Experimental Investigation of Lug Stresses and Failures

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CONNECTIONS AND HANDLING brackets, generally referred to as "lugs", have historically been designed with large end distances in order that pin failure result rather than failure of the lug material. The AISC *Specification for the Design, Fabrication and Erection of Structural Steel for Buildings*[1] in referring to end distances provides minimum values to be used in design, but provides no insight as to allowable loads if a smaller end distance is desirable. Often, it is desirable to lift and handle components of mechanical equipment through the use of existing holes in the components. Many such cases do not meet the minimum edge distance requirements of the AISC Specification.

Previous studies of lug stresses were generally limited in scope and provided no design formulation.[2,3,4,5,6] The most practical approach was presented by Jongbloed-Unterhorst,[7] predicting only the failure plane of the lug under pin load. Variables such as lug width and pin clearance were not treated.

PROBLEM STATEMENT

The purpose of this study, in general, is to improve the understanding of lug behavior under applied load. Lug specimens were tested to relate the effect of pin clearance and lug width variations to stress distribution and failure. Also, an attempt was made to predict the planes along which the shear failure of the lug would occur. Finally, design formulation and recommendations for practical design applications were made.

All experimental studies were limited to small end distances so that lug failure rather than pin failure would occur.

EXPERIMENTAL TECHNIQUE

The reflective photoelastic technique was utilized, wherein a thin layer of birefringent material is bonded to the steel test specimen. The birefringent material strains compatibly with the steel lug under load. Furthermore, the difference in the principal strains of the lug can be measured in the coating material through a direct read-out utilizing a reflection polariscope, a uniform field compensator, and a digital strain indicator. The reinforcing effect of the coating material can be neglected. There are several references for additional reading on this technique and its applications.[8,9,10,11]

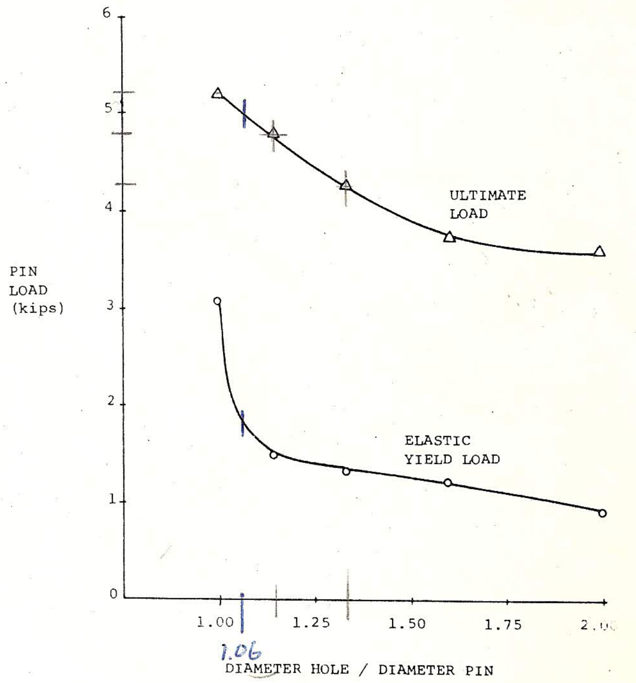

Fig. 1. Experimental results of pin clearance effect on lug elastic and ultimate capacity

EXPERIMENTAL RESULTS

Figure 1 graphs the effect of pin clearance on lug capacity. The elastic capacity and the ultimate capacity were plotted vs. the ratio of the pin hole diameter to the pin diameter. As noted from Fig. 1, pin clearance has a significant effect on lug behavior. Going from a 1-in.

