New Original ²⁶⁷⁴
Old Original Destroyed

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3	1	JA Tuck S2-24	<i>[Signature]</i>	14 Feb 1995					

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RELEASE AUTHORIZATION

Document Number: WHC-SD-WM-DA-158, REV 0

Document Title: STRONG-BACK SAFETY LATCH

Release Date: 3/6/95

**This document was reviewed following the
procedures described in WHC-CM-3-4 and is:**

APPROVED FOR PUBLIC RELEASE

WHC Information Release Administration Specialist:

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March 6, 1995

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7. Abstract This Design Alaysis provides stress calculation for selecting the required materials needed for the Strong-Back Safety Latch.	
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STRONG-BACK SAFETY LATCH ANALYSIS

October 1994

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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

Safety Latch	H-2-85424 Rev. 0 sht's 1 thru 3
Support & Stabilizer	H-2-85432 Rev. 0 sht's 1 thru 3
Latch Installation	H-2-85439 Rev. 0 sht 1 thru 2
Strong-Back	H-2-99132 Rev. 0 sht's 1 thru 4
Pump Run-in	H-2-71840 Rev. 2 sht's 1 thru 2

6.2 Literature

AISC, 1989, *Manual of Steel Construction - Allowable Stress Design*, Ninth Edition, American Institute of Steel Construction, Chicago.

C.G. Salmon, and J.E. Johnson, *Steel Structures Design and Behavior*, Third Edition, Harper Collins, New York.

DOE-RL, 1993, *Hanford Site Hoisting and Rigging Manual*, DOE-RL-92-36, U.S. Department of Energy, Richland Field Office, Richland, Wa.

Machinery's Handbook, Twenty-First Edition.

Omer W. Blodgett, 1976, *Design of Weldments*. Eighth Printing, Lincoln Arc Welding foundation, Cleveland, Ohio.

Roark, R.J., and W.C. Young, 1989, *Roark's Formulas for Stress and Strain*, Sixth Edition, McGraw-Hill, New York.

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 _v	vertical twisting force per inch for welds treated as lines, lb/in.
I _c	moment of inertia, 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 .
σ _a	allowable stress, psi.

σ_B bearing stress, psi.
 σ_b bending stress, psi.
 σ_w weld stress, psi.
 δ deflection, in.
 τ shear stress, psi.
 τ_t shear stress due to tension, psi.

7.2 Static Analysis

[1] Determining the weight of the Strong-Back and Pump.

8x8x3/16 sq. channel = 19.63 lb/ft
48'x 19.63 = 942lb

3" pipe sch 80 cs = 10.25 lb/ft
35'x 10.25 = 358lb

8x4x3/13 sq channel = 14.53 lb/ft
23' x 14.53 = 334lb

Total weight of Strong-Back = 1,634 lb

6" pipe sch 40 pipe, cs = 19.00 lb/ft
43'x 19 = 817lb

Pump motor = 500lb

1 1/2" Pump shaft = 6lb/ft
43' x 6 = 258lb

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, $Y_s = 92,000$ psi
Allowable stress $92,000/3 = 30,666$ psi

[2a] Calc. the surface area to be in contact with cantilever rod.

$$surface = .75 * 2\pi * \frac{46^0}{360^0} = 0.60 inch$$

[2b] Calc. bearing stress.

$$\sigma_B = \frac{1634}{(0.60 * 0.50)} = 5,446 lb/inch^2$$

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

$$\tau = \frac{1634}{0.5 * 1.50} = 2,178 lb/inch^2$$

[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:

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

$$k \text{ inside} = 1.52,$$

$$k \text{ outside} = 0.73$$

$$M = 1634 * 1.75 = 2,859 \text{ lb-inch}$$

$$I_c = \frac{0.5 * 1.5^3}{12} = 0.141 inch^4$$

$$\sigma_b = \frac{2859 * 0.75}{0.141} = 15,250 lb/inch^2$$

$$\sigma_{b \text{ inside}} = 15,225 * 1.52 = 23,181 lb/inch^2$$

$$\sigma_{b \text{ inside}} < \sigma_a \checkmark$$

[3] 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, $Y_s = 92,000$ psi.
Allowable = $92,000/3 = 30,666$ psi.

