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ON THE PLASTIC COLLAPSE OF HORIZONTAL SADDLE SUPPORTED STORAGE VESSELS

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ABSTRACT

The present paper summarises a comprehensive programme of work on collapse loads of horizontal cylindrical saddle supported storage vessels. A programme of tests was conducted on 40 model vessels that included both welded and loose saddles. Different collapse behaviours were observed depending largely on the radius to thickness ratio of the vessels. A range of theoretical approaches were explored and compared with the experimental results. The best theoretical comparison was then used to conduct a parametric survey covering a total of 218 cases. The results of the survey have been presented in the form of simple design graphs.

NOTATION

A	Longitudinal distance to the saddle support centre line from the end of the cylindrical shell (mm)
b_1	Width of saddle (mm)
L	Barrel length of the vessel (mm)
L_S	Longitudinal distance between saddle supports (mm)
R	Mean radius of vessel (mm)
t	Shell thickness of vessel (mm)
σ_y	Tensile yield strength of shell material (N/mm^2)
2α	Saddle embracing angle (degrees or radians)
P	Parametric collapse load by elastic-plastic analysis (kN)
P_{ex}	Experimental collapse load (kN)
P_{min}	Upper bound limit load by limit analysis (kN)
P_{inc}	Inscribed yield solution by limit analysis (kN)
P_{krup}	Krupka's simplified solution by limit analysis (kN)
P_{lb}	Lower bound limit load by elastic compensation method (kN)
P_{ub}	Upper bound limit load by elastic compensation method (kN)

INTRODUCTION

Horizontal vessels are widely used as storage vessels for liquids or gaseous products. Such vessels are commonly supported above ground by twin saddles, Fig 1a. The saddles may be either fitted loosely or welded to the vessel. Current design rules tend to limit the maximum stress in the saddle region to a particular value or to employ a Design-by-Analysis approach. The two approaches differ little in the final design. However, in the absence of a fatigue requirement it may be appropriate to base the design of the vessel on the plastic collapse load of the vessel. In this way the designer can find an allowable load directly from the collapse load by dividing by an appropriate factor, usually 1.5. This has the merit of avoiding the calculation or the categorisation of the stresses.

Plastic collapse loads have been investigated by the authors (1,2,3) and by Krupka (4,5,6,7,8) in a range of simple experimental tests on end supported steel cylinders loaded centrally by an external saddle load, Fig 1b. It was found that there are two main modes of collapse, a gradual plastic collapse and a more sudden elastic-plastic buckling failure. Plastic collapse occurs when the vessel radius to shell thickness is relatively small (typically R/t < 200) and is characterised by the sequential formation of plastic hinges which cause the eventual collapse of the vessel. The formation of plastic hinges is different for saddles which are welded to the vessel than for those where the vessel is placed loosely on the saddle. In the case of *welded saddles* the hinges occur at the horns and (usually) on one side of the nadir close to the saddle. *Loose saddles*, on the other hand, have symmetric hinges which form round the periphery of the saddle/vessel interface; this ultimately results in a localised indentation on the shell surface under the saddle, referred to as a ‘foot print’.

EXPERIMENTAL RESULTS

A typical cylindrical vessel is shown in Figure 1a and assumed to be fluid filled as this represents the worst loading case. The vessel is unstiffened and has two saddle supports each with a saddle reaction force of P. If a portion of the vessel is isolated and inverted it can be considered to be loaded through one saddle with force P as in Figure 1b. The force P can be treated as an applied force which represents the specific weights of the liquid and vessel material. The ends of the model were supported on saddles. This represents a convenient test arrangement.

The results of a programme of 40 experimental results on steel vessels using this inverted configuration, including both welded and loose saddles are reported in (1). The models covered a range of R/t ratios from 50 to 300 and length to radius ratios between 4 to 6. These were brought together with other similar experimental results to give a total experimental base for comparison of 70 tests. The collapse loads are simply the highest load sustained in the test.

