EFFECT OF LOW VELOCITY IMPACT ON DEFORMATION BEHAVIOUR OF METALLIC PRESSURE VESSEL

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ABSTRACT

The present study deals with the numerical investigations of aluminium pressure vessel behaviour when impacted by spherical and blunt steel projectiles. One of the main causes of vessel rupture is the reduction of the vessel strength caused by external impact or material defect. The safety of the pressure vessel is vital because of its high working pressure which is more than 20 MPa and any untoward incidents must be avoided. Therefore, the main goal of this study is to predict the deformation and vessel strength behaviors for varying projectile shape, target thickness and impact velocity under low velocity impact conditions. In the modelling process, Johnson-Cook plasticity model was selected for the vessel material (Al 6061-T6) and direct impacts were subjected on the mid and dome vessel zones with varying parameters. The dynamic explicit analysis of the problem was carried out using ABAQUS finite element code. It has been concluded that, the target vessel exhibited lower impact resistance for blunt projectile compared to spherical projectile under the same impact loading conditions. Also, the damage which occurred in the dome vessel zone was greater than on the cylindrical area of the vessel, which leads to a specific design requirement in the dome zone. Finite element simulation results were compared with the numerical results in the literature and a good agreement was found.

Keywords: impact behaviour, metallic vessel, damage, finite element analysis.

INTRODUCTION

In the last 20 years, there is a clear evidence that the number of NG-fuelled driving systems has been increasing steadily. This is due to the fact that NGVs emit very low pollutant. Moreover, they have low maintenance cost and less noise issues. However, there are serious concerns regarding the safety of NGVs in normal operation and in accidents. The NGVs are equipped with pressure vessels called cylinders. These pressure vessels should be designed not to rupture when fully filled during their service life [1]. From this outlook, there is a necessity to ensure more safety features in NGVs especially for pressure vessels.

For pressure vessels, the failure produced by low velocity impact of external objects is considered as one of the most critical safety problems which always occur through service life. These events can affect the structural stability resulting to dangerous situations. According to the standard EN 14427 [2], the ability of the designed vessel to resist the internal pressures as well as the external loading need to be tested accurately through experimental and numerical impact tests. Furthermore, one of the main causes needs to be considered concerning the vessel rupture is a reduction of the vessel strength caused by external impact, corrosion or material defect [3]. This aspect will be investigated in this paper.

Recently, with the advancement of computers, finite element analysis (FEA) became a more useful tool in the analysis of impact. Therefore, computer based simulation studies have become dominant amongst the researchers. FEA is a method widely used in design, analysis and optimization. FEA has proven to be an important tool to understand the deformation responses and behaviour of vessel damage under impact loads [4]. The FEA simulations are thus used to reduce costly experimental prototype models for impact analysis.

Therefore, this study is focused on the analysis of a thin vessel structure subjected to direct impact loading for both central and dome vessel regions to predict the damage effects and deformation through FEA.

The literature showed that low, mid and high velocity impacts are widely studied for metallic plates and shells but are restricted for pressure vessels. The studies carried out can be classified into analytical, experimental, numerical and empirical approaches. Hallquest, *et al.* [5] studied the impact simulation between a rigid sphere and a thin elastic-plastic plate using LS-DYNA. The study was claimed to be able to predict correctly the failure behaviour of the plate. However, the experimental validation was not carried out. Many literatures related to the simulation of metallic systems employed a variety of constitutive models for damage modelling of foreign objects, namely Johnson-Cook (J-C), Bammann and Armstrong-Zerilli material models. Due to high strength-to-density ratio, aluminium alloy has become a potential material for light tank (vessels). Borvik *et al.* [6] analyzed the impact of 7.62 mm projectile on double-layered steel targets using Lagrangian LS-DYNA simulations by assuming axisymmetric conditions. A good agreement was observed between simulations and experimental results.

Dey *et al.* [7] predicted the behaviour of double layered Weldox 700E plates impacted by an ogival projectile by using axisymmetric Langrangian simulations in LS-DYNA. The results showed a good agreement between numerical simulations and experimental results. The effect of projectile nose shape on impact was carried out by Kpenyigba *et al.* [8]. It was found that the ballistic limit is higher for hemispherical shaped projectile, followed by conical and blunt shaped projectiles, respectively.

