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Considerations Regarding Tube Wall Material

Steven P. Ahlen, Alex Marin, Bing Zhou
Boston University

March 1992

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We discuss several options for the PDT tube wall materials in our proposed pressure and temperature conditions. Physical properties of those options are presented, which, together with costs considerations, lead us to the conclusion that Aluminum PDTs should be the best choice.

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PDT-BU-92-5

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In selecting the wall material for the PDT's there is a trade-off between mechanical robustness and radiation length material thickness. This is seen in the table below where various parameters of 4 types of tubes are compared: type 304 wrought stainless steel (SS), type 2024-T4 aluminum (Al), polycarbonate plastic (PC), and P100 carbon fiber (CF). Each of the tubes has an assumed length of 4 meters, a radius of 2 cm, and 0.17% of a radiation length per wall (for an average of 4.3% of a radiation length for 8 layers of tubes).

Property	Unit	SS	Al	PC	CF
Wall Thickness	mils	1.15	6.0	22	13.3
Yield Stress	kpsi	42	47	8	325
Elastic Modulus	Mpsi	28	10.6	0.31	100
Radiation Length	cm	1.72	8.9	33	19.9
Density	g/cm^3	8.04	2.77	1.20	2.15
Poisson's Ratio		0.283	0.334	0.4	0.3
Weight/tube	lb	0.26	0.47	0.74	0.8
Maximum Gravitational Deflection at 0 psig	mm	1.35	1.23	18.2	0.10
Hoop Stress/Yield Stress at 29.4 psig	%	48	8.2	13.2	0.53
Buckling Pressure for Length = 4 m	psig	-0.032	-1.78	-2.70	-177
Buckling Pressure for Length = 12 cm	psig	-0.30	-8.5	-7.2	-590

The deflection calculation assumes a single tube clamped at each end with no excess pressure within the tube. Fig. 1 shows the formula used to calculate sag. The figure also indicates how the deflection is reduced if tubes are bonded in clusters of 3 or 6 close packed configurations.

Note that the hoop stress calculation assumes an overpressure of 2 atm within the tubes.

The results of buckling pressure calculations for the 4 tube types above are shown in Fig. 2 to 5, as a function of number of lobes in the buckling profile, and the length to radius ratio of the tube.

The deflection properties are changed if a tube is pressurized. Consider a tube fixed at one end, supported at the other, and subject to an internal pressure p. The deflection is obtained by multiplying the formula for D in Fig. 1 by a correction factor to get the deflection D*:

$$D^* = 0.0054 \times 384 \times D \times F(\varepsilon, r), \text{ where}$$

$$\varepsilon = Lp/(t\,R\,E) \text{ (see Fig. 1)}$$

$$r = x/L = \text{fractional position along tube}$$
(x = 0 is the supported end, and x = L is the clamped end)

F is a messy function which reduces to 1 in the limit
$\varepsilon = 0$ for $r = 0.4215$ (max. deflection point)

In the table below we give the deflection properties for the tubes considered before, this time for one end fixed, one end supported, with p = 29.4 psig:

Property	Unit	SS	Al	PC	CF
ε		5.36	3.81	11.65	0.834
r (maximum sag) = r_{max}		0.44	0.43	0.46	0.42
$F(\varepsilon, r_{max})$		0.42	0.588	0.135	0.97
D*	mm	1.18	1.51	5.11	0.21
D	mm	1.35	1.23	18.2	0.1
D*/F	mm	2.81	2.57	37.9	0.22

Note the 2nd to last row gives deflections for both ends fixed, with no internal pressure. The last row gives the deflection for one end fixed, with no internal pressure.

It will be important to design the tube support structure appropriately, to avoid overconstraining the tubes during pressurization (to avoid buildup of stresses and deformations and to permit reliable evaluation of mechanical stresses and strains to aid in alignment considerations). Perhaps this would best be accomplished by firmly fixing only one end, allowing the other end to move in the direction parallel to the tube).

We believe two of the four materials can be eliminated based on a comparison of the above properties:

(1) Stainless tubes of the required radiation length thickness are quite vulnerable to underpressure buckling (manifesting itself as a critical failure mode even at pressure differences as small as 0.9 inch of water, or as tube deformation through sloppy handling); they also have small safety margin for operation at over-pressure.

(2) Plastic tubes are not rigid enough (due to their large ratio of mass density to elastic modulus) to be handled reliably during installation or pressurization without extensive procedures to stabilize the tube arrays (for example by embedding large numbers of tubes in an adhesive matrix). The most disturbing aspect is the large change in deflection from zero pressure to operation at 29.4 psig (38 mm sag to 5 mm sag). The additional adhesive or

metallic structural elements required for stabilization would increase cost of design, testing and construction, would increase radiation length thickness, and would destroy the inherently simple nature of the design.

