Flat Panel Post-Buckling Analysis with Implicit Method using OptiStruct

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Introduction

Many commercial aircraft are designed so that fuselage skins can elastically buckle below limit load and continue to operate safely and efficiently (Fig. 1). This design regime makes for a very lightweight semi-monocoque structure compared to a non-buckling design. Therefore, predicting the local buckling, post-buckling behavior, and failures are critical to design and optimization of this kind of structure. The local panels buckle in a combination of compression and shear. Excess compression is redistributed to surrounding axial members (frames and stringers) and shear is continued to be carried by the buckled panels via tension parallel to the buckle waves. The compression redistribution and diagonal tension put special strength considerations on all involved structural components. This post-buckling behavior and the analysis method are both called intermediate diagonal tension (IDT).

Figure 1: Fuselage skin buckles below limit load but the airplane can still operate safely[1]

Challenges

Analysis of post-buckling behavior is currently carried out by OEMs using proprietary methods. The OEM methods usually take linear, static coarse mesh FEM results and post-process them with hand/spreadsheet calculation to estimate a post-buckled load distribution. This re-distribution is a semi-empirical method that has been in use for decades. These proprietary methods have been proven out by physical tests and have high reliability due to many years of in-service reliability and safety. However, these methods are conservative and most likely result in a heavier than necessary structure. Weight is one of the most import factors when designing aircraft and a small reduction in weight can bring some noticeable improvements in performance and operating economy.

Thus, an improved method for conducting IDT-based fuselage design analysis could result in lighter aircraft and quicker design time.

This paper is the first in a series toward Altair developing an advanced, correlated IDT method for design and optimization. Here Altair will explore using a large displacement solution to predict buckling and post-buckling behavior. With the increasing improvements in FE code efficiency, CPU speeds and core memory availability, it may be possible to run a post-buckled fuselage analysis or even optimization fully within the FEA realm rather than employing empirical post-processing methods.

Problem Formulation

To figure out an efficient and accurate way of predicting post-buckling structural behavior and failure, this paper investigates the implicit simulation method using OptiStruct, and tries to construct a set of best practices. This is the foundation of our potential numerical optimization of IDT structures in the future.

Experiments and Results

In this study, the buckling and post-buckling behavior of a flat panel were investigated. Three different quasi-static conditions were considered: compression with hinged boundary, shear with hinged boundary, and shear with attached frame. For each condition, the influence of imperfection, element size, and damping were studied. For the panel with attached frame model, the effect of contact was examined as well. Non-linear quasi-static large displacement analysis of OptiStruct 2018 was used for the simulation so that the geometric nonlinearity can be properly captured. To focus on the buckling and post-buckling behaviors, the material was assumed to be Aluminum with simple linear elastic properties (E=1.015e7psi, nu=0.33).

For the computing resources, all simulations were run with 1 core of Intel® i7-6820HQ @2.7GHz, with Window 7 operating system and in-core solution.

Flat panel compression model

A 6''x8''x0.04'' (152.4x203.2x1.016mm) aluminum flat panel was compressed on the shorter edges (Fig. 2). All edges were hinged, and load was applied as enforced displacement. The theoretical buckling stress is 1624ksi and the theoretical buckling load is 389.8lbf.[2]

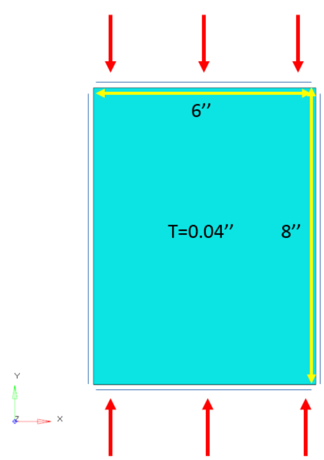

Figure 2: Flat panel compression model dimension and loading

In the simulation, the instant of buckling was identified by the out-of-plane displacement contour (Fig. 3) and the nonlinearity in the force-displacement curve. Because a linear material law was used, and there was no contact in this simple case, the only source of non-linear stiffness was buckling and the geometric nonlinearity associated with it. To simulate the geometric nonlinearity, large displacement analysis must be used. The effects of imperfection, element size, and numerical damping were studied with this model.

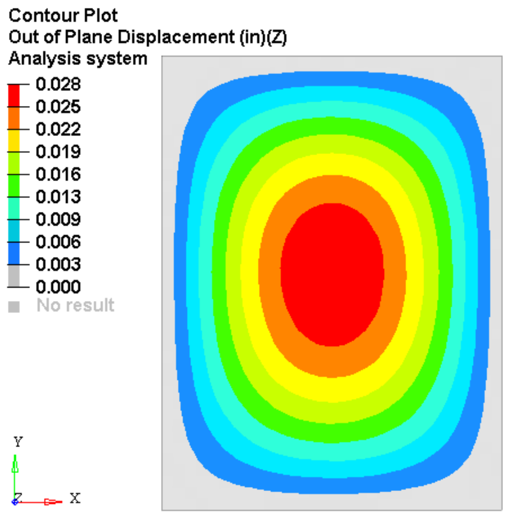

Figure 3: Out of plane displacement (in) shows that flat panel buckles under compression

Imperfection

The buckling is initiated by the imperfection either in geometry, material, or loading. A perfectly flat panel with uniform material and thickness and under uniform compression load will not buckle, because there is no imperfection to initiate buckling. Hence, it is important to study the influence of imperfection and figure out the suitable amount of imperfection that can lead to proper buckling behavior. In this research, imperfection was introduced by translating flat panel nodes randomly in both direction and magnitude. The maximum translating distances are 0%, 5%, 10%, and 15% of the panel thickness. For instance, maximum 5% thickness means all nodes on the panel were translated randomly by the magnitude of 0-5% of the panel thickness, and in random direction. To be consistent, the element size was kept as 0.2'' for the four models.

