

Research on Aluminum Structures for Crash Energy Absorption

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ABSTRACT

This research describes aluminum crash boxes for the purpose of absorbing crash energy. In this research, the load conditions of the boxes were calculated and the mechanisms of energy absorption were analyzed by experiment. This analysis achieved a crash box with a highly effective section shape. The research also achieved the additional goal of increasing the accuracy of crash simulation for evaluation of energy absorption by this crash box.

1. Introduction

The requirement for a vehicle with a lightweight body to absorb greater crash energy has become increasingly stringent. The use of extruded aluminum for the body frame structure offers one way of achieving this. Extrusion processing enables a variety of cross section shapes, thereby optimizing the advantages of aluminum as a material capable of absorbing crash energy.

A hexagonal beam with ribs which are located diagonally on a cross-section has been employed as an application example of an aluminum crash box⁽¹⁾(Fig. 1). There have been numerous research case studies in the past on determining the crash box cross section. Some research has looked at thin-walled structures, with a focus on the ratio of the thickness to width⁽²⁾, and recently, simulations have been frequently employed. However, there have been few studies that attempted to take advantage of the characteristics of section shape when extruded aluminum was used.

This research focused on the characteristics of the cross section shape of ribbed aluminum beams by experimental investigation and the establishment of simulation technology for realizing both greater energy absorption and weight reduction. Described for the first time in this report is a method for determining the load in the axial direction of the beam and the bending moment in order to evaluate the strength performance required of the crash box. Experiments are then conducted on components of different cross-sectional shape to evaluate the energy absorption

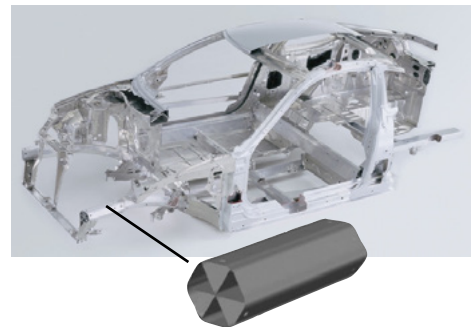


Fig. 1 Aluminum crash box

performance of each when installed at different angles in the frame structure. The experimental data are then used to increase the accuracy of crash simulation.

2. Load Applied to the Crash Box and Testing Apparatus

Generating a collapse mode in a frame involving continuous buckling from the end is effective for reducing weight and increasing the amount of energy absorbed⁽²⁾(Fig. 2). In addition, a shape that can realize a stable collapse mode with minimal load fluctuation during compression is preferable.

In practice, the front side frame that constitutes the crash box is installed at an angle as shown in Fig. 3 to provide

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adequate space for both the engine and suspension. As the frame angle increases, the moment increases by the offset of the axial direction at the locations where the crash box is attached to the frame structure against the load input during a crash. If the maximum bending moment of the frame is exceeded, the bending mode shown in Fig. 4 occurs, and the amount of energy that can be absorbed decreases.

It is therefore important to determine the load applied to the frame. A crash test was conducted in the form of a sled test to experimentally confirm the relationship between the buckling mode of the front side frame and the frame installation angle, and to determine the load required to generate the buckling mode. The equipment used in the crash test is shown in Fig. 5.

Equation (1) represents the bending moment applied to the frame in this test as represented by Fig. 6.

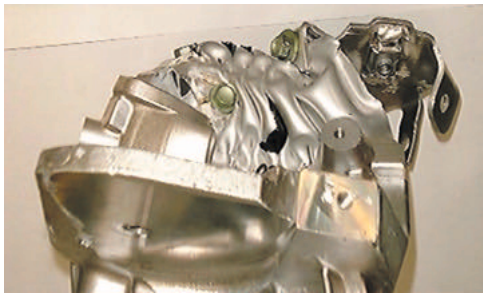


Fig. 2 Collapse mode of crash frame

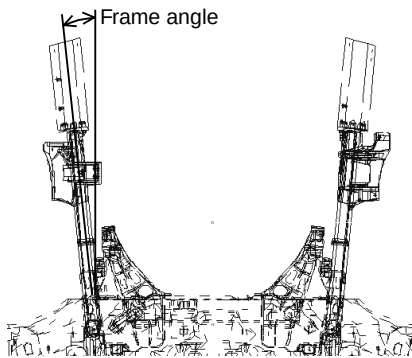


Fig. 3 Frame layout



Fig. 4 Bend mode of crash frame

$$M_z = F_{m2} \times (l_f \tan \theta) + F_y \times l_f + F_{m1} (l_m - l_{m1}) + \frac{F_{m1} (l_m^3 - l_{m1}^3)}{3l_m^2} + F_{ex} \times l_e \quad (1)$$

