 1.5S_f or 1.5S_n for optional type flanges designed as integral [Fig. 2-4 sketches (8), (8a), (9), (9a), (10), (10a), and (11)], also integral type [Fig. 2-4 sketch (7)] where the neck material constitutes the hub of the flange;

(b) longitudinal hub stress S_H not greater than the smaller of 1.5S_f or 2.5S_n for integral type flanges with hub welded to the neck, pipe or vessel wall [Fig. 2-4 sketches (6), (6a), and (6b)].

(2) radial flange stress S_R not greater than S_f;

(3) tangential flange stress S_T not greater than S_f;

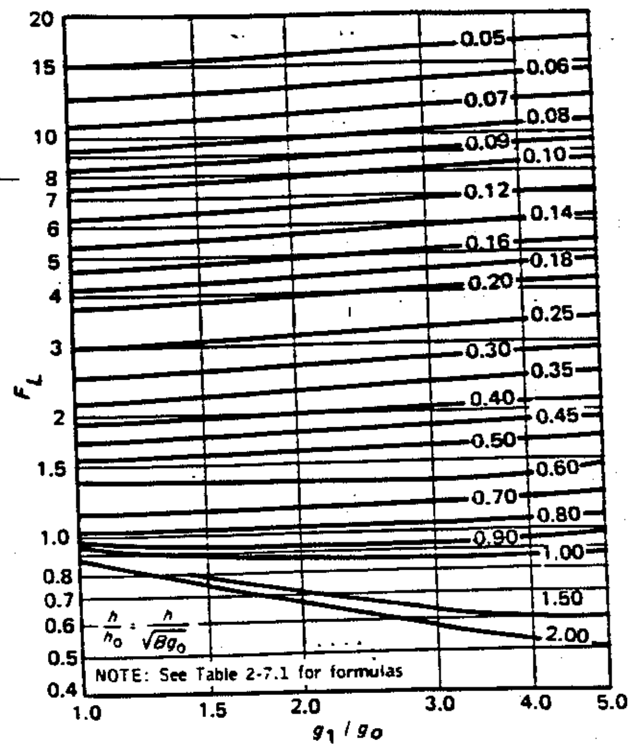
(4) also $(S_H + S_R)/2$ not greater than S_f and $(S_H + S_T)/2$ not greater than S_f.

(b) For hub flanges attached as shown in Fig. 2-4 sketches (2), (2a), (3), (3a), (4), and (4a), the nozzle neck, vessel or pipe wall shall not be considered to have any value as a hub.

(c) In the case of loose type flanges with laps, as shown in Fig. 2-4 sketches (1) and (1a), where the gasket is so located that the lap is subjected to shear, the shearing stress shall not exceed 0.8 S_n for the material of the lap, as defined in 2-3. In the case of welded flanges, shown in Fig. 2-4 sketches (3), (3a), (4), (4a), (7), (8), (8a), (9), (9a), (10), and (10a) where the nozzle neck, vessel, or pipe wall extends near to the flange face and may form the gasket contact face,

[1] When the flange material is cast iron, particular care should be taken when tightening the bolts to avoid excessive stress that may break the flange. The longitudinal hub stress has been limited to S_f in order to minimize any cracking of flanges. An attempt should be made to apply no greater torque than is needed to assure tightness during the hydrostatic test.

27



2.4

FIG. 2-7.4 VALUES OF F_L
(Loose Hub Flange Factors)

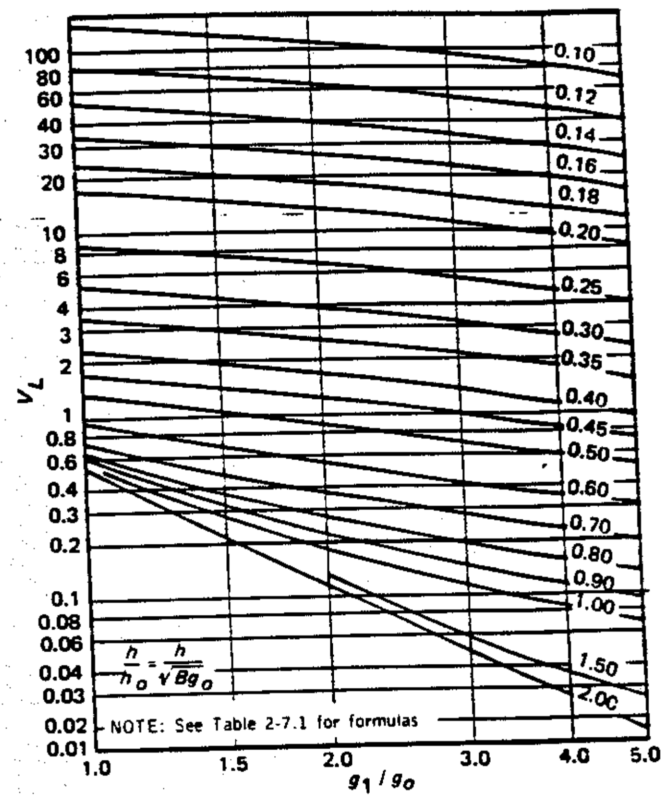


FIG. 2-7.5 VALUES OF V_L
(Loose Hub Flange Factors)

the shearing stress carried by the welds shall not exceed 0.8 S_n. The shearing stress shall be calculated on the basis of W_{m1} or W_{m2} as defined in 2-3, whichever is greater. Similar cases where flange parts are subjected to shearing stress shall be governed by the same requirements.

2-9 SPLIT LOOSE FLANGES[2]

Loose flanges split across a diameter and designed under the rules given in this Appendix may be used under the following provisions.

(a) When the flange consists of a single split flange or flange ring, it shall be designed as if it were a solid flange (without splits), using 200% of the total moment M_0 as defined in 2-6.

(b) When the flange consists of two split rings each ring shall be designed as if it were a solid flange (without splits), using 75% of the total moment M_0 as defined in 2-6. The pair of rings shall be assembled so that the splits in one ring shall be 90 deg. from the splits in the other ring.

(c) The splits should preferably be midway between bolt holes.

[2]Loose flanges of the type shown in Fig. 2-4 sketch (1) are of the split design when it is necessary to install them after heat treatment of a stainless steel vessel, or when for any reason it is desired to have them completely removable from the nozzle neck or vessel.

2-10 NONCIRCULAR SHAPED FLANGES WITH CIRCULAR BORE

The outside diameter A for a noncircular flange with a circular bore shall be taken as the diameter of the largest circle, concentric with the bore, inscribed entirely within the outside edges of the flange. Bolt loads and moments, as well as stresses, are then calculated as for circular flanges, using a bolt circle drawn through the centers of the outermost bolt holes.

