ANDREW SZYMULANSKI

R/D MECHANICAL DPT.

E - 832

1.8 m VACUUM WINDOW ;

93.75" I.D. VACUUM VESSEL

- ENGINEERING COMPUTATIONS

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" MECHANICAL SAFETY SUBCOMMITTEE GUIDLINE
FOR DESIGN OF THIN WINDOWS FOR VACUUM VESSEL "

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17. CHECKING THE REQUIRED WALL THICKNESS OF A COVER (OR BOTTOM) PLATE FOR VACUUM/PRESSURE TEST FIXTURE

18. DWGS.: 9220.832.MD-285382
MD-285383
MD-285391
MD-285394
MD-285387
MD-285389
MD-285390

19. REFERENCES

1. OBJECTIVE

TO DESIGN, FABRICATE AND TEST THE WINDOW, USING PROVEN DESIGN AND MANUFACTURING APPROACH OF Ø 48" AND Ø 30" VACUUM WINDOWS. THE WINDOWS WERE USED IN E-731, E-773 AND Ø 48 IN E-799.

THE NEW WINDOW HAS (1.8 M) 71" INSIDE DIAMETER AND WILL USE THE SAME IDEA OF SUPPORTING THE MYLAR FILM.

2. DESIGN DESCRIPTION

THE WINDOW ASSEMBLY COMPRISES MYLAR/KEVLAR/MYLAR, COMPOSITE SANDWICHED BETWEEN TWO STEEL FLANGES. THE FLANGES ARE BOLTED TOGETHER AND WHOLE ASSEMBLY IS BOLTED AGAIN TO THE FLANGE OF A VACUUM PIPING SYSTEM.

3. SYSTEM SCHEMATIC

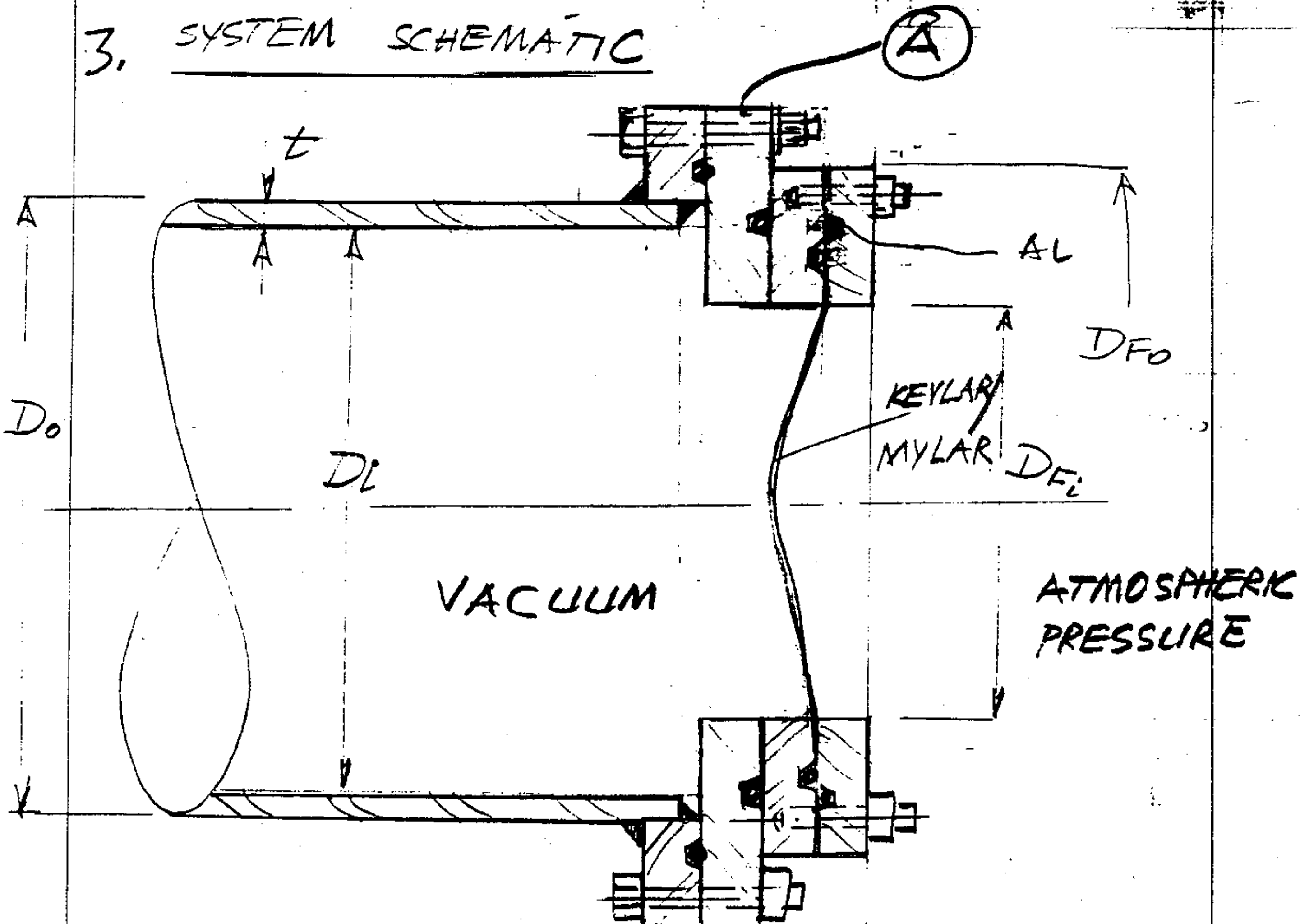

FIG. A1.

D_o – OUTSIDE DIAMETER OF THE PIPE $D_o = 94.5''$

D_i – INSIDE DIAMETER OF THE PIPE

$t = \frac{3}{8}''$

D_{Fo} – FLANGE OUTSIDE DIA.

D_{Fi} – FLANGE INSIDE DIA

$D_i = 93.75''$

$D_{Fi} = 1.8m = \underline{71''}$

REV.	DESCRIPTION	DRAWN / APPD.	DATE / DATE
A	1) ADDED ITEM 12 2) WAS 16.50 3) WAS 16.50" SQ 4) WAS 1.81 5) WAS 2.50 LG. P/N 9503	CLIFF PITTS	08-31-89

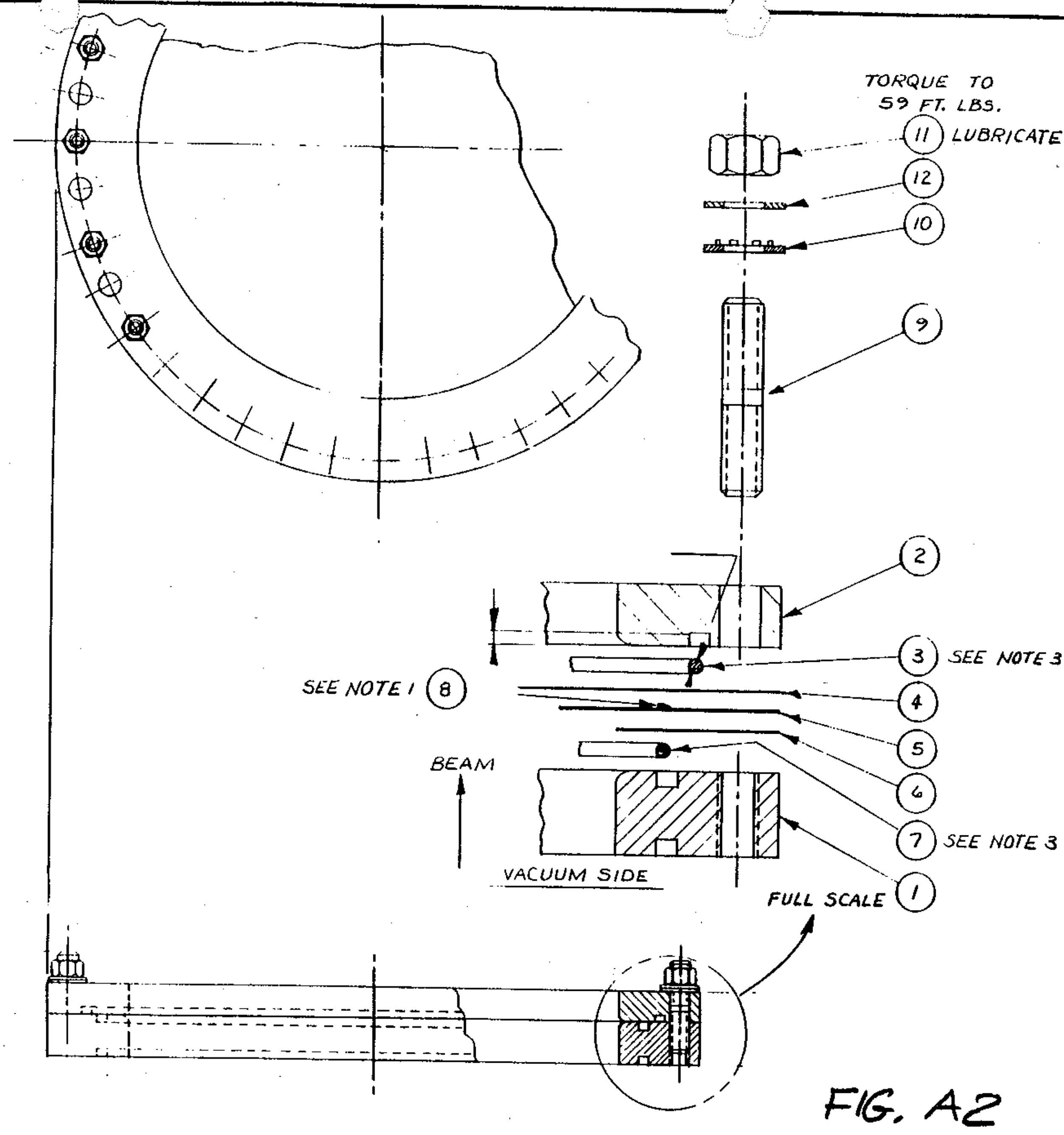

FIG. A2

ITEM NO.	PART NO.	DESCRIPTION OR SIZE	QTY. REQ.
12		FLAT WASHER GRADE 8	
11	—	HEX NUT GRADE 8	
10	—	LOAD INDICATOR WASHER ASTM F959 TYPE 325	
9	—		
8	—	EPOXY — RESIN EPON 826 "SHELL OIL PROD.", & EM 308 (SEE NOTE 1) "THIOKOL CHEMICAL CORP." (AVAILIABLE THRU TECH SERVICE)	A/R
7	—	O-RING ORANGE CORD .25 DIA. × 42" LG. APPROX.	1
6	MYLAR TYPE	MYLAR RING .005 THK. 12.00 I.D × 16.75" O.D.	1
5	—	KEVLAR-29 FABRIC STYLE 735 MILITARY SPEC: MIL-C-44050 SPECIFICATION: 1500 DENIER KEVLAR-29, 2×2 BASKET WEAVE, 35×34 COUNT 1800-1821LBS. TENSILE STRENGTH VENDOR: CLARK-SCHWEBEL (16.75" SQ.) FIBER GLASS CORP. P.O. BOX 851C WHITE PLAINS, N.Y. 10603	1
4	MYLAR TYPE	MYLAR .005 THK. × 14.75" SQ.	1
3	—	ALUM. WIRE ALLOY 1100 .188 DIA. × "LG. APPROX.	1
2		VACUUM WINDOW - COVER FLANGE	1
1		VACUUM WINDOW - INNER FLANGE	1

PARTS LIST

UNLESS OTHERWISE SPECIFIED			ORIGINATOR		
FRACTIONS ±	DECIMALS ±	ANGLES ±	DRAWN	CLIFF PITTS	08-15-89
1. BREAK ALL SHARP EDGES 1/64 MAX.			CHECKED	A. SZYMULANSKI	08.30.89
2. DO NOT SCALE DWG.			APPROVED	J.F. LINDBERG	8-30-89
3. DIMENSIONING IN ACCORD WITH ANSI Y14.5 STD's.			USED ON		
MAX. ALL MACHINED SURFACES			MATERIAL-		

FERMI NATIONAL ACCELERATOR LABORATORY
UNITED STATES DEPARTMENT OF ENERGY

ASSEMBLY, VACUUM WINDOW
RD./MECH. DEPT.

SCALE	FILMED	DRAWING NUMBER	REV.

NOTES
1. MIX RATIO 1:1, MININUM BATCH 200 GRAMS. USE WHEN CLEAR AND FREE OF AIR BUBBLES
2. ASSEMBLE O-RING ITEM 7 AND MYLAR RING ITEM 6 & KEVLAR FABRIC ITEM 6 TO ITEM 1. ON THE KEVLAR FABIC DIRECTLY OVER THE O-RING, LAY A BEAD OF EPOXY OVER THE ENTIRE DIA. OF THE O-RING, LEAVE THE EPOXY SOAK THROUGH THE KEVLAR FABRIC BEFORE COMPLETING REMAINING ASSEMBLY. THE REQUIRED TIME FOR CURING IS 24 HOURS.
3. WELD JOINT & FINISH TO BLEND MATING DIA.
4. VACUUM TEST: NO LEAK SHALL BE DETECTABLE ON THE MOST SENSITIVE SCALE OF A HELIUM LEAK DETECTOR WITH A MININUM SENSITIVITY OF 10^{-7} ATM. cc/SEC.

A Preliminary Information Memo

CHARACTERISTIC

INFORMATION DATA

Number 375

Date September 28, 1976

CHARACTERISTICS AND USES OF KEVLAR® 29 ARAMID

DU PONT COMPANY
TEXTILE FIBERS DEPARTMENT
WILMINGTON, DELAWARE, 19898

The information contained in this memo was prepared for rapid dissemination to meet special needs. Data and recommendations may be tentative and, as such, subject to change. For that reason, the information presented should be used only after consultation with appropriate Du Pont technical representatives to determine its currentness and validity.

THIS MEMO IS NOT INTENDED AS A PRODUCT SPECIFICATION

CHARACTERISTICS AND USES OF KEVLAR® 29 ARAMID

I. INTRODUCTION

KEVLAR is the registered trademark for one member of Du Pont's family of aromatic polyamic fibers*, which have been granted the generic name "aramid" by the Federal Trade Commission. KEVLAR 2 with a tensile strength of 400,000 lb/in² (2758 Mpa+) and modulus of 9 million lb/in² (62 000 MPa), especially suited for a number of industrial applications including ropes, cables, protective clothing, and coate fabrics. KEVLAR 49, which has a modulus of 19 million lb/in² (131 000 MPa) and the same tensile strengt as KEVLAR 29, is designed for the reinforcement of plastics and offers industry a new level of composi performance**.

KEVLAR 29 is supplied by Du Pont in filament yarns and staple fibers; product descriptions ar shown in Table I. Fabrics and nonwoven felts are also being produced commercially from these fibers and yarns

This bulletin describes the properties of KEVLAR 29 and typical applications, including fabrics an other products. More detailed technical information and current prices are available upon request.

TABLE I
YARNS OF KEVLAR® 29 ARAMID

Type 960
Yarn for Ropes and Cables with Special Finish for Improved Abrasion Resistance

Denier	Decitex*	Yield		Filament	Twist†	Nominal Yarn Diameter***	
		yd/lb	(m/kg)**			10^{-3} in	mm
1500	1670	2976	6000	1000	0	21.2	0.54
9000	10000	497	1000	4000	0	52.0	1.32
15000	17000	298	600	10000	0	67.1	1.70

Type 961
Yarn for Ropes and Cables with Standard Finish

Denier	Decitex*	Yield		Filament	Twist†	Nominal Yarn Diameter***	
		yd/lb	(m/kg)**			10^{-3} in	mm
1000	1110	4464	9000	666	0	17.3	0.44
1500	1670	2976	6000	1000	0	21.2	0.54
9000	10000	497	1000	4000	0	52.0	1.32
15000	17000	298	600	10000	0	67.1	1.70

Type 964
Yarn for Weaving Application

Denier	Decitex*	Yield		Filament	Twist†	Nominal Yarn Diameter***	
		yd/lb	(m/kg)**			10^{-3} in	mm
200	220	22320	45000	134	0	7.8 .00005	0.20
400	440	11160	22500	267	0	11.0 .00010	0.28
1000	1110	4464	9000	666	0	17.3 .00024	0.44
1500	1670	2976	6000	1000	0	21.2 .00035	0.54

A

† Yarns are designated Rotoset
* Numbers have been rounded to conform to product descriptions adopted for uniformity in packaging and labeling.
** m/kg = yd/lb x 2.016.
*** Assuming 70% packing factor.

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* NOMEX is also included in this generic fiber category (see Du Pont bulletin entitled "Properties of NOMEX Aramid Fiber").
** For further information see Du Pont brochure entitled "Characteristics and Uses of KEVLAR 49 Aramid High Modulus Organic Fiber".
+MPa = MN/m² = psi x 6895 x 10^{-6}

II. FIBER PROPERTIES

The physical properties of KEVLAR 29 aramid fiber compared with those of conventional industrial nylon, DACRON* polyester, fiberglass and stainless steel are shown in Table II. It can be seen that the tensile strength of KEVLAR* 29 is more than twice that of nylon or DACRON, 15% greater than that of "E"-glass, and 60% greater than that of steel. Modulus, or stiffness, is more than 10 times that of nylon, almost 5 times that of DACRON, and is almost equivalent to that of "E"-glass. The fiber elongation-to-break is quite low compared with that of other organic fibers, and the density, while higher than that of nylon or DACRON, is about 1/2 that of glass, and 1/5 that of steel.

TABLE II
COMPARATIVE YARN PROPERTIES

	KEVLAR® 29 Aramid	DU PONT Nylon Type 728	DACRON® Polyester Type 68	"E-HTS" Glass	Stainless Steel
Tensile Strength, lb/in^2 (MPa)*	400,000** (2758)	143,000** (985)	162,500** (1120)	350,000*** (2412)	250,000 (1724)
Modulus, lb/in^2 (MPa)	9,000,000 (62000)	800,000 (5512)	2,000,000 (13780)	10,000,000 (68900)	29,000,000 (199800)
Elongation to Break, %	4.0	18.3	14.5	3.5	2.0
Density, lb/in^3 (g/cm^3)†	0.052 (1.44)	0.041 (1.14)	0.050 (1.38)	0.092 (2.55)	0.284 (7.83)

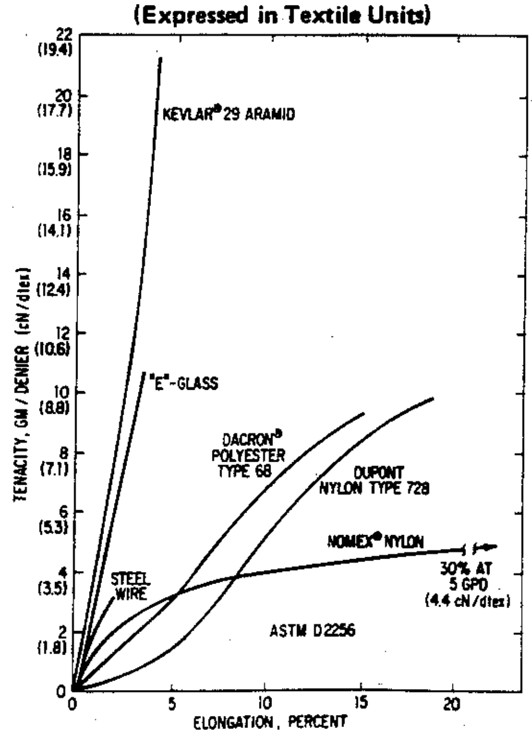
*MPa = MN/m^2 = lb/in^2 x 6.895 x 10^{-3}
**Unimpregnated twisted yarn test – ASTM D2256
***Impregnated strand test – ASTM D2343
†g/cm^3 = lb/in^3 x 27.68

The nearly linear stress/strain curve to failure of KEVLAR 29 is similar to that of glass, but unlike those of other organic fibers (Figure 1). Because it is relatively insensitive to fiber surface defects, the tensile strength of KEVLAR 29 is uniform along the length of the fiber; for example, a twisted 1500 denier (1670 dtex) yarn tested at 100 in. (2540 mm) gage length retains 95% of its 1 in. (25.4 mm) gage strength. KEVLAR 29, available in yarns of up to 15,000 denier** (16 700 decitex)***, is comprised of many continuous, round cross-section filaments each having a denier of about 1.5 (1.7 decitex) and a diameter slightly under 0.5 mil (0.013 mm).

Tensile strength is measured by ASTM D2256 using yarn samples twisted to 1.1 twist multiplier (T.M.) on 10 in. (254 mm) gage length at 50% per minute elongation rate. The formula for twist multiplier and the significant effect that twist level has on tensile strength are shown in Figure 2.

The 0.052 lb/in^3 (1.44 g/cm^3) density of KEVLAR 29 results in a higher specific† tensile strength than is currently available from any other material commercially available and a specific† molulus higher than that of glass fiber. These properties (Figure 3) form the key to market opportunities in ropes and cables and other uses where the ratio of strength to weight is important.

FIGURE 1
STRESS-STRAIN BEHAVIOR OF YARNS
(Expressed in Textile Units)



CONVERSION FACTOR $\frac{LB}{IN^2} = \left(\frac{GM}{DEN}\right) \times DENSITY \left(\frac{GM}{CM^3}\right) \times 12{,}800$

* Du Pont registered trademark
** Denier is weight in grams of 9000 meters
*** Decitex is weight in grams of 10,000 meters
† Tensile strength or modulus divided by density.

FIGURE 2
THE EFFECT OF TWIST ON THE TENSILE STRENGTH OF KEVLAR® 29 ARAMID
(Textile Units)

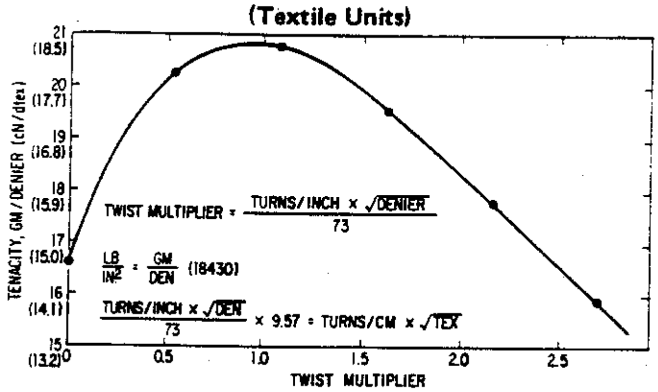
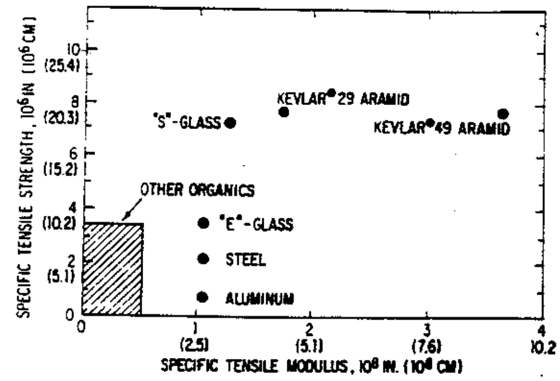


FIGURE 3
SPECIFIC TENSILE STRENGTH AND SPECIFIC TENSILE MODULUS OF FIBERS AND OTHER MATERIALS



KEVLAR** 29 also has high toughness which yields good textile processibility and high impact strength; for example, loop strength is 55% of straight breaking strength.

KEVLAR 29 has good thermal stability, retaining a high percentage of room temperature properties when tested up to 355° F (180°C) (Figure 4). The fiber exhibits virtually no shrinkage between room temperature and 320°F (160°C). KEVLAR 29 does not melt or support combustion under normal environmental conditions but will carbonize at about 800° F (427°C). At arctic temperatures of –50° F (–46°C), it exhibits essentially no embrittlement or degradation of fiber properties.

The chemical resistance of KEVLAR* 29 is excellent except in a few strong acids (Table III). The effect of ultraviolet light will vary with the thickness of the item exposed. Very thin fabric (4.5 mil, 0.114 mm), if exposed directly to Florida sunshine for a period of 5 weeks, will lose about half of its tensile strength

FIGURE 4
THE EFFECT OF TEMPERATURE ON THE TENSILE STRENGTH OF KEVLAR® 29 ARAMID

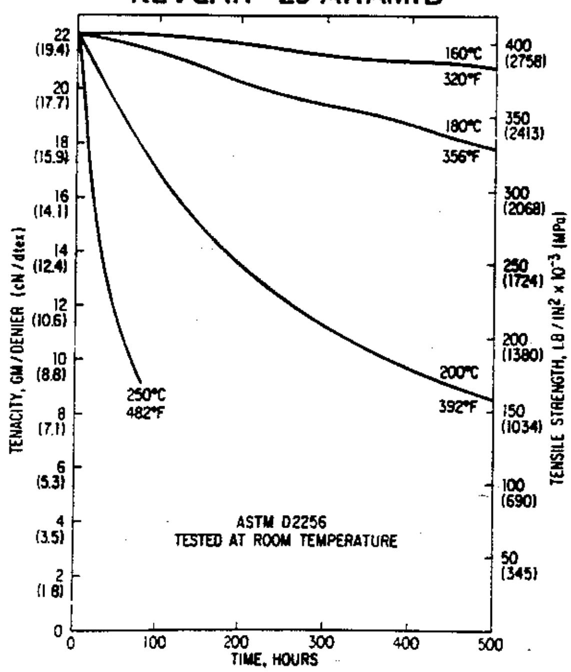


TABLE III
CHEMICAL RESISTANCE OF YARN OF KEVLAR® 29 ARAMID

Environment (100 hr* exposure at 70° F; 21°C)	Tensile Strength Loss %
ACIDS	
Formic (90%)	10
Hydrochloric (37%)	90
Hydrofluoric (10%)	12
Nitric (70%)	82
Sulfuric (70%)	100
OTHER CHEMICALS	
Brake Fluid (312 hr)	2
Greases (MoS_2 and Lithium base)	0
Jet Fluid (JP-4) (300 hr)	0
Ozone (1000 hr)	0
Tap Water	0
Boiling Water	0
Superheated Water 156°C (313°F) 80 hr	16

*Except where noted.

