Remarks on Configuration, Assembly and Stringing of the BABAR Drift Chamber

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1. General

2. Specific to the Inner and Outer Cylinders

Survey of Large Cylindrical Drift Chambers

Table 1. Survey of Large Cylindrical Drift Chambers

	JADE	Mark II	TASSO	CLEO	ARGUS	Benichou	VENUS	Mark III	OPAL	SLD	CLEO II	Mark II	CDF	KEDR	AMY	ZEUS	BES	KLOE
Reference	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Chamber size																		
R_min (mm)		373	320	172	150	400	250	140	250	200	175	192	277	125		162	155	~200
R_max (mm)	800	1509	1280	950	859	1700	1260	1140	1850	1000	945	1519	1380	535	787	850	1150	1000
L_max (mm)	3000	2692	3520	1930	2000	4500	3000	2340	4000	1800	2150	2300	3214	100	1792	2030	2120	3300
Wire/cell parameters																		
Sense wires	1536	3204	2340	5304	5940	12880	7104	528	3816	5120	12240	5832	6156	1512	9048	4608	2808	~12000
Other wires		9612	7020	15912	24588	64400	21312	1584	?	32640	36240	?	30448	14520	27144	19584	16572	~36000
Total layers	48	16	15	17	36	40	29	62	159	80	51	72	72	56	40	72	40	~48
Stereo layers		10	6	8	18	9	9	10		48	11	36	36	24	15	32	20	~24
R_min (mm)		414	367	213	180		286	175		238	199	246	309		155	182	195	
R_max (mm)		1448	1222	892			1213			961	901	1448	1320		639	794	1095	
Geometry	jet	square	square	square	square	hex	square	rect.	jet	jet	square	jet	jet	jet	hex	jet	jet	square
Max drift (mm)	70	18	16	6	18	10	10	30	250	30	7		40	30	6	25	31	15
Min. resolution (micron)	150	150	220	210	150		150	250	110	55	100	125	200	40	140	100	200	~100
dE/dx resolution (%)	10				5.6			15	3.5	7	7.1	7.2				6	9	
Endplates																		
Material	Rohacell	Al + H-C	Al	Al	Al	Al	Al	Al + H-C	Al	Al	Al	Al	Al	Gl-F	Al	Al	Al	Al or C-F
Thickness (mm)		76	35		30	5	21	76	28	5.1	31.8	50.8	50.8	21	30	20	40	
Geometry	flat + rib	flat + cone	flat	flat	flat	cone	sphere	flat	cone	parabola	flat	flat	flat	flat	stepped	flat	flat	sphere
Wire load (tonnes)	1.2		2.3		3.1	33.7	6.8		14	13.3	6.8	20	25	3.4	4.5	5.9		
Max deflection (mm)		8				3	0.6		0.5		7.9		1.4	3		*1.8*		~1
Outer cylinder																		
Material		Al	Al		Al	H-C	C-F	Al	Al	Al + H-C	Al + H-C	Al	Al	Gl-F	Al	Al	Al	rods
Thickness		6	6		6	25	5	6			8	12.7	6.4	5	posts	6	10	
Inner cylinder																		
Material		Lexan	Gl-F		C-F	foil	C-F	paper	foil	Al + H-C	C-F	Be	C-F	C-F	Kevlar	Al + foam	C-F	foil
Thickness		3	5		3.3		1	1			0.75	2	2	1.5	1	1.4 + 9	2	
Load bearing		no	yes			no	no	no	no	yes	no		yes		no	yes		no
Stringing		horiz.									vert.		horiz.			vert.		
Prestressing		rods								rods	rods		wires	external	rods	rods		
C-F - carbon-fiber/epoxy, Gl-F = glass-fiber/epoxy, H-C = honeycomb																		

$\Rightarrow$ many variations of drift-chamber concept can be built.

Changes in the BABAR Environment

Forward aerogel system will not be implemented

$\Rightarrow$ 20 cm more in z available for tracks with highest momenta.

PEP-II support tube now at least 1.2 X_0 thick

$\Rightarrow$ loss of precision linking between SVT and drift chamber. $\Rightarrow$ slight physics cost in making the drift chamber inner tube thick enough to be load bearing.

Additional resources available to drift chamber project.

$\Rightarrow$ Timely to review status of drift chamber.

Since the drift chamber may already be somewhat behind schedule, changes that incur delays are to be discouraged.