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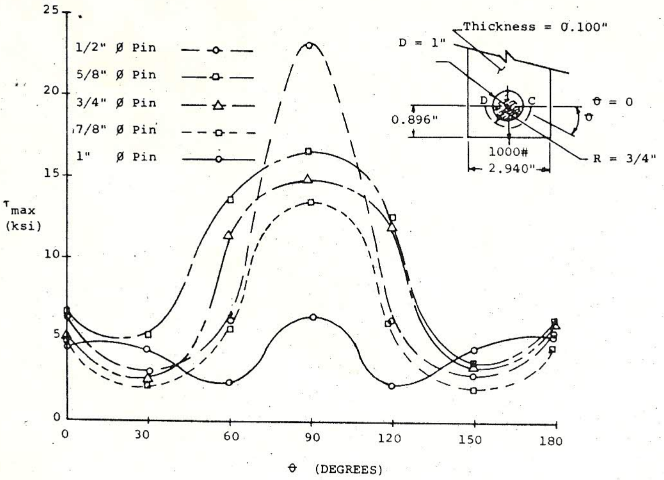

Fig. 2. Experimental results for maximum shear stress distribution on semicircular section C-D for 1000 kips pin load

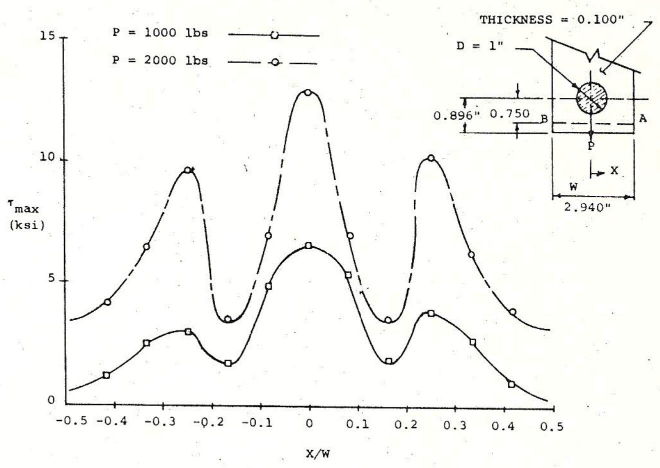

Fig. 4. Experimental results for maximum shear stress distribution on horizontal section A-B for 1-in. diam. pin

diameter snug fit pin to a ½-in. diameter pin reduced the ultimate capacity 31 percent and the elastic capacity 72 percent. The effect of clearance on elastic stress concentration was greatest for initial clearances. For example, going from 1-in. diameter sung fit pins to ⅞-in. diameter pins reduced the elastic capacity 52 percent. Further reductions in bolt sizes by ⅛-in. increments caused additional reductions in capacity of approximately 6.6 percent for each incremental pin size reduction.

Figures 2 and 3 are included to provide insight into the lug elastic shear stress distribution. Multiples of these shear stresses are produced by equal multiples of applied pin load as shown in Fig. 4.

Figure 5 graphs the effect of lug width on lug capacity. Only a graph of the lugs' ultimate capacity is given, since the variation of the elastic results was not considered to be of enough significance to be conclusive. As shown, lug width had a measurable effect on lug capacity. The ratio of lug width to pin hole diameter (W/D) was plotted as one parameter versus the lug capacity. The ultimate capacity of the lug was approaching a constant value at a W/D ratio of 4.5 and was felt to be sufficiently close to its infinite width solution. The narrowest lug tested had a width of 2½ times the pin hole diameter and had an ultimate capacity of approximately 90 percent of the widest lug tested.

Failure of the lugs with push fit pins was along two planes parallel to the axis of loading. The innermost point on each plane is located by a radius extending from the pin hole center at a 55 to 60 degree angle to the line of action. Pin clearance reduces this angle considerably.

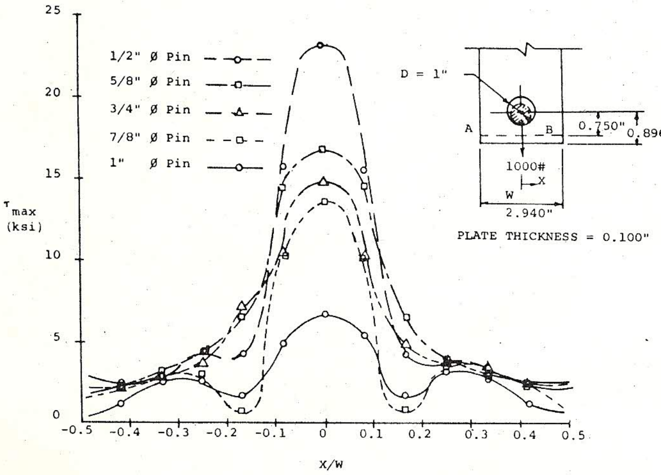

Fig. 3. Experimental results for maximum shear stress distribution on horizontal section A-B for 1000 kips pin load

