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Utility Technical Support, 71620

From: Originating Organization

Utility Technical Support, 71620

ETN-94-0165

Attached are the design calculations for the strong-back safety latch. Please distribute copies per block #17.

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<th>(A)</th>
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Approval Designator (F): GN DeSantis

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OFFICIAL RELEASE
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DATE MAR 06 1995
STRONG-BACK SAFETY LATCH ANALYSIS

October 1994

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Facility System Engineering

CHECKED BY: J.A. Tuck, Mechanical Engineer
Facility System Engineering

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TABLE OF CONTENT

1.0 PURPOSE .......................................................... 2
2.0 SCOPE .............................................................. 2
3.0 DISCUSSION ......................................................... 2
4.0 RESULTS ............................................................ 3
5.0 CONCLUSION ......................................................... 4
6.0 REFERENCES ......................................................... 4
   6.1 Drawings ......................................................... 4
   6.2 Literature ....................................................... 4
7.0 DESIGN CALCULATION ............................................ 5
   7.1 List of Symbols .................................................. 7
   7.2 Static Analysis ................................................ 8
7.3 Dynamic Analysis .................................................. 16

APPENDIX A
TABULATED STRESS .................................................. 21

APPENDIX B
WELDING PROCEDURE ............................................... 22
1.0 PURPOSE

To resolve the following safety issues and concerns with the strong-back.

1. Inadequate clamping between the strong-back and pump.
2. Strong-Back support legs has no locking device.
3. The crane might pull the stud-bolts out of the concrete in shop.
4. The crane might lift the flatbed trailer.

2.0 SCOPE

To resolve the above safety items by designing a safety latch, pivot brace and a rear leg stabilizer lock.

3.0 DISCUSSION

The following calculation decides the integrity of the safety latch that will hold the strong-back to the pump during lifting. The safety latch will be welded to the strong-back, and will latch to a 1 1/2" diameter cantilever rod welded to the pump baseplate (see figure 2, section 7.0).

The calculations assure the materials selected are of the appropriate strength to withstand the weight of the strong-back (1,634 lbs) during lifting. The weight of the pump will be carried by the crane at the lifting bails, which will be attached to the pump. A safety factor of (3) has been used as part of the design criteria, per the Hanford Site hoisting and Rigging Manual (DOE-RL 1993).

The safety latch has been designed so that an operator may release the jaws by pulling on a wire cable leading up to the latch, freeing the pump (see figure 2, section 7.0).

The pivot brace is designed to be mounted onto the flatbed trailer, and the floor of the shop. The Brace will allow the strong-back to pivot during lifting, and to move vertical, eliminating the issue of lifting the trailer or pulling the bolt out of the concrete floor (see section 6.0 for reference drawing).

The rear leg stabilizer lock will constrain the strong-back rear support by eliminating any forward, or lateral movements, allowing the operator to place the safety plate over the legs, and be bolted (see section 6.0 for reference drawing).
The current design of the strong-back holds the pump in place by two pins on the yoke and two friction clamps at the bottom of the shaft. The safety latch is designed to take the load (1,634 lbs) of the strong-back in the upright position if slippage or failure of the pins occurs. Therefore, all load points are based on a load of 1,634 lbs, being placed on the safety latch in the vertical position.

The safety latch will be welded to a piece of structural tubing which will be welded to the strong-back.

Each pump baseplate will be modified with a 1-1/2" diameter cantilever rod, approximately 8 inches long. The cantilever rod will be welded to the pump base. This will be the second link to the safety latch.

4.0 RESULTS

The static analysis (see section 7.2) was based on a using a safety factor of 3, it was found all materials selected for the safety latch components will support the strong-back.

By comparing all the calculated stresses to the tabulated yield points for each of the components and their welds (see Appendix A), the component first to yield would be the shoulder bolt, if a continuously increasing load were applied. The shoulder bolt is fabricated from ASTM A325, which yields at 92 ksi. The calculated stress for the shoulder bolt is approximately 25 ksi. This stress is based on the shoulder bolt taking the full static load.

