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INFLATABLE GASKET FOR THE 72-INCH BUBBLE CHAMBER

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ABSTRACT

A satisfactory glass-to-metal seal at liquid hydrogen temperatures has been developed for the large oval-shaped optical window of the 72-inch liquid hydrogen bubble chamber. Indium wire is held in contact with the chamber glass by an inflatable stainless steel member capable of 160 mils useful deflection.

During cooldown to liquid hydrogen temperatures, the stainless steel chamber shrinks about 3/16 in. more than the glass, causing a relative translation between the contacting surfaces. To avoid opening leaks, the gasket is not sealed until the cooldown is nearly completed. Only a nominal inflation pressure (~ 50 psig) is used during the cooldown, enough to keep the assembled parts in position without mashing the indium. At about 77°K, after most of the relative shrinkage has occurred, the inflation pressure is increased to between 400 and 600 psig, mashing the indium and making the seal.

Indium wire is used instead of lead because lead cannot be easily mashed at 77°K. We have also found that 99.999%-pure indium gives a better seal than the 99.9% pure, since it is softer. We have experienced no difficulty with indium sticking to the glass, possibly because the gasket is not pressurized in a vacuum and because the compression between indium and glass is made at low temperature.

With a double row of indium wires, such as we use, the bearing stress on the glass is only ~ 800 psi at an inflation pressure of 400 psig. Since the BSC-517/645 glass should be able to withstand a compressive stress on the order of 10,000 psi for this gasket geometry, an adequate margin of safety should exist.

Because a multiplicity of seals with pumpouts between each seal is used, it is not necessary that each seal be tight to a helium leak detector, as long as the vacuum pumps can handle the leaks. In practice we have found that after an initial period of chamber pulsing the seal improves. A vacuum in the range of 10^{-5} to 10^{-6} mm Hg can be held above the glass, with a liquid pressure below the glass varying from 95 to 30 psig. Pressures in the chamber-side and glass-side pumpouts vary from 10μ to 1000μ, and in the glass-edge pumpout the pressure is about 50μ.

During the first liquid hydrogen operation the chamber was cycled from room temperature to low absolute temperatures (27°K) three times with the same set of gaskets.

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I. INTRODUCTION

During the operation of the 72-inch liquid hydrogen bubble chamber there is a pressure varying from 95 to 30 psig in the 27°K liquid hydrogen below the glass and an insulating vacuum above the glass (Fig. 1). The first gasket developed to provide the glass-to-metal seal consisted of lead wires held in contact with the glass by an inflatable stainless steel tube (Fig. 2).[1] Short-distance irregularities in the contact surfaces would be sealed by the wire, whereas longer-distance variations would be accommodated by expansion or contraction of the inflation tube.

This first design proved to have a total useful deflection of only 10 mils with a calculated bending stress of 60,000 psi. Where the ends were lapped and soldered the stresses were about four times as great. Also the pressure connections were prone to leak, since they joined the inflation tube at a point of maximum stress.[2]

As the bubble chamber design progressed and as the design and assembly tolerances became known, it became evident that a gasket accommodation of about 160 mils would be needed. It also became apparent that lead would not be suitable as a sealing material, because it becomes stiff at low temperature and requires such high sealing pressures as would endanger the glass. During the cooldown to liquid hydrogen temperatures the chamber shrinks about 3/16 in. more than the glass, which causes a relative translation of the contacting surfaces. To avoid opening leaks, the gasket must not be sealed until the cooldown has nearly been completed.

The problem then was to design a vacuum seal that could seat and remain relatively tight at liquid hydrogen temperatures without requiring a prohibitive bearing stress. This report discusses the preliminary tests conducted to find a suitable design and sealing materials, and describes the inflatable gasket using indium seals which was finally used with the bubble chamber.

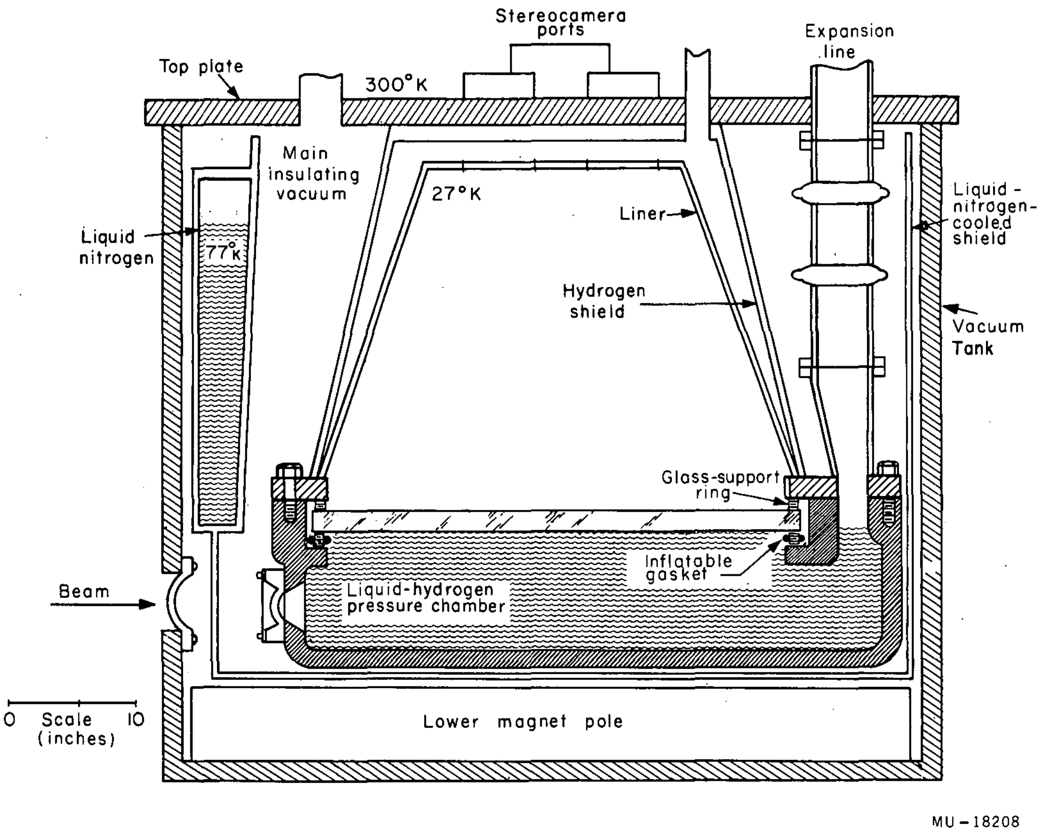

Fig. 1. Side elevation of the 72-in. bubble chamber.

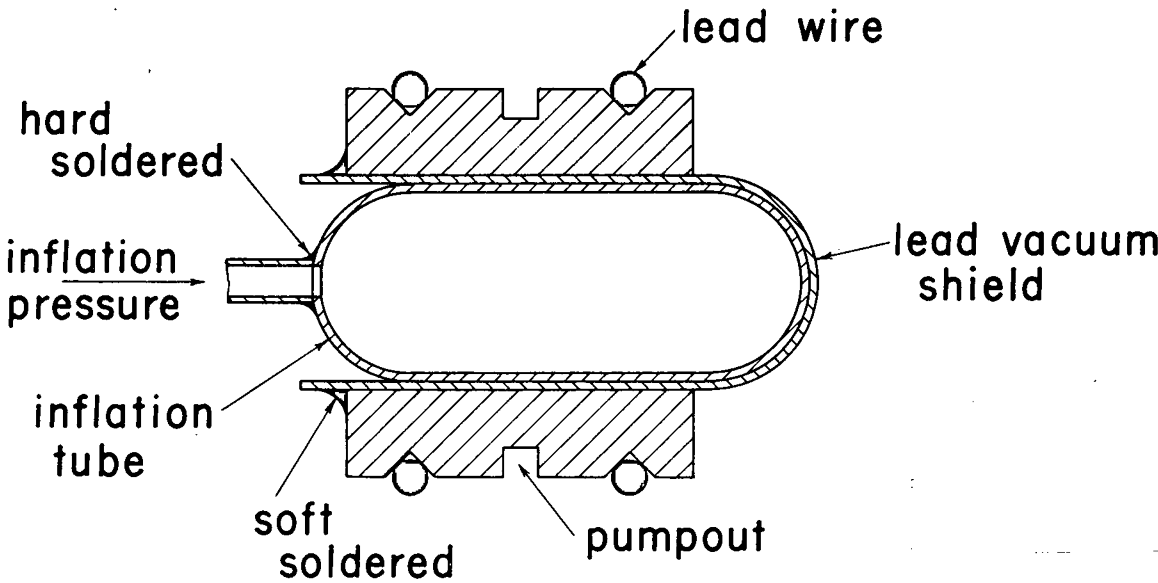

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Fig. 2. First experimental inflatable gasket. This gasket did not meet the design requirements.

II. PRELIMINARY STUDIES

A. Studies of Inflation-Tube Designs

1. Design Requirements

If the gasket were designed to accommodate both the mashing of the wire seals and the accumulation of the necessary construction tolerances of the assembled parts, as indicated in Table I, it should have a useful deflection of 172 mils. This should be ultrasafe, since it is unlikely that all four pieces would have the maximum tolerance. Also, if it should be necessary, the four layers of wires (Fig. 3) could be precompressed about 10 mils each, either before assembly or during the tightening of the clamping bolts, without jeopardizing the final seal. It seemed quite adequate then, to establish the gasket accommodation at about 160 mils total.

Preliminary tests at room temperature with a 3/16-in.-diameter indium wire indicated that a pressure of 225 lb/lin-in. is required to make a vacuumtight seal, i.e. one which gives no response when tested with a helium leak detector.[3] Two wires require 450 lb/lin-in., and provision should be made to supply twice this force. If the effective width of the gas support is 1 in., then the maximum gas pressure is of the order of 1000 psi.

2. Geometry of Curved Beams

An inflation tube of circular cross section is best able to withstand the 1000-psi pressure, because the pressure induces only direct tension in the skin and no bending stress. The stresses in curved beams were studied to find out if a total deflection of 160 mils could be obtained using an inflation tube small enough to fit into the allowed space.

The first design studied, one which could be analyzed as a 90° curved beam, was unsatisfactory. The next design, one analyzed as a 180° curved beam, appeared suitable and was therefore chosen for fabrication. These preliminary studies are summarized below.

a. 90° Curved Beam

A 90°-curved-beam design as shown in Fig. 4 looked attractive because the welds would be accessible. However, to obtain a 160-mil deflection in a tube thick enough to weld (~ 15 mils) would require an inflation tube with a radius of 0.8 in. (see Fig. 5). A gasket of this size would not fit the available space, since the largest acceptable radius is ~ 0.4 in. This design was therefore abandoned.

b. 180° Curved Beam

A 180°-curved-beam design (Fig. 6) was next studied. It was found that this design would give the required 160-mil deflection for inflation tube radii between 0.31 and 0.40 in. and thicknesses between 15 and 25 mils (see Fig. 7). In this interval the tube is thick enough to be welded, and the total stress does not exceed the ultimate strength of the stainless steel. This design was therefore selected. (See the Appendix for a discussion and for formulas on which Figs. 5 and 7 are based.)

Table I

Maximum accommodation requirements[a]	
Structural design tolerances:	
depth of chamber-window recess	32 mils
glass-support ring thickness	16
glass-window thickness	20[b]
inflatable-gasket minimum thickness	32
Wire-seal compression (maximum for four layers of wire)	72
Total maximum accommodation	172 mils

a. Based on the arrangement shown in Fig. 3.

b. Each side is flat within 10 mils, with a thickness variation of 20 mils.

