EFFICIENT NON-LINEAR MODELLING OF FUSELAGE PANELS

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Abstract

This paper reports on the development of a non-linear model for the buckling analysis of fuselage panels. The aim is to increase the accuracy of the global Finite Element model, accounting for variations in stiffness, as behaviour for some components becomes non-linear. The strategy is based on representing a typical fuselage stiffened panel with a single non-linear element. Each element captures both geometric and material non-linearity in its response. The non-linear data is obtained from a detailed FE model of a single stiffened panel. Larger segments of structure are represented by connecting many of these single non-linear elements together forming a framework. The response both of individual non-linear elements and the collective frameworks compare well with the detailed model results and physical tests of specimens. The non-linear models are highly efficient for analysing multiple load cases and take advantage of the repeatability of structure in a typical fuselage. The development of the elemental non-linear model is summarised here followed by a description of how larger segments of structure are represented and the comparison with detailed FE studies and experimental results.

1 Introduction

A typical aircraft fuselage is a stiffened shell construction consisting of several thousand structural elements and usually has several hundred load cases to be considered. The sheer size of this problem necessitates that simplifications are made in the design and analysis models. Even with simplifications it is a considerable task to account for non-linear behaviour such as buckling. For example, for a simple stiffened panel consisting of a single stringer length between frames with associated skin either side, the non-linear buckling analysis for one load case in compression takes several hours on a typical UNIX workstation (e.g. SGI O2, R5000 300 MHz, RAM 256 KB).

In the conventional approach to fuselage design the stress distribution throughout the entire structure is obtained from a highly idealised linear Finite Element Model. This model is a coarse mesh of beam and shell elements and is used primarily to obtain load paths and load magnitudes for use in detail design (Figure 1). This global FE model is fixed early on in the design process and typically only changes if major structural modifications are necessary.

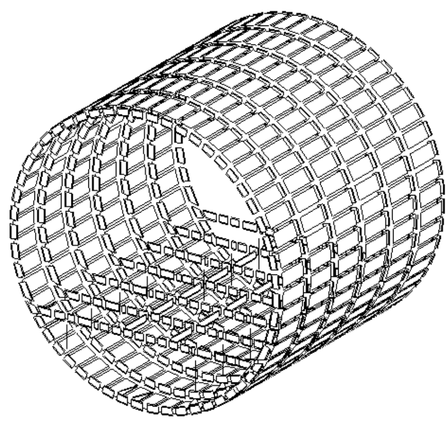


Fig. 1. Highly Idealised Fuselage Model

The loads obtained from the FE model are then used to stress individual structural members using conventional stressing methods. For buckling analysis these conventional methods [1,2] are based on empirical data and structural testing. In order to make use of the empirical design data the structure is analysed as a collection of plate and column elements with predefined edge/end conditions. The combination of empirical data and conservative boundary conditions results in conservative analysis results and potentially over designed structures [3,4].

Finite Element tools are now being developed to improve the accuracy and reliability of panel buckling and collapse analysis, with some success but the modelling and analysis times are still excessive and cannot yet replace existing approaches [5].

The strategy presented in this paper is based on representing a typical fuselage stiffened panel, Figure 2, with a single non-linear element. Each element captures both geometric and material non-linearity in its response. The non-linear data is obtained from a detailed Finite Element model of a single stiffened panel. The whole fuselage model then becomes a collection of such non-linear elements and is capable of accurately assessing the compression behaviour of the fuselage.

The procedure then is to carry out detailed FE analysis on individual stiffened panels to generate the non-linear model data and then to carry out an analysis of the new global model. Since the execution time for buckling analysis rapidly increases with model size [5,6] the analysis of all the individual panels is still less than that for the complete structure. Moreover since there is a significant degree of repeatability in the structure not every stiffened panel needs to be analysed in detail.

The following section describes the procedure for obtaining the non-linear data for a single non-linear element before describing how the global modelled is assembled and analysed. The global structure used as a benchmark for this work is a 36 by 34 inch flat fuselage panel used in a previous study [5] on the post buckling behaviour of fuselage panels using FE. This naturally provided a basis since both detailed FE and experimental results already exist for comparison with this new modelling approach.