*Du Pont registered trademark.

(Table IV). In thicker items, such as the 1/2 in. (12.7 mm) diameter rope shown in the table, the majority of the yarns are protected by the outer layer and the strength loss is minimal. Although self-screening may be sufficient for some applications, the addition of opaque jacketing may be required for increased UV resistance under critical conditions.

KEVLAR* 29 has an equilibrium moisture level of 7% at 72° F (22°C) at 55 R.H., and a negative coefficient of thermal expansion of $-2 \times 10^{-6}/°C$ ($-1.1 \times 10^{-6}/°F$).

KEVLAR 29 has excellent dynamic and static fatigue resistance (Table V, Figure 5), as well as stress relaxation behavior (Figure 6). Creep rate is equivalent to that of fiberglass, but unlike glass, is much less susceptible to creep-rupture, even at levels as high as 70% ultimate tensile strength. Additional information on creep, fatigue, and impact properties is available upon request.

TABLE IV
ULTRAVIOLET STABILITY OF KELVAR® 29 ARAMID

Product Form	Break Load	Strength Loss (%)
Fabric, 4.5 mil (0.114 mm) thickness		
Unexposed	300 lb/in (525 N/cm)	–
Florida Sun (5 weeks)	154 lb/in (270 N/cm)	49
Rope, ½" (12.7 mm) diameter		
Unexposed	11,400 lb (50,700 N)	–
Florida Sun (6 mo.)	10,260 lb (45,700 N)	10

TABLE V
EFFECT OF TENSION-TENSION FATIGUE ON KEVLAR® 29 ARAMID

Cycled Between (% of Ultimate Tensile Strength) High	Low	No. of Cycles	Break Load After Cycling Lb	(N)	Decrease in Tensile Strength Due to Fatigue
Control		–	124	552	–
74	45	1000	130	578	None
52	29	1000	137	610	None
31	8	1000	132	587	None
10	0	13×10^6	118	525	5%

1500 denier (1670 dtex) 2-ply yarn of KEVLAR® 29 was tested using air-actuated 4-D cord clamps on an Instron test machine, at 10" (254 mm) original gage length, 10% per minute elongation, and at 55% R.H. and 72° F (22°C).

FIGURE 5
CREEP OF YARNS OF KEVLAR® 29 AND KEVLAR® 49 ARAMID AT 50% OF ULTIMATE STRENGTH

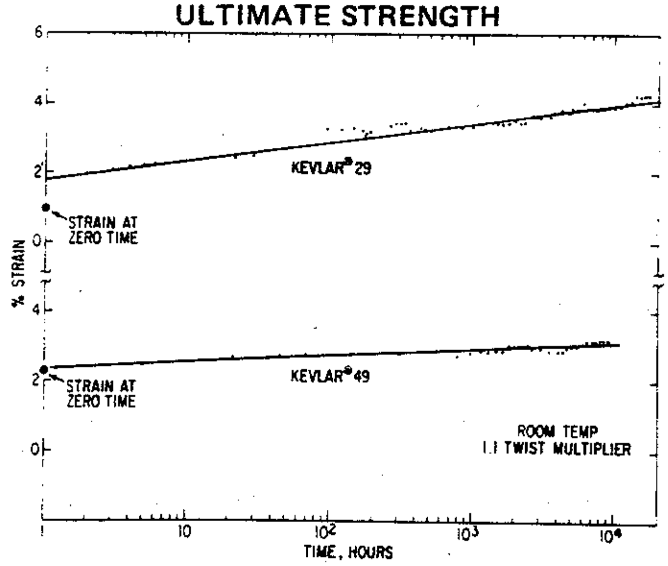
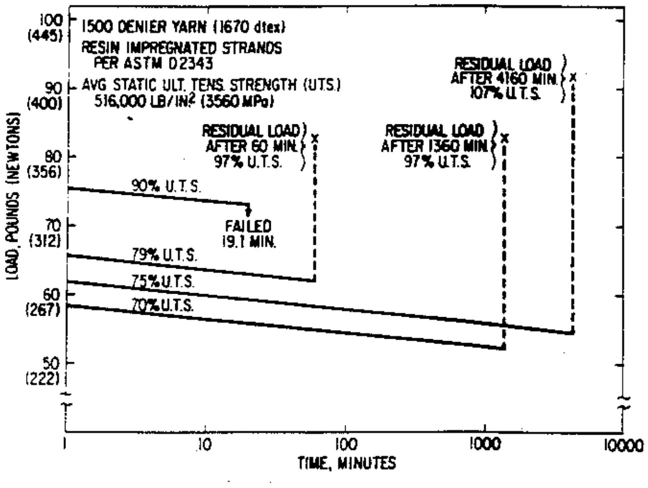


FIGURE 6
STRESS RELAXATION OF YARN OF KEVLAR® 29 ARAMID



*Du Pont registered trademark.

III. FABRICS

Woven fabrics of KEVLAR* 29 aramid have a balance of properties generally unattainable with traditional textile fibers or with fiberglass, including the high strength and stability of glass fabrics at significantly lower weight. They also have a balance of tensile and tear strengths superior to that attainable with other organic fibers, thus eliminating the necessity to over-construct fabrics to obtain high tear strength. Typical fabric properties are discussed under Applications.

Because of its unique combination of physical properties and organic composition, KEVLAR 29 is an inherently "tough" material. However, yarns, rovings, fabrics, and coated fabrics can be cut and trimmed using sharp, clean tools. Scissors must have close tolerances between cutting surfaces and serrated scissors** are preferred since they prevent the material from slipping out from between the cutting surfaces. Good quality canvas, upholstery, and carpet shears (for example, Wiss #4 I.S.) or electric scissors (for example, United Cloth Cutting Machine Co. #LIL 25159) will also satisfactorily cut fabrics of KEVLAR 29. The canvas shears are also available with serrations**.

Multiple plies of fabrics of KEVLAR 29 can be cut using a sabre saw with a special blade tipped with tungsten carbide**. Additional information on cutting is available upon request.

IV. APPLICATIONS

A. Ballistics

Fabrics of KEVLAR 29 are being extensively used in ballistic garments to protect law enforcement officers from handgun threats and in military flak jackets and helmets. KEVLAR 29 is also being evaluated in other ballistic protective end uses such as armored cars and trucks, bomb disposal blankets, military personnel carriers, bank teller cages, explosives storage and transportation cases, military aircraft, jet engine blade containment, and blankets and curtains surrounding high speed machinery. Fabrics may be used untreated in "soft" armor, or impregnated with thermoset, thermoplastic, or elastomeric resin systems and then molded into "flexible" or "hard" armor which have structural integrity as well as ballistic resistance.

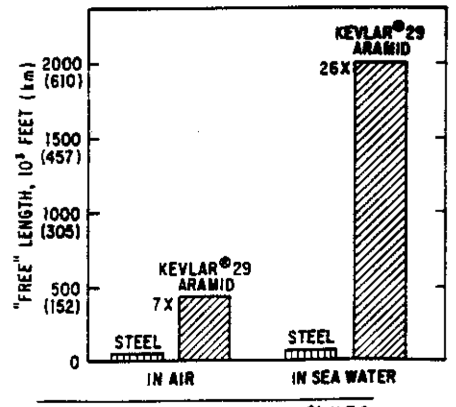
B. Protective Apparel

High strength, cut resistance, and thermal insulating characteristics of KEVLAR 29 make it attractive for protective apparel applications. Gloves made from continuous filament or spun yarn fabrics show good cut resistance and durability in metal and glass handling operations. When these same glove constructions are backed by felts of KEVLAR 29, additional thermal and puncture protection are obtained. Combinations of ballistic fabric and felt of KEVLAR 29 are being evaluated for lightweight protective aprons for meat cutters and knee pads for chainsaw operators.

C. Ropes and Cables

KEVLAR 29 is replacing steel in tension members and cables where very high specific tensile strength, low stretch, good electrical properties, cyclic and creep fatigue resistance, and toughness are important. The high strength-to-weight ratio is the primary reason for the use of KEVLAR 29 in very long cables of up to 5 miles (8 km), such as those used in oceanographic and aerospace markets. Figure 7 illustrates the "free" length of KEVLAR 29 that will support itself in both air and water, as compared with steel. KEVLAR 29 is also replacing other organic fibers such as DACRON* polyester or nylon in applications where low stretch, or reduced weight and diameter (at equivalent strength), is desirable.

FIGURE 7
"FREE" LENGTH COMPARISON
(Length at which Strength Member Breaks of Its Own Weight = Tensile Strength/Density)



BRAIDED ROPE EFFICIENCY = ~80% U.T.S.

*Du Pont registered trademark.
**Available for Technology Associates, Inc., P.O. Box 7163, Wilmington, Delaware 19803.

For additional information on "Kevlar" 29 aramid, please contact:

E. I. DU PONT DE NEMOURS & CO. (INC.)
TEXTILE FIBERS DEPARTMENT
KEVLAR® SPECIAL PRODUCTS
CENTRE ROAD BUILDING
WILMINGTON, DELAWARE 19898

E-21451

Printed in U.S.A

...all-aramid approach to aircraft design

...description by the company as the fastest, most fuel-efficient airplane of its kind is not an idle claim for the Avtek 400 business jet, nor does it augur poorly for the future of composites in the aircraft industry. In fact, the all-composite plane was able to surpass the goals its designers had set by virtue of the lower weight, lower cost, and better aerodynamic shapes afforded by composites.

Says Leo Windecker, vice president of research and a designer of the nine-seat plane for Avtek Corp. (Camarillo, Calif.), "Composites give designers a freedom that doesn't exist with flat aluminum sheets, which don't allow the combination of complex shapes. And they offer a cost advantage as well. The fuselage of the Avtek plane has six basic large pieces that are bonded together; a similar plane in aluminum would have required about 5000 small parts, all of which would have to be riveted together by hand."

For the first plane built entirely of aramid composites (load-bearing structure and secondary parts), Windecker is projecting excellent performance. The airframe of a comparable plane would weigh 4000 to 6000 lb; the Avtek 400 weighs 1000 lb. Whereas other turboprops could expect to climb 2000 to 3000 ft/minute, fly 1500 to 1600 miles, and get as much as 5.5 mpg, the Avtek offers a climbing rate of 5400 ft/minute, an approximate flying range of 2600 miles, and a maximum fuel economy of 13.1 mpg.

Dramatic reduction in fasteners and sleek aerodynamic shapes are offered by the use of composites in the construction of the Avtek 400. Inset: Leo Windecker. (Photo: Avtek Corp.)

All exterior surfaces use Kevlar skins over a Nomex honeycomb core, and all spars and other parts in which stiffness is primary are made from carbon/epoxy composites. "Kevlar's compressive strength of 40,000 psi is quite good," Windecker says, "but where we wanted to increase it we blended glass or carbon with Kevlar." Hybrids were used in frames surrounding the windows and doors and in areas where there were hard-point attachments.

Construction of the Avtek 400 involved designing with the concept of multiple stress paths and at low stress levels, Windecker says. Traditional skeletal construction with primary stress paths, at the stringers and ribs, for example, was replaced with many stress paths throughout the pressurized cabin made in right and left halves. "Each structure incorporated much redundancy, so if there should be an overwhelming load, there would be additional passageways for it to travel without causing failure. The use of much larger structural areas also meant that design was done at far lower stress levels."

In considering the aerodynamic stresses and loads to determine efficient use of Kevlar, Windecker cites as an example the aerodynamic stress level that the wings must support. For instance, the weight of the airplane times the anticipated force of gravity (3.8 G) plus a safety factor of 150 percent is the stress level to which the wings must be designed.

A modified wet-layup procedure is currently being used for the Avtek 400 prototype only; for full production, a method entailing the use of vacuum bagging, robots for wetting the cloth, and a microwave for curing is anticipated.

son to other materials, aramid's density of 1.44 g/cm³ is 43 percent lower than glass and 12 to 30 percent lower than graphite; its tensile strength of 525,000 psi translates into five times the strength of steel and two and a half times that of E-glass on a pound-for-pound basis. Its tensile modulus, while less than that of graphite, is roughly three times as great as S-glass. Compressive strength and modulus of elasticity are lower than those of graphite. Inherently flame resistant and unsusceptible to melting, Kevlar has a continuous-use temperature range of −320F to +400F. This combination of properties has resulted in the use of Kevlar in aircraft (including ultralight home-built versions), aerospace, marine, and industrial areas.

*

Two of the three types of Kevlar aramid will be covered here—the highest-modulus Kevlar 49, primarily intended as a continuous-filament plastic reinforcement for structural parts ranging from airplane propellers to auto bodies; and Kevlar 29, which is primarily used for apparel and rope but is also available in a short chopped form and as a highly fibrillated pulp product for plastic reinforcement. The latter two versions are selected for applications where frictional properties, chemical resistance, or tear strength, among other properties, take precedence over light weight. These materials are frequently used as asbestos replacements in brake blocks for heavy-duty trucks, automotive disk brakes, clutch facings, and gasket sheeting for industrial automotive use.

Experiments currently under way with short (6 mm), chopped-fiber, and pulp-form Kevlar for use in bulk molding compounds have thus far indicated that chopped fibers are preferable where high strength, modulus, and impact resistance are required; on the other hand, the fibrillated form offers greater fiber surface area and better fiber interlocking to provide a strong reinforcement. In either case, proper mixing techniques, which allow aramid to dwell a longer time than glass in the mixer for full "opening" of the fibers and their even distribution throughout the compound, can enhance performance.

Along with the dynamic parts in electrical, automotive, and industrial equipment in which short reinforcements are finding a market, elastomeric applications—including seals for dril

DEVELOPMENT

INC.

June 27, 1990

Mr. Andrew Szymulanski
Fermi-Lab
M/S 221
P.O. Box 500
Batavia, Illinois 60510

Dear Andrew:

It was nice discussing with you regarding your requirements of 130" wide Kevlar 29 fabric and I am providing the following information.

FABRIC DEVELOPMENT INC. is an independent research and development company, offering its services to fiber producers, different agencies of the government and aerospace industries in general.

We have a complete facility for developing and supplying specially woven, knit and braided fabrics for specific end use requirements. We can work with all available natural, synthetic, as well as high modulus and high strength yarns, such as Thornel 300, Kevlar 29, Kevlar 49, 's' glas, Ceramics, Quartz, Teflon, Nomex, etc.

We have developed and supplied various styles of fabrics from Kevlar 49 and 29, graphite, Nextel 312, Nicalon, Quartz, Spectra, Teflon, Gore-tex, etc. yarns to meet specific end use requirements.

I am enclosing our brochure describing our general activities in brief.

If I can be of any further assistance to you, please feel free to let me know.

Very truly yours,

Piyush A. Shah/mg

Piyush A. Shah
President
FABRIC DEVELOPMENT INC.

PAS:mg
Enc.

Phone: (215) 536-1420

1217 Mill Street
P.O. Box 462
Quakertown, PA 18951

Fax: (215) 536-1154

Look to Us for Endless Possibilities in Textile Development

E-832

PRODUCT DATA SHEET

CATEGORY

KEVLAR-29 FABRICS

CLARK-SCHWEBEL FIBER GLASS CORPORATION

5 CORPORATE PARK DRIVE WHITE PLAINS, N.Y. 10604 • Tel. 914 - 694 - 9090

No. 11D-11

DATE JULY 1, 199

PAGE 1 OF 1

STYLE	WEIGHT OZ./SQ.YD.	THICKNESS* INCHES	COUNT	TENSILE LBS./INCH WARP	FILL	YARN DENIER WARP	FILL	WEAVE
710	9.4	.017	24x24	1100	1200	1500	1500	PLAIN
713	8.3	.015	31x31	900	930	1000	1000	PLAIN
728	6.6	.012	17x17	778	810	1500	1500	PLAIN
730	5.5	.010	22x22	650	725	1000	1000	PLAIN
732	3.2	.006	32x32	450	430	400	400	PLAIN
733	3.1	.008	60x60	475	475	200	200	BASKET 2x2
735	13.8	.023	35x35	1800	1821	1500	1500	BASKET 2x2
739	14.5	.025	39x34	1600	1900	1500	1500	BASKET 2x2
740	2.1	.005	40x40	337	327	200	200	PLAIN
745	13.6	.024	17X17	1600	1800	3000	3000	PLAIN
748	18.8	.032	48x48	2200	2300	1500	1500	BASKET 8x8
755	17.5	.029	21X21	2000	2000	3000	3000	BASKET 4x4
759	18.9	.032	24X24	2700	2700	3000	3000	BASKET 4x4

*ALL VALUES ARE BASED ON CS-800 (SCOURED) FABRICS. ALL DATA IS TO BE USED AS A GUIDE AND ARE NOT GUARANTEED VALUES.

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CLARK-SCHWEBEL FIBER GLASS CORPORATION

5 CORPORATE PARK DRIVE WHITE PLAINS, N.Y. 10604 • Tel. 914 - 694 - 9090

CATEGORY KEVLAR-29 FABRICS

No. 11-17

DATE JAN. 1, 1991

PRICE PER LINEAR YARD
WITH CS-800 (SCOURED) FINISH

C-S STYLE	WIDTH	UP TO 499 YDS.	500-1,499 YDS.	1,500-4,999 YDS.	5,000-9,999 YDS.	10,000 + YDS.
710	50"	$29.45	$27.85	$26.75	$26.05	$25.65
713	50"	29.90	28.30	27.20	26.50	26.10
728	50"	21.80	20.20	19.10	18.40	18.00
730	50"	22.55	20.95	19.85	19.15	18.75
732	50"	19.45	17.85	16.75	16.05	15.65
735	50"	39.70	38.10	37.00	36.30	35.90
740	56"	22.15	20.55	19.45	18.75	18.35

CS-897 WATER REPELLENT FINISH SURCHARGE

ORDERS OF 150 MINIMUM
BUT LESS THAN 5,000 YARDS..........................$1.25 PER LINEAR YARD

5,000 YARDS OR OVER................................$1.00 PER LINEAR YARD

93.75" I.D. VACUUM VESSEL

4. DETERMINING THE ACTING FORCE DUE TO THE ATMOSPHERIC PRESSURE.

$$P = \frac{\pi D_i^2}{4}(p)$$

WHERE p - ATMOSPHERIC PRESSURE

$$P = \frac{\pi (93.75)^2}{4}(14.7)$$

$$P = 101,473 \text{ lbs}$$

4.1 SD-41 REQUIREMENTS:

VACUUM PRESSURE VESSEL SAFETY STANDARD SD-41

3. POLICY

3.1 Fermilab's contract with the Department of Energy stipulates that Fermilab conform to the current DOE 6430.1 standard. This standard includes the A.S.M.E. Pressure Vessels Code. The A.S.M.E. Code does not apply to vacuum vessels, but nevertheless we shall adhere to it to the extent practical for vacuum vessels in a laboratory environment. The design criterion for vacuum vessels at Fermilab shall be a minimum collapsing pressure of 30 psi differential, (15 psi external differential with a minimum safety factor of two).*

WINDOW
93.75" I.D. VACUUM VESSEL

$$P_{(30)} = \frac{\pi D_i^2}{4}(30)$$

$$P_{30} = \underline{207087.} \quad lbs$$

5. DETERMINING THE WALL THICKNESS OF FLANGE "A" FIG. A3

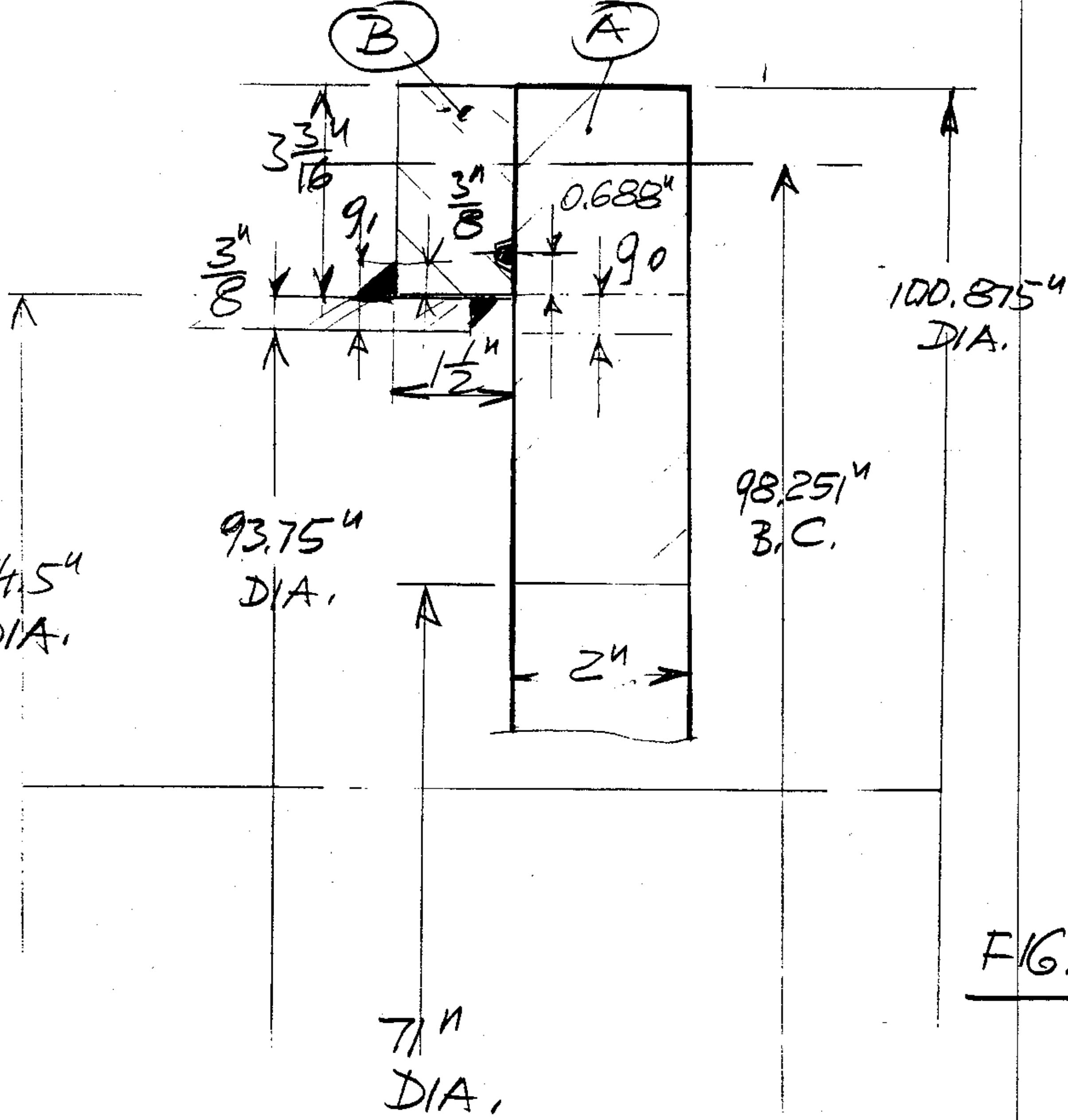

FIG. A3

COMPUTATION, ACCORDING TO: ASME APPENDIX 14 (14-20)

"INTEGRAL FLAT HEADS WITH A LARGE, SINGLE, CIRCULAR, CENTRALLY-LOCATED OPENING."

