Response of Calandria Vessel to Impulse Pressure Loading

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INTRODUCTION

The calandria vessel in a CANDU(CANada Deuterium Uranium) reactor contains the heavy water moderator and provides structural support to the fuel channels and other in core devices such as shut off rods. The calandria vessel is made up of a stepped cylindrical shell with a main shell and two sub- shells. The two sub shells are connected to the main shell by means of the annular plates at one end and to the calandria tube sheets at the other ends to form a closed shell to hold heavy water moderator. Under normal operating condition the calandria vessel is subjected to the internal pressure of the cover gas and to the dead weight of the moderator, fuel, in core devices and fuel channels. It is designed as a class III pressure vessel based on ASME code. A schematic of the calandria vessel along with the pressure boundary is shown in Fig 1.

In the accident analysis of CANDU reactors, under the class of single channel events, a number of accidents such as a simultaneous failure of pressure and calandria tube under normal operating condition and failures of a channel due to large fuel temperature excursions are considered. The channel failure causes a severe hydrodynamic transient within the moderator due to the discharge of high temperature coolant and/or fuel. The hydrodynamic transient in the moderator causes a short pressure transient in the calandria vessel. A worst-case pressure transient in the moderator following either a single channel event failure or due to multiple channel failure is shown in Fig 2. Under the accident condition, it is required to show that the calandria vessel can withstand the pressure transient and that the integrity of the core is assured.

STATIC STRESS ANALYSIS AND COLLAPSE PRESSURE.

To identify the highly stressed regions of the calandria vessel, a static stress analysis has been carried out using the computer code ABAQUS using the large displacement/ small strain and material nonlinear capabilities. For analysis purpose it is assumed that the calandria vessel is an axisymmetric shell which is rigidly joined to the calandria tube sheets. It is also assumed that the welds at the junctions with the annular plate are flush on both sides and hence do not introduce any discontinuity in structure. The material and geometry data of a typical calandria vessel is given in Table-1. To account for the material non-linearity in the analysis the following Ramberg-Osgood form of the stress-strain (σ - ε) relation is used.

$$\varepsilon = \sigma + A\,\sigma^n \qquad (1)$$

where ε = non-dimensional strain= $E\varepsilon/\sigma_y$

$\sigma = \sigma/\sigma_y$

σ_y = yield stress

E = Young's modulus

A and n are numerical constants.

A static stress analysis of the shell, within the elastic limit, indicates that the junction of the annular plate with the main shell and the sub-shell are the highly stressed points with the stress at the junction being about 1.5 times the stress in the main body of the shell. Hence it is realistic to assume that under elastic and small plastic stain limits the strain at the junction will be 1.5 times the strain in the main body of the shell. As the stresses at the junction are higher, under pressurisation by static or dynamic loading, these joints will reach the failure limit first. A lower bound estimate of the collapse pressure of the calandria vessel is obtained based on the static limit load analysis as per the rule NB.3213.25 of ASME pressure vessel code (Ref 1). The results of the limit load analysis, given in Fig 3, indicate that the maximum plastic strain (Fig 3c) at the junction (point B) is around 0.5 percent at the expected limit load of 2.7 MPa. The results also indicate that the rate of plastic straining at this point increases very rapidly when the internal pressure exceeds about 3.5 MPa level. Hence this pressure can be considered to be a realistic estimate of the gross failure pressure for calandria vessel for static loading.

Effects of Dynamic Loading

In order to assess the effects of dynamic loading a conservatively severe pressure transient is considered. The dynamic analysis was again carried out using the ABAQUS code. The pressure transient considered and the predicted plastic strain at the junction points are shown in Fig 4. These results indicate that up to about 3 MPa pressure there is little plastic strain. Beyond about 4 MPa the plastic straining at the point B is seen to be very rapid and hence this is assumed to be the dynamic failure pressure. Comparing the failure pressure predicted by the static stress analysis with the dynamic pressure required for plastic straining, it is concluded that the dynamic effects of the pressure loading tends to increase the failure pressure (by about 15%) due to inertia effects of the shell.

RESPONSE TO PRESSURE PULSE

As the assumed pressure loads due to multiple channel failure are of large magnitude and short duration (Fig 2b), these can be treated as impulse loads causing elastic/plastic deformation. The response of the shell is considered to be elastic even though the critical points may undergo some plastic strain. The effect of these loads is estimated by treating them as idealised impulse loads with known specific impulse and by applying a correction factor to take into account the finite magnitude of the pulse width as in Ref 2. As the correction factor is strongly dependent on the relative duration of impulse compared to the period of natural vibration, an estimate of the periods of vibration is carried out first.