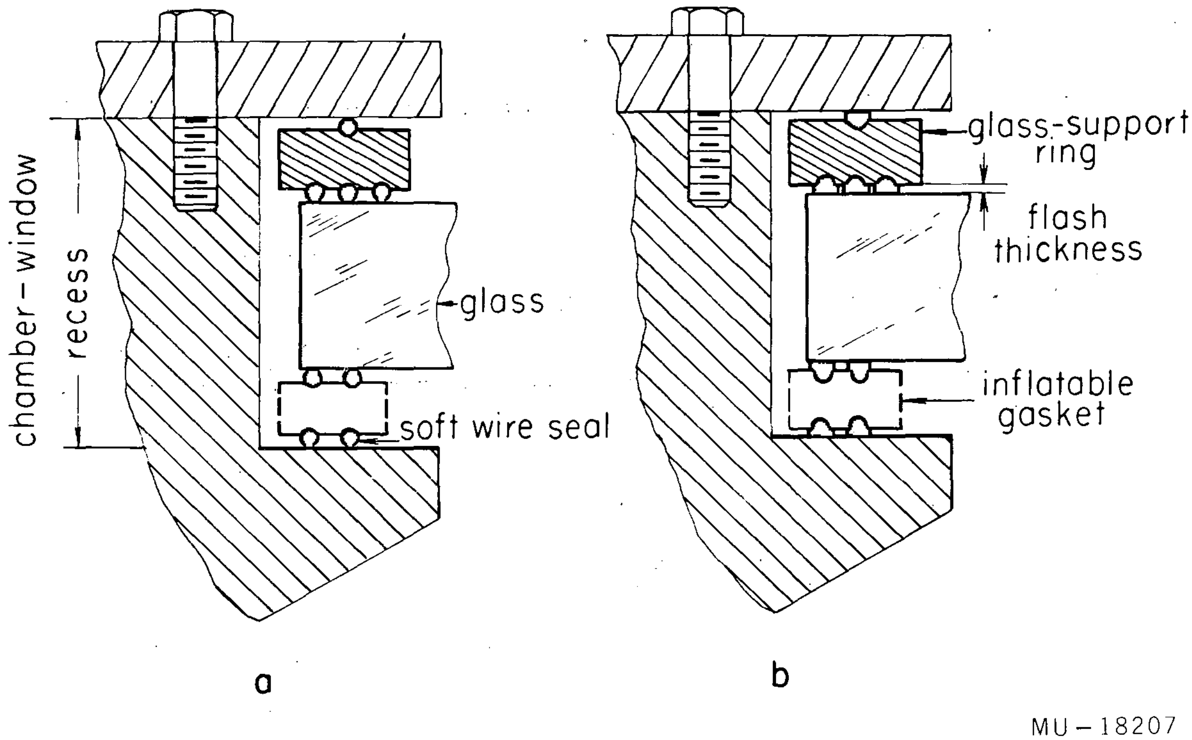

Fig. 3. Proposed arrangement of inflatable gasket and related parts.

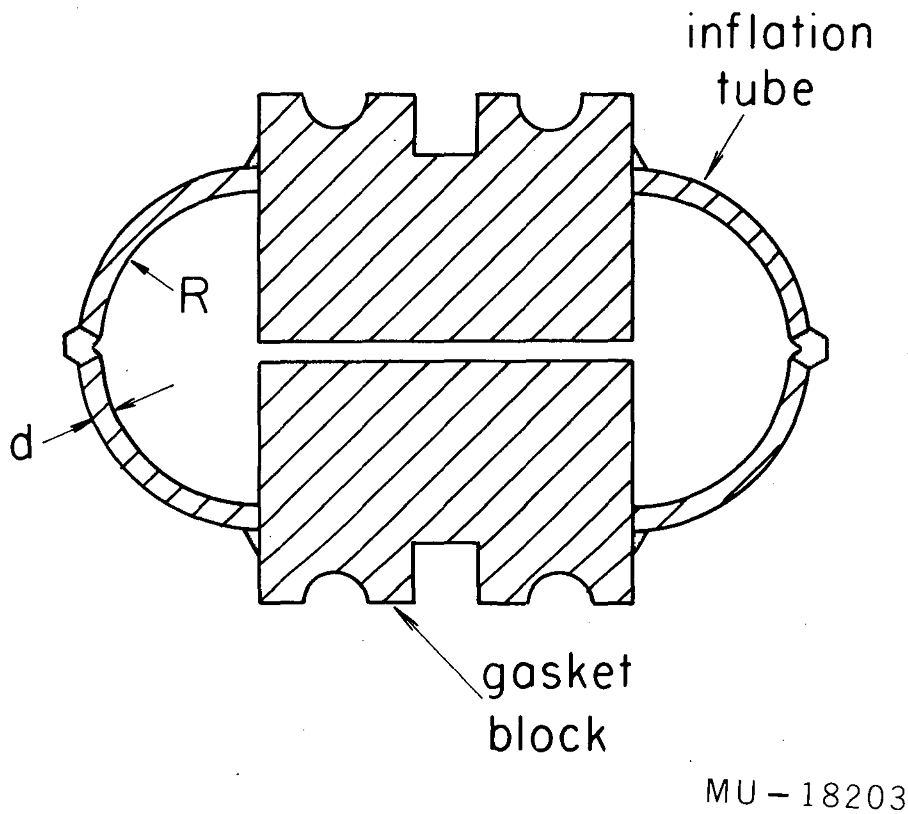

Fig. 4. Inflation-tube design analyzed as a 90° curved beam. This design is unsuitable for the requirements of the 72-in. bubble chamber.

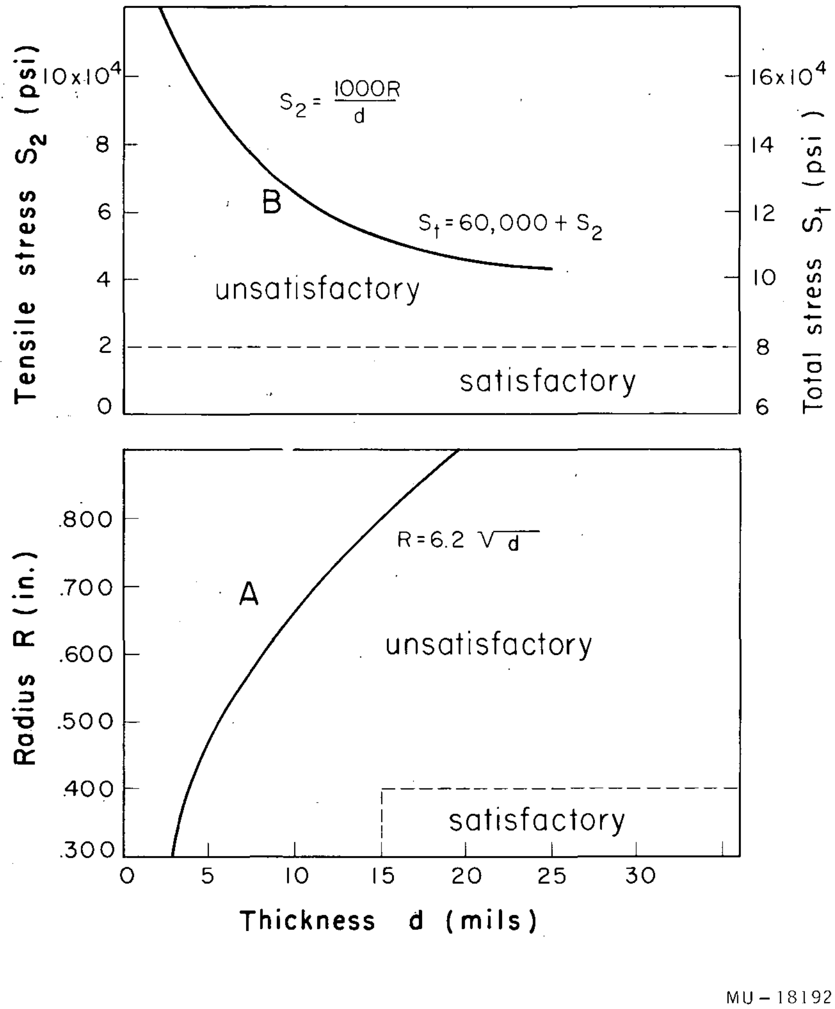

Fig. 5. Radius vs thickness and stress vs thickness in a stainless steel inflation tube designed to give 160 mils useful deflection, based on the design of Fig. 4 (90^o curved beam). Curve A is plotted from Eq. (21), and Curve B from Eq. (23) of the Appendix.

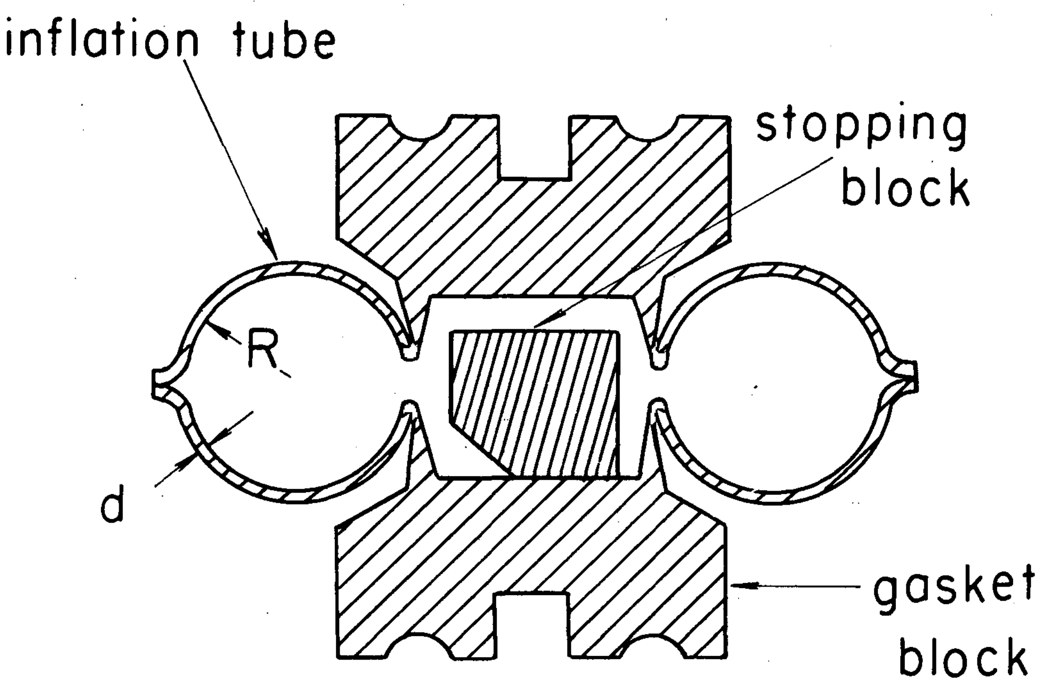

Fig. 6. Inflation-tube design analyzed as a 180° curved beam. This design was chosen for fabrication.

