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FPSO response to fast transient dynamic events

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FPSO response to fast transient dynamic events

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The Health and Safety Executive (HSE) have a keen interest in the response of Offshore Structures to air blast loading arising from gas explosions. Topside structures of many in-service platforms have traditionally been designed using static codes of practice, which mainly consider loads from the environment and from deck mounted equipment. Often no allowance was made for any dynamic pressure caused by explosive loading.

This report is the second in a series that seeks to harness the methodology and experience built up within the defence research communities concerned with air blast on warships. For many years QinetiQ Rosyth has been using theoretical and experimental methods to assess the response of warships to high explosive (HE) blast loading.

In the first report of the series, entitled 'Explosion resistance of floating offshore installations'1 QinetiQ carried out preliminary non-linear finite element (FE) analyses of four different types of floating offshore platforms.

With this second report, the deck structure of one floating production, storage and offloading unit (FPSO) is FE-analysed in more detail to assess the ability of the deck structure to withstand loading from an idealised gas explosion. The vessel chosen is ‘Vessel C’ of the first report.

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CONTENTS

EXECUTIVE SUMMARY

At the request of the Health and Safety Executive (HSE), QinetiQ Rosyth have adapted the technology developed for examining the effects of air blast on naval ships to the potential problem of gas explosions over the deck of an FPSO. This study is the second in a sequence and focuses in a more detailed way on:

- Spectral content of the gas explosion pressure pulse in relation to the natural elastoplastic response modes of the structure
- Influence of in-plane stresses in the deck arising from wave-induced hogging and sagging acting simultaneously with the gas explosion.

The report sets out the reasoning behind the approach adopted and the implied assumptions and limitations which are summarised as follows:

Outline of the methodology

Non-linear elastoplastic dynamic analysis using ABAQUS on a clamped one-quarter symmetry model of the top and sides of the central tank to predict:

Dynamic failure modes (if any)
Strains at panel edges.

Panel edge failure related to empirically determined failure strains in naval research:

Remote from intersection	10% strain
Within 60mm of an intersection	5% strain
Within 60mm of a weld HAZ	2% strain.

Worst case skewness (rise time/fall time) and pulse duration determined parametrically.
Wave pre-loads represented as an initial time = 0 uniform stress field.
Material modelled with bilinear stress strain curve without strain rate enhancement.

Assumptions and limitations

Idealised triangular pressure pulse, uniformly distributed over deck.
No effects arising from deck equipment weight or inertia and fluid in tanks.
Inertial resistance of ship either side of central tank sufficient to clamp the tank.
Clamping the tank will over-estimate damage into the deck and tank walls.
Effect of loss of in-plane stiffness of damaged deck can be examined as a separate, later event.
No consideration of residual strength of hull girder.
Edge failure data from symmetric ‘dynamic pull’ tests will apply to:

decks pushed onto a bulkhead by an explosion
decks fillet welded to a hull side.

Conclusions

The deck and central tank on an FPSO is likely to survive a gas explosion with a peak pressure of 0.2MPa and worst case spectral content. A permanent dent of 0.3 to 0.4 metres is anticipated.

A 0.4MPa peak pressure gas explosion over the deck of an FPSO will cause a 1 to 2 metre dent but the structure is not predicted to undergo any kind of dynamic buckling failure. Strains around the edges of panels where plate overlaps bulkheads or where plate is welded to the hull sides will be very high and may exceed the levels for which plates are expected to survive near joints and heat affected zones of welds.

Edge stresses and strains are not particularly sensitive to the skewness or duration of the gas pressure pulse. Permanent dent amplitude changes little with gas pulse skewness but increases monotonically with pulse duration (impulse).

Edge stress is not particularly sensitive to hogging or sagging stresses occurring simultaneously with the gas explosion. Sagging alleviates and hogging increases the edge stresses. Permanent dent amplitude increases monotonically under a sag moment simultaneous with the explosion and decreases in a simultaneous hogging moment.

The residual hull girder strength following a gas explosion over a cargo tank deck of an FPSO is likely to be determined by the dent amplitude in the deck, assuming the deck survives the explosive event in the prevailing sea state.

Recommendations

Dynamic, non-linear elastoplastic FE analysis of credible explosions over the deck of a central cargo tank should be carried out as a normal part of safety case assessments for FPSO operations. The combined effect of gas blast and wave loading should be considered.

If the hull around the tank is predicted to survive as regards dynamic buckling, an assessment should then be made of the strains around the edges of panels, especially the edges of the deck per se. These strains should be related to perceived failure strains for dynamically loaded plates near junctions and welds.

A mixed theory plus test program should be carried out to measure critical failure strains on non-welded and welded joints representative of FPSO construction. This should embrace cases with plate joints in tension, compression and shear.

In lieu of more relevant data, the edge failure strains quoted in this report should be used.

The residual strength of the hull with a deck damaged by the credible gas explosion should be evaluated for a range of sea states.

1. INTRODUCTION

1.1 BACKGROUND

The Health and Safety Executive (HSE) have a keen interest in the response of Offshore Structures to air blast loading arising from gas explosions. Topside structures of many in-service platforms have traditionally been designed using static codes of practice, which mainly consider loads from the environment and from deck mounted equipment. Often no allowance was made for any dynamic pressure caused by explosive loading.

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In the first report of the series, entitled '*Explosion resistance of floating offshore installations*'[1] QinetiQ carried out preliminary non-linear finite element (FE) analyses of four different types of floating offshore platforms.

With this second report, the deck structure of one floating production, storage and offloading unit (FPSO) is FE-analysed in more detail to assess the ability of the deck structure to withstand loading from an idealised gas explosion. The vessel chosen is 'Vessel C' of the first report.

1.2 STUDY OBJECTIVES

Specific objectives of the present study are as follows:

- Study the sensitivity of deck behaviour to variations in dynamic loading characteristics, namely: pulse shape, peak pressure and pulse duration.

- Consider the influence of in-plane deck stresses arising from realistic wave loading on the blast performance of the vessel.

- Improve the modelling and solution strategy where appropriate and consider the effects of truncated edge conditions.

1.3 LIMITATIONS

The analysis does not consider the residual strength of the FPSO after explosion. For example, the deck and FPSO may survive the explosion in a given sea state but the deck may then provide insufficient stiffness and strength to the hull girder to resist wave loading in higher sea states either on station or in transit. It is hoped that further studies will examine this related issue.

2 ASSESSMENT PROCEDURE

2.1 INTERPLAY BETWEEN THEORETICAL ANALYSIS AND EXPERIMENT

As in the previous report[1], the assessment procedure used in this study is entirely numerical and has no direct experimental validation. This approach contrasts with that adopted over the last few decades by the defence research community. With any new genre of structure or new analytical tool, extensive experimentation was traditionally used to validate the analysis for that new structural/load application. This was done either by means of

- Complete scale models e.g. a bulkhead plate exposed to air blast
- Test on a sub-feature at full scale e.g. plate failure near a joint.

2.1.1 Failure of plates near welded and non-welded joints

One area that has been found difficult to characterise by FE analysis alone is the failure around a flat bulkhead or deck under explosive load[2]. This is particularly difficult where high strains occur near weld heat affected zones (HAZs). Very often this will be near edges of panels where they abut stiffeners or at the edges of bulkheads or decks where they intersect the rest of the hull structure.

