EVALUATION OF IRON - BASED, STRESSED - SHELL CONTAINER

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ABSTRACT

The Iron-Based, Stressed-Shell container is being developed for immobilization and disposal of used fuel bundles from CANDU Nuclear Power Generating Stations, in an up to 1000-m deep underground vault. In this design, the bundles are placed in a basket, which is then placed in a thick steel cask covered by a thin titanium shell. The steel cask forms the structural component of the container, while the titanium shell provides corrosion protection. The outer surface of the container will be tapered to allow for a good fit between the two shells. A novel press-fit, self-locking and (possibly) weld-free closure has been developed. The closure provides an easier and a more reliable automated assembly process in the remote environment of a hot cell than a full penetration weld.

INTRODUCTION

The Iron-Based, Stressed-Shell (IBSS) Container is one of several containers considered for the immobilization and disposal of unreprocessed used fuel bundles from CANDU nuclear generating stations, in an underground vault[1]. The vault will likely be built in an up to 1000 m deep cavern in a plutonic rock formation in the Canadian Shield. Besides disposal, some possibilities for other, more general applications of the IBSS container have been considered[2].

A layout of the container is shown in Fig. 1. Its main component is a steel cask about 45 mm thick. The cask has an outside surface sloped to allow for easy overpacking with a 4.76 mm thick titanium shell. The bottom and top plates consist of two layers: a 76 mm-thick steel plate on the inside and 4.76 mm-thick titanium plate on the outside. The titanium shell is provided for corrosion protection only. The steel cask will be overpacked with the titanium shell before the container is placed in a hot cell where a basket with bundles will be placed inside it. The container is expected to reduce radiation on the outside surface significantly.

Several alternative designs are considered because several generic studies directly effecting the design have not been concluded.

DESIGN CONDITIONS

The container is designed to meet the following criteria representing the worst anticipated vault conditions:

1. The container should resist perforation by corrosion for at least 500 years in the highly saline water of the vault;

2. The container must be able to support an external pressure of 10 MPa in the event the vault is flooded with a 1000 m head of water.

3. The container will operate in temperatures of up to 150 C, because of the heat generated by the fuel bundles.

4. The container must be able to support an internal pressure of 0.1444 MPa that can form due to the temperature increase by 130°C.

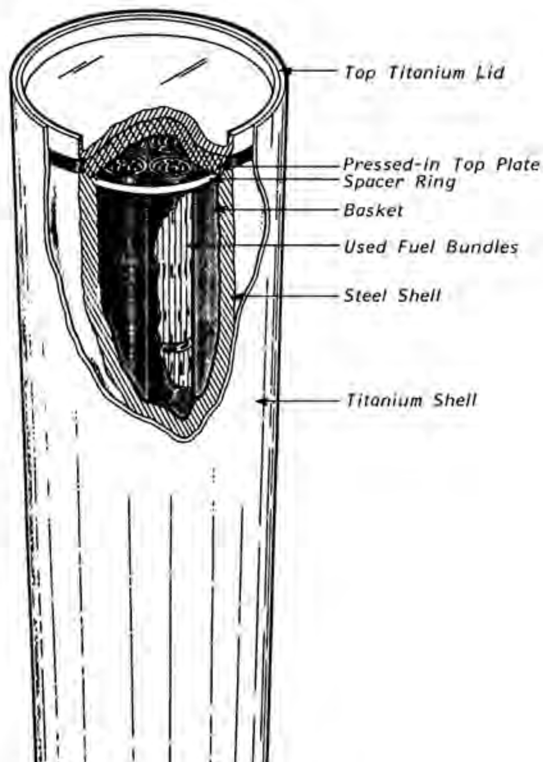


Fig. 1. IBSS Container - Layout Diagram.

CONTAINER DESIGN

The Iron-Based, Stressed-Shell (IBSS) container consists of a low-carbon steel basket assembly (see Fig. 1) which rests inside a steel cask. The basket has a capacity of 72 bundles. The container consists of two concentric layers of metallic shells: a thick mild steel cask provides the required strength while a thin titanium outer shell provides corrosion protection (see Fig. 1). The container is about 2250 mm high and has an inside diameter of 610 mm. The outside surface has a taper of about 0.15 to allow for easier overpacking of the steel canister by the titanum shell.