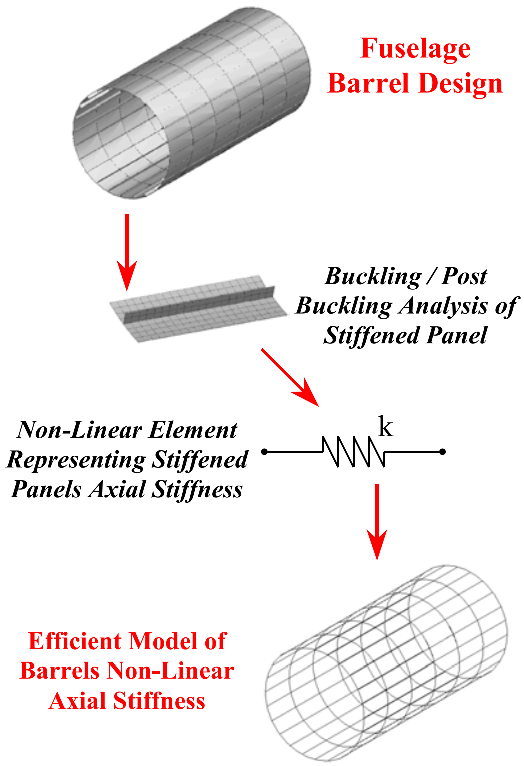


Fig. 2. Fuselage Barrel Modelling Strategy

2 Single Panel Modelling

A number of element types were considered as possible 1D non-linear entities, including beam elements with modified stress-strain data and spring elements with non-linear force displacement response properties [7]. Both element types produce identical output with the same input stiffness properties, however using spring elements requiring a minimum amount of data manipulation. Using a spring element as a 'non-linear super element' a stiffened panel's non-linear axial stiffness may be effectively modelled as a single spring. The major advantage of a non-linear spring element over conventional sub-structuring is that they are able to represent a structure's non-linear response to loading. The main concern to be addressed is then the generation of the spring input data.

A number of approaches may be used to generate a sub-structure's stiffness response including theoretical, numerical and experimental analysis methods. The technique presented within this paper focuses on the use of non-linear 3D-Shell FE models to generate the required stiffness data.

Stiffened panels are commonly model as a series of shell elements representing the plate sections and beam elements representing the stiffener sections (Figure 1 and 3a). This level of idealisation will accurately and efficiently represent the structure's linear stiffness but will fail to capture local cross-section translation and rotations and therefore fail to accurately model non-linear buckling and post buckling behaviour. In order to predict the collapse behaviour the structures cross-section must be modelled in detail.

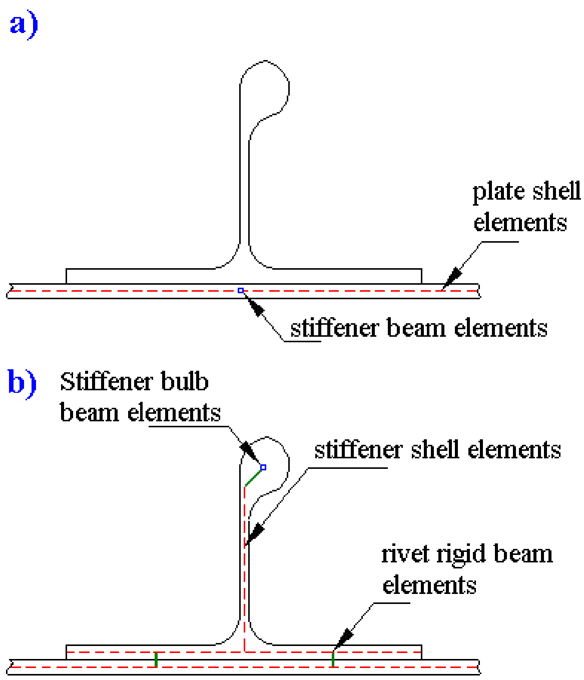
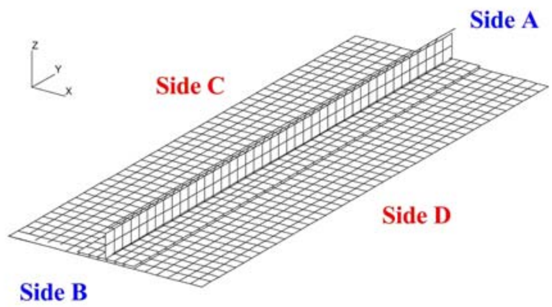


Fig. 3. Stiffener Idealisation

Lynch [5] developed a number of Finite Element modelling methodologies for the buckling and post buckling analysis of metallic fuselage panels. The idealisations proposed focus on modelling each section of the stiffener cross-sections with shell or beam elements (Figure 3b). The work investigated element selection, minimum required mesh density and the plate-stiffener interface idealisation. The work developed a computationally expensive analysis methodology capable of accurately predicting the structural response of flat fuselage stiffened panels loaded in compression.

Following the computational methodologies developed by Lynch and applying appropriate boundary conditions, which aim to match the true support/constraint conditions within the global structure, one may generate the required sub-structure stiffness data. It should be noted however that the applied sub-structure boundary conditions are based on the same simplifying assumptions followed as part of the conservative traditional analysis process. The model results will therefore tend towards under predicting the structure's local stiffness. An example of a typical sub-structure model is detailed in figure 4.



Boundary Conditions Table	
Side A	1) Negative uniform Y-Axis displacement load 2) Z-Axis translations restrained (in-plane translations)
Side B	1) Y-Axis translations restrained 2) Z-Axis translations restrained
Side C	Model material constraints 1) X-Axis translations restrained 2) Y and Z-Axis rotations restrained
Side D	Model material constraints 1) X-Axis translations restrained 2) Y and Z-Axis rotations restrained

Fig. 4. Sub-Structure Model

The individual stiffened panel models may be analysed considering full material and geometric non-linear behaviour. Figure 5 illustrates a typical axial load vs. end-shortening curve predicted by a sub-panel model.