While thick-shell FE elements can in principle be used to predict through-thickness shear and elastoplastic failure in shell structures, it had been found preferable to adopt a mixed ‘theoretical and experimental’ approach. Figure 1 shows a laboratory specimen used to assess the dynamic performance of a typical plate-stiffener joint. The specimen is first pre-tensioned to reflect the loaded span of a real bulkhead and loaded dynamically until the failure strain is reached.

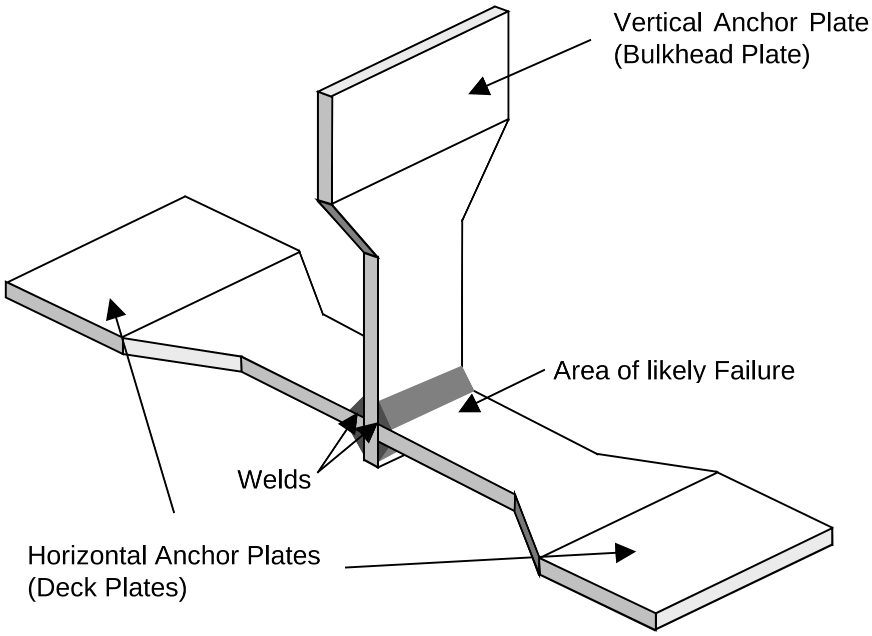


Figure 1 'Dynamic-pull' specimen

Strain rates applied are representative of those observed or expected for a particular explosive event. The laboratory 'dynamic pull' tests allow the relative merits of different types of joint to be evaluated. More importantly, for a specified type of joint, they allow the failure strain at the point of failure to be determined by:

- Direct measurement
- FE simulation of the test
- Corroborative simulation and test.

From this mixed programme, admittedly on a rather different set of stiffened steel plates, the following conclusions are drawn:

Panel failure under blast loading is best characterised in terms of the strain around the edges of panels near joints with stiffeners or with intersecting hull:

- Away from the edges, panels can take 10% strains without failing
- At non-welded T-joints, a safe allowable strain is 5%
- Near the HAZ of a welded T-joint, failure can occur at only 2% strain.

Armed with this data, non-linear elastoplastic FE analysis can then be used to simulate the explosive event on the complete structure and zones of high strain examined to assess the likelihood of failure due to that specific event.

By definition the 'dynamic pull' test seeks to pull one structural member off another plate, which is both orthogonal to it and symmetrically placed with respect to the direction of pull. The applicability of this data to the problem at hand might be questioned:

2.2 EXTENT OF FE MODEL

2.2.1 Structural details

Vessel C is a tanker refitted as an FPSO with overall length of approximately 250 metres. The analysis of this study focuses on the central tank, which has the following salient features:

- Length 24m
- Width 34m
- Draught 19m
- Main deck 30mm thick
- Frame spacing is approximately 4.8m
- Frames approximately 12mm thick.

Deep transverse frames support the hull at 4.8 metre spacing and these are stiffened by orthogonal small frames. Longitudinal small frames also stiffen the deck and double hull sides. A variety of different steels have been used.

2.2.2 Modelling options

Various options exist at least in principle for modelling the problem at hand:

Option 1	Full model of the entire FPSO with hydroelastic fluid loading
Option 2	Full model of the FPSO with added fluid mass or idealised restraint
Option 3	Detailed model of the central tank + beam model of the rest of the hull
Option 4	Detailed model of the central tank plus clamped boundaries
Option 5	Detailed model of the top and sides of the clamped central tank.

Feasibility and relative merits of these options are summarised as follows:

Option 1	Beyond the budget of the present study!
Option 2	Restraints impossible to specific reliably
Option 3	Attachment of idealised beam will give unrealistic inertial loads
Option 4	Will over-estimate blast energy going into the central tank
Option 5	Will over-estimate blast energy going into deck and tanks walls.

Options 1-3 are large calculations. Options 2-3 add little to reliable prediction. Options 4 and 5 have merit but the implication of the boundary restraint deserves discussion. Option 5 was used in reference 1 where the bases of the side walls were clamped.

By clamping the tank, it is implicitly assumed that the inertial resistance of approximately 100 metres of steel hull either side of the tank is sufficient to prevent the rest of the FPSO accelerating inwards during the time of the explosive impulse. On the basis of intuitive physical reasoning and past analytical experience corroborated by test, it is inferred that more damaging energy from the blast per se will go into the tank and in the case of Option 5 more energy will go into the deck and side walls. Option 5 is therefore a safe over-estimate of the likely damaging effects.

It should be emphasised that the assumed clamped boundary condition is not necessarily optimal as regards the simulation of what may happen to the hull shortly after the explosion. Within time constants more appropriate for wave loading, the hull may sag and impose additional compressive strain on the already damaged deck and side walls. Analysis with Options 4 and 5 will not allow the prediction of this latter effect i.e. residual strength to resist wave loading will not be predicted. But subsequent analysis of the response to wave loading with the damaged deck (and with different restraints) should be conservative because the deck has been damaged more than it would have been if the hull had been allowed to move!

With a really full tank with little entrapped air, load could be transferred from the deck to the double bottom of the hull by a hydraulic mechanism. However, HSE confirm that tank levels are unlikely to be sufficiently close to the deck for this effect to occur, obviating both the need to consider entrained fluid effects and therefore the need to explicitly model the double bottom of the hull.

A quick analysis of the original FE model[1] but with simple support restraints, allowing the side panels to rotate at the base, indicated very little change in deformation and stress. Option 5 was therefore chosen as the most damaging to the deck and the side walls but evidently not unreasonably over-conservative. Clamping the base prevents global displacements, simplifying the specification of boundary conditions.

2.2.3 Use of symmetry

The tank is symmetric about two planes and the loading is assumed to be a spatially uniform time varying pressure pulse i.e. it is assumed that there are no local directional blast effects which might punch through into a particular deck panel.

In addition, it is also assumed that inertial loads from deck equipment can be ignored. Heavy deck equipment is mounted directly over the inner, outer or transverse (bulkhead) walls of the tank so that load is transferred down into the hull by compression and shear rather than bending of the deck plate.

With these assumptions, one-quarter symmetry can therefore be invoked. This considerably reduces the size of model in terms of number of degrees of freedom and therefore reduces computation time.

Axial symmetry could have been applied as a moving-plane of symmetry on one end of the model. In fact, both ends have been modelled as symmetry planes by suppressing axial displacements and the rotations in the two orthogonal planes. A potential complication of this boundary restraint is that is it precludes some options for applying the static pre-loads associated with wave loading, namely:

- Applied forces will be immediately reacted by the constraints
- The forces associated with imposed displacements will be reacted by the constraints
- An initial pre-stress field will however be possible in some FE codes.