E-832/ kTEV

28

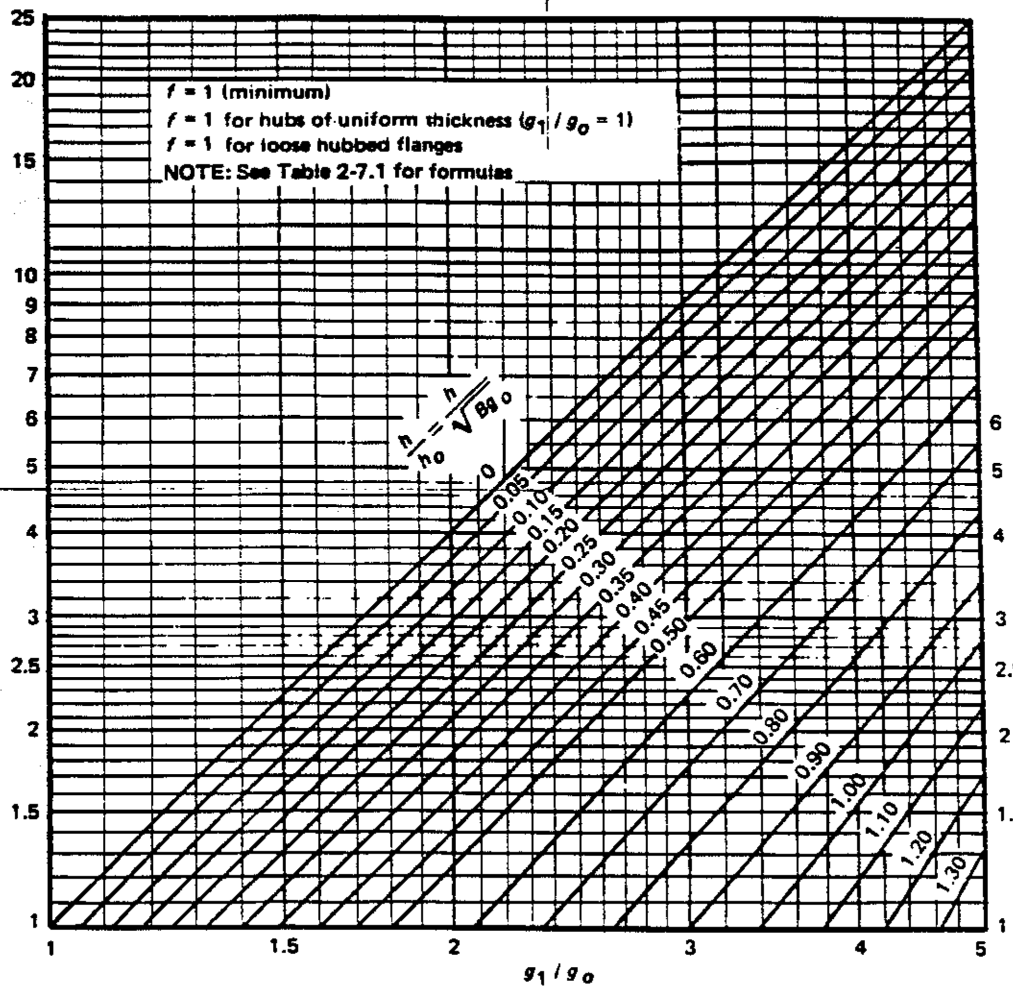


FIG. 2-7.6 VALUES OF f
(Hub Stress Correction Factor)

2.4

2-11 FLANGES SUBJECT TO EXTERNAL PRESSURES

(a) The design of flanges for external pressure only [see UG-99(f)][3] shall be based on the formulas given in 2-7 for internal pressure except that for operating conditions:

$$M_0 = H_D(h_D - h_G) + H_T(h_T - h_G) \quad (10)$$

For gasket seating,

[3]When internal pressure occurs only during the required pressure test, the design may be based on external pressure, and auxiliary devices such as clamps may be used during the application of the required test pressure.

$$M_0 = Wh_G \quad (11)$$

where

$$W = \frac{A_{m2} + A_b}{2} S_a \quad (11a)$$

$$H_D = 0.785B^2P_e \quad (11b)$$

$$H_T = H - H_D \quad (11c)$$

$$H = 0.785G^2P_e \quad (11d)$$

P_e=external design pressure, psi

See 2-3 for definitions of other symbols. S_a used in

29

TABLE 2-6
MOMENT ARMS FOR FLANGE LOADS UNDER OPERATING CONDITIONS

	h_D	h_T	h_G
Integral type flanges [see Fig. 2.4 sketches (5), (6), (6a), (6b), and (7)]; and optional type flanges calculated as integral type [see Fig. 2-4 sketches (8), (8a), (9), (9a), (10), (10a), and (11)]	$R + 0.5g_1$	$\dfrac{R + g_1 + h_G}{2}$	$\dfrac{C - G}{2}$
Loose type, except lap-joint flanges [see Fig. 2-4 sketches (2), (2a), (3), (3a), (4), and (4a)]; and optional type flanges calculated as loose type [see Fig. 2-4 sketches (8), (8a), (9), (9a), (10), (10a), and (11)]	$\dfrac{C - B}{2}$	$\dfrac{h_D + h_G}{2}$	$\dfrac{C - G}{2}$
Lap-type flanges [see Fig. 2-4 sketches (1) and (1a)]	$\dfrac{C - B}{2}$	$\dfrac{C - G}{2}$	$\dfrac{C - G}{2}$

Formula (11a) shall be not less than that tabulated in the stress tables (see UG-23).

(b) When flanges are subject at different times during operation to external or internal pressure, the design shall satisfy the external pressure design requirements given in (a) above and the internal pressure design requirements given elsewhere in this Appendix.

NOTE: The combined force of external pressure and bolt loading may plastically deform certain gaskets to result in loss of gasket contact pressure when the connection is depressurized. To maintain a tight joint when the unit is repressurized, consideration should be given to gasket and facing details so that excessive deformation of the gasket will not occur. Joints subject to pressure reversals, such as in heat exchanger floating heads, are in this type of service.

2-12 FLANGES WITH NUT-STOPS

(a) When flanges are designed per this Appendix, or are fabricated to the dimensions of ANSI B16.5 or other acceptable standards [see UG-44(a)], except that the dimension R is decreased to provide a nut-stop, the fillet radius relief shall be as shown in Fig. 2-4 sketches (10a) and (10b) except that:

(1) for flanges designed to this Appendix, the dimension g_1 must be the lesser of $2t$ (t from UG-27) or $4r$, but in no case less than $^1/_2$ in., where

r = the radius of the undercut

(2) for ANSI B16.5 or other standard flanges, the dimension of the hub g_o shall be increased as necessary to provide a nut-stop.

2-13 REVERSE FLANGES

(a) Flanges with a configuration as indicated in Fig. 2-13 shall be designed as reverse flanges, in conformance with the rules in 2-3 through 2-8, but with modifications as described in the following. Mandatory use of these rules is limited to $K \leq 2$. When $K > 2$, results become increasingly conservative and U-2(g) may be used.

The symbols and definitions in this paragraph pertain specifically to reverse flanges. Except as noted in (b) below, the symbols used in the equations of this paragraph are defined in 2-3.

The formulas for S_H, S_R, and S_{T1} correspond, respectively, to Formulas (6), (7), and (8) in 2-7, in direction, but are located at the flange *outside* diameter. The sole stress at the flange inside diameter is a tangential stress and is given by the formula for S_{T2}.

(b) Notation

B = inside diameter of shell, in.