CALCULATE THE OPERATING MOMENT M_o

$$M_o = W \frac{(C-G)}{2}$$

$W = W_{m1}$

$W_{m1} = H + H_p = 0.785\, G^2 P + (2b \times 3.14\, G m P)$

$G = 0.688(2) + 94.5$

$G = 95.876''$

$C = 98.251''$

$b = 0.315''$

$W_{m1} = 0.785(95.876)^2(30) + (2 \times 0.315 \times 3.14 \times 95.87 \times 0.5 \times 30)$

$W_{m1} = 216476.48 + (2844.75)$

$W_{m1} = \underline{212{,}105.35}$

$$M_o = 212{,}105.35 \frac{98.251 - 95.876}{2}$$

$M_o = \underline{251{,}875.1}$ [in lb]

FOR AN OPENING WITHOUT A NOZZLE:

$$(EQ)^* = \left(\frac{B_n}{t}\right) S_T^*$$

$B_n = 71''$

$$S_T = \frac{Y M_o}{t^2 B} - Z \cdot S_R$$

Y - FACTOR INVOLVING K (FROM FIG. 2.7.1)

t - FLANGE THICKNESS

$t = 2''$

A - OUTSIDE DIA. OF FLANGE

$A = 100.875''$

B - INSIDE DIA. OF FLANGE

$B = 71''$

$$K = \frac{A}{B}$$

$K = 1.42$

FROM FIG. 2.7.1

$Y = 5.8$

$Z = 3.1$

$$S_R = \frac{(1.33\,te + 1)\,M_o}{L\,t^2\,B}$$

$$e = \frac{F}{h_o}$$ FROM FIG. 2-7.2

$h = 0$

$h_o = \sqrt{B\,g_o} = \sqrt{(71)(0.375)}$

$h_o = 5.15$

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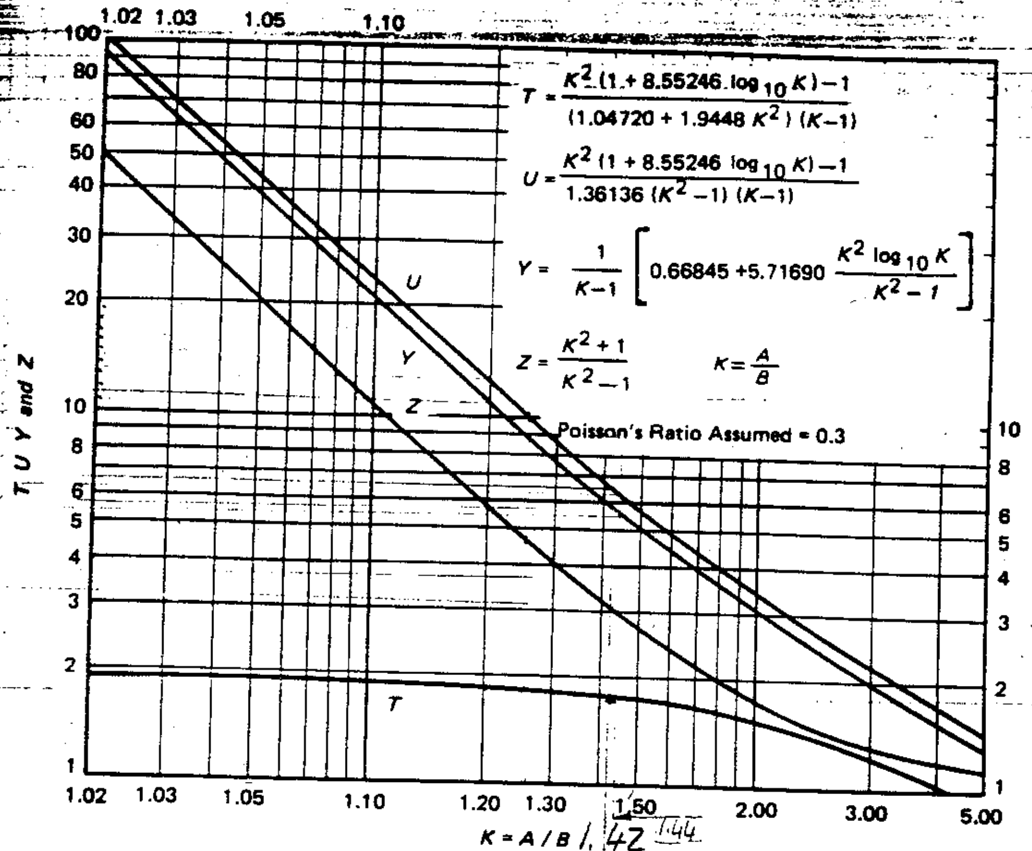


FIG. 2-7.1 VALUES OF T, U, Y, AND Z
(Terms Involving K)

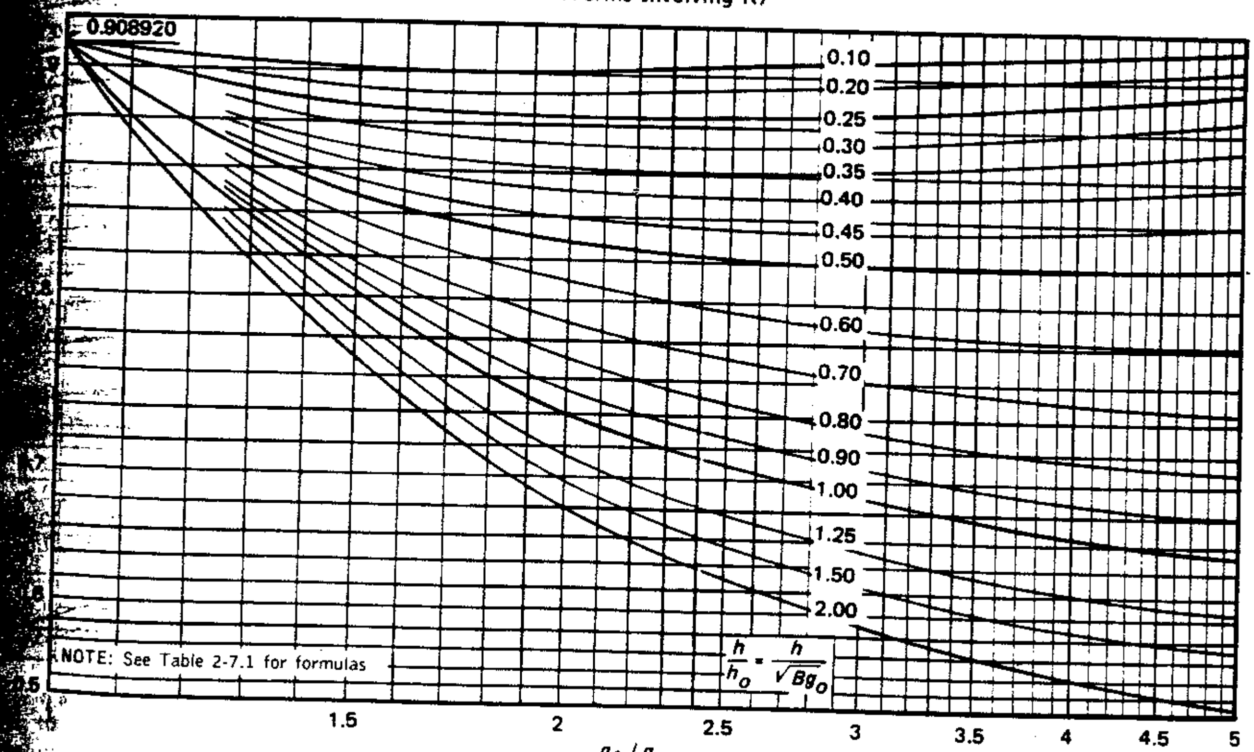


FIG. 2-7.2 VALUES OF F
(Integral Flange Factors)

$$\frac{h}{h_o} = \frac{0.}{5.15} = 0.0$$

$$\frac{g_1}{g_o} = \frac{0.75}{0.375} = 2$$

F FROM FIG 2-7.2

$$F \approx 0.918$$

L - factor

$$L = \frac{te+1}{T} + \frac{t^3}{d}$$

$t = 2.0"$

T FROM 2.7.1

$T = 1.75$

d - factor

$$d = \frac{U}{V} h_o g_o^2$$

$U = 6.8$

$V = 0.48$ FIG. 2.7.3

$$d = \frac{6.8}{0.48}(5.15)(0.375)^2$$

$$d = 10.25$$

WINDOW 26

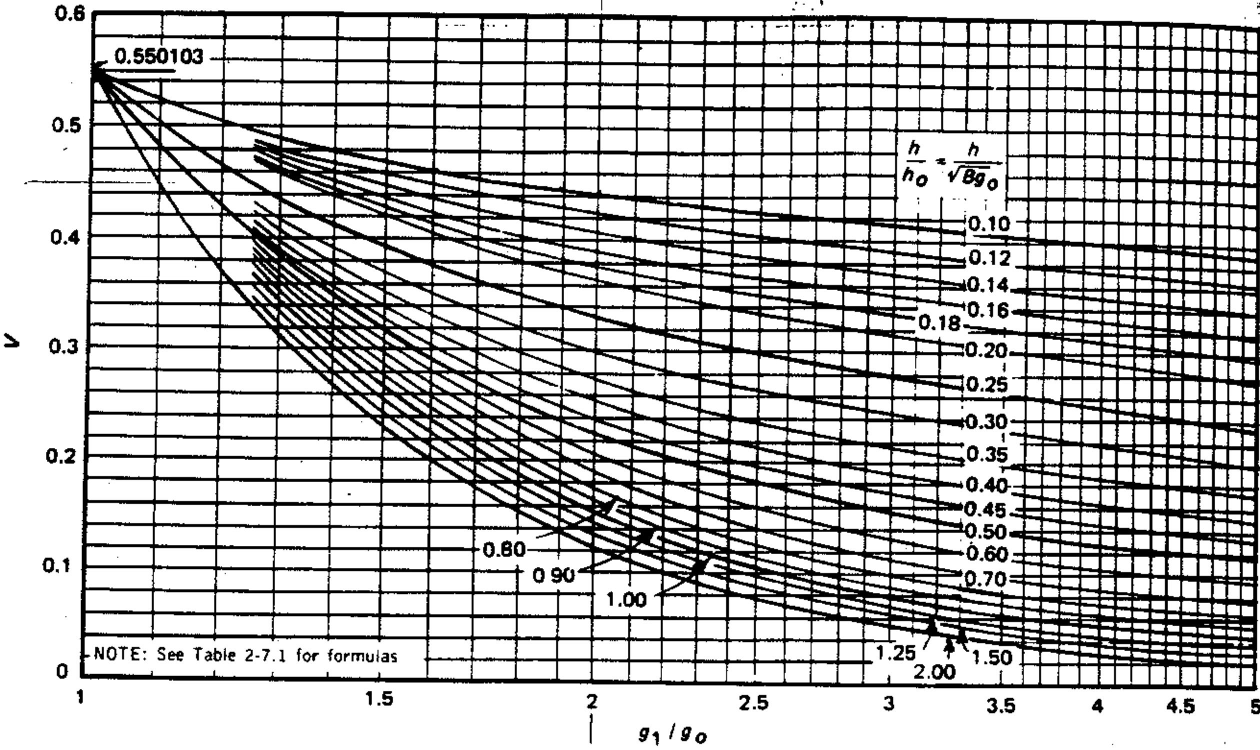


FIG. 2-7.3 VALUES OF V
(Integral Flange Factors)

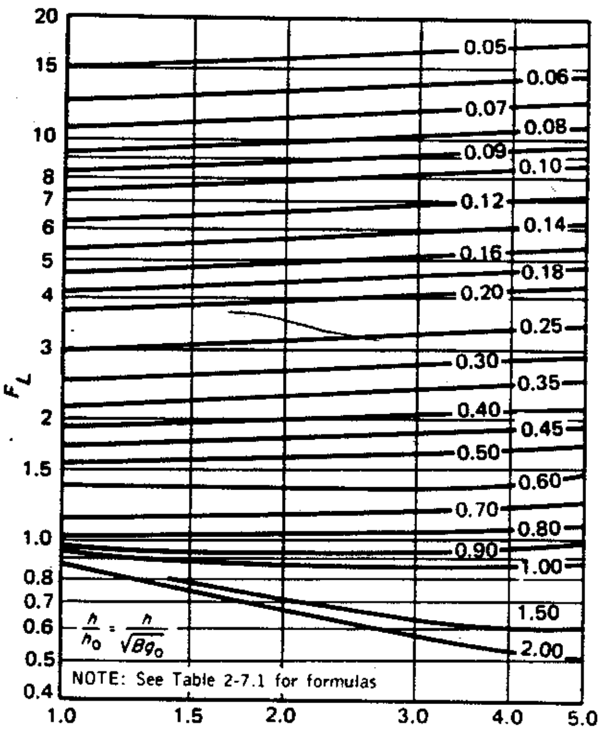


FIG. 2-7.4 VALUES OF F_L
(Loose Hub Flange Factors)

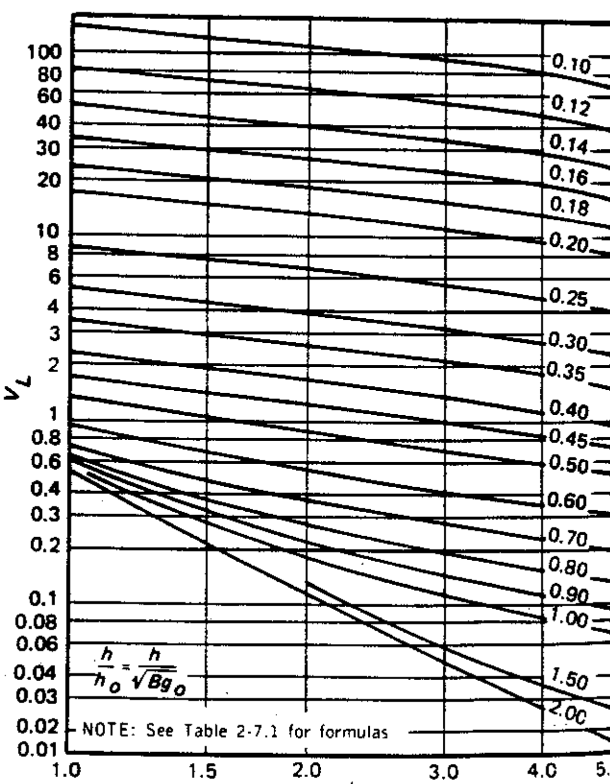


FIG. 2-7.5 VALUES OF V_L
(Loose Hub Flange Factors)

$$e = \frac{F}{h_o} = \frac{0.98}{5.15}$$

$$e = \underline{0.19}$$

L - factor

$$L = \frac{(2.0)(0.19)+1}{1.75} + \frac{(2.0)^3}{10.25}$$

$$L = 0.78 + 0.78$$

$$L = \underline{1.56}$$

$$S_R = \frac{(1.33\,t\,e+1)\ M_0}{L\ t^2\ B}$$

$$S_R = \frac{(1.33 \times 2.0 \times 0.19 + 1)\ 251{,}875.1}{1.56\ (2.0)^2\ (71)}$$

$$S_R = \underline{855.8} \text{ psi}$$

$$S_T = \frac{Y\ M_o}{t^2\ B} - Z\ S_R$$

$$S_T = \frac{(5.8)\ 251\,875.1}{(2.0)^2\ 71} - 3.1\,(855.8)$$

$$S_T = 5143.92 - 3.1\,(855.8)$$

$$S_T = \underline{2491.} \text{ psi}$$

EQ CALCULATION (FOR AN OPENING WITHOUT A NOZZLE)

$$EQ = \left(\frac{D_n}{t}\right) S_T$$

$$EQ = \left(\frac{71}{2.0}\right) 2491$$

$$EQ = 88,423.6$$

CALCULATE $\frac{EQ}{M_o}$:

$$\frac{EQ}{M_o} = \frac{88423.6}{251875.1}$$

$$\frac{EQ}{M_o} = 0.35 \qquad 0.352$$

CALCULATE M_H :

$$M_H = \frac{EQ}{\frac{1.74\, h_o V}{g_o^3\, B_1} + \frac{EQ}{M_o}\left(1 + \frac{Ft}{h_o}\right)}$$

$$= \frac{88423.6}{\frac{1.74 \times 5.15\,(0.48)}{(0.375)^3\; 94.5} + 0.35\left(1 + \frac{0.918\,(2.0)}{5.15}\right)}$$

$$M_H = \frac{88423.6}{1.33}$$

$$M_H = 66,487.37 \quad [\text{in lb}]$$

CALCULATE X_1

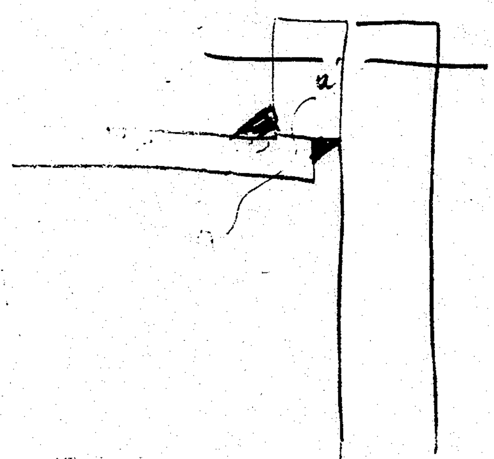
$$X_1 = \frac{M_o - M_H\left(1 + \frac{Ft}{h_o}\right)}{M_o}$$

$$X_1 = \frac{251,875.1 - 66087.87\left(1 + \frac{0.918(2.0)}{5.15}\right)}{251875.1}$$

X_1 = 0.64

LONGITUDINAL HUB STRESS IN SHELL

$$S_{HS} = (X_1)(EQ)\frac{1.10\, h_o\, f}{\left(\frac{g_1}{g_o}\right)^2 B_s V}$$



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

$\frac{h}{h_o} = 0$ $f = 4$

FIG. 2.7.6

$$S_{HS} = (0.64)(88423.6)\frac{(1.10)(5.15)(4)}{(2)^2\ 93.750(0.48)}$$

S_{HS} = 7124.79 psi

E-832 1.8 M VACUUM WINDOW 30.

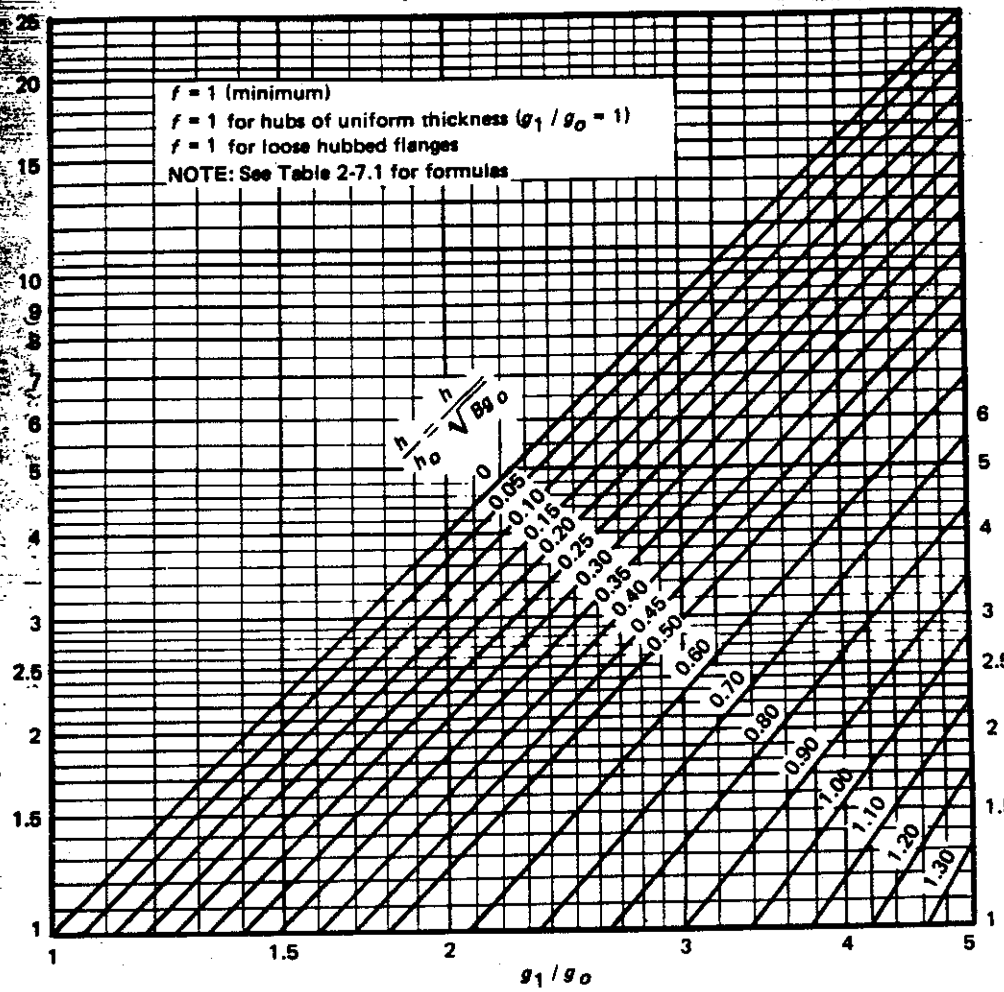


FIG. 2-7.6 VALUES OF *f*
(Hub Stress Correction Factor)

...eater of the values for A_{m1} and A_{m2} where $A_{m1} =$ ...S_b and $A_{m2} = W_{m2}/S_a$. A selection of bolts to be ...hall be made such that the actual total cross-...onal area of bolts A_b will not be less than A_m.

...nge Design Bolt Load W. The bolt loads used ...design of the flange shall be the values obtained ...Formulas (3) and (4). For operating conditions

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

...sket seating

P=0 REF. UG-34 P-33

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

...sed in Formula (4) shall be not less than that ...lated in Subsection C. In addition to the minimum ...ements for safety, Formula (4) provides a margin against abuse of the flange from overbolting. Since 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_bS_a, the flange may be designed on the basis of this latter quantity.

2-6 FLANGE MOMENTS

In the calculation of flange stresses, 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

RADIAL STRESS AT OUTSIDE DIA. :

$$S_{RS} = \frac{1.91 \times M_H \left(1 + \frac{F \cdot t}{h_o}\right)}{B_s t^2} + \frac{0.64\, F\, M_H}{B_s h_o t}$$

$$S_{RS} = \frac{1.91 \times 66087.87 \left(1 + \frac{0.918(2.0)}{5.15}\right)}{93.75(2.0)^2} + \frac{0.64 \times 0.918 \cdot 66087.87}{93.75(5.15) \cdot 2.0}$$

$$S_{RS} = 456.6 + 40.2$$

$$S_{RS} = \underline{496.8} \quad psi$$

TANGENTIAL STRESS AT OUTSIDE DIAMETER

$$S_{TS} = \frac{(X_1)(EQ)t}{B_s} - \frac{0.57\left(1 + \frac{Ft}{h_o}\right)M_H}{B_s t^2} + \frac{0.64\, F\, Z\, M_H}{B_s h_o t}$$

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

$$Z = \frac{1.42^2 + 1}{1.42^2 - 1}$$

$$Z = \underline{2.96}$$

$$S_{TS} = \frac{0.64(88423.6)(2.0)}{93.75} - \frac{0.57\left(1+\frac{0.98(2.0)}{5.15}\right)660878}{93.75(0.5)^2}$$

$$+\frac{0.64(0.918)(2.96)\,66087.87}{93.75(5.15)(2.0)}$$

$$S_{TS} = 1207.37 - 218?.?? + 119.0$$

$$S_{TS} = \underline{-853.98} \text{ psi}$$

- OPENING HEAD JUNCTURE

LONGITUDINAL HUB STRESS IN CENTRAL OPENING

$$S_{HO} = X_1 S_H$$

$$S_{HO} = 0.64(S_H)$$

$$S_H = \frac{f M_o}{L g_1^2 B}$$

$$S_H = \frac{(4)(251875.1)}{(1.56)(0.75)^2(71)}$$

$$S_H = \underline{\underline{16171.1}} \text{ psi}$$

$$S_{HO} = 0.64\,(16{,}171.1)$$

$$S_{HO} = \underline{10{,}349.5} \text{ psi}$$

RADIAL STRESS AT CENTRAL OPENING

$$S_{RO} = X_1 S_R$$

$$S_{RO} = (0.64)(855.8)$$

$$S_{RO} = \underline{547.7} \text{ psi}$$

TANGENTIAL STRESS AT DIAMETER OF CENTRAL OPENING

$$S_{TO} = X_1 S_T + \frac{0.64\, F Z_1 M_H}{B_s\, h_o\, t}$$

$$Z_1 = \frac{2K^2}{K^2 - 1}$$

$$Z_1 = \frac{2(1.42)^2}{(1.42)^2 - 1}$$

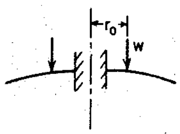
$$Z_1 = \underline{3.96}$$

$$S_{TO} = (0.64)(2491.) + \frac{0.64(0.918)(3.96)\ 66{,}087.7}{93.75(5.15)\ 2.0}$$

$$S_{TO} = 1594.2 + 159.2$$

$$S_{TO} = 1753.43 \text{ psi}$$

6. DETERMINING THE WALL THICKNESS OF FLANGE "B" (FIG. A3) (FIG. A4)

11. Outer edge free, inner edge fixed 	$y_b = 0 \quad \theta_b = 0 \quad M_{ra} = 0 \quad Q_a = 0$ $M_{rb} = \frac{-wa}{C_8}\left(\frac{r_oC_9}{b} - L_9\right)$ $Q_b = \frac{wr_o}{b}$ $y_a = \frac{-wa^3}{D}\left[\frac{C_2}{C_8}\left(\frac{r_oC_9}{b} - L_9\right) - \frac{r_oC_3}{b} + L_3\right]$ $\theta_a = \frac{-wa^2}{D}\left[\frac{C_5}{C_8}\left(\frac{r_oC_9}{b} - L_9\right) - \frac{r_oC_6}{b} + L_6\right]$	If $r_o = a$ (load at outer edge). Max $y = y_a = \frac{-wa^4}{bD}\left(\frac{C_2C_9}{C_8} - C_3\right)$ Max $M = M_{rb} = \frac{-wa^2}{b}\frac{C_9}{C_8}$ (For numerical values see case 1b after computing the loading at the inner edge)