Consideration of the options for applying pre-load has had a strong influence of the choice of software for the problem at hand.

2.3 Selection of software

2.3.1 Dynamics of the problem

Figure 2 shows a typical ‘normalised’ pressure pulse used to simulate a gas explosion. The pulse shown has the following characteristics:

- Rise time 60 milliseconds
- Decay time 240 milliseconds
- Duration 300 milliseconds

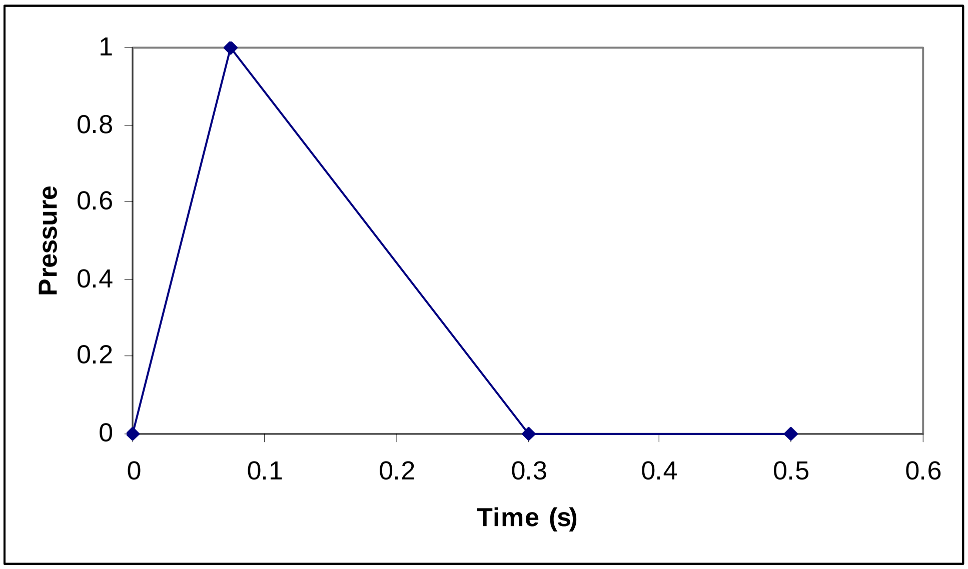


Figure 2 Typical idealised pressure pulse from a gas explosion

Reference 1 concluded that the structural problem lay in the ‘dynamic’ regime i.e.

- The periods of the normal modes were too long for the problem to be considered quasi-statically i.e. with a uniform static pressure on the deck

- The periods of the normal modes were not long enough for the problem to be considered as a simplified impulse problem

The need therefore was for a code that would allow:

- Non-linear dynamics including moment-destabilisation failure modes e.g. dynamic buckling with the hull folding in transversely as the deck deforms laterally

- Accurately predicted strains at the edges of panels to relate to the empirically determined failure strains

Two options for software were considered for a full non-linear elastoplastic dynamic analysis:

- ABAQUS/standard - uses only implicit solution methods
- LS DYNA - uses explicit solution thought likely to expedite solution.

2.3.2 Problem size

Experience at QinetiQ Rosyth suggests that for large and highly non-linear problems with high plastic strain, explicit codes are generally most efficient, thus favouring the use of LS DYNA.

However, by reducing the problem size, focussing solely on the central tank and using one-quarter symmetry, the number of degrees of freedom has been kept low and use of ABAQUS was felt to be feasible.

2.3.3 Static pre-load

Implicit methods can accept a static pre-load specified at an initial (time =0) stress field so the potential problem arising from imposed constraints (section 2.2.3) could be overcome with ABAQUS.

2.3.4 Modelling of stiffener webs

Analyses carried out over many years at QinetiQ Rosyth have highlighted the importance of selecting the optimal method of modelling the webs of all types of stiffener. In general, webs represented, as shell elements are preferred if either of the two following conditions apply:

- Shear deformation is high compared to bending deformation, typically when the loaded structural span is short in relation to the depth of the web

- Sidesway of the stiffener, lateral bending of the web and rotational restraint between the toe of the frame and the contiguous plate are important, often when trip buckling of the stiffener is driving the plate instability

However, if shell elements are used to model stiffener webs then it has been found desirable to have as many as eight elements across the stiffener web to accurately reflect the bending of the web. This increases problem size and creates meshing difficulties where stiffener webs intersect bulkheads and tank walls, inevitably favouring minimal use of shell webs. However, with some beam elements, the rotational stiffness is too high, producing an unrealistic clamping on the edge of attached plate.

Over many years structural analyses have been carried out by QinetiQ with ABAQUS models employing various options for stiffener modelling. These have been appraised and compared with experimental observation, leading to the conclusion that the ABAQUS B31 beam element is optimal if small frame webs are to be represented as discrete beams rather than shells**.** The preferred solution for the problem at hand was as follows:

- Use of shell elements for the deep transverse frames ideally with eight elements per web – shear deformation will be important for the deep frames

- Use of beam elements for the many small T-frames which stiffen the tank walls – shear deformation will be less important for the small stiffeners with longer length to depth ratios

2.3.5 ABAQUS selected

The non-linear finite element (FE) code ABAQUS was used for this study. The basis of this decision based on past experience is summarised as follows:

- Relatively small size problem allowing implicit solution
- Static pre-stress for wave load specified as an time=0 initial condition
- Good discrete stiffeners for small T frames

One additional advantage of using ABAQUS was that the existing model for Vessel C could be used as a starting point. This has been adapted for the present analysis using the finite element pre-processor PATRAN.

2.4 Material modelling

The steels used for the various components and their material properties are listed in Tables 5 and 6 of reference 1. Stress strain curves were bilinear, defined by the yield stress and strain and the ultimate strength and elongation.

3. RESULTS

3.1 CONVERGENCE STUDIES

Analyses have been performed with two mesh densities using both first order and second order elements. Figures 3 and 4 show the two mesh densities and typical displacements. The coarser mesh is the same as that used in reference 1 and is referred to as 'original mesh'. The 'fine mesh' has double the mesh density and with that mesh spacing has the perceived optimum of eight elements in the webs of the deep transverse frames.

Convergence analyses studies have been carried out at both low loads, for which strains are expected to elastic, and at high loads well into the elastoplastic regime.

Table 1a shows that in all cases peak displacements from a non-linear elastic analysis are within 10% of each other and the predicted frequency of the (elastic) fundamental mode is within 5%.

Table 1a Effect of mesh density and element order – elastic regime

Convergence case	*Displacement (m)*	*First natural frequency (Hz)*
Original mesh, 1st order	0.370	10.009
Original mesh, 2nd order	0.422	9.444
Fine mesh, 1st order	0.375	9.548
Fine mesh, 2nd order	0.406	9.427

In view of the largely heuristic objective of the present analysis and the uncertainty in some areas of interpretation, for example the applicability of the naval empirical data to the tanker deck scenario, it was decided that the original mesh with first order elements would be used for all subsequent analyses in order to reduce computation time. However, this judgement would be revised if for example:

- An actual case on the margins of acceptability was under consideration
- Accurate failure strain data were available.

The importance of these convergence studies should not be under-estimated. Figures 5 and 6 show the plastic strains for both coarse and fine mesh models with 0.4 MPa blast load. Data shown in Figures 3-6 is summarised in Table 1b.