Calculation of Frequencies of Vibration

The basic mode of vibration of the calandria vessel due to applied dynamic loading will be in the axisymmetric mode. The frequency of vibration in the purely radial axisymmetric mode is given by

$$\omega^2 = \frac{E}{\rho R^2} \qquad (2)$$

where: ρ - density of shell material
R - Mean radius of shell

The frequencies of vibration in the basic mode are estimated to be around 200 cycles/sec for the main shell and 225 cycles per sec for the sub-shell. These frequencies correspond to a period of vibration (τ) of the order of 5 msec.

When the pressure loading is not axisymmetric, the cylindrical shell can respond in modes with a large number of circumferential waves which tend to have lower frequencies. The lowest frequencies of natural vibration are calculated for the main and sub-shell using the method given in Ref 3 as:

$$\omega_n^2 = \frac{E\,\Delta_n}{\rho R^2(1-\nu^2)} \qquad (3)$$

where Δ_n = frequency parameter in mode n

$$= \frac{n^4 + (1-\nu^2)}{\Sigma} \left[\frac{K_1 R}{nL}\right]^4$$

Σ = $\frac{1}{12}\left(\frac{h}{R}\right)^2$

n = number of waves in circumferential direction

L = length of shell

ν = Poisson ratio

h = thickness of shell

K_1 = a constant=1.506π for a shell with fixed ends and with half axial wave.

For any value of n, knowing the geometry of the shell, the frequency ω_n can be calculated. To obtain the lowest frequency, the value of n is varied over a wide range and the corresponding frequencies calculated. The lowest frequencies for the main shell are around 60 to 64 cycles per second(cps) with the corresponding period (τ) being 15 to 16 msec with n=9,10. For the sub-shell the lowest frequencies of around 280 cps are obtained with n=16 or 17 with a period of 3.5 msec.

Impulse Response of Calandria Vessel.

In order to examine the impulse response of the shell, the dynamic equilibrium of the shell under the combined action of transient pressure, internal hoop stress and inertia is considered as in the case of vibration analysis. Considering a unit length of the vessel in the axial direction under transient pressure loading P(t), the equation of motion can be written in terms of the dynamic stress and inertia as:

$$2Rh\,\rho\,\frac{d^2u}{dt^2} + 2\sigma h = 2\,R\,P(t) \qquad (4)$$

where u = radial displacement

t = time

As the hoop stress σ is related to the radial displacement by

$\sigma = E\,\varepsilon = E\,u/R$, the equation of motion reduces to

$$\frac{d^2u}{dt^2} + \omega^2 u = \frac{P(t)}{\rho h} \qquad (5)$$

The general solution to equation (5) is

$$u = u_0 \cos \omega t + \dot{u}_0 \sin \omega t + \frac{1}{\rho h \omega}\int_0^t P(t)\,\mathrm{Sin}\,\omega(t-z)\,dz \qquad (6)$$

The response of a shell, initially at rest(ie, with $u_0=\dot{u}_0=0$), to a rectangular pressure pulse of magnitude P_0 acting for a duration ΔT, is obtained from the above equation as:

$$u = \frac{P_0}{\rho h \omega^2}\,[1- \cos \omega t] \qquad 0 \le t \le \Delta T \qquad (7)$$

and

$$u = \frac{P_0}{\rho h \omega^2}\,[\cos \omega(t -\Delta T) - \cos \omega t\,] \qquad t \ge \Delta T \qquad (8)$$

when ΔT is small compared to the period ($\tau=2\pi/\omega$), these solutions reduces to

$$u = \frac{I}{\rho\, h\, \omega}\,\sin \omega t \qquad (9)$$

where I = specific impulse = $P_0\,\Delta T$

The maximum strain in the shell is given by

$$\varepsilon_{max} = \frac{u}{R} = \frac{I}{R \rho h \omega} \qquad (10)$$

When the pressure pulse is applied over a period comparable to the period of vibration, the maximum strain in the shell is obtained by using the relation given in Ref 2 as

$$\varepsilon = K \varepsilon_{max} \qquad (11)$$

where K is a correction factor to account for finite pulse width.
The values of correction factor K (given in Ref 2) used in the present analysis are shown in Table 2

Using the above methodology, the strain in the main body of the calandria vessel is estimated due to a hypothetical impulse of 5 MPa magnitude for various pulse widths and two pulse shapes(ie rectangular and triangular shapes). These results shown in Table 3 indicate that large plastic strains occur in the body of the shell for the asymmetric impulse response .