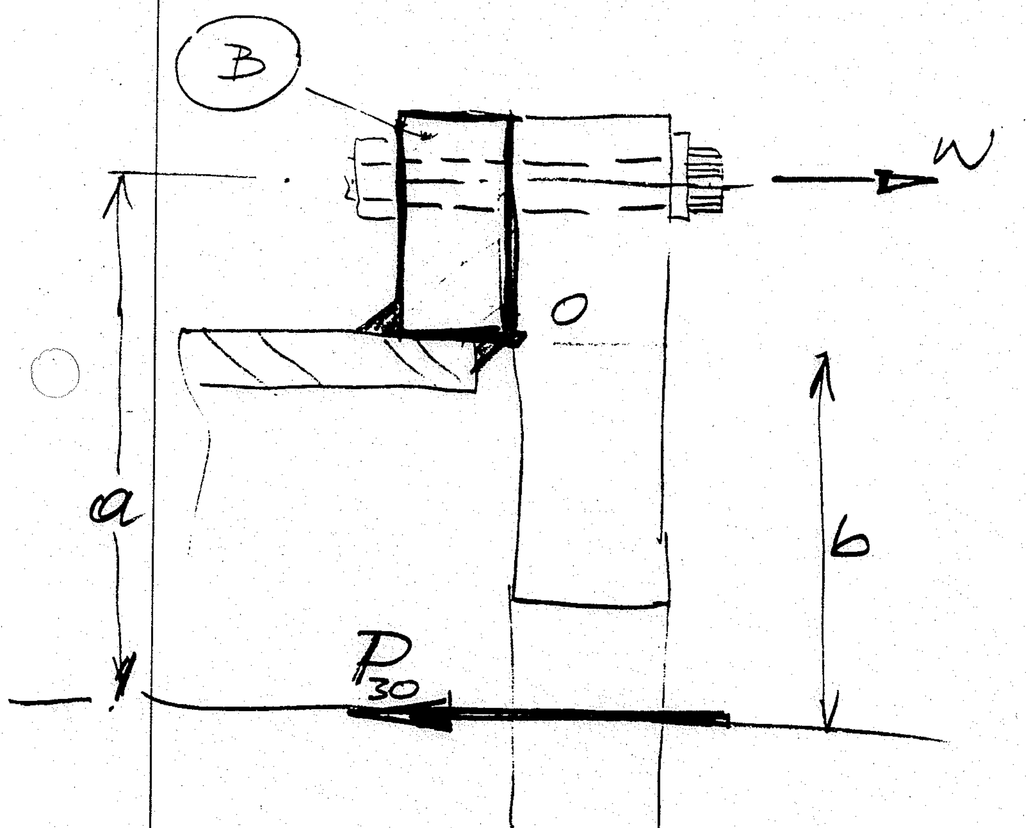


FIG. A4

$a = 49.125''$

$b = 47.250''$

$$\text{Max } M = M_{rb} = \frac{-wa^2}{b}\frac{C_9}{C_8}$$

$$C_8 = \frac{1}{2}\left[1 + \nu + (1-\nu)\left(\frac{b}{a}\right)^2\right]$$

$$C_9 = \frac{b}{a}\left\{ \frac{1+\nu}{2} \ln\frac{a}{b} + \frac{1-\nu}{4}\left[1-\left(\frac{b}{a}\right)^2\right]\right\}$$

$$C_8 = \frac{1}{2}\left[1 + 0.3 + (1-0.3)\left(\frac{47.25}{49.125}\right)^2\right]$$

$$C_8 = \underline{0.9737}$$

$$C_9 = \frac{47.25}{49.125}\left\{\frac{1+0.3}{2} \ln\frac{49.125}{47.25} + \frac{1-0.3}{4}\left[1-\left(\frac{47.25}{49.125}\right)^2\right]\right\}$$

$$C_9 = 0.9618\left\{0.025295 + 0.175\left[0.0748790\right]\right\}$$

$$C_9 = \underline{0.03693}$$

TABLE 24 ***Formulas for flat circular plates of constant thickness***

NOTATION: W = total applied load (pounds); w = unit line load (pounds per inch of circumference); q = load per unit area (pounds per square inch); M_o = unit applied line moment loading (inch-pounds per inch of circumference); θ_o = externally applied change in angular displacement (radians); y_o = externally applied step in the normal displacement (inches); y = vertical deflection of plate (inches); θ = slope of plate measured from the horizontal (radians); M_r = unit radial bending moment (inch-pounds per inch of circumference); M_t = unit tangential bending moment (inch-pounds per inch of radius); Q = unit shear force (pounds per inch of circumference); E = modulus of elasticity (pounds per square inch); ν = Poisson's ratio; γ = temperature coefficient of expansion (inches per inch per degree); a = outer radius (inches); b = inner radius for annular plate (inches); t = plate thickness (inches); r = radial location of quantity being evaluated (inches); r_o = radial location of unit line loading or the start of a distributed load (inches). F_1 to F_9 and G_1 to G_{19} are the several functions of the radial location r. C_1 to C_9 are plate constants dependent upon the ratio a/b. L_1 to L_{19} are loading constants dependent upon the ratio a/r_o. When used as subscripts, r and t refer to radial and tangential directions, respectively. When used as subscripts, a and b refer to an evaluation of the quantity subscripted at the outer edge and inner edge, respectively. When used as a subscript, c refers to the subscripted quantity evaluated at the center of the plate

Positive signs are associated with the several quantities in the following manner: Deflections y and y_o are positive upward; slopes θ and θ_o are positive when the deflection y increases positively as r increases; moments M_r, M_t, and M_o are positive when creating compression on the top surface; and the shear force Q is positive when acting upward on the inner edge of a given annular section

Bending stresses can be found from the moments M_r and M_t by the expression $\sigma = 6M/t^2$. The plate constant $D = Et^3/12(1-\nu^2)$. The singularity function brackets $\langle\ \rangle$ indicate that the expression contained within the brackets must be equated to zero unless $r > r_o$, after which they are treated as any other brackets.

General Plate Functions and Constants for Solid and Annular Circular Plates

$$F_1 = \frac{1+\nu}{2}\frac{b}{r}\ln\frac{r}{b} + \frac{1-\nu}{4}\left(\frac{r}{b} - \frac{b}{r}\right) \qquad C_1 = \frac{1+\nu}{2}\frac{b}{a}\ln\frac{a}{b} + \frac{1-\nu}{4}\left(\frac{a}{b} - \frac{b}{a}\right)$$

$$F_2 = \frac{1}{4}\left[1 - \left(\frac{b}{r}\right)^2\left(1 + 2\ln\frac{r}{b}\right)\right] \qquad C_2 = \frac{1}{4}\left[1 - \left(\frac{b}{a}\right)^2\left(1 + 2\ln\frac{a}{b}\right)\right]$$

$$F_3 = \frac{b}{4r}\left\{\left[\left(\frac{b}{r}\right)^2 + 1\right]\ln\frac{r}{b} + \left(\frac{b}{r}\right)^2 - 1\right\} \qquad C_3 = \frac{b}{4a}\left\{\left[\left(\frac{b}{a}\right)^2 + 1\right]\ln\frac{a}{b} + \left(\frac{b}{a}\right)^2 - 1\right\}$$

$$F_4 = \frac{1}{2}\left[(1+\nu)\frac{b}{r} + (1-\nu)\frac{r}{b}\right] \qquad C_4 = \frac{1}{2}\left[(1+\nu)\frac{b}{a} + (1-\nu)\frac{a}{b}\right]$$

$$F_5 = \frac{1}{2}\left[1 - \left(\frac{b}{r}\right)^2\right] \qquad C_5 = \frac{1}{2}\left[1 - \left(\frac{b}{a}\right)^2\right]$$

$$F_6 = \frac{b}{4r}\left[\left(\frac{b}{r}\right)^2 - 1 + 2\ln\frac{r}{b}\right] \qquad C_6 = \frac{b}{4a}\left[\left(\frac{b}{a}\right)^2 - 1 + 2\ln\frac{a}{b}\right]$$

$$F_7 = \frac{1}{2}(1-\nu^2)\left(\frac{r}{b} - \frac{b}{r}\right) \qquad C_7 = \frac{1}{2}(1-\nu^2)\left(\frac{a}{b} - \frac{b}{a}\right)$$

$$F_8 = \frac{1}{2}\left[1 + \nu + (1-\nu)\left(\frac{b}{r}\right)^2\right] \qquad C_8 = \frac{1}{2}\left[1 + \nu + (1-\nu)\left(\frac{b}{a}\right)^2\right]$$

$$F_9 = \frac{b}{r}\left\{\frac{1+\nu}{2}\ln\frac{r}{b} + \frac{1-\nu}{4}\left[1 - \left(\frac{b}{r}\right)^2\right]\right\} \qquad C_9 = \frac{b}{a}\left\{\frac{1+\nu}{2}\ln\frac{a}{b} + \frac{1-\nu}{4}\left[1 - \left(\frac{b}{a}\right)^2\right]\right\}$$

$$L_1 = \frac{1+\nu}{2}\frac{r_o}{a}\ln\frac{a}{r_o} + \frac{1-\nu}{4}\left(\frac{a}{r_o} - \frac{r_o}{a}\right) \qquad G_1 = \left[\frac{1+\nu}{2}\frac{r_o}{r}\ln\frac{r}{r_o} + \frac{1-\nu}{4}\left(\frac{r}{r_o} - \frac{r_o}{r}\right)\right]\langle r - r_o\rangle^0$$

$$L_2 = \frac{1}{4}\left[1 - \left(\frac{r_o}{a}\right)^2\left(1 + 2\ln\frac{a}{r_o}\right)\right] \qquad G_2 = \frac{1}{4}\left[1 - \left(\frac{r_o}{r}\right)^2\left(1 + 2\ln\frac{r}{r_o}\right)\right]\langle r - r_o\rangle^0$$

$$L_3 = \frac{r_o}{4a}\left\{\left[\left(\frac{r_o}{a}\right)^2 + 1\right]\ln\frac{a}{r_o} + \left(\frac{r_o}{a}\right)^2 - 1\right\} \qquad G_3 = \frac{r_o}{4r}\left\{\left[\left(\frac{r_o}{r}\right)^2 + 1\right]\ln\frac{r}{r_o} + \left(\frac{r_o}{r}\right)^2 - 1\right\}\langle r - r_o\rangle^0$$

$$L_4 = \frac{1}{2}\left[(1+\nu)\frac{r_o}{a} + (1-\nu)\frac{a}{r_o}\right] \qquad G_4 = \frac{1}{2}\left[(1+\nu)\frac{r_o}{r} + (1-\nu)\frac{r}{r_o}\right]\langle r - r_o\rangle^0$$

$$L_5 = \frac{1}{2}\left[1 - \left(\frac{r_o}{a}\right)^2\right] \qquad G_5 = \frac{1}{2}\left[1 - \left(\frac{r_o}{r}\right)^2\right]\langle r - r_o\rangle^0$$

$$L_6 = \frac{r_o}{4a}\left[\left(\frac{r_o}{a}\right)^2 - 1 + 2\ln\frac{a}{r_o}\right] \qquad G_6 = \frac{r_o}{4r}\left[\left(\frac{r_o}{r}\right)^2 - 1 + 2\ln\frac{r}{r_o}\right]\langle r - r_o\rangle^0$$

$$L_7 = \frac{1}{2}(1-\nu^2)\left(\frac{a}{r_o} - \frac{r_o}{a}\right) \qquad G_7 = \frac{1}{2}(1-\nu^2)\left(\frac{r}{r_o} - \frac{r_o}{r}\right)\langle r - r_o\rangle^0$$

$$L_8 = \frac{1}{2}\left[1 + \nu + (1-\nu)\left(\frac{r_o}{a}\right)^2\right] \qquad G_8 = \frac{1}{2}\left[1 + \nu + (1-\nu)\left(\frac{r_o}{r}\right)^2\right]\langle r - r_o\rangle^0$$

$$L_9 = \frac{r_o}{a}\left\{\frac{1+\nu}{2}\ln\frac{a}{r_o} + \frac{1-\nu}{4}\left[1 - \left(\frac{r_o}{a}\right)^2\right]\right\} \qquad G_9 = \frac{r_o}{r}\left\{\frac{1+\nu}{2}\ln\frac{r}{r_o} + \frac{1-\nu}{4}\left[1 - \left(\frac{r_o}{r}\right)^2\right]\right\}\langle r - r_o\rangle^0$$

$$L_{11} = \frac{1}{64}\left\{1 + 4\left(\frac{r_o}{a}\right)^2 - 5\left(\frac{r_o}{a}\right)^4 - 4\left(\frac{r_o}{a}\right)^2\left[2 + \left(\frac{r_o}{a}\right)^2\right]\ln\frac{a}{r_o}\right\}$$

$$G_{11} = \frac{1}{64}\left\{1 + 4\left(\frac{r_o}{r}\right)^2 - 5\left(\frac{r_o}{r}\right)^4 - 4\left(\frac{r_o}{r}\right)^2\left[2 + \left(\frac{r_o}{r}\right)^2\right]\ln\frac{r}{r_o}\right\}\langle r - r_o\rangle^0$$

$$L_{12} = \frac{a}{14{,}400(a - r_o)}\left\{64 - 225\frac{r_o}{a} - 100\left(\frac{r_o}{a}\right)^3 + 261\left(\frac{r_o}{a}\right)^5 + 60\left(\frac{r_o}{a}\right)^3\left[3\left(\frac{r_o}{a}\right)^2 + 10\right]\ln\frac{a}{r_o}\right\}$$

$$G_{12} = \frac{r\langle r - r_o\rangle^0}{14{,}400(r - r_o)}\left\{64 - 225\frac{r_o}{r} - 100\left(\frac{r_o}{r}\right)^3 + 261\left(\frac{r_o}{r}\right)^5 + 60\left(\frac{r_o}{r}\right)^3\left[3\left(\frac{r_o}{r}\right)^2 + 10\right]\ln\frac{r}{r_o}\right\}$$

$$L_{13} = \frac{a^2}{14{,}400(a - r_o)^2}\left\{25 - 128\frac{r_o}{a} + 225\left(\frac{r_o}{a}\right)^2 - 25\left(\frac{r_o}{a}\right)^4 - 97\left(\frac{r_o}{a}\right)^6 - 60\left(\frac{r_o}{a}\right)^4\left[5 + \left(\frac{r_o}{a}\right)^2\right]\ln\frac{a}{r_o}\right\}$$

$$G_{13} = \frac{r^2\langle r - r_o\rangle^0}{14{,}400(r - r_o)^2}\left\{25 - 128\frac{r_o}{r} + 225\left(\frac{r_o}{r}\right)^2 - 25\left(\frac{r_o}{r}\right)^4 - 97\left(\frac{r_o}{r}\right)^6 - 60\left(\frac{r_o}{r}\right)^4\left[5 + \left(\frac{r_o}{r}\right)^2\right]\ln\frac{r}{r_o}\right\}$$

$$L_{14} = \frac{1}{16}\left[1 - \left(\frac{r_o}{a}\right)^4 - 4\left(\frac{r_o}{a}\right)^2\ln\frac{a}{r_o}\right] \qquad G_{14} = \frac{1}{16}\left[1 - \left(\frac{r_o}{r}\right)^4 - 4\left(\frac{r_o}{r}\right)^2\ln\frac{r}{r_o}\right]\langle r - r_o\rangle^0$$

$$L_{15} = \frac{a}{720(a - r_o)}\left[16 - 45\frac{r_o}{a} + 9\left(\frac{r_o}{a}\right)^5 + 20\left(\frac{r_o}{a}\right)^3\left(1 + 3\ln\frac{a}{r_o}\right)\right]$$

$$G_{15} = \frac{r\langle r - r_o\rangle^0}{720(r - r_o)}\left[16 - 45\frac{r_o}{r} + 9\left(\frac{r_o}{r}\right)^5 + 20\left(\frac{r_o}{r}\right)^3\left(1 + 3\ln\frac{r}{r_o}\right)\right]$$

$$L_{16} = \frac{a^2}{1440(a - r_o)^2}\left[15 - 64\frac{r_o}{a} + 90\left(\frac{r_o}{a}\right)^2 - 6\left(\frac{r_o}{a}\right)^6 - 5\left(\frac{r_o}{a}\right)^4\left(7 + 12\ln\frac{a}{r_o}\right)\right]$$

$$G_{16} = \frac{r^2\langle r - r_o\rangle^0}{1440(r - r_o)^2}\left[15 - 64\frac{r_o}{r} + 90\left(\frac{r_o}{r}\right)^2 - 6\left(\frac{r_o}{r}\right)^6 - 5\left(\frac{r_o}{r}\right)^4\left(7 + 12\ln\frac{r}{r_o}\right)\right]$$

$$L_{17} = \frac{1}{4}\left\{1 - \frac{1-\nu}{4}\left[1 - \left(\frac{r_o}{a}\right)^4\right] - \left(\frac{r_o}{a}\right)^2\left[1 + (1+\nu)\ln\frac{a}{r_o}\right]\right\}$$

$$G_{17} = \frac{1}{4}\left\{1 - \frac{1-\nu}{4}\left[1 - \left(\frac{r_o}{r}\right)^4\right] - \left(\frac{r_o}{r}\right)^2\left[1 + (1+\nu)\ln\frac{r}{r_o}\right]\right\}\langle r - r_o\rangle^0$$

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W – UNIT LINE LOAD
(POUNDS PER INCH OF CIRCUMFERENCE)

$$\omega = \frac{P_{30}}{\pi(98.25)}$$

$$\omega = \frac{207\,087}{\pi(98.25)}$$

$$\underline{\omega = 670.91} \quad [\text{lbs/in}]$$

$$M = \frac{-670.91\,(49.125)^2}{47.25}\;\frac{0.036932}{0.9737}$$

$$M = -1299.7 \quad \text{lbin}$$

$$\sigma = \frac{6M}{t^2} \qquad \text{assuming } \sigma = 8000 \text{ psi}$$

$$t = \sqrt{\frac{6M}{\sigma}}$$

$$t = \underline{0.98''}$$

$t = 1\frac{1}{2}''$ <u>IS SELECTED</u>.

7. DETERMINING THE NUMBER AND SIZE OF BOLTS IN A & B FLANGE CONNECTION

COMPUTATION ACCORDING TO ASME SECTION VIII DIV. I B&PV CODE.

- THE MINIMUM INITIAL BOLT LOAD FOR GASKET SEATING CONDITIONS.

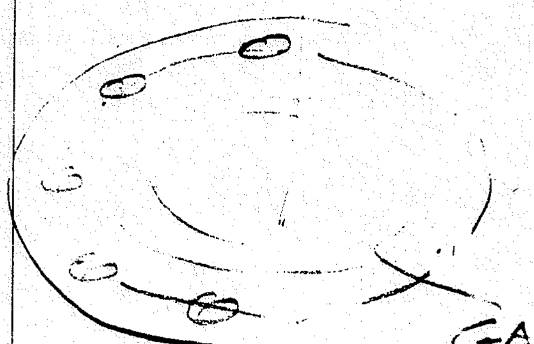
$$W_{m2} = 3.14\, b\, G\, y$$

$$G = 95\tfrac{7}{8}''$$

$$G = 95.875''$$

$$b = 0.310''$$

$$y = 200 \text{ psi}$$



$$W_{m2} = 3.14\,(0.310)\,95.875\,(200)$$

$$W_{m2} = \underline{18{,}665} \quad [\text{lbs}]$$

- THE REQUIRED BOLT LOAD FOR THE OPERATING CONDITIONS.

$$W_{m_1} = 0.785\, G^2 P + 2b \times 3.14\, G\, m P$$

W_{m_1} (FROM PAGE 22.)

$$W_{m_1} = \underline{212{,}105.35} \quad [\text{lbs}]$$

THE TOTAL CROSS-SECTIONAL AREA OF BOLTS

A_m

A_m – GREATEST OF A_{m_1} OR A_{m_2}

$$A_{m_1} = \frac{W_{m_1}}{S_b}$$

$$A_{m_2} = \frac{W_{m_2}}{S_a}$$

S_a, S_b } ALLOWABLE BOLT STRESS

SPEC. A 574

$$A_{m_1} = \frac{212\,105.3}{65\,000}$$

$$A_{m_1} = \underline{10.6}\ in^2$$

REQUIRED NUMBER OF BOLTS

$$n = \frac{A_{m1}}{A_i}$$

A_i CROSS SECTION AREA OF THE BOLT AT THREAD ROOT.

$$n = \frac{10.6}{0.302} = 35$$

$n = 60$ IS SELECTED

$\frac{3}{4}''$ - 10 UNRC FERRY CAP SCREWS ARE SELECTED

FERRY CAP [redacted]

7/16" (.4375") Diameter

Length in Inches	Coarse or Fine Thread Stocked	Package Quantity	Net Weight Per 1000 Pieces
⅝	C	100	47.9
¾	C	100	52.3
⅞	C	100	56.7
1	C-F	100	61.0
1¼	C-F	50	69.7
1½	C-F	50	79.8
1¾	C	50	91.3
2	C-F	50	102.0
2¼	C	50	113.0
2½	C	50	123.0
2¾	C	25	134.0
3	C	25	143.0
3¼	C	25	153.0
3½	C	25	163.0
4	C	25	182.0

1/2" (.500") Diameter

Length in Inches	Coarse or Fine Thread Stocked	Package Quantity	Net Weight Per 1000 Pieces
⅝	C	50	69.6
¾	C	50	75.4
⅞	C	50	81.2
1	C-F	50	87.0
1¼	C-F	50	98.6
1½	C-F	50	111.0
1¾	C-F	50	125.0
2	C-F	50	139.0
2¼	C-F	25	153.0
2½	C-F	25	166.0
2¾	C	25	180.0
3	C-F	25	194.0
3¼	C	25	207.0
3½	C	25	220.0
4	C	25	245.0
4½	C	25	271.0
5	C	25	296.0
5½	C	25	322.0
6	C	25	348.0
6½	C	10	—
7	C	10	—

5/8" (.625") Diameter

Length in Inches	Coarse or Fine Thread Stocked	Package Quantity	Net Weight Per 1000 Pieces
1	C-F	25	147
1¼	C-F	25	165
1½	C-F	25	183
1¾	C-F	25	203
2	C-F	25	224
2¼	C	25	245
2½	C-F	25	266
2¾	C	25	287
3	C	25	309
3¼	C	25	330
3½	C	25	351
4	C	25	395
4½	C	25	435
5	C	25	475
5½	C	10	515
6	C	10	555
6½	C	10	—
7	C	10	—

3/4" (.750") Diameter

Length in Inches	Coarse or Fine Thread Stocked	Package Quantity	Net Weight Per 1000 Pieces
1¼	C	25	252
1½	C	25	279
1¾	C	25	306
2	C	25	335
2¼	C	25	366
2½	C	25	397
2¾	C	25	428
3	C	25	459
3¼	C	25	490
3½	C	25	521
4	C	25	533
4½	C	25	541
5	C	25	700
5½	C	10	758
6	C	10	817
6½	C	10	—
7	C	10	—

7/8" (.875") Diameter

Length in Inches	Coarse or Fine Thread Stocked	Package Quantity	Net Weight Per 1000 Pieces
2	C	10	480
2¼	C	10	520
2½	C	10	562
2¾	C	10	604
3	C	10	646
3¼	C	10	688
3½	C	10	730
4	C	10	814
4½	C	10	898
5	C	10	979
5½	C	10	1059
6	C	10	1139
6½	C	10	—
7	C	10	—

1" (1.000") Diameter

Length in Inches	Coarse or Fine Thread Stocked	Package Quantity	Net Weight Per 1000 Pieces
2	C	10	641
2¼	C	10	691
2½	C	10	742
2¾	C	10	797
3	C	10	852
3¼	C	10	907
3½	C	10	962
4	C	10	1072
4½	C	10	1182
5	C	10	1292
5½	C	10	1400
6	C	10	1503
6½	C	10	—
7	C	10	—
8	C	10	—

NOTE: Selected sizes of Ferry Cap Countr-Bor® Screws in diameters from ½" through 1½" are available from stock to meet ASTM-A320/L7 and ASTM-A193/B7-B7M specifications.

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the thread. The length of the center portion shall be equal to the screw length minus two screw diameters (2D). The starting thread shall be chamfered and the junction between diameters A_e and D_e shall be rounded to a value equal to F maximum.

(8) **Fillet.** For all lengths of screws the form of the underhead fillet shall be optional, as depicted in Fig. 1, provided it is a smooth and continuous concave curve fairing into the bearing surface of the head and the screw shank within the envelope established by the limits for fillet extension, length, and juncture radius specified in Table 1A.

(9) **Length.** The length of the screw shall be measured parallel to the axis of screw from the plane of the bearing surface under the head to the plane of the flat of the point. The portion of the screw contained within dimension L is commonly called the shank. The basic length dimension on the product shall be the nominal length expressed as a two-place decimal.

(10) **Standard Lengths.** The standard length increments for socket head cap screws shall be as tabulated below:

LENGTH

Nominal Screw Size	Nominal Screw Length	Standard Length Increment
0 to 1 in., incl.	1/8 thru 1/4	1/16
	1/4 thru 1	1/8
	1 thru 3 1/2	1/4
	3 1/2 thru 7	1/2
	7 thru 10	1
Over 1 in.	1 thru 7	1/2
	7 thru 10	1
	Over 10	2

(11) **Length Tolerances.** The allowable tolerance on length shall be as tabulated below:

Nominal Screw Size	0 thru 3/8, incl.	7/16 thru 3/4, incl.	7/8 thru 1 1/2, incl.	Over 1 1/2
Nominal Screw Length	**Tolerance on Length**			
Up to 1 in., incl.	-0.03	-0.03	-0.05	...
Over 1 in. to 2 1/2, incl.	-0.04	-0.06	-0.10	-0.18
Over 2 1/2 to 6, incl.	-0.06	-0.08	-0.14	-0.20
Over 6	-0.12	-0.12	-0.20	-0.24

(12) **Threads.** Threads shall be Unified external threads with radius root. Class 3A UNRC and UNRF Series for screw sizes 0 (0.060 in.) through 1 in.; Class 2A UNRC and UNRF Series for sizes over 1 in. to 1-1/2 in., inclusive; and Class 2A UNRC Series for sizes larger than 1-1/2 in.