[Conversely, changes that simplify fabrication and reduce risk would tend to speed up construction and should be considered.]

Using the Space Vacated by the Aerogel

We should plan now to use this space.

1. Build chamber as presently planned, but slide it forward 20 cm.

2. Build chamber to present design but 20 cm longer.

3. Reconfigure forward endplate.

– Biconical design is structurally agressive ($\Rightarrow$ fabrication delays likely in my opinion).

– Wire load is $\approx 1/6$ atm. inward $\Rightarrow$ chamber is kind of a vacuum vessel.

– Favored mechancial structure of endplate is inwardly curved surface (as used in VENUS and SLD chambers).

– Carbon-fiber layup favors shape that can be deformed into a plane without stretching $\Rightarrow$ inward cone (as used in OPAL chamber).

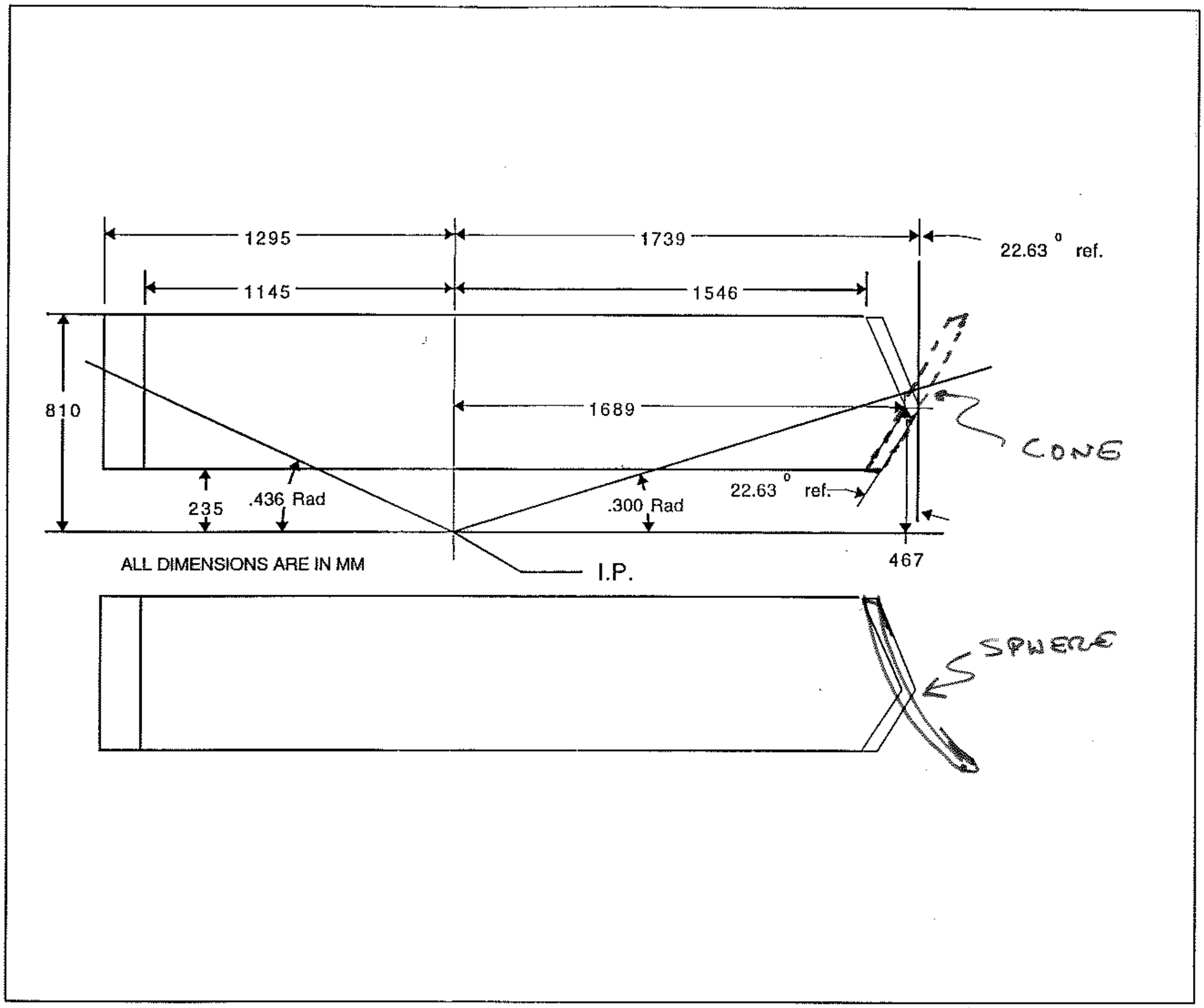


Figure 2: r-z dimensions of the BABAR drift chamber.