THEORETICAL ANALYSES

The following methods of analyses were used (2) to give estimates of the collapse loads for both welded and loose saddles.

a) Rigorous upper bound analysis (9).
b) As above, with an inscribed yield surface (9).
c) A simplified upper bound solution (5,6).
d) An elastic compensation method, lower bound (10,11).
e) An elastic compensation method, upper bound (10,11).
f) An elastic-plastic finite element method.

In the finite element model, for cases d), e) and f), symmetry boundary conditions were applied to the longitudinal and transverse sections of the geometry to produce a quarter model. The open ends of the model were constrained in the circumferential direction but were free to deform in the radial direction or rotate in their plane. This is an approximation to the experimental boundary conditions (1) where there was a degree of radial restraint imposed by thin rings inserted into the open ends of the test cylinder. The finite element analyses inherent in d) e) and f) above were conducted using ANSYS (12).

COMPARISON OF THEORY AND EXPERIMENT

Typical results for welded saddles are shown in Fig 2 and for loose saddles in Fig 3. Various theoretical results are also shown in the figures. Two lines have been drawn through the experimental results. The solid line covers results with R/t ratios that fail by plastic collapse; this shows a rising curve with increasing R/t. The dotted line covers regions where vessels fail by sudden elastic/plastic buckling; this is a gently falling curve as R/t increases. Although the theoretical values are valid only for plastic collapse, results have been included throughout the whole range of R/t for completeness. This serves to emphasise the transition in the buckling behaviour.

The main observation is that the elastic-plastic finite element collapse load gives the best approximation to the experimental results. Subsequently, it was decided to use this method to conduct a parametric survey on actual vessel configurations.

Essentially the above work assumes that collapse is a local phenomenon and a model length of approximately $4R$ is sufficient to avoid interaction effects from the ends of the model (3). While these results are useful for comparisons between theory and experiment, the values need to be treated with care when considering actual vessels.

PARAMETRIC STUDY

It will be appreciated that all of the above theoretical results are restricted since they either treat the saddle-supported problem as a local problem, or they have been configured to suit the experimental set-up. In order to conduct a parametric survey which is appropriate to actual vessels, it is necessary to include all of the factors which influence the collapse of horizontal vessels supported on twin saddle supports: these include,

(a) the fixture of the saddle and vessel, i.e. welded or loose
(b) saddle embracing angle (2α)
(c) saddle width (b_1)
(d) total length of the vessel (L)
(e) distance of the saddle centre profile from the vessel "head" (A)

Vessels of A/R ratio equal to 0.5, 1.0, 2.0 and 6.0 and values of $R\alpha/b_1$ (where α is in radians) of 2.0, 3.5, 5.0, 7.5 and 10.0 were examined. Various vessel radii of 130mm, 500mm, 1000mm and 4000mm were used. The saddle location was restricted to the quarter point on the vessel but with the vessel's total length varying from $2R$, $4R$, $8R$ and $24R$ (representing A/R of 0.5, 1.0, 2.0 and 6.0). The saddle-embracing angle was restricted to between 120° and 150°. The saddle width, saddle embracing angle and the vessel's radius were varied to include a range of values of the ratio $R\alpha/b_1$ namely 2, 3.5, 5.0, 7.5 and 10.0. The thickness of the vessels was such that the R/t ratio does not exceed 300 to adequately ensure the cases correspond to plastic collapse. The material property of the shell is assumed to be elastic-perfectly plastic with a yield strength of 300N/mm^2. A total of 105 vessels with welded saddles and 113 vessels with loose saddles were analysed to determine the various collapse loads.

Fig 4 shows some typical results for a welded saddle for a fixed value A/R = 1.0. There is a slight scatter in the results due to the combined geometrical parameter involving b_1/R and R/t. Results for loose saddles are generally similar but tend to be slightly lower for a given geometry. Other values of A/R are given in (3). The collapse loads reduce as A/R increases so that when A/R = 6 the values are approximately $^1/_3$ of those shown in Fig 4.