Basket

The basket consists of nineteen 4-in Schedule 10 tubes concentrically arranged, as illustrated in Fig 2. Each pipe (except the centre pipe) can contain four fuel bundles. A spoked wheel arrangement at the bottom of the basket supports the bundles inside the pipes.

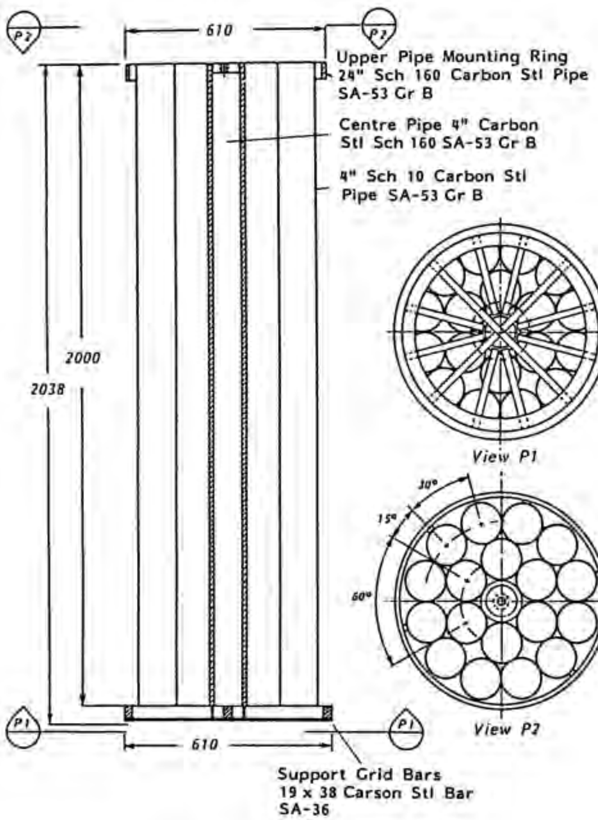


Fig. 2. Basket Layout.

Steel Cask

The thickness of the steel cask's cylindrical shell varies from 44.5 mm at the bottom to 50 mm at the top, as shown in Fig. 3. The shell could be made by centrifugal casting. The bottom plate will be 76 mm thick and would be welded to the cylindrical shell with a full penetration weld. The top of the cylindrical shell would be prepared for the press-fit closure.

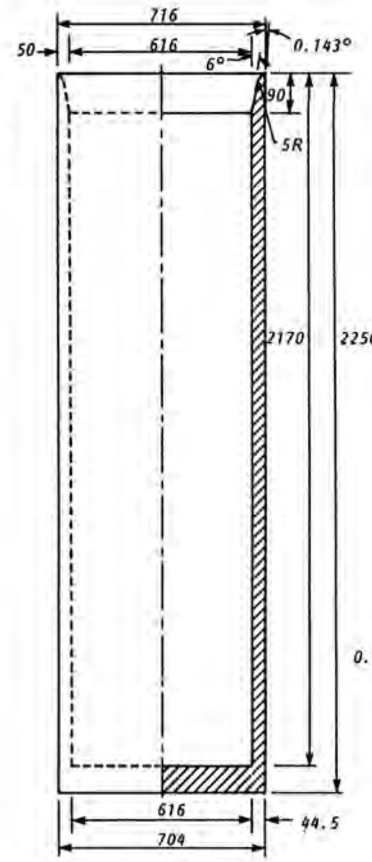
The top plate would be machined from a 76 mm-thick steel plate to match the press-fit closure dimensions.

Titanium Shell

The titanium shell would be made of 4.76 mm thick B265 grade 2 titanium plate (see Fig. 4). The bottom and top plates would be formed by pressing. The bottom plate would be pressed in place when the steel cask is inserted into the shell and welded in place using resistance bonding.

Container Handling

We expect that the container would be handled in both the horizontal and upright positions. The horizontal handling may be preferred while the container is being loaded with fuel and during transportation in the vault. Other processes, including emplacement of the container in a borehole requires upright handling.