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

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

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

$$M_{\max} = wl/4$$

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

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

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

[3d] Calc. the bending stress.

$$\sigma_b = M_{\max}/Z$$

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

$$\sigma_b < \sigma_s \checkmark$$

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

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

[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, $Y_s=36,000$ psi

Allowable: $36,000/3 = 12,000$ psi

3/8" bolts, ASTM A307, $Y_s=36,000$ psi

Allowable: $36,000/3 = 12,000$ psi

[4a] Calc. bearing stress in the 3/4" diameter hole.

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

$$\sigma_B < \sigma_s \checkmark$$

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

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

$$\tau < \tau_s \checkmark$$

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

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

$$\tau < \tau_s \checkmark$$

[4d] Calc. shear stress in 3/8 in bolt.

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

$$\tau < \tau_s \checkmark$$

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

$$\sigma_B = \frac{408}{0.375 \times 0.25} = 4,352 \text{ lb/inch}^2 \quad \sigma_B < \sigma_s \checkmark$$

[4f] Calc. tension in one angle support.

$$\tau_t = P/A_n$$

$$A_n = t(\omega_g - N\omega_h)$$

t = plate thickness.

ω_g = gross width of plate.

ω_h = diameter of hole

N = number of holes in the section being considered

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

$$\tau_t < \tau_a \checkmark$$

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

$$f = \frac{P}{L}$$

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

[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 ✓

[4i] Calc stress induced in the weld.

$$\sigma_w = \frac{68}{0.187 * 0.707}$$

$$\sigma_w = 514 \text{ psi}$$

[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, $Y_s = 97,000$ 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

$$M_{\max} = 1634 * 2.375 = 3,880 \text{ lb-inch}$$

Calc. the section modulus.

$$Z = \frac{\pi * 1.50^3}{32} = 0.331 \text{ inch}^3$$

Calc. the bending stress.

$$\sigma_b = \frac{3880}{0.331} = 11,724 \text{ lb/inch}^2$$

$$\sigma_b < \sigma_s \checkmark$$

[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.

$$Z_w = 1 * 3 + \frac{3^2}{3} = 6.0 \text{ inch}^2$$

[6a] Calc. force in the weld due to bending.

$$f_b = \frac{4902}{6.0} = 817 \text{ lb/inch}$$

[6b] Calc. the required weld size using a 8016-B2.

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

$$w = \frac{817}{0.707 * 12000} = 0.096 \text{ inch}$$

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

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

$$\sigma_w = \frac{817}{0.313 * 0.707}$$

$$\sigma_w = 3,691 \text{ psi}$$

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

Deflection δ = 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 = ?

$$P = \frac{\delta G d^4}{8 D^3 N} = \frac{1.6 * 11.5E6 * 0.0418^4}{8 * 0.327^3 * 10}$$

$$P = 20 \text{ lb}$$

For two springs in parallel

$$P_T = 2 * 20 = 40 \text{ lbs}$$

$$K = P/\delta = 20/1.6 = 12.50 \text{ lbs/in}$$

[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.

$$\text{Allowable } 36,000/3 = 12,000 \text{ psi}$$

Calc. the maximum bending.

$$M_{\max} = 1634 * 7$$

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

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.

$$Z_w = 8 * 8 + \frac{8^2}{3} = 85.3 \text{ inch}^2$$

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

$$w = \frac{134}{0.707 * 18000} = 0.011 \text{ inch}$$

Design weld is 3/16 (0.188) > 0.011 ✓

[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 attach 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.

$$V = \sqrt{2 * 387 * 0.25} = 14 \text{ inch/s}$$

$$KE = \frac{1634 * 14^2}{2 * 387} = 414 \text{ inch-lb}$$

[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.

$$U = \frac{Y_s^2 * I * L}{6 * E * c^2}$$

Yield stress (Y_s) = 97,000 psi

Distance (L) from the center line of the jaw to the load = 1.75 in.