From the point of view of application in design situations, it may be useful to have the parametric results in a more directly useable form. Accordingly the best fit curves for the data, have been characterised in terms of a simple power law of the form,

$$\frac{P}{\sigma_y t^2} = K_1\left(\frac{R\alpha}{b_1}\right)^n \qquad (1)$$

Values of K_1 and n are given in Table 1 for the welded saddle cases for A/R = 1.0. Values for other A/R ratios are given in (3). It is in fact possible to further condense the parametric collapse load results by increasing the combination of geometric parameters, albeit this results in some additional scatter. The results are shown in Figures 5 and 6 for typical welded and loose saddle cases against the grouped parameter $\frac{b_1}{\sqrt{Rt}}\left(\frac{R\alpha}{b_1}\right)$. This has the merit of allowing all the results to be shown neatly on one graph. Again these may be fitted with a simple power law of the form,

$$\frac{P}{\sigma_y t^2} = K_2\left[\frac{b_1}{\sqrt{Rt}}\left(\frac{R\alpha}{b_1}\right)^{0.93}\right]^m \qquad (2)$$

for the welded case and

$$\frac{P}{\sigma_y t^2} = K_2\left[\frac{b_1}{\sqrt{Rt}}\left(\frac{R\alpha}{b_1}\right)^{q}\right]^m \qquad (3)$$

for the loose saddle case. The values of K_2 and m are given in Table 2 for both the welded and loose saddle cases (with the values of q identified in Fig. 6 for the loose case).

CONCLUDING COMMENTS

The results of the parametric study are useful tools in determining the collapse load of twin saddle supported vessels that may fail by plastic collapse. The validity of these curves is restricted to vessels supported by saddles with embracing angles of 120° to 150° and to the range of parameters covered. It must be emphasised that they are only valid for failure by plastic collapse; they are not relevant to situations where elastic buckling or fatigue are likely failure modes.

Although the parametric results are for vessels that are supported by twin saddles at the quarter points, they may also be used for vessels which are not supported at the quarter points. A simple approach would be to use the *A/R* ratio for that particular vessel since the distance between the supports does not greatly influence the collapse load (3). One must ensure that the appropriate load is used in the calculation.

For the purposes of practical design, one approach would be to reduce the collapse load obtained from the design curves by a factor, say 1.5, to obtain a working load. The total load (fluid and vessel weight) acting on one saddle should be less than this working load. If the total load required exceeds the allowable working load, then the design and the allowable working load may be achieved by altering the vessel/saddle parameters.

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Table 1. Graph curve-fit constants for welded saddle with *A/R*=1.0

$R\alpha/b_1$	K_1	n
10	71.198	1.4604
7.5	45.598	1.5391
5	27.737	1.4136
3.5	16.301	1.4558
2	7.6067	1.4741

Table 2. Graph curve-fit constants for saddle condensed data

	Welded		Loose	
A/R	K_2	m	K_2	m
0.5	2.58	1.56	5.07	1.363
1.0	3.00	1.47	4.89	1.340
2.0	3.22	1.30	4.25	1.233
6.0	3.08	1.08	2.66	1.126

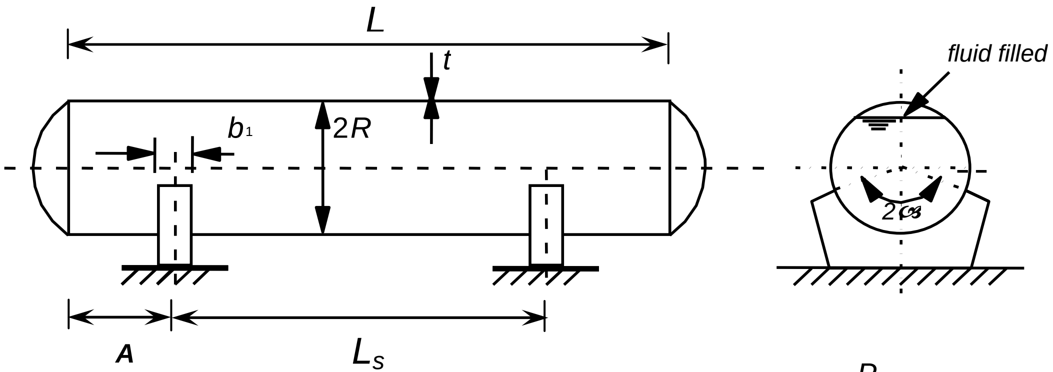

Figure 1(a)

