Having a Blast!

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Abstract:

Freudenberg Oil & Gas Technologies (FO&GT) supplies complex, metal to metal sealing solutions to the oil & gas and energy industries. As part of its core business, supporting the development and expansion of further qualification procedures for its clients, FO&GT have specified a blast chamber for its new factory.

Using traditional hand calculations and finite element methods undertaken with Abaqus/CAE, work has been undertaken on an initial developmental FE model aiming to create a realistic, correlating, ballistic-type simulation model to be used in conjunction with a physical test program, focusing on the energy absorption characteristics of a blast chamber.

Constructed of high strength steel columns, with horizontal wooden sleepers reinforced by steel panels, the blast chamber has an internal volume of 450m³ to cope with large pressure vessels and equipment; being designed to withstand up to 0.2 MJ of impact energy. Dampers placed in the walls and floor allow for large amounts of chamber flexure, maintaining the desired containment characteristics for internal projectile(s).

A range of experimental elasto-plastic, visco-elastic and hyper-elastic material models combined with ductile damage and eroding surface contact, are used within the explicit finite element model, assessing the large displacements expected from high speed impacts, ensuring containment.

To aid the flexure characteristics of the blast chamber both on a global and local level, FO&GT have utilized in-house expertise with a derived ‘Freudenberg rubber model’ for the dampers, together with factory test data depicting deflection performance for the wooden sleepers used in its walls.

Initial FEA results have shown good correlation with traditional calculation methods under current loadings conditions.

Keywords: Explicit, Abaqus/CAE, Ductile damage, Visco-elastic, Hyper-elastic, Erosion, Elastomer.
1. Introduction

Freudenberg Oil & Gas Technologies Ltd (FO&GT) specializes in a range of high precision metal to metal sealing solutions for use in the oil, gas and energy industries. Oil and gas pipeline and pressure vessel operation requires high integrity sealing solutions to cope with the fluctuating demands of transport media, pressure and temperature to match the campaign life stipulated by the client.

Components such as seal rings, pipe connectors and flanges, integrated within Optima subsea connectors, pig launcher receiver assemblies and chemical processing vessels, can be subjected to amongst other loadings, immense internal pressures up to 25 ksi. These components are therefore required to be subjected to the rigors of extensive qualification processes for the individualistic demands of many and varied high profile clients.

Testing of highly pressurized components presents a wide range of problems in its own right; how to contain systems that can produce high velocity projectiles with many mega-joules of kinetic energy in the event of catastrophic component failure.

As part of a factory relocation and upgrade, FO&GT have decided to bring all component qualification programs in-house to reduce costs and to have more control of the testing processes, allowing the provision of a more bespoke service to the clients that demand such a service. To ensure this, a containment chamber needs to be specified and built. FO&GT’s own design of blast chamber is based around limited available data, both structurally and ballistically. In this respect it is quite conservatively designed, ensuring a generous factor of safety within the finalized blast chamber specification.

This paper addresses the task of designing and analyzing a ballistic chamber with limited real world knowledge and experience, utilizing largely simplistic material models derived and developed from physical testing for various wooden, rubber and steel components. FEA simulation has been undertaken on a range of material models for wood [1],[2], ductile metals such as mild steel [3] and aluminium [4], as well as rubber [5], where high levels of detail have been able to predict variations of Young’s modulus through fibrous materials, as well as the ‘cup and cone’ phenomenon [6] with the necking and failure of specimens of steel bar.