The dynamic analysis (see section 7.3) is a conservative approach based on the strong-back free falling 1/4 inch and causing the safety latch to come in contact with the cantilever rod, thereby shock loading the safety latch components. As stated above the first component to yield is the shoulder bolt. Assuming the stress-strain curve for the shoulder bolt is elastic and perfectly plastic, and assuming that all the kinetic energy is absorbed by the shoulder bolt, the analysis shows that the shoulder bolt can absorb 6.7 in-lb in the elastic range, and 6,261 in-lb in the plastic range using an ultimate strain of 0.11 in/in. The kinetic energy that needs to be absorbed is only 414 in-lb and is far less then the ultimate energy which can be absorbed by the shoulder bolt. Therefore: based on this conservative approach, if all of the kinetic energy is applied to the shoulder bolt, fracture would not occur. It should be noted that the kinetic energy is actually absorbed by all the safety latch components and not just the shoulder bolt. Since the shoulder bolt is the first component to yield (weakest component), once yielding is initiated, and assuming an elastic perfectly plastic stress strain curve for the bolt, no additional load (or stress) will be applied to the remaining components. The shoulder bolt will continue to deform into the plastic range until all the kinetic energy is dissipated by deformation of the shoulder bolt. This analysis shows that the shoulder bolt will not fracture (reach the ultimate strain) if the strong-back falls 1/4 inch.
5.0 CONCLUSION

The purpose of the analysis is to resolve the safety issues dealing with the strong-back listed above in section 1.0, by analyzing the static and dynamic loading to the safety latch during lifting. The static and dynamic analysis shows that the safety latch will have the integrity to hold the strong-back to the pump if the friction clamps were to fail and the pump was to become free from the strong-back. This analysis proves the safety latch will function as designed, and will meet the requirements of the Lifting and Rigging Manual for under the hook lifting for static loading. This analysis also shows the safety latch is also capable of sustaining shock loading induced by the strong-back falling 1/4 inch.

6.0 REFERENCES

6.1 Drawings

<table>
<thead>
<tr>
<th>Description</th>
<th>Reference</th>
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<tbody>
<tr>
<td>Safety Latch</td>
<td>H-2-85424 Rev. 0 sht’s 1 thru 3</td>
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<td>Support &amp; Stabilizer</td>
<td>H-2-85432 Rev. 0 sht’s 1 thru 3</td>
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<tr>
<td>Latch Installation</td>
<td>H-2-85439 Rev. 0 sht 1 thru 2</td>
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<tr>
<td>Strong-Back</td>
<td>H-2-99132 Rev. 0 sht’s 1 thru 4</td>
</tr>
<tr>
<td>Pump Run-in</td>
<td>H-2-71840 Rev. 2 sht’s 1 thru 2</td>
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6.2 Literature


7.0 DESIGN CALCULATION

Figure 1
Safety Latch Assembly
Figure 2
Safety Latch Weldment
7.1 List of Symbols

- $A$ area of cross section, in$^2$.
- $c_h$ horizontal distance from c.g. to weld, in.
- $c_v$ vertical distance from c.g. to weld, in.
- $f$ force per inch for welds treated as lines, lb/in.
- $f_a$ allowable force per inch for welds treated as lines, lb/in.
- $f_b$ bending force per inch for welds treated as lines, lb/in.
- $f_s$ shear force, force per inch for welds treated as lines, lb/in.
- $f_r$ vertical twisting force per inch for welds treated as lines, lb/in.
- $I_c$ moment of interia, in$^4$.
- $k$ stress factor for curved beams.
- $L_w$ length of weld, in.
- $M$ bending moment, in-lb.
- $P$ Force, lb.
- $U$ maximum energy, in-lb
- $U_u$ ultimate energy, in-lb
- $w$ weld leg length, in.
- $Y_s$ yield strength, psi.
- $Y_u$ ultimate stress, psi.
- $Z$ section modulus, in$^3$.
- $Z_w$ section modulus for welds treated as lines, in$^2$.
- $\sigma_a$ allowable stress, psi.
7.2 Static Analysis