Civil Engineering Department:

J.-H. Prévost, author of FEA program DYNAFLOW on UNIX machines.

D.B. Billington, expert on domes; book: *Thin Shell Concrete Structures.*

THIN SHELL CONCRETE STRUCTURES

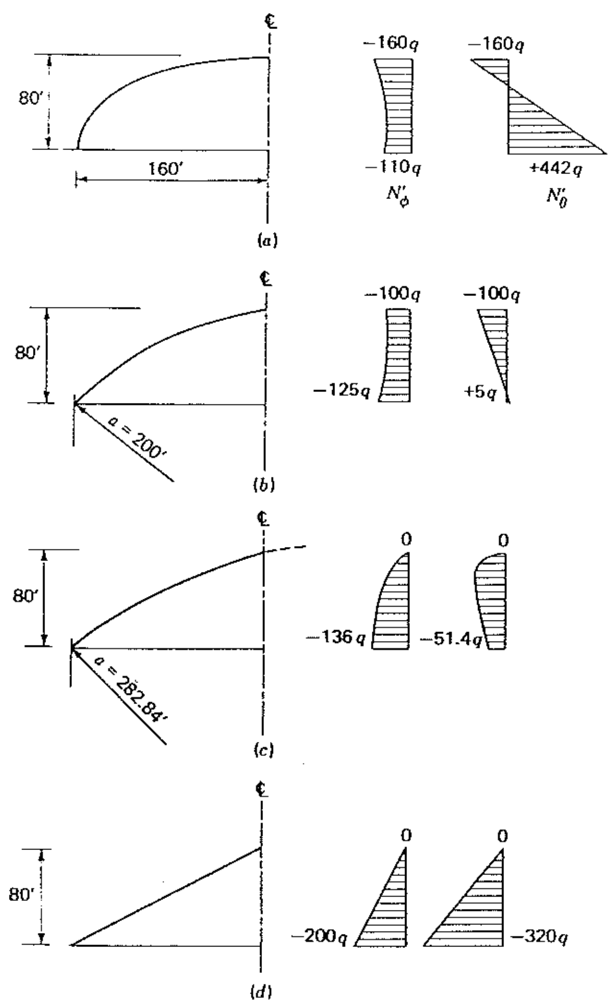


Fig. 3-12 Membrane stress resultants on various domes. (*a*) Elliptical dome. (*b*) Spherical dome. (*c*) Conoidal dome. (*d*) Conical dome.