Mechanical considerations suggest carbon fiber as the preferred material. Reliable procedures for applying aluminized cathode surfaces could probably be devised, but their final cost would probably be excessive (estimates of stainless or aluminum tubes are in the several million dollar range for just the tubes for the GEM barrel; carbon fiber tubes would likely be an order of magnitude more expensive, which would be unacceptable).

Thus, our conclusion is that aluminum is the preferred tube material. In our opinion, the primary limitation of these is their vulnerability to permanent deformation if abused in handling. A measure of this is given by the 12 cm buckling pressure parameter quoted in the first table. This is the overpressure which must be applied uniformly over a 12 cm tube length that results in tube collapse. Of course "banging" the tube, or "pressing" on the tube would have different pressure specifications, but the 12 cm crush pressure is probably useful in comparing tubes of different types. Our group has had experience with making and using an array of drift tubes for a balloon experiment (EXAM) in which the type 304 stainless steel tubes were 3.7 m long, with a radius of 1.9 cm, and a wall thickness of 6 mils. The buckling characteristics of such a tube are shown in Fig. 6. The 12 cm buckling pressure is seen to be -25 psig, compared to -8.5 psig for a 6 mil wall Al tube with 2 cm radius. Our experience with the 1.9 cm radius stainless tubes is that a little care is all that is required to maintain the wall integrity, although sloppiness did result in walls occasionally being seriously "dinged". Greater care would be required with the 6 mil aluminum tubes, but probably not by a large amount. The crush pressure is quite sensitive to wall thickness. In Fig. 7 we show the properties for an aluminum tube with 12 mil wall thickness. The 12 cm crush pressure is -48 psig for this tube. By slightly compromising the radiation length, in going from 4.3% for 8 tubes to 6.6%, the crush pressure would go from -8.5 psig to -25 psig, the same as the stainless steel tubes used by us in the past. We are confident based on past experience that such tubes have strong enough walls to withstand reasonable handling.

In choosing tubes for the central drift tube (CDT) array for CDF, Steve Errede et al. (NIM **A268**, 92, 1988) selected cylindrical stainless steel tubes over tubes of other shapes and other materials. Their two reasons for rejecting aluminum were a propensity for this material to experience electrical discharges at large voltages (these were employed in limited streamer mode for CDF), and failure of the aluminum tubes to match the straightness achieved with stainless tubes. We feel that the former limitation is not a problem for pressurized tubes used in the proportional or limited proportional mode. A great deal of experience exists in our group and elsewhere with aluminized plastic wall tubes which have performed reliably under such gain conditions. The potential straightness problem needs to be investigated through interactions with manufacturers of aluminum tubes.

We conclude by summarizing information provided us by Brian Smith on the availability of metallic tubes. As of October 1991, Brian had identified one tube manufacturer who would make quotes on drift tube prices (Kilsby Roberts of Indianapolis, the same manufacturer who had provided the tubes to us for EXAM). According to Brian, standard thicknesses of down to 18 mils were available for aluminum tubes, with special orders for aluminum of 12 mils thickness, and 6 mils for stainless being possible). The prices (for 18 mil Al tubes and 6 mil stainless) were:

PRICES FOR LARGE ORDERS	SIZE	WALL THICKNESS	PRICE
Aluminum	1 inch OD	18 mils	$1.06/ft
Aluminum	1.5 inch OD	18 mils	$1.68/ft
Stainless Steel	1 inch OD	6 mils	$2.00/ft
Stainless Steel	1.5 inch OD	6 mils	$2.12/ft

In small orders (less than 1000 ft of tubes), the price for stainless tubes was $4.20/ft. For reference, the EXAM tubes cost $3.49/ft for 5000 ft of tubing in 1984.

The specifications for the stainless tubes were a straightness of 0.75 mm over 3 ft length (or 0.37 mm over 3 ft for 5% increase in cost), and a roundness of +/- 75 microns.

An advantage of the aluminum tubes is that they can be extruded and thus be made seamlessly, while the stainless tubes are wrapped, welded, and made nearly seamless inside by pulling a plug through the inside after they are made.

