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^{16}N and ^8Li target chambers safety limits

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1 Introduction

This document calculates the safety limits (due to gas pressure) of the two target chambers built for housing the target material. The chambers were built for the production and transport of the radioactive nuclei (^{16}N and ^8Li) by way of capillaries all the way to the deck area.

We emphasise on the safety aspect of those chambers. Those having a small volume and being operated at low pressure (no more than 60 psiG), would fall into CSA Standard B51 under the classification "Registered as a category H fitting AND inspected by the manufacturer". Furthermore, the target chambers are exempt from "the Government of Ontario Boilers and Pressure Vessels Act" under sections:

- 2.d) a pressure vessel having a capacity of 1.5 cubic feet or less
- 2.f) a pressure vessel having an internal diameter of 6 inches or less

Once more, the target chambers are exempt from the scope of ASME Section VIII Division 1 under:

- U1 C-9) vessels having an inside diameter, width, height, or cross section diagonal not exceeding 6 inches, with no limitation on length of vessel or pressure.

2 Chambers description

The ^{16}N and ^8Li chambers are shown on figures 1 to 17. It is to be noted that the drawings are scaled to actual size. Each chamber is basically made out of 2 cylindrical shells held in place by two plates at each ends. The chambers are made of Stainless Steel 303 and 304. In the following calculations we used austenitic steel (SS 304, 18% Cr 8% Ni). The maximum operating gas pressure is 60 psiG. The most sensitive part of the chambers to gas pressure are the inside cylinder because of their small wall thickness.

3 Preliminary calculations

3.1 Collapsing pressure

The design calculations are taken from Mark's Standard Handbook for Mechanical Engineering, 9th edition. The collapsing pressure for a cylinder is given by:

$$W_c = k \cdot E \cdot \left(\frac{t}{d}\right)^3 \quad (1)$$

Parameters	¹⁶ N chamber		⁸ Li chamber	
	Inner Cyl.	Outer Cyl.	Inner Cyl.	Outer Cyl.
Diameter (in)	1.850	4.250	1.810 A/F	4.200
Lenght (in)	3.110	4.500	3.110	4.500
Thickness (in)	0.050	0.125	0.030	0.100

Table 1: Dimensions used in the calculations for stress values for the target chambers.

where E is Young's modulus and is equal to 27.6×10^6 psi, t is the shell's thickness and d is the outside shell diameter. k is a factor depending on the ratio of length to radius and on the ratio of diameter to thickness of the cylinder. Table 1 gives the physical dimensions of the two chambers. We used $k = 16$ for both chambers (see table 5.2.64 in the above reference). Therefore we readily obtain values for the buckling or collapsing pressure of the inner cylinders of

$$W_c(^{16}N) = 8700 \text{ psi}, \quad (2)$$

$$W_c(^8Li) = 2000 \text{ psi}. \quad (3)$$

Values which are well above the maximum operating pressure of the chambers. The collapsing pressure for ⁸Li is smaller than the one for the ¹⁶N due to the fact that the ⁸Li inner cylinder is thinner. In fact, the ⁸Li inner cylinder as an hexadecimal shape on the outside but a circular shape on the inside. So, for the benefit of the calculations, we have used the formulás for cylindrical shape plus the minimum thickness encountered. Hence, the result is a lower bound of the actual collapsing pressure.

3.2 Stress inside the Stainless Steel due to internal pressure

The outside cylinders are subject to internal pressure. The stress inside the stainless steel due to internal pressure is given by

$$S = \frac{P \cdot r}{t}, \quad (4)$$

where P is the pressure, r the radius of the outer cylinder and t the thickness. With $P_{max} = 60$ psiG we find immediatelly

$$S(^{16}N) = 1020 \text{ psi}, \quad (5)$$

$$S(^8Li) = 1260 \text{ psi}. \quad (6)$$

Those are to be compared with the yield strenght S_Y (permanent deformation) and the tensil strenght S_T (rupture) of stainless steel from table 5.11 of Mark's.

$$S_Y = 30\,000 \text{ psi}, \quad (7)$$

$$S_T = 85\,000 \text{ psi}. \quad (8)$$

4 ASME codes and standards

The American Society of Mechanical Engineers (ASME) codes and standards section VIII division 1 formulates standard rules for the construction of steam boilers and pressure vessels. We have used some elements of the code to get estimates on the safety of the target chambers. The chambers were built by STC in Ottawa, which is recognized for its machining quality but is not an ASME shop since they do not have the obligatory tracking of material as demanded by ASME.