Acceptability shall be based upon System 22, ANSI B1.3.

Class 3A does not provide a plating allowance. When plated products are required it is recommended that they be procured from the manufacturer. (See 1.8.)

(13) **Thread Length, L_T.** The length of thread shall be measured, parallel to the axis of screw, from the extreme point to the last complete (full form) thread. The thread length on socket head cap screws shall be as defined by Table 1B and notes thereto.

(14) **Grip Gaging Length, L_G.** The grip gaging length is the distance, measured parallel to the axis of screw, from the bearing surface of the head to the first complete (full form) thread under the head. (See Table 1B.)

(15) **Body Length, L_B.** The body length is the length, measured parallel to the axis of screw, of the unthreaded portion of the shank. (See Table 1B.)

(16) **Screw Point Chamfer.** The point shall be flat or slightly concave and chamfered. The plane of the point shall be approximately normal to the axis of the screw. The chamfer shall extend slightly below the root of the thread and the edge between the flat and chamfer may be slightly rounded. The included angle of the point should be approximately 90 deg. Chamfering of screw sizes up to and including size 8 (0.164 in.) and lengths below .75d shall be optional.

(17) **Material**

(a) ***Steel, alloy.*** Cap screws shall be fabricated from an alloy steel and shall conform in all respects to ASTM Specification A574, Alloy Steel Socket Head Cap Screws.

(b) ***Steel, corrosion resistant.*** Cap screws shall be fabricated from austenitic corrosion resistant steel of types 18-8, AISI 384, or equivalent types, and shall have a minimum tensile strength of 80,000 psi for sizes up to and including 5/8 in. and 70,000 psi for sizes larger than 5/8 in.

(18) **Surface Roughness.** For alloy steel screws of sizes up to and including 5/8 in., and nominal lengths equal to or less than 8 times the basic screw diameter,

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TABLE UCS-23
MAXIMUM ALLOWABLE STRESS VALUES IN TENSION FOR CARBON AND LOW ALLOY STE[illegible]

(Caution: See UW-12 for vessels constructed under Part UW)

	Spec. No.	Grade	Nominal Composition	P-No.	Group No.	Notes	Specified Min. Yield ksi	[illegible]
	Bolting — All Carbon and Low Alloy Steels							
	SA-193	B5	5Cr–½Mo			(7)	[illegible]	[illegible]
		B7	1Cr–⅕Mo ≤ 2½ in.	...	...	(7)	[illegible]	[illegible]
			1Cr–⅕Mo > 2½ in. and ≤ 4 in.	...	...	(7)	[illegible]	[illegible]
			1Cr–⅕Mo > 4 in.	...	...	(7)	[illegible]	[illegible]
		B7M	1Cr–⅕Mo ≤ 2½ in.	...	...	...	[illegible]	[illegible]
		B16	1Cr–½Mo–V ≤ 2½ in.	...	...	(7)	[illegible]	[illegible]
			1Cr–½Mo–V > 2½ in. and ≤ 4 in.	...	...	(7)	[illegible]	[illegible]
			1Cr–½Mo–V > 4 in.	...	...	(7)	[illegible]	[illegible]
	SA-307	B	C	...	...	(8)	[illegible]	[illegible]
S83 W83	SA-320	L43	1¾Ni–¾Cr–¼Mo	...	...	(7)(11)	[illegible]	[illegible]
		L7	1Cr–⅕Mo	...	...	(7)(32)	[illegible]	[illegible]
		L7A for 2½ in. dia. and under	1Cr–¼Mo	...	...	(7)(33)	[illegible]	[illegible]
		L7M	1Cr–⅕Mo ≤ 2½ in.	...	...	...	[illegible]	[illegible]
	SA-325	Types 1 and 2	C; ½ in.–1 in., incl.	...	...	(7)	[illegible]	120
		Types 1 and 2	C; 1⅛ in.–1½ in., incl.	...	...	(7)	[illegible]	105
S83	SA-354	BC	C	...	...	(7)	[illegible]	115
		BD	C	...	...	(7)	125	[illegible]
S84	SA-420	WPL-6	C–Mn–Si	1	1	(18)(38)	35	60
		WPL-9	2Ni–1Cu	9A	1	(10)(38)	46	[illegible]
		WPL-3	3½Ni	9B	1	(38)	35	65
	SA-449	1 in. & under	C	...	...	(7)	92	120
		> 1 in. and ≤ 1½ in.	C	...	...	(7)	81	105
		> 1½ in. and ≤ 3 in.	C	...	...	(7)	58	[illegible]
	SA-574	≤ ½ in.	...	...	...	(7)(34)(35)	...	180
		> ½ in.	...	...	...	(7)(34)(35)	[illegible]	[illegible]
S85	**Fittings — All Carbon and Low Alloy Steels**							
S84	SA-234	WP1	C–½Mo	3	1	(19)(38)	30	55
		WP12	1Cr–½Mo	4	1	(38)	30	60
		WP11	1¼Cr–½Mo–Si	4	1	(38)	30	60
		WP22	2¼Cr–Mo	5	1	(38)	30	60
S84 S85	SA-234	WP5	5Cr–½Mo	5	2	(10)(38)	30	60
		WP7	7Cr–½Mo	5	2	(10)(38)	30	60
		WPB	C–Si	1	1	(18)(38)	35	60
		WP9	9Cr–1Mo	5	2	(10)(38)	30	60
		WPC	C–Si	1	2	(18)(38)	40	70
		WPR	2Ni–1Cu	9A	1	(10)(38)	40	63

present in sufficient quantity to assure that the specified strength properties are met after oil quenching and tempering. As a guide for selecting material, an alloy steel should be capable of meeting the specified mechanical requirements if the "as oil quenched" core hardness one diameter from the point is equal to or exceeds 25 HRC + (55 × carbon content).

6.4 Application of heats of steel to which bismuth, selenium, tellurium, or lead has been intentionally added shall not be permitted.

6.5 Chemical analyses shall be performed in accordance with Methods A 751.

7. Mechanical Properties

7.1 The hardness of finished screws shall be 39 to 45 HRC for 0.500 in. and smaller and 37 to 45 HRC for 0.625 in. and larger. This shall be only the mechanical requirements for screws that are shorter than three times the diameter or that have insufficient threads for tension testing.

7.2 Screws, other than those exempted in 7.1 and 7.3, shall meet the proof load and tensile requirements in Tables 2 and 3. The screws shall be tension tested with a wedge of the angle specified in Table 5 under the head. To meet the requirements of the wedge test, there must be a tensile failure in the body or threaded section. For the purpose of this test, failure means separation into two pieces. Screws threaded to the head shall pass the requirements for this test if the fracture that caused failure originated in the thread area, even though it may have propagated into the fillet area or the head before separation.

7.3 Screws having a diameter larger than 1.250 in. shall be preferably tested in full size and shall meet the requirements of Tables 2 and 3. When equipment of sufficient capacity is not readily available, screws shall meet 170 ksi, min, tensile strength, 153 ksi, min, yield strength at 0.2 % offset, and 8 % elongation on specimens machined in accordance with Test Methods F 606.

8. Metallurgical Requirement

8.1 *Carburization or Decarburization:*

8.1.1 There shall be no evidence of carburization or total decarburization on the surfaces of the heat-treated screws when measured in accordance with 11.3.

8.1.2 The depth of partial decarburization shall be limited to the values in Table 4 when measured as shown in Fig. 1 and in accordance with 11.3.

9. Dimensions

9.1 Unless otherwise specified, the product shall conform to the requirements of ANSI B18.3.

9.2 Unless otherwise specified, threads shall be Unified standard: Class 3A, UNRC and UNRF series for screw sizes 060 through 1 in. inclusive; Class 2A, UNRC and UNRF series for sizes over 1 in. to 1.500 in. inclusive; and Class 2A UNRC series for sizes larger than 1.500 in. in accordance with ANSI B1.1.

10. Workmanship, Finish, and Appearance

10.1 *Discontinuities*—The surface discontinuities for these products shall conform to Specification F 788 and the additional limitations specified herein.

TABLE 2 Tensile Requirements for Coarse Thread Screws

Screw Dia (D), in.	Threads/in.	Tensile Load, min, lbf[A]	Stress Area, in.[2] [B]	Proof Load (Length Measurement Method), min, lbf[C]
0.073	64	473	0.00263	368
0.086	56	666	0.00370	518
0.099	48	877	0.00487	682
0.112	40	1 090	0.00604	846
0.125	40	1 430	0.00796	1 110
0.138	32	1 640	0.00909	1 270
0.164	32	2 520	0.0140	1 960
0.190	24	3 150	0.0175	2 450
0.250	20	5 730	0.0318	4 450
0.3125	18	9 440	0.0524	7 340
0.375	16	13 900	0.0775	10 800
0.4375	14	19 100	0.1063	14 900
0.500	13	25 500	0.1419	19 900
0.625	11	38 400	0.226	30 500
0.750	10	56 800	0.334	45 100
0.875	9	78 500	0.462	62 400
1.000	8	103 000	0.606	81 800
1.125	7	129 000	0.763	103 000
1.250	7	165 000	0.969	131 000
1.375	6	196 000	1.155	156 000
1.500	6	239 000	1.405	190 000
1.750	5	323 000	1.90	256 000
2.000	4½	425 000	2.50	338 000
2.250	4½	552 000	3.25	439 000
2.500	4	680 000	4.00	540 000
2.750	4	838 000	4.93	666 000
3.000	4	1 010 000	5.97	806 000
3.250	4	1 210 000	7.10	958 000
3.500	4	1 420 000	8.33	1 120 000
3.750	4	1 640 000	9.66	1 300 000
4.000	4	1 880 000	11.08	1 500 000

[A] Values based on 180 ksi for 0.500 and smaller and 170 ksi for 0.625 and larger and stress area in accordance with Footnote B.

[B] Stress areas based on Handbook H-28 (U. S. Department of Commerce) as follows:

$$A_s = 0.7854\,[D - (0.9743/n)]^2$$

where:

A_s = stress area,
D = nominal screw size, and
n = threads/in.

[C] Values based on 140 ksi for 0.500 and smaller and 135 ksi for 0.625 and larger and stress area in accordance with Footnote B.

10.2 *Socket Discontinuities:*

10.2.1 Depth of discontinuities in the socket area will be permissible within the limits of Condition 1 provided they do not affect the usability and performance of the screw. Discontinuities exceeding these limits are not acceptable.

10.2.2 Longitudinal discontinuities must not exceed $0.25T$ in length. Permissible and nonpermissible discontinuities are shown in Fig. 2.

NOTE 2—T = actual key engagement.

10.3 *Permissible Head and Body Discontinuities*—Discontinuities as defined above are permitted in the locations illustrated in Fig. 3 to the depths shown below. These discontinuities are permitted, provided they do not affect the usability and performance of the screw. All discontinuities are to be measured perpendicular to indicated surfaces.

10.4 *Conditions for Permissible Discontinuity Depths:*

8. DETERMINING THE REQUIRED NUMBER OF BOLTS FOR "C" — "D" FLANGE ASSEMBLY, "C" AND "D", FIG. A6

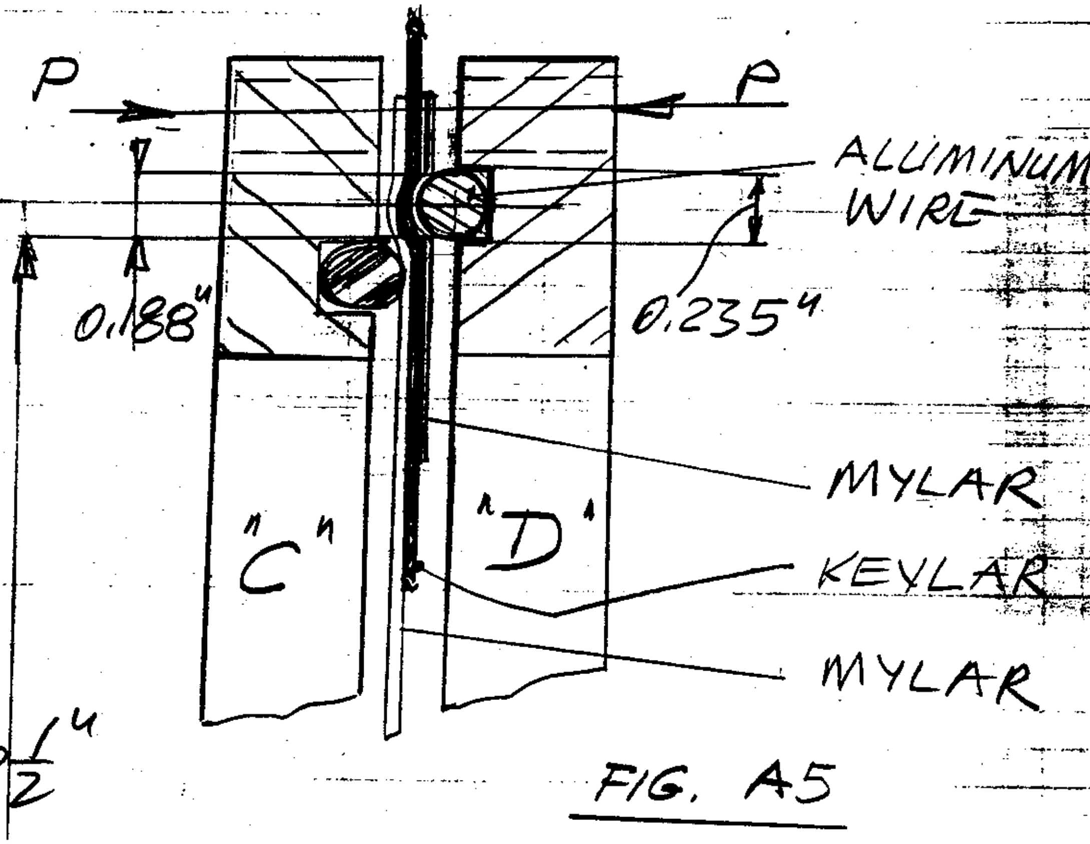


FIG. A5

THE ALUMINUM WIRE RING COMPRESSIVE STRESS DUE TO CLAMPING FORCE "P" SHOULD NOT EXCEED THE TENSILE STRENGTH FOR THAT MATERIAL.

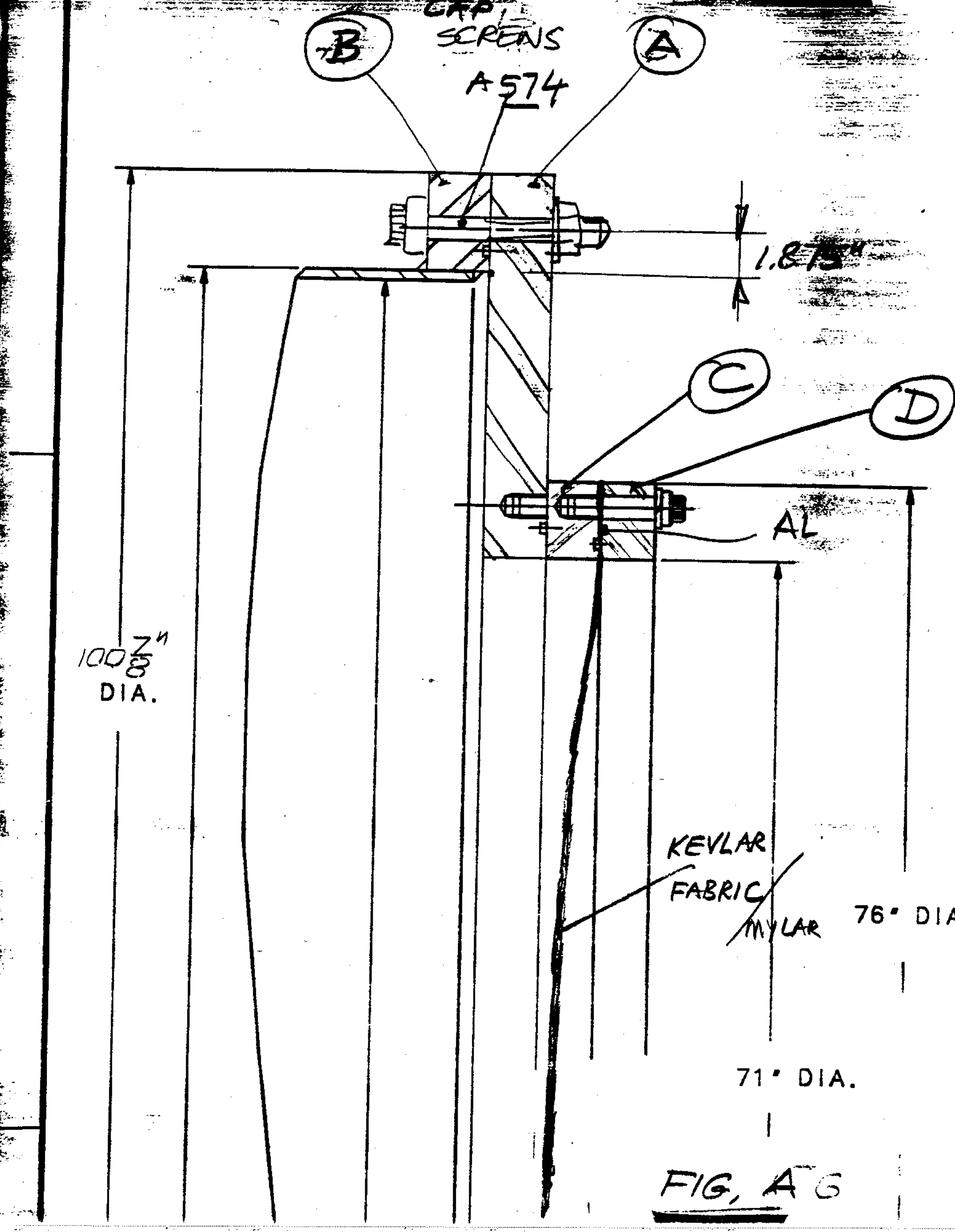
ALUMINUM TENSILE STRENGTH OF THE ALLOY 1100 (WIRE)

$$\sigma_T = 13000 \text{ psi}$$

ASSUMING $\sigma_c = 10\,000$ psi,

THE REQUIRED CLAMPING FORCE

$$P = A(\sigma)$$


CAP SCREWS
A574
B
A
1.875"
C
D
AL
100 7/8" DIA.
KEVLAR FABRIC/MYLAR
76" DIA
71" DIA.

FIG, A6

$$P = \pi \left(72.5 + \frac{0.188}{2} \right) 10000$$

$$\underline{P = 2{,}280{,}607.7} \quad [lbs]$$

$\frac{3}{4}$" FERRY CAP BOLTS WILL BE USED FOR THIS ASSEMBLY.

ASSUMING $\tilde{\sigma}_W$ - WORKING STRESS

FOR THE BOLTS :

$$\tilde{\sigma}_W = 65{,}000 \text{ psi}$$

THE SINGLE BOLT CLAMPING FORCE WILL BE :

$$P_i = \tilde{\sigma}_W \cdot A_{BOLT}$$

$$P_i = 65000 \, (0.302)$$

$$P_i = \underline{19630}$$

REQUIRED NUMBER OF BOLTS :

$$n = \frac{P}{P_i} = 116.1$$

$n = \underline{120}$ IS SELECTED.

9. DETERMINING THE REQUIRED TORQUE VALUE FOR SINGLE CLAMPING BOLT.

$$T = K\, d\, F_{PT} \qquad [\text{REF.}]$$

REF. : STANDARD HANDBOOK OF MACHINE DESIGN
J.E. SHIGLEY
CH. R. MISCHKE

WHERE :
K - NUT FACTOR (TABLE 23-5)
d - NOMINAL DIAMETER OF THE BOLT
F_{PT} - TARGET PRELOAD lbs (kN)

TABLE 23-5 Nut Factors

Lubricant or coating on the fastener	Source	Nut factor: Reported mean	Nut factor: Reported range
1. Cadmium plate	1	0.194–0.246	0.153–0.328
2. Zinc plate	5	0.332	0.262–0.398
3. Black oxide	1	0.163–0.194	0.109–0.279
4. Baked on PTFE	1	0.092–0.112	0.064–0.142
5. Molydisulfide paste	2	0.155	0.14–0.17
6. Machine oil	2	0.21	0.20–0.225
7. Carnaba wax (5% emulsion)	2	0.148	0.12–0.165
8. 60 Spindle oil	2	0.22	0.21–0.23
9. As received steel fasteners	3	0.20	0.158–0.267
10. Molydisulfide grease	3	0.137	0.10–0.16
11. Phosphate and oil	3	0.19	0.15–0.23
12. Copper-based anti seize compound	3	0.132	0.08–0.23
13. As received steel fasteners	4	0.20	0.161–0.267
14. Plated fasteners	4	0.15	
15. Grease, oil, or wax	4	0.12	

SOURCES:
1. Values given are typical results from a very large and unpublished set of experiments on ASTM A193 B7 studs treated with various coatings. The tests were made in 1979–1980. Mean values for K varied with the diameter of the studs tested and the torques applied in various test series.
2. Kazuo Maruyama, Makoto Masuda, and Nobutoshi Ohashi, "Study of Tightening Control Methods for High Strength Bolts," *Bulletin of the Research Laboratory Precision Machine Selection*, Tokyo Institute of Technology, N46, September 1980, pp. 27–32.
3. John H. Bickford, *An Introduction to the Design and Behavior of Bolted Joints*, Marcel Dekker, Inc., New York, 1981, p. 429.
4. *Fastener Standards*, 5th ed., Industrial Fastener Institute, Cleveland, Ohio, 1970, p. N-16.
5. Edwin Rodkey, "Making Fastened Joints Reliable—Ways to Keep 'em Tight," *Assembly Engineering*, March 1977, p. 24.

$T = 0.137\,(0.75)\,19630$

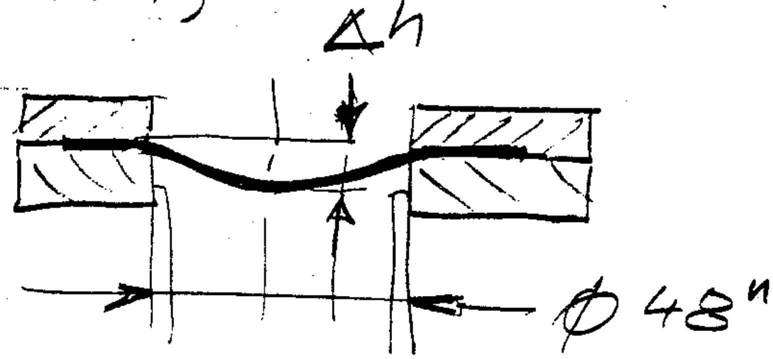
$T = 2016.9$ lbin

$T = \underline{168}$ [Ft.lb.]