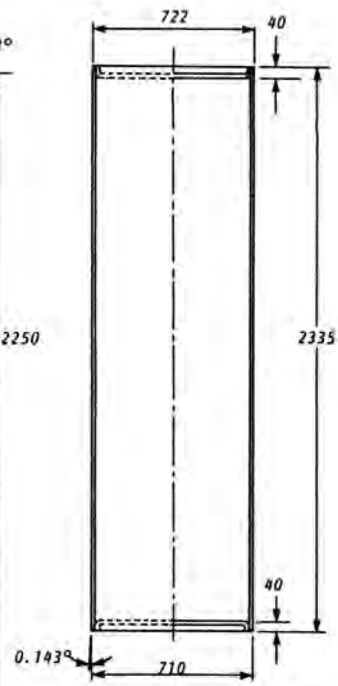


Fig. 4. Titanium Shell.

Fig. 3. Steel Cask Design.

When handled in the vertical orientation, the container can be lifted using the ring at the top of the container. The gripping could be provided by three clamps positioned 120° apart. Horizontal handling may be provided by a supporting cradle which will also be used to rotate the container between horizontal and vertical positions.

If a backup handling method is required a large electromagnet could be used to lift the container from either vertical or horizontal orientation.

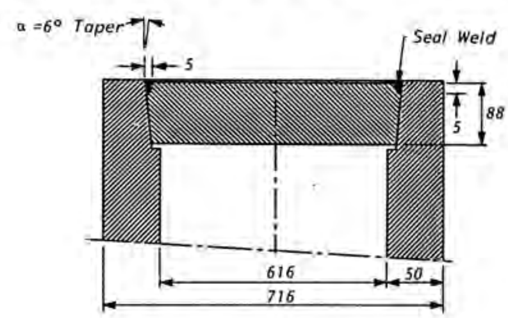


Fig. 5. Seal-Welded Closure.

CLOSURE SYSTEM

Design

We considered several closure designs for sealing the IBSS container in the hot cell. The conventional system has a full penetration weld between the top lid and the cylindrical shell. This weld would be 76 mm thick. The difficulty in making or repairing such a weld in a hot cell makes it unattractive. A novel closure system design which requires only limited welding is proposed.

The closure is based on press-fitting a slightly tapered lid into a similarly tapered area at the top of the steel cask. The taper should not exceed 6° to provide the self-locking effect. The results of our preliminary tests indicate that the closure has adequate strength, but its leak rate, although low, may be unacceptable for disposal. The final closure arrangement may have a small seal weld shown in Fig. 5.

Leak Rate

A test program was conducted to determine if the closure could be made leak-tight without welding. The leak-rate seems to decrease as:

- the thicknesses of lid and cylindrical shell increase
- the accuracy of machining improves.

The test program was performed using 6 inch pipes of various schedules and using different plate thicknesses. The lowest leak rate was below 10^{-6} atm mm³/sec and was measured on a Schedule XX pipe with a 63.5 mm thick lid. The highest leak rate was measured during the tests of 6-inch schedule 80 pipes closed with 25 to 50 mm-thick lids. It was about 1 atm mm³/sec. This high leak rate was found to be highly localized, suggesting a manufacturing flaw or contamination of surfaces in contact. The results of these tests are presented in more detail in Table I.

TABLE I

Leak-Rate Test Results

Test No*	Pipe (in)	Sched	Top Plate Thickness (mm)	Closure Force (kN)	Leak Rate** atm mm³/sec
1.1	6	10	25	156	4.8
2.1	6	40	25	178	0.011
3.1	6	80	25	245	4.8
3.2	6	80	25	267	1.0
3.3	6	80	38	334	1.0
3.4	6	80	51	356	1.0
4.1	6	XX	63	602	10^{-6}
4.2	6	XX	63	614	10^{-6}

Notes: * Test Number - first digit - pipe size or schedule
second digit - plug size or sample number

** Leak can be detected between 5 and 10^{-6} atm mm³/sec

The small seal weld is not expected to require inspection because it is introduced as the second seal. A thin metallic foil could be used to reduce the leak-rate. To study this, a single test was performed using aluminium foil. Its results were inconclusive. A few small and highly localized leaks were detected. Tin foil may be a potential alternative. The leak rate tests are continuing.