THIN SHELL CONCRETE STRUCTURES

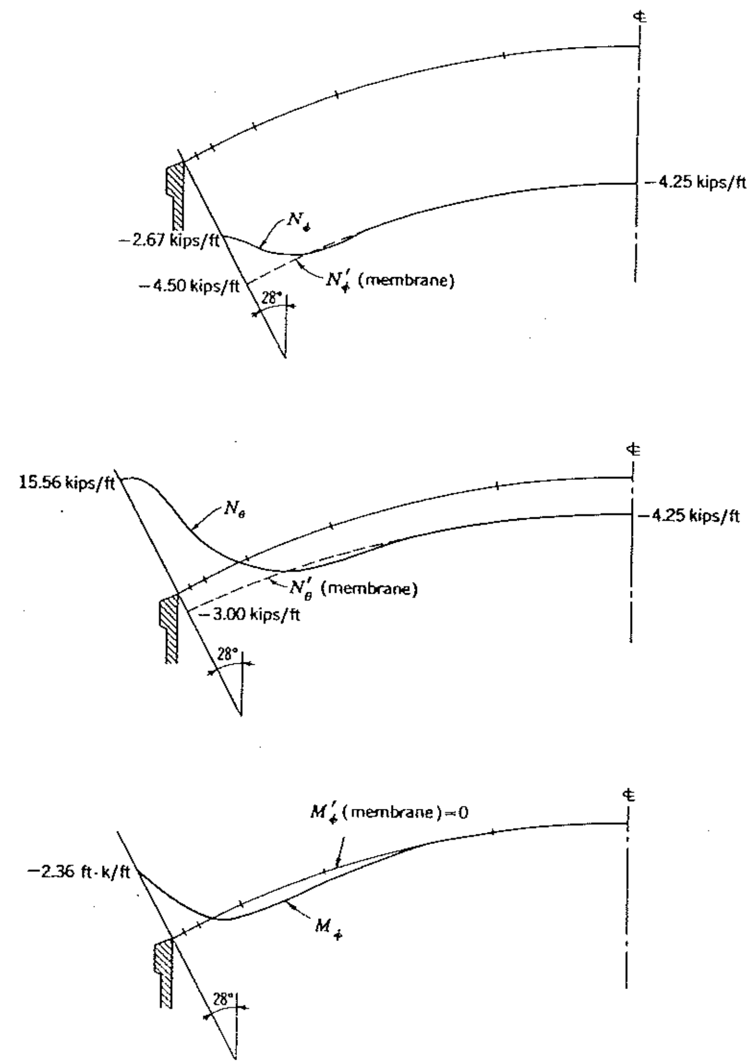


Fig. 4-15 (*a*) Dome-ring-wall system under dome loads.

I strongly encourage a backup solution to the foward endplate based on an inward cone or sphere – possibly made of aluminum.

[3rd of 3 SLD Al blanks available: 2m $\times$ 2m $\times$20 cm: "When in doubt, hog it out".]

The Inner Cylinder

(General remarks, details later)

PEP-II structural concerns and rf shielding issues will likely add, not reduce, mass of the support tube.

A load-bearing inner cylinder could reduce endplate axial deformation under wire load to $\approx 100\ \mu$, and radial deformation to $\approx 50\ \mu$m.

10 x safety factor against buckling $\Rightarrow$ inner cylinder of be 1.5 mm carbon fiber (if modulus of $1.4 \times 10^{11} = 2E_{\rm al}$); $\Rightarrow 0.5\%\ X_0$.

[TASSO, SLD, CDF and ZEUS chambers have load-bearing inner cylinders.]

Outer cylinder need be only 1.6 mm thick carbon fiber (total wall thickness less than present design).

Benefits If Chamber Is Assembled Before Stringing

Endplate-cylinder joints could be leak tested and load tested (and repaired) prior to stringing.

Prestressing of endplate not needed.

Chamber would be strung vertically,

$\Rightarrow$ No inside robots or inside wire handling needed.

Semiautomatic wire feeding as in CLEO II and ZEUS.

No need to rotate chamber about horizontal axis for wire repair.

Costs

Only one crew can string stereo wires $\Rightarrow$ longer stringing time.

Tall cleanroom needed (but needed anyway for vertical wire repair).

Inner Cylinder Calculations

(see note of 2/7/96 by KTM, on DC Web Page)

Analytic calculations available for flat, annular plates,

(*Roark's Formulas*, by Young, 6th ed., McGraw-Hill, 1989).

Implemented in PC program *TK Solver*.

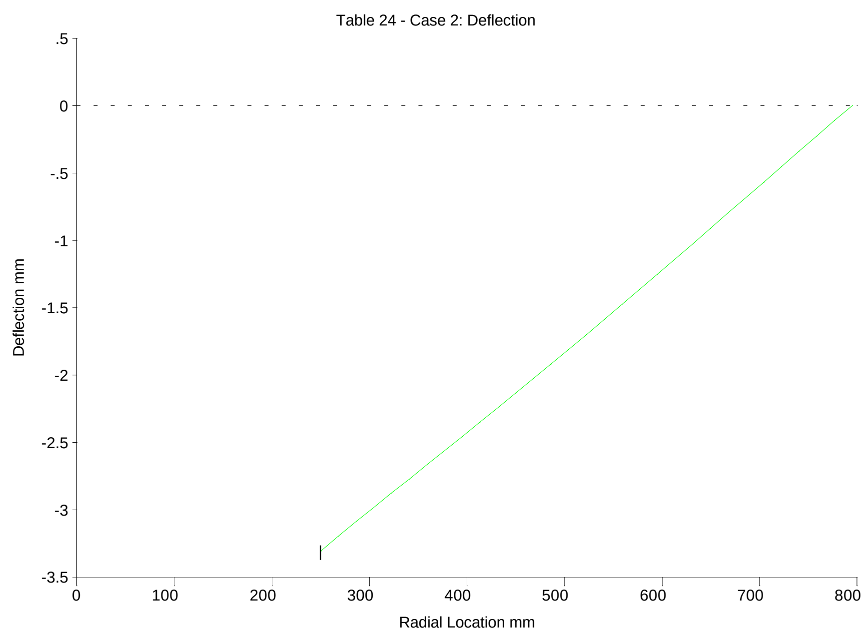
Flat Al plate, 3.2 cm thick, $r_1 = 25$ cm, $r_2 = 80$ cm.