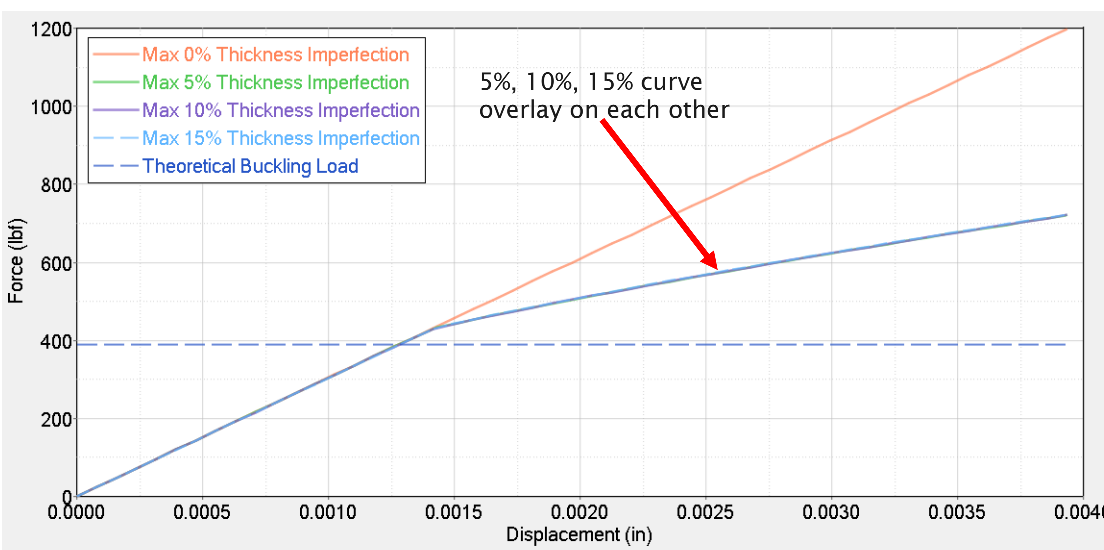

Figure 4: Force-deflection curve of flat panel compression with different imperfection levels

Figure 4 shows that the imperfection is necessary for simulating flat panel buckling because the panel without imperfection didn't buckle. Models with all three levels of imperfections buckled at the same 427.6lbf load, which is 10%

higher than the theoretical value. Random imperfection up to 15% of panel thickness didn't affect the global stiffness. Simulations with a 5% maximum imperfection percentage had non-linear solver convergence issues, but higher imperfection percentages did not.

Element size

Element size is another key factor that could affect buckling, because the structure can be artificially harder to buckle if there are not enough elements to form the initial buckling shape. Based on the previous observation of the imperfection's effects, a maximum 10% panel thickness imperfection was applied to all models for this study. Table 1 summarizes the element sizes and corresponding number of elements per initial buckling wavelength that were investigated. The initial buckling wave shape is related to the structure geometry and loading. In this case, the initial buckling wavelength is equal to the longer panel edge length.

Table 1: element sizes/number of elements per initial buckling wavelength for flat panel compression

Study	Element Size (in)	# element per initial buckling wavelength
1	0.1	80
2	0.2	40
3	0.4	20
4	0.8	10
5	1.6	5

Figure 5 shows the force-deflection curves of models with different element sizes. The buckling load and post-buckling stiffness became insensitive to element size for 0.4'' or smaller elements, which corresponds to 20 elements per initial buckling wavelength. 0.1'', 0.2'', and 0.4'' models buckled at 427.6lbf, which is 10% higher than the theoretical value. The models with 0.8'' elements and 1.6'' elements could not converge at the full load.

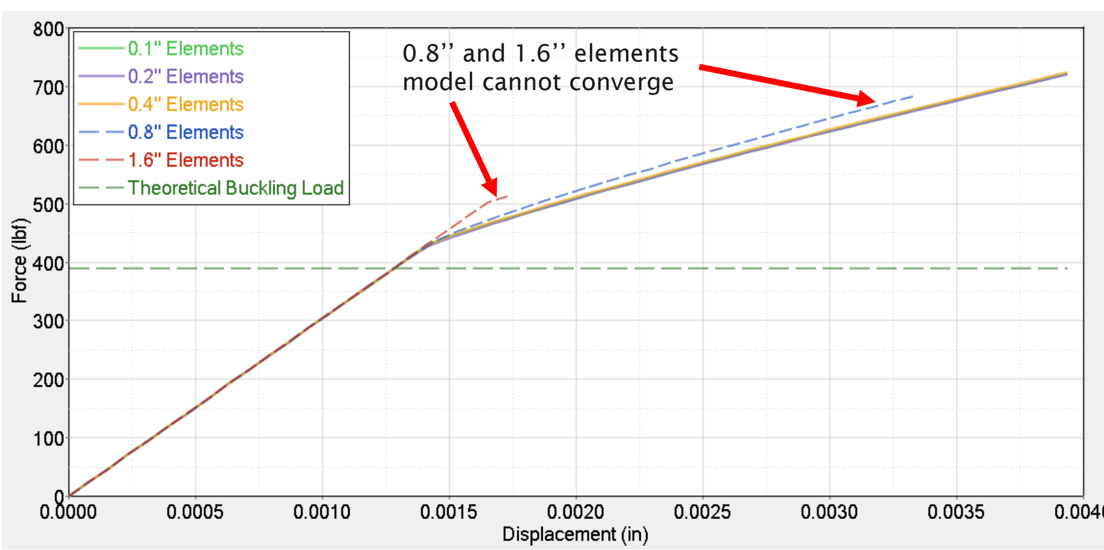

Figure 5: Force deflection curve of flat panel compression with different element sizes