The symbols used have the following definitions:

- θ : Frame angle
- F_y : Horizontal force resulting from bumper beam
- F_{m1}, F_{m2} : Barrier load
- F_{ex} : Engine forward and backward inertia force
- l_{m1} : Distance from bumper beam load point to bumper center
- l_{m2} : Distance from bumper beam load point to frame load point
- l_f : Distance from barrier to frame attachment point
- l_e : Distance from frame attachment point to mount attachment point
- M_z : Moment applied to frame attachment point (clockwise rotation is positive)

Engine inertia force can be calculated from experimental values.

$$F_{ex} = m_{ex} \cdot a_{ex} \quad (2)$$

- m_{ex} : Engine mass
- a_{ex} : Forward and backward direction engine acceleration

In addition, experiments on each part were conducted separately for horizontal force F_y of the bumper beam, and applied to equation (1) after obtaining the relationship between deformation and load.

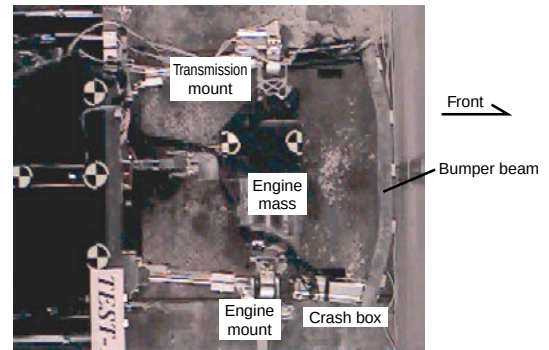


Fig. 5 Crash test equipment for frame research

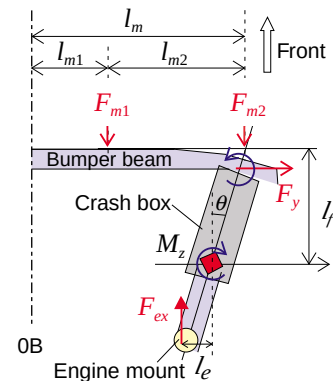


Fig. 6 Calculate moment for crash test

The results from the crash test indicated a collapse mode occurring at a frame angle of 5.6 degrees, and a bending mode occurred at a frame angle of 7 degrees. The following is the comparison of each mode.

Fig. 7 shows the energy absorption performance. The load in the axial direction is the sum of F_{m1} and F_{m2} , and differences due to the buckling mode became prominent at a post-crash deformation of 180 mm or more. Fig. 8 shows moment M_z . In the case of a frame angle of 7 degrees, moment M_z increased with a deformation of 180 mm, and the bending mode was initiated at 2.9 kN·m.

Calculation of the load applied to the frame was made possible by equation (1) from the experimental values, and the relationship between the limiting load for onset of the collapse mode and the frame installation angle could therefore be determined from the crash test data.

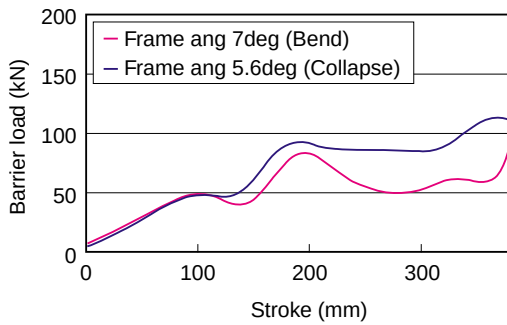


Fig. 7 Stroke-barrier load for crash test

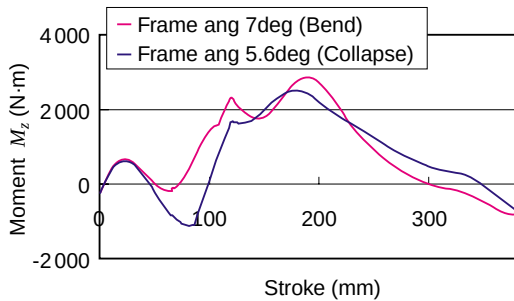


Fig. 8 Stroke-calculated moment M_z for crash test

3. Analysis of Frame Performance

3.1 Test Piece

A hexagonal cross section shape has been found suitable for achieving a stabilized collapse mode⁽¹⁾. Previous studies on rectangular sections have indicated the effect of the corner radius on the collapse load.⁽³⁾ An attempt was therefore made to evaluate the hexagonal cross section by focusing on the corner areas of the section while changing the shape.