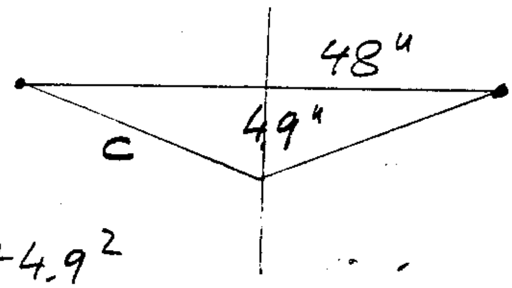
(196) Ft/lb FOR 7/8" BOLTS

10. DETERMINING REVULAR ELONGATION

USING THE DATA OF E-731 EXPERMENT - SPECIFICALLY THE 48"ϕ WINDOW TEST RESULTS (E731/E621 VACUUM WINDOW DSGN. REPORT)



$\Delta h_{max} = 4.9"$



$c^2 = 24^2 + 4.9^2$

$c = \underline{24.49"}$

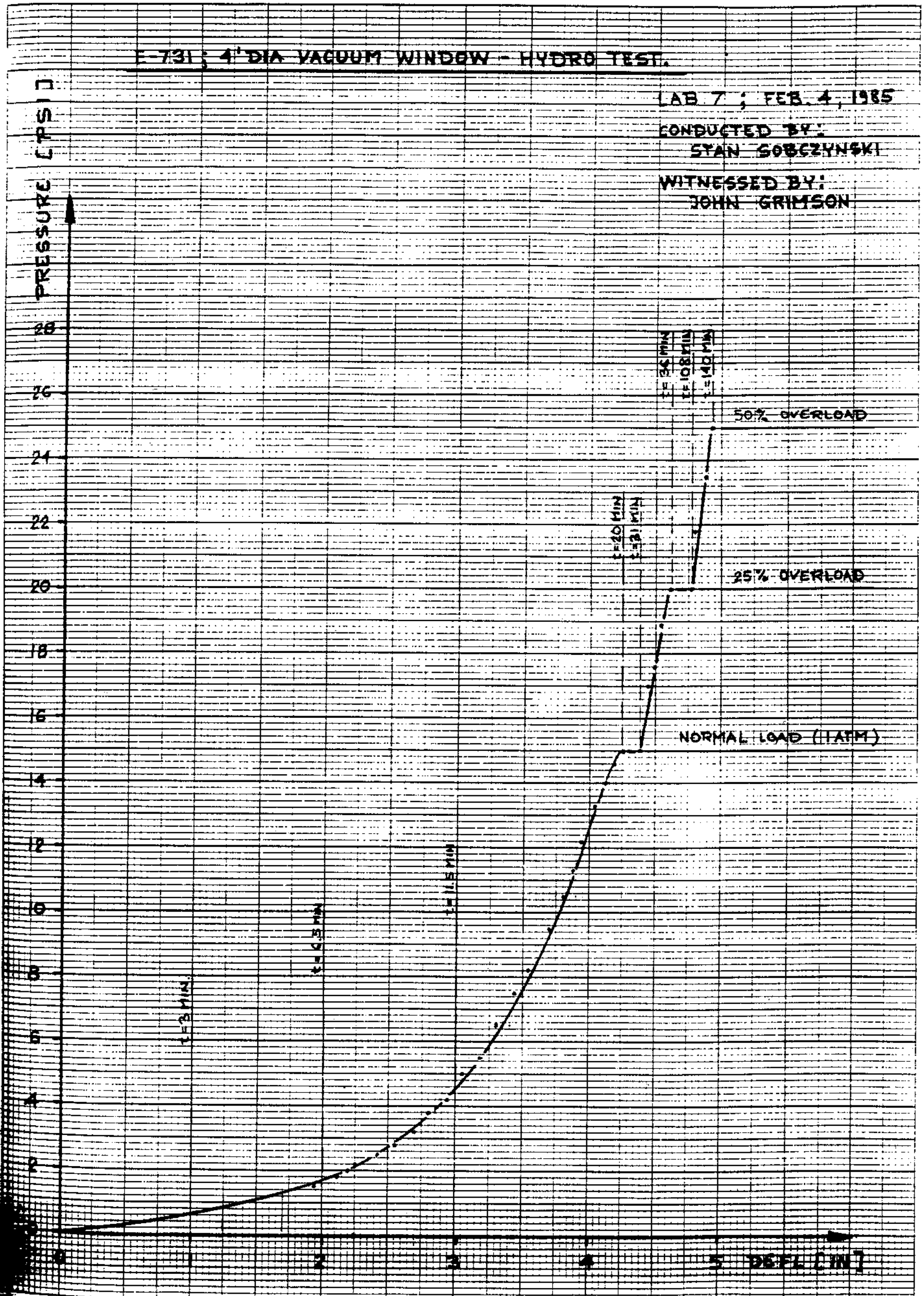
Elongation

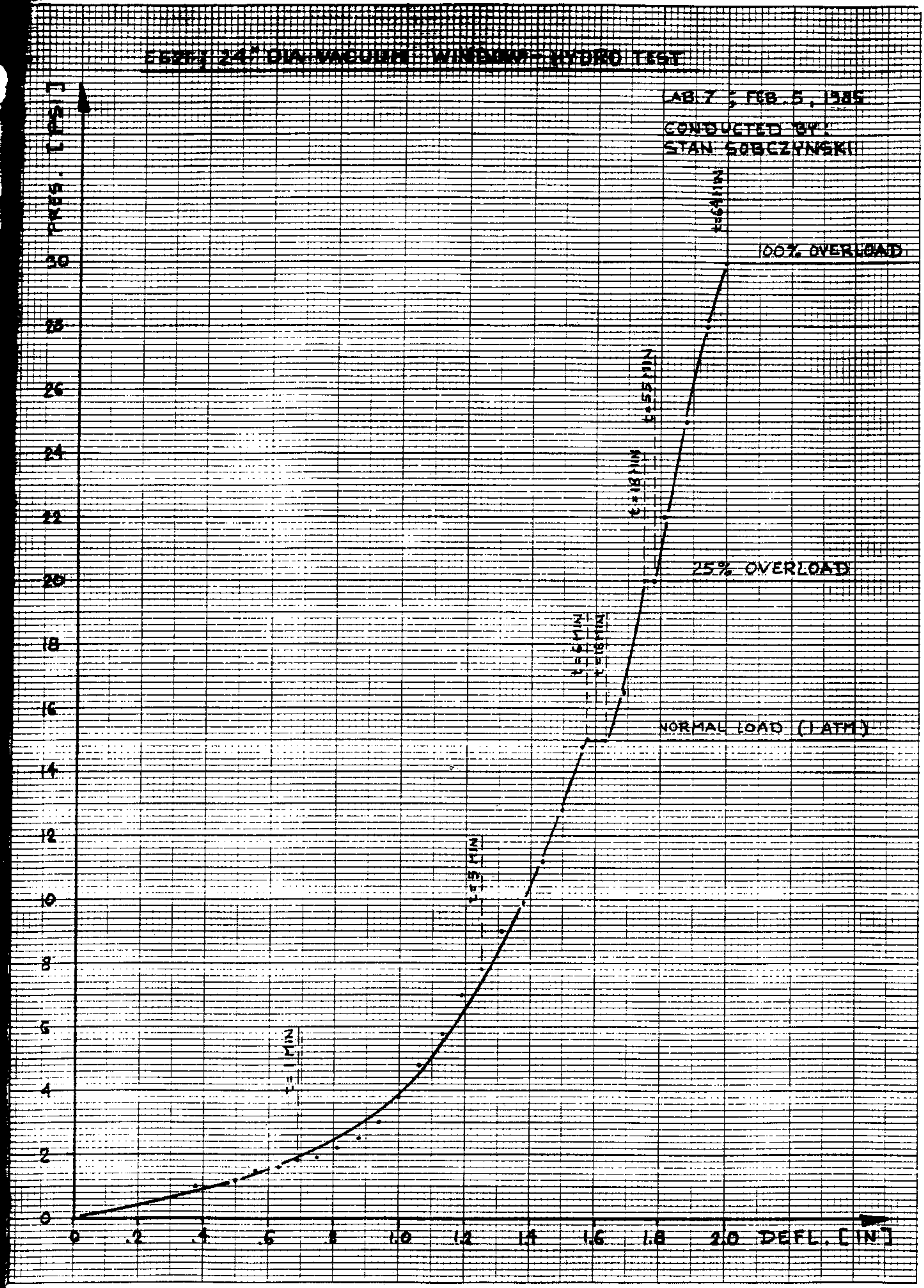
$$\varepsilon = \frac{\Delta L}{L} \cdot 100$$

$$\varepsilon = \frac{(24.49)2 - 48}{48} \cdot 100$$

$$\varepsilon = 2.06\ \%$$

E-832 REFERENCE DATA

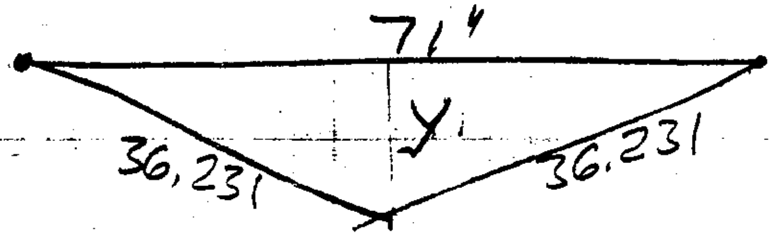



E627; 24" DIA VACUUM WINDOW HYDRO TEST
LAB 7 ; FEB. 5, 1985
CONDUCTED BY:
STAN SOBCZYNSKI
PRES. [PSI]
DEFL. [IN]
0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
1.8
2.0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
100% OVERLOAD
25% OVERLOAD
NORMAL LOAD (1 ATM)
t = 1 MIN
t = 5 MIN
t = 6 MIN
t = 16 MIN
t = 18 MIN
t = 55 MIN
t = 64 MIN

FOR 71" DIA. WINDOW THE CORRESPONDING "Y" WILL BE:

$71 - 100\%$
$x - 2.06$

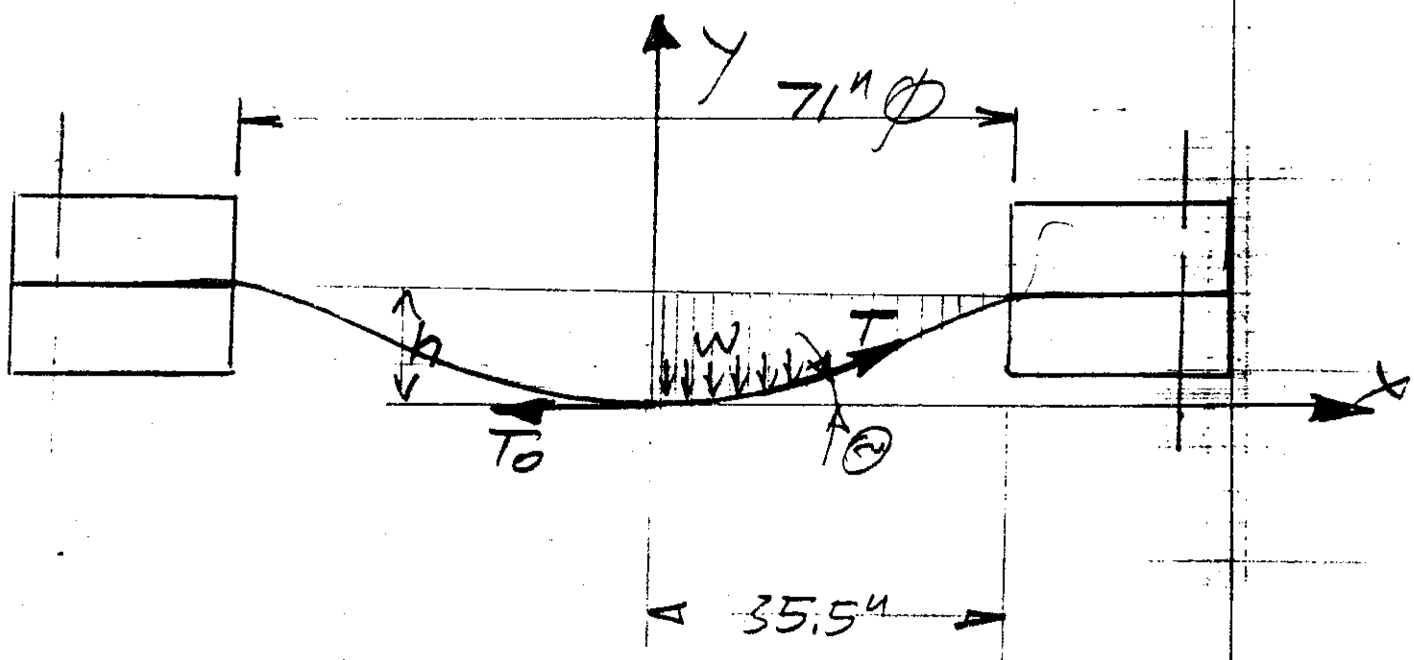
$x = 1.462$



$$36.231^2 = h^2 + 35.5^2$$

$$Y = \underline{7.24"}$$

11. EVALUATION OF THE APPROXIMATE TENSION STRESS VALUE IN THE KEVLAR FABRIC.



THE VALUE OF "Y" FROM PREVIOUS SECTION:

$$Y = \underline{7.24''}$$

$$y = \frac{wx^2}{2T_o} \quad \text{[REF. 5]}$$

DIVIDING WHOLE KEVLAR FABRIC WINDOW AREA ON 10 ELEMENTS, THE CENTER PIECE WILL HAVE LARGEST SPAN AND MAXIMUM SAG.

THE LOAD "W"

$$W = \frac{\frac{\pi(71)^2}{4}(\text{ATM, PRESSURE})}{10}$$

$$W = \frac{58200}{10}$$

$$W = 5820 \text{ lbs}$$

$$w = \frac{5820}{(2)(36.74)}$$

$$w = 79.20 \text{ lbs/in} \quad (7.1'' \text{ WIDE STRIP})$$

WITH 2 ATM PRESSURE

$$w = 79.20(2)$$

1.8m. 'ACLIM WINDOW'

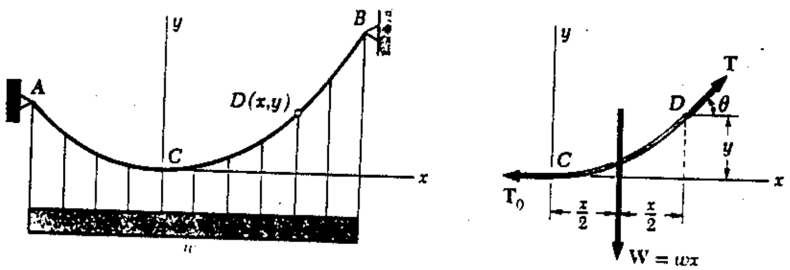
56

[5]

MECHANICS FOR ENGINEERS
F.P. BEER / E.R. JOHNSTON, JR.
THIRD EDITION

From the relations (7.5), it appears that the horizontal component of the tension force **T** is the same at any point and that the vertical component of **T** is equal to the magnitude W of the load measured from the lowest point. Relations (7.6) show that the tension T is minimum at the lowest point and maximum at one of the two points of support.

***7.8. Parabolic Cable.** Let us assume, now, that the cable AB carries a load *uniformly distributed along the horizontal* (Fig. 7.16*a*). Cables of suspension bridges may be assumed loaded in this way, since the weight of the cables is small compared with the weight of the roadway. We denote by w the load per unit length (*measured horizontally*) and express it in N/m or in lb/ft. Choosing coordinate axes with origin at the lowest point C of the cable, we find that the magnitude W of the total load carried by the portion of cable extending from C to the point D of coordinates x and y is $W = wx$. The relations (7.6) defining the magnitude and direction of the tension force at D become



(*a*) (*b*) **Fig. 7.16**

$$T = \sqrt{T_0^2 + w^2x^2} \qquad \tan\theta = \frac{wx}{T_0} \tag{7.7}$$

Moreover, the distance from D to the line of action of the resultant **W** is equal to half the horizontal distance from C to D (Fig. 7.16*b*). Summing moments about D, we write

$$+\circlearrowleft \Sigma M_D = 0: \qquad wx\frac{x}{2} - T_0 y = 0$$

$$y = \frac{wx^2}{2T_0} \tag{7.8}$$

$$7.24 = \frac{(2)79.20(35.5)^2}{(2)\;T_0}$$

$$T_0 = \frac{(2)79.20(35.5)^2}{7.24\;(2)}$$

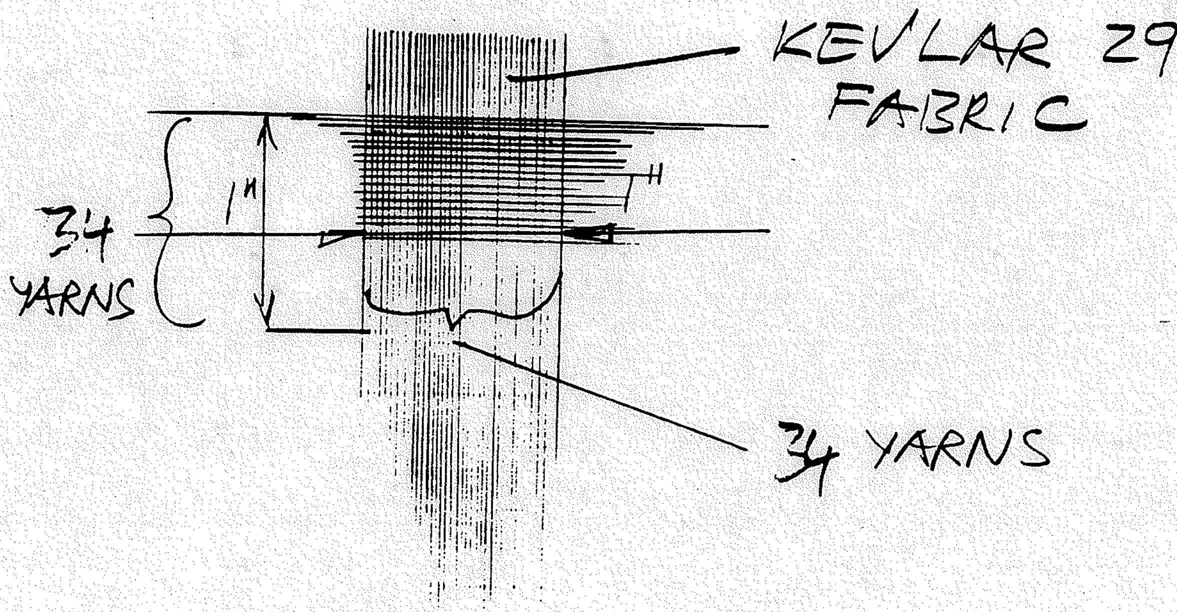
$T_0 = \underline{13786.16}$ [lbs]

$T = \sqrt{T_0^2 + W^2 x^2}$ [REF. FORMULA (7.7)]

$T = \sqrt{13786.16^2 + (2 \times 79.2)^2\, 35.5^2}$

$T = \underline{14888.9}$ [lbs]

MAX. TENSION $T_{max} = T$



YARN CONSTRUCTION :

1000 FILAMENTS / YARN

NOMINAL YARN DIA = 0.021"

NOMINAL CROSS SECTION AREA A_n = 0.0003549 in^2

CONSIDERING PACKING FACTOR OF THE FILAMENTS (70%)

THE THEORETICAL CROSS SECTION AREA OF A YARN

$$A_t = A_n(0.7)$$

$$A_t = \underline{0.000248429}\ in^2$$

CROSS SECTION AREA OF THE 7.1" WIDE STRIP

$$A_i = A_t(34) \times 7.1$$

$$A_i = 0.059\ in^2$$

E-832 1.8 M VACUUM WINDOW 5

CHARACTERISTICS AND USES OF KEVLAR® 29 ARAMID

I. INTRODUCTION

KEVLAR is the registered trademark for one member of Du Pont's family of aromatic polyamide fibers*, which have been granted the generic name "aramid" by the Federal Trade Commission. KEVLAR 29, with a tensile strength of 400,000 lb/in² (2758 Mpa+) and modulus of 9 million lb/in² (62 000 MPa), is especially suited for a number of industrial applications, including ropes, cables, protective clothing, and coated fabrics. KEVLAR 49, which has a modulus of 19 million lb/in² (131 000 MPa) and the same tensile strength as KEVLAR 29, is designed for the reinforcement of plastics and offers industry a new level of composite performance**.

KEVLAR 29 is supplied by Du Pont in filament yarns and staple fibers; product descriptions are shown in Table I. Fabrics and nonwoven felts are also being produced commercially from these fibers and yarns.

This bulletin describes the properties of KEVLAR 29 and typical applications, including fabrics and other products. More detailed technical information and current prices are available upon request.

TABLE I
YARNS OF KEVLAR® 29 ARAMID

Type 960
Yarn for Ropes and Cables with Special Finish for Improved Abrasion Resistance

Denier	Decitex*	Yield yd/lb	Yield (m/kg)**	Filament	Twist†	Nominal Yarn Diameter*** 10^{-3} in	Nominal Yarn Diameter*** mm
1500	1670	2976	6000	1000	0	21.2	0.54
9000	10000	497	1000	4000	0	52.0	1.32
15000	17000	298	600	10000	0	67.1	1.70
					0		

Type 961
Yarn for Ropes and Cables with Standard Finish

Denier	Decitex*	Yield yd/lb	Yield (m/kg)**	Filament	Twist†	Nominal Yarn Diameter*** 10^{-3} in	Nominal Yarn Diameter*** mm
1000	1110	4464	9000	666	0	17.3	0.44
1500	1670	2976	6000	1000	0	21.2	0.54
9000	10000	497	1000	4000	0	52.0	1.32
15000	17000	298	600	10000	0	67.1	1.70

Type 964
Yarn for Weaving Application

Denier	Decitex*	Yield yd/lb	Yield (m/kg)**	Filament	Twist†	Nominal Yarn Diameter*** 10^{-3} in	Nominal Yarn Diameter*** mm
200	220	22320	45000	134	0	7.8 .00008	0.20
400	440	11160	22500	267	0	11.0 .00010	0.28
1000	1110	4464	9000	666	0	17.3 .00014	0.44
1500	1670	2976	6000	1000	0	21.2 .00025	0.54

†Yarns are designated Rotoset
*Numbers have been rounded to conform to product descriptions adopted for uniformity in packaging and labeling.
**m/kg = yd/lb x 2.016.
***Assuming 70% packing factor.

3 A

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*NOMEX is also included in this generic fiber category (see Du Pont bulletin entitled "Properties of NOMEX Aramid Fiber").
**For further information see Du Pont brochure entitled "Characteristics and Uses of KEVLAR 49 Aramid High Modulus Organic Fiber".
+MPa = MN/m² = psi x 6895 x 10^{-6}

$$\sigma = \frac{T_{max}}{A_i}$$

$$\sigma = \frac{14888.9}{0.059}$$

$$\sigma = \underline{252354.23} \text{ psi}$$

THE CASE WHICH WAS CONSIDERED, INCLUDED LONGITUDINAL YARNS ONLY;
WITH THE CROSS YARNS PATTERN, THE LOAD WILL BE SPREADED EVENLY

THE ACTUAL STRESS IN THE FIBER:

$$\sigma_a = \frac{252354.23}{2} = \underline{126177.1} \text{ psi}$$

THIS NUMBER WILL TRANSLATE ITO SAFETY FACTOR OF 3.1 (FOR KEVLAR).

CHECKING THE WINDOW PARAMETERS

12. ACCORDING TO :

"MECHANICAL SAFETY SUBCOMMITTEE GUIDELINE FOR DESIGN OF THIN WINDOWS FOR VACUM VESSEL"
(PRELIMINARY TM-1380)

1.1 KEVLAR ALLOWABLE TENSILE STRESS SEE APPENDIX 1

$S = 0.5\, F_u$

F_u = ULTIMATE TENSILE STRENGTH

$F_u = 400\,000$ psi

$S = 0.5\,(400\,000)$

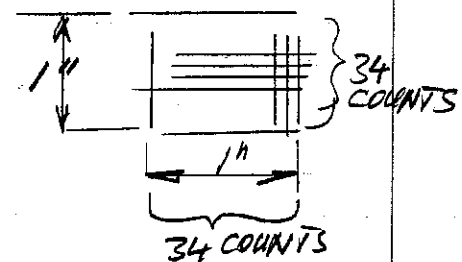
$S = \underline{200\,000}$ psi

$\sigma_a = 126,177.1$ psi

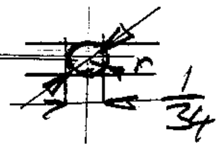
$\sigma_a < S$

12. CHECKING THE REQUIRED MYLAR THICKNESS

$$S > 0.423\left(E q^2 \frac{a^2}{t^2}\right)^{\frac{1}{3}}$$



MYLAR



$r = \dfrac{\frac{1}{34}}{2}$

UNSUPPORTED MYLAR ELEMENT
IS RESTRICTED TO THE SQUARE AREA OF $\frac{1}{34}$".

$S = 0.5\,F_u \qquad S = 0.5(26600) = 13300$ psi

$$13300 = 0.423\left(500000 \cdot 29^2 \frac{0.014^2}{t^2}\right)^{\frac{1}{3}}$$

$$\log 13300 = \log 0.423 + \frac{1}{3}\left[\log\left(500000 \cdot 29^2 \frac{0.014^2}{t^2}\right)\right]$$

$$\log 13300 = \log 0.423 + \frac{1}{3}\left[\log 500000 + 2\log 29 + 2\log 0.014 - 2\log t\right]$$

$$\log 13300 = \log 0.423 + \frac{1}{3}\log 500000 + \frac{2}{3}\log 29 + \frac{2}{3}\log 0.014 - \frac{2}{3}\log t$$

$$\frac{2}{3}\log t = \log 0.423 + \frac{1}{3}\log 500000 + \frac{2}{3}\log 29 + \frac{2}{3}\log 0.014 - \log 13300$$

$$\frac{2}{3}\log t = -0.3736 + 1.8996 + 0.9749 - 1.2359 - 4.1238$$

$$\frac{2}{3}\log t = -2.8588$$

$$\log t = \frac{(-2.8588) \cdot 3}{2}$$

$$\log t = -4.2882$$

$$10^{-4.2882} = t$$

$$\underline{t = 0.000051"}$$

APPENDIX 1 (P1 OF 10)

DRAFT - FOR COMMENT ONLY

Preliminary TM-1380
October 31, 1990

Mechanical Safety Subcommittee Guideline for Design of Thin Windows for Vacuum Vessel

by: Jeffrey L. Western

1. INTRODUCTION

This guideline specifies the usage of thin windows for vacuum vessels in terms of their design and application at Fermilab.

2. SCOPE

2.1 This guideline is to be applied to vacuum vessels when the pressure differential across the window is less than fifteen (15) psid.

$$dP \leq 15 \text{ psid} \qquad \textbf{(equation 2.1)}$$

2.2 This guideline will encompass thin windows consisting of circular, square and rectangular geometries.

2.3 This guideline shall fall under the scope of the "Fermilab Safety Manual", Vacuum Pressure Vessel Safety (5033).

2.4 Excluded from this guideline is windows for LH_2 targets which are included under "Guidelines for the Design, Fabrication, Testing, Installation and Operation of LH_2 Targets".