Damping

Numerical damping can help the implicit solver converge when instability such as buckling happens. On the other hand, the damping is artificial, and it could affect the simulation result if too much damping is introduced. In this research, constant damping was applied because it is recommended for buckling stabilization. Since the flat panel compression model didn't have convergence difficulty up to 0.04'' displacement even without damping, damping factor of 0.01, 0.1, and 1 were studied. To be consistent between models, all models in the section had 0.4'' element size and 10% imperfection.

As shown in Fig. 6, stabilization damping factor up to 0.1 didn't affect the global structural behavior too much. The stabilization energy ratio is defined as the ratio between the stabilization energy and the strain energy. The stabilization energy is artificial, but the strain energy is physical, so a smaller value means less artificial error introduced. Figure 7 explains that the stabilization energy percentage was 0.0018% for a damping factor of 0.1, and it was 0.54% for a damping factor of 1.

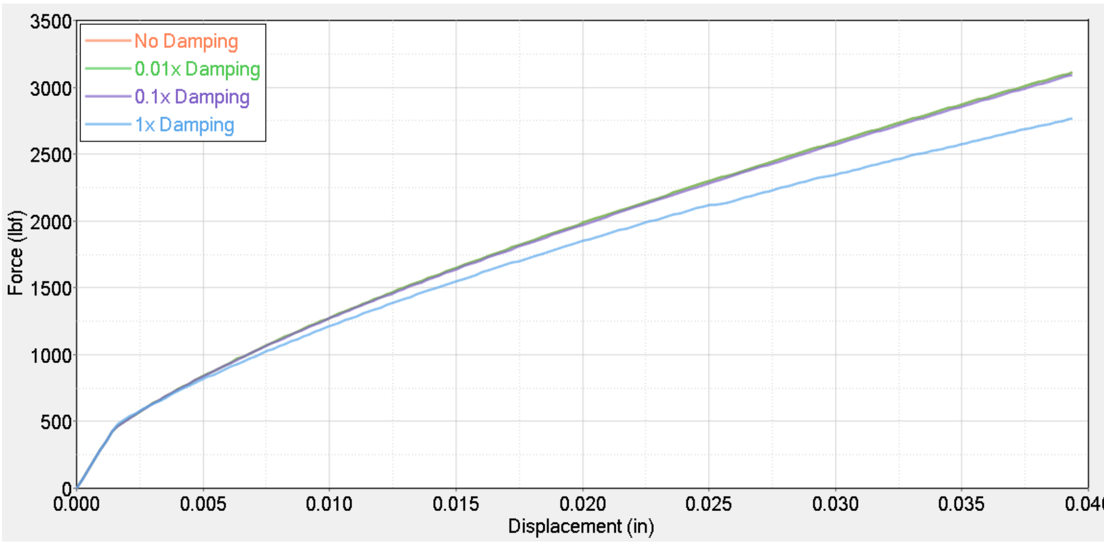

Figure 6: Force deflection curve of flat panel compression with damping

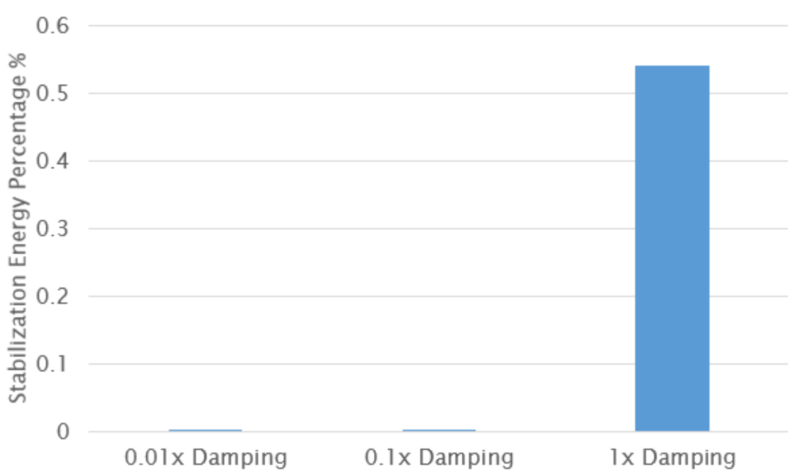

Figure 7: The ratio between stabilization energy and strain energy for flat panel compression

Comparing to linear buckling solution

Linear buckling simulation is an eigen value solution. It is very fast to solve but it can only predict buckling load and buckling mode. The result and computation time of linear buckling analysis are compared with those of non-linear quasi-static solution in Table 2. The computation time of non-linear quasi-static solution is directly related to the load magnitude. The CPU time of the model with 0.04'' enforced displacement, which is sufficient to determine the buckling load, is referenced below.