Fig. 9 shows the cross section shape and corner area of the hexagonal section that were studied. R_1 and R_2 (Fig. 10) respectively indicate the internal and external radius in the three-legged corner area of the section. Table 1 shows the cases of radii tested, in which "Area" refers to the cross sectional area.

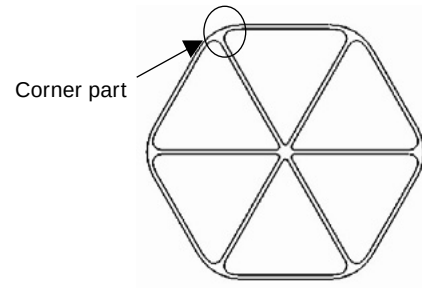


Fig. 9 Section view of hexagonal frame

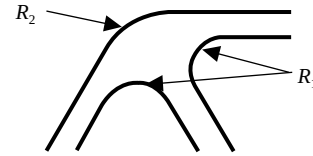


Fig. 10 Detail of corner part

Table 1 Scales of test-specimen

Case	R_1 (mm)	R_2 (mm)	Area (mm ²)
A	2	11	894
B	3	18	907
C	4.5	30	914

3.2 Compression Load

The collapse load was measured for each of the cases shown in Table 1. A summary of the test equipment is shown in Fig. 11. An AUTOGRAPH AG-10TD (Shimadzu Co., Ltd.) was used for the compression test. The compression load was measured with a load cell installed on the cross head, and the distance between the load face of the compression jig and the stage on which the test piece was mounted is designated as the compression stroke.

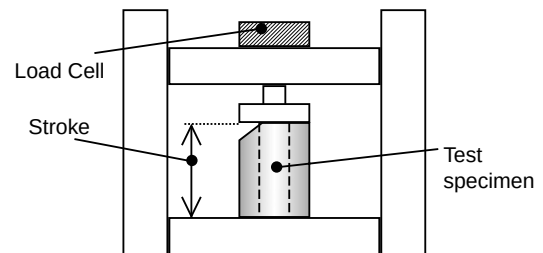


Fig. 11 Test equipment set-up

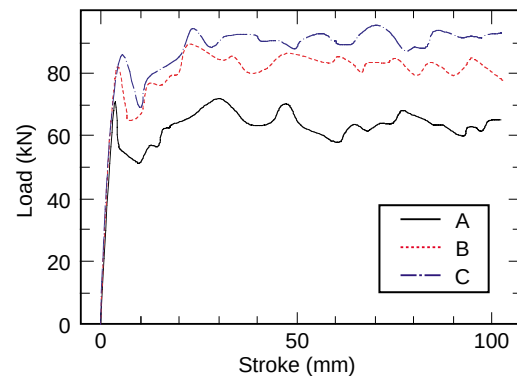


Fig. 12 Relationship between collapse (buckling) load and stroke

Fig. 12 shows the relationship between collapse load and stroke. In the figure, the solid line, dotted line and single dot broken line represent the test results of Case A, B and C, respectively. The collapse/buckling load can be seen to increase depending on the area between the inside corner part radius R_1 and outside corner part radius R_2 .

3.3 Maximum Bending Moment

The maximum bending moment was measured next, a schematic illustration of the test being shown in Fig. 13. Three-point bending was used for the test, the bending load being measured by compression test equipment in a similar manner to that for measuring the collapse load. Equation (3) was used to calculate maximum bending moment M_{Smax} from the measured load.

$$M_{Smax} = W_{Smax} a(l-a)/l \quad (3)$$

W_{Smax} is the peak value for the bending load in equation (3). In order to prevent localized deformation at jig mount points due to the hollow structure of the test pieces, the test piece length was set at $a = 2.5$ m and $l = 5.0$ m to be sufficiently long in comparison with the strength of the shearing force. In addition, iron inserts were inserted at the load point and support points on both ends to prevent local cross sectional deformation.