In this paper, a parametric study was performed numerically to determine the effects of parameters such as impact velocity, vessel thickness and geometry of the projectile on the deformation and damage behaviour due to low velocity impact loading.

PROBLEM DESCRIPTION AND MODEL VALIDATION

a) Problem description

Simulations were carried out on 6061-T6 aluminium alloy vessels of 3 mm and 5 mm thickness using blunt and spherical shaped projectiles. Hard steel projectile of 0.531 kg mass were kept constant for two projectile types, viz. a 15-mm-diameter blunt shaped projectile with a depth of 28 mm and a 50-mm-diameter spherical shaped projectile as shown in Figure-1. The projectiles were impacted normally on the central and dome regions of the target vessel at a low velocity. The impact velocities selected for the steel impactor were 13 m/s and 15 m/s with a constant drop height of 0.02 m.

The vessel considered has a length of 300 mm, an internal diameter of 150 mm and a variable thickness. The vessel was modelled as a 3-D deformable body as shown in Figure-2 using the non-linear FE code ABAQUS Explicit. The impactor was modelled as a rigid body with only one allowable translational displacement and all other translational and rotational degrees of freedom were fixed. A single node reference point was selected and assigned with the mass and initial velocity. The contact algorithm used to simulate contact interaction between all components was the "general contact algorithm". The effect of friction between the projectile and target was considered as negligible.

The vessel structure was modelled by using 4 nodes linear tetrahedron (C3D4) elements with 3 integration points along the thickness direction of the vessel. The vessel was meshed using tetrahedron shaped solid elements which comprised of 28573 elements with an element size of 0.004 m.

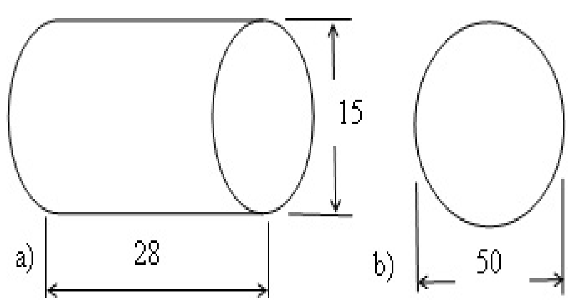

Figure-1. Projectile shapes used in the simulations (a) blunt projectile (b) spherical projectile (All dimensions in mm).

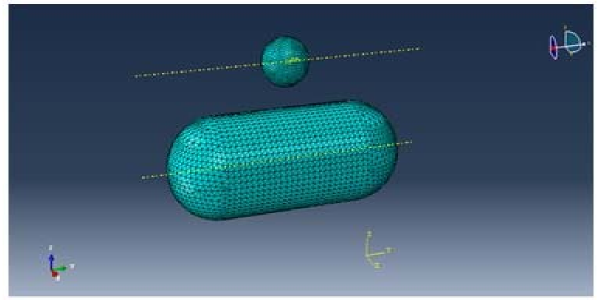

Figure-2. Vessel model used in this analysis and mesh density distribution.

b) Johnson-cook constitutive model

The Johnson-Cook (JC) model was primarily developed from the collected test data at different temperatures and strain rates for a wide range of test samples. The JC model was introduced in 1983 and was initially prepared for the application in the computational work. In this paper, the JC model was selected to predict the material response. Equation. (1) describes the JC model which states the Ludwik law as a multiplicative law of three uncoupled terms. These terms are dependent on the plastic strain, the strain rate and the temperature,

$$\sigma = \left[A + B\varepsilon^{n}\right]\left[1 + C\ln\frac{\dot{\varepsilon}}{\dot{\varepsilon}_{o}}\right]\left[1-\left(\frac{T-T_{o}}{T_{m}-T_{o}}\right)^{m}\right] \quad (1)$$

where A is the yield stress at a reference strain rate and temperature, B is the strain hardening modulus, n is the strain hardening exponent, C is the strain hardening rate constant, ε is the equivalent plastic strain, $\dot{\varepsilon}$ is the equivalent plastic strain rate, $\dot{\varepsilon}_o$ is the reference strain rate, T is the experimental temperature, T_o is the reference temperature, T_m is the melting temperature, and m is the thermal softening constant.