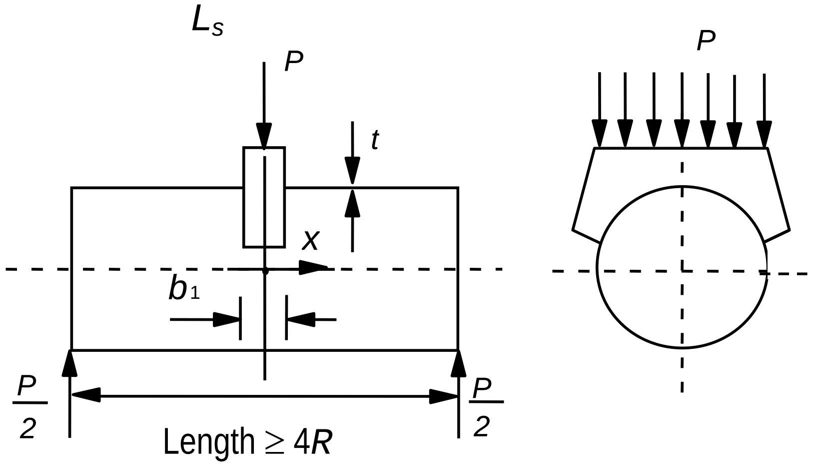

Figure 1(b)

Figure 1. Geometric details of saddle supported vessel and simple test arrangement