Fig. 14 shows the results of the bending moment test. These lines represent the same design cases as in Fig. 12. Although stroke distance to buckling increased with increased corner radius (cases A to C) similar to correlation between stroke to collapse and corner radius, correlation between the maximum bending moment M_{Smax} and corner radius is low. Fig. 15 shows a comparison of the results for the maximum bending moment between the three-point bending and crash tests. The difference in shape of these two curves is believed to have been due to the crash test measurement of the bending strength being under an axial load, while the three-point bending test did not involve an axial load.

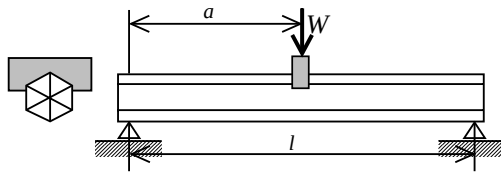


Fig. 13 Test set-up for bending frame

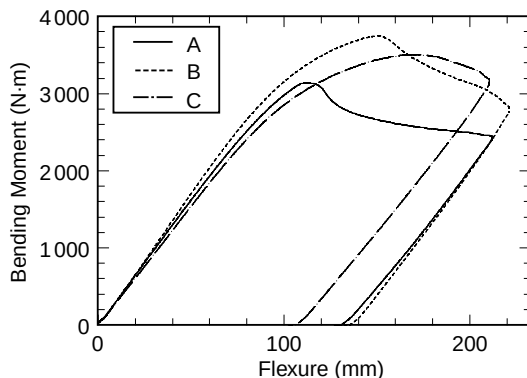


Fig. 14 Relationship between Bending moment and Flexure

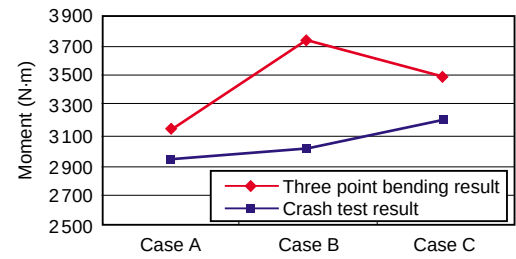


Fig. 15 Correlation between three point bending and crash test result

3.4 Mode Determination Conditions

This analysis focuses on the threshold value for the frame-buckling mode. This study evaluated the maximum moment arm, L_m , which was determined from the frame angle and frame length. L_m is defined by equation (4).

$$L_m = M_{max} / F_{ave} \quad (4)$$

The symbols used have the following definitions:

M_{max} : Maximum bending moment obtained from the crash test

F_{ave} : Average compression load

The results of evaluation of results for the performance of the framelayout from the collapse load and bending moment are shown in Table 2. Although Case B demonstrated the largest value, there was only a very small difference compared to Case A, and conversely, the difference compared to Case C was only about 10%.

The foregoing results indicate little change in the moment arm due to changes in the shape of the corner part. One factor behind this is that, in the case of the collapse load of Case C relative to Case A, it increased just under 30% while the sectional area remained constant. While a change in the shape of the corner part had an effect on the collapse load, it had little effect on the installation angle that determines the bending mode. The following provides a discussion that focuses on this point.

Table 2 Result of lay out performance evaluation test

Case	F_{ave} (kN)	M_{max} (kN·m)	Moment arm length (m)
A	69	3.1	45×10^{-3}
B	80	3.7	46×10^{-3}
C	89	3.5	39×10^{-3}

4. Discussion and Theoretical Model

4.1 Examination of the Collapse Load

Fig. 16 shows a cross section of the corner part, while Fig. 17 shows the relationship between its sectional area and the collapse load. The results show a linear relationship between the sectional area of the corner part and the collapse load. Based on this, the cross section was divided into three parts, and the energy absorbed by each was measured in order to calculate to what extent the corner area was responsible for absorbing energy.

Fig. 18 shows how the section was divided into three parts comprising (1) the internal rib, (2) the external wall,