\[ \sigma_b \text{ bearing stress, psi.} \]
\[ \sigma_b \text{ bending stress, psi.} \]
\[ \sigma_w \text{ weld stress, psi.} \]
\[ \delta \text{ deflection, in.} \]
\[ \tau \text{ shear stress, psi.} \]
\[ \tau_t \text{ shear stress due to tension, psi.} \]

\[
\begin{align*}
8\times8\times3/16\text{ sq. channel} &= 19.63\text{ lb/ft} \\
48' \times 19.63 &= 942\text{ lb} \\
3'' \text{ pipe sch 80 cs} &= 10.25\text{ lb/ft} \\
35' \times 10.25 &= 358\text{ lb} \\
8\times4\times3/13\text{ sq channel} &= 14.53\text{ lb/ft} \\
23' \times 14.53 &= 334\text{ lb} \\
\end{align*}
\]

Total weight of Strong-Back = 1,634 lb

\[
\begin{align*}
6'' \text{ pipe sch 40 pipe, cs} &= 19.00\text{ lb/ft} \\
43' \times 19 &= 817\text{ lb} \\
\end{align*}
\]

Pump motor = 500lb

1 1/2'' Pump shaft = 6lb/ft
43' \times 6 = 238\text{lb}

Total weight of pump = 1,575lb

Total combined weight = 3,210lb

[2] Calculate the bearing, shear and bending stress due to a load of 1,634 lbs on one jaw.

Material: 1/2'' plate, AISI 4130, Ys = 92,000 psi
Allowable stress \( 92,000/3 = 30,666 \text{ psi} \)
[2a] Calc. the surface area to be in contact with cantilever rod.

\[
surface = 0.75 \times 2\pi \times \frac{46^\circ}{360^\circ} = 0.60\text{ inch}
\]


\[
\sigma_b = \frac{1634}{(0.60 \times 0.50)} = 5,446\text{ lb/inch}^2
\]

[2c] Calc. the shear stress in the plate due to the upward load.

\[
\tau = \frac{1634}{0.5 \times 1.50} = 2,178\text{ lb/inch}^2
\]

[2d] Calc. the stress due to bending in one jaw.

Assumption: calculate the bending stress for a straight cantilever beam, and multiply it by the stress factor \((k)\) for a curved beam (Reference: Machinery's Handbook, 24 Edition, pg. 242). From pg. 242, for a rectangular cross section:

\[
r/c = 1.75/0.75 = 2.3 \text{ therefore}
\]

\[
k \text{ inside} = 1.52,
\]

\[
k \text{ outside} = 0.73
\]

\[
M = 1634 \times 1.75 = 2,859\text{ lb-inch}
\]

\[
I_c = \frac{0.5 \times 1.5^3}{12} = 0.141\text{ inch}^4
\]

\[
\sigma_b = \frac{2859 \times 0.75}{0.141} = 15,250\text{ lb/inch}^2
\]

\[
\sigma_{b \text{ inside}} = 15,225 \times 1.52 = 23,181\text{ lb/inch}^2
\]

\[
\sigma_{b \text{ inside}} < \sigma_a \checkmark
\]
The following calculations analyze the shear, bending and deflection of the shoulder bolt. The shoulder bolt is assumed to be simply supported, loaded at the center.

Material: 3/4" ASTM A325 type 3, Ys = 92,000 psi. Allowable = 92,000/3 = 30,666 psi.

[3a] Calc. shear stress due to a load of 1,634 lbs, by two clips.

$$\tau = \frac{1634}{(2\pi \times 0.375^2)} = 1,850 lb/inch^2$$

[3b] Calc. the maximum bending stress in the shoulder bolt, assuming a simple supported beam.

$$M_{max} = \frac{wl}{4}$$
$$M_{max} = \frac{2.5 \times 1634}{4} = 1,021 lb-inch$$

[3c] Calc. the section modulus for the shoulder bolt.

$$Z = \frac{\pi \times 0.75^3}{32} = 0.041 inch^3$$

[3d] Calc. the bending stress.