Table 2: flat panel compression result and computation time comparison between methods

	Theoretical calculation	Linear buckling	Non-linear quasi-static with large displacement
Buckling Force	389.8lbf	432.5lbf	427.6lbf
CPU time	N/A	1s	9s

For flat panel under compression, maximum 10% panel thickness imperfection and at least 20 elements per initial buckling wavelength were recommended. The force-deflection curves were not affected much by damping if the stabilization factor was 0.1 or less. In addition, the linear buckling solution and the non-linear large displacement solution predicted very close buckling loads, and they were about 10% higher than the theoretical calculation value, which was acceptable.

Flat panel shear model

In this model, a shear load was applied to all edges of a 6''x8''x0.04'' (152.4x203.2x1.016mm) aluminum flat panel (Fig. 8). All edges were hinged, and the load was applied as nodal force. The theoretical buckling stress is 3022psi and the theoretical buckling load is 725.3lbf along the shorter edge, which corresponds to 120.1lbf/in in terms of shear flow.

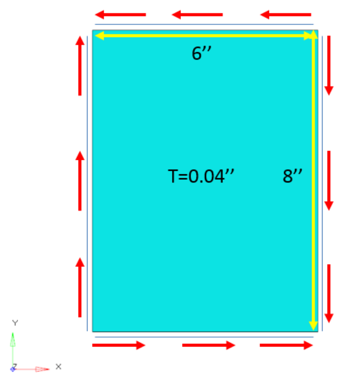

Figure 8: Flat panel shear model dimension and loading

Imperfection

Similar to the previous study, imperfection was introduced by translating flat panel nodes randomly in both direction and magnitude. The maximum translating distances were 0%, 5%, 10%, and 15% of the panel thickness. The element size was kept as 0.2'' for the four models.

As shown in Fig. 9, the flat panel did not buckle under shear when no imperfection was present. The model with a maximum 5% panel thickness imperfection predicted a buckling load of 870.5lbf, while the panel with 10% and 15% imperfection buckled at 725.5lbf, which is within 1% of theoretical value. Random imperfection up to 15% of panel thickness didn't affect the global stiffness.

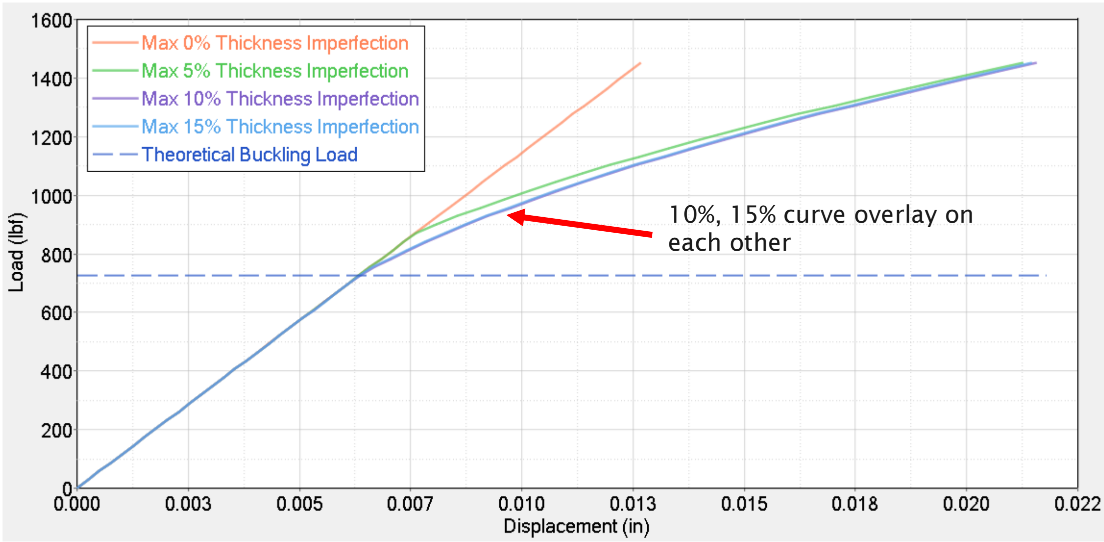

Figure 9: Force-deflection curve of flat panel shear with different imperfection levels

Element size

In this section, element sizes of 0.2'', 0.4'', and 0.8'' were investigated, as explained in Table 3. Based on the observation of the influence of imperfection, a maximum 10% panel thickness imperfection was applied to all models in this study.

Table 3: element sizes/number of elements per initial buckling wavelength for flat panel shear

Study	Element Size (in)	# element per initial buckling wavelength
1	0.2	32
2	0.4	16
3	0.8	8

Figure 10 demonstrates the force-deflection curves of models with different element sizes. At least 16 elements per initial buckling wavelength (0.4'' element size) were necessary to predict accurate buckling load. The model with 0.2'' element size predicted the same buckling load but slightly lower post-buckling stiffness. Both 0.2'' and 0.4'' models buckled at 725.5lbf, which is within 1% of theoretical value. The model with 0.8'' elements could not converge at the full load.