Modulus reduced 15% for hole.

When supported at outer radius only,

max deflection = 3.3 mm under 3500 kg wire load

(deflection angle = 6 mrad).



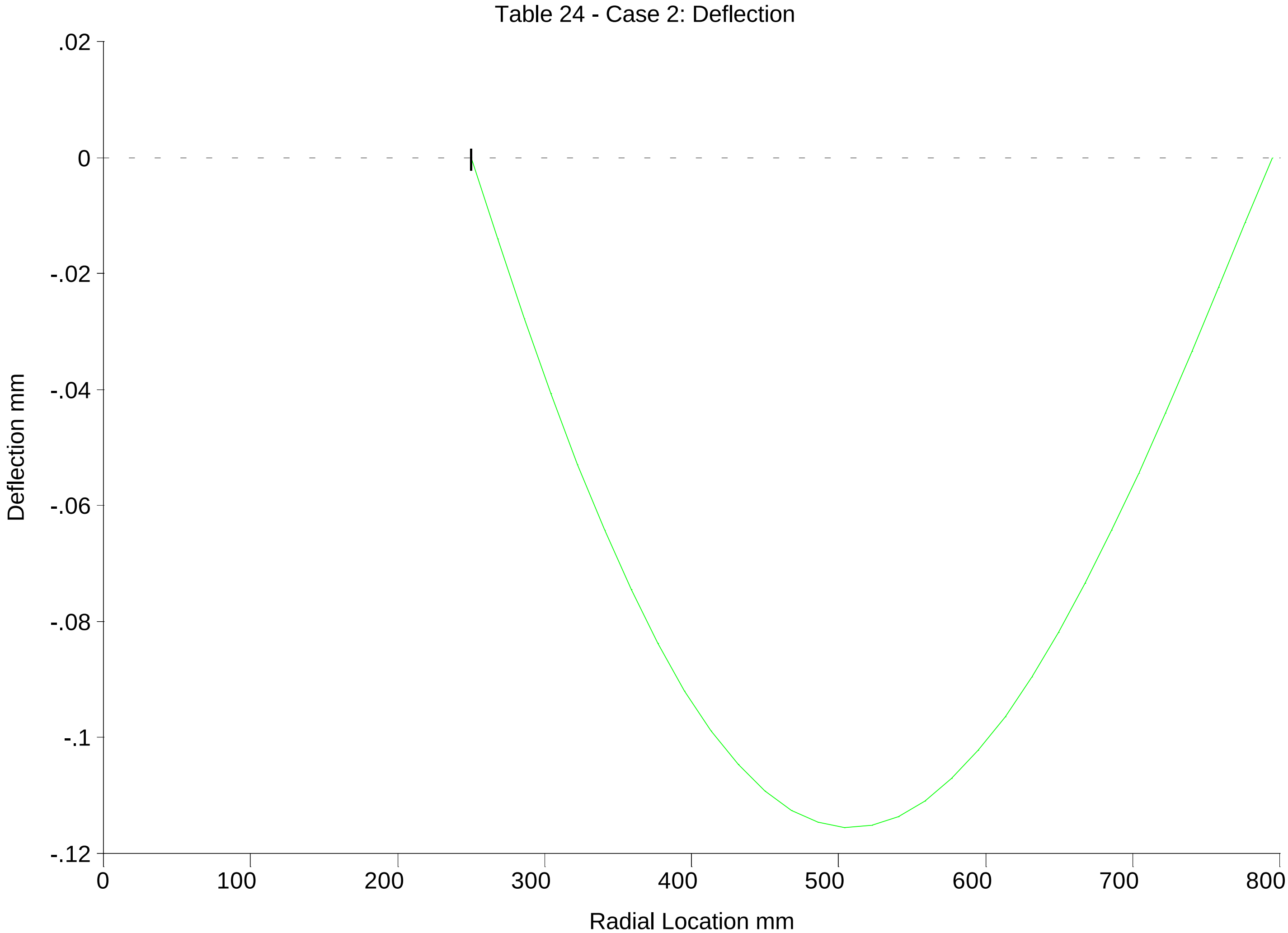
St	Input	Name	Output	Unit	Comment
					BABAR rear endplate, supported at outer edge
					Table 24 - Roark & Young (6 ed) [file tab24.tk]
					Formulas for flat circular plates of [mod by ktm]
					constant thickness [file 242.tk also mod]
		case	'Case_2a		Reference number
	4	matnum			Material Number (See Material Table)
		matl	"Aluminum		Material name
	5.857E10	E		Pa	Young's Modulus (reduced 15% for holes)
		nu	.3		Poisson's ratio
	.795	a		m	Outer Radius
	.25	b		m	Inner Radius
		Area	1.7892156	m^2	Area of plate
	.032	t		m	Plate Thickness
	3500	L		kgf	Load on plate
		q	19170.412	Pa	Uniformly distributed pressure
		D	175752.91	N-m	Plate Constant + E*t^3/12/(1-nu^2)
					AT OUTER EDGE:
		ya	0	m	Deflection
		tha	.0061928	rad	Radial Slope Angle
		Mra	0	N-m/m	Radial Bending Moment
		Qa	-6866.685	N/m	Shear Force Density
		La	-3500	kgf	Shear Force
					AT INNER EDGE:
		yb	-.0033073	m	Deflection
		thb	.0060856	rad	Radial Slope Angle
		Mrb	0	N-m/m	Radial Bending Moment
		Qb	0	N/m	Shear Force Density
		Lb	0	kgf	Shear Force