B' = inside diameter of reverse flange, in.

$$d_r = U_r h_{0r} g_0^2 / V$$

$$e_r = F / h_{0r}$$

F = factor (use h_{0r} for h_0 in Fig. 2-7.2)

f = factor (use h_{0r} for h_0 in Fig. 2-7.6)

H = total hydrostatic end force on attached component, lb

$$= 0.785G^2P$$

H_D = hydrostatic end force on area inside of flange, lb

$$= 0.785B^2P$$

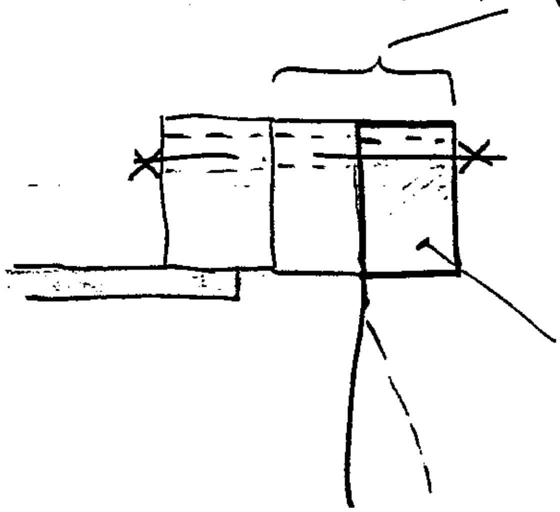
H_T = difference between hydrostatic end force on attached component and hydrostatic end force on area inside of flange, lb

$$= H - H_D$$

III. FLANGE ANALYSIS

DWG. MD-285394

ITEM (2)



B.1 CLASSIFICATION

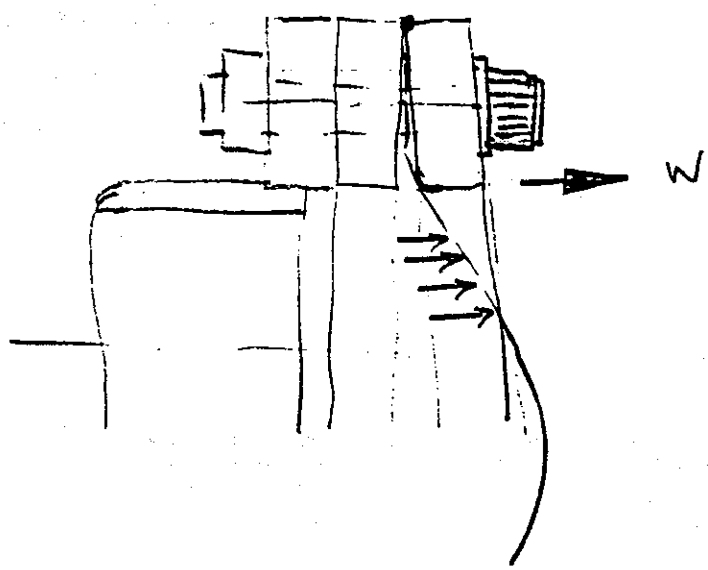
THE ANALYSIS IN SUBSECTION A OF THIS CALCULATION SHOWED VERY MARGINAL ANGULAR DEFLECTION OF THE TEST VESSEL FLANGE.

HAVING THAT AS A BASE, WE WILL TREAT THE WINDOW ASSEMBLY AS A "FIXED" CASE AND FORMULAS FOR FLAT CIRCULAR PLATES OF CONSTANT THICKNESS WILL APPLY.

"FORMULAS" — R.J. ROARK
W.C. YOUNG
FORMULAS FOR STRESS AND STRAIN
(Fifth edition)

THE FLANGE / FLANGE METAL CONTACT WILL OCCUR UNDER FOLLOWING CONDITIONS:

a) ANGULAR DISLOCATION OF THE FLANGE AS SHOWN IN FIG. 7a



b) THE WASHER INSTALLATION

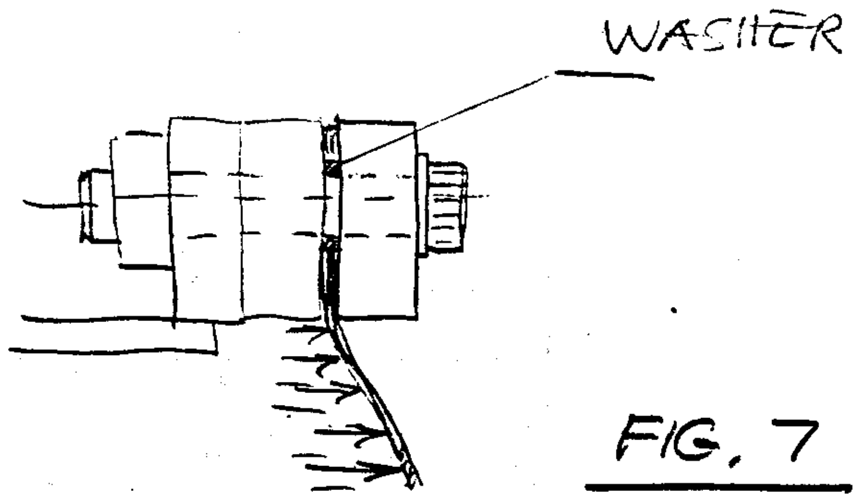
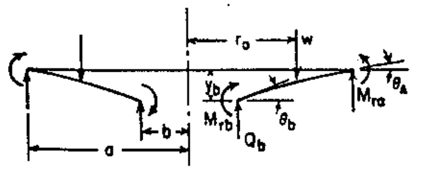


FIG. 7

BOTH CASES WILL BE INVESTIGATED DURING THE TESTS. THE CASE b IS CONSIDERED AS A BASE FOR FOLLOWING ANALYSIS; TREATING THE FLANGE AS "THE FIXED" CASE — ROARK AND YOUNG FORMULAS FOR STRESS AND S

III.2 ANALYSIS

Case 1. Annular plate with a uniform annular line load of w lb/in at a radius r_o



General expressions for deformations, moments, and shears:

$$y = y_b + \theta_b r F_1 + M_{rb}\frac{r^2}{D}F_2 + Q_b\frac{r^3}{D}F_3 - w\frac{r^3}{D}G_3$$

$$\theta = \theta_b F_4 + M_{rb}\frac{r}{D}F_5 + Q_b\frac{r^2}{D}F_6 - w\frac{r^2}{D}G_6$$

$$M_r = \theta_b\frac{D}{r}F_7 + M_{rb}F_8 + Q_b r F_9 - wrG_9$$

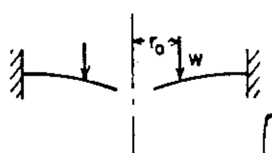
$$M_t = \frac{\theta D(1-\nu^2)}{r} + \nu M_r$$

$$Q = Q_b\frac{b}{r} - w\frac{r_o}{r}\langle r - r_o\rangle^0$$

For the numerical data given below, $\nu = 0.3$

$$y = K_y\frac{wa^3}{D} \qquad \theta = K_\theta\frac{wa^2}{D} \qquad M = K_M wa \qquad Q = K_Q w$$

TABLE 24 *Formulas for flat circular plates of constant thickness* (*Cont.*)

Case no., edge restraints	Boundary values
1e. Outer edge fixed, inner edge free 	$M_{rb} = 0 \quad Q_b = 0 \quad y_a = 0 \quad \theta_a = 0$ $y_b = \frac{-wa^3}{D}\left(\frac{C_1L_6}{C_4} - L_3\right)$ $\theta_b = \frac{wa^2}{DC_4}L_6$ $M_{ra} = -wa\left(L_9 - \frac{C_7L_6}{C_4}\right)$ $Q_a = \frac{-wr_o}{a}$