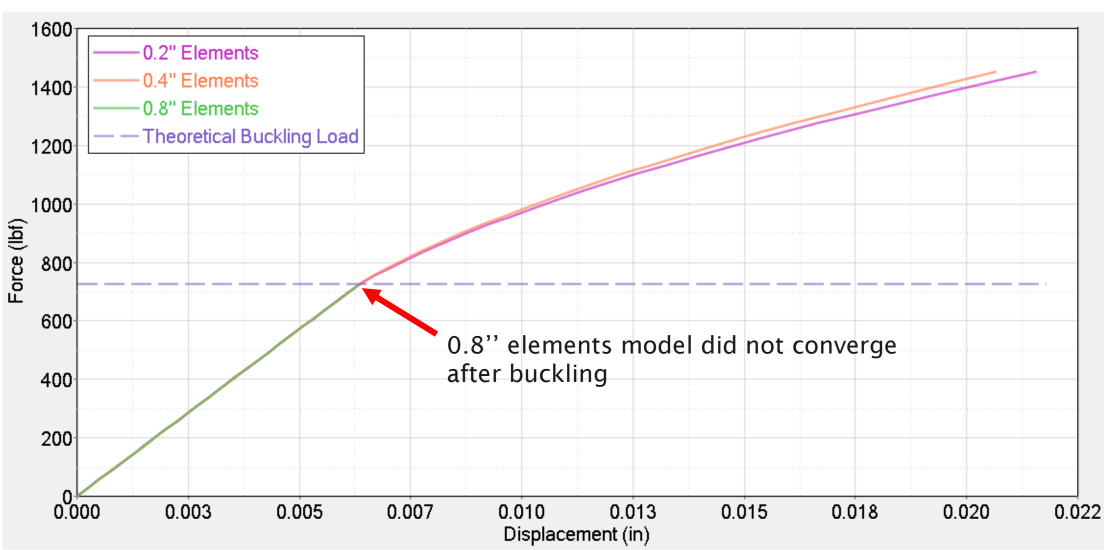

Figure 10: Force-deflection curve of flat panel shear with different element sizes

Damping

As shown in Fig. 11, the flat panel under shear model failed to converge at 2698lbf without damping, which indicates that this model had higher instability and was more difficult to converge. Therefore, damping factors of 0.1, 1, and 10 were studied. Figure 11 shows that a stabilization damping factor up to 10 didn't affect the global stiffness too much. The stabilization energy percentage was 0.51% for damping factor of 1, and it was 1.1% for damping factor of 10 (Fig. 12).

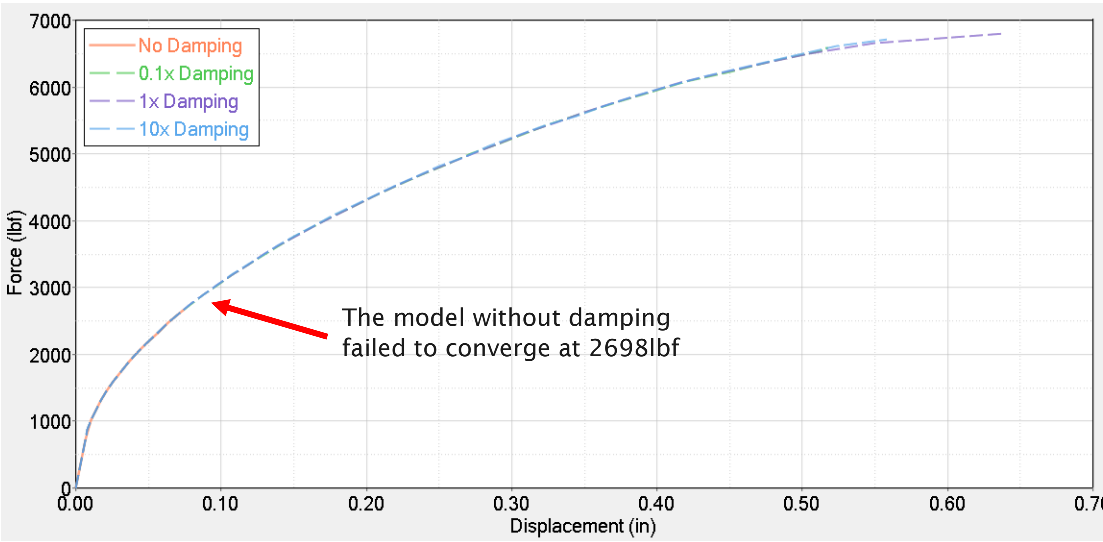

Figure 11: Force deflection curve of flat panel shear with stabilization damping factors

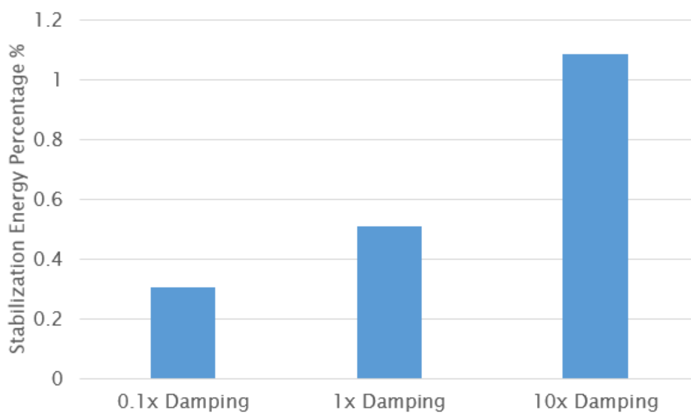

Figure 12: The ratio between stabilization energy and strain energy for flat panel shear

Comparing to linear buckling solution

The result and computation time of linear buckling analysis for flat panel shear are shown in Table 4 for reference. In the non-linear quasi-static simulation, the shear flow was applied up to 241.5lbf/in, which is sufficient to observe buckling.