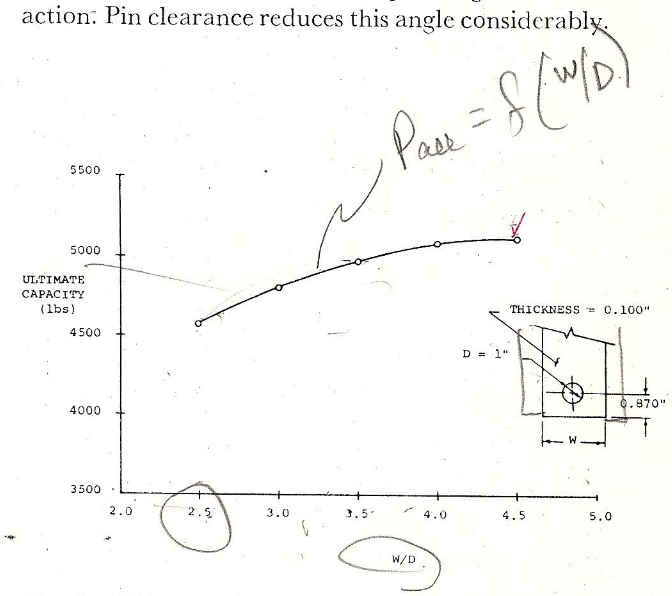

Fig. 5. Ultimate capacity of mild steel lugs for varying width-to-hole-diameter ratios

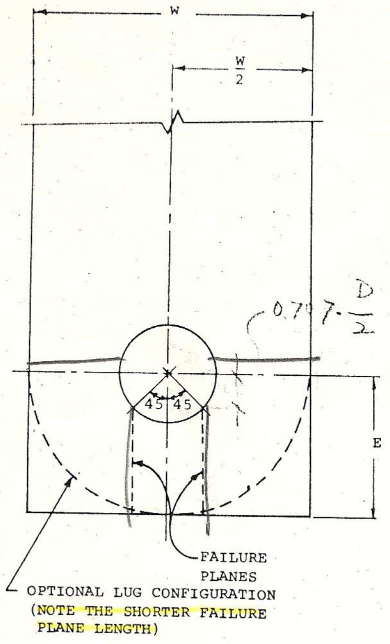

Fig. 6. Lug configurations and shear failure planes

RECOMMENDATIONS ON LUG CAPACITY

The following recommended analysis procedure is based on the lug and pin configuration with a reduction factor to account for pin clearance and an amplification factor to account for width variations. This formulation is restricted to situations where the lug width W divided by 2 is greater than or equal to the distance E (see Fig. 6). Standard checks for pin shear failure and lug bearing or tension failure should also be made.

To determine the allowable lug capacity, a failure plane angle of 45 degrees is used, since it is a trigonometrically easy angle to work with and is only slightly conservative. The failure plane areas are computed and multiplied by the shear stress capacity of the material. Next, the appropriate factor of safety is selected and finally, from the graphs provided in Figs. 7 and 8, a lug width amplification factor and a pin clearance reduction factor are selected.

If the shear stress capacity of the lug material is unstated, 60 percent of the ultimate tensile stress can be used for low strength carbon steels. Yield theories, namely the generally accepted Huber-von Mises-Hencky "energy of distortion" theory,[12] sets the shear yield point equal to the tension yield divided by $\sqrt{3}$. Also considering the fact that steel specifications have permitted shear stresses to be two-thirds of the values permitted for tension for the past fifty years,[13] 60 percent of the ultimate tensile stress seems reasonable.