The properties of 6061-T6 aluminium alloy selected for the vessel model are shown in Table-1.

Table-1. Summary of Al 6061-T6 properties [9].

Parameter	Description	Value
E	Young's modulus	70 GPa
ν	Poisson's ratio	0.33
A	Material parameter	340 MPa
B	Material parameter	1018 MPa
n	Strain hardening exponent	0.7789
C	Material parameter	0.0568
m	Temperature power coefficient	0.2526
ε	Equivalent plastic strain	-
$\dot{\varepsilon}$	Equivalent plastic strain rate	-
$\dot{\varepsilon}_o$	Reference strain rate	0.0001 s^{-1}
ρ	Density	2700 kg/m^3
T_m	Melting temperature	855 K
T_o	Reference temperature	298 K
C_p	Specific heat	486 J/kg-K

c) Modal validation

A numerical model was created for the impact and crash analysis of a striker plate impactor with a velocity of 15.6 m/s and a mass of 275 kg on a target square tube of 75 mm × 75 mm with a thickness of 2 mm, as shown in Figure-3. The tube which was made of A36 steel has a length of 350 mm and was imposed to the same analysis as S-300 square tube conditions as in Ref. [10]. The analysis was carried out with a FE code using ABAQUS Explicit and a good agreement was achieved. This agreement can be observed through the force time graph shown in Figure-4 where the average force was 111 kN which is very close to the reference case value of 115.7 kN.

In this paper the analysis was performed in the same manner as that of the validation case, with regards to general contact algorithm, boundary conditions, modelling parts and plasticity model except for the differences in striker, target shapes and material selection.

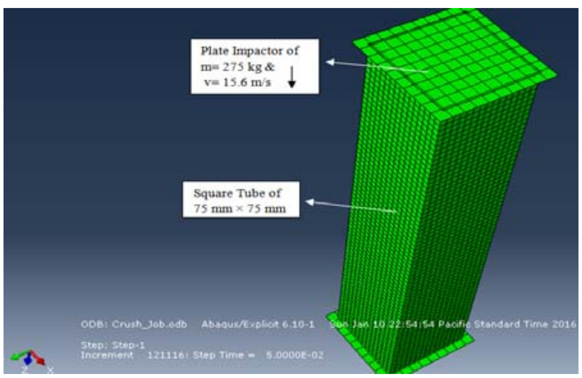

Figure-3. Finite element analysis setup for the validation case.

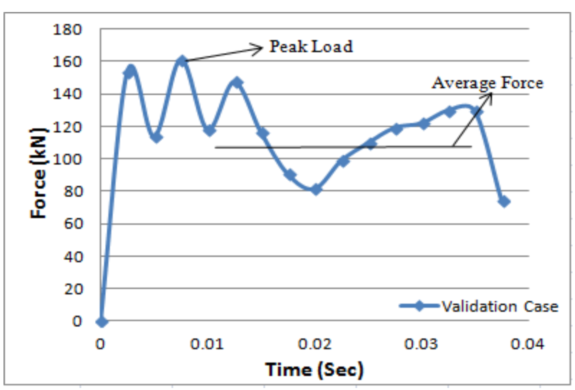

Figure-4. Force-time curve of the validation case study.

RESULTS AND DISCUSSION

An investigation of the low velocity impact load level at which a metallic vessel will be damaged was presented. The peak load is described as the load level, obtained from the load–time history or load–displacement plot, at which a sudden load drop occurs due to the loss in the specimen stiffness as a result of target level damage.

Finite element analysis was carried out on a full vessel by using ABAQUS. The boundary condition was assigned as a fixed fixture along the bottom longitudinal y-axis. The impact was subjected exactly at the top, at the middle and dome sections of the vessel. The results obtained from the numerical predictions are presented accordingly. The results presented are in the form of maximum von Mises stress and maximum deformation along with the load-time curves for all studied cases.

Figure 5 shows the maximum von Mises stress of 628.8 MPa with corresponding maximum deformation of 10.5 mm for direct spherical impactor on vessel center section at 13 m/s impact velocity on 3 mm thickness target vessel.