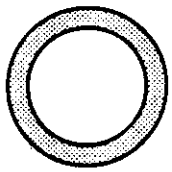
Mechanical Considerations for Pressurized Drift Tubes

Gravitational Sag:

Max Deflection = $D = WL^3/(384\,E\,I)$

W = load (assumed uniform)
L = tube length
E = elastic modulus
I = moment of inertia

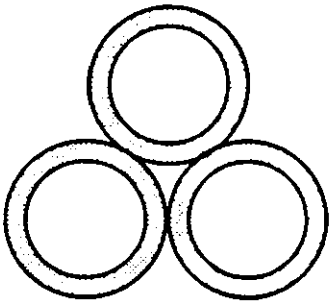
For a single thin walled tube:



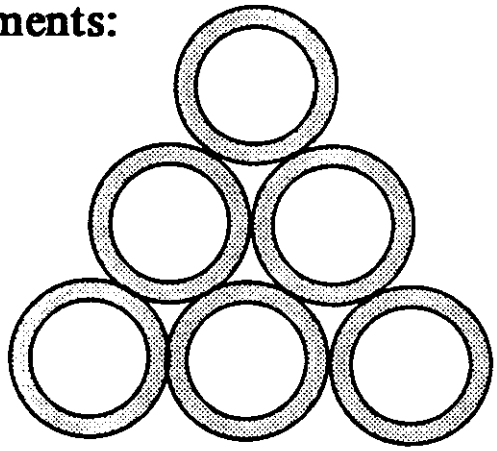
$$D = 4.7\text{ mm}\,(\rho/(\text{g/cm}^3))(L/4\text{ m})^4\,(\text{Mpsi}/E)\,(2\text{ cm}/R)^2$$

ρ = mass density of tube material
R = tube radius

For other, close-packed arrangements:



$$\text{Sag} = 3\,D/7$$



$$\text{Sag} = 6\,D/26$$

Stress From Internal Pressure:

$$\text{Hoop Stress} = R\,p/t$$

$$\text{Meridional Stress} = R\,p/(2\,t)$$

R = radius, p = gauge pressure, t = wall thickness

Fig. 1

Fig. 2

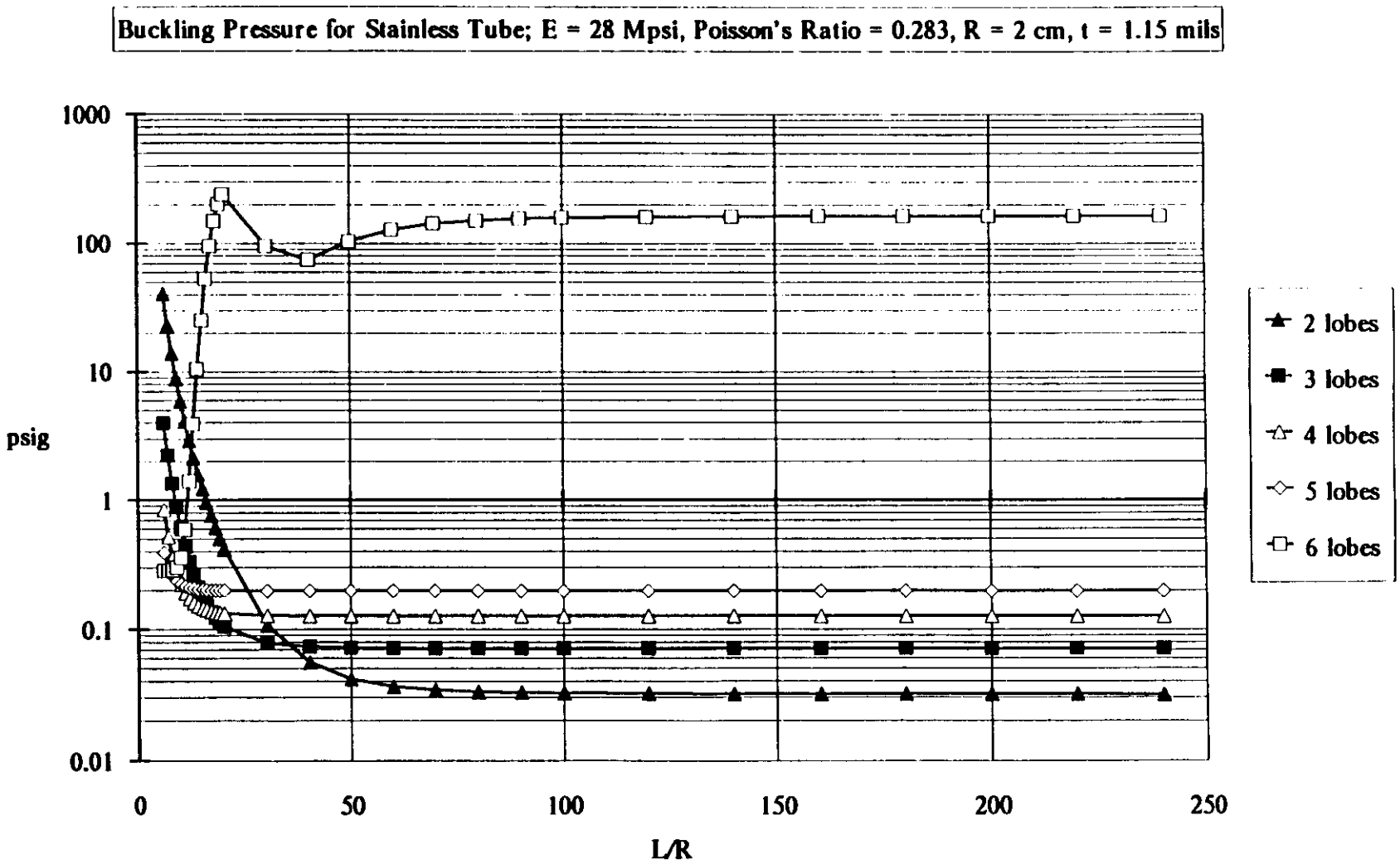

Buckling Pressure for Stainless Tube; E = 28 Mpsi, Poisson's Ratio = 0.283, R = 2 cm, t = 1.15 mils
1000
100
10
1
0.1
0.01
psig
0
50
100
150
200
250
L/R
2 lobes
3 lobes
4 lobes
5 lobes
6 lobes