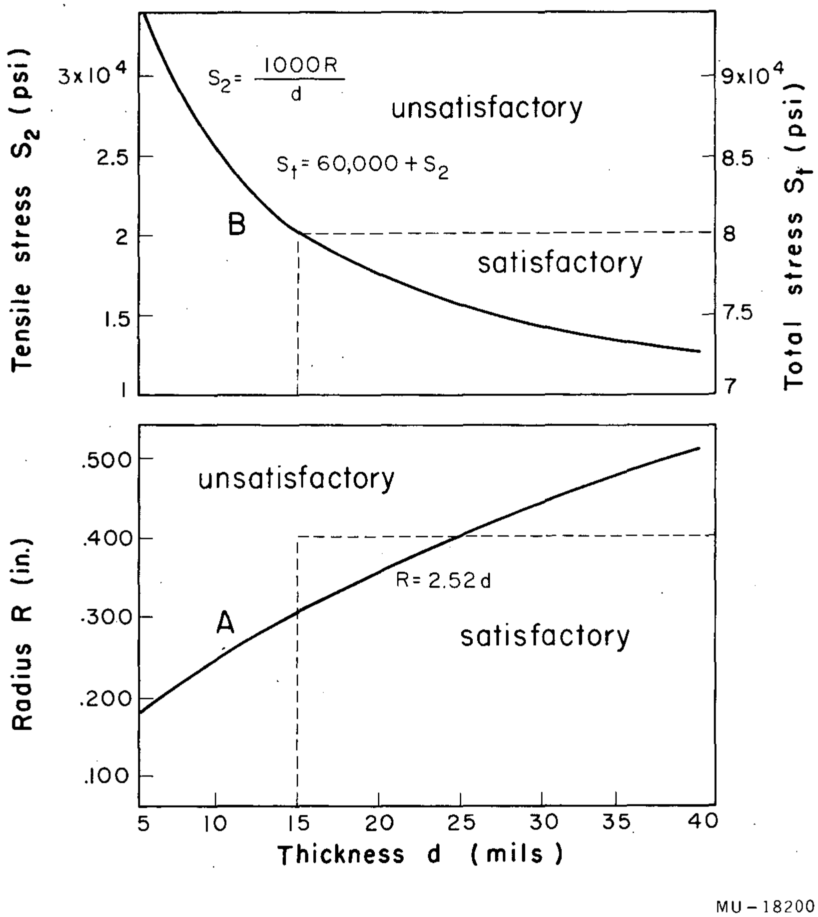

Fig. 7. Radius vs thickness and stress vs thickness in a stainless steel inflation tube designed to give 160 mils useful deflection, based on the design of Fig. 6 (180° curved beam). Curve A is plotted from Eq. (41), and Curve B from Eq. (23) of the Appendix.

B. Experiments With Metal Seals

1. Indium and Lead

A comparison was made between indium and lead at different loads and temperatures to determine the sealing characteristics of each.[4] Figure 8 shows the experimental arrangement and Table II summarizes the results. Lead did not make an acceptable seal even at room temperature and a load of 6020 lb. At this load the unit pressure on the glass was 4090 psi.

Other tests showed that the compressive stress needed to break a 10-in. bubble chamber window is about 12,500 psi (Fig. 9). Furthermore, Kropschot reports that borosilicate optical glass, such as that used with the 72-in. chamber, has a 1% probability of breaking at a tensile stress of 6000 psi at room temperature (8800 psi at 27°K).[5] From this it appears that if lead were used, a safety factor of only about 2 would exist. This is not regarded as adequate when one considers the uncertainties involved.

On the other hand, indium made an acceptable seal at room temperature with a load of only 1000 lb, and became tight (i.e. showed no response on a helium leak detector) at 2000 lb. With indium the unit pressure on the glass was 970 psi at the 2000-lb load, well under the breaking stress of the glass.

From these tests it appeared that indium might be a suitable material for the wire seals used with the inflatable gasket. Consequently, further tests were conducted with indium.

2. Further Tests With Indium

a. Effect of Impurities

Tests were carried out to see what effect impurities have on the sealing properties of indium.[6] The experimental arrangement used is shown in Fig. 10 and the results are presented in Table III.

The data reveal that impurities have little effect on the sealing properties of indium at room temperature, but from 197° down to 77° K the 99.999%-pure indium gives a better seal since it is softer than the 99.9% pure. (See Fig. 11.)

b. Stiffness

(1) Flash thickness vs time. Figure 12 shows the results of tests conducted to determine how flash thickness decreases with time for a given inflation pressure.[7] These curves do not show the true zero, because the same indium wire was used throughout the test, the wire being progressively mashed by higher pressures.

These curves reveal two things: pressure is more important than time in producing a given flash thickness; and there must either be sufficient resilience built into an indium-seal joint to follow its relaxation, or the joint must be accessible for intermittent tightening.

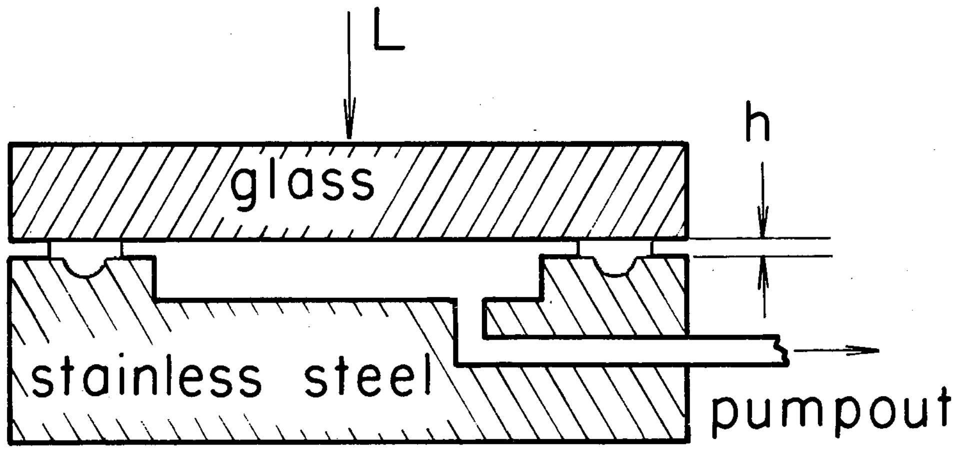

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Fig. 8. Arrangement for preliminary tests with indium and lead seals.

Table II

Sealing properties of indium and lead (at 300° K)

Load, L (lb)	Time from start of compression (min)	Vacuum[a]	Flash thickness, h (mils)
Lead			
1600	0	8μ	(almost no deformation)
2345	10	(see footnote b)	
2800	75	150	
2800	135	70	
6020	145	10	(some deformation)
6020	170	8	
Indium			
1190	0	1	(some deformation)
2100	15	0	
3500	40	0	32
3500	70	0	28

a. Except for the first value, which is in microns of Hg, all values are for readings on a helium leak detector (h. l. d.).

b. The pump on the helium leak detector could carry the full leak.

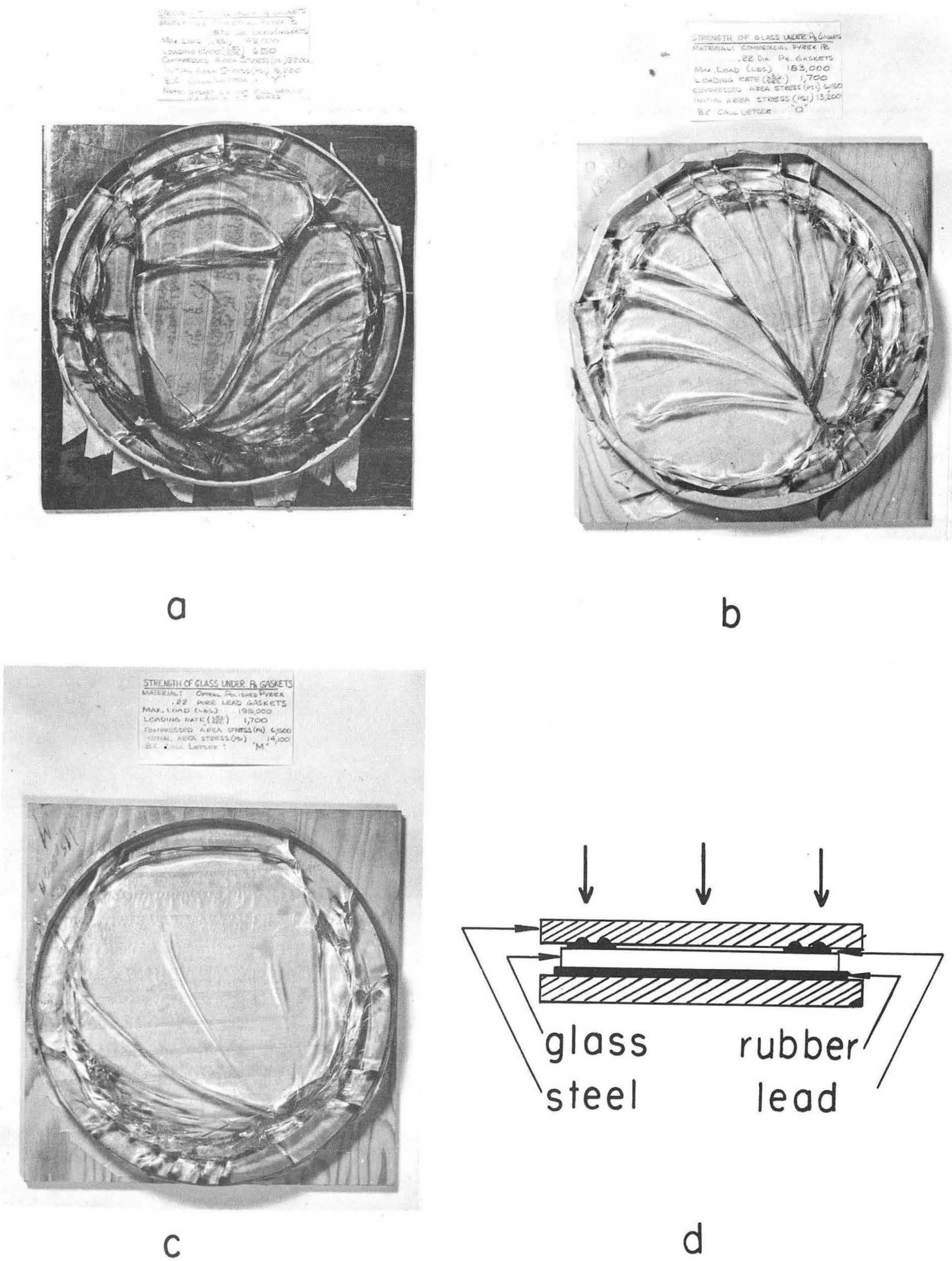

Fig. 9. In preliminary tests, Pyrex windows broke at the following compressive stresses: (a) 14,000 psi; (b) 13,000 psi; (c) 9,000 psi (since the wire did not fill the groove, the glass hit the lower metal plate). The test arrangement is shown in (d). The windows were 10-1/2 in. in diameter and 1-1/2 in. thick. The lead gaskets were 0.22 in. in diameter.