Special cases

If $r_o = b$ (load at inner edge),

b/a	0.1	0.3	0.5	0.7	0.9
K_{y_b}	−0.0143	−0.0330	−0.0233	−0.0071	−0.0003
K_{θ_b}	0.0254	0.0825	0.0776	0.0373	0.0048
$K_{M_{ra}}$	−0.0528	−0.1687	−0.2379	−0.2124	−0.0911
$K_{M_{tb}}$	0.2307	0.2503	0.1412	0.0484	0.0048

(*Note:* $|M_{ra}| > |M_{tb}|$ if $b/a > 0.385$)

III.2.1 DESIGN DATA

$a = \frac{74.375}{2}$

$a = \underline{37.187}$

$b = \frac{71}{2}$

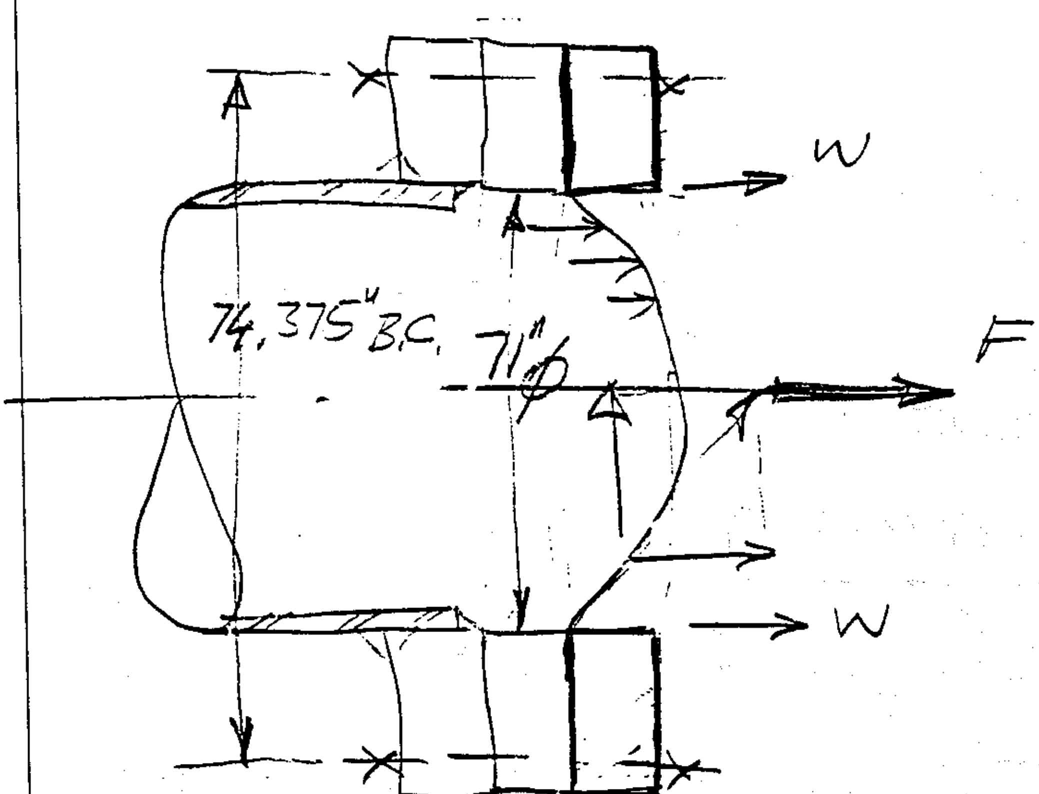
$b = \underline{35.5}$ $\quad r_o = \underline{35.5}$

$r_o = b$

MAX. HYDROSTATIC PRESSURE

$p = 45$ psi

FIND THE FORCE W.



$$F = \frac{\pi (71)^2}{4}(45)$$

$$F = \underline{178163.6} \quad [lbs]$$

$$W = \frac{F}{\pi (71)} = \frac{178163.6}{\pi (71)}$$

$$W = \underline{798.75} \quad \left[\frac{lbs}{in}\right]$$

L_6 ...

L_9 ...

C_4 ...

C_7 ...

$$L_6 = \frac{r_o}{4a}\left[\left(\frac{r_o}{a}\right)^2 - 1 + 2\ln\frac{a}{r_o}\right]$$

$$L_6 = \frac{35.5}{4(37.187)}\left[\left(\frac{35.5}{37.187}\right)^2 - 1 + 2\ln\frac{37.187}{35.5}\right]$$

$$L_6 = \underline{0.000997704}$$

$$L_9 = \frac{r_o}{a}\left\{\frac{1+\nu}{2}\ln\frac{a}{r_o} + \frac{1-\nu}{4}\left[1 - \left(\frac{r_o}{a}\right)^2\right]\right\}$$

$$L_9 = \frac{35.5}{37.187}\left\{\frac{1+0.3}{2}\ln\frac{37.187}{35.5} + \frac{1-0.3}{4}\left[1 - \left(\frac{35.5}{37.187}\right)^2\right]\right\}$$

$$L_9 = \underline{0.043621994}$$

$$C_4 = \frac{1}{2}\left[(1+\nu)\frac{b}{a} + (1-\nu)\frac{a}{b}\right]$$

$$C_4 = \frac{1}{2}\left[(1+0.3)\frac{35.5}{37.187} + (1-0.3)\frac{37.187}{35.5}\right]$$

$$C_4 = \underline{0.987144939}$$

$$C_7 = \frac{1}{2}(1-\nu^2)\left(\frac{a}{b} - \frac{b}{a}\right)$$

$$C_7 = \frac{1}{2}(1-0.3^2)\left(\frac{37.187}{35.5} - \frac{35.5}{37.187}\right)$$

$$C_7 = \underline{0.042263331}$$

$$M_{ra} = (-798.75)(37.187)\left[0.043621994 - \frac{0.042263331 \times 0.000997704}{0.987144939}\right]$$

$$M_{ra} = \underline{-1294.44}$$

$$\text{MAX } \sigma = \frac{6\, M_{ra}}{t^2}$$

$$\sigma = \frac{6\,(1294.44)}{(1.75)^2}$$

$$\sigma = \underline{2536} \quad psi$$

USING FORMULA FOR SPECIAL CASE WHEN:

$r_o = b$ (load at inner edge)

$$\frac{b}{a} = \frac{\frac{71}{2}}{37.187} = 0.9546$$

$$M_{ra} = K_{Mra} W a$$

$$M_{ra} = (-0.09)(798.75)(37.187)$$

$$M_{ra} = \underline{2673.3} \quad [in.lbs]$$

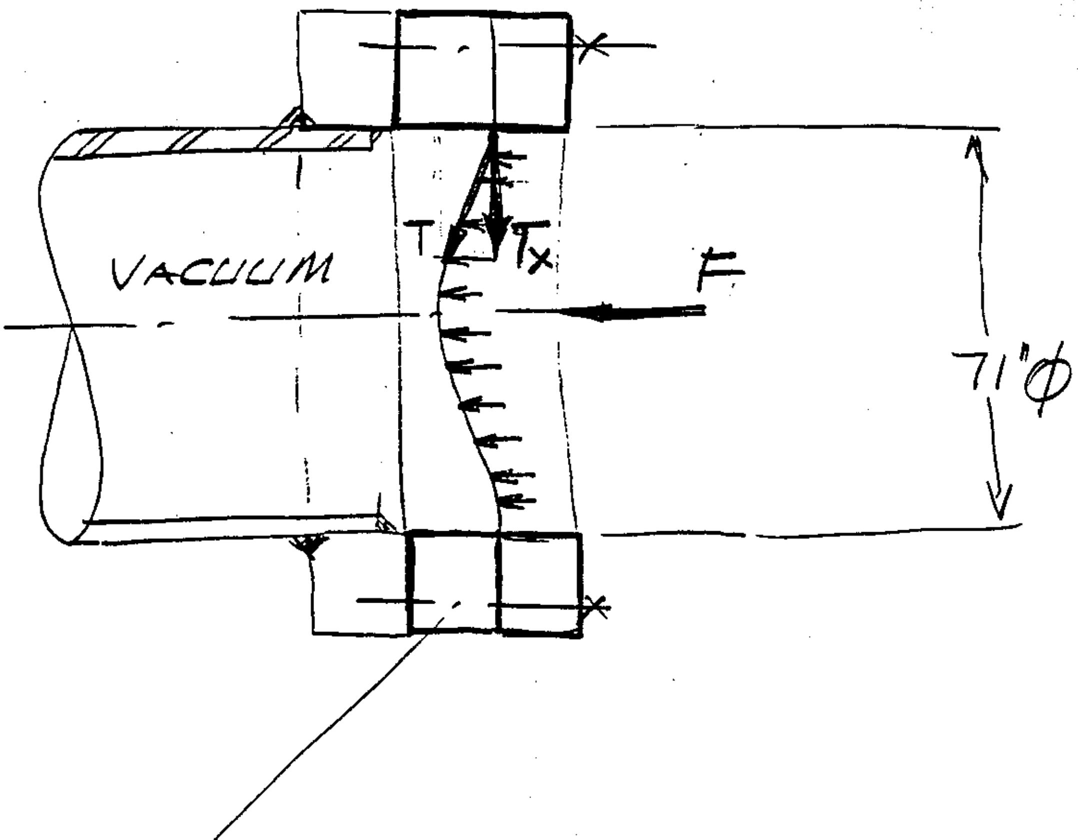
Max σ in this case:

$$\sigma = \frac{6\,(2673.3)}{(1.75)^2}$$

$$\sigma = \underline{5237.4} \quad psi$$

THE ABOVE COMPUTATION USES COEFFICIENT OF -0.0911 FOR $b/a = 0.9000$.
THIS IS CONSERVATIVE APPROACH.
BOTH METHODS OF ANALYSIS GIVE A SAFE RANGE OF THE STRESSES.

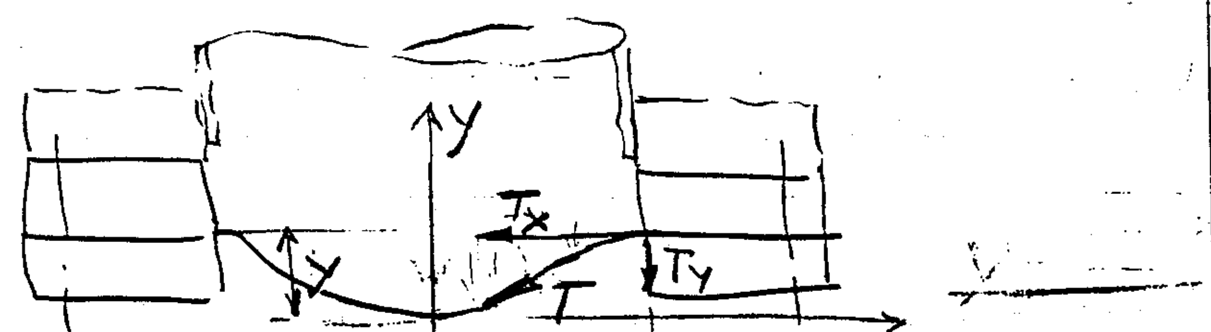
IV. CHECKING THE ~~HOOP~~ STRESS IN THE WINDOW ASSEMBLY