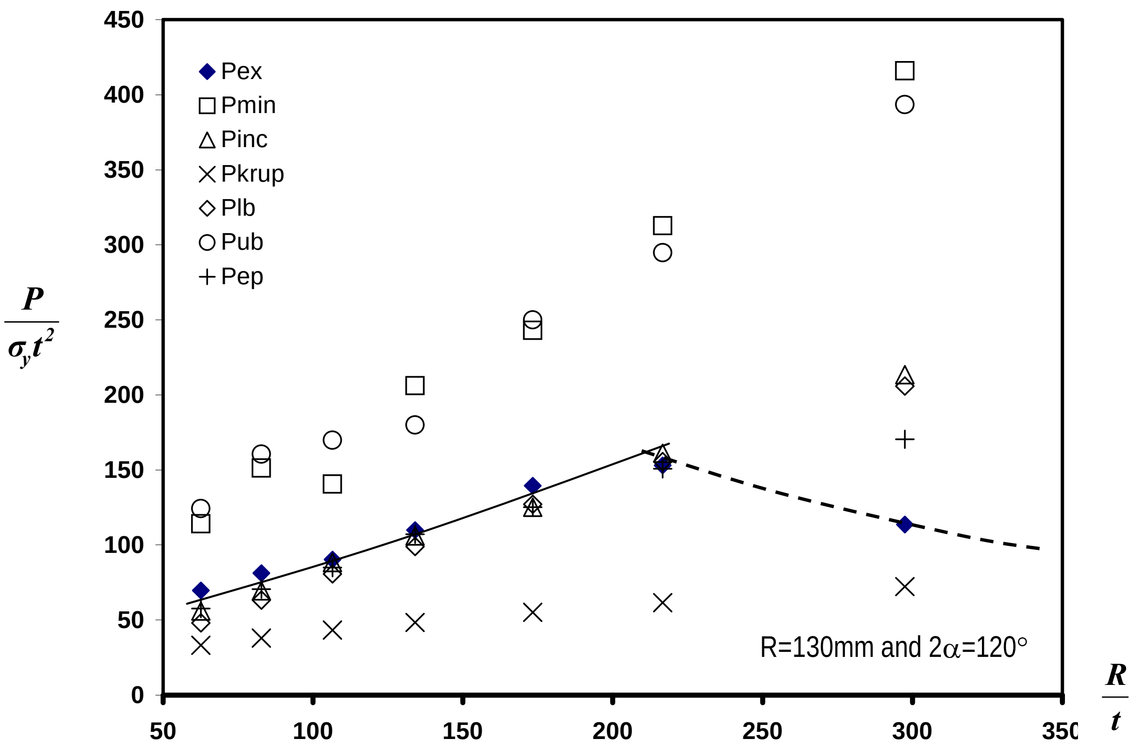

Figure 2. Collapse Loads for Vessels with Welded Saddles

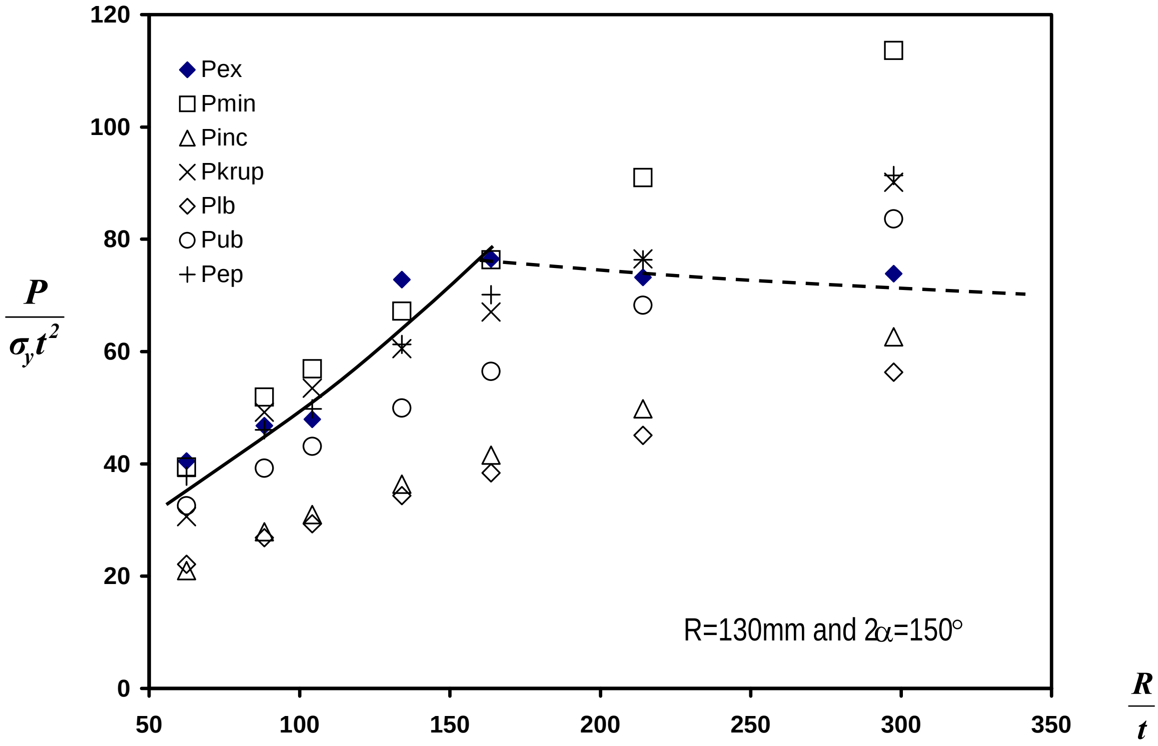

Figure 3. Collapse Loads for Vessels with Loose Saddles

