APPLYING MSC/PATRAN AND MSC/NASTRAN TO IMPROVE THE DESIGN PERFORMANCE OF LARGE BULK MATERIALS HANDLING MACHINES

R C Morgan BE MEng Sc FIE Aust CP Eng
Principal Consultant - Structural, BHP Engineering Pty Ltd
Dr S Loganathan BscEng MEng PhD MIE Aust CP Eng
Senior Supervising Engineer, BHP Engineering Pty Ltd
Dr F Gatto BE MEng Sc PhD MIE Aust CP Eng
Senior Supervising Engineer, BHP Engineering Pty Ltd

ABSTRACT

Bulk materials handling machines such as stackers, bucket wheel reclaimers, ship loaders and ship unloaders are complex moving structures with onerous loading and service conditions. Due to such factors there is the potential for many failure modes and problem areas on these machines.

Machines are normally purpose designed for the particular application at a port, mine or industrial facility. In the current environment, high reliability standards are required in order to provide operational security for business and safety to personnel.

Dynamic, buckling and non linear behaviour of such structures can all contribute to failures and operational problems. Hence a sophisticated analysis and modelling tool is required in order to provide adequate representation under service conditions.

Due to the complexity and size of these structures, models with large numbers of degrees of freedom are required for analysis. It was previously very time consuming and costly to carry out such work, putting it out of the realms of commercial application. However the compatibility of MSC/PATRAN with CAD systems and the ease of building large models, together with recent developments in the efficiency of the solvers in MSC/NASTRAN, opens new frontiers for engineers to critically examine the design of new structures before they are built.

Similarly, with the application of MSC/PATRAN and MSC/NASTRAN, existing machines can be critically examined and cost effective life extension strategies developed to deliver major economic benefits to owners.

1.0 INTRODUCTION

The authors have been involved in many different aspects of large materials handling machines including stackers, reclaimers, shiploaders and shipunloaders. This involvement can be classified into three categories viz:

- Failure Investigations
- Proof Engineering

· Life Extension

The structures are usually quite complex comprising a combination of trusses, tension stays, plated box components as well as mechanical items such as slew rings, bearings, pins, wheels and drives. In many cases structural design standards, which have traditionally been developed for building works, are not directly applicable. Hence a more fundamental engineering approach is required in order to effectively design such machines to be economical as well as meeting safety requirements.

Traditionally, hand calculations using empirical formulae and more recently using simplistic computer modeling based on a frame analogy were adopted for the analysis of such structures. However the high incidence of failure of such machines (Ref 1) demands a more rigorous design approach.

The compatibility of MSC/PATRAN with computer aided drafting systems (CAD) and ease of creating complex geometry in MSC/PATRAN provides an order of magnitude reduction in modelling time compared to other commercially available systems. This has enabled the authors to apply detailed Finite Element Modelling to such structures on a commercial basis. Similarly improvements in the solver efficiency of MSC/Nastran now enables the authors to carry out the appropriate level of analysis in a realistic time frame for commercial design applications.

Analysis types commonly used are:

· Linear static analysis - Used to determine stress levels in components of the structures

· Linear (eiginvalue extraction) buckling analysis - used to determined load factors in platework structures.

· Non-linear analysis - Used to replicate failure modes including large displacement and material yielding behaviour.

MSC/Patran's post processing features allow very efficient visualisation of all aspects of the structural behaviour so that critical effects are not overlooked.

The remainder of the paper describes a critical application which demonstrates the power and accuracy of MSC/PATRAN and MSC/NASTRAN for modelling and analysis of these types of structures.

2.0 FAILURE INVESTIGATION

2.1 Problem Definition

The authors carried out an analysis to replicate the failure of a mast structure on a materials handling machine. The mast had buckled locally and rotated due to loading imposed during maintenance activities.

The aim of the analysis was to verify the stress levels at various regions of the mast and to determine the load factor under failure conditions. Operational wind loads and ore loads were not included in this investigation. Therefore, it was only necessary to model the components of the mast that contributed to the stiffness of the structure. It was noted that the geometry of the machine and the design loads were symmetric about an axis parallel to longitudinal direction of the boom. Hence, it was decided to model only one leg of the mast (ie, half model) in conjunction with a symmetric boundary condition in order to investigate the mast.

The finite element model was created using existing drawings to provide the dimensional information. The model was built using MSC/PATRAN. Centreline dimensions were used for the entire modelling of the mast. Where a connection was made in the structure, either by weld or bolting, it was assumed that the connection was perfect. However, the intermittent weld shown in the drawing for the flange and web stiffeners and diaphragm was incorporated as an element discontinuity (ie, artificially induced crack) in the model. Where the diaphragm was not connected to the tension side of the flange, no connection was made in the model. The mast was modelled above the rigid bolted joint at the slewing platform.

2.2 Analysis

The analysis was carried out using MSC/NASTRAN. The model largely consisted of MSC/NASTRAN'S QUAD4 (four noded isoparametric quadrilateral element with optional coupling of bending and membrane stiffness) elastic plate elements and, where unavoidable, TRIA3 (three noded isoparametric triangular element with optional coupling of bending and membrane stiffness) elastic plate elements. For QUAD4 and TRIA3, the forces and stresses were evaluated at the centroid of the element based on the element coordinated system. A relatively fine mesh (typical element size 125mm x 120mm) was used in the regions where the stress gradients were believed to be critical for elastic buckling. Triangular plate elements were generally avoided for better accuracy. Detailed plots of the finite element model are presented in Appendix A

Plate coupons were removed from a total of 6 locations on the mast webs and flanges and tensile tests were carried out in a certified laboratory. On this basis a minimum yield stress of 280 MPa and ultimate tensile stress of 475 MPa could be adopted for strength calculations and non-linear analysis.