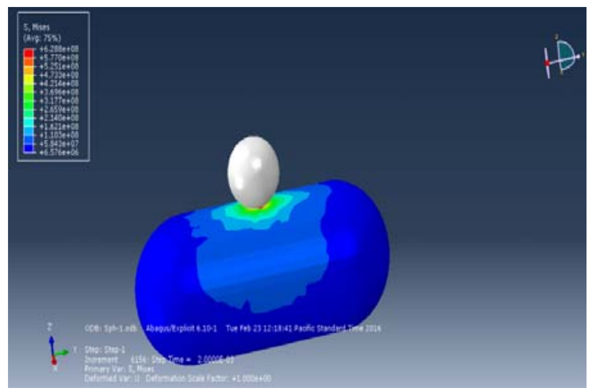

Figure-5. von Mises stresses for direct spherical impactor at the center section for vessel thick. 3 mm and V= 13 m/s.

Figure-6 shows that the maximum von Mises stress and deformation for the same case except the velocity of 15 m/s as 654.2 MPa and 13.3 mm, respectively. The stresses increase with increasing impact energy for a constant thickness.

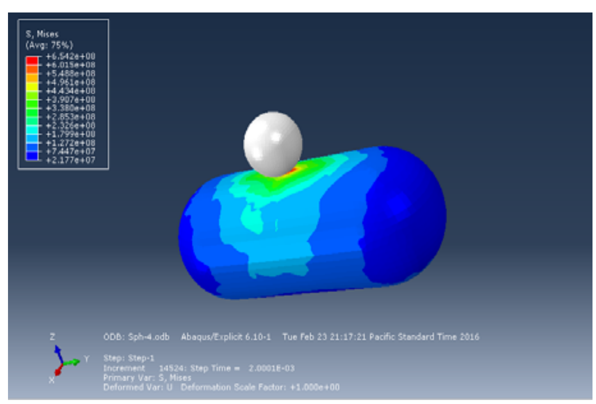

Figure-6. von Mises stresses for direct spherical impactor at the center section for vessel thick. 3 mm and V= 15 m/s.

Also Figures-7 and 8 display the maximum von Mises contours for the direct impact of spherical and blunt impactor on the dome section, for impact velocity of 13 and 15 m/s at constant thickness of 3 mm, respectively.

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The above few samples of results and the summary of all studied cases are summarized in Tables-2, 3, 4 and 5. It has been shown that for all cases, when the target thickness is increased from 3 mm to 5 mm, the von Mises stresses and deformations will decrease accordingly. This behaviour is completely true as compared with the literature [11].

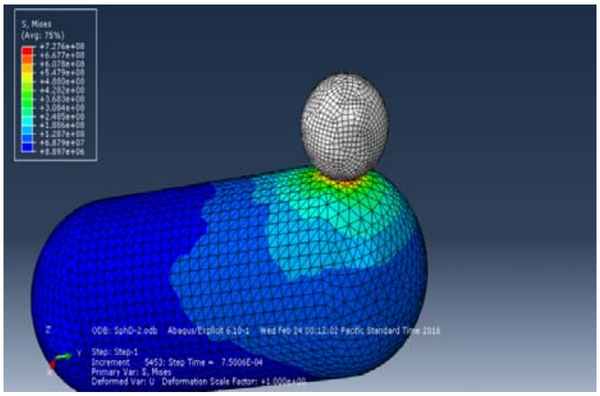

Figure-7. von Mises stresses for direct spherical impactor at the dome section for vessel thick. 3 mm and V= 13 m/s.

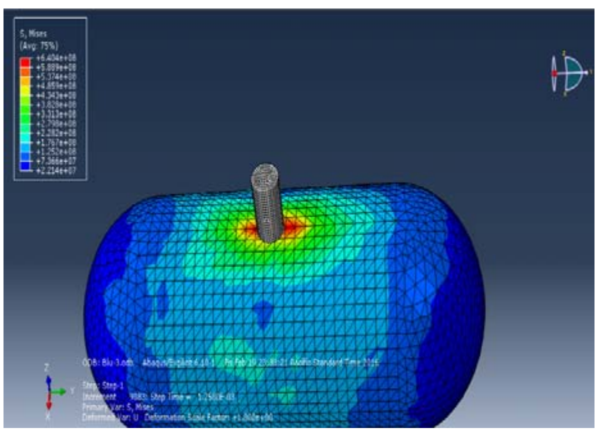

Figure-8. von Mises stresses for direct blunt impactor at the center section for vessel thick. 3 mm and V= 15 m/s.