$$\sigma_b = \frac{M_{max}}{Z}$$
$$\sigma_b = \frac{1021}{0.041} = 24,664 lb/inch^2$$

$$\sigma_b < \sigma_a \checkmark$$

[3e] Calc. deflection in the shoulder bolt (information only).

$$\delta = \frac{1021 \times 2.5^3}{38 \times 30E6 \times 0.016} = 0.001 inch$$

[4] The following calculations, analyze the stresses from the applied load, for one side plate of the safety latch housing.
Material: 1/4" plate, ASTM A36, Ys=36,000 psi
Allowable: 36,000/3 = 12,000 psi

3/8" bolts, ASTM A307, Ys=36,000 psi
Allowable: 36,000/3 = 12,000 psi


\( \sigma_B = \frac{1634}{2 \times 0.25 \times 0.75} = 4,357 \text{ lb/inch}^2 \)

\( \sigma_B < \sigma_s \checkmark \)

[4b] Calc. shear stress in the 1/4 plate above the 3/4" bolt hole.

\( \tau = \frac{1634}{2 \times 0.25 \times 0.875} = 3,734 \text{ lb/inch}^2 \)

\( \tau < \tau_s \checkmark \)

[4c] Calc. shear stress in 1/4" plate, above the 3/8" bolt hole.

\( \tau = \frac{1634}{4 \times 0.25 \times 0.50} = 3,268 \text{ lb/inch}^2 \)

\( \tau < \tau_s \checkmark \)


\( \tau = \frac{408}{\pi \times 0.187^2} = 3,695 \text{ lb/inch}^2 \)

\( \tau < \tau_s \checkmark \)

[4e] Calc. bearing stress in one 3/8" diameter hole.

\( \sigma_B = \frac{408}{0.375 \times 0.25} = 4,352 \text{ lb/inch}^2 \)

\( \sigma_B < \sigma_s \checkmark \)
[4f] Calc. tension in one angle support.

\[ \tau_i = \frac{P}{A_n} \]
\[ A_n = t(\omega_g - N\omega_h) \]
\[ t = \text{plate thickness.} \]
\[ \omega_g = \text{gross width of plate.} \]
\[ \omega_h = \text{diameter of hole} \]
\[ N = \text{number of holes in the section being considered} \]

\[ \tau_i = \frac{408}{0.25[0.75-(0.375+0.062)]} = 5,222 \text{lb/inch}^2 \]

\[ \tau_i < \tau_s \checkmark \]

[4g] Calc. the transverse force induced in the fillet welds of the housing.

\[ f = \frac{P}{L} \]

\[ f = \frac{1634}{(2*9.75+2*2.25)} = 68 \text{lb/inch} \]

[4h] Calc. the required weld size using a 6016 welding rod.
The allowable shear stress for a 6016 rod is 18,000 psi.

\[ w = \frac{P}{(0.707*\sigma_{aw})} \]

\[ w = \frac{1634}{(0.707*18000)} \]

\[ w = 0.128 \text{ inch} \]

design weld size 3/16 (0.187) > 0.128 \checkmark

[4i] Calc stress induced in the weld.
\[ \sigma_w = \frac{68}{0.187 \times 0.707} \]

\[ \sigma_w = 514 \text{ psi} \]

[5] Calculate the bending in the 1 1/2" diameter lever rod, welded on the pump base plate.

Material: 1 1/2" diameter rod, AISI 4130, Ys=97,000 psi
Allowable = 97,000/3 = 32,333 psi
Assumption: The moment will be taken 2.375" in from the right end of the lever rod, and not at the welded end, since the support bar relieves the stresses at the weld.

