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ENGINEERING DATA TRANSMITTAL

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				STEEL WASTE DISPOSAL BOX							
				LIFTING BAIL DESIGN"							
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2	1	Design Authority	WM BOEHNKE	11/19/99	H1-11			INDEPENDENT REVIEWER			
2	1	Design Agent	BD FLANAGAN	11/22/99	H1-11	2	1	DL RILEY	DL RILEY	11/19/99	H1-11
2	1	Cog. Eng.	WM BOEHNKE	11/19/99	H1-11	3	6	BL SASSER	BL SASSER	11/19/99	L1-02
2	1	Cog. Mgr.	JA REEVES	11/23/99	H1-11	3	6	RL BOGART	RL BOGART	11-19-99	L1-02
2	1	QA	JA CHARBONEAU	11/19/99	H1-11	3	6	HE REW	HE REW	11/19/99	L1-06
2	1	Safety	T ROMANO	11/23/99	T4-05	2	1	TL ERICKSON	TL ERICKSON	11/23/99	L1-02
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ENGINEERING SAFETY EVALUATION FOR 22 TON STEEL DISPOSAL BOX LIFTING BAIL DESIGN

WM BOEHNKE

WASTE MANAGEMENT TECHNICAL SERVICES

Richland, WA 99352

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Analysis on the modified lifting bails for the 22 Ton Steel Waste Disposal Box.

Abstract:

The objective of this analysis is to design and analyze the lifting bail of the 22 Ton Steel Waste Disposal Box (SWDB). The new design takes the original lifting bail and adds a hinge allowing the top portion of the bail to fold over towards the lid. This modification only applies to the lifting bails and does not effect the attachment of the bails to the box, this analysis applies only to the lifting bails hinge design.

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Karen A. Noland 11/23/99
Release Approval Date

Release Stamp

Approved For Public Release

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Preparer: BD Flanagan <i>BD Flanagan</i>	Date: 9/22/99
Checker: DL Riley <i>DL Riley per Tekon</i>	Date: 9/22/99

1.0 OBJECTIVE

The objective of this analysis is to design and analyze the lifting bail of the 22 Ton Steel Waste Disposal Box. The new design takes the original lifting bail and adds a hinge allowing the bail to fold over towards the lid. A spring is used to hold the top of the bail at a 52 degree angle so the bail can be remotely pushed to the vertical position. Since the new design does not change the attachment to the box or the top of the bail which interfaces with the original lifting yoke, this analysis is limited to the design of the hinge.

The lifting bails must withstand a factor of safety for lifting of 3:1 to yield.

2.0 REFERENCES

- AISC, 1989, *Manual of Steel Construction*, Allowable Stress Design, 9th Edition, American Institute of Steel Construction, Inc., Chicago, Illinois.
- PNL, 1988, *22 Ton Steel Waste Disposal Box Arrangement*, drawing H-3-56431, Rev. 1, Pacific Northwest Laboratory, Richland, Washington.
- Shigley, Joseph E, and Charles R. Mischke, 1989, *Mechanical Engineering Design*, Fifth Edition, McGraw-Hill, Inc. New York, New York.
- DWS, 1999, *Lifting Bail Spring*, (Fax to D. Reid, 9/23/99, 13:29), 800-238-3510, Diamond Wire Spring Co., Tyler, Texas.

3.0 ASSUMPTIONS, RESULTS, AND CONCLUSIONS

The lifting bail design consists of three main components. The components are the top and bottom parts of the bail, the hinge pin, and the spring. The maximum weight of the package that is used in the calculations is conservatively estimated to be 30,000 lb on each lifting bail (60,000 lb total). The margin of safety for each evaluation includes a lifting factor of 3:1 to yield.

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The hinge pin is made of AISI 41L40 with a yield stress of 120,000 psi. A two step process was used to determine the bending of the pin for this design. The first step used the formulas for continuous beams, with the theory of three moments. The second step utilizes the simple beam formulas for a uniformly distributed load and variable end moments. The end moments are taken from the moments determined from the first step and added in the second step to find the maximum moment for each span. The moments are symmetric about the center so only one side of the maximum moments are determined. The maximum moment is found on the forth span to be 2,151 in-lb. The maximum bending stress in the pin is thus 32,706 psi which gives a margin of safety of 0.22. The largest shear stress of the pin is 6,603 psi with a margin of safety of 2.03. The largest bending stress at the reaction points is 9,032 psi with a margin of safety of 3.43. From the bending and shear stresses the principle stresses are determined and are shown to be 9,871 psi and 6,603 psi for shear. The margin of safety is 3.05 and 0.22 respectively. The bearing stress of the pin is 4,898 psi with a margin of safety of 3.08.