Fig. 16 Focused cross section of area corner

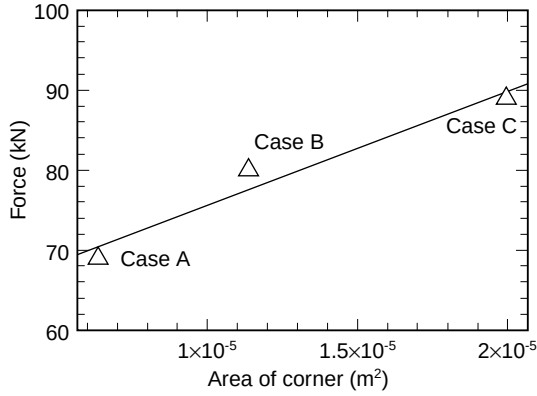


Fig. 17 Relationship between force and area of corner

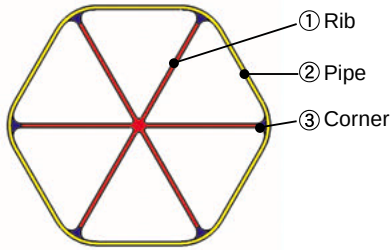


Fig. 18 Definition of divided parts

and (3) the corner area. Parts (1) and (2) were cut out for direct measurement. Each of these parts was cut to a length such that it would deform according to the buckling pitch of the collapse mode of the complete section. The amount of absorbed energy was calculated by multiplying the load and displacement. The contribution of part (3) was calculated by subtracting the amounts of energy absorbed by parts (1) and (2) from the total amount of energy absorbed by the complete section.

Fig. 19 shows the average amount of absorbed energy per unit of weight (referred to as the crash energy density) for Cases A, B and C of the cross section.

Equation (5) was empirically derived from the results of the experiment for estimating the sectional area and collapse load of each part.

$$F_{ave} = C (A_{rib} + 2A_{pipe} + 6A_{corner}) \quad (5)$$

where,

- A_{rib} : Rib sectional area
- A_{pipe} : Pipe sectional area
- A_{corner} : Corner part sectional area
- C : Constant value, 4.72×10^7

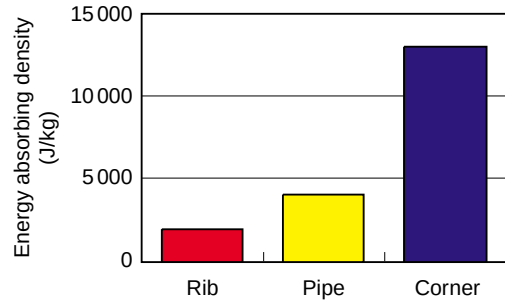


Fig. 19 Energy absorbing density of divided parts

The results of applying equation (5) to Cases A, B and C are shown in Fig. 20. The correlation plotted on the vertical axis uses the value obtained by dividing the calculated value with the experimental value. The error was 10% or less for all cases, thereby confirming the validity of equation (4) for calculating the collapse/buckling load of the aluminum crash boxes.

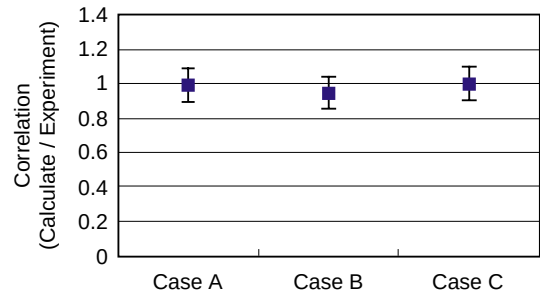


Fig. 20 Correlation of collapse load calculation

4.2 Effect of Cross Section Shape on the Energy Absorption Density

The cross sectional area of the corner part made a considerable contribution to the collapse load. Therefore, a new shape was produced by maximizing inside radius R_1 and optimizing outside radius R_2 with the aim of achieving a collapse load of 150 kN with equation (5) (Fig. 21).

Fig. 22 shows the collapse loads for the new shape and

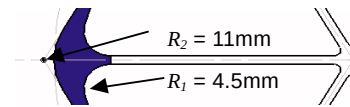


Fig. 21 Detail of new shape

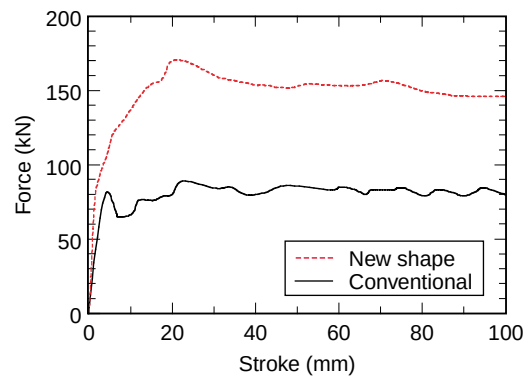


Fig. 22 Relationship between collapse load and stroke (Conventional and new shape)

for the Case B shape. The sectional area of the new shape was 1169 mm² due to the dimensional changes in the corner region. The collapse load for the new shape was an average of 150 kN, roughly double that for the Case B shape.