Calc. the maximum moment
\[ M_{\text{max}} = 1634 \times 2.375 = 3,880 \text{ lb-in} \]

Calc. the section modulus.
\[ Z = \frac{\pi \times 1.50^3}{32} = 0.331 \text{ inch}^3 \]

Calc. the bending stress.
\[ \sigma_b = \frac{3880}{0.331} = 11,724 \text{ lb/inch}^2 \]

\[ \sigma_b < \sigma_a \checkmark \]

[6] Calculate the section modulus of the weld, for the cantilever rod. Reference pg. 276 of Salmon and Johnson, Steel Structures Design and Behavior, for the section modulus. The weld is assumed to be a straight line.
\[ Z_w = 1 \times 3 + \frac{3^2}{3} = 6.0 \text{ inch}^2 \]

[6a] Calc. force in the weld due to bending.
\[ f_b = \frac{4902}{6.0} = 817 \text{ lb/inch} \]

The allowable weld shear is the allowable of the base metal 12,000 psi.

\[ w = \frac{817}{0.707 \times 12000} = 0.096 \text{ inch} \]

The design weld is 5/16 (0.313) > 0.096 ✓

[6d] Calc. the induced stress in the weld.

\[ \sigma_w = \frac{817}{0.313 \times 0.707} \]

\[ \sigma_w = 3,691 \text{ psi} \]

[7] Calculate the required spring constant, used for opening and closing the jaws.

Deflection \( \delta \) = 1.60 (open jaw)

O.D. of spring = 0.187

Center line D = 0.375 - 0.0418 = 0.327

Gage d (19 Ga) = 0.0418

Modulus G = 11.5E6 psi

Number of coils N = 10

Spring Const. \( K \) = ?

\[ P = \frac{\delta Gd^4}{8D^3N} = \frac{1.6 \times 11.5E6 \times 0.0418^4}{8 \times 0.327^3 \times 10} \]

\[ P = 20 \text{ lb} \]

For two springs in parallel

\[ P_T = 2 \times 20 = 40 \text{ lbs} \]

\[ K = \frac{P}{\delta} = \frac{20}{1.6} = 12.50 \text{ lbs/in} \]

[8] The following calculations analyzes the induced stresses in the safety latch housing support, which will be welded to the back side of the strong-back.

Material 8x8x3/16x5" lg. steel tube ASTM A36, \( Y_s = 36,000 \)

Section Modulus \( Z = 14.6 \text{ in}^3 \) Reference pg I-95 of the AISC.

Allowable 36,000/3 = 12,000 psi
Calc. the maximum bending.

\[ M_{\text{max}} = 1634 \times 7 \]

\[ M_{\text{max}} = 11,438 \text{ lb-inch} \]

Calc. the bending stress

\[ \sigma_b = \frac{11438}{14.6} = 783 \text{ lbs/inch}^2 \]

\[ \sigma_b < \sigma_a \checkmark \]

[8a] Calc. the section modulus for the weld.
Reference pg. 276 of Salmon and Johnson, Steel Structures Design and Behavior, for the section modulus.

\[ Z_w = 8 \times 8 + \frac{8^2}{3} = 85.3 \text{ inch}^2 \]

Calc. force due to bending in the weld.

\[ f_B = \frac{11438}{85.3} = 134 \text{ lb/inch} \]

Calc. the required weld size using a 6016 rod.
the allowable shear stress is 18,000 psi

\[ w = \frac{134}{0.707 \times 18000} = 0.011 \text{ inch} \]

Design weld is 3/16 (0.188) \( > 0.011 \checkmark \)

[8b] Calc. the induced stress in the weld.

\[ \sigma_w = \frac{134}{0.188 + 0.707} \]

\[ \sigma_w = 1,008 \text{ psi} \]
7.3 Dynamic Analysis

[1] The conditions for analyzing the dynamic loading on the safety latch and the cantilever rod are:

1. The strong-back must be lifted off the trailer or shop floor.
2. The friction clamps must fail.

If both conditions occur, the strong-back safety latch (which is attached to the strong-back) would drop onto the cantilever rod (which is attached to the pump) causing shock loading.

The following calculation determines the energy absorbed first in the elastic then in the plastic range of each component (if required).