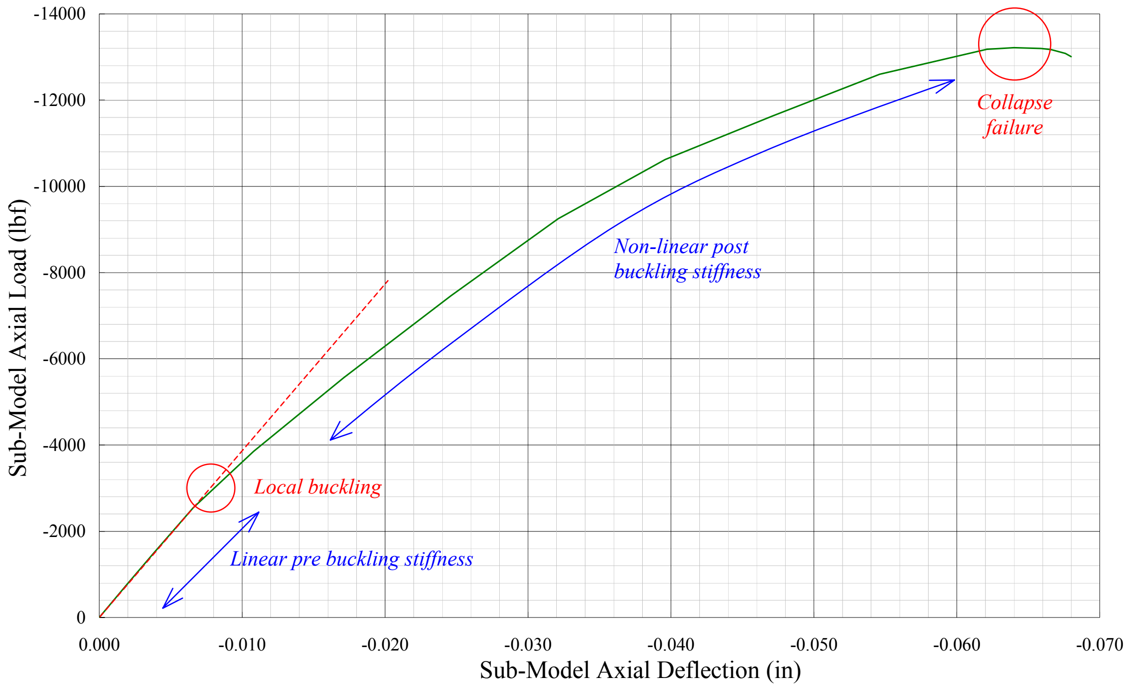


Fig. 5. Sub-Structure Results

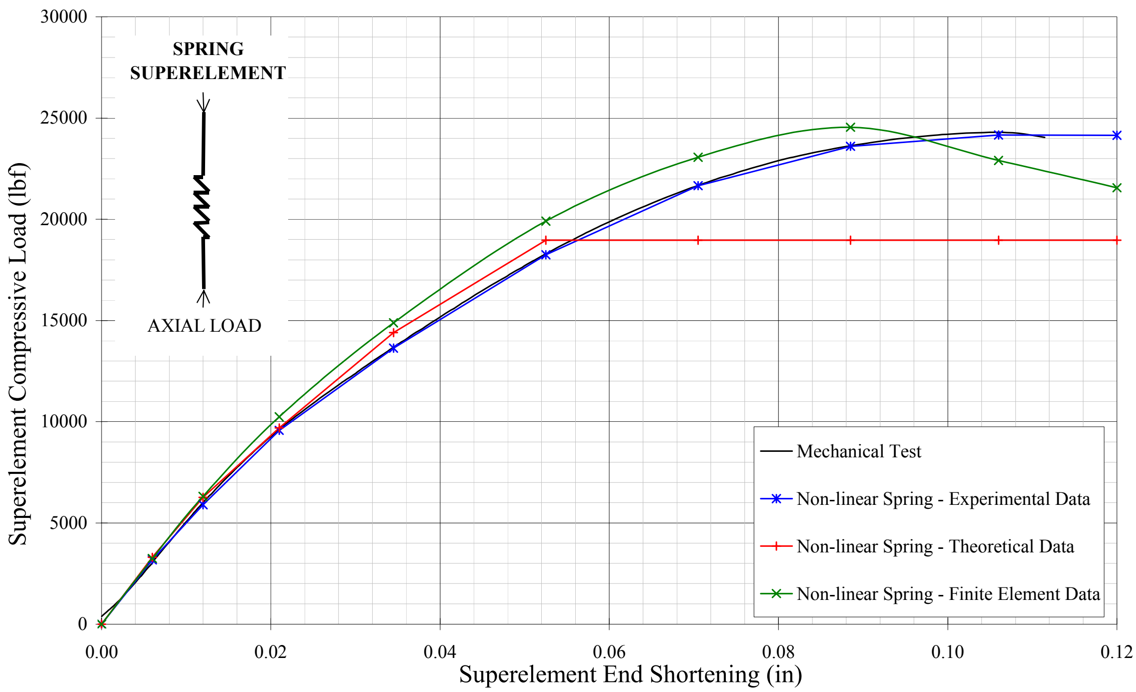


Fig. 6. Non-linear Spring Results [7]

Murphy [7] considered sub-model data generated using experimental tests and theoretical analysis as well as detailed FE models. The sub-structuring procedure detailed above was then validated against experimental sub-component tests, figure 6. The work concluded that the spring non-linear sub-structuring technique detailed above may be used to accurately model sub-component axial stiffness at a fraction of the cost of detail 3D-Shell modelling techniques. It was noted that the true value of the non-linear sub-structuring methods would only be achieved when coupling non-linear spring elements to model large stiffened panel structures.