3. GENERAL GUIDELINES

3.1 A thin window is a diaphragm plate where the deflection is greater than 1/2 of the thickness. From Article 10.11, page 457 "Formulas for Stress and Strain" Sixth Edition.

$$y > t/2 \qquad \textbf{(equation 3.1)}$$

3.2 The allowable stress for thin windows shall be the most stringent of the following:

$$S = 0.5 F_u \qquad \textbf{(equation 3.2a)}$$

or

$$S = 0.9 F_y \qquad \textbf{(equation 3.2b)}$$

3.2.1 Where:

S = allowable stress (psi)
F_u = ultimate tensile strength (psi)
F_y = yield strength or stress to produce 5% elongation (psi)

3.3 Thin windows shall not be exposed to cyclic loading greater than 1000 load cycles. Negative load cycling is not allowed unless design and testing verifies performance.

3.4 The mounting flange shall have an edge radius to prevent the window from tearing.

3.5 Material documentation: Vendor material certification/ verification shall be included in window documentation.

3.6 Multi-layer mylar window: Mylar windows with a thickness greater than 0.010" shall have that thickness built up from multiple layers of mylar, with no single layer more than 0.010" thick.

4. GUIDELINES FOR FLEXIBLE MATERIAL WINDOWS

4.1 Flexible material circular windows such as mylar/kapton and titanium/stainless steel less than 0.003" (t < 0.003"). Design condition: held not fixed.

4.1.1 The allowable stress for circular windows shall be greater than the following: Derived from Equations 1 and 2, page 477, case number 4, page 478 "Formulas for Stress and Strain" Sixth Edition.

$$S > 0.423\,(Eq^2a^2/t^2)^{1/3} \qquad \text{(equation 4.1a)}$$

and the deflection is:

$$y = 0.662\,a\,(qa/Et)^{1/3} \qquad \text{(equation 4.1b)}$$

4.1.2 Where:

- t = thickness of window (inch)
- a = radius of window measured at O-ring on flange (inch)
- q = uniform pressure on window (psid)
- S = allowable stress (psi)
- E = Young's modulus of window material (psi)
- y = window deflection (inch)

4.2 Flexible material rectangular windows such as mylar/kapton and titanium/stainless steel less than 0.003" (**$t < 0.003$"**). Design condition: held not fixed.

4.2.1 The allowable stress for rectangular windows shall be greater than the following: From Brookhaven National Laboratory "Occupational Health and Safety Guide", Section 1.4.2, appendix B.

$$S > K_1\,(E\,(qa/t)^2)^{1/3} \qquad \text{(equation 4.2a)}$$

and the deflection is:

$$y = K_2\,(qa^4/Et)^{1/3} \qquad \text{(equation 4.2b)}$$

4.2.2 Where:

- t = thickness of window (inch)
- K_1 = stress constant based on ratio a/b (see table below)
- K_2 = deflection constant based on ratio a/b (see table below)
- a = short side of rectangular window measured at O-ring (inch)

b = long side of rectangular window measured at O-ring (inch)
q = uniform pressure on window (psid)
S = allowable stress (psi)
E = Young's modulus of window material (psi)
y = window deflection (inch)

4.2.3 Constant table for values of K for rectangular windows:

b/a	K_1	K_2
1.0	0.271	0.320
1.1	0.292	0.331
1.2	0.306	0.339
1.3	0.316	0.344
1.4	0.323	0.348
1.5	0.329	0.351
1.6	0.332	0.353
1.7	0.336	0.355
1.8	0.338	0.356
1.9	0.340	0.357
2.0	0.340	0.357
3.0	0.346	0.360
> 3.0	0.346	0.360

Note: K_1 values for maximum stress at center of window.

5. GUIDELINES FOR RIGID MATERIAL WINDOWS

5.1 Rigid material circular windows such as titanium and stainless steel greater than 0.003" (t > 0.003"). Design condition: held and fixed.

5.1.1 The allowable stress for circular windows shall be greater than the stress obtained by solving by trial and error equation 5.1b for deflection and then substitute into 5.1a. See equations 1 and 2, page 477, case number 3, page 478 "Formulas for Stress and Strain" Sixth Edition.

$$S > E(t/a)^2[K_3(y/t) + K_4(y/t)^2] \qquad \text{(equation 5.1a)}$$

and the deflection can be calculated from:

$$qa^4/Et^2 = K_1(y/t) + K_2(y/t)^3 \qquad \text{(equation 5.1b)}$$

5.1.2 Where:

- **t** = thickness of window (inch)
- **a** = radius of window measured at O-ring on flange (inch)
- **q** = uniform pressure on window (psid)
- **S** = allowable stress (psi)
- **E** = Young's modulus of window material (psi)
- **y** = window deflection (inch)
- K_1 = $5.33/(1-\nu^2)$
- K_2 = $2.6/(1-\nu^2)$
- K_3 = $2.0/(1-\nu)$
- K_4 = 0.976
- ν = poisson's ratio

Note: K values for maximum stress at center of window.

5.2 Rigid material rectangular windows such as titanium and stainless steel greater than 0.003" **(t > 0.003")**. Design condition: held and fixed.

5.2.1 The allowable stress for rectangular windows shall be greater than the stress obtained by solving by trial and error equation 5.2b for deflection and then substitute into 5.2a. From Brookhaven National Laboratory "Occupational Health and Safety Guide", Section 1.4.2, appendix C.

$$S > E(t/a)^2[K_3(y/t) + K_4(y/t)^2] \qquad \text{(equation 5.2a)}$$

and the deflection can be calculated from:

$$qa^4/Et^2 = K_1(y/t) + K_2(y/t)^3 \qquad \text{(equation 5.2b)}$$

5.2.2 Where:

t = thickness of window (inch)
K_1 = stress constant based on ratio a/b (see table below)
K_2 = deflection constant based on ratio a/b (see table below)
a = short side of rectangular window measured at O-ring (inch)
b = long side of rectangular window measured at O-ring (inch)
q = uniform pressure on window (psid)
S = allowable stress (psi)
E = Young's modulus of window material (psi)
y = window deflection (inch)

5.2.3 Constant table for values of K for rectangular windows:

b/a	K_1	K_2	K_3	K_4
1.0	72.5	30.5	22.3	2.7
1.1	60.9	27.6	21.3	2.7
1.2	53.2	25.7	20.4	2.7
1.3	47.9	24.6	19.7	2.7
1.4	44.3	23.7	19.3	2.7
1.5	41.5	23.1	18.9	2.7
1.6	39.8	22.7	18.7	2.7
1.8	37.5	22.2	18.3	2.7
2.0	36.1	22.0	18.0	2.7
> 2.0	35.2	21.4	17.6	2.7

Note: K_3 and K_4 values for maximum stress at the midpoint of the long edge of window.

6. RESPONSIBILITY / DOCUMENTATION

The responsibility and documentation for thin windows shall follow the "Fermilab Safety Manual", Vacuum Pressure Vessel Safety (5033).

APPENDIX A

EXAMPLE 1: Circular flexible window

Circular 3" diameter by 0.005" thickness Mylar window under vacuum.

t	=	0.005": thickness of window (inch)
a	=	1.5": radius of window measured at O-ring on flange (inch)
q	=	14.7 psi: uniform pressure on window (psid)
S	=	$0.5\ F_u = 0.5 \times 25{,}000$ psi = <u>12,500 psi</u> or $0.9\ F_y = 0.9 \times 15{,}000$ psi = 13,500 psi: allowable stress (psi)
E	=	500,000 psi: Young's modulus of window material (psi)
y	=	window deflection (inch) <= to be calculated

From TM-1380 Page 2, Section 4.1, Circular window with edge held but not fixed.

$$S > 0.423\,(Eq^2a^2/t^2)^{1/3} \qquad \textbf{(equation 4.1a)}$$

$$12{,}500 \text{ psi} > 0.423\,[(500{,}000)(14.7)^2(1.5)^2/0.005^2]^{1/3}$$

$$12{,}500 \text{ psi} > \underline{9{,}029 \text{ psi}} \Rightarrow \text{Adequate}$$

and the deflection is:

$$y = 0.662\,a\,(qa/Et)^{1/3} \qquad \textbf{(equation 4.1b)}$$

$$y = 0.662\,(1.5)\,[(14.7)(1.5)/(500{,}000)(0.005)]^{1/3}$$

$$y = \underline{0.205 \text{ inch}} \Leftarrow \text{Deflection}$$

and:

$$y > t/2 \qquad \textbf{(equation 3.1)}$$

$$0.205'' > 0.005''/2$$

$$0.205'' > 0.0025'' \Rightarrow \text{Adequate}$$

APPENDIX A

EXAMPLE 2: Square flexible window

Square 3" by 0.009" thickness Kapton window under vacuum.

t	=	0.009": thickness of window (inch)
a	=	3.0": short side of rectangular window measured at O-ring (inch)
b	=	3.0": long side of rectangular window measured at O-ring (inch)
q	=	14.7 psi: uniform pressure on window (psid)
S	=	$0.5\ F_u = 0.5 \times 25{,}000$ psi = 12,500 psi or $0.9\ F_y = 0.9 \times 12{,}500$ psi = <u>11,250 psi</u>: allowable stress (psi)
E	=	400,000 psi: Young's modulus of window material (psi)
y	=	window deflection (inch) <= to be calculated

From TM-1380 Page 3, Section 4.2, Rectangular window with edge held but not fixed.

from table 4.2.3: for b/a = 3.0/3.0 = 1 ==> $K_1 = 0.271$ & $K_2 = 0.320$

$$S > K_1\,[E(qa/t)^2]^{1/3} \qquad \textbf{(equation 4.2a)}$$

$$11{,}250 \text{ psi} > 0.271\,[(400{,}000)(14.7)^2(3.0)^2/(0.009)^2]^{1/3}$$

$$11{,}250 \text{ psi} > \underline{5760 \text{ psi}} \Rightarrow \text{Adequate}$$

and the deflection is:

$$y = K_2\,(qa^4/Et)^{1/3} \qquad \textbf{(equation 4.2b)}$$

$$y = 0.320\,[(14.7)(3.0)^4/(400{,}000)(0.009)]^{1/3}$$

$$y = \underline{0.221 \text{ inch}} \Leftarrow \text{Deflection}$$

and:

$$y > t/2 \qquad \textbf{(equation 3.1)}$$

$$0.221" > 0.009"/2$$

$$0.221" > 0.0045" \Rightarrow \text{Adequate}$$

APPENDIX A

EXAMPLE 3: Circular rigid window

Circular 6" diameter by 0.005" thickness stainless steel (work hardened 302 S.S.) window under vacuum.

t	=	0.005": thickness of window (inch)
a	=	3.0": radius of window measured at O-ring on flange (inch)
q	=	14.7 psi: uniform pressure on window (psid)
S	=	$0.5\ F_u = 0.5 \times 250{,}000$ psi = <u>125,000 psi</u> or $0.9\ F_y = 0.9 \times 150{,}000$ psi = 135,000 psi: allowable stress (psi)
E	=	30E6 psi: Young's modulus of window material (psi)
v	=	0.3 : poisson's ratio
y	=	window deflection (inch) <= to be calculated

From TM-1380 Page 4, Section 5.1, Circular window with edge held and fixed.

from section 5.1.2: for $v = 0.3 \Rightarrow K_1 = 5.86,\ K_2 = 2.86,\ K_3 = 2.86,\ K_4 = .976$

deflection is: $qa^4/Et^4 = K_1(y/t) + K_2(y/t)^3$ **(equation 5.1b)**

$$14.7(3.0)^4/30E6(.005)^4 = 5.86\ (y/.005) + 2.86\ (y/.005)^3$$

y = <u>0.14 inch</u> <= Deflection......solved by trial and error

where: y > t/2 **(equation 3.1)**

0.14" > 0.005"/2

0.14" > 0.0025" => Adequate

stress is: $S > E(t/a)^2[K_3(y/t) + K_4(y/t)^2]$ **(equation 5.1a)**

125,000 psi > $(30E6)(.005/3.0)^2[2.86(.14/.005) + 0.976(.14/.005)^2]$

125,000 psi > <u>38,817 psi</u> => Adequate

APPENDIX A

EXAMPLE 4: Rectangular rigid window

Rectangular 6" x 9" by 0.006" thickness titanium (6AL-4V) window under vacuum.

t	=	0.006": thickness of window (inch)
a	=	6.0": short side of rectangular window measured at O-ring (inch)
b	=	9.0": long side of rectangular window measured at O-ring (inch)
q	=	14.7 psi: uniform pressure on window (psid)
S	=	$0.5\ F_u = 0.5 \times 130{,}000$ psi = 65,000 psi or $0.9\ F_y = 0.9 \times 120{,}000$ psi = 108,000 psi: allowable stress (psi)
E	=	16E6 psi: Young's modulus of window material (psi)
v	=	0.3 : poisson's ratio
y	=	window deflection (inch) <= to be calculated

From TM-1380 Page 5, Section 5.2, Rectangular window with edge held and fixed.

from table 5.2.3: for b/a = 1.5 => $K_1 = 41.5$, $K_2 = 23.1$, $K_3 = 18.9$, $K_4 = 2.7$

deflection is: $qa^4/Et^4 = K_1(y/t) + K_2(y/t)^3$ **(equation 5.2b)**

$$14.7(6.0)^4/16E6(.006)^4 = 41.5\,(y/.006) + 23.1\,(y/.006)^3$$

y = 0.21 inch <= Deflection......solved by trial and error

where: y > t/2 **(equation 3.1)**

0.21" > 0.006"/2

0.21" > 0.003" => Adequate

stress is: $S > E(t/a)^2[K_3(y/t) + K_4(y/t)^2]$ **(equation 5.2a)**

65,000 psi $> (16E6)(.006/6.0)^2[18.9(.21/.006) + 2.7(.21/.006)^2]$

65,000 psi > 63,504 psi => Adequate

13. CHECKING THE STRESS VALUES FOR 2" THICK FLANGE (A) FIG A7
(DWG. 9220, 832, MD - 285383)

NOMENCLATURE: ASME, SECTION VIII, DIV. I, APPENDIX 14.

OPERATING MOMENT:

$$M_o = W \frac{(C - G)}{2}$$

$$G = 1.563 (2) + 94.5$$

$$G = 97.626''$$

$$C = 100''$$

$$b = 0.315''$$ $$W = W_{m_1}$$

$$W_{m_1} = 0.785 (97.626)^2 (30) + (2 \times 0.315 \times 3.14 \times 97.626 \times 0.5 \times 3$$

$$W_{m_1} = 224{,}451.18 + 2896.8$$

$$\underline{W_{m_1} = 227{,}348.0}$$

$$M_o = 227{,}348.0 \; \frac{100 - 97.626}{2}$$

$$\underline{M_o = 269{,}862.11} \quad [lb\,in]$$

FOR AN OPENING WITHOUT A NOZZLE:

$$(EQ)^* = \left(\frac{B_n}{t}\right) S_T^*$$

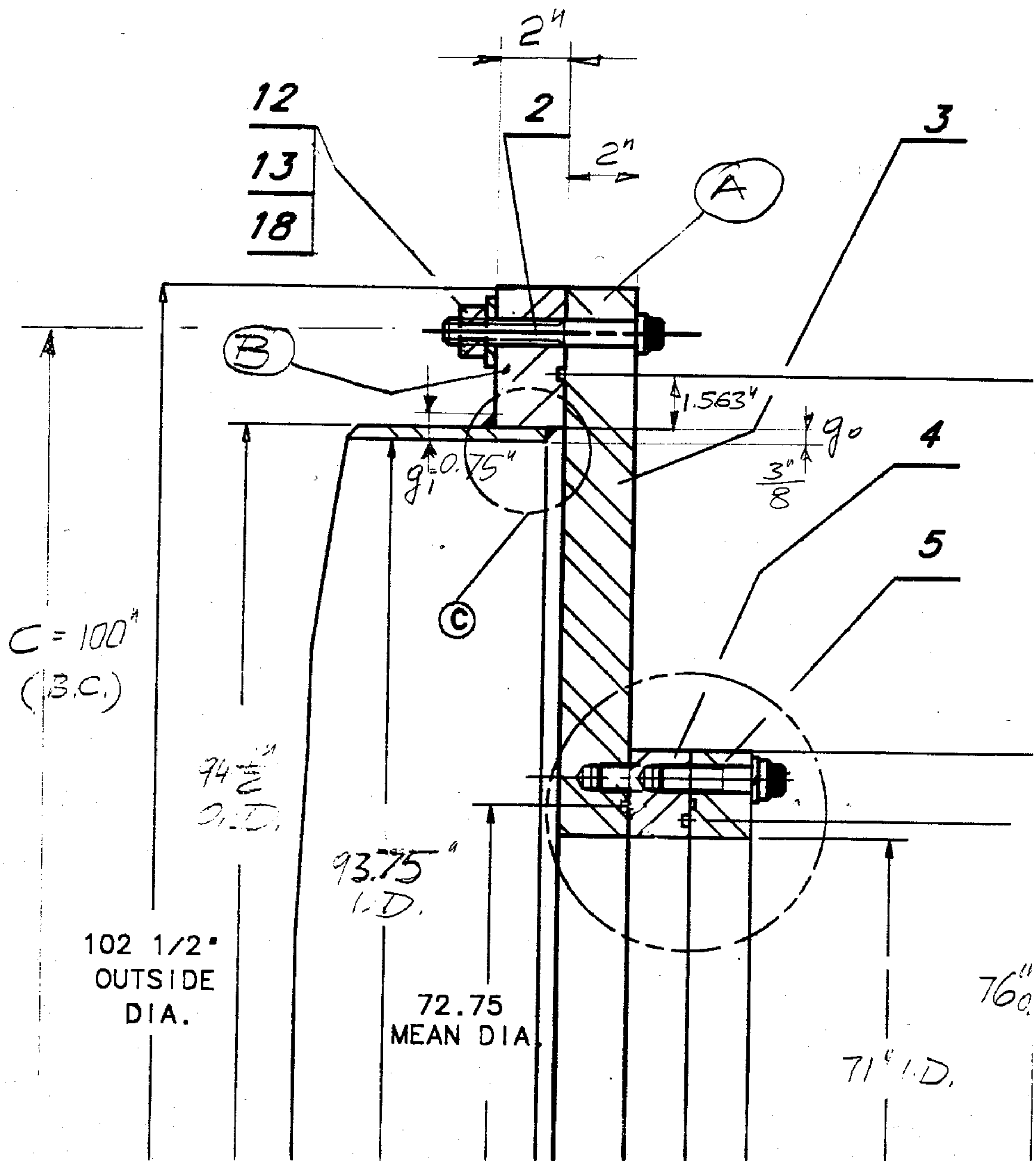
$$B_n = \underline{71''}$$

$$S_T = \frac{Y M_o}{t^2 B} - Z \cdot S_R$$

E-832

1.8 M VACUUM WINDOW
ASSEMBLY (A)
DWG. 9220, 832, MD-285383

FIG. A7



y - FACTOR INVOLVING K (FROM FIG, 2.7.1)

t - FLANGE THICKNESS

$t = 2''$

A - OUTSIDE DIA OF FLANGE

$A = 102.5''$

B - INSIDE DIA OF FLANGE

$B = 71''$

$$K = \frac{A}{B}$$

$K = 1.44$

FROM FIG. 2.7.1
PAGE 24

$Y = 5.5$

$Z = 3$

$$S_R = \frac{(1.33te + 1) M_o}{L t^2 B}$$

$e = \frac{F}{h_o}$ FROM FIG. 2.7.2

$h = 0$

$h_o = \sqrt{B \cdot g_o} = \sqrt{(71)(\frac{3}{8})}$

$h_o = 5.15$

$$\frac{h}{h_o} = \underline{0.0}$$

$$\frac{g_1}{g_o} = \frac{0.75}{0.375} = \underline{2}.$$

F — FROM FIG. 2-7.2

$$F \approx \underline{0.915}$$

L — factor

$$L = \frac{te + 1}{T} + \frac{t^3}{d} \qquad \underline{t = 2''}$$

T FROM 2.7.1

$$T = \underline{1.7}$$

d — factor

$$d = \frac{U}{V} h_o g_o^2 \qquad U = \underline{6.25}$$

FIG. 2-7.3

$$V = \underline{0.45}$$

$$d = \frac{6.25}{0.45}(5.15)(0.375)^2$$

$$d = \underline{9.42}$$

$$e = \frac{F}{h_o} = \frac{0.915}{5.15}$$

$$e = \underline{0.177}$$

$$L = \frac{(2)(0.177) + 1}{1.7} + \frac{(2)^3}{9.42}$$

$$L = \underline{1.645}$$

$$S_R = \frac{(1.33\,te + 1)\,M_o}{L\,t^2\,B}$$

$$S_R = \frac{[1.33(2)(0.177 + 1]\,269{,}862.11}{1.645\,(2)^2\,(71)}$$

$$S_R = \underline{849.6} \text{ psi}$$ ✓

$$S_T = \frac{y\,M_o}{t^2\,B} - Z\,S_R$$

$$S_T = \frac{(5.5)\,269862.11}{(2)^2\,71} - 3\,(849.6)$$ ✓

$$S_T = 5226.2 - 2548.8$$

$$S_T = \underline{2677.4} \text{ psi}$$

EQ CALCULATION

(FOR AN OPENING WITHOUT A NOZZLE)

$$EQ = \left(\frac{B_n}{t}\right) S_T$$

$$EQ = \left(\frac{71}{2}\right) 2677.4$$

$$EQ = \underline{95047.7}$$

$$\frac{EQ}{M_o} = \frac{95047.7}{269862.11}$$

$$\frac{EQ}{M_o} = \underline{0.352}$$

CALCULATE M_H

$$M_H = \frac{EQ}{\frac{1.74\, h_o V}{g_o^3\, B_1} + \frac{EQ}{M_o}\left(1 + \frac{F\, t}{h_o}\right)}$$

$$M_H = \frac{95047.7}{\frac{1.74\,(5.15)\,0.45}{(0.375)^3\; 94.5} + 0.352\left(1 + \frac{0.915\,(2)}{5.15}\right)}$$

$$M_H = \frac{95047.7}{0.80917672 + 0.4770}$$

$$M_H = \underline{73894.83}$$

CALCULATE X_1

$$X_1 = \frac{M_O - M_H\left(1 + \frac{F\,t}{h_o}\right)}{M_O}$$

$$X_1 = \frac{269\,862.11 - 73894.83\left(1 + \frac{0.915 \times 2}{5.15}\right)}{269\,862.11}$$

$$X_1 = \underline{0.628}$$

✓

LONGITUDINAL HUB STRESS IN SHELL

$$S_{HS} = X_1(EQ)\,\frac{1.10\ h_o\ f}{\left(\frac{g_1}{g_o}\right)^2 B_s\ V}$$

$$S_{HS} = 0.628\,(95047.7)\,\frac{1.10\,(5.15)\ f}{\left(\frac{g_1}{g_o}\right)^2 B_s\ V}$$

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

$$\frac{h}{h_o} = 0$$

$$f = \underline{4}$$

FIG. 2.7.6.

$$S_{HS} = 0.628 \times (95047.7) \frac{1.10(5.15)(4)}{(2)^2 \cdot 93.750 \cdot (0.45)}$$

$$S_{HS} = \underline{8015.2} \text{ psi}$$

RADIAL STRESS AT OUTSIDE DIA. :

$$S_{RS} = \frac{1.91 \times M_H \left(1 + \frac{Ft}{h_o}\right)}{B_s t^2} + \frac{0.64 F M_H}{B_s h_o t}$$

$$S_{RS} = \frac{1.91(738948)\left(1 + \frac{0.915(2)}{5.15}\right)}{93.75(2)^2} + \frac{0.64(0.915)738948}{93.75(5.15)(2)}$$

$$S_{RS} = 510.10 + 44.8$$

$$S_{RS} = \underline{555.0} \text{ psi}$$

TANGENTIAL STRESS AT OUTSIDE DIAMETER

$$S_{TS} = \frac{(X_1)(EQ)t}{B_s} - \frac{0.57\left(1 + \frac{Ft}{h_o}\right) M_H}{B_s t^2} +$$

$$+ \frac{0.64 F Z M_H}{B_s h_o t}$$

$$Z = \frac{K^2 + 1}{K^2 - 1} = \frac{1.44^2 + 1}{1.44^2 - 1} = \underline{2.862}$$

$$S_{TS} = \frac{0.628(95047.7)(2)}{93.75} - \frac{0.57\left(1+\frac{0.915(2)}{5.15}\right)73894.83}{93.75\;(2)^2}$$

$$+\frac{0.64(0.915)(2.862)\;73894.83}{93.75(5.15)\;2}$$

$$S_{TS} = 1273.38 - 152.23 + 128.2$$

$$S_{TS} = \underline{1249.4} \quad psi$$

- OPENING HEAD JUNCTURE

LONGITUDINAL HUB STRESS IN CENTRAL OPENING.

$$S_{HO} = X_1 \, S_H$$

$$S_{HO} = 0.628(S_H)$$

$$S_H = \frac{f\,M_o}{L\,g_1^2\,B} = \frac{4(269{,}862.11)}{1.645(0.75)\,71}$$

$$S_H = \underline{12{,}323.} \quad psi$$

$$S_{HO} = 0.628(12{,}323)$$

$$S_{HO} = \underline{7738.8} \quad psi$$

RADIAL STRESS AT CENTRAL OPENING

$$S_{RO} = X_1 \, S_R$$

$$S_{RO} = 0.628\,(849.6)$$

$$S_{RO} = \underline{533.54} \quad \text{psi}$$

TANGENTIAL STRESS AT DIAMETER OF CENTRAL OPENING.