Flat Plate, Supported Both Edges

3500 kg load distributed 1400 kg on inner cylinder,

2100 kg on outer.

Max deflection = 115 μm (max angle = 0.8 mrad).



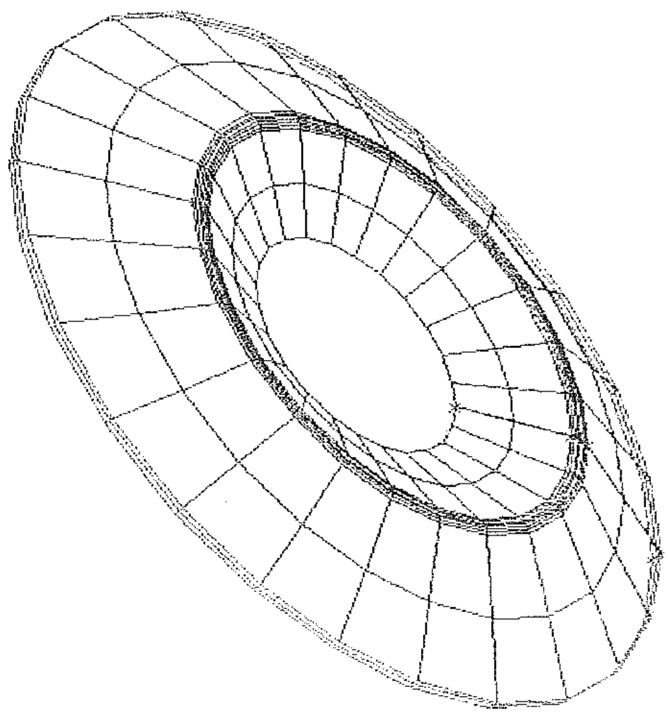
St	Input	Name	Output	Unit	Comment
					BABAR rear endplate, supported both edges
					Table 24 - Roark & Young (6 ed) [file tab24.tk]
					Formulas for flat circular plates of [mod by ktm]
					constant thickness [file 242.tk also mod]
		case	'Case_2c		Reference number
	4	matnum			Material Number (See Material Table)
		matl	"Aluminum		Material name
	5.857E10	E		Pa	Young's Modulus (reduced 15% for holes)
		nu	.3		Poisson's ratio
	.795	a		m	Outer Radius
	.25	b		m	Inner Radius
		Area	1.7892156	m^2	Area of plate
	.032	t		m	Plate Thickness
	3500	L		kgf	Load on plate
		q	19170.412	Pa	Uniformly distributed pressure
		D	175752.91	N-m	Plate Constant + E*t^3/12/(1-nu^2)
					AT OUTER EDGE:
		ya	0	m	Deflection
		tha	6.1746E-4	rad	Radial Slope Angle
		Mra	0	N-m/m	Radial Bending Moment
		Qa	-4127.383	N/m	Shear Force Density
		La	-2103.758	kgf	Shear Force
					AT INNER EDGE:
		yb	0	m	Deflection
		thb	-.0007872	rad	Radial Slope Angle
		Mrb	0	N-m/m	Radial Bending Moment
		Qb	8710.98	N/m	Shear Force Density
		Lb	1396.2424	kgf	Shear Force