Once the sub-model has been analysed the next stage is the discretisation of the structure's axial load vs. axial end-shortening curve into non-linear spring input data. In practice this is simply a list of load-deflection points entered as spring properties within an analysis input deck [8].

Once all the sub-components of the global structure are modelled and the spring data generated, the next stage is the assembly of the global spring model.

3 Assembling the Global Model

The aim of the global modelling scheme is simply to assemble the sub-panel spring elements in their appropriate location and account of the lateral and longitudinal interaction between springs via the frame structure. Any global model built with the sub-panel axial spring element will only be capable of modelling axial behaviour, any additional lateral stiffness detail add at the global model stage with not added to the value of the model.

Considering the 1D axial nature of the non-linear spring elements used to model the stiffened panel sections the fuselage frames are simply modelled as rigid bodies, which are free to translate axially (Figure 7). The frames are also constrained from out-of-plane translations, which is consistent with fuselage frame design philosophy [9] and the boundary conditions applied to the sub-panel models. Each sub-panel spring is attached to the fore and aft frame structure, this method of coupling the non-linear spring elements although simple, is effective as shall be demonstrated in the next section.

In order to assess the accuracy and cost of the global stiffness modelling methods the flat specimen detailed in figure 8 was analysed and stiffness results compared with detail FE analysis data and experimental test data. The specimen consists of a of a 36 by 34 inch skin, stiffened by six stringers and two frame segments.

Two specimens were tested in a 150 ton capacity Avery hydraulic compression testing machine. A half inch thick cerrobends base was cast on to specimens, producing fully clamped boundary conditions at each end. The ends were subsequently machined flat and perpendicular to the skin to ensure that uniform axial loads were applied. Frame support fixtures were designed to eliminate frame out-of-plane deflections, while allowing axial displacements. This was achieved by bolting specially built trolleys onto the protruding specimens frames. The trolleys then rolled vertically between support tracks.

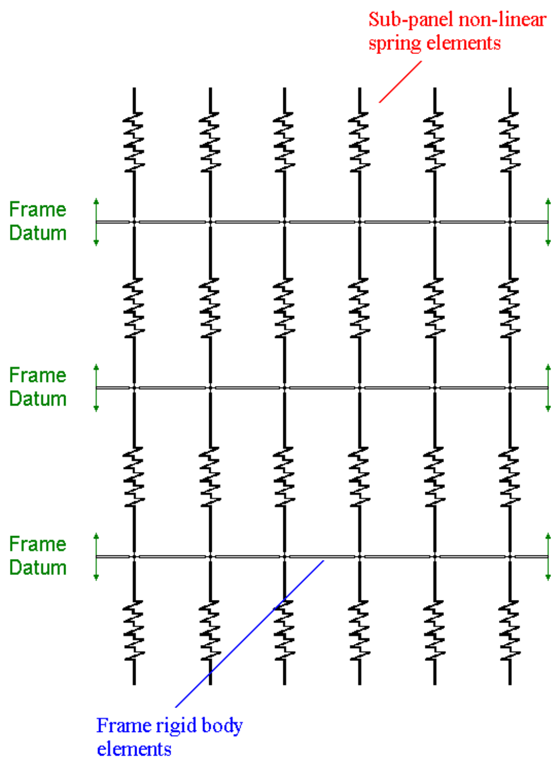


Fig. 7. Global Modelling Scheme

A full non-linear analysis of the specimen was also performed using the 3D-Shell modelling methods detailed in Section 2. The full specimen FE analysis modelled non-linear material and geometric behaviour and consisted of approximately 16,000 beam and shell elements plus 600 rigid beam elements representing the rivets.