[1a] Calculate Kinetic energy if the strong-back were to fall 0.25 inches.

\[ V = \sqrt{2 \times 387 \times 0.25} = 14 \text{ inch/s} \]

\[ KE = \frac{1634 \times 14^2}{2 \times 387} = 414 \text{ inch-lb} \]

[1b] Calculate maximum energy which can be absorb in one jaw of the safety latch without deforming in the elastic range, using AISI 4130 (reference the Design of Weldments, page 3.1-5 Table 3) is.

\[ U = \frac{Y_s^2 \times I \times L}{6 \times E \times c^2} \]

Yield stress \( (Y_s) = 97,000 \text{ psi} \)
Distance \( (L) \) from the center line of the jaw to the load = 1.75 in.
\( I = .141 \text{ in}^4 \)
\( E = 30E6 \text{ psi} \)
\( c = 0.75 \text{ in} \)

\[ U = \frac{97000^2 \times .141 \times 1.75}{6 \times 30E6 \times 0.75^2} = 23 \text{ inch-lb} \]

23 in-lb < KE, which exceeds the elastic region. Therefore, calculate the ultimate energy which can be absorbed:
\[ U_u = \frac{Y_u + Y_y}{2} \epsilon_u A L \]

From the Stress - Strain curves (reference ASM, Atlas of Stress Strain Curves, pg 222).
Elongation \( \epsilon_u = 0.11 \text{ in/in} \) at a strain rate of 60 in/in/min.
Cross section area \( (A) \) of the jaw = \((0.5\times1.5)=0.75 \text{ in}^2 \)
Distance \( (L) \) from the center line of the jaw to the load = 1.75 in.
Ultimate stress \( (Y_u)=115,000 \text{ psi.} \)

\[ U_u = \frac{97000 + 115000}{2} \times 0.11 \times 1.75 \times 0.75 = 15,303 \text{ inch-lb} \]

\( 15,303 \text{ in-lb} > KE \checkmark \)

[2] Calculate the maximum energy which can be absorbed in the shoulder bolt. Utilizing the same two formulas which were used for the Jaw.

\( I = 0.016 \text{ in}^4 \)
\( L = 1.25 \text{ in} \)
\( E = 30E6 \text{ psi} \)
\( c = 0.375 \text{ in} \)

\( U = 6.7 \text{ in-lb} < KE, \text{ exceeds the elastic range.} \)

[2a] Calculate the ultimate energy which can be absorbed.
Elongation \( \epsilon_u = 0.11 \text{ in/in} \).
Cross section area \( (A) \) of the jaw = \((\pi\times0.375^2)=0.44 \text{ in}^2 \)
Distance \( (L) = 1.25 \text{ in} \).
Ultimate stress \( (Y_u)=115,000 \text{ psi.} \)
Yield stress \( (Y_y)=92,000 \text{ psi.} \)

\[ U_u = \frac{92000 + 115000}{2} \times 0.11 \times 0.44 \times 1.25 = 6,261 \text{ inch-lb} \]

\( U_u = 6,261 \text{ in-lb} > KE \checkmark \)

[3] Calculate the maximum energy which can be absorbed in the cantilever rod utilizing the same two formulas.

\( I = 0.25 \text{ in}^4 \)
\( L = 3 \text{ in} \)
E=30E6 psi
c= 0.75 in
Yield stress \(Y_u=97,000 \text{ psi}\)

\[U=70.0 \text{ in-lb} < KE\]

[3b] Calculate the ultimate energy which can be absorbed.
Elongation \(\varepsilon_u=0.11 \text{ in/in.}\)
Cross section area \(A=(\pi*0.75^2)=1.76 \text{ in}^2\)
Distance \(L=3.0 \text{ in.}\)
Ultimate stress \(Y_u=115,000 \text{ psi}\).

\[U_u=61,546 \text{ in-lb} > KE\]

[4] Calculate the maximum energy which can be absorbed in one plate of the housing.

I= 122 \text{ in}^4
L= 4.875 \text{ in}
E=30E6 psi
c= 2.66 in
Yield stress \(Y_u=36,000 \text{ psi}\)

\[U= 605 \text{ in-lb} > KE\]

[5] Calculate the maximum energy which can be absorbed in the housing support.