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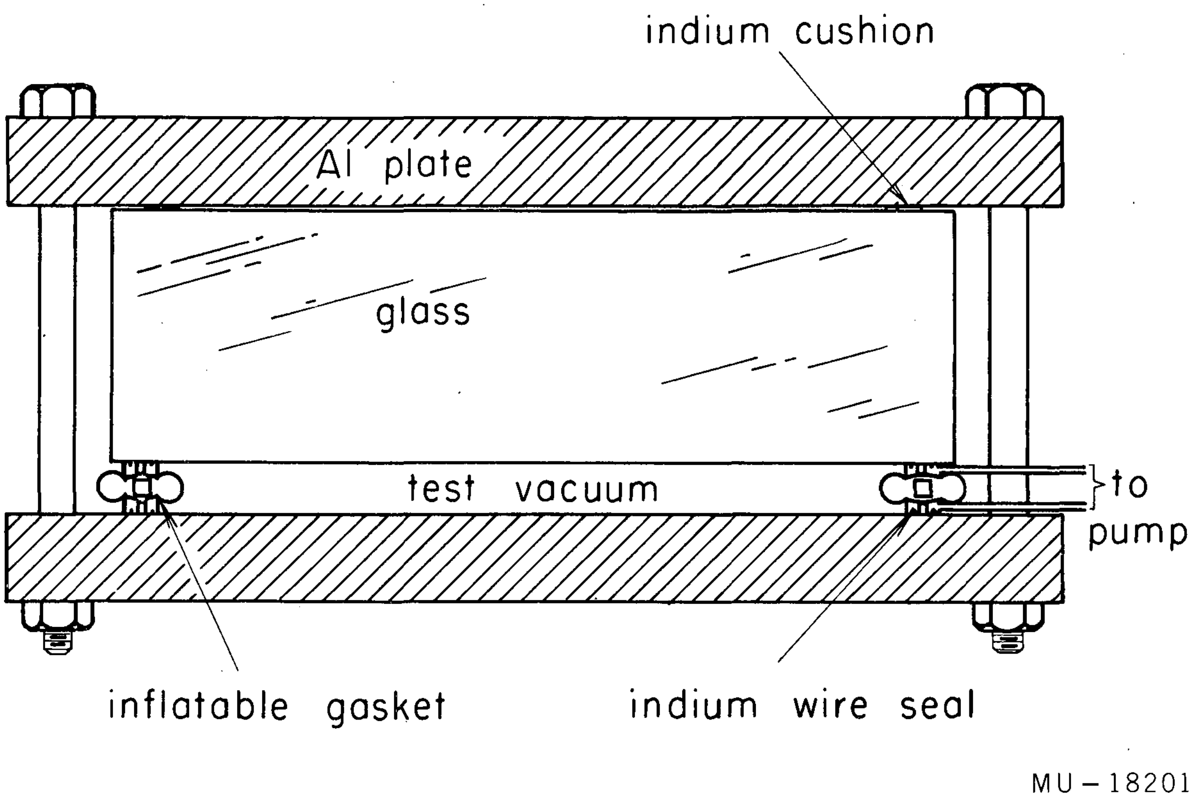

Fig. 10. Arrangement for preliminary tests to determine the effect of impurities on the sealing properties of indium.

Table III

Effect of impurities on the sealing properties of indium

	99.9% Pure			99.999% Pure		
Temp. (^{o}K)	Infl. pressure (psig)	Vacuum[a] Glass side	Vacuum[a] Al side	Infl. pressure (psig)	Vacuum[a] Glass side	Vacuum[a] Al side
300	100	5.5μ	5.5μ	50	21μ	16μ
265	100	4μ	4μ	50	30μ	30μ
236	100	1000μ	--	50	33μ	33μ
230	100	1000μ	5μ	50	36μ	36μ
197	100	1000+μ	38μ	50	130μ	42μ
90	100	--	--	100	3"Hg	90μ
90	100	--	--	150	2.5"Hg	72μ
90	100	--	--	200	0.4"Hg	58μ
77	200	no seal	no seal	250	0.4"Hg	64μ
77	300	no seal	150μ	300	500μ	--
77	400	24"Hg	35μ	400	100μ	45μ
77	500	20"Hg	25μ	300	--	45μ
77	700	12"Hg	21μ	200	--	48μ
77	800	3"Hg	14μ	125	--	50μ
77	900	0.4"Hg	--	100	--	54μ
fixture failed				75	--	70μ
				lost seal -- parts shifted		
77				200	400μ	0.2"Hg
77				300	400μ	1000μ
77				400	200μ	1000μ
77				500	95μ	700μ
77				550	81μ	700μ
77				300	85μ	700μ

a. Ion-gage readings at the pump manifold.

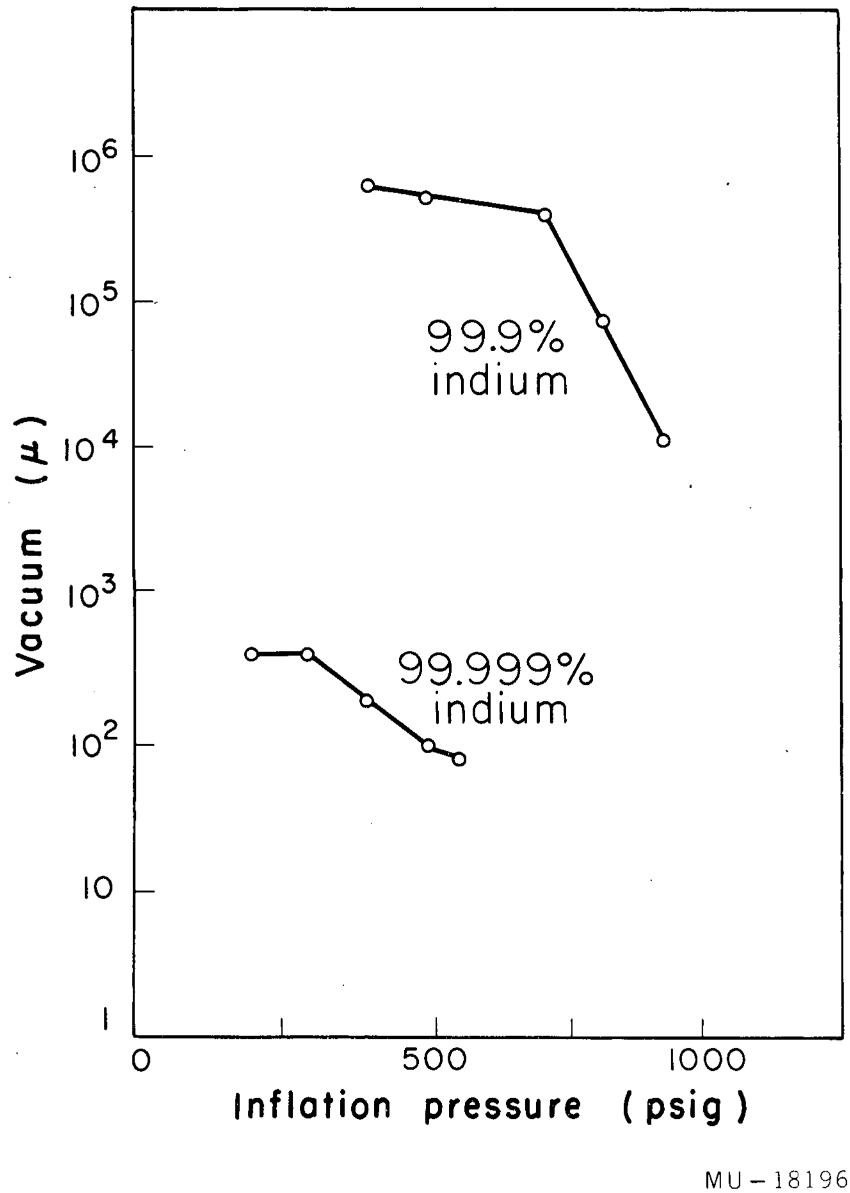

Fig. 11. The effect of impurities on the sealing properties of indium at 77° K (based on data from Table III).

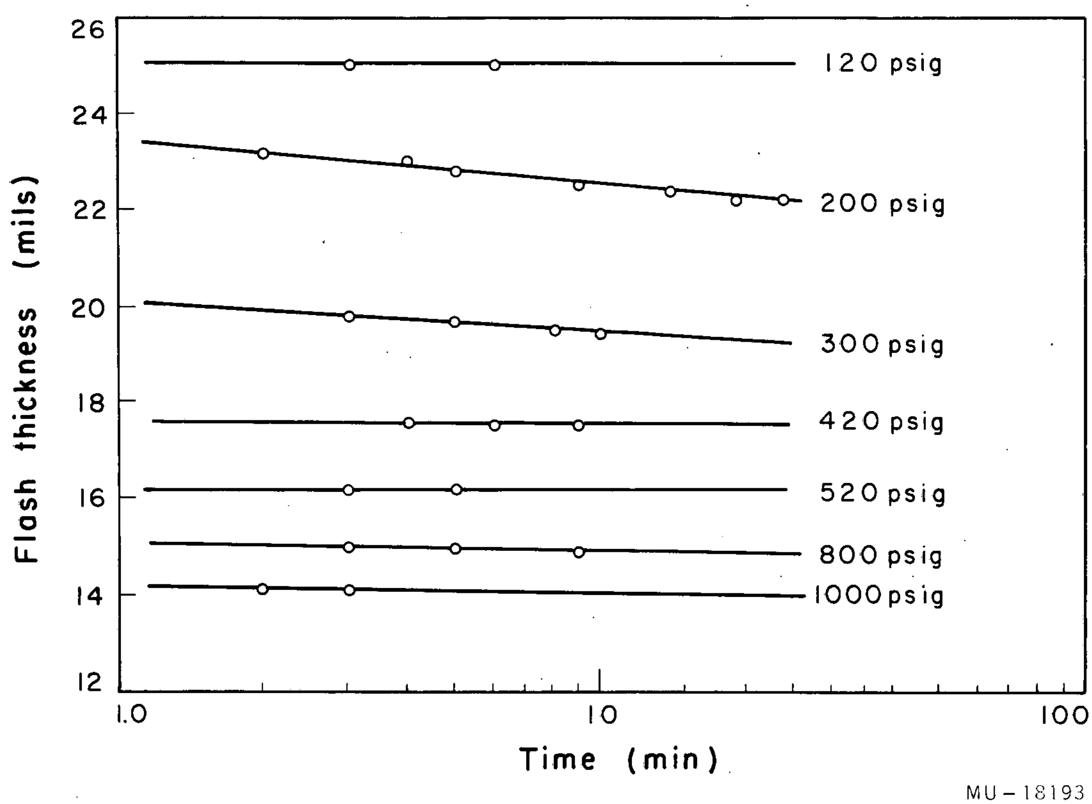

Fig. 12. Flash thickness vs time at different pressures for indium wire (at 300° K).

(2) Flash thickness vs pressure. Figure 13 shows flash thickness plotted against pressure at two different temperatures, 300° and 77°K.

It is evident that to get a good seal the indium must not be set until after the cooldown has been nearly completed. For instance, if we were to apply an inflation pressure of 500 psi at room temperature, the indium would be compressed to 16.5 mils. During cooldown leaks would inevitably be opened, owing to the differential contraction between the stainless steel chamber and the glass. To obtain any further seating at 77°K would require an inflation pressure of nearly 4400 psi, enough to induce dangerous stresses in the glass. It would also be difficult to make a gasket which could withstand such a high inflation pressure.

This principle was illustrated in a test in which the indium was first set with a pressure of 500 psig at room temperature. The best vacuum attainable at 77°K was 0.12 in. Hg using a 1000-psig inflation pressure. When no setting was induced at room temperature, a vacuum of 28μ was obtained at 77°K using a 1000-psig inflation pressure.

Thus we see that from a leak-rate point of view it is not desirable to apply more than a nominal inflation pressure at room temperature.

c. Miscellaneous

(1) Surface finish. No tests were conducted to determine the effect of surface finish on the inflation pressure required to make a tight seal, but there is an excellent reference on this topic.[8] In all the preliminary tests the glass and metal surfaces were highly polished. The fact that a few scratches were not removed from the aluminum plates might account for the poorer vacuums attainable on the aluminum side in the tests at 77°K after the parts had shifted.*

(2) Surface cleanliness. Two tests were made with indium wire as it came off the spool, leaving the light grey tarnish on the indium undisturbed. The wire was laid on a surface cleaned with acetone. Airborne lint or hair would infariably fall on a gasket surface producing leaks. These leaks were large enough so that the best vacuum at the pump was 50μ at room temperature with an inflation pressure of 1000 psig.

In a later test the indium wire was polished by wiping it with a rag soaked in acetone. The wire and groove were then flooded with acetone; this eliminated the lint and hair, but there was no indication that polishing the indium had any value.