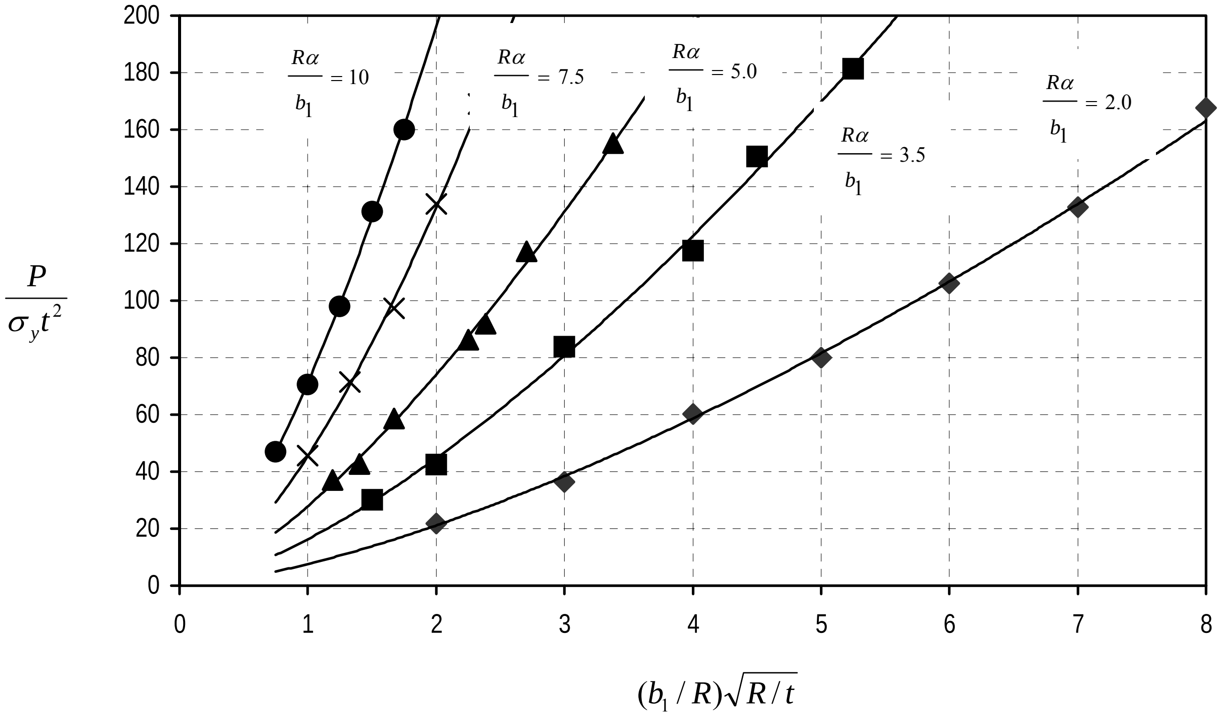

Figure 4. Collapse Loads for Vessels with Welded Saddles for *A/R*=1.0

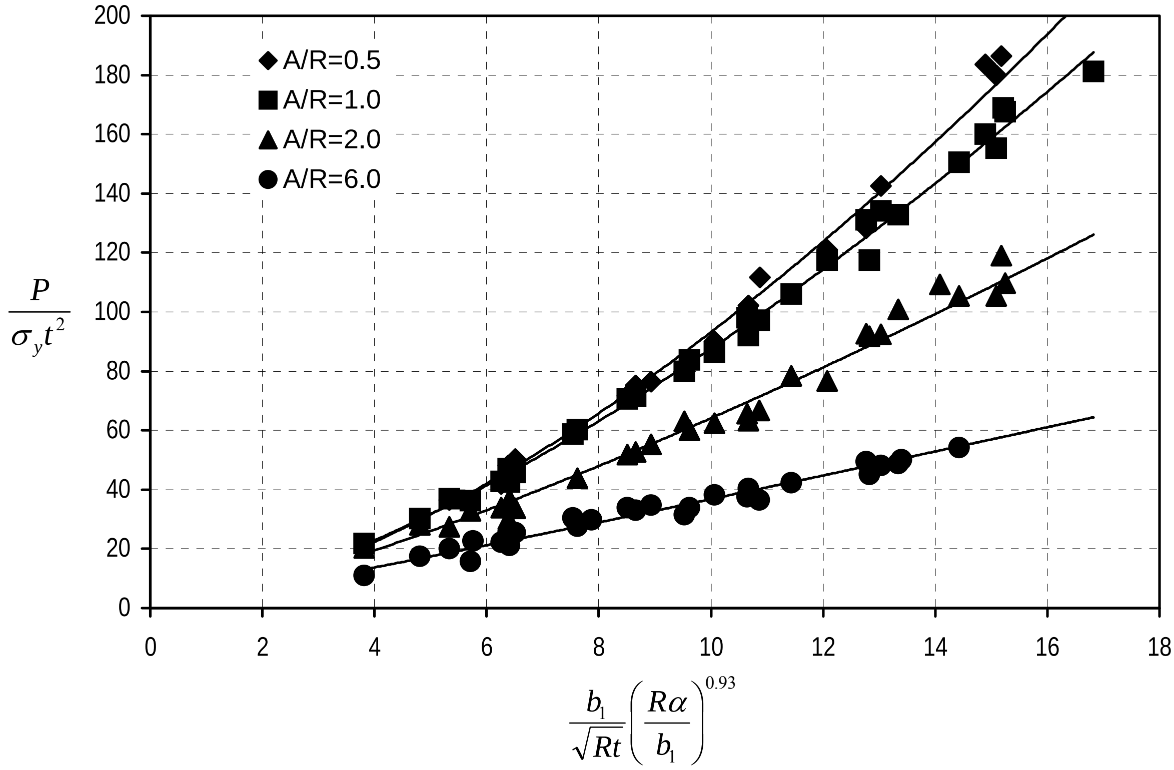