Table 1b Effect of mesh density and element order – elastoplastic regime

Mesh	*Peak displacement in m*	*Strain in %*
Coarse 'original'	1.78	5.61
Fine	2.06	7.70

Displacements are 15% bigger with the refined mesh but edge strains are 37% bigger than those seen on the coarse mesh.

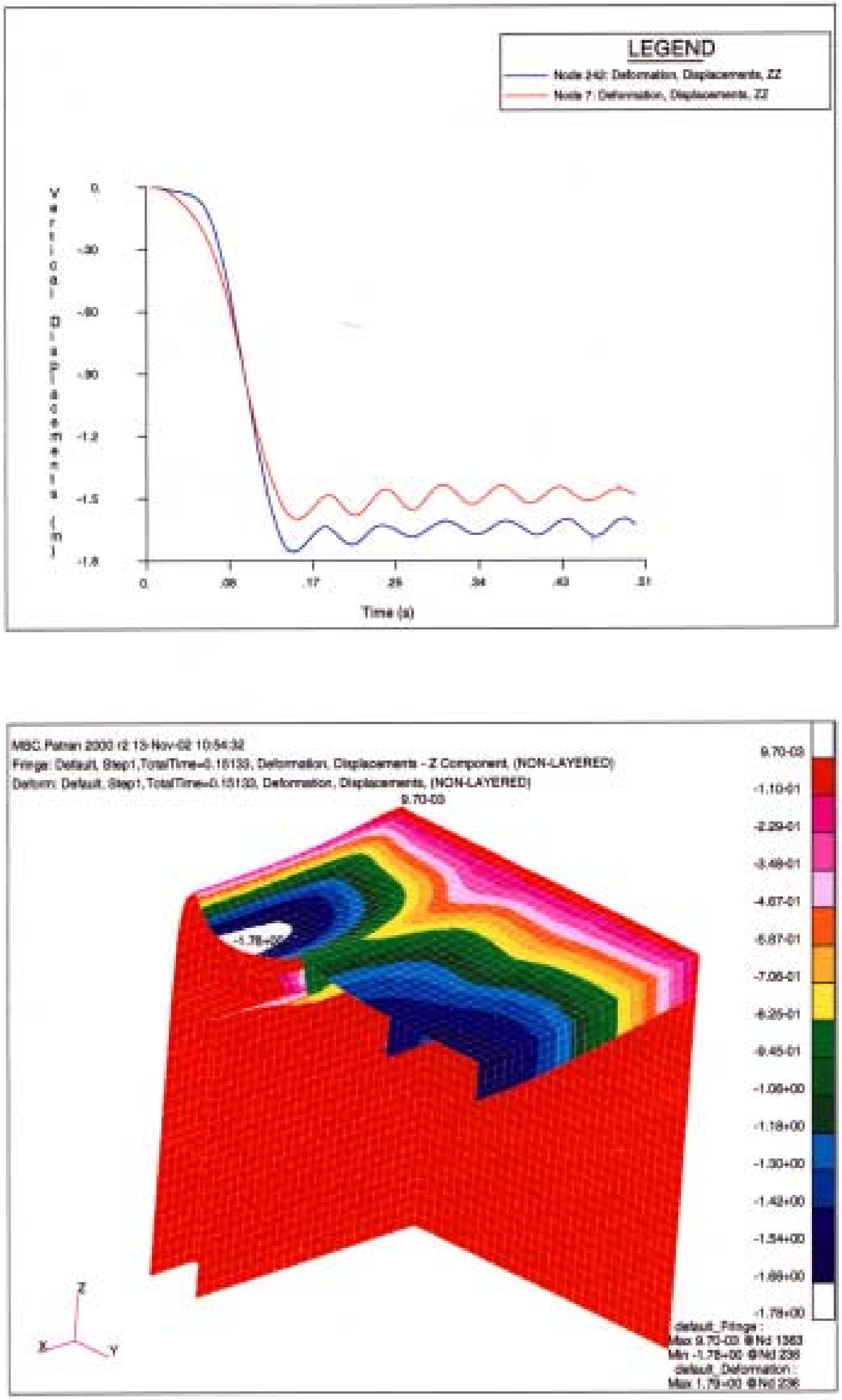


Figure 3 Coarse 'original' mesh and displacements with 0.4 blast load

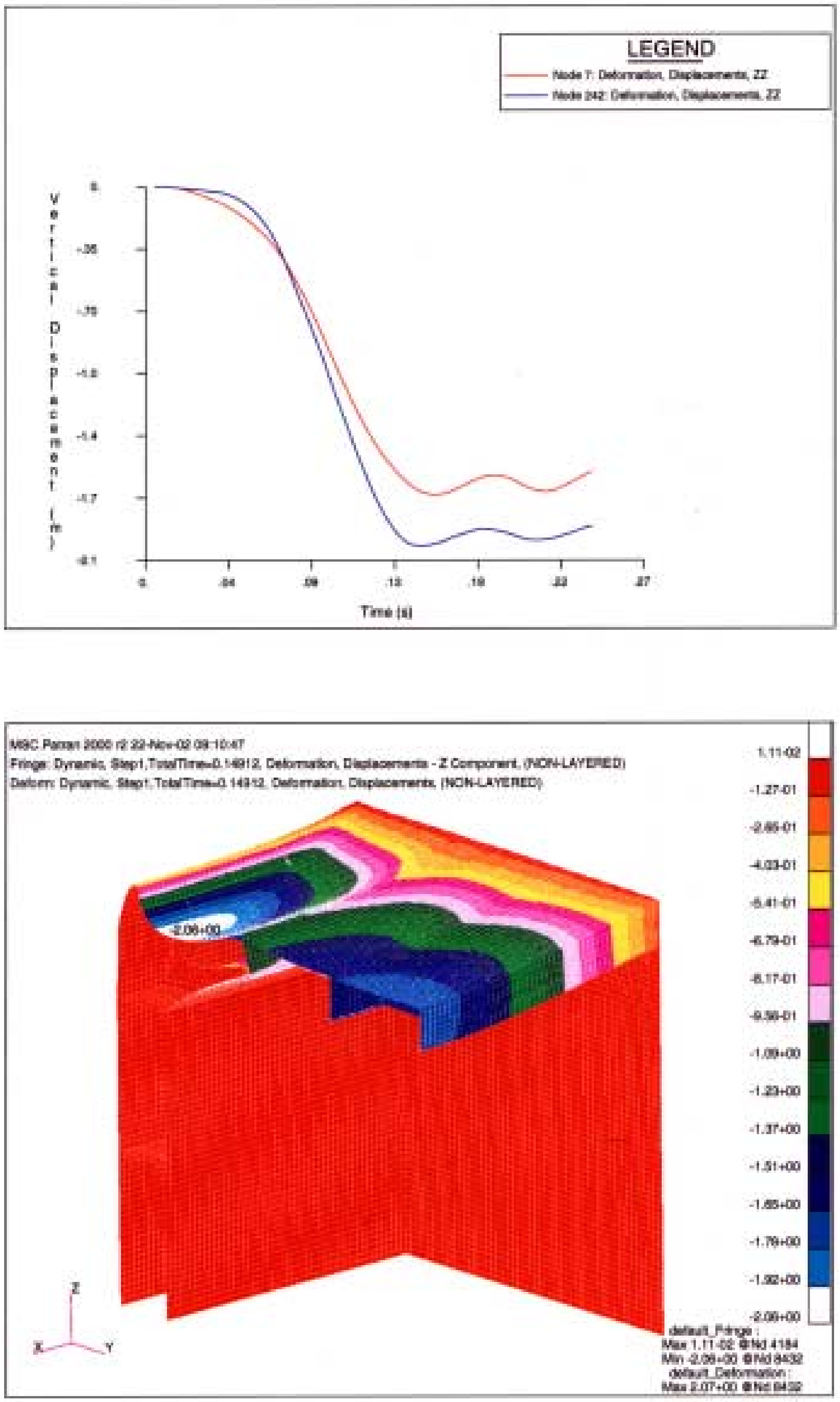


Figure 4 Fine mesh and displacements with 0.4 MPa blast load

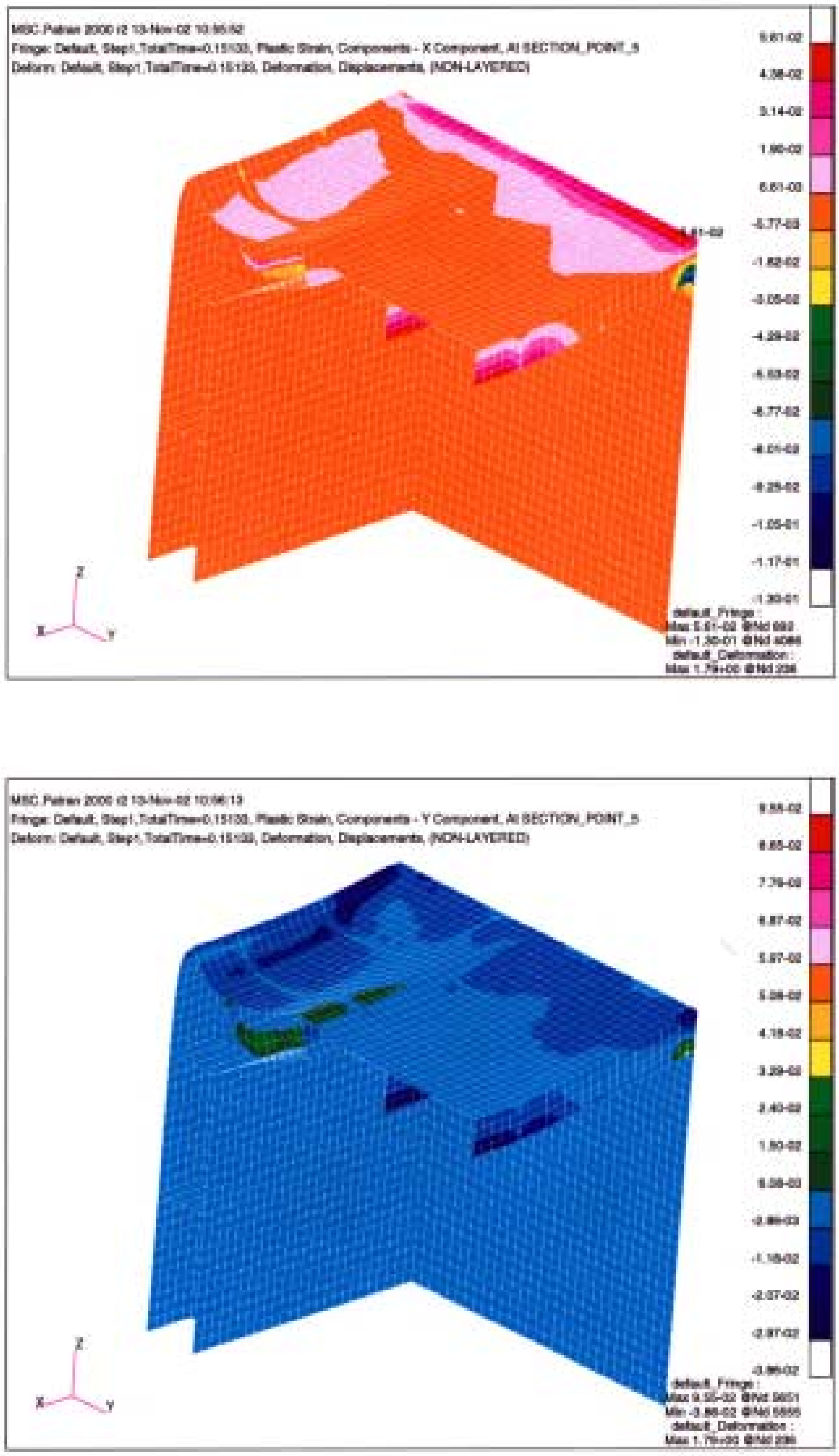
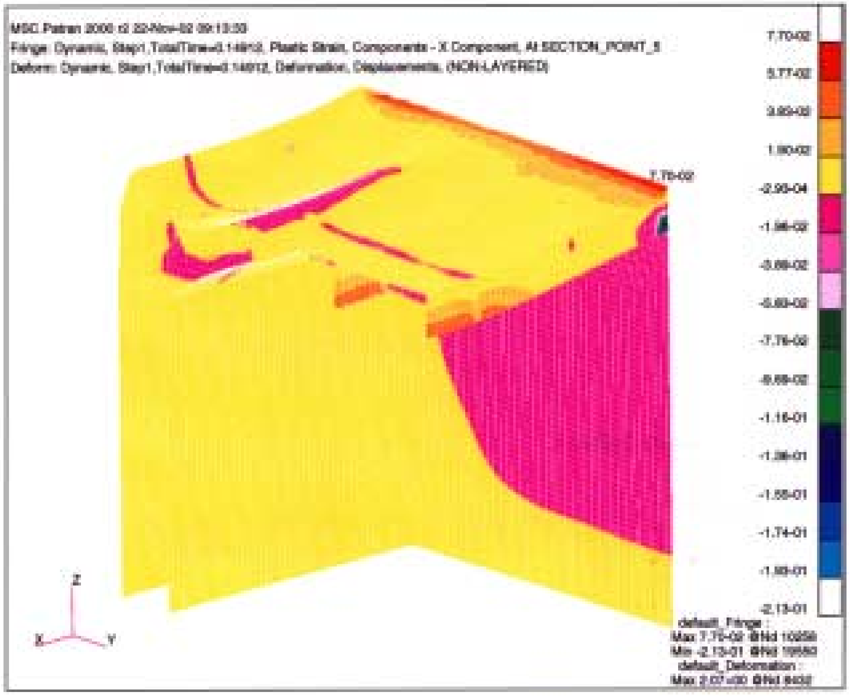