Estimation of Impulse Required For Failure.

The method used for estimating the impulse required for causing failure of the calandria vessel is described here. The method is based on the assumptions that;

a) the response of the main shell is in symmetric mode,
b) the response of the shell is elastic
c) the failure of the vessel is due to plastic strain accumulation at the highest stress points (ie the junctions of main shell and annular plate) due to successive impulse loads. The plastic strain limit for failure is assumed to be 10%.
d) Any strain hardening of the material due to plastic straining is neglected.
This assumption is very conservative regards the plastic strain accumulation.

Based on these assumptions, consider the symmetric response of the shell due to a rectangular pressure pulse (with P_{max}= 5 MPa and duration ΔT= 15 msec). Using the correction factor K of 0.1, the peak strain in the shell is estimated from equations (10) and (11) to be around 0.7 percent. Based on the results of the static stress analysis, the strain at the critical point is assumed to be 1.5 times the above value at 1.05 percent. Subtracting the elastic strain component from the total strain, the plastic strain at the critical point is obtained as 0.6 %. Neglecting the material strain hardening effect, the response of the shell in successive impulses is assumed to be identical. Consequently, it is assumed that the same amount of plastic strain will be accumulated in each successive pulse. Based on the assumed failure strain of about 10 percent plastic strain, the number of impulses required for failure is calculated to be around 16 which is equivalent to a specific impulse of 1.2 MPa-sec. Similar calculations with a triangular pulse indicate that the specific impulse for failure is about 0.44 MPa-sec.

RESULTS AND CONCLUSIONS

Based on the limit load analysis, the lower bound pressure at which plastic straining begins (based on ASME code) is estimated to be about 2.7 MPa. The detailed static analysis also indicates that the failure pressure (or the pressure at which rapid plastic straining occurs at the critical point) for the vessel is around 3.5 MPa. When the dynamic effects due to inertia are included, the estimated failure pressure is seen to be higher. These failure pressures are seen to be much higher than any peak pressure estimated in all types of single channel accidents considered. If the loading is assumed to consist of a series of impulse loadings, the specific impulse required for failure is estimated to be in the range of 1.2 MPa-sec for rectangular pulses and 0.44 Mpa-sec for triangular pulses. The calculated specific impulse in a single channel failure (equal to the area under the pressure transient in Fig 2a) is around 0.1 MPa-sec. Based on these static, dynamic and impulse load analyses, it is concluded that pressure transients resulting from the failure of a fuel channel in any single channel event does not result in calandria vessel failure.

REFERENCES

1. ASME Boiler and Pressure Vessel code Section III. 1986 edition Table 1-6.0 and Table I-2.2.

2. Karpp R.R., Duffey T.A. and Neal T.R. "Response of Containment Vessel to Explosive blast loading." Los Alamos Scientific Laboratories Report La-8082, UC-38. June 1980.

3. Krauss H. "Thin Elastic Shells " John Wiley and Sons, 1967.

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Table-1.
Calandria Vessel Material and Geometry Data.

Main Shell		Sub Shell		Annular plate	
Diameter =	7.6 m	Diameter =	6.76 m	Outside Diameter =	7.65 m
Length =	4.01 m	Length =	0.965 m	Inside Diameter =	6.8 m
Thickness =	28.6 mm	Thickness =	28.6 mm	Thickness =	19 mm

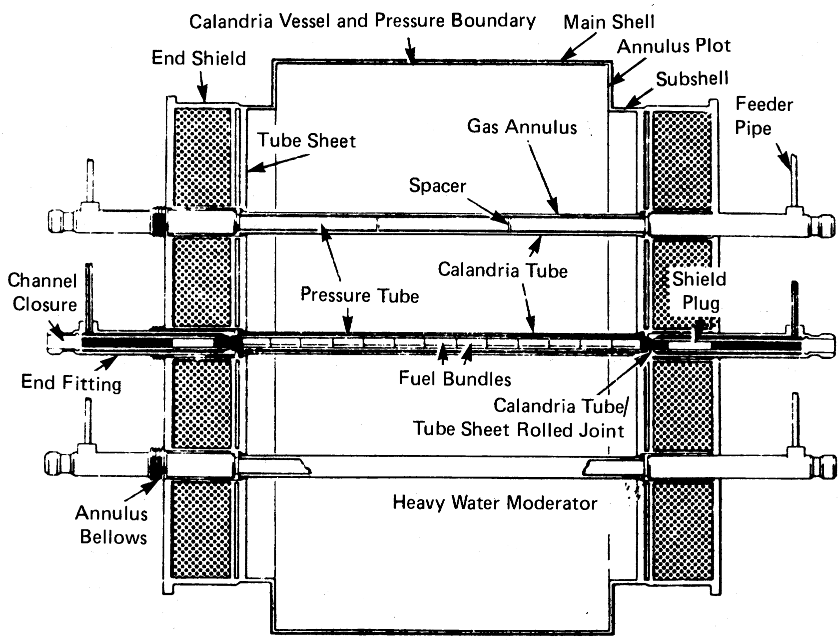
Material - 304 L Austenitic Stainless Steel
Young's Modulus E- 186 GPa,
Yield Stress σ_y - 360 MPa.
Poisson Ratio ν - 0.3
Density ρ 7800 kg/m^3
Values of numerical constants in equn (1)
A=1.22 and n=8.