In order to reduce complication and the potential to get carried away with extreme levels of detail within the utilized material models, FO&GT have made a conscious effort to keep material models as simplistic as possible during the verification process of virtual and physical testing. Material models are derived exclusively within the FE domain in order to generate parity between the virtual and physical world. This paper documents material model development, its application to the physical chamber and performance comparisons of the initial explicit FE model(s) with theoretical hand calculations.
2. Blast Chamber Layout & Specification

The blast chamber has an internal volume of about 450 m$^3$ to accommodate the majority of large pressure vessel components and equipment and is nominally 4.5 m tall by 9 m wide and 11.2 m long. The blast chamber is illustrated in Figure 1;

![Figure 1: Left to right; Images of the blast chamber from outside, above looking inwards and outwards at the blast chamber doors.](image)

Blast chamber construction (see Table 1) comprises high strength steel vertical columns (1) on a concrete base (2), with horizontally arranged fully floating wooden sleepers specified in either oak (0 to 2 m high) or pine (2 to 4.5 m high) (3). Reinforced by steel panels (20 mm thick behind the oak and 6 mm thick behind the pine) (4), the chamber is designed to absorb around 0.2 MJ of energy before a breach of containment. Between each set of columns and steel panels sits a plethora of machined polyurethane dampers (5). Heavy duty disc springs (6) are bolted (7) into the base plate of each main column, designed to react and damp column flexure as much as possible during projectile impact. Component material specifications are identified in Table 1.

<table>
<thead>
<tr>
<th>Part</th>
<th>Material Specification</th>
<th>Young’s Modulus (GPa)</th>
<th>Poisson’s Ratio</th>
<th>Density (Tonne/mm$^3$)</th>
<th>Yield Stress (MPa)</th>
<th>Elasto-Plastic Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>(1)</td>
<td>S355</td>
<td>202</td>
<td>0.3</td>
<td>7.85x10$^{-9}$</td>
<td>355</td>
<td>Non-linear</td>
</tr>
<tr>
<td>(2)</td>
<td>Concrete</td>
<td>40</td>
<td>0.21</td>
<td>2.40x10$^{-7}$</td>
<td>40</td>
<td>-</td>
</tr>
<tr>
<td>(3)</td>
<td>Pine/Oak</td>
<td>3.952/8.866</td>
<td>0.35/0.35</td>
<td>5.28x10$^{-9}$/7.39x10$^{-10}$</td>
<td>28/121</td>
<td>Bi-linear/Bi-linear</td>
</tr>
<tr>
<td>(4)</td>
<td>S355</td>
<td>202</td>
<td>0.3</td>
<td>7.85x10$^{-9}$</td>
<td>355</td>
<td>Non-linear</td>
</tr>
<tr>
<td>(5)</td>
<td>Polyurethane</td>
<td>-</td>
<td>-</td>
<td>1.20x10$^{-9}$</td>
<td>-</td>
<td>Non-linear</td>
</tr>
<tr>
<td>(6)</td>
<td>Mild Steel (S355)</td>
<td>202</td>
<td>0.3</td>
<td>7.85x10$^{-9}$</td>
<td>355</td>
<td>Non-linear</td>
</tr>
<tr>
<td>(7)</td>
<td>Class 8.8 Steel</td>
<td>200</td>
<td>0.3</td>
<td>7.85x10$^{-9}$</td>
<td>800</td>
<td>Non-linear</td>
</tr>
</tbody>
</table>

Blast chamber design/construction ensures the wooden sleepers sacrificially absorb the initial impact energy of the projectile before load transfers into the steel blast plates. Upon impact with the blast plates, load is transferred into the polyurethane dampers and then into the steel...
framework around the chamber. The movement of the framework is dampened against the disc springs in each column base absorbing the remaining energy in the system.

3. Material Model Development

A large amount of work has gone into developing material models both from a ballistic perspective for the steel blast panels, and a brittle failure ideology for the wooden sleepers. Due to the shear scope of the material models available within the extensive property options, a decision was made early on to attempt to optimize a more simplistic material model for material specifications, hoping to limit potential sources of error as much as possible.

3.1 Pine & Oak Sleepers; Geometry & 3 Point Bending

The bend test rig layouts, together with the nominal dimensions of the wooden sleepers were replicated within Abaqus/CAE as a three point bend test. The bend test rigs contained a framework for supporting the wooden sleepers at both ends, together with metal pins to constrain the ends of the sleepers from moving in the vertical direction. A pressurized ram was positioned in the center of each sleeper with a flat metal plate that pushed upwards with an increase in ram pressure. Ram pressure was increased for both grades of wood until splintering and then fracture (snapping) was witnessed.