The chambers were made of stainless steel 303 and 304. Both types of stainless steel being very close in mechanical properties. SS303 is non-code material so we have used SS304 properties as given in the specification for base materials in ASME section II, to perform the stress calculations.

4.1 UG23 - Maximum allowable stress values

The maximum allowable stress value, defined as the maximum unit stress permitted in a given material used in a vessel constructed under the section VIII rules, are given in subpart 1 of section II part D. From table 1A one finds that the maximum allowable stress values range from 14 000 to 17 000 psi for stainless steel 304¹.

The maximum allowable longitudinal compressive stress is defined as the smaller of the following values

- the maximum allowable tensile stress value permitted (for a lower bound we will choose 14 000 psi)
- the value B as calculated by the procedure outlined in UG23. In doing so and with reference to table HA1 we find $B = 14\,000$ psi. This value depends on the wall thickness of the cylindrical shell. It represents the upper limit of stress permitted and cannot exceed the allowable tensile stress value. Otherwise a larger value must be chosen for the thickness of the shell.

So for the above calculation, the maximum allowable stress value of 14 000 psi will be chosen.

4.2 UG27 - Thickness of shells under internal pressure

For cylindrical shells, the maximum allowable working pressure is the lesser of the circumferential stress or the longitudinal stress.

¹All quoted values are for the temperature range at which the chambers are operated i.e. -20 to +100 °F

4.2.1 Circumferential stress

The circumferential stress is calculated as

$$P_C = \frac{S \cdot E \cdot t}{R + 0.6t}, \quad (9)$$

where R is the inside radius of the shell, S is the maximum allowable stress value as given in UG23 and E is the joint efficiency which is 1 since there are no joints in the target chambers.

$$P_C(^{16}N) = 840 \text{ psi}, \quad (10)$$

$$P_C(^8Li) = 680 \text{ psi}. \quad (11)$$

4.2.2 Longitudinal stress

The longitudinal stress is calculated as

$$P_L = \frac{2 \cdot S \cdot E \cdot t}{R - 0.4t} \quad (12)$$

$$P_L(^{16}N) = 1800 \text{ psi}, \quad (13)$$

$$P_L(^8Li) = 1400 \text{ psi}. \quad (14)$$

4.3 UG28 - Thickness of shells and tubes under external pressure

The maximum allowable external working pressure is given in UG28 by

$$P_a = \frac{4 \cdot B}{3(D_0/t)}, \quad (15)$$

where D_0 is the outside diameter of the shell. B is determined from the ratios of outside diameter to thickness of the shell and length to outside diameter of the shell. From the code we readily find $B = 13\,000$ psi. Hence the maximum allowable external pressures are

$$P_a(^{16}N) = 470 \text{ psi}, \quad (16)$$

$$P_a(^8Li) = 290 \text{ psi}. \quad (17)$$

5 Conclusion

The results displayed in table 2 and 3 clearly demonstrate that the target chambers maximum operating pressure of 60 psiG is well below the maximum operating pressures derived from the ASME division VIII code. As expected, it is the inner cylinder

	¹⁶ N chamber	⁸ Li chamber
Collapsing press. (psi)	8700	2000
Max. allowable stress (psi)	14 000	14 000
Max. external press. (psi)	470	290

Table 2: Results of stress values for the inner cylinders of the target chambers. The top row are the results from Mark's and the bottom rows are from the ASME calculations.

	¹⁶ N chamber	⁸ Li chamber
S (psi)	1020	1260
S_Y (psi)	30 000	30 000
S_T (psi)	85 000	85 000
Max. allowable stress (psi)	14 000	14 000
Max. internal press. (psi)	840	680

Table 3: Results of stress values for the outer cylinders of the target chambers. The 3 first rows give the results from Mark's and the 2 bottom rows are from the ASME calculations.

of both chambers which is the limiting factor of the operating pressure. Even so, the maximum operating pressure is almost 8 times less than the maximum allowable pressure for the ¹⁶N chamber and almost 5 times less than the maximum allowable pressure for the ⁸Li chamber. So that we can safely conclude that the target chambers are operated well below the safety limits as drawn by ASME.

It is to be noted that there is more than an order of magnitude difference between the collapsing pressure as calculated by Mark's and the maximum allowable external pressure as calculated using the ASME code (see table 2). This strongly suggest that the ASME code's calculation already contains several safety factors that would account for the difference. Unfortunately the code is unclear as when and what safety factors are used.

Both chambers have now been operated regularly over the past year at a pressure of 60 psiG. Some test were done at 100 psiG for several hours. No problems were reported. No leaks were measured.