Symmetric boundary constraints were applied to the geometric symmetry plane of the half model. Fixed boundary constraints were imposed at the bolted joints of the mast. These boundary constraints were appropriate for the analyses undertaken.

Effect of Initial Plate imperfections

The initial plate imperfection or crookedness was included as an initial displaced shape in accordance with values as recommended in the Merrison Committee report (Ref 2). Plate imperfections can in many cases significantly affect the real buckling behaviour of plates.

Effect of Residual Welding Stresses

Equivalent initial out-of-plane imperfections due to residual welding stresses were calculated in accordance with the recommendations in the Merrison report (Ref 2). Initial residual stresses can in many cases typically considerably effect both stress distribution in a linear analysis and the real buckling behaviour.

The symmetric model of the mast was analysed using MSC/NASTRAN. The finite element model statistics are as follows:

No. of Node Points	=	10 625
No. of QUAD4 Elements	=	10 473
No. of TRIA3 Elements	=	15
No. of Degrees of Freedom	=	54 647

Four analyses were carried out for the investigation and are summarised below.

Analysis 1 - Self Weight Analysis

- Linear static analysis using SOL 101 Solver due to gravity load only.

Analysis 2 - Linear Static Analysis

- Linear static analysis using SOL 101 Solver due to failure load case.

Analysis 3 - Linear Buckling Analysis

- Linear buckling analysis using SOL 105 Solver (ie, eigenvalue extraction) due to failure load case (first five modes only).

Analysis 4 - Geometric and Material Non-linear Analysis.

- Non-linear static analysis using SOL 106 Solver due to failure load case.

- To determine the load factor for plate buckling incorporating large displacement and material non linearity which in turn predicts the post-buckling behaviour of the mast.

2.3 Discussion of Results

Linear Static Analysis

Analysis results for the design load combination are shown graphically in the form of colour stress plots in Appendix A. Major and Minor Principal Stresses and Von Mises Stresses for the critical regions are presented. The maximum stresses at flange and web regions between the mast bolted joint and counterweight boom pivot at the mast are presented in Table 2.1. The maximum mast deflection for this load combination is about 153 mm.

Table 2.1 Maximum Flange and Web Stresses

Mast Component	Von Mises Stress (MPa)	Major Principal Stress (MPa)	Minor Principal Stress (MPa)
Flange (forestay side)	230	214	-233
Flange (backstay side)	290	120	-223
Web (inner side)	290	120	-223
Web (outer side)	230	214	-223

Note: - The above stresses are average values obtained from the colour contour plots depicted in Appendix A.

Linear buckling Analysis

The first five buckling modes for the design load combination were calculated by the MSC/Nastran analysis and the first three buckling modes are shown graphically in Appendix A. A summary of the eigenvalues or load factors from sensitivity analyses analysis is tabulated in Table 2.2. To examine the effect of discontinuous stiffeners, intermittent welds and diaphragm holes a further analysis was carried out on a model which did not include these features. The Lanczos method was used to determine both the eigenvalues and corresponding eigenvectors or mode shapes.

Table 2.2 Elastic Buckling Load Factors

Mode Number	Load Factor			
	No. diaphragm holes, intermittent welds, and discontinuous stiffeners	Cross Beam Reaction Ratio (Lower face: Upper face)		
		2:1	1:1	0:1
1	1.047	1.023	1.023	1.021
2	1.064	1.038	1.038	1.037
3	1.173	1.054	1.058	1.070
4	1.196	1.119	1.142	1.144
5	1.208	1.141	1.164	1.165

The results of the buckling analysis which give a minimum load factor of 1.02 show that the assumed failure loads are of the correct order to have caused the failure. The first buckling mode as shown in Appendix A is primarily in the web of the mast. Deflection in the shape of this mode correlate with the observed buckling mechanism on site.

Non-linear Analysis Results

The geometric and material non-linear analysis was carried out for the failure load case. The effects of plate imperfections and residual stresses were included as noted above. The stress-strain behaviour of the mast flange and web was obtained from test data and included in this analysis. The objective was to determine whether or not the post buckling effects the overall behaviour of the mast. The non-linear analysis is an incremental-iterative method and the summary of the first twenty load steps are shown in Table 2.3. The incremental algorithm adopted in this analysis is well known Crisfield's Arc Length Method in conjunction with the conventional Newton-Raphson Method as the iterative scheme.

Table 2.3 - Load Factors obtained from Non-linear Analysis

Load Step Number	Required Number of Iterations for Convergence	Load Factor
1	4	0.050
2	6	0.100
3	5	0.150
4	6	0.199
5	6	0.249
6	7	0.299
7	7	0.348
8	10	0.397
9	11	0.422
10	7	0.446
11	10	0.470
12	10	0.494
13	16	0.496
14	7	0.497
15	8	0.500
16	10	0.502
17	5	0.505
18	8	0.511
19	8	0.522
20	12	0.533

The deformed shape from the non-linear analysis correlates with the failure mode observed in the field as may be seen from the plots and photographs in Appendix A.

2.4 Conclusions

The non linear analysis showed a reduced load factor compared to the linear buckling analysis, demonstrating the importance of second order deformation effects on plate buckling. Hence, the non-linear buckling analysis can be considered as a lower bound.

Comprehensive verification checks were carried out to verify the model and analysis.

From this investigation it was concluded that the assumed failure mechanism was a good representation of the actual behaviour.

This investigation was effectively a full scale test which has given the authors a great deal of confidence in the power and accuracy of MSC/PATRAN and MSC/NASTRAN to model and analyse large bulk materials handling machines.

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