(3) Indium sticking to surfaces. We found that the indium would stick to the contact surface when it was purposely flattened at room temperature by a high inflation pressure. Sticking could be avoided by making the compression between the indium and glass at low temperature and in the presence of a contaminant liquid or gas.

*See Table III.

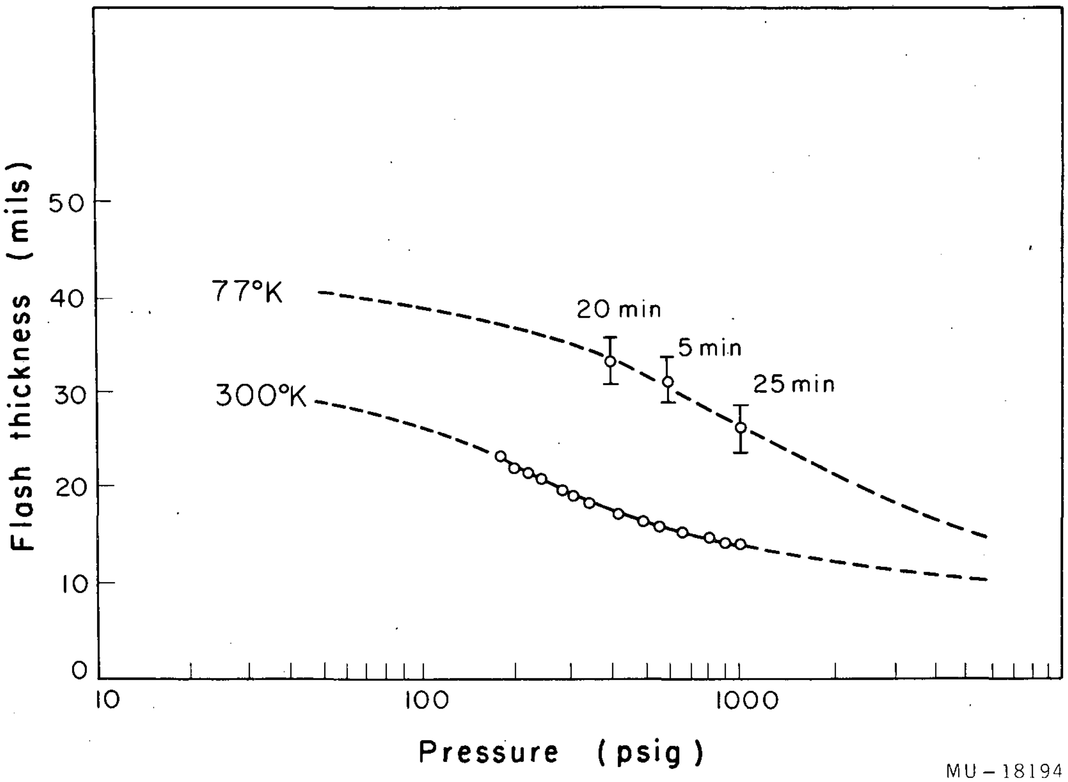

Fig. 13. Flash thickness vs pressure for indium wire (at 300° and 77° K). For the 300°K curve all thicknesses were taken after 30 minutes at a given pressure.

3. Summary of Tests With Indium and Lead

Indium was seen to be softer at 20°K than lead is at room temperature. Satisfactory seals can be made with indium at low tempertures and at moderate pressures so that the glass window is not endangered. The 99.999%-pure indium was found to be definitely superior to the 99.9% as a sealing material at liquid hydrogen temperatures.

III. THE DESIGN ADOPTED

A. Description

Figure 14 is a cross section and Fig. 15 a plan view of the inflatable gasket installed in the chamber. Table IV gives the gasket dimensions and materials.

Table IV

Dimensions and characteristics of the inflatable gasket

maximum accommodation	160 mils
load to compress gasket 80 mils	31.3 lb/lin-in.
maximum test inflation pressure (at 77°K)	800 psig
inflation gas	helium (reactor grade)
dimensions:	
inflation-tube radius (R)	0.375 in.
inflation-tube thickness (d)	19 mils
materials:	
inflation tube	type-305 stainless steel
gasket block	type-304 stainless steel
diameter of indium wire	0.150 in.

Note that in the plan view the middles of the long straight sides of the gasket are crowded outward at the time of assembly until the contact bars touch the sides of the chamber. Without this support, the pressure under the glass could cause the gasket to skid, producing a leak. The curved ends of the gasket are capable of restraining the hydraulic force by a hoop tension in the gasket and therefore require no contact against the chamber.

The gasket is protected from over-compression by an internal stop, but there is no protection from over-extension. The gasket must not be pressurized except in an environment which provides the necessary restraint.

The groove (Fig. 16) for the indium is made is made with enough depth of parallel sides to allow an inversion of the gasket assembly without the indium falling out of the groove, providing the indium is given a small initial mashing. The cross-sectional area of the groove is made less than that of the indium wire.

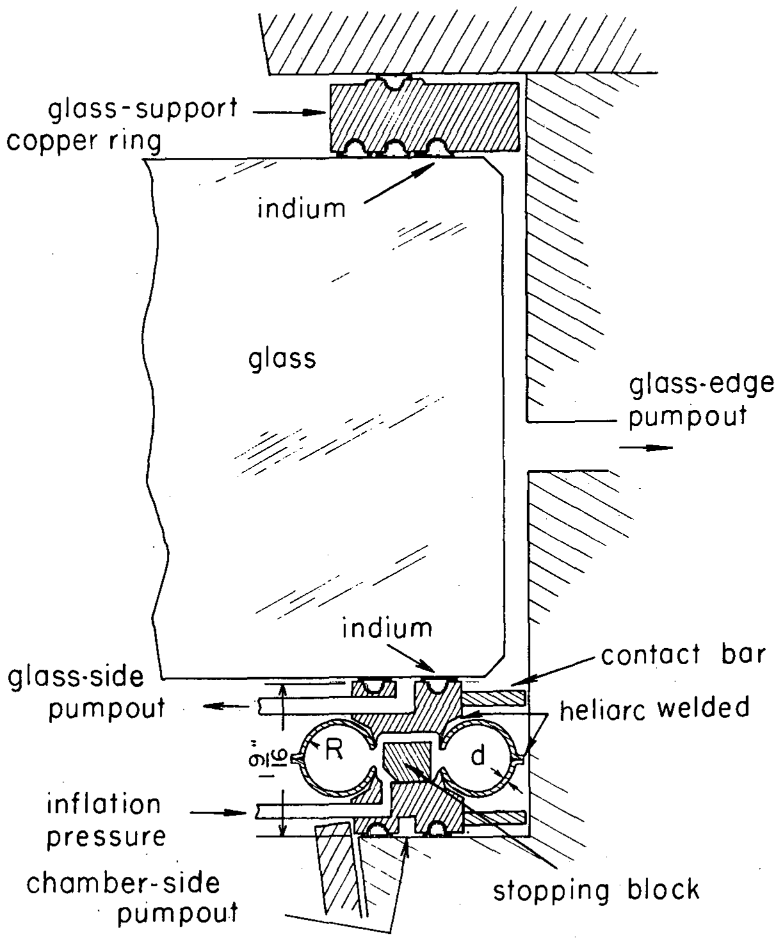

Fig. 14. Cross section of the inflatable gasket and related parts of the chamber rim and glass, as installed.

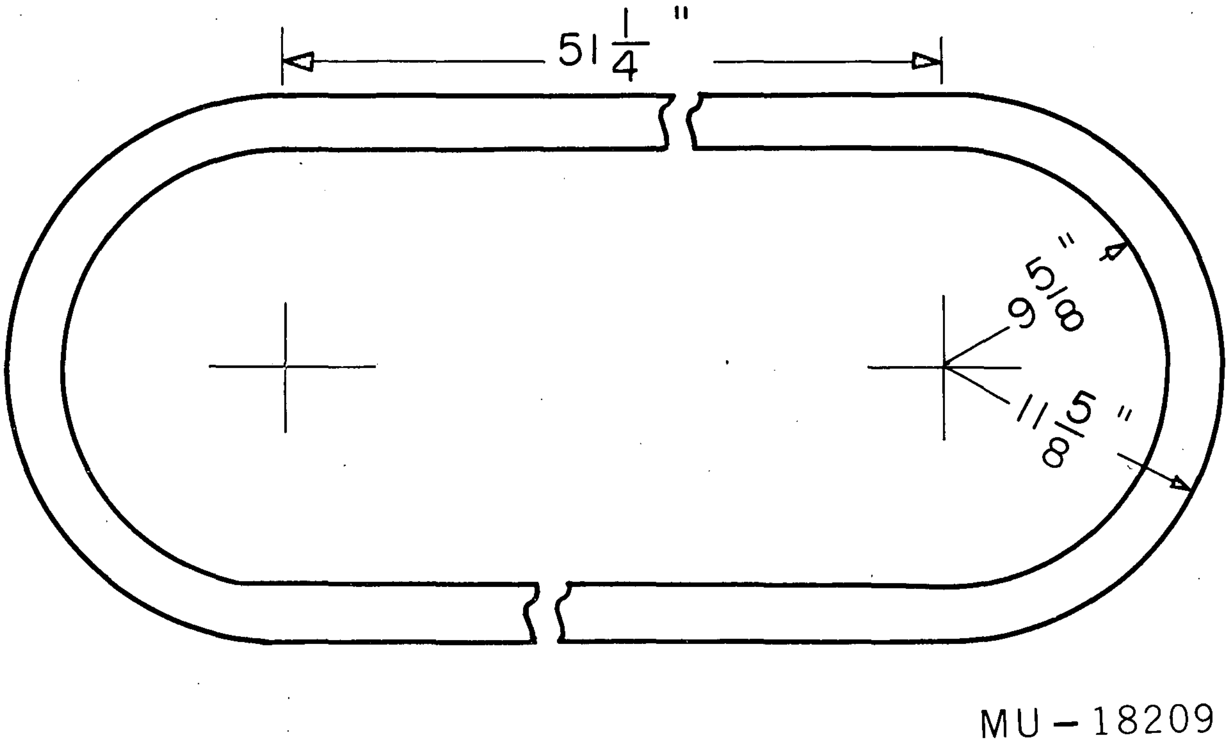

Fig. 15 (a) Plan view of inflatable gasket (simplified).