Table-2. Summary of results for direct impact of spherical projectile on the vessel center.

Impact Velocity (m/sec)	Vessel Thick. (mm)	Max. von Mises stress (MPa)	Max. deformation (mm)	Max. Force (kN)
15	3	654.0	11.6	5.1
13	3	628.8	10.5	3.2
15	5	621.2	8.0	2.3
13	5	613.8	7.5	2.0

Table-3. Summary of results for direct impact of spherical projectile on the dome center.

Impact Velocity (m/sec)	Vessel Thick. (mm)	Max. von Mises stress (MPa)	Max. deformation (mm)	Max. Force (kN)
15	3	773.1	13.5	21.8
13	3	727.6	11.2	14.5
15	5	673.4	8.5	12.1
13	5	669.4	7.7	9.8

Table-4. Summary of results for direct impact of blunt projectile on the vessel center.

Impact Velocity (m/sec)	Vessel Thick. (mm)	Max. von Mises stress (MPa)	Max. deformation (mm)	Max. Force (kN)
15	3	640.4	18.2	20.1
13	3	626.0	17.3	18.2
15	5	613.9	15.3	14.3
13	5	542.0	15.0	13.3

Table-5. Summary of results for direct impact of blunt projectile on the dome region.

Impact Velocity (m/sec)	Vessel Thick. (mm)	Max. von Mises stress (MPa)	Max. deformation (mm)	Max. Force (kN)
15	3	768.1	18.9	24.9
13	3	737.0	18.3	21.7
15	5	736.2	16.0	20.5
13	5	616.9	15.5	18.6

Generally, in this study, the von Mises stresses exceeded the yield stress of the selected material for most cases of impact velocities. Also it was observed that, the damage which occurred due to spherical projectile at the dome section is greater than the central section for the same mass and impact velocity. This occurs because the area of contact for spherical impactor on the dome section is smaller as compared to the center section.

Typical force–time graphs for each type of analyzed case in this study are presented in Figures-9 –12. The force here was assigned to the selected reference point of the deformable target surface. Each figure predicts the force-time behaviour for variable velocity and thickness of target for direct central and dome impact conditions.

The peak load (F_{max}) was considered as an important parameter for the damage of structure at low-velocity and low-energy impacts. The induced damage due to blunt impactor at the dome section was higher than for the central section for each case represented in Figures-9 and 10. The maximum forces at the dome and central section were 24.9 kN and 20.1 kN, respectively for the

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case of 3 mm target thickness and 15 m/s impact velocity. Hence, the vessel structure exhibited more impact resistance in terms of failure due to contact stresses in the vessel center location.

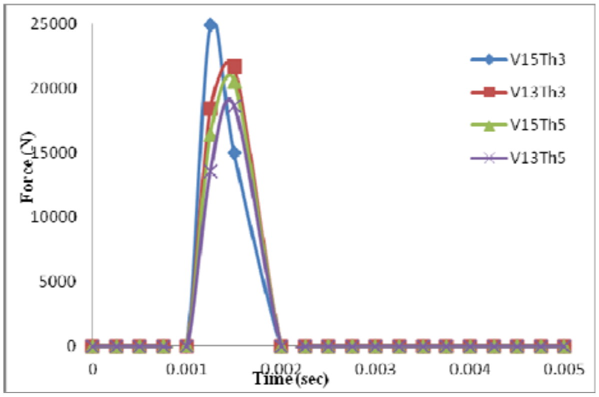

Figure-9. Force-Time curves for direct impact of blunt projectile on the dome section.