Figure 5 Coarse ‘original’ mesh – plastic strains with 0.4 MPa blast load



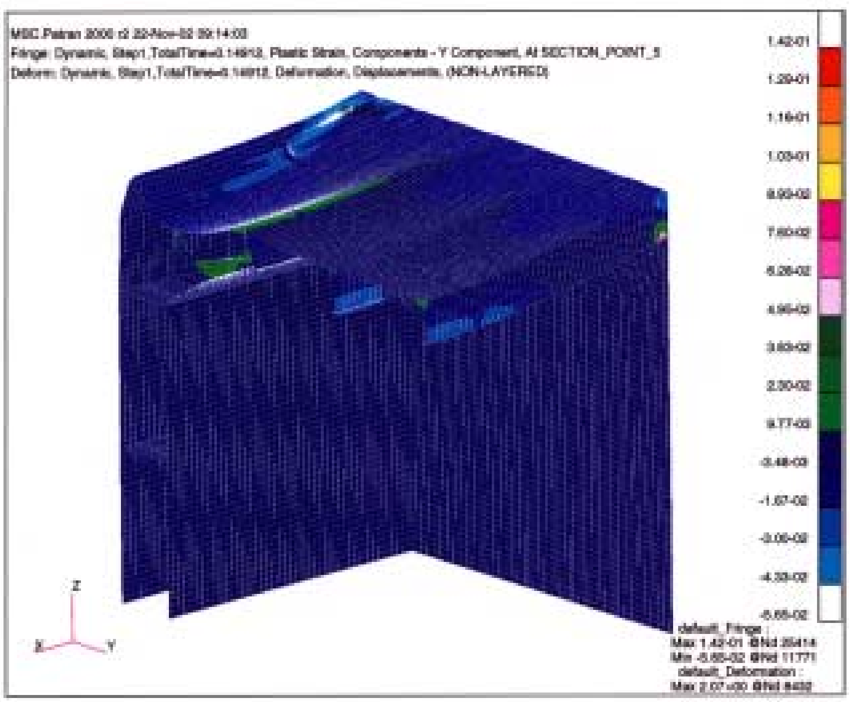


Figure 6 Fine mesh – plastic strains with 0.4 MPa blast load

3.2 PRESSURE PULSE SKEWNESS

Analyses have been carried out for various pressure rises and decay times within the anticipated range for a gas explosion. Tables 2 to 4 show data for two values of peak pressure:

- 0.2MPa which was expected to give stresses well within the elastic limit
- 0.4MPa which was anticipated to be on the margins of survivability

Table 2 Dynamic pressure pulse characteristics - pulse skewness

Load case	*Peak Pressure (MPa)*	*Rise time (ms)*	*Decay time (ms)*	*Duration (ms)*
ELS1	0.2	50	200	250
ELS2	0.2	75	175	250
ELS3	0.2	100	150	250
ELS4	0.2	125	125	250
ELS5	0.2	50	75	125
ELS6	0.2	75	100	175
ELS7	0.2	100	125	225
FLS1	0.4	50	200	250
FLS2	0.4	75	175	250
FLS3	0.4	100	150	250
FLS4	0.4	125	125	250
FLS5	0.4	50	75	125
FLS6	0.4	75	100	175
FLS7	0.4	100	125	225

Table 3 Elastic limit results – pulse skewness

Case	*Max displacement (m)*	*Max x stress (E8Pa)*	*Max y stress (E8Pa)*
ELS1	0.3	2.56	2.80
ELS2	0.351	2.53	2.21
ELS3	0.409	3.14	2.48
ELS4	0.432	3.31	2.52
ELS5	0.2	3.28	2.75
ELS6	0.262	3.03	2.54
ELS7	0.381	2.87	2.54

Table 4 Failure results – pulse skewness

Case	*Max displacement (m)*	*Max x stress (E8Pa)*	*Max y stress (E8Pa)*	*Max x strain*
FLS1	1.62	4.59	3.28	0.0543
FLS2	1.69	4.50	2.71	0.0554
FLS3	1.70	4.58	2.97	0.0566
FLS4	1.66	4.57	2.91	0.0630
FLS5	1.14	4.14	2.92	0.0493
FLS6	1.48	4.40	2.83	0.0497
FLS7	1.66	4.54	2.80	0.0555

In Tables 3 and 4, the x-stress is in the fore-and-aft direction and the y-stress is in the port-and-starboard direction. The results are remarkably insensitive to ‘skewness’ and any attempts to pick out a ‘worst case’ are hampered by the fact that worst case displacements, stresses and strains do not coincide. The case ‘rise time/decay time = 1/4’ has been taken as the ‘worst case’ on the basis of highest stresses and high displacement.

3.3 PRESSURE PULSE DURATION

A further series of calculations have been carried out with the ‘worst case’ skewness but with pulse duration varied from 100 milliseconds up to 300 milliseconds. In each case the peak pressure was 0.4 MPa. Data are given in Tables 5 and 6.

- As might be expected the peak displacements increase monotonically with the impulse i.e. longer duration gives rise to a bigger dent!

- The increase in stress with pulse duration is less marked but almost monotonic.

- Maximum x-strains are all in the 5% region and vary little and with no indication of monotonic change with impulse.

Table 5 Dynamic pressure pulse characteristics - pulse duration

Load case	*Peak pressure (MPa)*	*Skewness factor*	*Duration (ms)*
FLD1	0.4	Worst case	100
FLD2	0.4	Worst case	125
FLD3	0.4	Worst case	150
FLD4	0.4	Worst case	175
FLD5	0.4	Worst case	200
FLD6	0.4	Worst case	225
FLD7	0.4	Worst case	250
FLD8	0.4	Worst case	275
FLD9	0.4	Worst case	300

Table 6 Effect of loading duration - results

Case	*Max z Disp (m)*	*Max x Stress (E8Pa)*	*Max y stress (E8Pa)*	*Max von Mises stress (E8Pa)*	*Max x Plastic strain*	*Max y Plastic strain*	*Max x strain*
FLD1	1.01	4.02	3.13	4.68	0.0585	0.066	0.0588
FLD2	1.16	4.19	3.28	4.75	0.0577	0.0837	0.0582
FLD3	1.27	4.34	3.27	4.79	0.0481	0.0901	0.0499
FLD4	1.39	4.27	3.31	4.48	0.0522	0.0978	0.0528
FLD5	1.52	4.47	3.74	4.76	0.0504	0.0954	0.0524
FLD6	1.62	4.56	3.41	4.83	0.052	0.0961	0.054
FLD7	1.69	4.59	3.26	4.85	0.0532	0.0963	0.0551
FLD8	1.75	4.6	3.16	4.85	0.054	0.0941	0.056
FLD9	1.79	4.65	2.85	4.86	0.0561	0.0955	0.0582

3.4 EFFECT ON IN-PLANE STRESS IN THE DECK DUE TO WAVELOAD

Analysis has been performed on four cases of deck in-plane stress with the pressure pulse skewness and duration of the worst case spectral case FLD9. These cases are as follows:

HOG100	100MPa tensile stress
HOG200	200MPa tensile stress
SAG100	100MPa compressive stress
SAG200	200MPa compressive stress

Peak displacements, edge stresses and strains are given in Table 7. For comparison, data is also listed for Case FLDG i.e. the ‘still water’ case with no pre-stress in the deck. Figures 7 and 8 give a variety of plots of displacement and strain with wave and blast loading. The figures can be compared Figures 3-6 which give still-water data. The Y-strains shown are for the top surface of the deck and they are compressive.