Table 4: flat panel shear result and computation time comparison between methods

	Theoretical calculation	Linear buckling	Non-linear quasi-static with large displacement
Buckling Force	725.3lbf	748.8lbf	725.5lbf
CPU time	N/A	2s	8s

For flat panel under shear, maximum 10% panel thickness imperfection and at least 16 elements per initial buckling wave were recommended. In addition, the stabilization factor of 1 or less did not affect the results much, but helped the model converge at higher load. Moreover, the non-linear quasi-static solution predicted a buckling load that was very close to the theoretical value and the linear buckling simulation result, indicating high result accuracy.

Flat panel shear with attached frame model

A 9.97''x7.97''x0.04'' (253.2x202.4x1.016mm) aluminum flat panel was connected to a frame by rivets. The frame had an "I" cross-section with 1.98'' (50.4mm) flange width 2.16'' (55.0mm) web height. The rivets distributed along both the inner side and the outer side of the flanges, and the rivets distribution are indicated as pink dots in Fig. 13. Shear force was applied to all edges of the flat panel, and hinged boundary condition was applied along its outermost perimeter. The frame imposed a boundary around the region that does not contact with the frame, which is defined as the effective buckling region. The dimension of the effective buckling region is 4''x6'' (101.6x152.4mm), as shown in Fig. 13. If hinged boundary condition is assumed to be applied to the effective buckling region, by applying the same flat panel under shear formula[2], the theoretical buckling stress is 7005psi. This corresponds to 280.2lbf/in in shear flow and the theoretical buckling load is 2233lbf along the shorter edge. Hinged assumption could be good if not taking the contact into consideration, but the contact between the frame and panel could make the boundary a mixture of hinged and clamped condition.

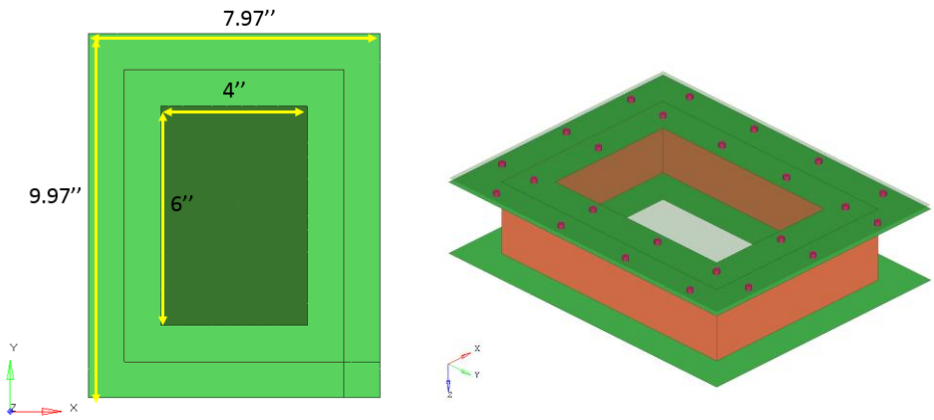

Figure 13: The dimension of flat panel with attached frame

Imperfection

Similar to the previous studies, imperfection was introduced by translating flat panel nodes randomly in both direction and magnitude. The maximum translating distances are 0%, 5%, and 10% of the panel thickness. To be consistent, the element size was kept as 0.2'' for the three models, and no contact was applied.

A line corresponding to linear behavior was added in Fig. 14 to help identify when the force-deflection curves bent and buckling initiated. As shown in the plot, the force-deflection curves were very close to each other with different imperfection levels. The flat panel with frame did not need artificial imperfection to buckle because the rivet connections between the frame and flat panel introduced imperfect boundaries and initiated the buckling. The buckling load predicted by the model is 2244lbf, which is 5% higher than the theoretical value.

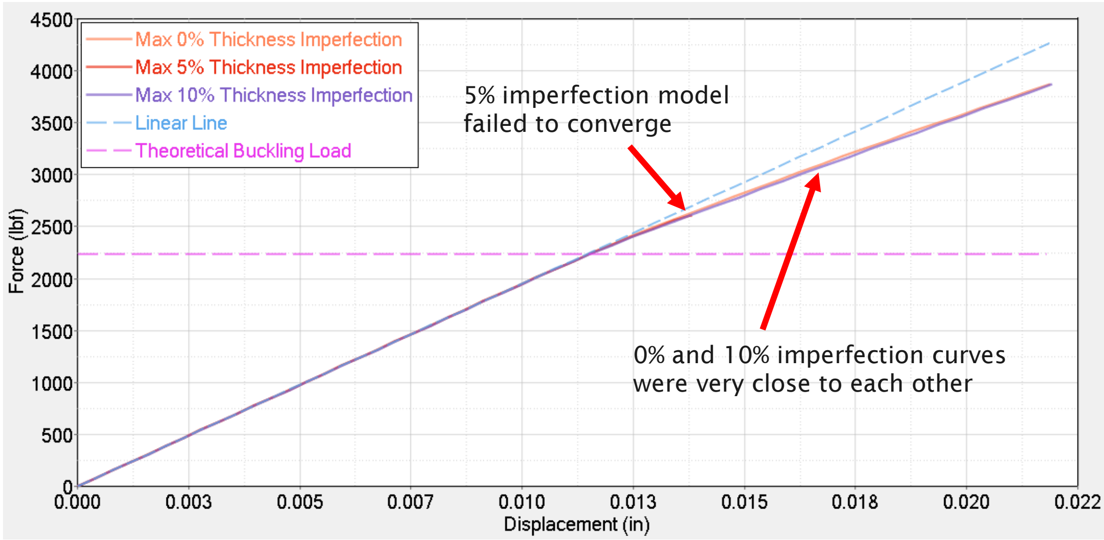

Figure 14: Force-deflection curve of flat panel shear with frame with different imperfection levels