While the crash energy density was 3.1×10^4 (J/kg) for the Case B shape, it was 4.8×10^4 (J/kg) for the new shape, an improvement of 54%.

4.3 Maximum Bending Moment

A theoretical formula was also proposed for the maximum bending moment. Fig. 23 shows the test piece after the bending test. Measurements were taken for the cut-out parts above and below the neutral axis at the center of the test piece. A wall-buckling mode similar to collapse deformation caused by compression appeared in the region above the neutral axis, while plastic deformation caused by tension occurred below.

On the basis of these observations, an assumption of the compressive region plastic deformation by the collapse/buckling load, while the tensile region is defined as the moment resulting from fully plastic conditions as shown in Fig. 24. Their experimental values were then compared. The collapse load in the compressive region is thought to have acted on the centroid in the compressive half of the cross section. The maximum bending moment can be represented by equation (6). G and A_{all} represent the distance from the centroid to the center of the entire cross section and the total area of the hexagonal cross section.

$$M_{max} = \frac{A_{all}}{2} G \sigma_y + \frac{F_{ave}}{2} G \quad (6)$$

Fig. 25 shows the correlation between the results calculated with equation 6 based on yield stress $\sigma_y = 180$

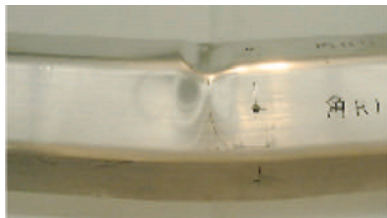


Fig. 23 Test specimen after bending test

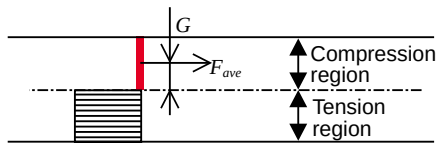


Fig. 24 Definition of bending moment calculation

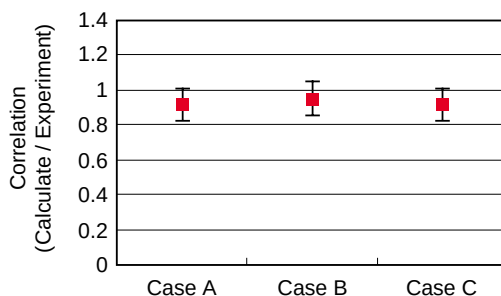


Fig. 25 Correlation of bending moment calculation

Mpa and the experimental values obtained from the crash test. The difference between the calculated values and experimental values was at the 7% level or less, indicating that it was possible to obtain satisfactory results by calculation.

The collapse load and bending strength could thus be suitably calculated by the equations, focusing on the area of the corner part.

5. Establishment of a Predictive Simulation Model

The findings from the experimental analysis in section 4 indicated the desirability of a model that could accurately simulate the buckling of a strut with internal ribs in the corner region.

The stress distribution in the thickness direction of the strut corner part was analyzed with solid model in RADIOSS simulation software in order to establish a highly accurate predictive technique, while a study was also conducted on a simpler modeling method.

5.1 Solid Model Analysis

A model was produced from solid elements and then analyzed in detail in order to elucidate the stress distribution in the corner part. Calculations were first made for one, two, three and four plies each in the thickness direction by using the 1/6 model shown in Fig. 26 in order to determine the number of element plies that would be needed to provide sufficiently high accuracy.

Comparison between calculated results of each ply are shown in Fig. 26. The comparisons of the deformed shape and internal stress distribution indicated that the results of sufficient accuracy could be obtained with a minimum of three plies. A graph of the load versus deformation during compression with a minimum of three plies is shown in Fig. 27. Concerning the average values for the collapse load, the calculated values were 2% lower than the experimental results.