![Figure 2: Bend test rig configuration for testing the Oak sleeper.](image)

The two sets of bend tests for the pine and oak were undertaken (four sleepers each) with nominal sleeper dimensions of 127x254 mm and 62x216 mm respectively, with lever arms of 1000 mm and 1100 mm respectively. The orientation of the wood in the test rig was such that displacement due to bending occurs through the direction perpendicular to the grain [1],[2].

The difficulty of testing wooden samples is the lack of homogeneity from sample to sample due to the presence of knots and differing grain structure. Testing was conducted where orientation of the fiber direction was considered as consistently as possible based on the sleepers available. The pine and oak sleepers were subject to load after they had been dried, being more representative of sleeper condition over time after installation in the walls of the blast chamber.
3.2  Pine & Oak Sleepers; Material Data Generation

Ram pressure vs. ram displacement was recorded until the point of wood failure (snapping). These results were then converted to bi-linear elasto-plastic true stress vs. true strain for input into Abaqus/CAE. The graphs were used to generate a Young’s Modulus for the oak and pine materials, and to predict the fracture strain variable(s) used in the ductile damage sub-option. Widely available poisons ratio values of 0.35 were selected for both the pine and oak.

![Figure 3: Representation of the typical fracture pattern seen in the pine (left) and oak (right) sleepers.](image)

![Figure 4: Example of generated bi-linear elasto-plastic true stress vs. true strain data for the pine and oak sleepers for input into Abaqus/CAE.](image)

The differing grain structure from wood sample to sample makes direct comparison of any useful material data extremely difficult. As a result, the trend of comparative material data from test and sample variations is considered more important and useful in these initial stages of material model development. Figure 4 is supported by the data presented in both [1] and [2], and therefore
indicates that these initial material curves can be useful for the initial assessment of the global model.

3.3 Pine & Oak Sleepers; Local Mesh Discretization

Both the pine and oak models were run with a variety of mesh sizes to generate the correlation between the real world failure point and that within the simulation model until a mesh independent solution was obtained.

![Figure 5: Abaqus/CAE representation of the oak bend test.](image)

![Figure 6: Time shots of the V.M. stress plot (MPa) during fracture test of the oak.](image)

![Figure 7: V.M. stress plot (MPa) before (left), just after fracture (right) of the pine sleeper.](image)
For the oak sleepers, factory testing showed the assumed failure point to be after a ram displacement of 173 mm. The FEA simulation model predicted this to be 194 mm, giving a discrepancy of 12.6%. For the pine sleepers, a ram displacement of 57 mm was required to cause failure, with the FEA simulation predicting 61.6 mm, generating a discrepancy of 8.1%. These failure point displacements proved mesh independent below a mesh size of 20 mm for both materials, and were subsequently carried over into the global simulation model, giving an adequate representation of the three point bend test(s).

3.4 Steel Blast Plates

In order to develop a material model for the steel plates lining the outer boundary of the blast chamber walls, as well as the main vertical support columns, a set of ballistic tests were carried out based upon those found in a health and safety Executive (HSE) report [7] from the ballistic research laboratory (BRL). The tests conducted by the BRL present data for the required thickness of a mild steel plate to resist the penetration of a given dimension and mass projectile, travelling at a specified velocity.

With limited published information, comparisons were made between two randomly selected BRL tests. These tests were necessary to generate indicative initial material models for the steel components in both solid and shell element forms for the global simulation model. The first test (figure 8) was undertaken for a full 3D blast plate with multiple C3D8R elements through its thickness. A spherical ended projectile with a diameter of 63 mm was fired at a velocity of 80 ms\(^{-1}\) into a 19.3 mm mild steel plate (S355). The second test (figure 9) takes the same projectile and fires this at 102 ms\(^{-1}\) into a 26.7 mm steel plate with the same properties as the first, this time comprising of S4R shell elements.