Contact

The purpose of this study was to check the influence of simulating the contact between the flat panel and the frame. Three models were investigated: no contact, node to surface contact, and surface to surface contact. For all three models, 0.2'' element size was used and no imperfection was applied.

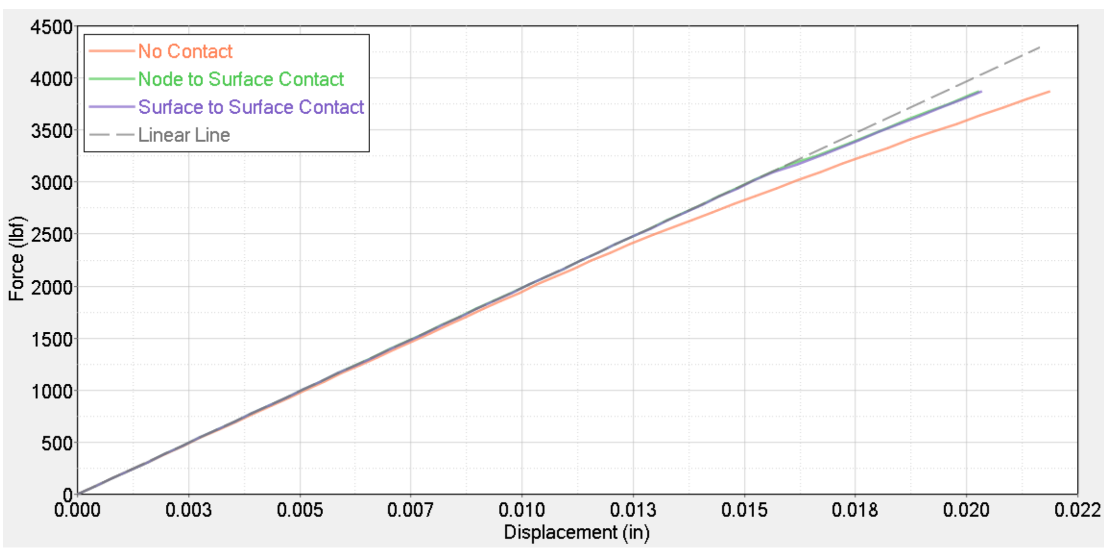

Figure 15: Force-deflection curve of flat panel shear with frame with different contact boundaries

A linear line was added in Fig. 15 to help identify buckling load. The plot shows that the buckling load and post-buckling stiffness were very close between the node-to-surface contact model and the surface-to-surface contact model. On the other hand, the models with contact had higher buckling load and post-buckling stiffness compared to the no-contact model. The panel with contact buckled at 3147lbf because the contact made the boundary of the effective buckling region something between hinged and clamped situation.

As for the solving speed, with the same hardware configuration, the node-to-surface contact model took 10 minutes to solve while the surface-to-surface contact model took 16 minutes to solve.

Damping

In this study, all models have 0.2'' element size and no contact. The model failed to converge at 21357lbf without damping and the load increments were small because of the convergence difficulty. Therefore, damping factors of 1, 10, and 100 were studied to help the model converge while monitoring the accuracy. As shown in Fig. 16 and Fig. 17, stabilization damping factor up to 10 didn't affect the global structural behavior too much, and the stabilization energy percentage was 0.38% in this case. Figure 16 also shows that the model with a damping factor of 1 led to higher stabilization energy percentage than a damping factor of 10. This happened because the damping energy was not only dependent on the damping force but also the moving path under the force. The model struggled but still could not converge at full load with a damping factor of 1.