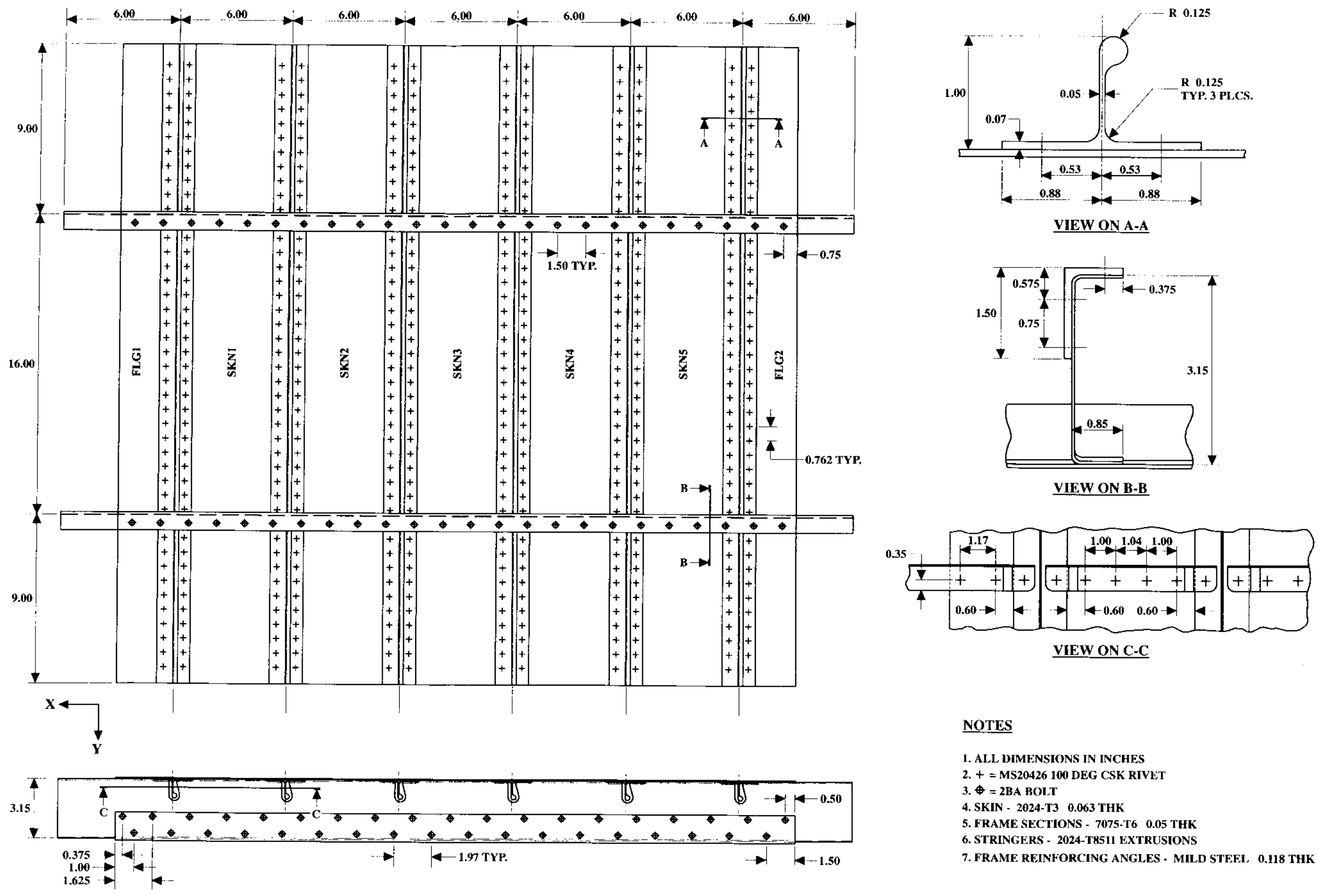


Fig. 8. Specimen [5]

The model required three man-days to build and in its completed state consisted of approximately 90,000 degrees of freedom. The loads and boundary conditions applied to the full specimen model were design to be as representative of the experimental test as possible.

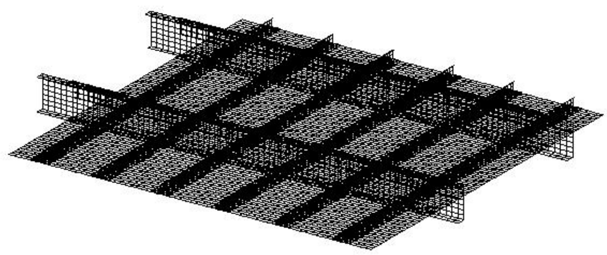


Fig. 9. Specimen FE Model

The first step in the analysis of the specimen is the division of the structure geometry into sub-sections. Each frame bay is divided into a series of sub-sections along the centre skin lines. Figure 10 details the sub-division, the analysis only requires 9 sub-section models considering structural repetition. A sub-model is required for each geometry set (top, middle and bottom sub-sections) and each geometry model must be analysed for a series of boundary conditions (left, centre, and right sub-sections). Only 50% of total specimen geometry is sub-modelled.

Once the structure has been sub-divided the sub-section models may be built and analysed following the procedures laid out in Section 2. The resulting axial loads vs. end-shortening curves for the nine sub-models are presented in figure 12.

Considering figure 12 there are three distinct groups of curves, one for the top frame bay sub-models (L1, C1 and R1), one for the middle frame bay sub-models (L2, C2 and R2), and one for the bottom frame bay sub-models (L3, C3 and R3). The sub-section cross-sectional area, length and material properties clearly define the structures pre buckling stiffness. The sub-sections post-buckling stiffness and failure loads and modes are clearly influence by the applied boundary conditions.