Table 2.
Variation of correction factors (K) with non-dimensional pulse width.

$\frac{\Delta T}{\tau}$	Rectangular Pulse	Triangular Pulse
0.0	1.0	1.0
0.5	0.5	0.7
1.0	0.3	0.44
1.5	0.2	0.34
2.0	0.15	0.32
3.0	0.1	0.3

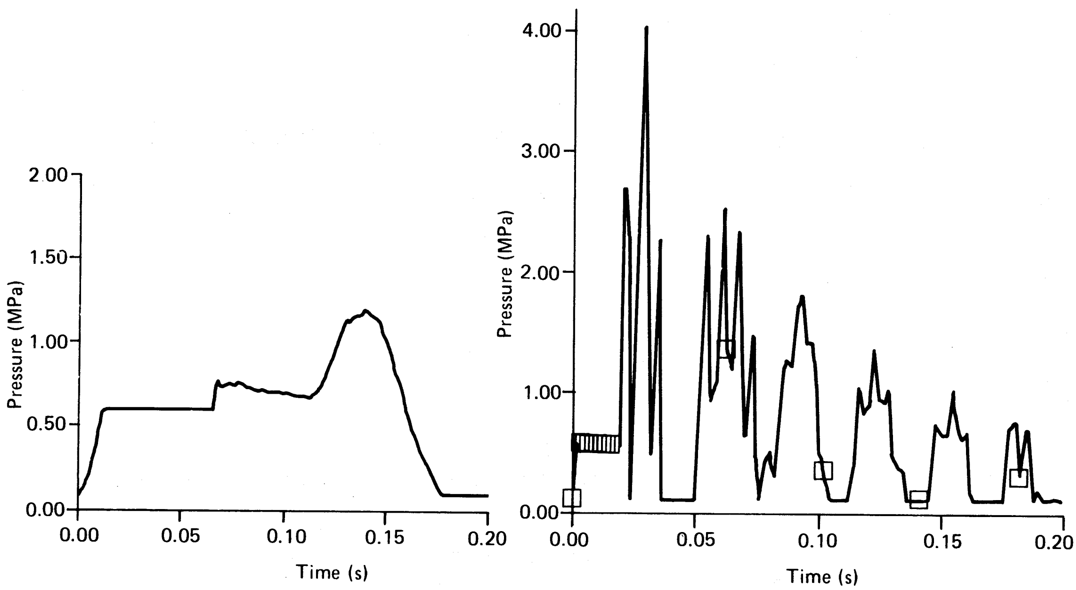
Table 3 .
Strain in the Calandria Vessel (%) due to a 5 MPa Pressure Pulse.

	Rectangular Pulse		Triangular Pulse	
Pulse Duration ΔT msec	Symmetric Response	Asymmetric Response	Symmetric Response	Asymmetric Response
5	0.7	6	0.46	2.73
10	0.7	6	0.7	5.3
15	0.7	6.8	1.22	4.5