I= 59.6 \text{ in}^4
L= 7.0 \text{ in}
E=30E6 psi
c= 4 in
Yield stress \(Y_u=36,000 \text{ psi}\)

\[U= 187.7 \text{ in-lb} < KE\]

[5b] Calculate the ultimate energy which can be absorbed.
Elongation \(\varepsilon_u=0.35 \text{ in/in.}\)
Cross section area \(A=5.86 \text{ in}^2\)
Distance \(L=7.0 \text{ in.}\)
Ultimate stress \(Y_u=60,000 \text{ psi}\).

\[U_u= 689,000 \text{ in-lb} > KE\]
If shock loading were to occur, possible deformation could result, but failure in the material would not since the ultimate energy which could be absorbed is greater the kinetic energy (reference ASM, Atlas of Stress Strain Curves).

[6] Calculate the required weld size (w) due to shock loading in each of the components.

Calc. safety factor from the applied load.

\[ sf = 1634 \times 3 = 4902 \text{ lb} \]

Calc. the moment for the cantilever rod.

\[ M = 4902 \times 3 = 14,706 \text{ in-lb} \]
Calculated bending stress due to impact.

Section modulus for the weld = 6 inch\(^2\). From section 7.1 step [6].

\[ \sigma_b = \frac{4902}{6} = 2,451 \text{ lb/inch} \]

Calc. the shear stress due to shock loading.

Length of weld = 8 inch.

\[ \tau = \frac{4902}{8} = 613 \text{ lb/inch} \]

Calc. the total stress.

\[ \sigma_b + \tau = 3064 \text{ lb/inch} \]

Calc. the required weld size.

welding rod 8016-B2
The allowable weld shear stress for the base metal is 14,400 psi

\[ w = \frac{3064}{0.707 \times 14400} = 0.30 \text{ inch} \]

Design weld size 5/16" (0.312) > 0.30 ✔

[7] Calculate the required weld size for the safety latch housing.

Calc. transverse stress on weld.

Length of weld 22 inch.
\[
\tau = \frac{14706}{22} = 668 \text{ lb/inch}
\]

Calc. required weld size.
welding rod 8016
The allowable shear stress for a 6016 rod is 18,000 psi.
\[
w = \frac{668}{0.707 \times 18000} = 0.05 \text{ inch}
\]

Design weld size 3/16" (0.187) > 0.05 ✔

[8] Calculate the required weld in the safety latch housing support.

Calc. the moment.
\[M = 4902 \times 7 = 34,321 \text{ inch-lb}\]

Calc. the bending stress due to impact.
Section modulus for the weld = 85.3 inch². From section 7.1 step [8]
\[
\sigma_b = \frac{34321}{85.3} = 402 \text{ lb/inch}
\]

Calc. the shear stress due to impact.
Length of weld = 32 inch.
\[
\tau = \frac{34321}{32} = 1,072 \text{ lb/inch}
\]

Calc. the total stress due to impact.
\[
\sigma_b + \tau = 1,474 \text{ lb/inch}
\]

Calc. required weld size.
welding rod 6016
The allowable weld shear stress for a 6016 rod = 18,000 psi.
\[
w = \frac{1474}{0.707 \times 18000} = 0.116 \text{ inch}
\]

Design weld size 3/16 inch.
3/16 (0.187) > 0.116 ✔
APPENDIX A
TABULATED STRESS

<table>
<thead>
<tr>
<th>Stress psi</th>
<th>Jaw, AISI 4130, Ys=92,000 psi</th>
<th>Shoulder Bolt, ASTM A325, Ys=92,000 psi</th>
<th>Side Plate, ASTM A36, Ys=36,000 psi</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\sigma_b$</td>
<td>23,181</td>
<td>24,664</td>
<td>-</td>
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<tr>
<td>$\tau$</td>
<td>2,178</td>
<td>1,021</td>
<td>3,268</td>
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<td>$\sigma_w$</td>
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<td></td>
<td>Lever Rod</td>
<td>Housing Support</td>
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<td>783</td>
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<tr>
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<tr>
<td>$\sigma_w$</td>
<td>3,691</td>
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<td>385</td>
</tr>
</tbody>
</table>

* The stress shown represents the stress in one jaw. The actual safety latch consist of two jaw, therefore the actual stress is one half the stress shown.
APPENDIX B
WELDING PROCEDURE