Closure Strength

The closure is self-locking, meaning that a force is required to open it. The force to open the closure is dependent on: the force used to press the lid into the cylinder, friction between the shell and the lid, and on the slope between the contacting surfaces, as discussed below. Both force and internal pressure were used to open the closure. The force ranged from 93 KN to 180 KN and internal pressures ranged from 2.5 kPa to 20.7 MPa, as described in Table II. The opening force is usually between 0.75 and 0.5 times of the force used to to seal the closure.

TABLE II

Closure Strength

Test No	Pipe Size (in)	Schedule	Top Plate Thickness (mm)	Surface Finish (RMS)	Closure Force (kN)	Load to Open: Force (kN)	Load to Open: Pressure MPa	Friction Factor
1.1	6	10	25	125	155	93.4	2.50	0.42
2.1	6	40	25	125	191	45	-	0.17
3.1	6	80	25	125	290	-	7.24	0.30
3.2	6	80	25	125	245	-	6.55	0.33
4.1	6	XX	64	125	614	-	20.70	0.49

Corrosion Consideration

The basic corrosion protection will be provided by the titanium shell. If this shell is breached at the closure, the crevice between the surfaces in contact would be conducive to corrosion. The weld and the compressive stresses at the mechanical seal would provide some (limited) protection.

Analysis

A 6-inch Schedule 80 pipe with a 38 mm closure plate has been selected for an analytical evaluation of the closure. The assumed force diagram is shown in Fig. 6. Based on the analysis, the force (P) required to press the lid in is a function of a friction coefficient (μ) and angle (α):

$$P = 2\pi rF\ (\mu + \alpha)$$

where force F is the force between the contacting surfaces, shown in Fig. 6.

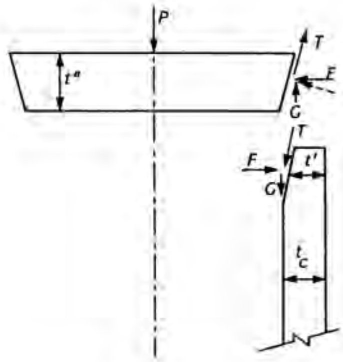


Fig. 6. Force Diagram of the Closure.

During removal of the lid, the direction of the friction force reverses. Thus the pull-out force is equal to

$$W = 2\pi rF\ (\mu - \alpha)$$

The ratio of the two forces depends only on the angle of contact surfaces and friction coefficient:

$$P/W = (\mu + \alpha)/(\mu - \alpha)$$

Friction Factor

The self-locking closure operates on friction. The contacting surfaces should be carefully prepared to obtain a suitable friction factor. The friction factor calculated from our tests ranged between 0.30 and 0.45 for a rough machined surfaces in an as-machined condition (machining oil not removed). As friction coefficient is reduced, the ratio of P/W increases reducing the strength of the closure. For this reason, a system where the friction coefficient is below 0.3 is not recommended. On the other hand, increasing the friction coefficient above 0.6 will not improve the ratio significantly and it may result in low contact stresses. Higher forces would be required for sealing such a connection and it could be more sensitive to any defects, making tolerances more stringent. The measured range of friction coefficients is considered the most suitable.

Angle of Contact

The angle of contact between the shell and the lid determines the strength of the self-locking closure. For example, if the friction factor is 0.4, the pull-out force W becomes zero for the angle α equal to 22°. As the angle is reduced, the ratio of closing and pull-out forces P/W is reduced. An angle of 5° to 6° is considered optimum.

Contact Stresses

Average radial stresses between the two surfaces in contact are generally low, while the shell hoop stress at the closure is expected to be above the yield strength of the material. In case of the 6-inch Schedule 80 specimen with a 38 mm lid, the radial stress did not exceed 51 MPa and the hoop stress was above 380 MPa, assuming a linear elastic behaviour (yield stress is equal to 245 MPa). The effect of the friction factor and angle of contact on forces and stresses is shown in Table III.