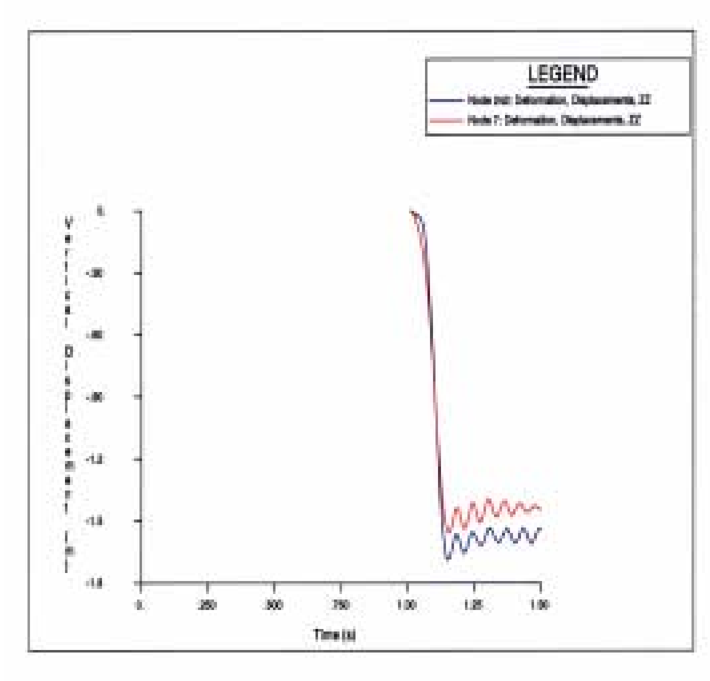
Table 7 Peak displacements, Von Mises stress and strains – wave loading

Case	*Displacement (m)*	*Von Mises stress (MPa)*	*X-strain – bulkhead*	*Y-strain - side*
HOG200	1.71	457	6.7 to 9.1%	-2.2%
HOG100	1.70	303	6.9 to 9.0%	-2.1%
Still water	1.78	420	4.6 to 5.8%	-2.1%
SAG100	1.84	420	4.6 to 5.9%	-2.2%
SAG200	1.92	460	5.4 to 7.6 %	-2.7%

The positions of peak displacements are shown clearly on the figures. The peak x-stress refers to the deck plate where it wraps over the bulkhead. The peak y-stress quoted is in the deck close to the side of the hull.

There is a clear increase in peak displacement moving from ‘hog’ (with the deck in tension) to ‘sag’ (with the deck in compression).

However, the data is less clear cut when considering Von Mises stress and the strains around the edges of the deck. If anything, the sag cases appear to be less damaging as regards edge failure. If the deck is in a slight compression, the tension arising from the wrap of the plating over the bulkhead will have a lesser net effect in initiating failure locally.



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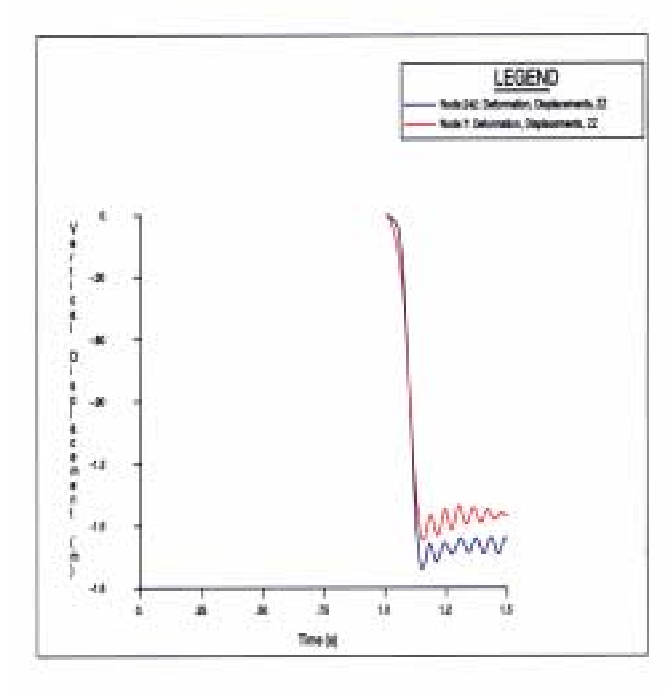

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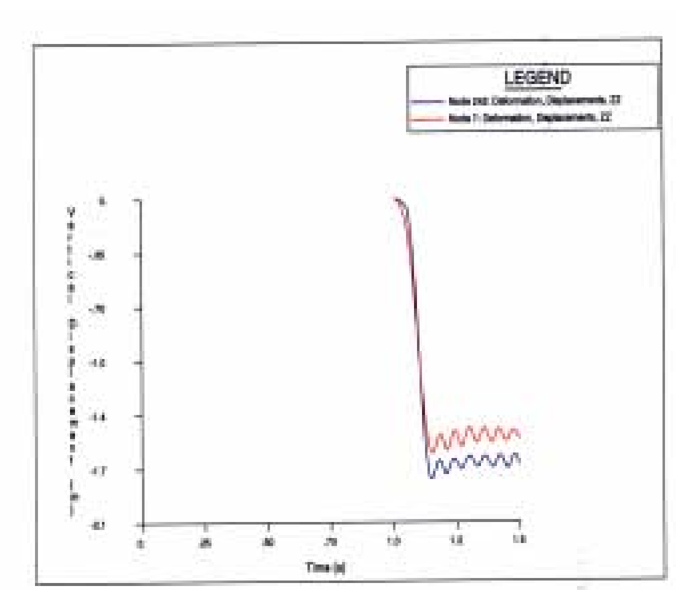

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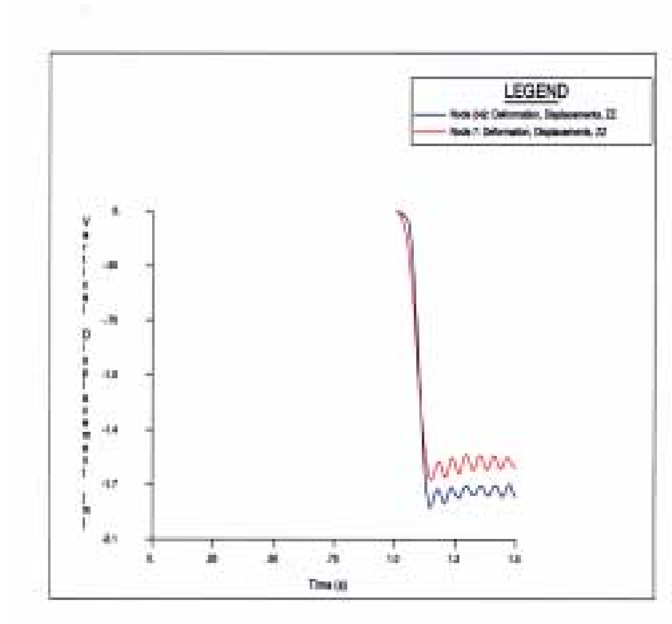

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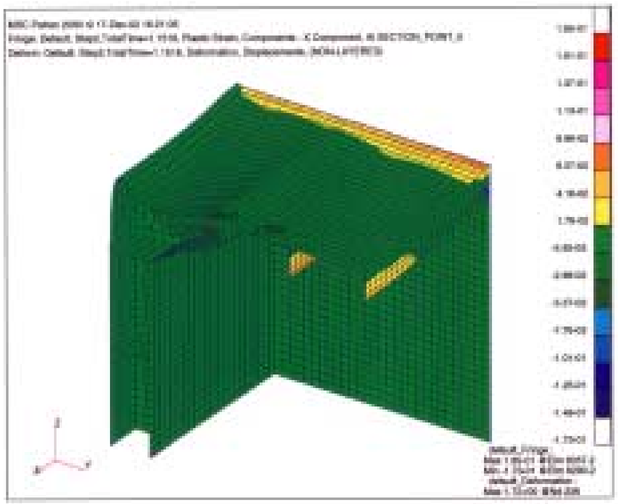
Figure 7 Effect of wave load on displacement