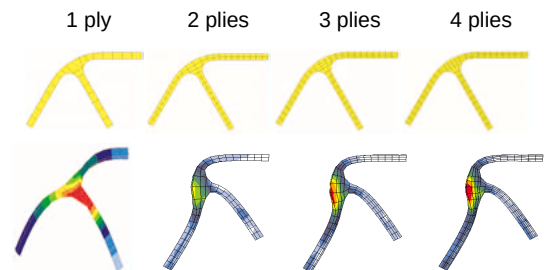


Fig. 26 Element plies of thickness and simulation results for deformation and stress

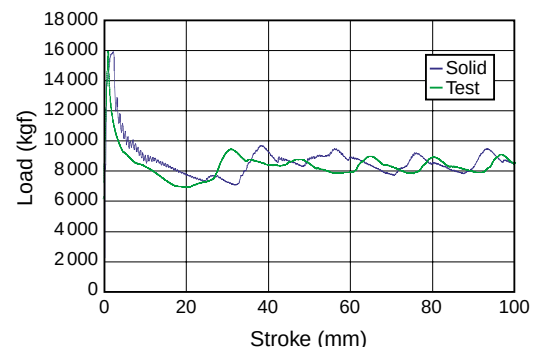


Fig. 27 Simulation result for solid model

The calculated stress distribution generated in the corner part is shown in Fig. 28. The stress was found to be greatest in the corner part and was generated in the thickness direction through the thickness, so the stress in the thickness direction had a significant effect on determining the collapse/buckling load.

Although the results calculated with the solid model were satisfactory in terms of their accuracy, there was the shortcoming of a single member requiring a considerable amount of calculation time. Therefore, two other methods were examined with the aim of reducing the calculation time. The first was a method for obtaining the equivalent load characteristics by using shell elements, and the other enabled increased accuracy of integration in the thickness direction, even with one ply, by tuning the solid model elements.

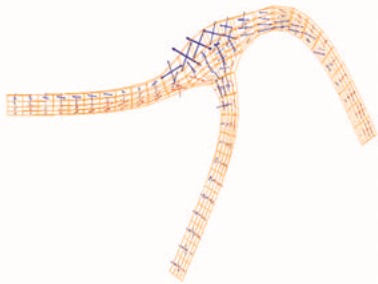


Fig. 28 Maxmum principal stress

5.2 Establishment of a Tuned Shell Model

In the case of an ordinary shell model, load is transmitted through a single node of the internal rib. Since the action of the load in the thickness direction was therefore considered to be different, modifications were made as shown in Fig. 29.

- (1) Shell elements were added around the internal curve of the corner part to cause force to be transmitted in the thickness direction around the periphery of the corner part.
- (2) The thickness of added shell elements were adjusted to give the same collapse/buckling load as that obtained in the experiments.

This thickness adjustment was based on the relationship between the sectional area of the corner part in the actual strut and that in the tuned shell model. As shown in Fig. 30, results were obtained for the relationship between A_0 and A_1 in the form of equation (7), with A_0 being the sectional area of the actual strut and A_1 being the sectional

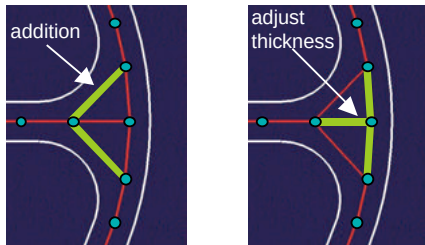


Fig. 29 Section shape of shell model

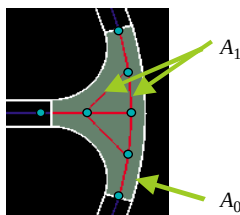


Fig. 30 Section shape of shell model

area of the additional parts and adjusted-thickness parts of the tuned shell model.

$$A_1 = 1.5 \times A_0 \quad (7)$$

Fig. 31 shows the calculated results for Case B when A_0 was 25.9 mm² and A_1 was 40.0 mm². The difference between the calculated results and those of the individual frame compression test was within 8% in terms of the average compressive load.

Differences in the shape of the corner part ($R_1 = 11$ and 30 mm) can also be represented with the tuned shell model based on the same relationship (Fig. 32), and a comparison between the calculated and experimental values is shown in Table 3. The use of a tuned shell model focused on the corner part enabled highly accurate modeling with about the same calculation time as that required for a conventional shell model.