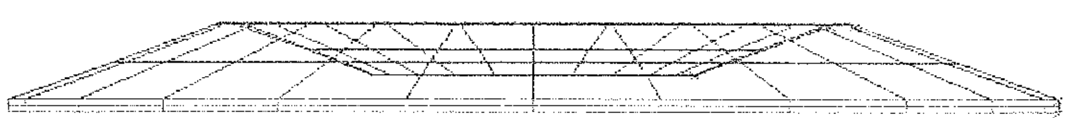
FEA Studies of Front Endplate

By C. Lu, using ALGOR running on a Pentium PC.

Factor of 8 reduction in axial deformation when endplate supported both at inner and outer radii.

Details in another talk.





Buckling of Thin Cylindrical Shells

$$F_{\max} = \frac{4\pi^2 EI}{l^2}, \qquad \text{(Euler)}$$

For a tube, the momentum of inertia is $I = \pi r^3 t$,

$$F_{\max} = \frac{4\pi^3 E r^3 t}{l^2}, \qquad \text{(tube)}.$$

Thin tubes buckle into a higher-order mode:

$$F_{\max} = \frac{2\pi E t^2}{\sqrt{3(1-\nu^2)}}, \qquad \text{(Timoshenko,1910)},$$

Short wavelength $\Rightarrow$ no dependence on r or l.