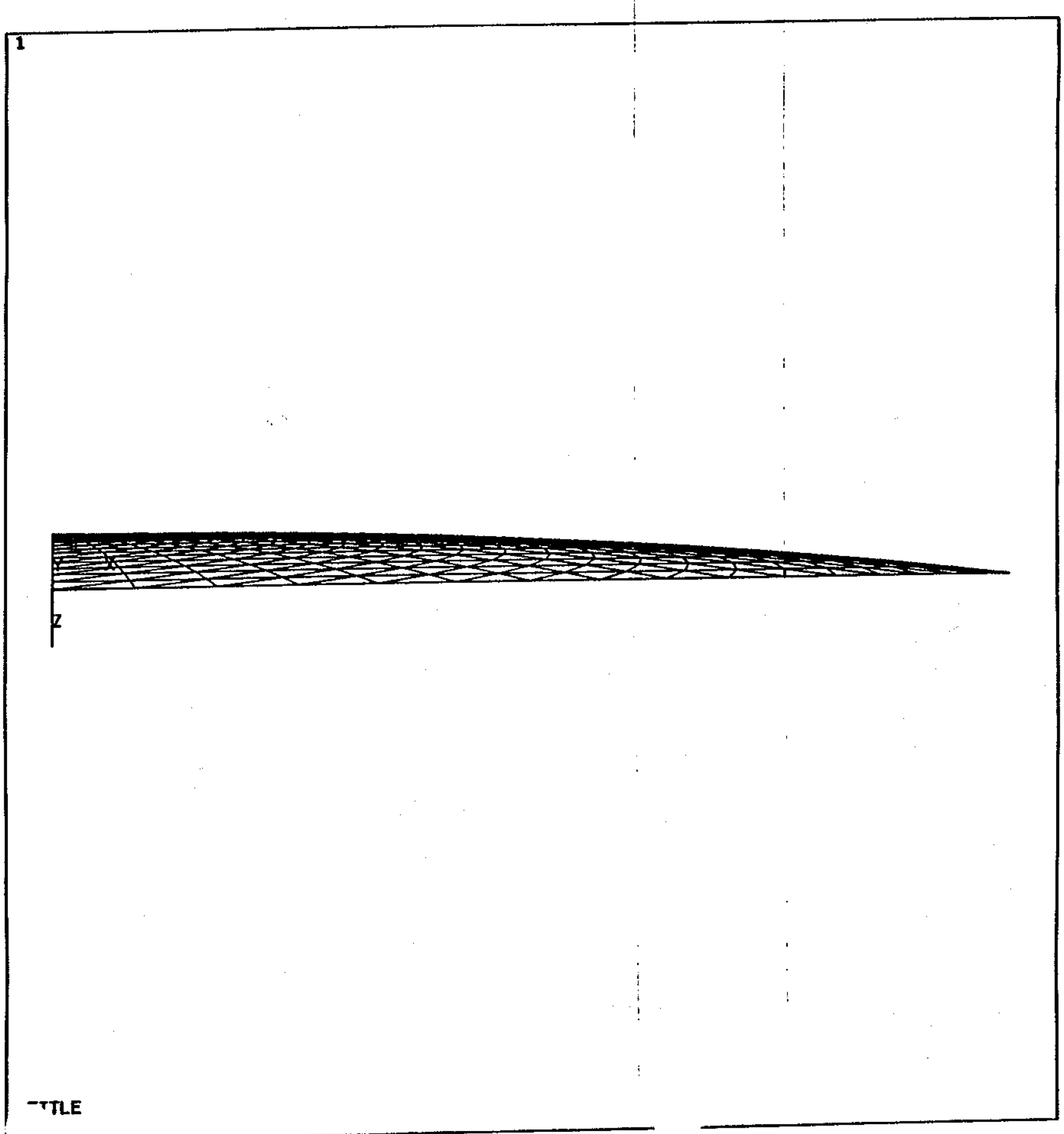


WINDOW ASSEMBLY
DWG. MD-285394

IV. 1 FIND THE RADIAL FORCE T_x (VACUUM CASE)

THE COMPUTATION HAS BEEN DONE USING ANSYS FINAL ELEMENT PROGRAM.





ANSYS 4.4A1
NOV 17 1992
17:17:32
PLOT NO. 1
POST1 DISPL.
STEP=2
ITER=20
DMX =7.7

DSCA=0.253055
YV =1
DIST=19.487
XF =17.715
YF =17.715

$D_{max} = 7.7''$. For 45 psi

since: $D \propto p^{\frac{1}{3}}$

for $p = 15$ psi $D = 5.28''$

Then;

For $p = 45$ psi:

$D \approx \left(\frac{45}{15}\right)^{\frac{1}{3}} \cdot 5.28''$

$= 7.615''$

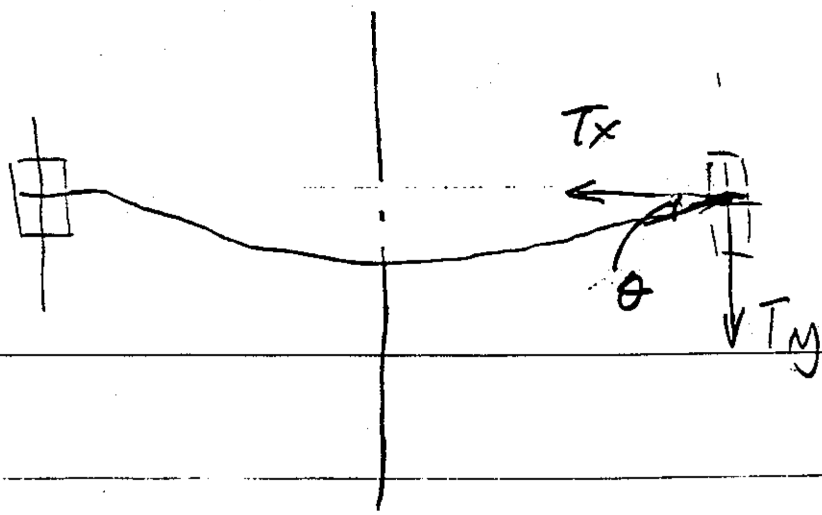
$\boxed{\widetilde{ANSYS} = 7.7''}$ ok ✓

To: A.Szymulanski
From: A.Lee
Subject: Force Tx and Ty

FINAL ELEMENT ANALYSIS
ANSYS 4.4A1

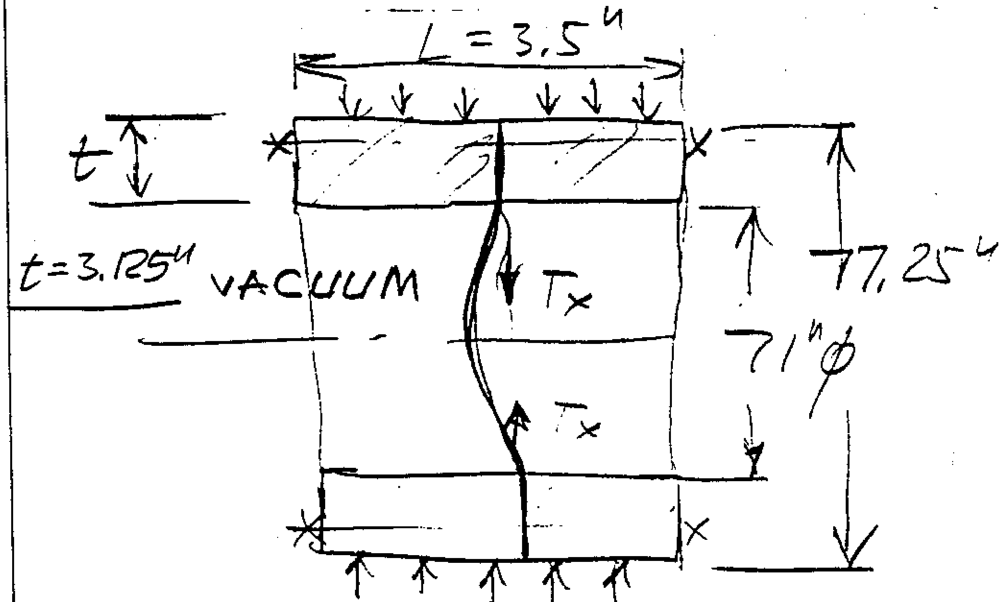
Case 1: p=45 psi.   Tx=2.9628E5 lbf ; Ty = 1.6824E5 lbf; angle=29.6

Case 2: p=14.7 psi. Tx=1.5488E5 lbf ; Ty = 5.652E4 lbf; angle=20.04



FIND THE RADIAL STRESS IN THE WINDOW FLANGES:

LET TREAT THE ASSEMBLY AS VERY SHORT VESSEL WITH EXTERNAL PRESSURE.



P = 14.7 [psi]

T_x – (FROM PAGE 39)

T_x = 154,880. [lbs]

CONVERTING THE RADIAL FORCE TO EXTERNAL PRESSURE:

$$p_e = \frac{T_x}{\pi(77.25)(3.5)}$$

$$p_e = \frac{154\,880}{\pi(77.25)(3.5)}$$

$$p_e = 182.33 \text{ [psi]}$$

TOTAL EXTERNAL PRESSURE

$$p_T = 14.7 + 182.33 = 197.03 \text{ [psi]}$$

- FIND THE CRYTICAL PRESSURE FOR THIS ARRANGEMENT -

$$P_c = \frac{S_y\, t}{R_o}$$

REF - PRESSURE VESSEL DESIGN HANDBOOK H.H. BEDNAR

(DESIGN OF CYLINDRICAL SHELL — SHORT CYLINDERS EXTERNAL PRESSURE page 50)

$S_y = 12700$

$$P_c = \frac{12700\,(3.125)\,(2)}{77.25}$$

$$P_c = \underline{1027.5} \; [psi] \qquad \frac{\frac{lbs}{in^2}\,(\cancel{in})}{\cancel{in}}$$

TOTAL EXTERNAL PRESSURE DERIVATIVE STRESS:

$$197.03 = \frac{S_y\, t}{R_o}$$

$$\underline{S_y = \sigma_v}$$

$$\sigma_v = \frac{(197.03)(77.25)/2}{3.125}$$

$$\sigma_v = \underline{2435.4} \; [psi]$$

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4.