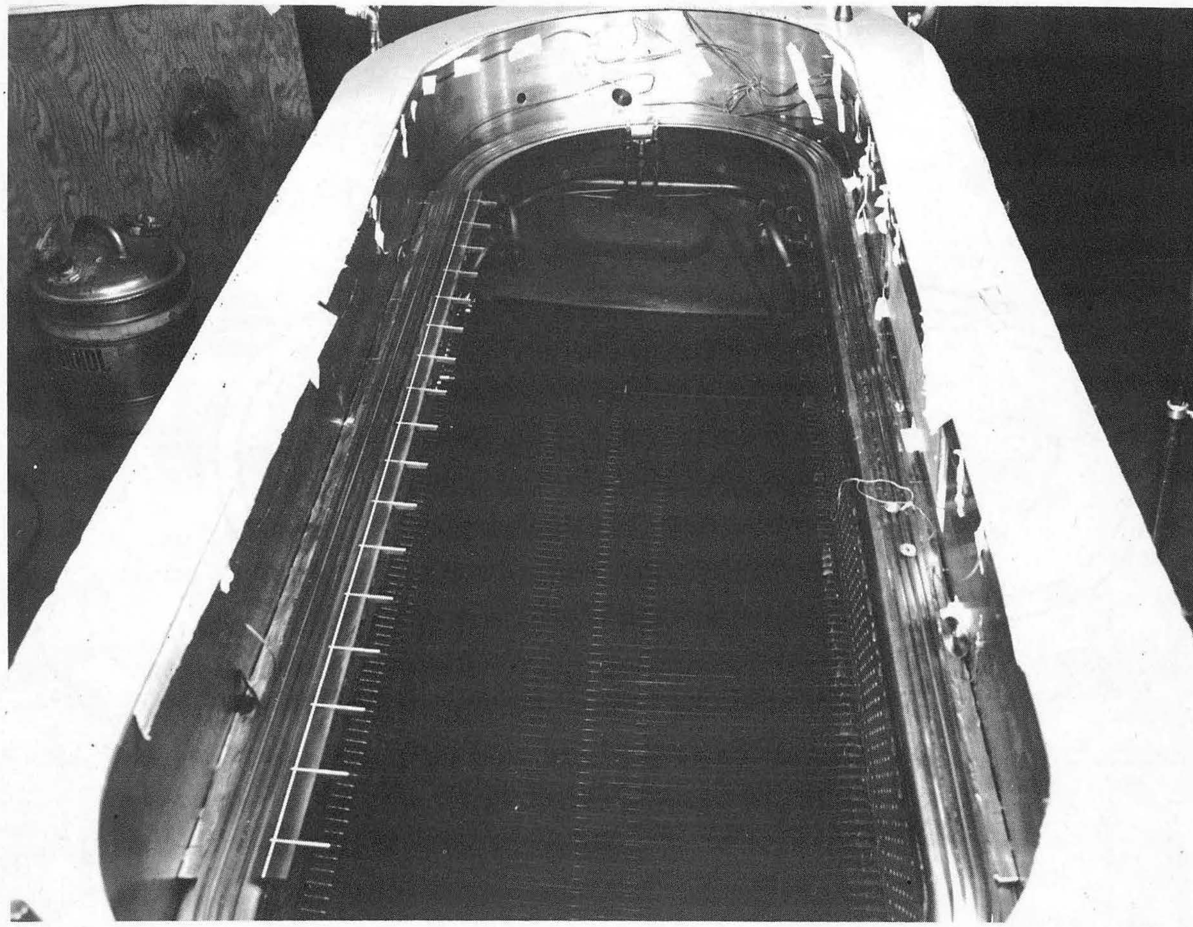

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Fig. 15 (b) The 72-in. chamber showing the inflatable gasket in position.

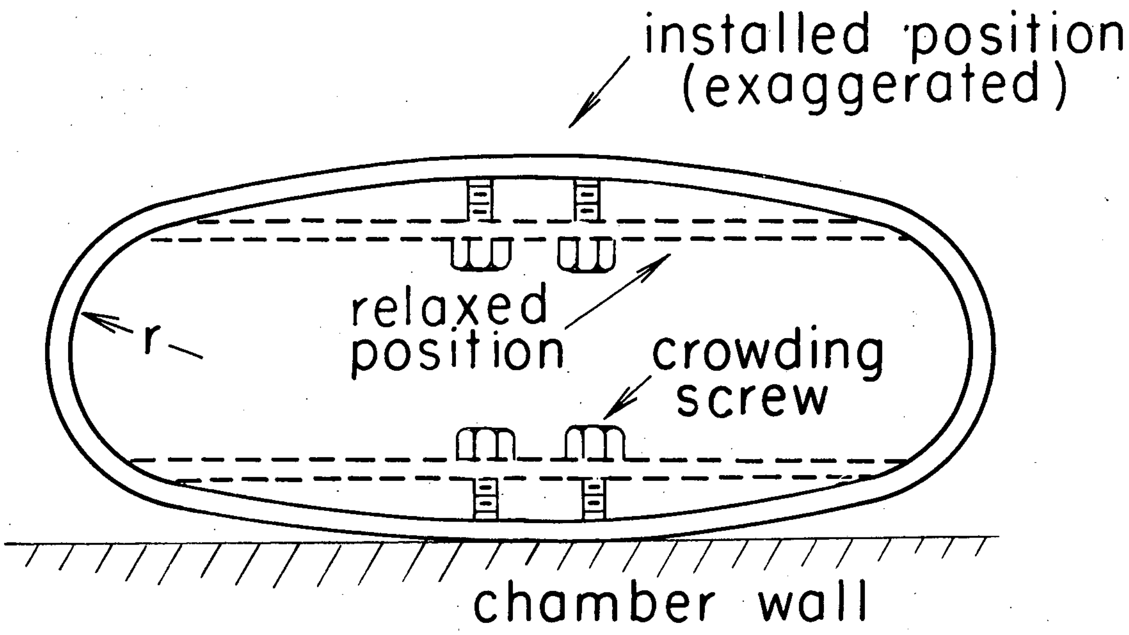

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Fig. 15 (c) Plan view of the inflatable gasket (exaggerated), showing how the middles of the straight sides of the gasket are crowded outward at the time of assembly until the contact bars touch the chamber wall.

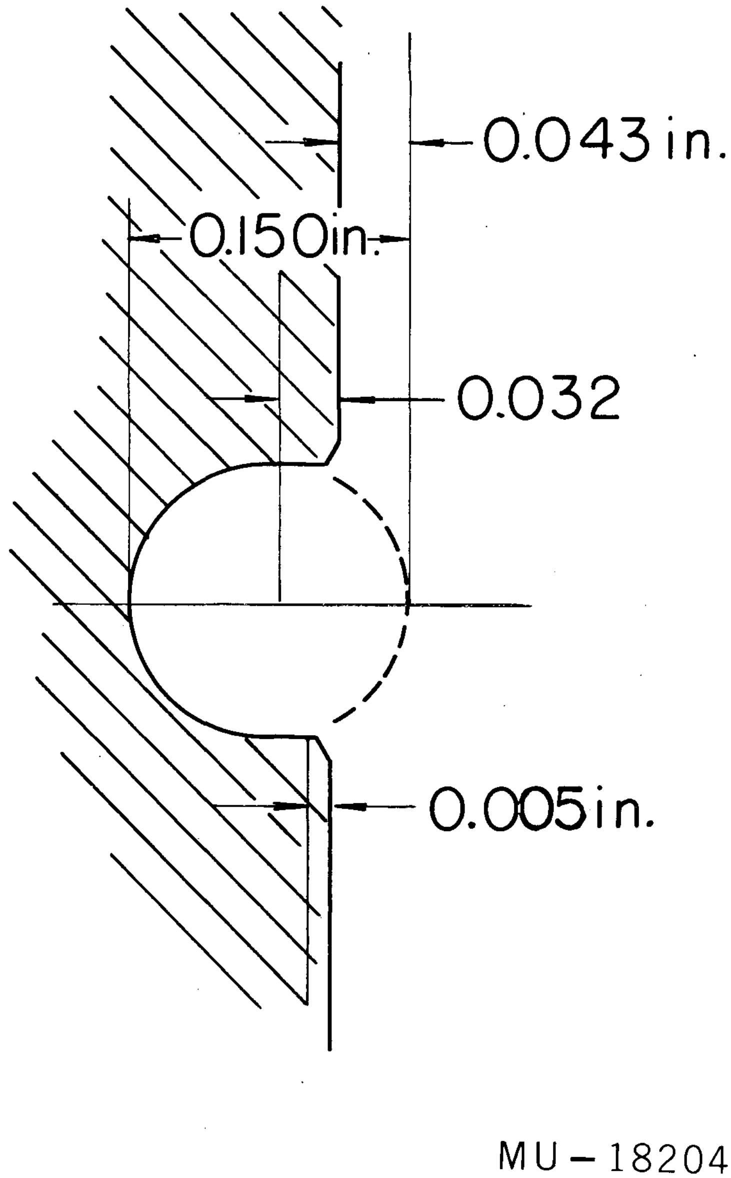

Fig. 16. Cross section of groove for indium wire.

B. Construction Problems

Austenitic stainless steel was selected for the gasket because it is paramagnetic and has good mechanical properties at low temperatures.

The gasket block (see Fig. 17) was made of 304 stainless bar stock. Each bar was rolled to the plan view shape of Fig. 15a and the ends were arc welded together using a 310 stainless lime-coated electrode.

The inflation-tube pieces (A, B, C, and D of Fig. 17) were made of 305 stainless sheet. Each sheet was stretched over a male die to form the half-round section, and these pieces were heliarc welded to the machined gasket block; no filler rod was added. Each subassembly, consisting of a gasket block and its attached two inflation-tube pieces was vacuum checked and then straightened. The gasket surface was made to fit a flat surface with less than a 5-mil gap.

The necessity for flatness is apparent when we consult Fig. 12. If we expect a normal flash thickness of the extruded indium of 16 mils, the required inflation pressure is 520 psig. At the places where the gap is 21 mils, the pressure can only be equivalent to 230 psig; if it were more, the indium would extrude until the pressure was relieved to this value. If the inflation pressure is increased to 1000 psig, the equivalent pressure at the 5-mil thicker zones increases only to 320 psig. Thus for a 500-psig increase in inflation pressure, the effective increase is only 90 psig. The foregoing statement is strictly true only at 300°K and for such short-length gaps that the gasket block can not bend to decrease the gap. Obviously if only finger pressure is required to flatten the subassembly, the required inflation pressure to flatten is small. We do not have sufficient points on the 77°K curve, Fig. 13, to be sure of the slope; but if we use the points at 26 mils and 31 mils flash thickness, the corresponding pressures are 1000 psig and 600 psig. This indicates that the flatness requirement at 77°K may be less critical than at 300°K.

The completed subassemblies are welded together (see "side weld" of Fig. 17), with heliarc and no filler rod. The inside of the tube is flooded with inert gas to encourage the formation of an internal fillet. This assembly was pressure-tested at 800 psig with water. The gasket was next flexed through its 160-mil range six times and vacuum checked.

A permeability check of the gasket after a temperature cycle to 77°K and back to 300°K showed an increase in some places from 1.02 to as high as 1.6. These changes occurred where the 304 bar had been cold worked by bending to the required radius.

A literature search then revealed that the chemical specifications for 304 are so broad as to include both a stable austenitic steel and also a metastable austenitic steel. The metastable steel can be converted partially to martensite and ferrite, and both of these forms are magnetic.

A second inflatable gasket was made from carefully selected 304 stock that was cold worked and soaked 12 hours at 77°K, and still has a permeability of less than 1.02.