Fig. 3

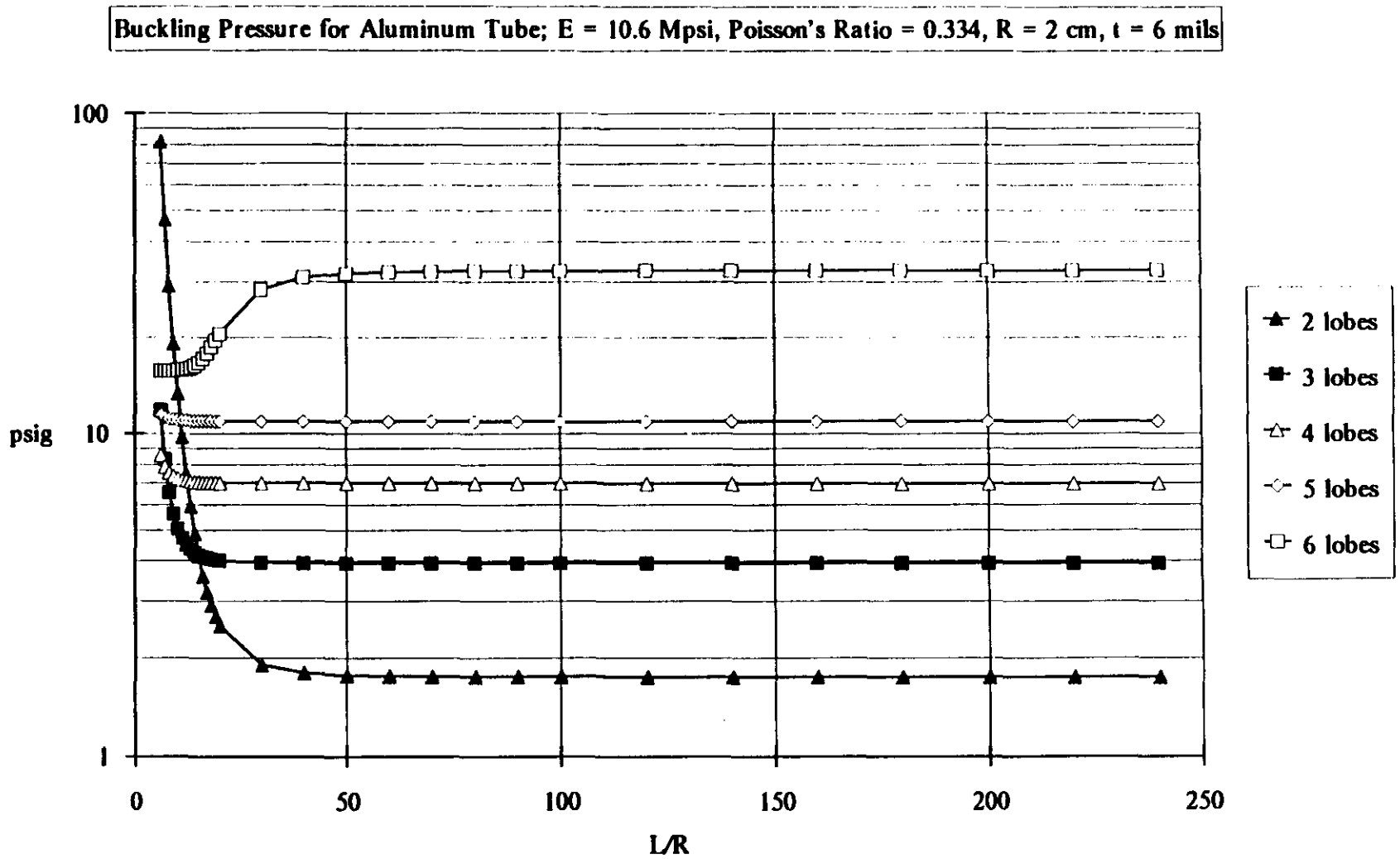

Buckling Pressure for Aluminum Tube; E = 10.6 Mpsi, Poisson's Ratio = 0.334, R = 2 cm, t = 6 mils
psig
100
10
1
0
50
100
150
200
250
L/R
2 lobes
3 lobes
4 lobes
5 lobes
6 lobes