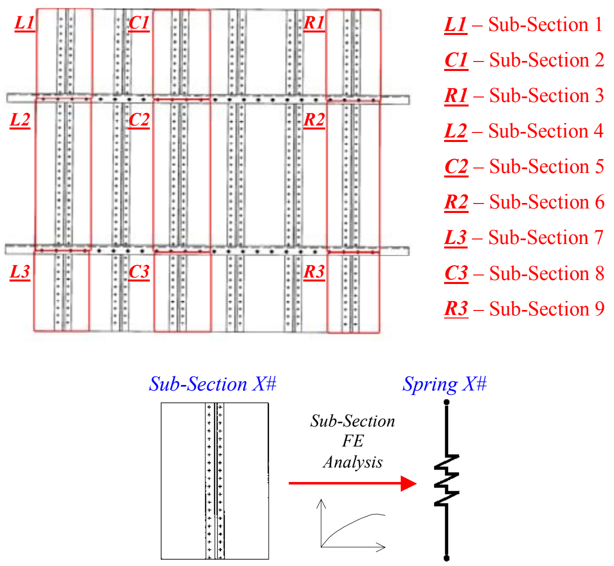


Fig. 10. Specimen Sub-division Scheme

From the basic load vs. specimen end-shortening data we may create the load response data for the non-linear springs.

4 Global Model Analysis

The final stage is then the assembly of the sub-section springs to form the global model. Figure 11 schematically illustrates the specimens global spring model. The global model in this case is loaded and constrained to replicate the specimen's mechanical test set-up, in order to benchmark global model accuracy.