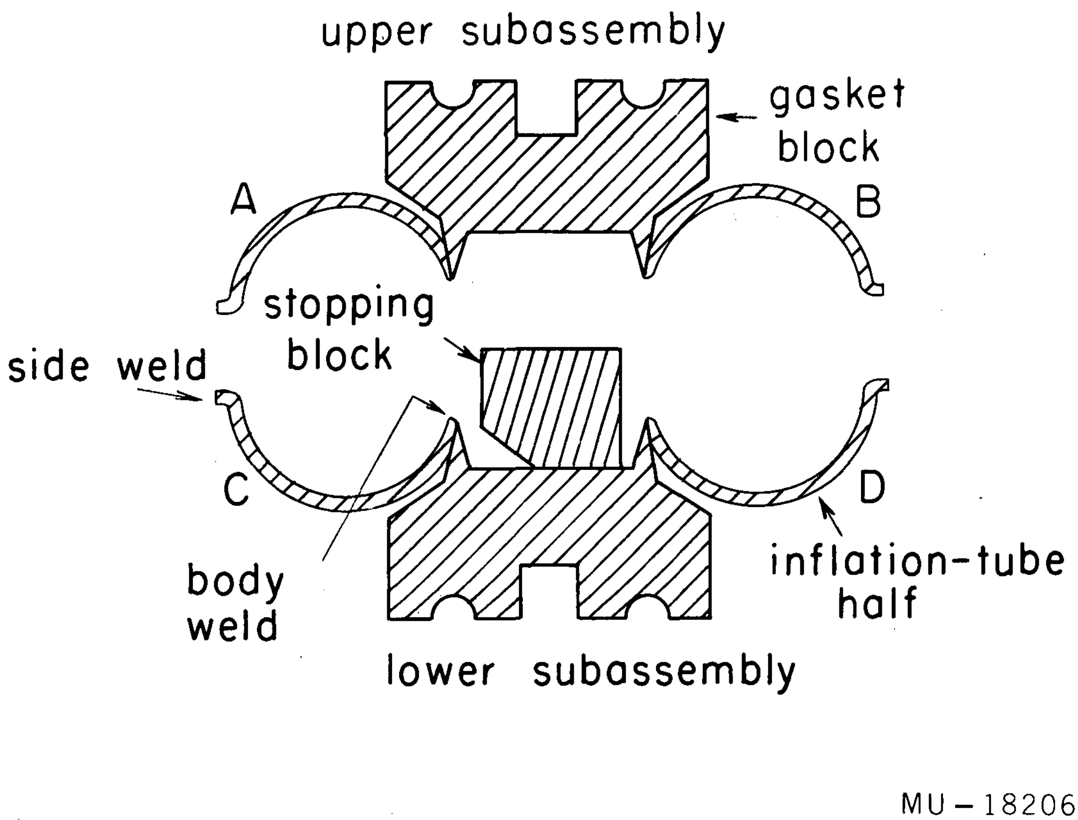

Fig. 17. Inflatable-gasket subassemblies.

IV. EXPERIMENTAL RESULTS

A. Assemby

The quality of the gasket seal depends to a large extent on the cleanliness of the assembly area and on maintaining the air free from dust and dirt. Just before the glass is installed, the gasket should be cleaned with solvent. The glass should then be installed immediately to reduce the probability of collecting dirt.

When the gasket, glass, and the glass-support ring have been installed and clamped by the hydrogen shield, the gasket is inflated to 50 psig. Higher pressures for a final seal are not used until after most of the relative shrinkage has occurred. Clips attached to the chamber prevent the inflatable gasket from being pushed into the chamber by a differential lateral pressure.

B. First Operation -- With Liquid Nitrogen

During the cooldown the helium inflation pressure was maintained at 50 psig. To assist with cooling the glass, the regions above and below the glass and at the edge of the glass were maintained at 5 psig of nitrogen. The pressure in the glass-edge pumpout was about 42μ and about 125μ in the chamber-side pumpout.

At 77°K the inflation pressure was increased to 200 psig to make the seal. When this was done, the pressure in the glass-side pumpout remained about the same, but in the chamber-side pumpout it improved to 55μ. The pressure in the glass-edge pumpout was 42μ. These pressures were measured at the pump manifold with the pump operating, so that the actual pressures at the gasket were probably in the millimeter range. The base pressure of the pump is 20μ and its rating is 15 scfm.

When the chamber was filled with liquid nitrogen, the inflation pressure was increased to 400 psig. The chamber was then pressure-tested to an internal pressure of 145 psig (155 psig across the glass). The seal was maintained.

In the pulsing cycle of the bubble chamber, the pressure of the liquid changes from about 95 to 30 psig and back again to 95 psig in about 100 msec. In this first operational test the chamber was pulsed about 5000 such cycles. The inflatable gasket proved successful.

C. Second Operation With Liquid Nitrogen

The chamber was filled with one atmosphere of nitrogen gas at 300°K, with 50 psig of helium in the inflatable gasket. The rate of pressure rise in the hydrogen shield was measured to determine if the indium gaskets between the glass-support ring and the glass were sealing. The indicated leak rate was 3.5 micron-liters per second over a 9-1/2-hour period (no cold-traps were in the circuit). The base pressure of the hydrogen shield was 15μ. The rate of pressure rise in the glass-side pumpout was 0.8μℓ/sec, and in the chamber-side pumpout it was 70 μℓ/sec. The leak was across the inner gasket between the chamber and the indium.

After the chamber was cooled to 77°K, it was filled with liquid nitrogen. The inflation pressure was increased to 370 psig. The chamber was then pressure-tested to 155 psig across the glass. The pressure in the main vacuum tank changed from 30 to 90×10^{-6} mm Hg, which was better than in the first nitrogen run when it went as high as 340×10^{-6} mm.

The chamber pressure was then decreased to 96 psig. The rate of pressure rise in the glass-edge pumpout was 7 $\mu \ell$/sec. The pressure in the chamber-side pumpout was 65μ. The chamber pressure was decreased to 80 psig, and after 12 hours the main vacuum was 12×10^{-6} mm.

D. Liquid Hydrogen Operation

After the chamber was pressure tested and operated briefly with liquid nitrogen, it was emptied and refilled with liquid hydrogen. During chamber pulsing the rate of pressure rise in the glass-edge pumpout varied from 160 to 0.56 $\mu \ell$/sec; the pressures ranged from 350 μ to 22 μ. This seal improved with operation.

The pressure in the chamber-side pumpout varied from 350 to 65μ. In the glass-side pumpout no rate of rise was detected, and the pressure was essentially the base pressure of the manifold.

Helium is supplied to the gasket through a pressure regulator. If the regulator is set so that a constant pressure is maintained, the helium consumption is high because helium is lost during each warm-up of the chamber of only a few degrees. However, if the regulator is set so that it allows the inflation pressure to vary somewhat, the helium consumption is greatly reduced. A large helium consumption requires that bottles be changed while the chamber is cold. This involves the risk of mixing air with the helium, which would freeze in the cold line.

When the chamber was first filled with liquid hydrogen, bubbling was seen from the inflatable gasket, indicating a helium leak into the hydrogen. The leak was so small that it could not be detected from the rate of fall of pressure in the 2000-psig helium supply bottle. After the run, an analysis was made of the hydrogen, but no traces of helium were detected. The leak could not be observed with a helium leak detector using 160 psig of helium pressure in the gasket. The leak was considered too small to risk damaging the gasket by welding, and it still remains in the gasket.

When the chamber was being inspected after the first liquid hydrogen run, no defects were found in the inflatable gasket. The indium was everywhere mashed until the width of the gasket surface was 3/16-in. except for a few spots where the wire had necked-down and the gasket surface was only ~ 1/8-in. wide. It is believed that these necked-down spots were caused during cooldown when the indium shrinks at a very much faster rate than the stainless steel.

The chamber was reassembled and is now in operation again. The chamber has been pulsed over 100,000 times, with the inflatable gasket performing very well.

ACKNOWLEDGMENTS

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APPENDIX

DERIVATION OF FORMULAS RELATING TO INFLATION-TUBE DESIGN*

1. 90° Curved Beam

Figure 18 shows the forces, moments, and deflections in an inflation-tube design which is analyzed as a 90° curved beam (because of the constant slope at points A and B). (The slope at point B is constant because of the weld at that point; the slope at point A is also constant since it is a point of symmetry.)

The amount of deflection, dy_F, produced by applying an external force, F, at the extreme end of the beam (point B) is:

$$dy_F = \pi F R^3 / 4EI, \tag{1}$$

where R is the radius of the curved beam, E is the modulus of elasticity of the material, and I is the moment of inertia.[10]

If the slope at point B were free to change, the beam would bend downward to an angle θ_F owing to the moment M produced by the force F. Because the slope at point B must remain fixed, a correction moment, M_C, is introduced which bends the beam upward to an angle θ_M equal to θ_F. This upward moment, M_C, causes a deflection dy_M in the upward direction. Therefore, the actual deflection, dy, is

$$dy = dy_F - dy_M. \tag{2}$$

To find the actual deflection, dy, we must first determine θ_F. We write

$$\theta_F = \int \frac{M ds}{EI} = \int \frac{F x ds}{EI}$$

$$= \frac{F}{EI} \int_0^{\frac{\pi}{2}} (R \sin\theta) R \, d\theta$$

$$= \frac{FR^2}{EI} \Big[-\cos\theta \Big]_0^{\frac{\pi}{2}}$$

$$\theta_F = \frac{FR^2}{EI}. \tag{3}$$

*These calculations ignore the hoop stresses induced by the deflection of the gasket, but are probably adequate where the ratio of the inflation-tube radius, R, to the inflatable-gasket-end radius, r (see Fig. 15), is less than 0.1.

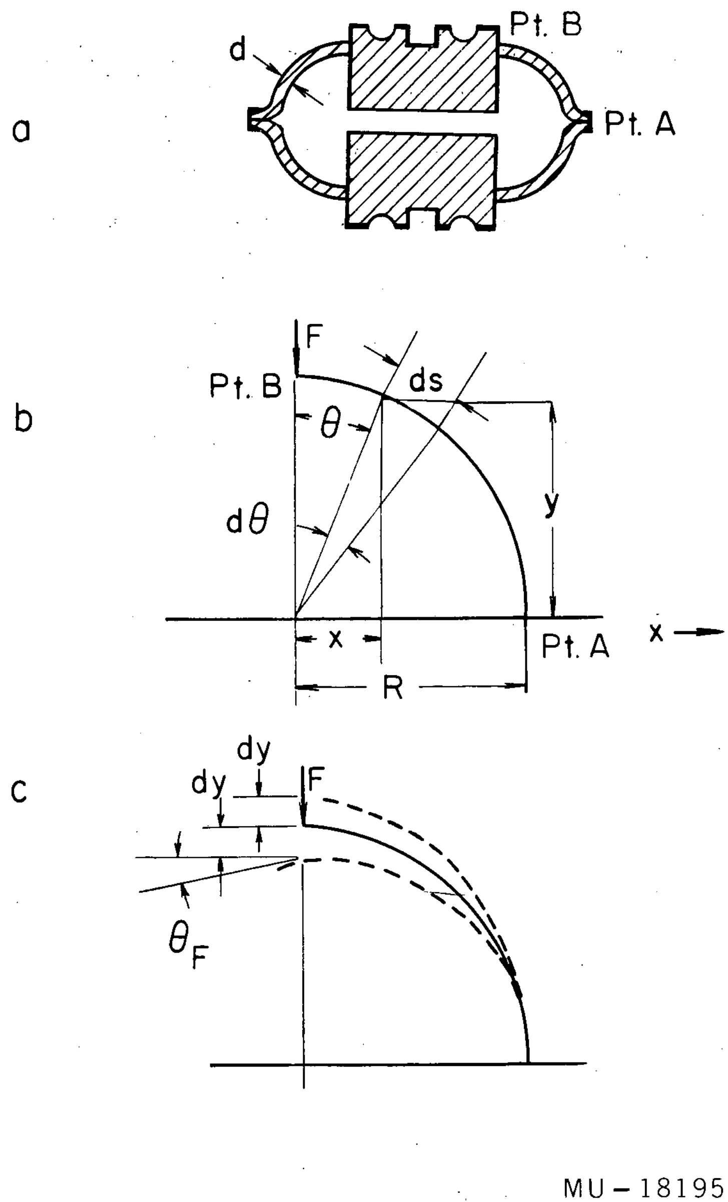


Fig. 18. Inflation-tube design which can be analyzed as a 90° curved beam, showing forces, moments, and deflections (A downward force is considered positive.)