Both sets of analysis were undertaken with identical non-linear elasto-plastic material data generated from ASME VIII Div.2 ANNEX 3-D. Once a mesh independent solution had been obtained, the displacement at failure criteria was optimized varied to obtain the limiting value for full penetration of the projectile through the steel plate. The material models used do not include additional variables such as stress triaxiality [3],[6], and it is recognized that these may have an effect on the penetration performance of the projectile through the material under consideration.

Ballistic comparisons were performed with Simpson integration criteria, where an identical number of integration points through the shell thickness were specified, duplicating the through thickness element count in the solid 3D plate. The performance of the shell and solid plate FE models were not contradictory, with the simplification from solid to shell, not fundamentally altering the deflection characteristics of the steel plate. Once the failure criteria had been established, the optimized material properties were then checked for a mesh independent solution once again. The lack of symmetry shown in figure 8 and figure 9 is the result of the off-center impact of the projectile.

Figure 8 and figure 9 shows snapshots over time of an identical geometry, 19.8 kg projectile hitting a solid and shell element plate of identical cross sectional area, but different thicknesses.
Figure 8: Time-shots of the impact test for the 3D solid 19.3 mm steel plate.

Figure 9: Time-shots of the impact test for the shell element 26.7 mm steel plate.

3.5 Disc Springs

The disc springs used on the vertical pillar base plates comprise standard (off the shelf) items of 100 mm in diameter, stacked to a height of approximately 130 mm. The disc springs have been simplified for inclusion within the FEA model (Figure 10) by means of a spring arrangement connecting two solid bodies. The total compressed thickness of the individually stacked springs is calculated and drawn as solid geometry (A) that is free to displace in the axial direction of the bolt shaft that secures all the disc springs into the pillar base plate and concrete floor.

The compression distance afforded by the springs is modelled as a ‘gap’, with 4 (spring) beam elements (B) positioned in the center of every quadrant of each set of disc springs. A spring stiffness $k$ value of 250 N/mm is specified for each beam feature. When this gap is reduced to zero, the ‘solid block’ comes into contact with the fixed upper portion of the bolt and head, and
resembles a locked out disc spring. Any further load is now taken directly by the bolt shaft. The ultimate tensile strength (UTS) of the bolt material (shown during factory pull tests to be around 90 kN on average) then becomes the limiting factor in the displaced performance of the vertical pillar and base plate. Non-linear elasto-plastic material data is generated from ASME VIII Div.2 ANNEX 3-D.

![Image of actual disc springs and FEA representation]

**Figure 10:** Actual disc springs (covered with blue plastic) on the blast chamber base plates (left), FEA representation of the springs (right).

The FE representation of the disc spring allows compression behavior to act more like a bushing, giving the spring stack movement with multiple degrees of freedom (DOF); likely in the physical world. Whilst sufficient mesh detail is required in the bolt shaft to give a reasonable prediction of the point of bolt failure (should it occur), this component proved largely immune to the size of the mesh discretization used.

### 3.6 Dampers

![Image of in-situ machined damper and FEA representation]

**Figure 11:** In-situ machined damper (left), FEA representation of dampers between steel plates and main framework (right).

The polyurethane dampers used between the steel plates and framework of the blast chamber are comprised of an 80ShoreA material specification. The data generated from standard ASTM D412-06a test methods, together with extensive in-house curve fitting and prediction methods developed
by Freudenberg GmbH, ensures that the dynamic nature of highly loaded and flexible elastomers is captured with a high degree of accuracy. To aid increased flexibility in the dampers, additional machining is undertaken to aid transient response.

4. Outline of the FEA Model

4.1 FEA Sub-Modelling

The FEA modelling methodology started with a series of local, less complex sub models of different interacting parts of the blast chamber assembly. These models were a combination of those assessing interaction through contact alone, whilst others concerned the optimization of a set of material properties (such as the steel blast plates). The sub-modelling methodology covered eroding contact, ductile damage, hyper and visco-elasticity, as well as optimization of the mass scaling function in components where element edge length was a limiting factor in the progress of the explicit solution. Sub-models were built up into three identical sets of blast chamber panel sections sharing common vertical columns with spring washers in the baseplates and dampers supporting the steel blast panels to those central columns.