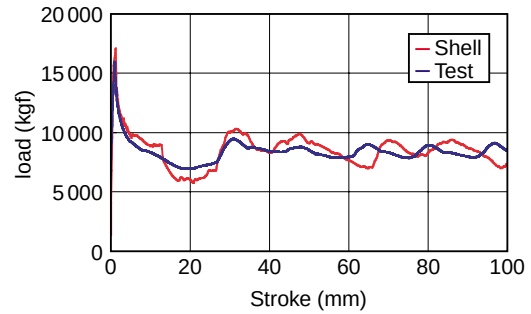


Fig. 31 Load and stroke result for the shell model modified

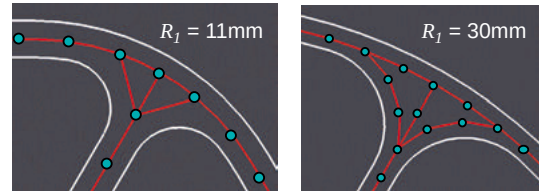


Fig. 32 Section shape of tuned shell model

Table 3 Result of loads

Case \ Load	Experiment F_{ave} (kN)	Calculate F_{ave} (kN)
A	69	69
B	80	78
C	89	92

5.3 Thick Shell Elements

Thick shell elements were examined in an attempt to modify the elements that would enable integration of the relationship between the stress and strain for changes in the thickness direction. Fig. 33 shows a model of the corner part with solid elements and thick shell elements. The thick shell elements represent thickness with a shape function in the same manner as solid elements.

However, in order to reduce the calculation time, a second-order shape function providing intermediate nodes was used in the width and length directions, while a first-

order shape function was used for the thickness direction. This was done to enlarge the shape that could be represented by a single element, while also enabling a model to be produced with a smaller number of elements even for the same shape.

Next, calculation of stress inside the elements is described. Fig. 34 shows a single thick shell element. Integration points were provided at $3 \times 3 = 9$ locations on the outer surface and setting of the direction through the thickness was made at maximum of 9 locations. The analytical results for the internal stress in a solid model indicated that three or more plies were necessary in the thickness direction in order to obtain adequate accuracy. Namely, it was considered that calculations could be made with sufficient accuracy if integration points were provided at six or more locations in the thickness direction in the case of thick shell elements.

Fig. 35 shows a comparison between the calculated results for thick shell elements and those for the solid model (three plies in the thickness direction). These two sets of calculated results can be seen to closely agree. In addition, the calculation time was only about 1/30 of that required for a conventional solid model.

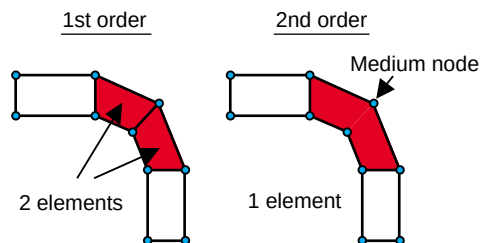


Fig. 33 Corner modelings

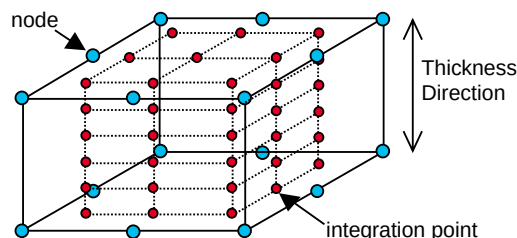


Fig. 34 Integration points

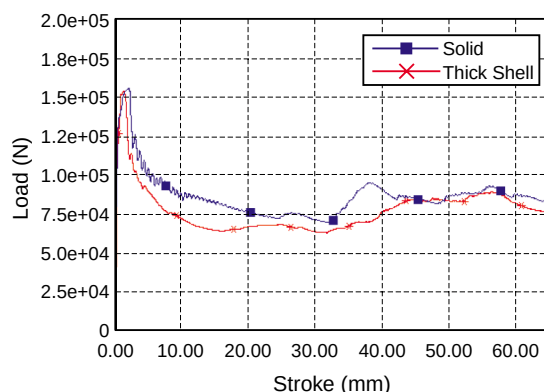


Fig. 35 Load and stroke result for thick shell model

6. Conclusion

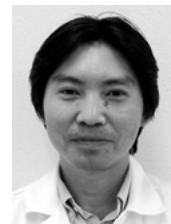
A study was made by experimental tests and simulation methods on the energy-absorbing members of a ribbed aluminum crash box. The following results were obtained.

- (1) Using a sled test, the bending moment of the compressive load applied to the box was calculated, and a method was established for calculating the moment at the time of bending mode of the box.
- (2) An analysis of the energy-absorbing box determined the factors and locations of compressive load and bending strength in relation to the cross-sectional shape, and it was found that the corner part made a significant contribution. An empirical equation was established that made calculation within an error of 10% possible.
- (3) Various simulation techniques were studied to identify the best match with the experimental results, and a suitable modeling method was then established.

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