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Built-Up Members in Plastic Design

TEST FIXTURE FOR RECTANGULAR PLATES IN POST BUCKLING RANGE

by

Donald R. Rutledge

Fritz Engineering Laboratory Report No. 248.25
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Fritz Engineering Laboratory
Department of Civil Engineering
Lehigh University
March 1969
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ABSTRACT

A description of a test fixture designed to investigate the large deflection post-buckling behavior of long rectangular plates is presented. The fixture simulates the support of the plate by longitudinal stiffeners. It permits the load and deformation of the plate alone to be measured. Compressive loads are applied in the plane of the plate at two opposite ends. Sketches suggesting methods of applying lateral loading are included.
1. INTRODUCTION

A theoretical elastic-plastic analysis of a longitudinally stiffened plate-panel subjected to an axial compressive load and a uniformly distributed lateral loading is described in Refs. 1, 2, and 3. The problem considered is characterized by a large panel width to thickness ratio (b/t ratio) so that the plate component buckles under loads smaller than the ultimate loads for the total stiffened plate panel.

The plate is subdivided into sub-panels by a series of regularly spaced longitudinal stiffeners and the behavior of a typical repeating element of the structure is assumed to be representative of the behavior of the entire structure. For example, Fig. 1 illustrates a stiffened plate structure which may be a portion of a ship bottom between large transverse bulkheads. Figure 2 shows the representative repeating element with the loads acting on it.

It is assumed that the stiffener does not buckle. However, the top flange formed by the plate can buckle long before the failure load for the total structure is reached. If the behavior of this plate component before and after buckling can be described by some load-deformation relationship, the problem of determining the ultimate strength analysis of a beam-column consisting of a stiffener and a plate with different material properties. The needed material property of the plate component is the average axial stress-
edge strain relationship (edge strain is axial strain along the plate-stiffener junction).

The behavior of the plate component before it buckles and in the elastic post-buckling range is well known. However, no known theoretical and very little experimental information is available for the inelastic post-buckling behavior, particularly after the ultimate load. The test fixture described in this report was designed to furnish this information.

For the problem that was considered in the references, the magnitude of lateral loading was believed to be too small in comparison with the axial load to significantly affect the average stress-edge strain relationship of a single plate panel. Thus, the fixture design presented in this report does not include a means of applying lateral loading to the specimen plate panel. Suggestions for ways in which lateral loading may be supplied, if desired, appear in Chapter 3.

Detailed drawings of the fixture are available on request from Professor A. Ostapenko, Fritz Engineering Laboratory, Lehigh University, Bethlehem, Pennsylvania.
2. TEST FIXTURE-ASSUMPTIONS AND DESCRIPTION

A test fixture is described for experimental determination of the average stress-edge strain relationship of a rectangular plate compressively loaded in the plane of the plate. The purpose of the fixture is to restrain the plate as it would be restrained by longitudinal stiffeners and yet to permit only the load-deformation behavior of the plate itself to be measured. The form of the fixtures described is a result of some inevitable problem simplifications and assumptions, as follows:

a. The torsional rigidity of the tee stiffeners (see Figs. 1 and 2) is assumed to be small. In the test fixture a panel is restrained in its original plane along the side edges by longitudinal KNIFE EDGES*. Five such panels are shown in Fig. 3 as they would be restrained in the fixture.

b. Any panel of interest is continuous to the sides with identical panels. Two panels either side of the test panel were considered sufficient to provide the boundary condition of continuity.

c. The panels are compressively loaded in the plane of the plate in the direction of the stiffeners on the free ends

* Note: Parts of the fixture identified by name in Fig. 9 are capitilized.
as shown in Fig. 3. A uniform end displacement is imposed and the resulting average stress in the center panel is to be determined. Actual contact with the plate ends is made with grooved BEARING INSERTS mounted on loading PLATENS. The grooves restrain the loaded ends in the original plane of the plate yet allow free rotation and transverse expansion of these ends. Figure 4 shows a portion of a BEARING INSERT in contact with the plate to be tested.

d. For the problem of primary concern (ship bottom plating), the magnitude of lateral loading is small compared to the magnitude of axial load. No provision for the application of lateral loading is made in the fixture design. Suggestions are offered in Chapter 3 for tests with lateral loading.