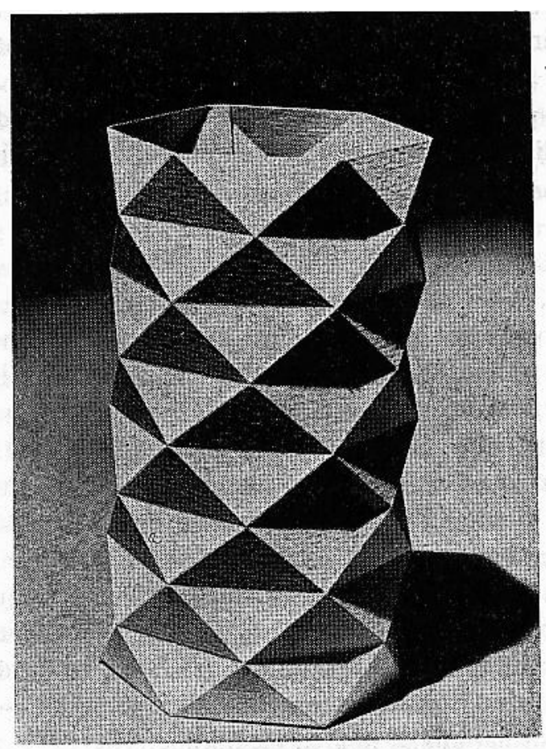


Fig. 2.6. The Yoshimura-pattern

Summary of Buckling Data

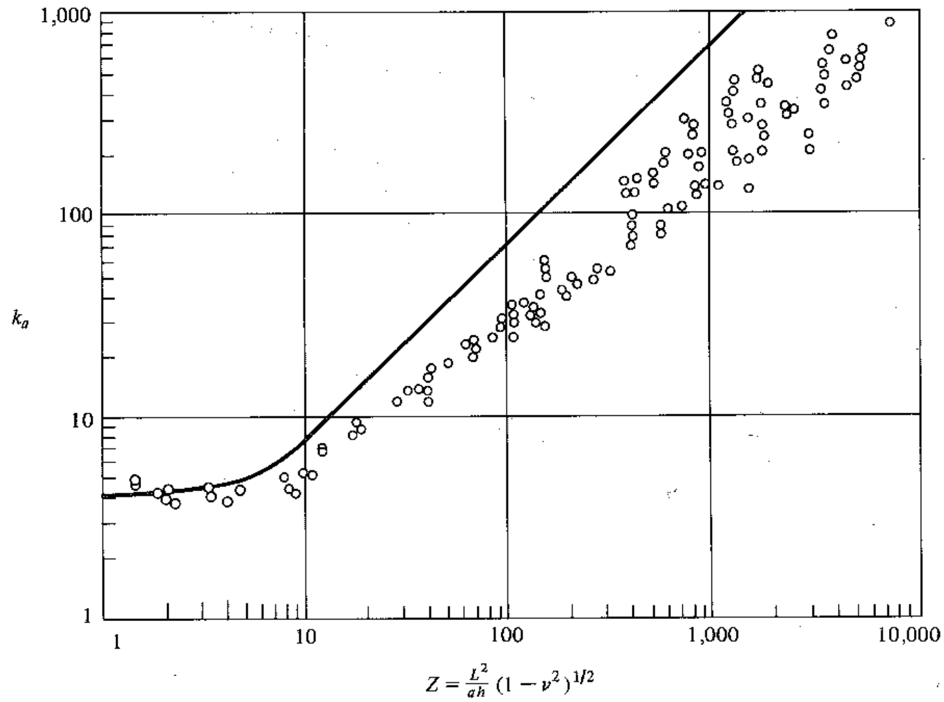


FIGURE 5.14
Comparison of theoretical and experimental values for cylinders subjected to axial compression.

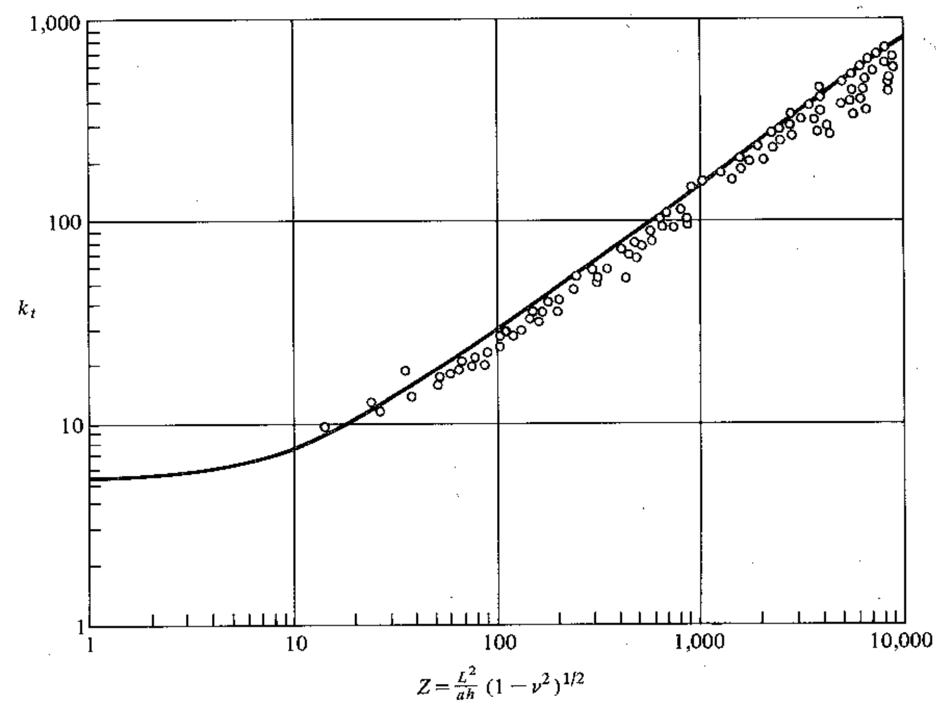


FIGURE 5.15
Comparison of theoretical and experimental values for cylinders subjected to torsion.

Semi-Empirical Fit to Buckling Data

$$F_{\max} = \frac{\pi^3 E t^{9/4} r^{1/4}}{6 l^{1/2} (1-\nu^2)^{5/8}}, \qquad \text{(semi-empirical)},$$

which is a translation of the fit $k_a = Z^{3/4}$ to the data on both axial and torsional buckling.

Laboratory Test

Cylinder of radius 6″ and length 36″ from a sheet of G-10 about 0.018″ = 450 μm thick.

The seam of the tube consisted of a 1″ overlap secured with pop rivets every inch.

A data sheet lists the modulus of G-10 as 2.5×10^6 psi.

We assume the Poisson ratio is 0.3. The calculated buckling force is then 425 pounds.

G-10 Tube Buckled Under 715-Lb Load

$\Rightarrow$ Semi-empirical formula gives a lower bound on the buckling force.