PRESSURE VESSEL DESIGN HANDBOOK

Second Edition

Henry H. Bednar, P.E.

VNR VAN NOSTRAND REINHOLD COMPANY
New York

6

KTEV E-832 | A. S2. | 43.

using 0.6P instead of 0.5P, where E is the Code weld joint efficiency and S is the allowable Code stress. See also reference 169.

Both σ_t and σ_L are principal stresses, without any shear stress on the side of the differential element. However, at other section planes the shear stress appears as in the following example.

Example 3.1. If a spiral-welded pipe (Fig. 3.5) is subjected to internal pressure P, determine the normal and shear forces carried by a linear inch of the butt weld.

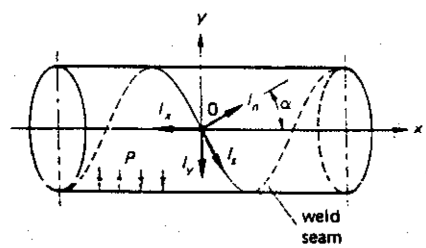


Fig. 3.5.

At point O,

$$l_x = PR/2 \quad \text{and} \quad l_y = PR \text{ lb/in.}$$

Normal tension is

$$l_n = (l_x + l_y)/2 + [(l_x - l_y)\cos 2\alpha]/2$$
$$= (PR/4)(3 - \cos 2\alpha).$$

Shear is

$$l_s = [(l_x - l_y)\sin 2\alpha]/2 = -(PR/4)\sin 2\alpha.$$

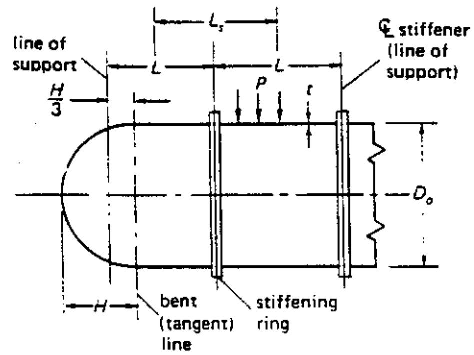
As a practical rule the minimum thickness of *carbon steel* cylindrical shells is not less, for fabrication and handling purposes, than the thickness obtained from the following empirical formula:

$$\text{min.}\, t = [(D_i + 100)/1000] \text{ in.}$$

where D_i is the shell inside diameter in inches.

Under Uniform External Pressure

To compute membrane compressive stresses in a cylindrical shell under uniform external pressure the internal pressure formulas can be used, if the pressure P is



L = distance between two lines of support
L_s = sum of the half of distances from stiffener to the lines of support on each side of stiffener
H = depth of head

Fig. 3.6.

replaced by $-P$. However, thin-wall vessels under external pressure fail at stresses much lower than the yield strength due to instability of the shell. In addition to the physical properties of the construction material at the operating temperature, the principal factors governing the instability and the critical (collapsing) pressure P_c are geometrical: the unsupported shell length L, shell thickness t, and the outside diameter D_o, assuming that the shell out-of-roundness is within acceptable limits (see Fig. 3.6).

The behavior of thin-wall cylindrical shells under uniform external pressure P differs according to cylinder length.

1. *Very Long Cylinders.* Subjected to a critical pressure P_c, the shell collapses into two lobes by elastic buckling alone (see Fig. 3.7), independent of the supported length L. The stiffeners or the end closures are too far apart to exercise any effect on the magnitude of the critical pressure. The only characteristic ratio is t/D_o and the collapsing pressure is given by the following equation [118]:

$$P_c = [2E/(1 - \nu^2)](t/D_o)^3$$

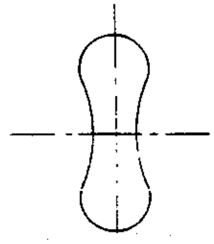


Fig. 3.7. Two-lobe collapse of a pipe under external pressure. The lobes may be irregular.

KTEV E·832 49

where E is the modulus of elasticity and for $\nu = 0.3$:

$$P_c = 2.2E(t/D_o)^3. \tag{a}$$

The minimum unsupported length beyond which P_c is independent of L is called the critical length L_c and is expressed by the equation

$$L_c = 1.14(1 - \nu^2)^{1/4} D_o(D_o/t)^{1/2}$$

and for $\nu = 0.3$

$$L_c = 1.11 D_o(D_o/t)^{1/2}.$$

2. *Intermediate Cylinders with $L < L_c$.* If the length L is decreased below the critical distance L_c, the critical pressure P_c and the number of lobes in a complete circumferential belt at collapse n will tend to increase and will become dependent on two characteristic ratios (t/D_o) and (L/D_o).

In the formula for P_c from reference 106.

$$P_c = KE(t/D_o)^3 \text{ psi}$$

the value of the collapse factor K and the number of lobes n that would produce the minimum P_c for given D_o/t and L/R_o can be read from the charts plotted for carbon steel shells (for $\nu = 0.3$) in refs. 120 and 121.

3. *Short Cylinders.* If L becomes short enough the cylinder will fail by plastic yielding alone at high stresses close to the yield strength of the material. The ordinary membrane stress formulas can be used: $P_c = (S_y t/R_o)$. This type of failure is common only in heavy-wall cylindrical shells. At this point the influence of L on P_c becomes very small or negligible. In the design of the vacuum thin-wall pressure vessels only the cases 1 and 2 have practical significance.

The theoretical elastic formulas for the critical pressure P_c at which intermediate cylinders would collapse under radial uniform external pressure or under uniform radial combined with uniform axial pressure, are derived in refs. 25 and 118. However, their solutions depend on n, the number of lobes at collapse, and they are cumbersome for a routine design.

To eliminate the dependency on n and to simplify the whole procedure of computation for the wall thickness t for the vacuum vessels at different design temperatures, the Code adopts the following procedure (UG-28). For cylindrical shells with $L < L_c$ the more accurate elastic formulas are replaced by the U.S. Experimental Model Basin formula, which is independent of n and is of sufficient

KTEV E832 46

accuracy:

$$P_c = \frac{2.42E}{(1-\nu^2)^{3/4}} \left[\frac{(t/D_o)^{2.5}}{(L/D_o) - 0.45(t/D_o)^{0.5}}\right]$$

where $0.45(t/D_o)^{0.5}$ can be disregarded and for all practical cases, using $\nu = 0.3$, P_c can be simplified to

$$P_c = 2.60E(t/D_o)^{2.5}/(L/D_o). \tag{b}$$

If equations (a) and (b) are substituted for pressure in the tangential stress formula $P = 2S(t/D_o)$, we obtain a set of two equations giving the tangential stress S_c at collapse:

$$S_c/E = 1.1(t/D_o)^2 \tag{1}$$

for cylinders with $L > L_c$ and

$$S_c/E = 1.30(t/D_o)^{1.5}/(L/D_o) \tag{2}$$

for cylinders with $L \leqslant L_c$. Both equation (1) (vertical) and (2) (slanting) are plotted with S_c/E values as variables on the abscissa and (L/D_o) values as variables on the ordinate for constant ratios (t/D_o) in Fig. UGO-28.0 (see Appendix A2). For better clarity the plot is labeled as (D_o/t).