Fig. 4

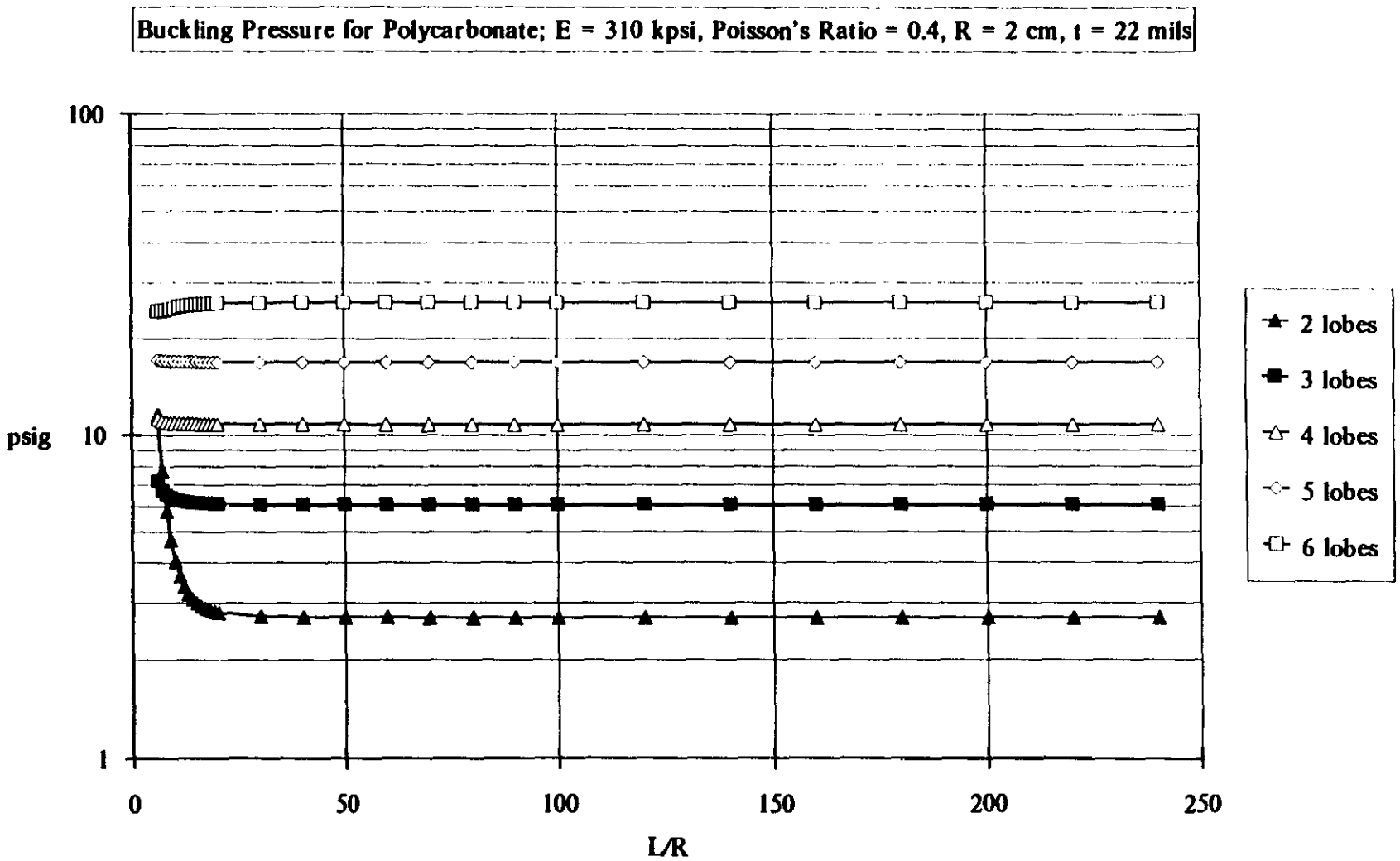

Buckling Pressure for Polycarbonate; E = 310 kpsi, Poisson's Ratio = 0.4, R = 2 cm, t = 22 mils
psig
100
10
1
0
50
100
150
200
250
L/R
2 lobes
3 lobes
4 lobes
5 lobes
6 lobes