The behavior of a stiffened plate panel is principally influenced by the material properties, the length to width ratio of the panel (L/b ratio), and the panel width to plate thickness ratio (b/t ratio). The test fixture presented here divides a plate specimen into panels 12 inches wide and 40 inches long (L/b=3.33). A change in the L/b ratio would require fixture modification; the bolt holes in the FIXTURE BEAM which determine the spacing between KNIFE EDGE SUPPORTS must be relocated, and corresponding new milled grooves which accept the ends of the KNIFE EDGE SUPPORTS
must be provided in the UPPER BEARING INSERT. The b/t ratio is more easily varied, within limits, by changing the specimen plate thickness. The milled grooves in the BEARING INSERTS are designed to accept gage 9 through gage 13 plates (United States Standard Gage). Thus, b/t ratios between 80 and 134 are represented.

Studies of reports by Botman and Besseling describing tests conducted in fixtures very similar to the one described here indicate that an important cause of unsuccessful tests was inadequate attention to the demands on the KNIFE EDGES. Some tests could not be carried to completion because large frictional forces between KNIFE EDGES and the specimen plate produced erroneous load measurements; others were terminated by buckling of the plate between opposing KNIFE EDGES resulting in a total specimen to KNIFE EDGE lockup.

The solution to the frictional problem was adopted from Besseling. Small brass cylinders form the "knives" of the KNIFE EDGES as shown in Fig. 5. These cylinders, placed with 1/4 inch gaps for free movement and heavily lubricated for friction reduction and adhesive purposes, allow free panel shortening, rotation at KNIFE EDGES, and transverse expansion.

The second problem of the plate local buckling between two opposite KNIFE EDGES, is solved by providing adequately rigid banks of KNIFE EDGES. The required KNIFE EDGE rigidity was determined from a simple model developed by studying photographs of the
tests performed by Botman\(^{(6)}\). A simply supported panel, with \(L/b = 3.33\), will elastically buckle into three half sine waves. The intersection of the anti-symmetric waves at a KNIFE EDGE produces nodes at which local buckling between opposing KNIFE EDGES tends to occur under high axial loads. After buckling between KNIFE EDGES begins, it produces a mechanism which can apply ever greater loads to the KNIFE EDGES until finally the plate folds upon itself in a Zee configuration.

The schematic drawing in the left half of Fig. 6 shows a cross section of a compressively loaded plate at a KNIFE EDGE location. Four local buckling nodes are forming; they are resisted by the contact forces with the KNIFE EDGES (schematically, the springs). A representative mechanism is also shown in Fig. 6. Both, normal forces and tangential frictional forces are supplied by the KNIFE EDGES. However, since care has been taken to design the KNIFE EDGES as frictionless as possible, the frictional forces are assumed equal to zero (the assumption is conservative since a greater total spreading force is applied to each KNIFE EDGE).

Without friction, \(N_1 = N_{iH} = N\) and \(P_0 = P_f\). The equation of equilibrium is:

\[
N = P_0 B \left( \frac{1}{A} + \frac{n}{c} \right)
\]

where

\[
A = (L_a^2 - B^2)^{\frac{1}{2}}
\]

\[
B = (S_o - t) + 2
\]
\[ C = L - nL_a \]
\[ P_0 = c \cdot (bt) \cdot \sigma_y \]
\[ L_a = \text{plate folding lever arm} \]
\[ n = \text{number of half-sine waves in the elastically buckled plate panel.} \]
\[ c = \text{a coefficient} \]
\[ S_o = \text{initial separation between opposing KNIFE EDGES} = \text{deflection of a KNIFE EDGE} \]
\[ \sigma_y = \text{yield stress of the plate material} \]

The test panels will elastically buckle into three half-sine waves, therefore \( n \) is equal to three and \( \Delta \) is the deflection of a KNIFE EDGE at the third points. The normal force \( N \) on the KNIFE EDGES is most severe when \( t \) and \( L_a \) are small. Let \( t \) equal the thickness of the 13 gage plate. The plate cannot fold on itself if \( L_a \) is less than \( 2t \), thus

\[ L_a = 2t \]

The length of the test panel \( L \) is 40 inches.

At the beginning of the test, it is reasonable to assume that the KNIFE EDGES can be placed in contact with the flat plate specimen, so that

\[ S_o = t \]

assuming the yield stress for the plates to be 33 ksi ASTM A245, grade C, the coefficient \( c \) is estimated to be 0.25. A value which reflects the facts that the average maximum axial stress in plates
with large b/t is much less than \( \sigma_y \) and that only a portion of the total edge load directly acts to propagate local buckling between the opposite KNIFE EDGES.