TABLE III

Closure Loads

Friction Factor	Angle δ(°)	Force to Close P(kN)	Force to Open W(kN)	P/W
0.8	6	100	77	1.3
0.6	6	100	70	1.4
0.45	6	100	62	1.6
0.3	6	100	48	2.1
0.1	6	100	0	–
0.3	17	100	0	181.1
0.3	10	100	26	3.8
0.3	6	100	48	2.1
0.3	3	100	70	1.4
0.3	1	100	89	1.1

STRESS ANALYSIS

Stability Consideration

The largest hydrostatic loading postulated for the container is an external pressure of 10 MPa anticipated in a flooded 1000 m deep vault The minimum thickness of the steel shell is determined by its strength to resist buckling. Paragraph 3133 of Section III, Division 1 of the ASME Boiler and Pressure Vessel Code requires that this thickness be not less than 44.5 mm. The reinforcing effect of the end plates was not included in the stability analysis.

Analysis of External Pressure

An approximate stress analysis of the closure was presented above. The analysis indicate a tensile radial stress at the rim of the closure to be equal to 50% yield and the hoop stress at yield. Since the top plate can be considered rigid in the radial direction, a tensile stress is expected in the cylindrical shell at the top closure plate. This tensile stress is amplified further by the large bending moments present at both top and bottom discontinuity areas. Although the connection between the top plate and the shell is only mechanical, it will be assumed rigid for the purpose of this analysis.

The stress analysis is based on the classical approach using discontinuity equations to determine the interaction loads and stresses. The mathematical model of the container and areas where stresses are calculated are shown in Fig. 7.

The titanium shell is assumed to have no strength. Its deformations will be the same as those in the steel cask.

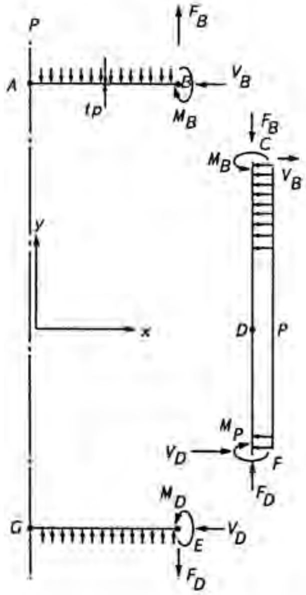


Fig. 7. Load Diagram.

TABLE IV

Summary of Stress Distribution in Steel Shell External Pressure

*	Stresses (MPa)			Deformations (mm)		
	σr	σa	σ_h or τ	Sr	Sa***	θ(°)
A	150	-5	0	0.0	-1.1	0.0
B	-145	-5	24	0.0	-0.5	0.08
C	-5	-320	33	0.0	-0.5	0.08
D	-5	-38	-75	-0.133	-0.25	0.0003
E	-130	-5	23	0.0	0.0	0.1
F	-5	-357	35	0.0	0.0	0.1
G	108	-5	0	0.0	0.6	0.0

* Location at which stresses and deformations are calculated.

** σ_h applies to location D. τ applies to all other locations.

*** Total deformation relative to the rim of bottom plate.

The deformations and stresses in the steel cask are shown in Table IV. The largest stress is anticipated in the cylindrical shell at the bottom plate. On the outside surface the stress is compressive and equal to 360 MPa, assuming linear elastic behaviour in the post-yield region. According to the Code this stress is produced by the secondary bending and is limited to 3 Sm or 390 MPa (ultimate strength). The stress in the cylindrical shell at the *top plate* is the largest in the axial direction and ranges from tensile 250 MPa on the inside to compressive 325 MPa on the outside. The circumferential stress is tensile at 195 MPa, due to the pressing of the lid. The stress intensity on the inside surface is tensile at 390 MPa. The stress intensities in the flat plates are below 150 MPa, while the allowable stress is equal to 195 MPa.

In summary, all stresses evaluated in the steel cask, for the external pressure of 10 MPa, are below the limiting values recommended by the ASME Pressure Vessel Code.

Stresses in the titanium shell will be caused by the deformation of the steel shell (as detailed in Table IV) and by the differential thermal expansion between steel and titanium. The differential thermal expansion in the axial direction is 1.12 mm and 0.18 mm in the radial direction for 130°C temperature increase.