The top and bottom bails were evaluated for direct tensile stress, shear tear-out stress, and bearing stress using AISI 1040 with a yield stress of 42,000 psi. Both the top and bottom bails have the same area for hinge teeth. Therefore, stresses will be the same for both top and bottom bails. The direct tensile stress is 3,820 psi with a margin of safety of 2.67. The shear tear-out stress is 4,193 psi with a margin of safety of 0.67. The bearing stress is 4,881 psi with a margin of safety of 1.87.

The maximum moment required to lift the top bail by the spring was calculated to be 825 in-lb about the center of the pin. At 60 degrees the moment required is 413 in-lb. Spring $\phi 2.0 \times 3.86 \times \phi .374$ 17-7 by Diamond Wire Spring Co. was determined to sufficiently provide the required moments and allows the lifting bail to be positioned at a steady state angle of 52 degrees.

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Evaluation of Pin

Using the equation for the theorem of three moments (AISC 1989, 2-294):

$$M_1 L_1 + 2M_2 (L_1 + L_2) + M_3 L_2 + \frac{w L_1^3}{4} + \frac{w L_2^3}{4}$$

Assume there are no gaps between the hinge teeth.

$$F := 30000$$

$$L_1 := 1.17 \quad L_2 := 1.75 \quad L_3 := 4.4$$

$$w_1 := \frac{F}{6 \cdot 1.17}$$

$$w_2 := \frac{-F}{4 \cdot 1.75}$$

$$w_1 = 4273.5$$

$$w_2 = -4285.71$$

$$w_3 := 0$$

$$M_1 := 0$$

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Guess values
for Mathcad:

$$M_1 := 0 \quad M_7 := -150 \quad M_{13} := 0$$

$$M_2 := 500 \quad M_8 := 600$$

$$M_3 := 450 \quad M_9 := 450$$

$$M_4 := 450 \quad M_{10} := 450$$

$$M_5 := 600 \quad M_{11} := 500$$

$$M_6 := -150 \quad M_{12} := 200$$

Given due to symmetry

$$M_2 = M_{11} \quad M_4 = M_9 \quad M_6 = M_7$$

$$M_3 = M_{10} \quad M_5 = M_8$$

Equation 1

$$M_1 \cdot L_1 + 2 \cdot M_2 \cdot (L_1 + L_2) + M_3 \cdot L_2 + \frac{w_1 \cdot L_1^3}{4} + \frac{w_2 \cdot L_2^3}{4} = 0$$

Equation 2

$$M_3 \cdot L_1 + 2 \cdot M_4 \cdot (L_1 + L_2) + M_5 \cdot L_2 + \frac{w_1 \cdot L_1^3}{4} + \frac{w_2 \cdot L_2^3}{4} = 0$$

Equation 3

$$M_5 \cdot L_1 + 2 \cdot M_6 \cdot (L_1 + L_3) + M_7 \cdot L_3 + \frac{w_1 \cdot L_1^3}{4} + \frac{w_3 \cdot L_3^3}{4} = 0$$

Equation 4

$$M_7 \cdot L_1 + 2 \cdot M_8 \cdot (L_1 + L_2) + M_9 \cdot L_2 + \frac{w_1 \cdot L_1^3}{4} + \frac{w_2 \cdot L_2^3}{4} = 0$$

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Equation 5

$$M_9 \cdot L_1 + 2 \cdot M_{10} \cdot (L_1 + L_2) + M_{11} \cdot L_2 + \frac{w_1 \cdot L_1^3}{4} + \frac{w_2 \cdot L_2^3}{4} = 0$$

Equation 6

$$M_{12} - w_1 \cdot \frac{L_1^2}{8} + \frac{M_{11} + M_{13}}{2} - \frac{(M_{11} - M_{13})^2}{2 \cdot w_1 \cdot L_1^2} = 0$$

M_1
 M_2
 M_3
 M_4
 M_5
 M_6
 M_7
 M_8
 M_9
 M_{10}
 M_{11}
 M_{12}
 M_{13}

:= Find($M_1, M_2, M_3, M_4, M_5, M_6, M_7, M_8, M_9, M_{10}, M_{11}, M_{12}, M_{13}$)

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Results:

$$\begin{aligned}
 M_1 &= 0 & M_6 &= -154.84 & M_{11} &= 559.1 \\
 M_2 &= 559.1 & M_7 &= -154.84 & M_{12} &= 478.4 \\
 M_3 &= 437.65 & M_8 &= 594.05 & M_{13} &= 0.02 \\
 M_4 &= 424.56 & M_9 &= 424.56 & & \\
 M_5 &= 594.05 & M_{10} &= 437.65 & &
 \end{aligned}$$

Largest moment is $M_5 = M_8 = 594.05$ in-lb.