Sustitution of all values in Eq. 1 produces

\[
N = 8.88 \Delta \left[ \frac{1}{(0.008 - \Delta^2)^{\frac{1}{2}}} + 0.152 \right] \tag{2}
\]

This equation is plotted in Fig. 7. The figure shows that \( N \) increases rapidly and non-linearly with an increasing \( \Delta \). Since the ratio \( \frac{N}{\Delta} \) can be linearly related to the moment of inertia of a KNIFE EDGE, it is obvious that any local buckling mechanism which forms will propagate unchecked to a total fold without any increase in applied loads. Therefore, the mechanism cannot be allowed to begin. This implies that the stiffness of all KNIFE EDGES at the third points must be greater than \( \frac{dN}{d\Delta} \bigg|_{\Delta=0} \) from Eq. 2. This initial slope of the curve in Fig. 7 is 95 kips per inch. All KNIFE EDGES were designed with third-point stiffnesses of 125 kips per inch so that local buckling between opposite KNIFE EDGES is unlikely.

Figure 8 shows the assembled fixture, with specimen, ready for test. The fixture is shown mounted on a mobile pedestal. A dial gage rig for measuring lateral deflections is in place. Figure 9 is an exploded view of the fixture and Fig. 10 illustrates the assembly sequence.
A sample estimate of the costs for the fixture and three test specimens is given in Table 1.
3. SUGGESTIONS FOR TESTING WITH LATERAL LOADS

The fixture described in Chapter 2 cannot supply lateral loading to the plate specimen. The fixture may be modified slightly to apply lateral loading with a contained air pillow scheme as shown in Fig. 11. The pillow container would, of course, be provided for the bank of KNIFE EDGES on only one side of the plate specimen. The adequacy of the structure to carry the additional loads must be carefully checked.

An extremely simple (in concept, at least) method to test a plate loaded axially and laterally is shown in Fig. 12. If the joints between the cavity forming partitions and the plates are accessible enough for welding and the open ends are capped with an airtight (or water tight) platen (through which the axial loads can be applied) and then if the load supporting action of the stiffening membranes can be made determinate, an economic combined loading test can be performed. One possible way to determine the contribution of the stiffener is to select a low yield material for the plates and a high yield material for the stiffener. If stiffener buckling is avoided by proper dimensioning, then its linear contribution in support of the total axial load may be simply determined from the axial shortening data.
4. TABLES
# Costs Estimate for the Test Fixture and Specimens

<table>
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<tr>
<th>Test Fixture</th>
<th>Cost per Test</th>
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<tr>
<td>Parts requiring machining</td>
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<tr>
<td>Assembly and minor parts</td>
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<td>(material &amp; labor)</td>
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<tr>
<th>Plate Specimens</th>
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<tr>
<td>3 Primary specimens</td>
<td>$120</td>
<td>$360</td>
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<td>@ 40 ea. (includes edge machining &amp; Coupon preparation)</td>
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<td></td>
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<tr>
<td>3 Backup specimens</td>
<td>$60</td>
<td>$180</td>
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<tr>
<td>@ 20 ea. (does not include coupon preparation)</td>
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<tr>
<th>Instrumentation</th>
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<tr>
<td>(All based upon assumption that research personnel will install and wire gages)</td>
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<td>Strain gages:</td>
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<td>12 - A7 rosettes @ $10 ea.</td>
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<td>92 - A7 linear gages @ $3 ea.</td>
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<td>Misc. busses, tape, cement, etc.</td>
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<tr>
<td>Special extension tips for lateral deflection dial gages (or potentiometers)</td>
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<tr>
<td>Total</td>
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<tr>
<td>Total, 3 tests:</td>
<td>$7,630</td>
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## Table 1 Sample Cost Estimate
5. FIGURES
Fig. 1 A Longitudinally Stiffened Plate Structure
Fig. 4  Edge Load Applied by Grooved Inserts
Fig. 5 Friction Reducing Cylinders on Knife Edge
Fig. 6 Assumed Mechanism for Local Buckling Beneath Knife Edges
Fig. 7 Knife Edge Rigidity Requirement

\[ \frac{N}{\Delta} = f(I) \]
Fig. 9 Basic Test Fixture for Unstiffened Plate Panels
Fig. 10 Test Fixture Assembly Sequence
Fig. 11 Modification of Fixture for Providing Lateral Load
Fig. 12 A Simple Pressurized Test Structure
6. REFERENCES

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2. Vojta, J.F., and Ostapenko, A.
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4. Davidson, H.L.

5. Besseling, J.F.

6. Botman, M.
7. ACKNOWLEDGEMENTS

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