The anticipated stresses in the titanium shell are low in comparison with allowable values, as shown in Table V. The largest stress intensity, including both thermal and mechanical effects, is expected in the cylindrical shell away from discontinuity and is expected to be compressive at below -150 MPa. Since this stress includes thermal effects, the allowable stress is 3 Sm or 252 MPa.

TABLE V

Summary of Stress Distribution in Titanium Shell External Pressure

Location	Stresses (MPa)			Deformations (mm)		
	σr	σa	σ_h	δr	δa	$\theta(°)$
A	-56	-5	-56	0.0	-1.1	0.0
B	-56	-5	-56	0.0	-0.5	0.08
C	-5	-45	-42	0.0	-0.5	0.08
D	-5	-35	-60	-0.133	-0.25	0.0003
E	-56	-5	-56	0.0	0.0	0.1
F	-5	-45	-42	0.0	0.0	0.1
G	-56	-5	-56	0.0	0.6	0.0

The largest stress due to the external pressure alone is in the cylindrical shell away from discontinuity and is equal to -37.2 MPa, while the yield stress is 175 MPa. Stresses at discontinuities and in the flat plates are below 2 MPa.

Analysis of Internal Pressure

The internal pressure in the container may reach 0.1444 MPa if the temperature of the container increases from 20°C at the time of its sealing to 150°C in the vault. The external pressure is assumed to be negligable. The closure tests have shown that the steel cask can support this pressure. However, if a leak in the cask allows the pressure to be released, the resulting stresses in the titanium shell will be high but within the allowables, as shown in Table VI.

TABLE VI

Stresses Due to Internal Pressure in Titanium Shell

*	Stresses (MPa)			Deformation		Loads (kN/mm) or (kN)*		
	σr	σa	σ_h or τ	δ(mm)	$\theta(°)$	V	F	M
A	140	0	0	13.5	0.0	0.0	0.0	526
B	185	0	2	0.0	0.7	21.7	9.0	683
C	0	183	5	0.0	0.7	21.7	9.0	683
D	0	4	7	0.05	0.0	0.0	0.0	0
E	180	0	2	0.0	0.8°	21.3	7.8	661
F	180	0	2	0.0	0.8°	21.3	7.8	661
G	132	0	0	12.2	0.0	0.0	0.0	479

* Location where stresses and deformations are calculated.

Stresses During Lifting

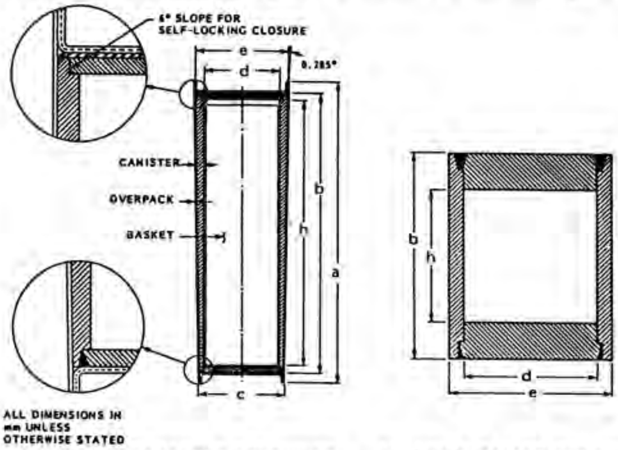
The container with 72 bundles of used fuel is expected to weigh about 42 kN (10,000 lbs). Due to the tapered outer shell, the load will not be transmitted to the bottom lid. Stresses in the titanium shell need only be calculated in two areas: tearing of the lifting ring and uniform stress in the titanium shell near the top of the container. The shear stress in the titanium shell at the supporting ring is 41 MPa. The allowable shear stress is equal to 0.6 Sm or 50 MPa. The uniform stress in the titanium shell is within 5 MPa.

ALTERNATIVE DESIGNS

The design proposed so far is the "reference design". We are proposing several alternative container designs within the IBSS project, because a final decision has not been made regarding:

- the preferred bundle capacity - 72 or 96;
- the preferred container geometry;
- the actual corrosiveness of vault environment;
- the preferred corrosion protection system;
- the preferred material: steel or cast iron.