Not to Scale

FIGURE 1
Schematic of CANDU Calandria Vessel



a) Due to Single Channel Failure

b) Due to Multiple Channel Failure

FIGURE 2
Maximum Pressure Transients in the Moderator

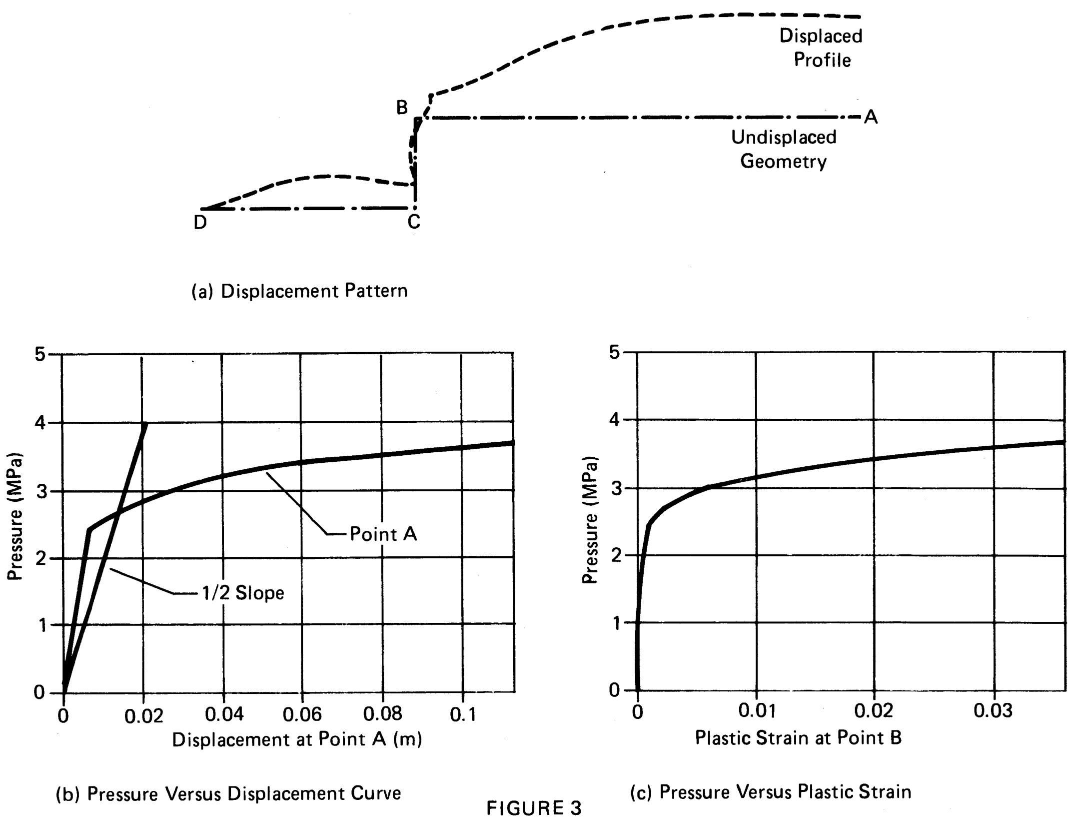


FIGURE 3
Limit Load Analysis

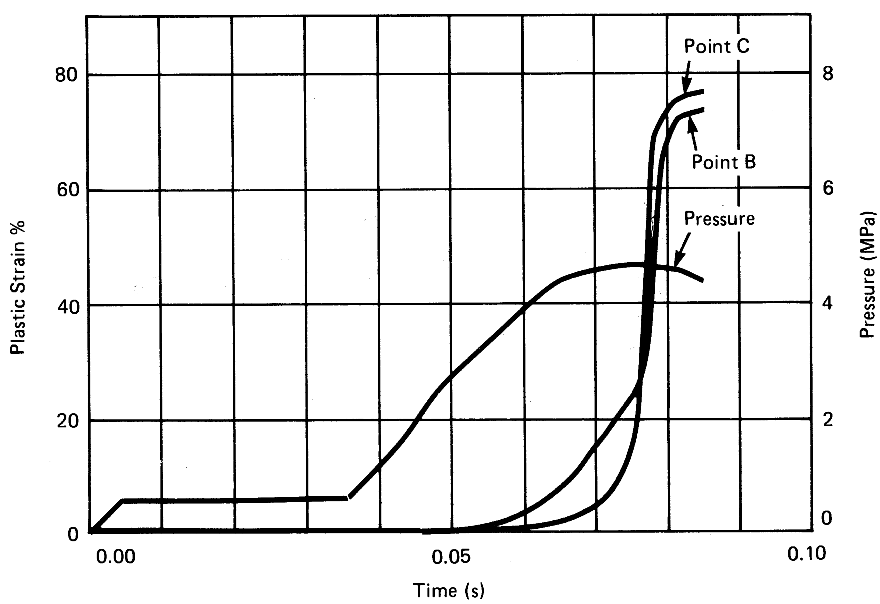


FIGURE 4
Plastic Strain Due to Dynamic Loading