Fig. 5

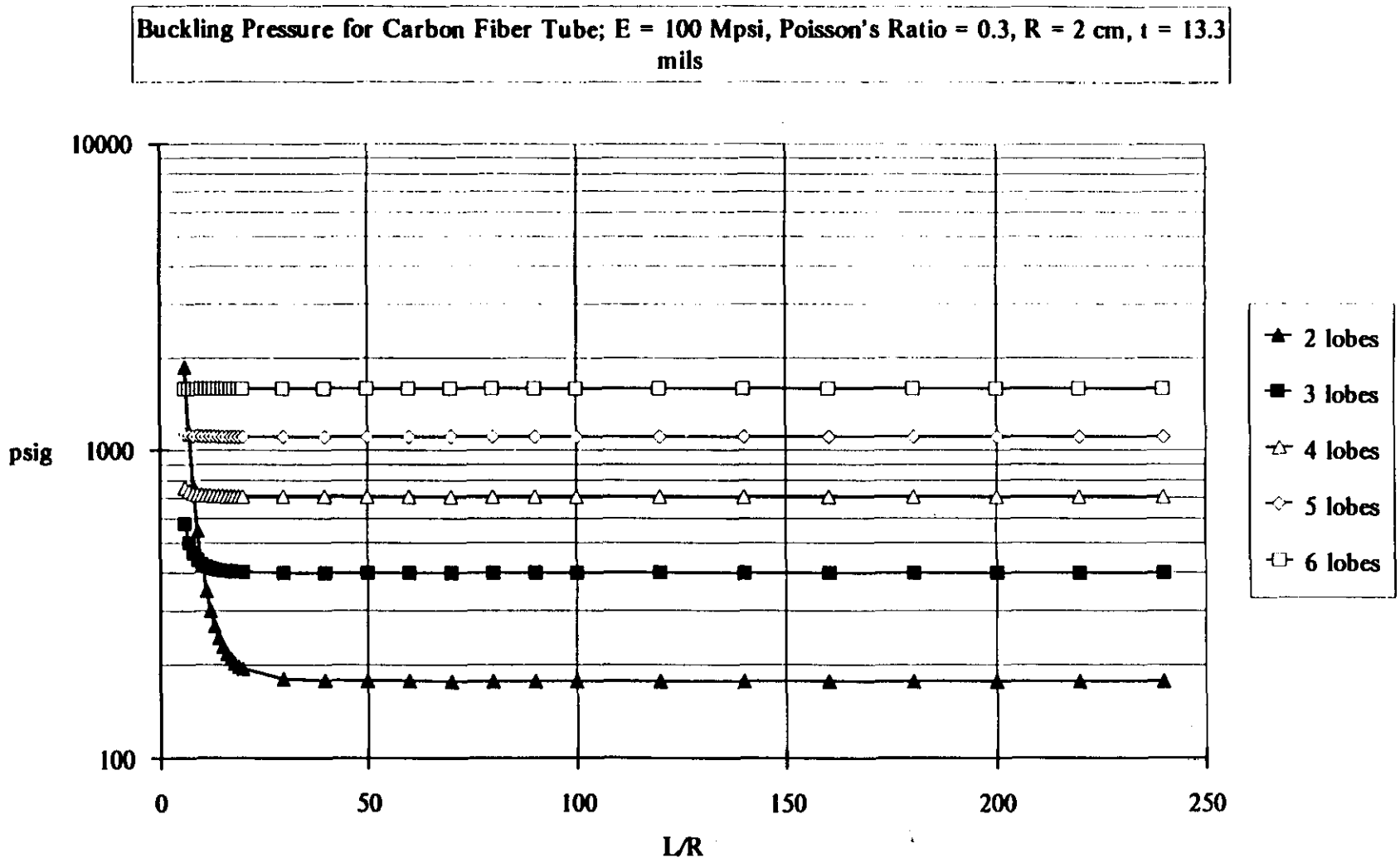
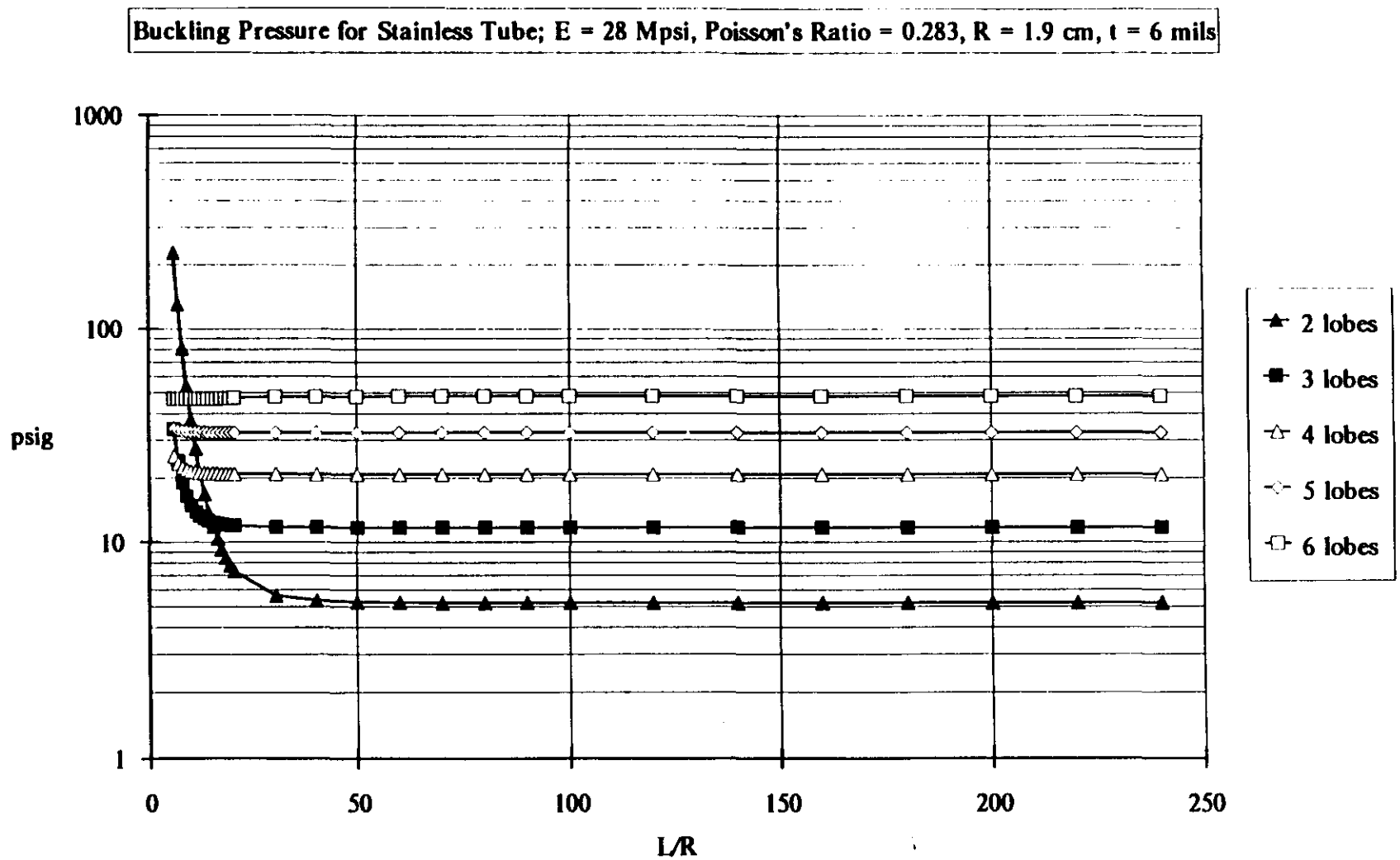

Buckling Pressure for Carbon Fiber Tube; E = 100 Mpsi, Poisson's Ratio = 0.3, R = 2 cm, t = 13.3 mils
psig
10000
1000
100
0
50
100
150
200
250
L/R
2 lobes
3 lobes
4 lobes
5 lobes
6 lobes