4.2 Mesh Discretization

Of particular importance to FOGT was a reliable solution that would solve in a computationally efficient timescale. This was driven largely by the size of the physical environment being considered. Care was taken in specifying the element sizes for individual components within Abaqus/Explicit solution, where mesh sizing had to be controlled to a far greater extent than in Abaqus/Standard analyses.

![Figure 12: Illustration of mesh discretization around the vertical support column base and disc springs (left), outside wall of the blast chamber showing steel blast plates and rubber damper detail (right).](image)

The simulation model of the blast chamber contains approximately 1.02x10^6 nodes and 795x10^3 elements, containing around 91% C3D8R hexahedral elements, with the remaining 9% S4R linear quadrilateral elements.
4.3 Boundary Conditions

Two projectiles are fired at a height of 1 m in the middle of the central blast chamber panel. The first projectile (blind hub) with a mass of 1,778 kg is specified with a velocity of 15 m/s. To ensure containment of much smaller objects (largely due to having greater impact loads at the point of contact) an 8.5 kg projectile (blind hub) was fired at 217 m/s, equivalent to the same 0.20 MJ of energy. All components are run at ambient conditions (20°C) and are subjected to gravity. The friction co-efficient assumed between all components of the simulation is 0.15, with the eroding contact algorithm selected for components subjected to the highest impact loads in the simulation to aid failure.

5. Simulation Limitations

Material parameters that have been generated are kept as low in number and as simplistic as possible for material model development. It is noted that variables such as stress triaxiality and strain rate dependency [3],[4],[6] would need to be considered in greater detail if ultimate accuracy was the driver for the simulation. As FO&GT are trying to generate an overview of blast chamber performance and consider its functionality up to a maximum impact energy of 0.2 MJ, this level of detail is felt unnecessary at this stage of the development process.

Mass scaling has been specified for very small sections of components where stable time increments exceeding much more than a few points above 10^{-7} seconds are present. Components of interest were broken down into smaller geometric regions, tied together, then specified with semi-automatic mass scaling throughout the time-step, with an order of magnitude larger than the best values achieved via alteration of the mesh size alone. This allowed the solution to run over a much quicker time period. Whilst the effects of mass scaling are considered small in the initial global model, further investigation will need to be carried out in due course to assess its impact on the final solution.

6. FEA Validation

In order to justify the simulation techniques and methodology adopted within this first attempt at the analysis of the blast chamber, it is important to assess the ability of the blast chamber design to absorb a given amount of energy through component flexure, but secondly it is also important to assess the type of projectile that may cause such component flexure, and ultimately, failure.

The presented values suggest that the amount of energy going into the dampers is significantly reduced and that the steel blast plates are flexing to a greater degree than that assumed in the calculations and are therefore being subjected to greater load than estimated in the hand calculations. This is difficult to predict theoretically as the FE simulation considers elasto-plastic material behavior, whereas the hand calculations do not. The same argument is put forward for the difference in stresses for the main support column.
Theoretical calculations do not consider the surface area of the impact face of the projectile, but an impact load acting at a given height above the ground. This is thought to be influential (however it is not known by how much at present) in the overall deflections seen in the pine and oak sleepers, the energy transferred through these, and the final deflection of the steel blast plates. Further investigations will need to be undertaken to gather more data.

Table 2: Comparison of key variables in the blast chamber design between theory and FEA simulation results with 1,778 kg projectile at 15 ms$^{-1}$.