To find θ_M we write

$$\theta_M = \int \frac{M_C ds}{EI} = \frac{M_C}{EI} \int_0^{\frac{\pi}{2}} R\, d\theta$$

$$\theta_M = \frac{\pi M_C R}{2EI} . \tag{4}$$

Setting Eq. (3) equal to Eq. (4), we obtain

$$\frac{FR^2}{EI} = \frac{\pi M_C R}{2EI} .$$

Solving for M_C, we have

$$M_C = \frac{2FR}{\pi} . \tag{5}$$

To find dy_M we write

$$dy_M = \int \frac{M_C x ds}{EI}$$

$$= \frac{M_C}{EI} \int_0^{\frac{\pi}{2}} (R \sin\theta) R\, d\theta$$

$$= \frac{M_C R^2}{EI} \int_0^{\frac{\pi}{2}} \sin\theta\, d\theta$$

$$dy_M = \frac{M_C R^2}{EI} . \tag{6}$$

Substituting the value obtained for M_C from Eq. (5) into Eq. (6), we have

$$dy_M = \frac{2FR}{\pi} \frac{R^2}{EI}$$

$$dy_M = \frac{2FR^3}{\pi EI} . \tag{7}$$

To find dy, the actual deflection, we subtract Eq. (7) from Eq. (1):

$$dy = \frac{\pi F R^3}{4EI} - \frac{2FR^3}{\pi EI}$$

$$= \left(\frac{\pi}{4} - \frac{2}{\pi}\right) \frac{FR^3}{EI}$$

$$dy = \frac{0.15\ FR^3}{EI} \quad . \tag{8}$$

To find the maximum bending moment, M_{max}, we write an expression for the total moment, M_t, at any point, x:

$$M_t = Fx - M_C$$

$$M_t = Fx - \frac{2}{\pi}\ FR. \tag{9}$$

Letting x = 0 (point B), we obtain

$$M_{t_B} = \frac{2}{\pi}\ FR$$

$$M_{t_B} = 0.636\ FR. \tag{10}$$

Letting x = R (point A), we have

$$M_{t_A} = FR - \frac{2}{\pi}$$

$$M_{t_A} = 0.364\ FR. \tag{11}$$

Thus we see that the maximum bending moment is at point B. Thus M_{max} equals 0.636 FR.

Therefore, to find the maximum bending stress per unit length, S_1, we write

$$S_1 = \frac{M_{max}}{Z}\ , \tag{12}$$

where Z is the section modulus. By definition Z equals $bd^2/6$, where b is the width of the beam and d is its thickness. Substituting this relationship for Z into Eq. (12), we have

$$S_1 = \frac{6M_{max}}{bd^2} = \frac{6(0.636)FR}{bd^2}$$

$$S_1 = \frac{3.816\ FR}{bd^2} \quad . \tag{13}$$

The deflection of one arm of the 90° curved beam is 2 dy in both compression and extension. Since the inflation-tube design of Fig. 14 uses two 90° curved beams, the total deflection of the gasket is 4 dy. Assuming a required accommodation of 160 mils, we write

$$4\ dy = 0.160\ \text{in.}$$

Solving for dy we obtain

$$dy = 0.040\ \text{in.} \tag{14}$$

Setting Eq. (14) equal to Eq. (8), we have

$$0.040 = \frac{0.15\ FR^3}{EI} \quad .$$

Solving for F, we obtain

$$F = \frac{0.040\ EI}{0.15\ R^3}$$

$$F = \frac{0.267\ EI}{R^3} \quad . \tag{15}$$

By definition I is $bd^3/12$; substituting this relationship for I into Eq. (15), we have

$$F = \frac{0.267\ E\ bd^3}{12\ R^3}$$

$$F = \frac{0.022\ E\ bd^3}{R^3} \quad . \tag{16}$$

For a beam of unit width (b = 1 in.), Eq. (16) becomes

$$F = \frac{0.022\ Ed^3}{R^3} \quad . \tag{17}$$

Substituting this value for F into Eq. (13), we have

$$S_1 = \frac{(3.816)(0.022)Ed^3R}{bd^2R^3}$$

$$S_1 = \frac{0.085\ Ed}{R^2}, \tag{18}$$

since b equals 1.

To find the tube thickness, d, we solve Eq. (18):

$$d = \frac{S_1R^2}{0.085E}. \tag{19}$$

Let us calculate d for a stainless steel tube, where E is 30,000,000, and where we will assume a maximum bending stress, S_1, of 60,000 psi. Then Eq. (19) becomes

$$d = \frac{60{,}000\ R^2}{0.085(30{,}000{,}000)}$$

$$d = 0.024\ R^2, \tag{20}$$

or that R is

$$R = 6.52\sqrt{d}. \tag{21}$$

The tensile stress, S_2, is

$$S_2 = \frac{RP}{d}, \tag{22}$$

where P is the inflation pressure. If the maximum inflation pressure is 1000 psi, then Eq. (22) becomes

$$S_2 = \frac{1000\ R}{d}. \tag{23}$$

2. 180° Curved Beam

Figure 19 shows the forces and moments in an inflation-tube design which is analyzed as a 180° curved beam. (This design is analyzed as a 180° curved beam because of the fixed slope at points A and B.) Here we have the following relationship:

$$\begin{aligned} x &= R - q \\ &= R - R\cos\theta \\ x &= R(1 - \cos\theta). \end{aligned} \tag{24}$$

To find the moment, M, produced by the downward force, F, we write

$$M = Fx. \tag{25}$$

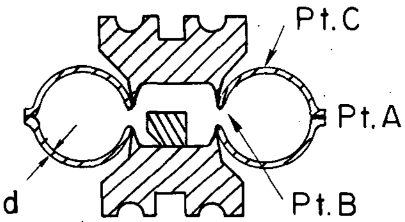
To find dy_F, the deflection produced by the downward force, we write

$$\begin{aligned} dy_F &= \int \frac{Mx}{EI}\, ds \\ &= \int \frac{Fx^2}{EI}\, ds = \int \frac{Fx^2}{EI}\, R\, d\theta \\ &= \frac{FR^3}{EI}\int_0^{\pi} (1 - \cos\theta)^2 d\theta \\ dy_F &= \frac{4.72\, FR^3}{EI}. \end{aligned} \tag{26}$$

To find θ_F, the angle to which the beam would bend downward if the slope were free to change, we write

$$\begin{aligned} \theta_F &= \int \frac{M}{EI}\, ds = \int \frac{Fx}{EI}\, ds \\ &= \frac{F}{EI}\int R(1 - \cos\theta)\, R\, d\theta \\ &= \frac{FR^2}{EI}\int_0^{\pi} (1 - \cos\theta)\, d\theta \\ \theta_F &= \frac{\pi FR^2}{EI}. \end{aligned} \tag{27}$$

a



b

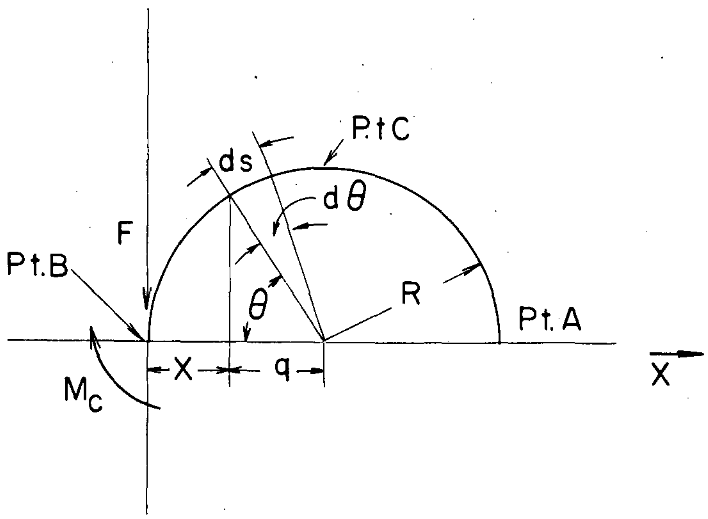


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Fig. 19. Inflation-tube design which can be analyzed as a 180^{o} curved beam, showing forces and moments. (A downward force is considered positive.)

Because the slope at point B is not free to change, the beam is bent upward by a correction moment, M_C, to an angle θ_M equal to θ_F. To find θ_M, we write

$$\theta_M = \int \frac{M_C}{EI}\, ds$$

$$= \frac{M_C}{EI} \int_0^{\pi} R\, d\theta$$

$$\theta_M = \frac{\pi M_C R}{EI} . \qquad (28)$$

Setting θ_M equal to θ_F, we obtain

$$\frac{\pi M_C R}{EI} = \frac{\pi F R^2}{EI} ,$$

which yields

$$M_C = FR. \qquad (29)$$

To find dy_M, we write

$$dy_M = \int \frac{M_C x}{EI}\, ds$$

$$= \int \frac{FR}{EI}\, R(1 - \cos\theta)\, R\, d\theta$$

$$dy_M = \frac{\pi F R^3}{EI} . \qquad (30)$$

To find the actual deflection, dy, we write

$$dy = dy_F - dy_M \qquad (2)$$

$$= \frac{4.72\, F R^3}{EI} - \frac{\pi F R^3}{EI}$$

$$dy = \frac{1.57\, F R^3}{EI} . \qquad (31)$$

To find the maximum bending moment, we first write an expression for the total moment, M_t, at any point, x:

$$M_t = Fx - M_C$$

$$M_t = FX - FR. \tag{32}$$

At point B, x is 0; Eq. (32) becomes

$$M_{t_B} = (-)\ FR.$$

At point C, x is R; Eq. (32) becomes

$$M_{t_C} = 0.$$

At point A, x is 2R; Eq. (32) becomes

$$M_{t_A} = FR.$$

Thus we see that the maximum bending moment, M_{max}, is FR.

Therefore, to calculate the maximum bending stress, S_1, we write

$$S_1 = \frac{M_{max}}{Z} \tag{12}$$

$$= \frac{FR}{bd^2/6}$$

$$= \frac{6\,FR}{d^2} \tag{33}$$

for a unit width (b = 1 in.). Next we write

$$Z = \frac{I}{c}\ , \tag{34}$$

where c is the distance from the neutral axis to the extreme fiber. Substituting Eq. (34) into Eq. (12), we have

$$S_1 = \frac{M_{max}(c)}{I}$$

$$S_1 = \frac{FR\ c}{I}\ . \tag{35}$$

Since c is d/2, we obtain

$$S_1 = \frac{FR}{I}\left(\frac{d}{2}\right).$$

Solving for F, we have

$$F = \frac{2S_1 I}{Rd} \quad . \tag{36}$$

Substituting this value for F into Eq. (31), we have

$$dy = \frac{1.570\, R^3}{EI} \quad \frac{2S_1 I}{Rd}$$

$$= \frac{3.14\, R^2 S_1}{Ed} \quad . \tag{37}$$

If we let the maximum bending stress, S_1, equal 60,000 psi, and if we use a stainless steel tube where the modulus of elasticity, E, is 30,000,000, then Eq. (37) becomes

$$dy = \frac{3.14\ R^2 (60,000)}{30,000,000\ d}$$

$$dy = \frac{0.00628\, R^2}{d} \quad . \tag{38}$$

Solving Eq. (38) for d, we obtain

$$d = \frac{0.00628\ R^2}{dy} \quad . \tag{39}$$

If we require a total deflection, 4 dy, of 160 mils, then the thickness, d, is

$$d = \frac{4(0.00628)R^2}{0.160}$$

$$d = 0.156\ R^2, \tag{40}$$

or that R is

$$R = 2.52\,\sqrt{d}\,. \tag{41}$$

The tensile stress, S_2, again is found by Eq. (23):

$$S_2 = \frac{1000\ R}{d} \quad , \tag{23}$$

for a maximum inflation pressure of 1000 psi.

PRINT LIST

7P5706	Inflatable gasket.
7P3176	Side elevation of the 72-in. bubble chamber (cross section).
7P3186	End elevation of the 72-in. bubble chamber (cross section).
7P3166	Plan view of the 72-in. bubble chamber (shield removed).

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