Fig. 6


Buckling Pressure for Stainless Tube; E = 28 Mpsi, Poisson's Ratio = 0.283, R = 1.9 cm, t = 6 mils
1000
100
10
1
psig
0
50
100
150
200
250
L/R
2 lobes
3 lobes
4 lobes
5 lobes
6 lobes

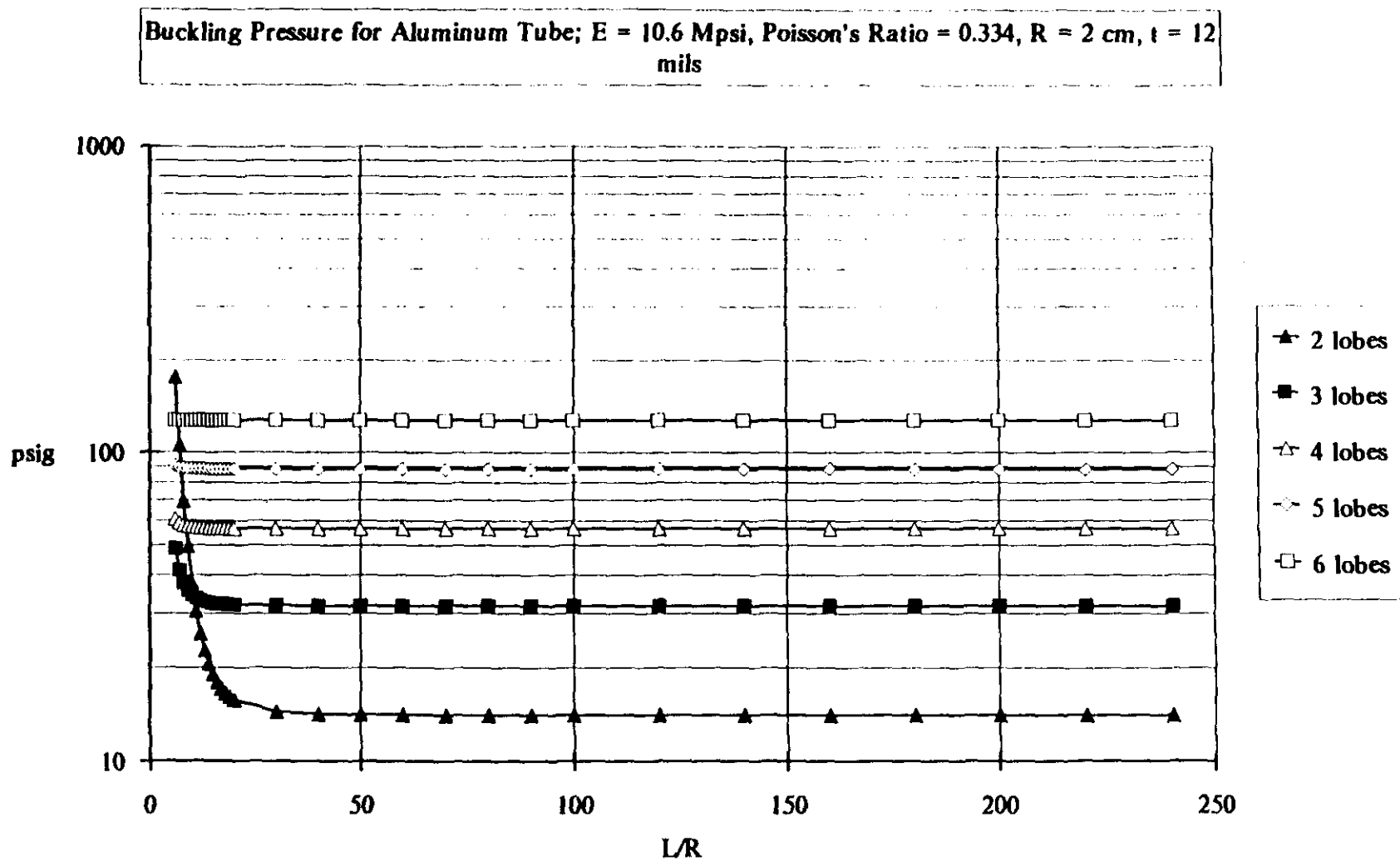

Buckling Pressure for Aluminum Tube; E = 10.6 Mpsi, Poisson's Ratio = 0.334, R = 2 cm, t = 12 mils
1000
100
10
psig
0
50
100
150
200
250
L/R
2 lobes
3 lobes
4 lobes
5 lobes
6 lobes

Fig. 7