Find maximum moment between spans

Span 1

$$M_{\max 1} := \frac{w_1 \cdot L_1^2}{8} - \frac{M_1 + M_2}{2} + \frac{(M_1 - M_2)^2}{2 \cdot w_1 \cdot L_1^2}$$

$$M_{\max 1} = 478.42$$

Span 2

$$M_{\max 2} := \frac{w_2 \cdot L_2^2}{8} - \frac{M_2 + M_3}{2} + \frac{(M_2 - M_3)^2}{2 \cdot w_2 \cdot L_2^2}$$

$$M_{\max 2} = -2139.57$$

Span 3

$$M_{\max 3} := \frac{w_1 \cdot L_1^2}{8} - \frac{M_3 + M_4}{2} + \frac{(M_3 - M_4)^2}{2 \cdot w_1 \cdot L_1^2}$$

$$M_{\max 3} = 300.16$$

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Span 4

$$M_{\max 4} := \frac{w_2 \cdot L_2^2}{8} - \frac{M_4 + M_5}{2} + \frac{(M_4 - M_5)^2}{2 \cdot w_2 \cdot L_2^2}$$

$$M_{\max 4} = -2151.02$$

Span 5

$$M_{\max 5} := \frac{w_1 \cdot L_1^2}{8} - \frac{M_5 + M_6}{2} + \frac{(M_5 - M_6)^2}{2 \cdot w_1 \cdot L_1^2}$$

$$M_{\max 5} = 559.58$$

Span 6

$$M_{\max 6} := \frac{w_3 \cdot L_3^2}{8} - \frac{M_6 + M_7}{2}$$

$$M_{\max 6} = 154.84$$

Maximum moment is $M_{\max 4} = -2151.02$ in-lb.

Find reaction forces to determine shear:

Span 1

$$R_1 := \frac{w_1 \cdot L_1}{2} - \frac{M_1 - M_2}{L_1}$$

$$R_1 = 2977.87$$

$$R_{2a} := \frac{w_1 \cdot L_1}{2} + \frac{M_1 - M_2}{L_1}$$

$$R_{2a} = 2022.13$$

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Span 2

$$R_{2b} := \frac{w_2 \cdot L_2}{2} + \frac{M_2 - M_3}{L_2}$$

$$R_{3a} := \frac{w_2 \cdot L_2}{2} - \frac{M_2 - M_3}{L_2}$$

$$R_{2b} = -3680.6$$

$$R_{3a} = -3819.4$$

Span 3

$$R_{3b} := \frac{w_1 \cdot L_1}{2} + \frac{M_3 - M_4}{L_1}$$

$$R_{4a} := \frac{w_1 \cdot L_1}{2} - \frac{M_3 - M_4}{L_1}$$

$$R_{3b} = 2511.19$$

$$R_{4a} = 2488.81$$

Span 4

$$R_{4b} := \frac{w_2 \cdot L_2}{2} - \frac{M_4 - M_5}{L_2}$$

$$R_{5a} := \frac{w_2 \cdot L_2}{2} + \frac{M_4 - M_5}{L_2}$$

$$R_{4b} = -3653.15$$

$$R_{5a} = -3846.85$$

Span 5

$$R_{5b} := \frac{w_1 \cdot L_1}{2} + \frac{M_5 - M_6}{L_1}$$

$$R_{6a} := \frac{w_1 \cdot L_1}{2} - \frac{M_5 - M_6}{L_1}$$

$$R_{5b} = 3140.07$$

$$R_{6a} = 1859.93$$

Span 6

$$R_{6b} := \frac{M_6 - M_7}{L_3}$$

$$R_{7a} := \frac{M_6 - M_7}{L_3}$$

$$R_{6b} = 0$$

$$R_{7a} = 0$$

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Add reactions together for each span

$$R_2 := R_{2a} + R_{2b}$$

$$R_4 := R_{4a} + R_{4b}$$

$$R_6 := R_{6a} + R_{6b}$$

$$R_3 := R_{3a} + R_{3b}$$

$$R_5 := R_{5a} + R_{5b}$$

$$R_1 = 2977.87$$

$$R_2 = -1658.47$$

$$R_3 = -1308.21$$

$$R_4 = -1164.34$$

$$R_5 = -706.78$$

$$R_6 = 1859.93$$

Maximum reaction force is $R_1 = 2977.87$ lbf.