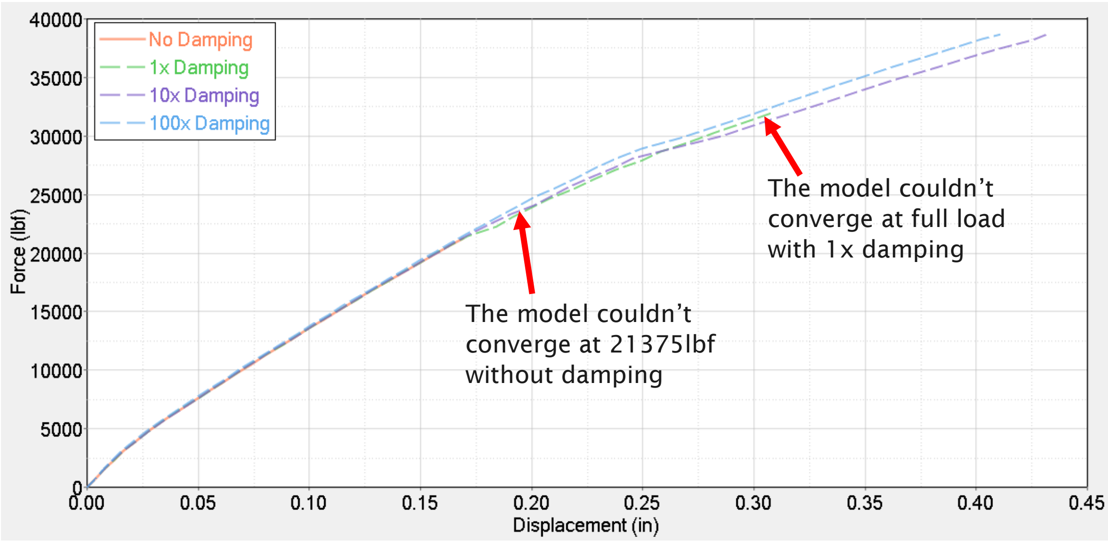

Figure 16: Force-deflection curve of flat panel shear with frame with different damping factors

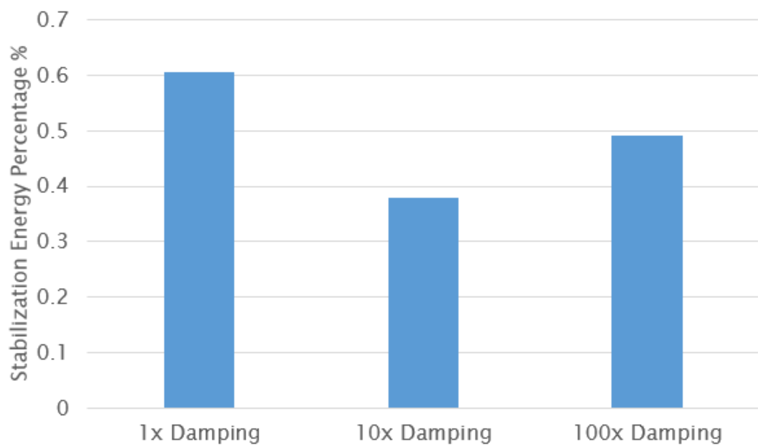

Figure 17: The ratio between stabilization energy and strain energy for flat panel shear with frame

Comparing to linear buckling solution

For a flat panel with frame under shear, the result and computation time of linear buckling analysis are compared with those of non-linear quasi static (no contact) solution in Table 5. In the non-linear quasi-static simulation, the shear flow was applied up to 483.6lbf/in, which is sufficient to observe buckling.

Table 5: flat panel shear with frame result and computation time comparison between methods

	Theoretical calculation	Linear buckling	Non-linear quasi-static with large displacement
Buckling Force	2233lbf	2240lbf	2244lbf
CPU time	N/A	4s	147s

For a flat panel under shear with attached frame, artificial imperfection was not necessary for the structure to buckle. On the other hand, contact had a significant influence on the buckling load and post-buckling behaviors. Because this model was more complicated, higher stabilization factor was required for the model to converge at the full load. Moreover, non-linear quasi-static solution predicted a buckling load that agreed with the theoretical calculation and the linear buckling simulation result well, which validated the result accuracy.

Conclusion

In this study, the influence of several modeling factors on buckling and post-buckling behavior were investigated using OptiStruct. The studied factors include imperfection, element size, numerical damping, and contact, and the best practices were summarized in Table 6.

Table 6: Best Practices for Modeling Post-Buckling Behavior

Factors	Best Practice
Imperfection	Necessary for simple flat geometry, but not required for complex structures
Element size	No less than 16-20 elements per initial buckling wavelength
Stabilization	Beneficial for convergence at high load, keep damping energy ratio less than 0.4-1%
Contact	Noticeable influence on the results, and should be included when possible

The imperfection is critical to initiate buckling, so it has significant influence on flat and simple geometries. For more complex structures, the geometry itself acts as irregularity or imperfection, and therefore, there is no need for introducing artificial imperfections.

At least 16-20 elements are necessary to accurately represent the initial buckling wave, and hence to predict accurate buckling load. Coarser elements will predict artificially higher buckling load and higher post-buckling stiffness. Because this requirement is relative to the initial buckling wavelength, the exact element size requirement may vary from case to case depending on the geometry and loading.

Numerical stabilization is not required for OptiStruct quasi-static solution to simulate initial buckling, but it stabilizes the solution after the structure buckles and helps the simulation converge at higher loads. Nevertheless, stabilization will introduce artificial damping into the simulation and cause numerical error, so it is important to monitor and evaluate the influence of the damping. The ratio between damping energy and strain energy is a measure of the influence of damping, but it is hard to draw a clear threshold of the acceptable damping energy ratio based on the observations in this study. It seems that limit varies from problem to problem, but it is preferable to keep it below 0.4-1.0%.

Contact between adjacent structures could play a big role in buckling and post-buckling behavior. In addition to connectors, contact imposes a stronger interaction between parts, and hence increases the buckling load and post-buckling stiffness. Contact should be included whenever it is physical. In the case that was studied, model with node-to-surface contact is faster to solve than the one with surface-to-surface contact.

In the next white paper, all the best practices will be applied to an IDT beam model with shear load, and the results from simulation will be compared with test results.

References

1. http://robedgcumbe.com/ripples-in-the-skin/

2. Airframe Stress Analysis and Sizing, Michael C. Y. NIU. Example 1, p.462