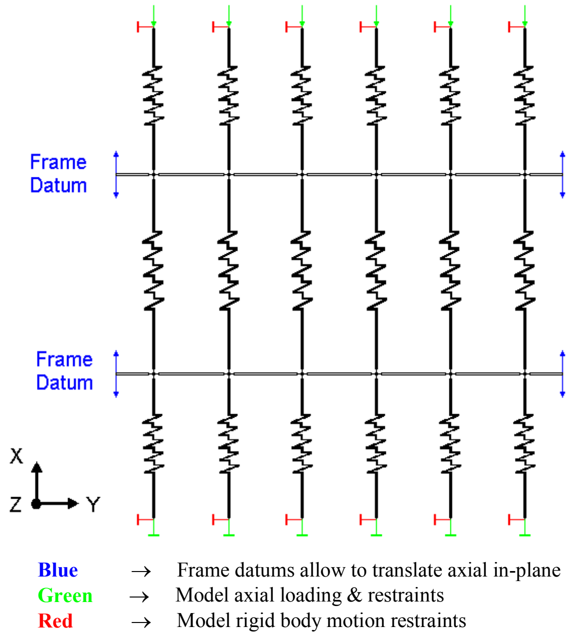


Fig. 11. Global Spring Model

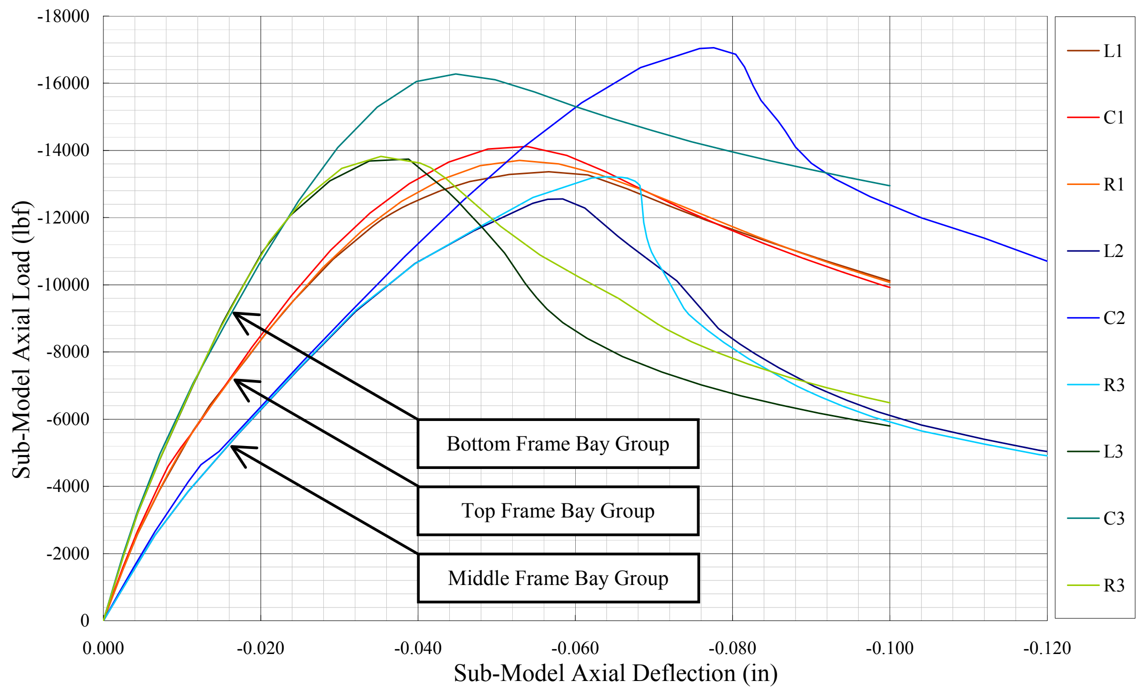


Fig. 12. Specimen Sub-Model Results

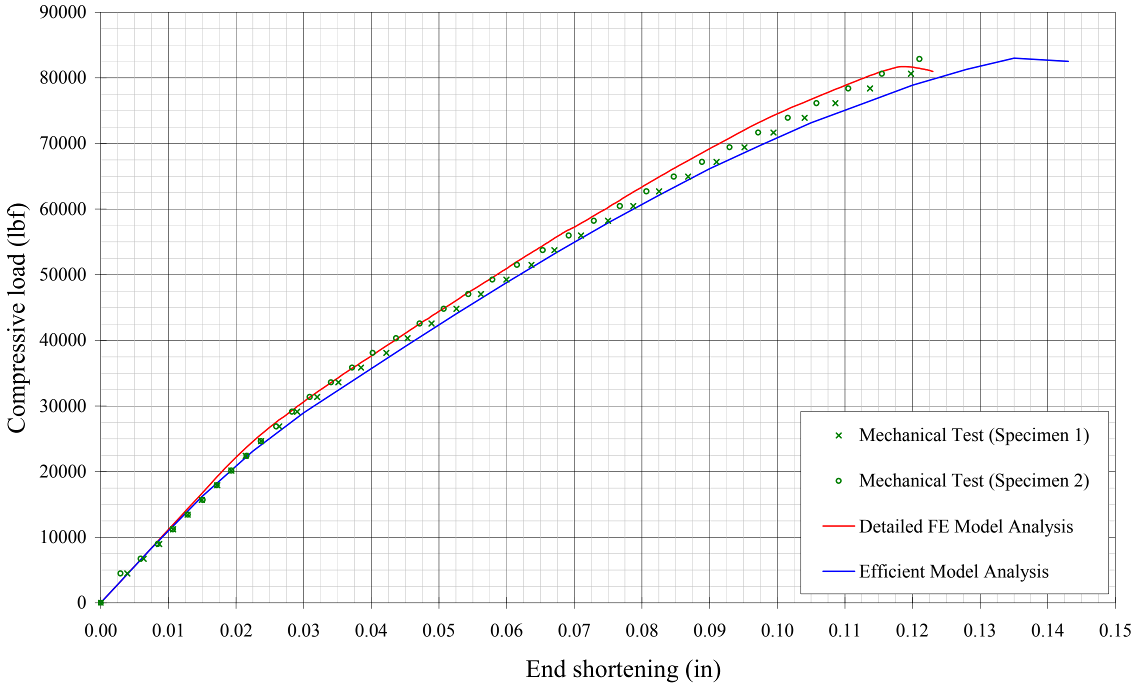


Fig. 13. Global Model Results

5 Results

The global model stiffness results are presented in Figure 13 along with experimental results and detailed FE results. Table 1 presents the computation cost of both the efficient and detailed specimen FE models.

Based on the experimental stiffness curves local specimen skin buckling occurs at approximately 25,000 lbf and specimen failure occurs at 80,640 lbf for Specimen 1 and 82,880 lbf for Specimen 2. Both specimens fail through the same mechanism of stringer local-flexural buckling, ending in a convex specimen collapse (specimen skin side curving out, stringer side curving in, figure 14). The experimental specimen's post failure stiffness data was not captured during the tests, therefore no negative slope is seen at the end of the experimental curves, figure 13.

The detailed FE model predicts local buckling at approximately 22,550 lbf, some 10% lower that the experimental value. The FE model however predicts the same stringer local-flexural failure mode as seen in the tests and predicts failure at a load of 81,741 lbf, 1.37% lower that Specimen 2 and 1.37% higher that Specimen 1. The detailed FE model stiffness output closely matches the experimental pre buckling gradient, and slightly over predicts the post buckling value. Due to the high predicted post buckling stiffness the FE model also slightly under predicts the specimen's end-shortening deflection at failure.

Fig. 14. Experimental Failure Mode

Again based solely on the load vs. end-shortening curves the efficient global model predicts the specimens local buckling load at approximately 23,750 lbf, 5% lower that the experimental value. The efficient model accurately predicts the structures pre buckling stiffness and slightly under predicts the post buckling stiffness, as expected.

The model predicts the specimens failure load as 83,038 lbf, 2.9% higher that Specimen 1. Due to the under prediction of specimen post buckling stiffness and the over prediction of specimen failure load the model also over predicts the specimens end-shortening deflection at failure.

The efficient model predicts specimen stiffness within 5% of experimental values up to 85% failure load, and within 7.5% up to 95% failure load.

Comparing the model size (Degrees Of Freedom – DOFs) for the full specimen model and a single sub-model, table 1 and the required analysis space (temporary analysis file size plus executable size) and time (wallclock time), it is clear that there is a non-linear relationship between model size and model cost when modelling non-linear buckling collapse behaviour.

Considering the size of temporary analysis files and analysis executables, the full specimen model requires an expensive UNIX server box for its analysis, whereas the sub-models and global spring model could have been analysed on a much less expensive desktop UNIX box or PC.

The total time required to built all sub-models and the global spring model was 1.5 man-days, half the time required to built the full specimen model.

Finally the total wallclock analysis time for the efficient modelling procedure was 1,124 sec, compared to 21,317 sec for the analysis of the full specimen model. This shows a possible reduction in compute time of 95%.