The factor of safety should be a function of the loading cycle and the absolute degree of certainty attached to

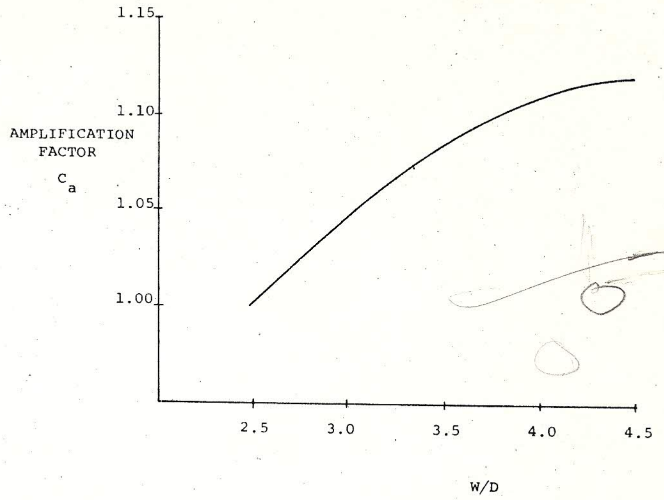

Fig. 7. Width variation amplification factor

the item being handled. As noted from the effect of pin clearance on the lug elastic strength, care should be exercised to minimize lug clearance, particularly in cyclic loading situations, to avoid fatigue failures. Factors of safety for handling devices are often 5 or even larger. A minimum factor of safety of 3 is recommended with larger values justifiable as stated above.

In Fig. 8, a graph of the elastic strength reduction and the ultimate strength reduction are provided, as well as a weighted average recommended for normal applications. It was felt that the ultimate capacity curve should be used. However, by using the minimum factor of safety and maximum clearance, allowable pin loads were very nearly equal to the elastic capacity. For this reason, the elastic strength reduction curve was averaged

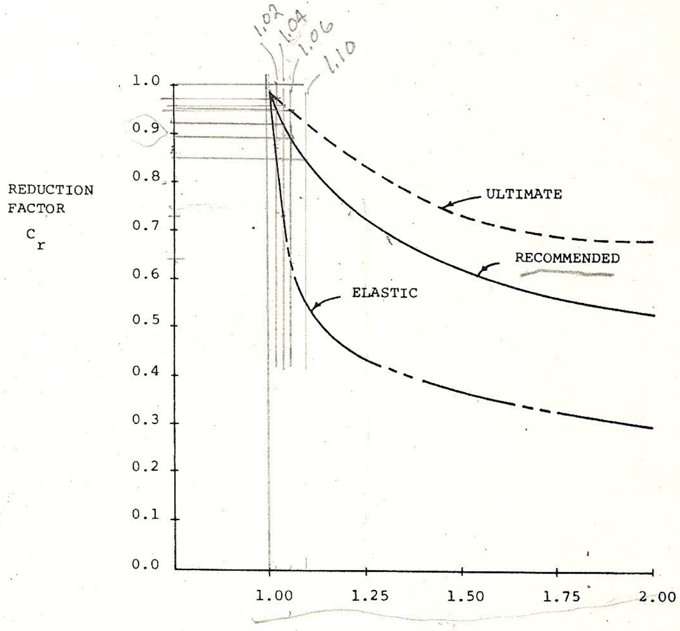

Fig. 8. Pin clearance reduction factor

with twice the ultimate strength reduction curve, providing what was felt to be the best clearance reduction curve. The recommended curve is not to preclude the use of either the elastic strength or ultimate strength reduction curves when appropriate.

In equation form, the allowable pin load P_a is:

$$P_a = \frac{AV_uC_aC_r}{FS} \qquad (1)$$

Where:

A = area of shear failure planes
V_u = ultimate shearing stress
C_a = width variation amplification factor
C_r = pin clearance reduction factor
FS = factor of safety

The area can be expressed in terms of the end distance E, the pin hole diameter D, and the thickness of the lug t as:

$$A = (2E - 0.707D)t \qquad (2)$$

The width variation amplification factor can conservatively be taken as 1. Substituting these values in Eq. (1), the allowable pin load becomes:

$$P_a = \frac{(2E - 0.707D)tV_uC_r}{FS} \qquad (3)$$

For application to semicircular ended lugs, as per the optional configuration shown in Fig. 6, the area term changes.

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$$P_{all} = \frac{(2E - .7D)\,t\,F_y}{2\sqrt{3}}$$

gives F.S. = 1.5 on yield if $C_r = 0.75$

$V_u = F_y/\sqrt{3}$