Since S_c/E is here treated as a variable factor A, this *geometric chart* can be used for all materials. To introduce the particular physical properties of the material, an additional material chart is required, relating the value of the collapsing ratio S_c/E to the collapsing pressure P_c for the particular material. The material chart is actually a strain-stress curve (S_c/E)-$(S_c = P_cD/2t)$ for the material at a design temperature. To obtain the coordinate in terms of the allowable working pressure $P_a = P_c/4$ with a safety factor of four, the hoop formula is again employed:

$$S_c = P_cD_o/2t = (4P_a)\,D_o/2t$$

and

$$P_aD_o/t = S_c/2.$$

The material chart is then plotted against the same abscissa S_c/E, called factor A (see Fig. A2.2 in Appendix A2), for a specific material and design temperature as ordinate P_aD_o/t, called factor B. Factor A ties the two plots together. Since

IV, 2 45 psi PRESSURE CASE

$$T_x = \underline{296\,280} \quad [\text{lbs}] \quad \left(\text{FROM PAGE 39}\right)$$

CONVERTING RADIAL FORCE TO EXTERNAL PRESSURE:

$$P_e = \frac{T_x}{\pi(77.25)(3.5)}$$

$$P_e = \frac{296280}{\pi(77.25)(3.5)}$$

$$P_e = \underline{348.8} \quad [\text{psi}]$$

EXTERNAL PRESSURE DERIVATIVE STRESS:

$$348.8 = \frac{S_y\, t}{R_o} \qquad S_y = \sigma_p$$

$$\sigma_p = \frac{348.8\,(R_o)}{t}$$

$$\sigma_p = \frac{348.8\left(\frac{77.25}{2}\right)}{3.125}$$

$$\sigma_p = \underline{4311.} \quad \text{psi}$$

V. CHECKING THE BOTTOM PLATE THICKNESS

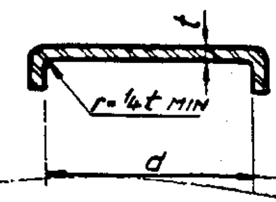
INTERNAL OR EXTERNAL PRESSURE

FORMULAS

NOTATION

P = Internal or external design pressure PSI **E = Joint effiency**
d = Inside diameter of shell, inches
S = Maximum allowable stress value of material, PSI, Page 159
t = Minimum required thickness of head, exclusive of corrosion allowance, inches
t_h = Actual thickness of head exclusive of corrosion allowance, inches
t_r = Minimum required thickness of seamless shell for pressure, inches
t_s = Actual thickness of shell, exclusive of corrosion allowance, inches

A

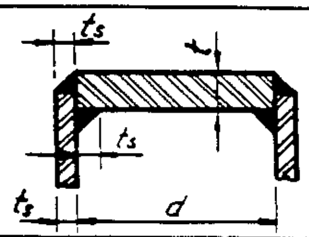


CIRCULAR FLAT HEADS

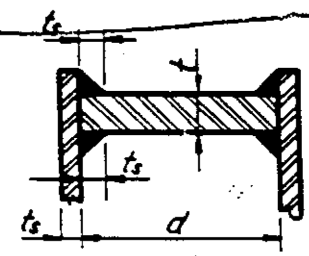
$$t = d\sqrt{0.13\,P/SE}$$

This formula shall be applied:
1. When d does not exceed 24 inches
2. t_h/d is not less than 0.05 nor greater than 0.25
3. The head thickness t_h is not less than the shell thickness t_s

B



C



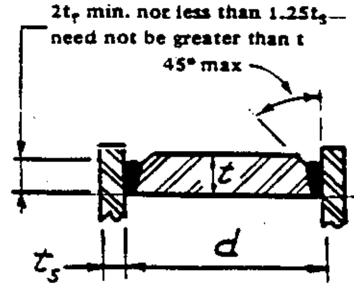
$$t = d \times \sqrt{CP/SE}$$

$$C = 0.33 \times \frac{t_r}{t_s}$$

C MIN. = 0.20

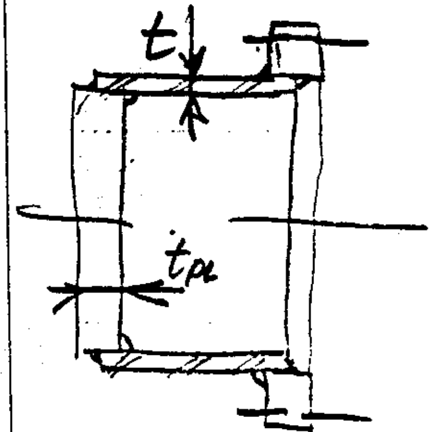
If a value of t_r/t_s less than 1 is used in calculating t, the shell thickness t_s shall be maintained along a distance inwardly from the inside face of the head equal to at least $2\sqrt{dt_s}$

D



Non-circular, bolted flat heads, covers, blind flanges Code UG-34; other types of closures Code UG-35

— FIND THE REQUIRED SHELL WALL THICKNESS.



$$t = \frac{P \cdot R}{SE + 0.4P}$$

R = OUTSIDE RADIUS

R = 35.5

P = 45 psi

S – STRESS VALUE ASSUMED 12700 psi

E = 0.7

$$t = \frac{45(35.5)}{12700(0.7) + 0.4(45)}$$

$$t = 0.179$$

$t = \frac{3}{8}$" IS SELECTED

for seamless shell

E = 1

t_r = 0.132

• FIND THE BOTTOM PLATE THICKNESS

$$t_{PL} = d \times \sqrt{\frac{CP}{SE}}$$

$$C = 0.33 \frac{0.132}{0.375}$$

$$C = 0.116$$

$$t_{PL} = 70.25 \sqrt{\frac{0.2\,(45)}{12700\,(0.7)}}$$

$$t_{PL} = \underline{2.23''}$$

$2\frac{1}{4}''$ THICK PLATE IS SELECTED.

THE CORRESPONDING STRESS WILL BE:

$$2.25 = 70.25 \sqrt{\frac{0.2\,(45)}{\sigma_{PL}\,(0.7)}}$$

WHERE σ_{PL} – STRESS IN THE PLATE

$$2.25 = 70.25 \left[\frac{0.2\,(45)}{\sigma_{PL}\,(0.7)}\right]^{\frac{1}{2}}$$

$$0.03202847 = \left(\frac{12.85714286}{\sigma_{PL}}\right)^{\frac{1}{2}}$$

$$\log 0.03202847 = \frac{1}{2}\log 12.85714286 - \frac{1}{2}\log \sigma_{PL}$$

$$\log \sigma_{PL} = 4.0978$$

$$\sigma_{PL} = 10^{4.0978}$$

$$\sigma_{PL} = 12{,}525.6 \quad PSI$$

VI. DRAWINGS SPECIFICATION

1.	1.8m Window Assembly	9220.832.MD-285394
2.	Vacuum Window Upstream Flange	9220.832.MD-285389
3.	Vacuum Window Upstream Flange	9220.832.MD-285390
4.	Vessel Assembly	9220.832.MD-285391
5.	Test Vessel	9220.832.MD-285386
6.	Vacuum Vessel Flange	9220.832.MD-285387
7.	Flange Bolt Plug	9220.832.MB-327687