<table>
<thead>
<tr>
<th>Data Assessed</th>
<th>Theory</th>
<th>FEA</th>
<th>(+/-) %</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deflection of polyurethane damper</td>
<td>18 mm</td>
<td>9.08 mm</td>
<td>-49 %</td>
<td>Average of 18 dampers around the steel plate before disc spring is energized.</td>
</tr>
<tr>
<td>Average bolt force</td>
<td>23,125 N</td>
<td>22,347 N</td>
<td>-4 %</td>
<td>To resist column moment at maximum deflection</td>
</tr>
<tr>
<td>Deflection of column</td>
<td>3.31 mm</td>
<td>7.83 mm</td>
<td>+137 %</td>
<td>Recorded at height 1 m above ground before disc spring is energized.</td>
</tr>
<tr>
<td>Stress in main support column</td>
<td>519.7 MPa</td>
<td>373.2 MPa</td>
<td>-28 %</td>
<td>Resulting from deflection (theory limited to elastic stress)</td>
</tr>
</tbody>
</table>

7. Blast Chamber Simulation Results

Figure 13 and figure 14 show the performance of the blast chamber over a step time of 0.21 seconds with the 1,778 kg projectile travelling at 15ms$^{-1}$. The results show that whilst there is no penetration of the projectile through the steel blast plate, there is significant damage to the L-shaped horizontal reinforcing plates and the larger steel blast plates (where stresses are very close to the UTS of the material). Large deflections are seen in the base plates of the vertical columns, as well as in the pine and (especially) the oak sleepers.

The simulation results presented in figure 13 and figure 14 indicate that the design and relative position of the blast chamber components to one another (such as they’re positioning and spacing through the overall thickness of the blast chamber wall), cumulatively allows these individual components to withstand impact energies up to 0.2 MJ, preventing a breach of containment.

The finished simulation results (t=0.21 secs) show that whilst the oak sleepers are subjected to high levels of deflection under a 0.2MJ impact from a large surface area projectile, they do not fracture, maintaining their form and would not necessarily need replacing after a high energy impact from a large projectile. It is also shown that the polyurethane dampers do not sustain significant damage during impact and are able to cope with the high levels of dynamic load resulting from this simulation.
Figure 13: Left to right; Illustration of deflection and stress through the central plane of the middle blast chamber panel at time $t=0$, $t=0.014$, $t=0.049$, $t=0.0805$ secs.

Figure 14: Left to right; Illustration of deflection and stress through the central plane of the middle blast chamber panel at time $t=0.1085$, $t=0.1435$, $t=0.1820$, $t=0.210$ secs.

Further simulation work (see figure 15) undertaken on the much smaller projectile mentioned in Section 4.3 (8.5 kg at 217 ms$^{-1}$) again showed no penetration of the projectile through the outer steel plates of the blast chamber. This indicates that the blast chamber appears robust enough to
withstand not just a given amount of impact energy, but also a much more concentrated impact load, as would be the case with a heavily reduced frontal surface area of this much smaller projectile. The larger impact force suggests there is enough energy for the projectile to penetrate the oak sleeper (see figure 15) in the initial part of the impact, but that the projectile is sufficiently slowed down that the steel blast plate is able to withstand the remaining energy in the system, even at such high impact velocities.

![Figure 15: Top left to bottom right; illustration of deflection and stress through the symmetry plane of the middle blast chamber panel from t=0 to t=0.1855 secs.]

8. Conclusion and Future Work

This paper has documented the detailed set-up required to produce a working simulation model representative of a custom designed blast chamber for projectiles under high pressures and velocities. This has been achieved through the use of simplistic elasto-plastic material models for pine and oak wood calibrated to physical tests, analysis models built to capture the features of ballistic testing of steel plates, and the use of established, company specific, dynamic elastomeric material models.
The simulations that have been carried out under the most extreme conditions have shown that blast chamber integrity is maintained up to a total impact energy of 0.2 MJ, with sacrificial failure of a selection of structural components.

Based on the flexure seen within the global model, additional design changes will be incorporated to further strengthen the containment characteristics of the structure. Further factory testing will also be scheduled on a mock-up wall section (probably in the guise of a drop test) to provide additional correlation data with the hand calculations already undertaken and assessed by this FE simulation work. The model that has been created will also be made available for future reference and use when specific high energy tests are known as part of new risk assessment criteria.

9. References


10. Acknowledgements

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