	Model database size (MB)	Number of model DOFs	Input file size (MB)	Temporary analysis file size (MB)	Analysis increments	Analysis iterations	Analysis executable size (MB)	Wallclock time (sec)	Output file size (Total) (MB)
Detail FE Model	10.97	90852	3.32	88.6	61	212	178.6	21317	231.4
Efficient Model (Sub-sections)									
L1	1.97	4524	0.15	4.73	43	120	18.2	89	5.33
C1	2.32	7740	0.24	4.74	36	103	18.2	73	4.60
R1	1.86	4140	0.14	4.73	38	114	18.2	87	4.80
L2	-	4524	0.15	9.25	56	162	24.9	248	11.36
C2	-	7740	0.24	9.26	35	111	24.9	136	7.75
R2	-	4140	0.14	9.25	55	199	24.9	254	11.18
L3	-	4524	0.15	4.15	40	129	17.3	83	4.58
C3	-	7740	0.24	4.16	36	120	17.4	70	4.35
R3	-	4140	0.14	4.15	26	134	17.3	56	3.23
(Spring model)	1.94	120	0.006	0.68	20	20	12.6	28	0.25
Total	8.09	49332	1.63	55.1	-	-	-	1124	57.4

All pre and post processing performed on a SGI Octane™ with a 250 MHz R10,0000 processor and 512 MB of RAM. All analysis preformed on a Sun Enterprise™ 3500, with six 400 MHz UltraSPARC II™ processors and 6,144 MB of RAM.

Table. 1. Computational Analysis Cost Parameters

6 Conclusion

The aim of the work presented is the development of efficient modelling techniques, which may be utilised to improve global model accuracy by accounting for sub-component non-linear behaviour. A simple non-linear sub-structuring strategy has been presented and results show great potential in increasing global model accuracy.

The benchmark global model results match experimental stiffness result extremely closely, with the efficient model predicting specimen stiffness within 5% of experimental values up to 85% failure load, and within 7.5% up to 95% failure load. Considering traditional global fuselage barrel models only account for linear behaviour this is a potential step forward for the accuracy of these models.

The simple non-linear spring modelling strategy may also be used to reduce the required effort to built and run a complex 3D-Shell model of a stiffened panel structure. The modelling strategy reduces analysis run times but also reduces analysis job sizes therefore reducing the computational power required and consequently reduces the cost of required computational hardware.

Considering the potential of the modelling strategies introduced there are still a number of issues requiring investigation. The first is the development of additional strategies to accurately and efficiently model the non-linear behaviour of the structure when loaded in shear, bending or combinations of compression, shear and bending.

Secondly considering the cost saving with respect to large detailed non-linear FE models, can the developed modelling methods be used along with mixed element coupling procedures [10, 11] to reduce the cost of large structure FE modelling, with the use of developed sub-structuring [12] or sub-modelling [12] techniques.

References

[1] Bruhn E F. *Analysis and Design of Flight Vehicle Structures*. 1st edition, Tri-State Offset Company, 1973.

[2] ENGINEERING SCIENCES DATA UNITS. *Structures Sub-Series*. ©ESDU International Ltd.

[3] Lynch C, and Sterling S G. A Finite Element Study of the Postbuckling Behaviour of a Flat Stiffened Panel. *21st ICAS Congress*, Melbourne Australia, ICAS-98-7,7,3, 1998.

[4] Domb M M, Elliott W G and Leigh B R. Modelling of stiffener Crippling Phenomena Using Finite Element Analysis. *Canadian Aeronautics and Space Journal*, Vol. 44, No. 4, pp 256-262, 1998.

[5] Lynch C J. *A Finite Element Study of the Postbuckling Behaviour of a Typical Aircraft Fuselage Panel*. PhD Thesis, Queen's University Belfast, Belfast, N. Ireland, 2000.

[6] Geiro A, Grundy P, Cannon S and Nguyen V. Advanced Finite Element Techniques for Hull Structural Analysis. *8th International Offshore and Polar Engineering Conference*, Montreal Canada, Vol. IV pp 320-325, 1998.

[7] Murphy A and Price M. Efficient Finite Element Modelling of Buckling and Post-Buckling Behaviour of Flat Stiffened Panels. *NAFEMS World Congress 2001*, Lake Como Italy, 2001.

[8] Anonymous. *ABAQUS / Standard User's Manual*. Version 6.1, Hibbitt, Karlsson and Sorenson, 2000.

[9] Niu M C. *Airframe Structural Design*. 1st edition, Hong Kong Conmilit Press Ltd, 1988.

[10] Price M, Murphy A and Armstrong C G. The use of Sub-Model Idealisations for Hierarchical Optimisation in Aircraft Structures. *2nd Altair HyperWorks UK Conference*, Towcester England, 2000.

[11] McCune R W, Armstrong C G, Robinson D J. Mixed Dimensional Coupling in Finite Element Models. *International Journal of Numerical Methods in Engineering*, Vol. 49, pp 725-750, 2000.

[12] Cook R D, Malkus D S and Plesha M E. *Concepts and Applications of Finite Element Analysis*. 3rd edition, Wiley & Sons, 1989.

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