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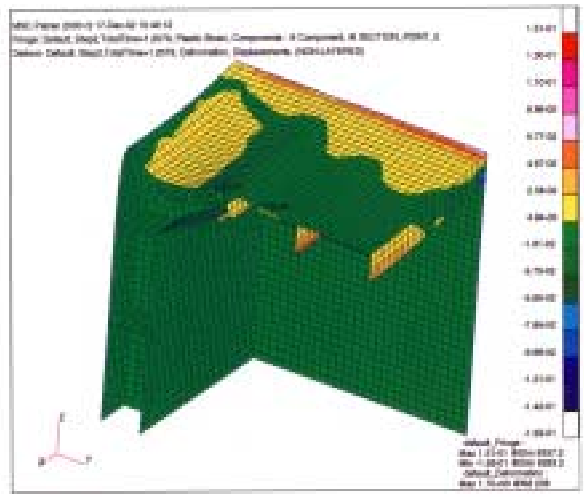

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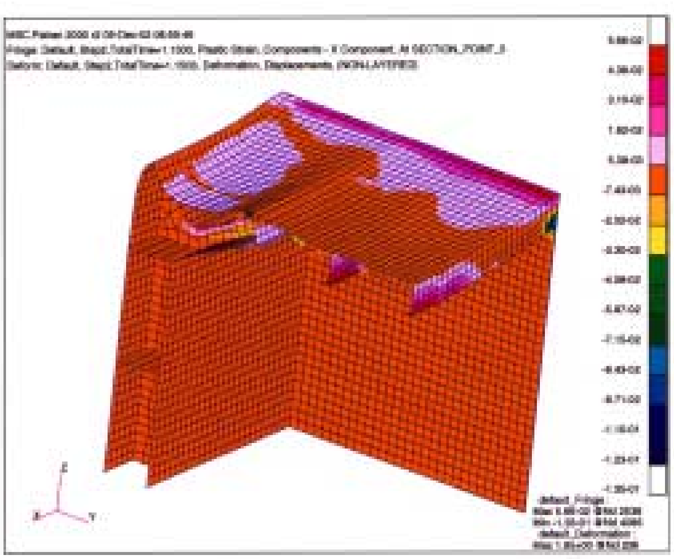

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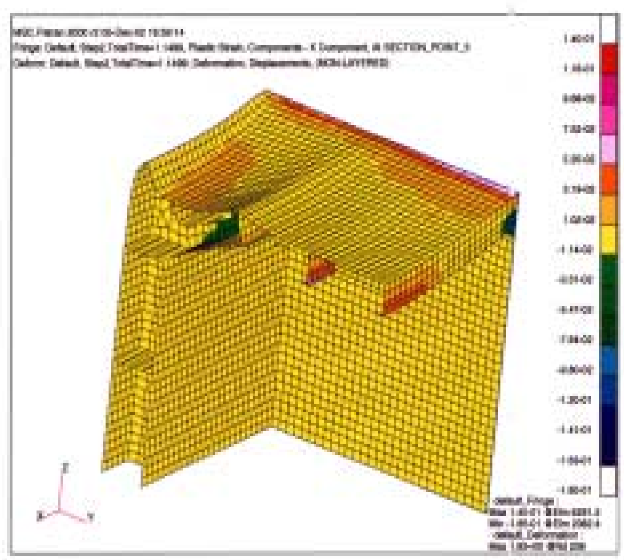

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Figure 8 Effect of wave load on X-strain

4. DISCUSSION

When interpreting the inferences of the previous section, it should be remembered that most of the FE data was acquired with the coarser of the two models. It has been shown that the coarse model under-predicts the displacement by 10% and 15% in the elastic and elastoplastic regimes, respectively. Plastic strains could be under-predicted by 30% to 40% with the coarse model.

Notwithstanding this loss of accuracy, the deck has been seen to not suffer any kind of catastrophic failure of a dynamic buckling kind. It has a permanent dent but is predicted by the analyses to be stable.

The criterion for panel edge failure must be imposed subjectively on the data and it too has attendant uncertainties having been derived from analysis and test on somewhat thinner plate in different types of steel. The important strain data for the cases listed in Table 7 may be summarised as follows:

- Deck strains are well above 5% in the x-direction where the deck goes over the bulkhead, presumably without a nearby weld.

- Deck y-strains are in the range 2 to 3% near the tank sides where there is a fillet weld between the deck and the hull.

Notwithstanding the finite but stable permanent set predicted in the FE analysis, it seems quite likely that edge failure will occur with a gas explosion leading to a peak pressure of 0.4MPa. Tensile strains in the deck where it wraps over the bulkhead are likely to cause rupture. Wave induced hogging would appear to increase the chance of edge failure while sagging will alleviate it somewhat, although the blast-induced edge strains are not particularly sensitive to either the nature or magnitude of the wave load.

Peak stresses quoted in Table 3 are just touching yield with a 0.2 MPa gas pressure pulse. It follows that strains, including those around the edges will be in the region of 1.5%. The deck structure would be expected to survive a 0.2 MPa peak pressure gas blast, although there will be a permanent dent in the deck.

When considering the residual strength of the hull to wave loading, hogging and sagging will affect the damage mechanisms differently, with much larger dents occurring when a blast is simultaneous with a sag moment. Smaller dents will occur with a hogging moment. The minimum residual strength following a credible explosion will therefore be determined by the sag-induced compressive load seen by the deck after it has been damaged by an explosion while undergoing simultaneous mid-hull wave-induced sag.

If the hull survives the explosive event, it may yet fail under higher sagging wave moment because of the reduced in-plane stiffness and strength of the deck. The contribution of the damaged deck to the hull girder second moment of area should be examined.

5. CONCLUSONS

Dynamic non-linear elastoplastic FE analysis has shown that the deck and central tank on an FPSO is likely to survive a gas explosion with a peak pressure of 0.2MPa. A permanent dent of 0.3 to 0.5 metres is expected.

FE simulation of a 0.4MPa peak pressure gas explosion over the deck of an FPSO suggests that a large permanent dent of perhaps 1.8 metres will result but the structure is not predicted to undergo any kind of dynamic buckling failure. However, the strains around the edges of panels where plate overlaps bulkheads or where plate is welded to the hull sides are very high and exceed the levels for which plates are expected to survive near joints and near heat affected zones of welds. The deck would therefore not be expected to survive a 0.4MPa gas explosion based on best-available failure data.

Edge stresses and strains are not particularly sensitive to the skewness or duration of the gas pressure pulse.

Permanent dent amplitude is not particularly sensitive to gas pulse skewness but it increases monotonically with pulse duration (impulse).

Edge stress is not particularly sensitive to hogging or sagging stresses occurring simultaneously with the gas explosion. Sagging alleviates and hogging increases the edge stresses.

Permanent dent amplitude increases monotonically with a sag moment simultaneous with the explosion and decreases with a hogging moment.

The residual hull girder strength following a gas explosion over a cargo tank deck of an FPSO is likely to be determined by the dent amplitude in the deck, assuming the deck survives the explosive event in the prevailing sea state.

6 RECOMMENDATIONS

As part of safety case assessments for FPSO operations, non-linear elastoplastic FE analysis should be carried out for credible explosions over the deck of a central cargo tank.

The combined effect of gas blast and wave loading should be considered.

If the hull around the tank survives, an assessment should then be made of the strains around the edges of panels and the edges of the deck per se. These strains should be related to perceived failure strains for dynamically loaded plates near junctions and heat affected zones of welds.

A mixed theory plus test program should be carried to measure critical failure strains on non-welded and welded joints representative of FPSO construction.

In lieu of more relevant data, the failure strains quoted in this report should be used.

The residual strength of the hull with a deck damaged by the credible gas explosion should be evaluated for a range of sea states.

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