Figure 5. Condensed Plot of Collapse Loads for Vessels with Welded Saddles

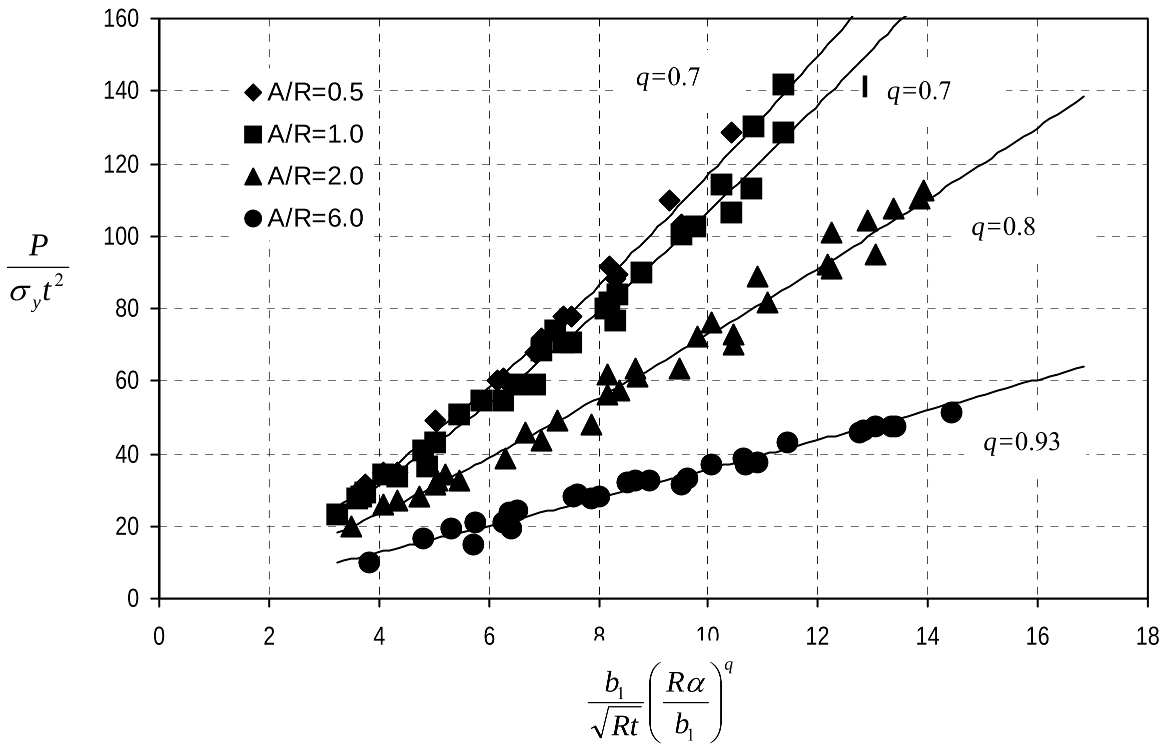

Figure 6. Condensed Plot of Collapse Loads for Vessels with Loose Saddles