Two basic types of containers are proposed. If the decision is made that a low strength, corrosion resistant shell is required, the above discussed container made of steel or cast iron will be used with a protective overpack. However, if the vault environment is less corrosive, an additional thickness of steel or cast iron might be provided as a corrosion allowance. In this case, our container could be 50 to 150 mm thick, with up to 100 mm allowed for corrosion. The titanium overpack would not be required in this arrangement. Dimensions and configurations of several alternative container designs are illustrated in Fig. 8.



a) OVERPACKED CONTAINER b) THICK-WALLED CONTAINER

CONTAINER SIZES

TYPE OF CONTAINER*	NUMBER OF BUNDLES	CONTAINER DIMENSIONS (mm)					
		a	b	c	d	e	h
50 mm SHELL	72	2335	2255	710	616	722	2080
50 mm SHELL	96	2335	2255	825	720	837	2080
150 mm SHELL**	72	-	2380	-	616	816	2080
150 mm SHELL**	96	-	2380	-	720	1050	2080
6 - LAYER**	96	3350	3250	655	558	669	3125

* CONTAINERS HAVE FOUR LAYERS OF BUNDLES EXCEPT AS STATED

** SHELL IS NOT TAPERED. (SEE FIGURE b)

Fig. 8. Approximate Container Sizes and Layouts.

Material Selection

AISI 1020 L steel is currently proposed for the "reference design" although it may not be the most suitable selection. A metallurgical evaluation should be conducted to determine if steel or cast iron would be more suitable. Steel offers several advantages: higher strength, lower brittleness, easier welding, etc. Cast iron has lower cost and is easier to cast.

Bundle Capacity

Most of the containers currently considered in the Canadian program have the capacity of 72 bundles arranged in four layers of 18 bundles per layer. This arrangement will allow for a convenient size container having a small diameter and acceptable height. A diagram of a 72-bundle tube layout is illustrated in Fig. 9.

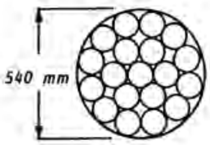


Fig. 9. Test Arrangement in 72-Bundle Basket.

Bundles are stored at the stations in modules of 96 bundles per module therefore, it would be advantageous if the disposal container had the same capacity of 96 bundles. Two baskets having the capacity of 96 bundles are illustrated in Fig. 10.

Fig. 10. Tube Arrangement in 96-Bundle Basket.

Integration

The container is also being considered for use in integration of several stages of the used fuel disposal scenario including transportation and storage, as illustrated in Fig. 11.

POTENTIAL ADVANTAGES OF THE DESIGN

The container offers several potential advantages:

- dual engineered barriers increasing container durability, strength and corrosion protection.

- low stresses in the titanium shell reduce the posibility of stress corrosion cracking of titanium.

- the large empty space inside the container could be filled with radionuclide retaining substances to minimize radionuclides migration when the container is ultimately penetrated.

- retrievability of the container is possible.

- high strength container for the rough handling environment possible in the vault offers an additional safety.

- the container offers significant radiation shielding.

FUTURE ACTIVITIES

The IBSS developmental program is expected to continue for a few more years. The following items are planned for the near future:

1. Conduct a detailed stress analysis of the reference container design and the closure system.

2. Continue the closure test program, including evaluation of the effects of:

   - various friction coefficients,
   - various angles of contact,
   - surface finish,
   - impact loading,
   - repeatable opening and closing.

3. Further refinement of the reference and alternative designs.

CONCLUSIONS

The concept of the Iron-Based, Stressed-Shell Container is well developed. The design is anticipated to offer a technically viable and economical container for disposal of the used fuel from the CANDU Nuclear Power Stations.

The IBSS concept is flexible, allowing for easy implementation of changes which might be introduced to the program in the future. The concept is technically viable and does not require development of new technology.

Several alternative designs are also proposed to account for possible changes as more information becomes available from the geological and metallurgical programs.

ACKNOWLEDGEMENTS

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