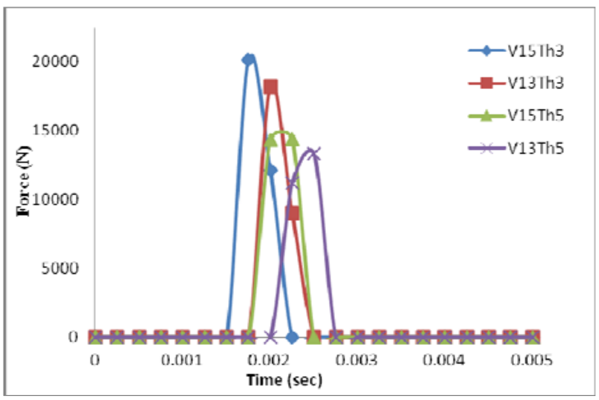

Figure-10. Force-Time curves for direct impact of blunt projectile on the vessel center section.

The prediction of damage due to the impact of spherical projectile at the dome and central sections under the same specifications was also included. Figures-11 and 12 show the behaviour of the force-time curves for the direct impact at two zones for different target thickness and impact velocity. It was observed from the results that the damage which occurred on the dome section was also higher than for the central section for each case study. Hence, the von Mises stress values were higher at the dome zone than that at the central zone under the same impactor shape. The forces illustrated by the curves were also shown and the reason for such behaviour was due to the small area of the dome section as compared to the central section. This behaviour gives an indication that the dome region is the critical and weakest area of the vessel structure. Therefore, it exhibited a negative impact resistance for both spherical and blunt projectiles under the same impact loading. Furthermore, the contact stresses due to the impact of blunt impactor were higher than the spherical impactor. The reason was that the contact pressure is inversely proportional to the radius of impactor, and the radius of blunt impactor was lower than the sphere impactor.

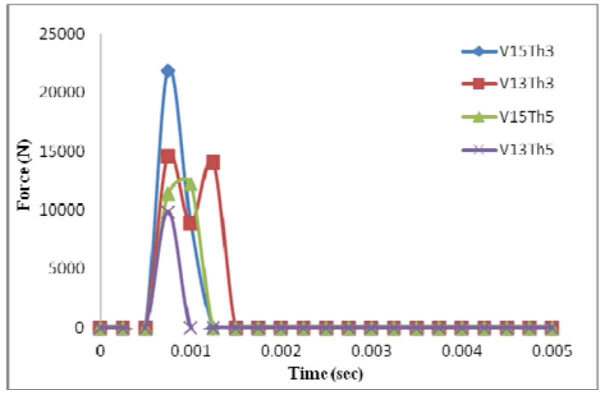

Figure-11. Force-Time curves for direct impact of spherical projectile on the dome zone.

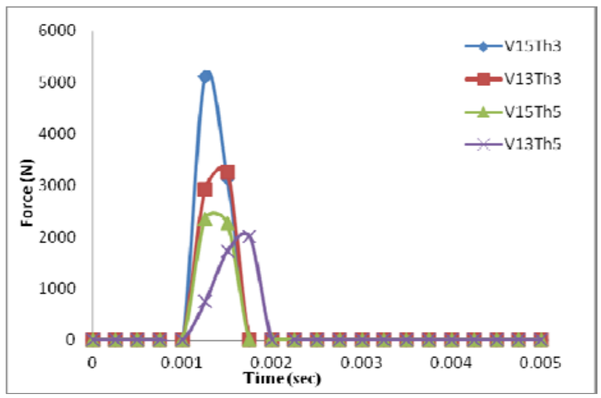

Figure-12. Force-Time curves for direct impact of spherical projectile on the center zone.

CONCLUSIONS

- The maximum von Mises stresses and deformations decrease when the vessel thickness increases. Therefore, the damage of the structure becomes less for thicker target vessel according to the studied impact conditions.
- As expected, with the increase of impactor velocity, the contact force magnitude also increases, due to which the damage of the impacted vessel increases.
- Generally, the contact forces and displacements due to the impact of blunt projectile are greater than the spherical impactor. Therefore, the target vessel exhibited low impact resistance for blunt projectile compared to spherical projectile under the same impact loading conditions.
- It has been observed that, the values of stresses and deformations were very high in the dome zone as compared to the vessel cylindrical zone for the same impactor and hence the damage in this area was also greater. Therefore, the pressure vessels require specific design and increased thickness at this area.

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