$$S_{TO} = X_1 S_T + \frac{0.64\, F Z_1\, M_c}{B_s\, h_o\, t}$$

$$Z_1 = \frac{2K^2}{K^2 - 1}$$

$$Z_1 = \frac{2\,(1.44)^2}{1.44^2 - 1}$$

$$Z_1 = \underline{3.862}$$

$$S_{TO} = 0.628\,(2677.4) + \frac{0.64\,(0.915)\,3.862\,(73894.83)}{93.75\,(5.15)\,(2)}$$

$$S_{TO} = 1681.4 + 173.06$$

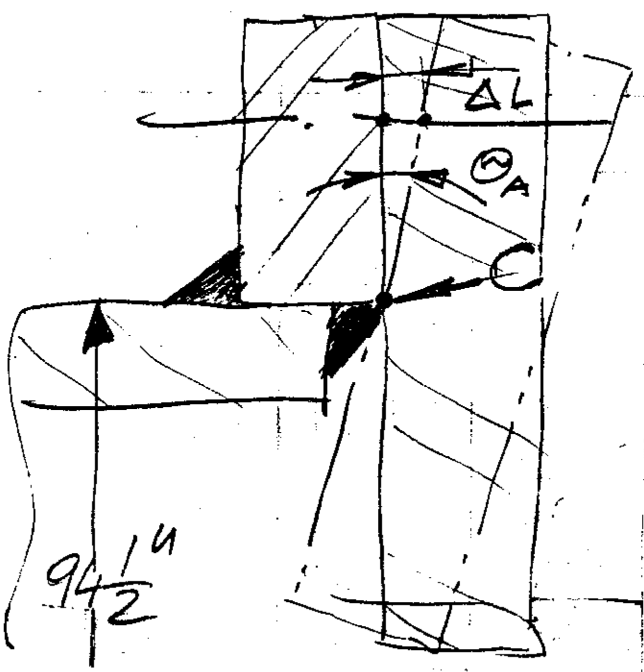
$$S_{TO} = \underline{1854.46} \quad \text{psi}$$

14. CHECKING THE REQUIRED NUMBER OF BOLTS IN A & B FLANGE CONNECTION, BASED ON DEFLECTION OF FLANGE Ⓐ.

THIS ANALYSIS IS BASED ON THE FOLLOWING ASSUMPTIONS:

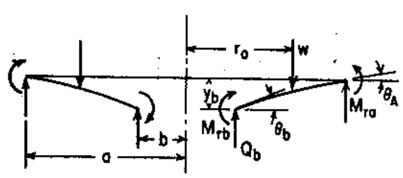
a/ PLATE'S SUPPORT CONTACT IS AT POINT "C"

b) THE DEFLECTION y_b WILL PRODUCE THE ΔL DISLOCATION AT FLANGE BOLT CIRCLE

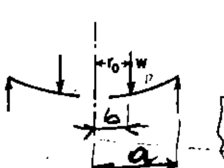


Case 1. Annular plate with a uniform annular line load of w lb/in at a radius r_o

General expressions for deformations, n



For the numerical data given below, ν

Case no., edge restraints	Boundary values	Special cases
1a. Outer edge simply supported, inner edge free 	$M_{rb} = 0 \quad Q_b = 0 \quad y_a = 0 \quad M_{ra} = 0$ $y_b = \frac{-wa^3}{D}\left(\frac{C_1 L_9}{C_7} - L_3\right)$ $\theta_b = \frac{wa^2}{DC_7} L_9$ $\boxed{\theta_a = \frac{wa^2}{D}\left(\frac{C_4 L_9}{C_7} - L_6\right)}$ $Q_a = -w\frac{r_o}{a}$	Max $y = y_b$ Max $M = M_{tb}$ If $r_o = b$ (load at inner edge),

b/a	0.1	0.3	0.5	0.7	0.9
K_{y_b}	−0.0364	−0.1266	−0.1934	−0.1927	−0.0938
K_{θ_b}	0.0371	0.2047	0.4262	0.6780	0.9532
K_{θ_a}	0.0418	0.1664	0.3573	0.6119	0.9237
$K_{M_{tb}}$	0.3374	0.6210	0.7757	0.8814	0.9638

$$W = \frac{P_{30}}{\pi(71)}$$

$$W = \frac{207087}{\pi(71)} = \underline{928.42} \left[\frac{lbs}{in}\right]$$

$$a = \frac{94.5}{2}$$

$$a = \underline{47.25''}$$

$$b = \frac{71}{2} \qquad \underline{r_o = b = 35.5}$$

$$b = \underline{35.5''}$$

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

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

$$D = \frac{Et^3}{12(1-\nu^2)}$$

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

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

$$C_4 = \frac{1}{2}\left[(1+0.3)\frac{35.5}{47.25} + (1-0.3)\frac{47.25}{35.5}\right]$$

$$C_4 = \frac{1}{2}\left[0.9767 + 0.9316\right]$$

$$\underline{C_4 = 0.9541}$$

$$C_7 = \frac{1}{2}\left(1-0.3^2\right)\left(\frac{47.25}{35.5} - \frac{35.5}{47.25}\right)$$

$$C_7 = \frac{1}{2}(0.91)(0.5796)$$

$$\underline{C_7 = 0.2637}$$

$$D = \frac{30\,000000\,(2)^3}{12\,(1-0.3^2)}$$

$$\underline{D = 21,978,021.98}$$

$$L_6 = \frac{35.5}{4(47.25)}\left[\left(\frac{35.5}{47.25}\right)^2 - 1 + 2\ln\frac{47.25}{35.5}\right]$$

$$L_6 = 0.1878\left[0.5644 - 1 + 0.5718\right]$$

$$\underline{L_6 = 0.02558}$$

$$L_9 = \frac{35.5}{47.25}\left\{\frac{1+0.3}{2}\ln\frac{47.25}{35.5} + \frac{1-0.3}{4}\left[1-\left(\frac{35.5}{47.25}\right)^2\right]\right\}$$

$$L_9 = 0.7513\left\{0.1858 + 0.07621\right\}$$

$$L_9 = \underline{0.1968}$$

$$L_7 = \frac{1}{2}(1-\nu^2)\left(\frac{a}{r_o} - \frac{r_o}{a}\right)$$

$$L_7 = \frac{1}{2}(1-0.3^2)\left(\frac{47.25}{35.5} - \frac{35.5}{47.25}\right)$$

$$L_7 = 0.455(1.3309 - 0.7513)$$

$$L_7 = \underline{0.2637}$$

$$L_4 = \frac{1}{2}\left[(1+\nu)\frac{r_o}{a} + (1-\nu)\frac{a}{r_o}\right.$$

$$L_4 = \frac{1}{2}\left[(1+0.3)\frac{35.5}{47.25} + (1-0.3)\frac{47.25}{35.5}\right]$$

$$L_4 = \underline{0.4796}$$

$$\Theta_a = \frac{(928.42)(47.25)^2}{21{,}978{,}021.98}\left(\frac{(0.9541)(0.1968)}{0.2637} - 0.02558\right)$$

$$\Theta_a = 0.094310383\left(0.686467327\right)$$

$$\Theta_a = \underline{0.0647} \text{ RADIANS}$$

C) THE DISLOCATION ΔL WILL ALLOW US TO FIND THE ALGULAR DISPLACEMENT

$\theta_o = \theta_a$

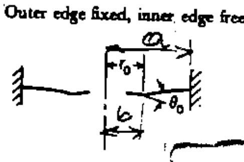
RESOLVING THIS CASE AS :

" OUTER EDGE FIXED, INNER EDGE FREE",

AND FINDING MOMENT AT FLANGE B.C, WILL GIVE US IN RESULT

THE REQUIRED BOLT CLAMPING FORCE.

6e. Outer edge fixed, inner edge free



$M_{rb} = 0 \quad Q_b = 0 \quad y_a = 0 \quad \theta_a = 0$

$$y_b = \theta_o a\left(\frac{C_1 L_4}{C_4} - L_1\right)$$

$$\theta_b = \frac{-\theta_o L_4}{C_4}$$

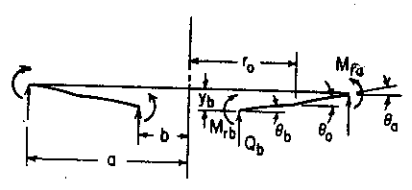
$$M_{ra} = \frac{\theta_o D}{a}\left(L_7 - \frac{C_7}{C_4} L_4\right)$$

$Q_a = 0$

b/a	0.1		0.5		0.7
r_o/a	0.5	0.7	0.7	0.9	0.9
K_{y_b}	0.0534	0.2144	0.1647	0.3649	0.1969
K_{y_o}	−0.0975	−0.0445	−0.0155	−0.0029	−0.0013
K_{θ_b}	−0.2875	−0.2679	−0.9317	−0.9501	−1.0198
$K_{M_{ra}}$	−2.6164	−2.4377	−1.6957	−1.7293	−1.3257

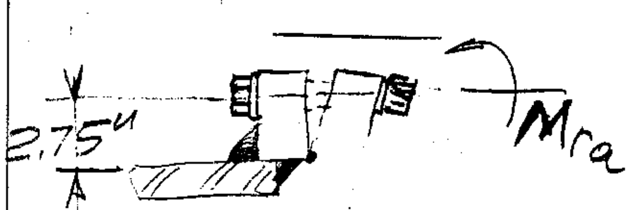
Case 6. Annular plate with an externally applied angular displacement θ_o on an annulus with a radiu

General expressions for deformations, momen



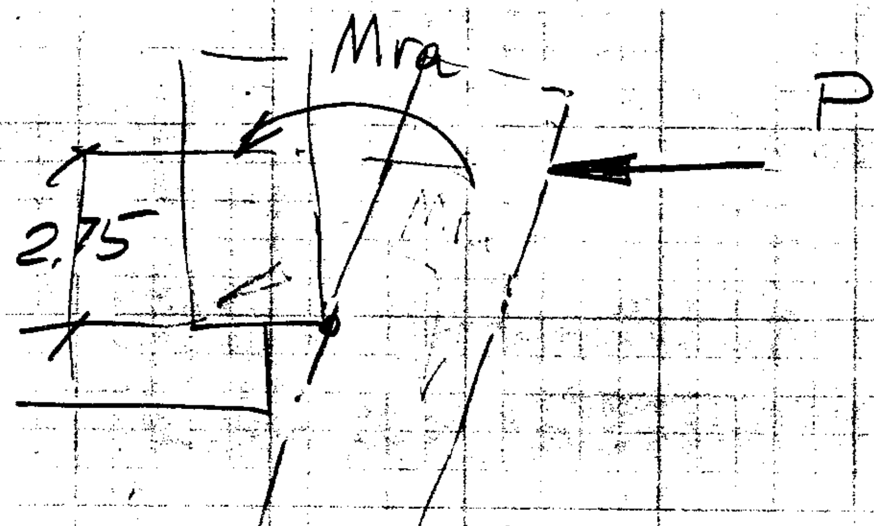
For the numerical data given below, $\nu = 0.3$

$y = \cdot$



$$M_{ra} = \frac{0.0647(21,978,021.98)}{47.25}\left(0.2637 - \frac{0.2637}{0.9541}\, 0.4796\right)$$

$$\underline{M_{ra} = 3946.78} \quad [\text{in lbs}]$$



$$M_{ra} = P(2.75)$$

$$\text{Min } P = \frac{M_{ra}}{2.75}$$

SELECTED $\frac{3}{4}$"-10 UNRC FERRY CUP BOLTS ARE ADEQUATE TO COVER THIS REQUIREMENT.

15. DETERMINING THE CRITICAL LENGTH OF THE VACUUM VESSEL

$$L_c = 1.14\left(1-\nu^2\right)^{\frac{1}{4}} D_o \left(\frac{D_o}{t}\right)^{\frac{1}{2}}$$ REF. [3] PAGE 50

$$L_c = 1.14\left(1-0.3\right)^{\frac{1}{4}} 94.5 \left(\frac{94.5}{0.375}\right)^{\frac{1}{2}}$$

$$L_c = 1.0427\left(1500.1\right)$$

$$L_c = \underline{1564.19''}$$

THE MINIMUM UNSUPPORTED LENGTH OF THE VESSEL WITH 94.5 O.D AND 3/8" WALL THICKNESS IS 1564.19" WHAT REPRESENTS MUCH HIGHER THAN REQUIRED FOR TESTING PURPOSE, NUMBER.

16. DETERMINING THE REQUIRED THICKNESS OF A FLAT ENCLOSURE HEAD OF A TEST VACUUM VESSEL

INTERNAL OR EXTERNAL PRESSURE

FORMULAS

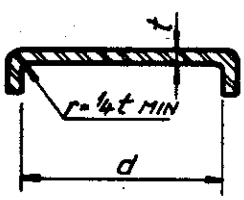
p. 26

REF. [4] PAGE 26

NOTATION

- P = Internal or external design pressure PSI **E = Joint effiency** p. 142
- d = Inside diameter of shell, inches
- S = Maximum allowable stress value of material, PSI, Page 19
- 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

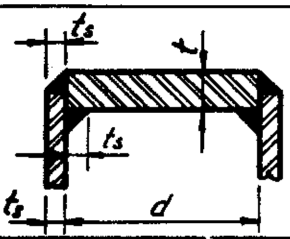


CIRCULAR FLAT HEADS

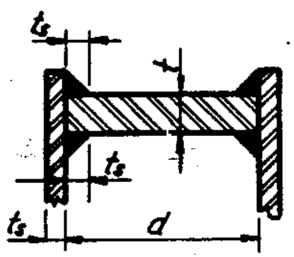
$$t = d\sqrt{0.13P/SE}$$

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



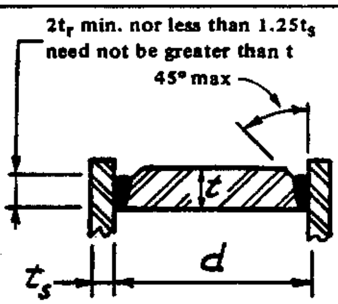
$$t = d \times \sqrt{CP/SE}$$

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

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

E - 832   1.8 M. VACUUM WINDOW   92

$d = 93.750"$

$t_s = 0.375"$ ACTUAL WALL THICKNESS

CHECKING THE MINIMUM SHELL WALL THICKNESS

EXTERNAL PRESSURE

FORMULAS

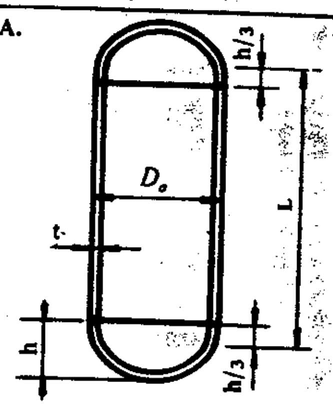
NOTATION

P = External design pressure, psig.
P_a = Maximum allowable working pressure, psig.
D_o = Outside diameter, in.
L = Length of a vessel section, taken as the largest of the following: inches (see figures A and B)

1. Distance between the tangent lines of the heads plus one third of the depth of the heads if stiffening rings are not used.
2. The greatest distance between any two adjacent stiffener rings.
3. The distance from the center of the first stiffening ring to the head tangent line plus one third of the depth of the head.
4. The distance from the first stiffening ring in the cylinder to the cone-to-cylinder junction.

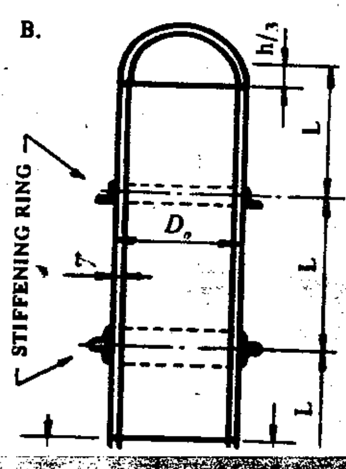
t = Minimum required wall thickness, in.

A.



VESSEL
WITHOUT STIFFENING RING

B.



CYLINDRICAL SHELL
Seamless or with Longitudinal Butt Joints

When D_o/t equal to or greater than 10 the maximum allowable pressure:

$$P_a = \frac{4B}{3(D_o/t)}$$

The value of B shall be determined by the following procedure:

1. Assume a value for t; (See pages 47-49) Determine L/D_o and D_o/t
2. Enter Fig. UGO-28.0 (Page 40) at the value of L/D_o. Enter at 50 when L/D_o is greater than 50, and at 0.05 when L/D_o is less than 0.05.
3. Move horizontally to the line representing D_o/t. From the point of intersection move vertically to determine the value of factor A.
4. Enter the applicable material chart (pages 41-45) at the value of A. Move vertically to the applicable temperature line*.
5. From the intersection move horizontally and read the value of B.

Compute the maximum allowable working pressure, P_a.

If the maximum allowable working pressure is smaller than the design pressure, the design procedure must be repeated increasing the vessel thickness or decreasing L by stiffening ring.

*For values of A falling to the left of the applicable temperature line, the value of P_a can be calculated by the formula:

100
10000
10
10000

ASSUMED $t = 0.20''$

$R = 46.875''$

$L = 36''$

$$\frac{L}{D_o} = \frac{36}{94.5} = 0.38$$

$$\frac{D_o}{t} = \frac{94.5}{0.20} = 472.5$$

FACTOR "A" FROM PAGE 40 [REF.]

$$A = \frac{3.8}{10\,000}$$

$$A = 0.00038$$

FACTOR "B" FROM PAGE 41

$$B = 5400$$

$$P_a = \frac{4B}{3\left(\frac{D_o}{t}\right)}$$

$$P_a = \frac{4(5400)}{3(472.5)} = 15.233 \text{ psi}$$

O.K.

THE MIN. WALL THICKNESS

$t_r = 0.2''$

$$\frac{t_r}{t_s} = \frac{0.2}{0.375''} = 0.533$$

$C = 0.33(0.533)$

$C = \underline{0.175}$

THE REQUIRED HEAD THICKNESS

$$t = d\sqrt{\frac{CP}{SE}}$$

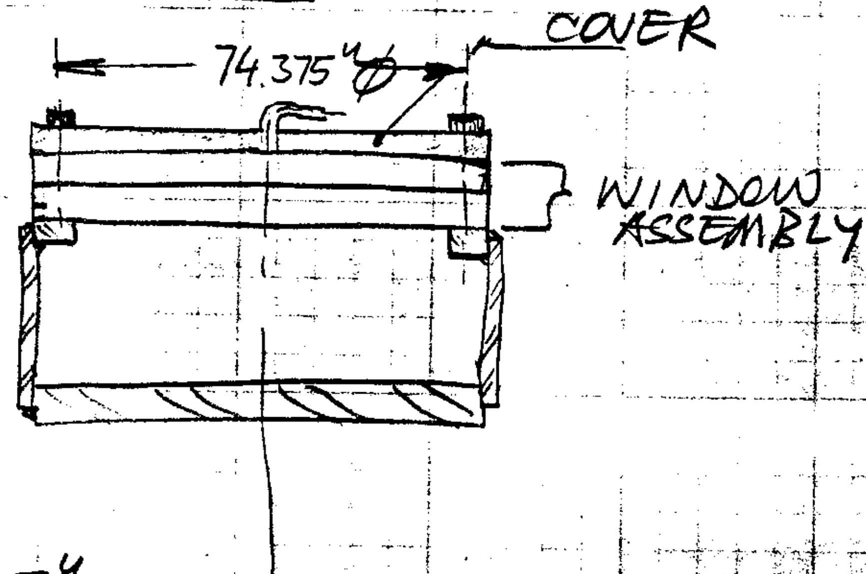
$$t = 93.75\sqrt{\frac{0.175\,(35)}{(12700)\,(0.55)}}$$

$t = \underline{1.84''}$

2" THICKNESS OF THE PLATE IS SELECTED

17.

CHECKING THE REQUIRED WALL THICKNESS OF A COVER PLATE FOR VACUUM / PRESSURE TYPE ARRANGEMENT.



$r_o = \underline{0.25"}$

$q = \underline{15}$ psi

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

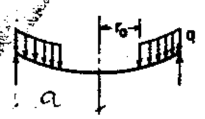
$a = \underline{37.187}$

$$L_{14} = \frac{1}{16}\left[1 - \left(\frac{r_o}{a}\right)^4 - 4\left(\frac{r_o}{a}\right)^2 \ln\frac{a}{r_o}\right]$$

$$L_{14} = \frac{1}{16}\left[1 - \left(\frac{0.25}{37.187}\right)^4 - 4\left(\frac{0.25}{37.187}\right)^2 \ln\frac{37.187}{0.25}\right]$$

$$L_{14} = \frac{1}{16}\left[0.9999 - 0.000904323\right]$$

$$L_{14} = \underline{0.06244}$$

$LT_\theta = \frac{-wr^2}{D}G_9$ $LT_M = -wrG_9$ $LT_Q = \frac{-wr_o}{r}\langle r - r_o\rangle^0$	9b. Fixed	$y_c = \frac{-wa^3}{2D}(L_6 - 2L_3)$ $M_c = wa(1+\nu)L_6$ $M_{ra} = \frac{-wr_o}{2a^2}(a^2 - r_o^2)$ $y_a = 0 \quad \theta_a = 0$	see table 9b below
10. Uniformly distributed load from r_o to a  $LT_y = \frac{-qr^4}{D}G_{11}$ $LT_\theta = \frac{-qr^3}{D}G_{14}$ $LT_M = -qr^2G_{17}$ $LT_Q = \frac{-q}{2r}(r^2 - r_o^2)\langle r - r_o\rangle^0$	10a. Simply supported	$y_a = 0 \quad M_{ra} = 0$ $y_c = \frac{-qa^4}{2D}\left(\frac{L_{17}}{1+\nu} - 2L_{11}\right)$ $M_c = qa^2L_{17}$ $\theta_a = \frac{q}{8Da(1+\nu)}(a^2 - r_o^2)^2$ $Q_a = \frac{-q}{2a}(a^2 - r_o^2)$	see table 10a below
	10b. Fixed	$y_c = \frac{-qa^4}{2D}(L_{14} - 2L_{11})$ $M_c = qa^2(1+\nu)L_{14}$ $M_{ra} = \frac{-q}{8a^2}(a^2 - r_o^2)^2$	see table 10b below

9b:

r_o/a	0.2	0.4	5 0.6	0.8
K_{y_c}	−0.02078	−0.02734	−0.02042	−0.00744
K_{M_c}	0.14683	0.12904	0.07442	0.02243
$K_{M_{ra}}$	−0.09600	−0.16800	−0.19200	−0.14400

(*Note:* If r_o approaches 0, see case 17)

10a:

$y = K_y\frac{qa^4}{D} \quad \theta = K_\theta\frac{qa^3}{D} \quad M = K_Mqa^2$

r_o/a	0.0	0.2	0.4	0.6	0.8
K_{y_c}	−0.06370	−0.05767	−0.04221	−0.02303	−0.00677
K_{θ_a}	0.09615	0.08862	0.06785	0.03939	0.01246
K_{M_c}	0.20625	0.17540	0.11972	0.06215	0.01776

Note: If $r_o = 0$, $G_{11} = \frac{1}{64}$ $G_{14} = \frac{1}{16}$ $G_{17} = \frac{(3+\nu)}{16}$

$y_c = \frac{-qa^4(5+\nu)}{64D(1+\nu)}$ $M_c = \frac{qa^2(3+\nu)}{16}$ $\theta_a = \frac{qa^3}{8D(1+\nu)}$

10b:

r_o/a	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
K_{y_c}	−0.01563	−0.01336	−0.00829	−0.00334	−0.00054
K_{M_c}	0.08125	0.06020	0.03152	0.01095	0.00156
$K_{M_{ra}}$	−0.12500	−0.11520	−0.08820	−0.05120	−0.01620

Note: If $r_o = 0$, $G_{11} = \frac{1}{64}$ $G_{14} = \frac{1}{16}$ $G_{17} = \frac{(3+\nu)}{16}$

$y_c = \frac{-qa^4}{64D}$ $M_c = \frac{qa^2(1+\nu)}{16}$ [illegible]

$$M_c = 15(37.187)^2(1+0.3)\,0.06244$$

$$M_c = \underline{1683.75} \text{ lb in}$$

$$M_{ra} = \frac{-(15)}{8(37.187)^2}\left(37.187^2 - 0.25^2\right)^2$$

$$M_{ra} = 0.001355873\,(1912164.793)$$

$$M_{ra} = \underline{2592.65} \text{ lb in}$$

$$\sigma = \frac{6M}{t^2}$$

$$12000 = \frac{6(M_{max})}{t^2}$$

$$t = \sqrt{\frac{6M}{12000}}$$

$$t = \underline{1.13''}$$

$t = \underline{1\tfrac{1}{2}}$ IS SELECTED

INTERNAL PRESSURE

$$P = \frac{S E t}{R - 0.4 t}$$

E → FROM
page 145
E. MEGYESY
PRESSURE VESSEL
H. B.
7 EDITION

$$P = \frac{12000(0.6)(0.375)}{35.5 - 0.4(0.375)}$$

$$P = \frac{2700}{35.35}$$

$$P = 76.37 \text{ psi } [\text{max.}]$$