$I = .141 \text{ in}^4$

$E = 30E6 \text{ psi}$

$c = 0.75 \text{ in}$

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

23 in-lb < KE, which exceeds the elastic region. Therefore, calculate the ultimate energy which can be absorbed:

$$U_u = \frac{Y_s + Y_u}{2} * \epsilon_u * A * L$$

From the Stress - Strain curves (reference ASM, Atlas of Stress Strain Curves, pg 222).

Elongation $\epsilon_u = 0.11$ in/in at a strain rate of 60 in/in/min.

Cross section area (A) of the jaw = $(0.5 * 1.5) = 0.75$ in²

Distance (L) from the center line of the jaw to the load = 1.75 in.

Ultimate stress (Y_u) = 115,000 psi.

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

15,303 in-lb > KE ✓

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

$I = 0.016$ in⁴

$L = 1.25$ in

$E = 30E6$ psi

$c = 0.375$ in

$U = 6.7$ in-lb < KE, exceeds the elastic range.

[2a] Calculate the ultimate energy which can be absorbed.

Elongation $\epsilon_u = 0.11$ in/in.

Cross section area (A) of the jaw = $(\pi * .375^2) = 0.44$ in²

Distance (L) = 1.25 in.

Ultimate stress (Y_u) = 115,000 psi.

Yield stress (Y_s) = 92,000 psi.

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

$U_u = 6,261$ in-lb > KE ✓

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

$I = 0.25$ in⁴

$L = 3$ in

$E=30E6$ psi
 $c=0.75$ in
Yield stress (Y_y)= $97,000$ psi

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

[3b] Calculate the ultimate energy which can be absorbed.

Elongation $\epsilon_u=0.11$ in/in.

Cross section area (A) = $(\pi \cdot 0.75^2)=1.76$ in²

Distance (L) = 3.0 in.

Ultimate stress (Y_u)= $115,000$ psi.

$$U_u=61,546 \text{ in-lb} > KE \checkmark$$

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

$I=122$ in⁴

$L=4.875$ in

$E=30E6$ psi

$c=2.66$ in

Yield stress (Y_y)= $36,000$ psi

$$U=605 \text{ in-lb} > KE \checkmark$$

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

$I=59.6$ in⁴

$L=7.0$ in

$E=30E6$ psi

$c=4$ in

Yield stress (Y_y)= $36,000$ psi

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

[5b] Calculate the ultimate energy which can be absorbed.

Elongation $\epsilon_u=0.35$ in/in.

Cross section area (A) = 5.86 in²

Distance (L) = 7.0 in.

Ultimate stress (Y_u)= $60,000$ psi.

$$U_u=689,000 \text{ in-lb} > KE \checkmark$$

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.

$$sf = 1634 * 3 = 4902 \text{ lb}$$

Calc. the moment for the cantilever rod.

$$M = 4902 * 3 = 14,706 \text{ in-lb}$$

Calc. the bending stress due to impact.

Section modulus for the weld = 6 inch². From section 7.1 step [6].

$$\sigma_b = \frac{14707}{6} = 2,451 \text{ lb/inch}$$

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

$$w = \frac{3064}{0.707 * 14400} = 0.30 \text{ inch}$$

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.

$$w = \frac{668}{0.707 * 18000} = 0.05 \text{ inch}$$

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

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

Calc. the moment.

$$M = 4902 * 7 = 34,321 \text{ inch-lb}$$

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.

$$w = \frac{1474}{0.707 * 18000} = 0.116 \text{ inch}$$

Design weld size 3/16 inch.

3/16 (0.187) > 0.116 ✓

**APPENDIX A
TABULATED STRESS**

Stress psi	*Jaw, AISI 4130 $Y_s=92,000$ psi	Shoulder Bolt, ASTM A325 $Y_s=92,000$ psi	Side Plate ASTM A36 $Y_s=36,000$ psi
σ_b	23,181	24,664	-
τ	2,178	1,021	3,268
σ_w			513
	Lever Rod		Housing Support
σ_b	14,809		783
τ			
σ_w	3,691		385

* 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