Determine Shear Stresses at reaction points from the beam:

$$A_{pin} := \frac{\pi}{4} \cdot .875^2 \quad A_{pin} = 0.601 \text{ in}^2$$

$$\tau_1 := \frac{4}{3} \cdot \frac{R_1}{A_{pin}} \quad \tau_1 = 6602.95 \text{ psi}$$

$$\tau_2 := \frac{4}{3} \cdot \frac{R_2}{A_{pin}} \quad \tau_2 = -3677.39 \text{ psi}$$

$$\tau_3 := \frac{4}{3} \cdot \frac{R_3}{A_{pin}} \quad \tau_3 = -2900.75 \text{ psi}$$

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$$\tau_4 := \frac{4}{3} \frac{R_4}{A_{pin}} \quad \tau_4 = -2581.74 \text{ psi}$$

$$\tau_5 := \frac{4}{3} \frac{R_5}{A_{pin}} \quad \tau_5 = -1567.17 \text{ psi}$$

$$\tau_6 := \frac{4}{3} \frac{R_6}{A_{pin}} \quad \tau_6 = 4124.1 \text{ psi}$$

Largest Shear Stress is at $\tau_1 = 6602.95 \text{ psi}$

Determine the bending stress at the reaction points:

$$c = 0.44 \text{ in} \quad I = 0.0288 \text{ in}^4$$

$$\sigma_1 := \frac{M_1 \cdot c}{I} \quad \sigma_1 = 0 \text{ psi}$$

$$\sigma_2 := \frac{M_2 \cdot c}{I} \quad \sigma_2 = 8500.97 \text{ psi}$$

$$\sigma_3 := \frac{M_3 \cdot c}{I} \quad \sigma_3 = 6654.36 \text{ psi}$$

$$\sigma_4 := \frac{M_4 \cdot c}{I} \quad \sigma_4 = 6455.26 \text{ psi}$$

$$\sigma_5 := \frac{M_5 \cdot c}{I} \quad \sigma_5 = 9032.29 \text{ psi}$$

$$\sigma_6 := \frac{M_6 \cdot c}{I} \quad \sigma_6 = -2354.23 \text{ psi}$$

Largest bending stress at the reaction points is at $\sigma_5 = 9032.29 \text{ psi}$

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Determine the maximum principal stresses at the reaction points:

$$\sigma_{p1} := \frac{\sigma_1}{2} + \left[\left(\frac{\sigma_1}{2} \right)^2 + \tau_1^2 \right]^{.5} \quad \sigma_{p1} = 6602.95 \text{ psi}$$

$$\sigma_{p2} := \frac{\sigma_2}{2} + \left[\left(\frac{\sigma_2}{2} \right)^2 + \tau_2^2 \right]^{.5} \quad \sigma_{p2} = 9870.97 \text{ psi}$$

$$\sigma_{p3} := \frac{\sigma_3}{2} + \left[\left(\frac{\sigma_3}{2} \right)^2 + \tau_3^2 \right]^{.5} \quad \sigma_{p3} = 7741.3 \text{ psi}$$

$$\sigma_{p4} := \frac{\sigma_4}{2} + \left[\left(\frac{\sigma_4}{2} \right)^2 + \tau_4^2 \right]^{.5} \quad \sigma_{p4} = 7360.79 \text{ psi}$$

$$\sigma_{p5} := \frac{\sigma_5}{2} + \left[\left(\frac{\sigma_5}{2} \right)^2 + \tau_5^2 \right]^{.5} \quad \sigma_{p5} = 9296.48 \text{ psi}$$

$$\sigma_{p6} := \frac{\sigma_6}{2} + \left[\left(\frac{\sigma_6}{2} \right)^2 + \tau_6^2 \right]^{.5} \quad \sigma_{p6} = 3111.68 \text{ psi}$$

Largest Principal Stress is $\sigma_{p2} = 9870.97 \text{ psi}$

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Determine Maximum Principal Shear Stress:

$$\tau_{p1} := \left[\left(\frac{\sigma_1}{2} \right)^2 + \tau_1^2 \right]^{.5} \quad \tau_{p1} = 6602.95 \text{ psi}$$

$$\tau_{p2} := \left[\left(\frac{\sigma_2}{2} \right)^2 + \tau_2^2 \right]^{.5} \quad \tau_{p2} = 5620.48 \text{ psi}$$

$$\tau_{p3} := \left[\left(\frac{\sigma_3}{2} \right)^2 + \tau_3^2 \right]^{.5} \quad \tau_{p3} = 4414.12 \text{ psi}$$

$$\tau_{p4} := \left[\left(\frac{\sigma_4}{2} \right)^2 + \tau_4^2 \right]^{.5} \quad \tau_{p4} = 4133.16 \text{ psi}$$

$$\tau_{p5} := \left[\left(\frac{\sigma_5}{2} \right)^2 + \tau_5^2 \right]^{.5} \quad \tau_{p5} = 4780.33 \text{ psi}$$

$$\tau_{p6} := \left[\left(\frac{\sigma_6}{2} \right)^2 + \tau_6^2 \right]^{.5} \quad \tau_{p6} = 4288.8 \text{ psi}$$

Largest Principal Shear Stress is $\tau_{p1} = 6602.95 \text{ psi}$

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Determine Bending Stresses from maximum bending moments between reaction points:

Maximum moment is: $M_{\max 4} = -2151.02 \text{ in} \cdot \text{lb}$

$$c := .5 \cdot \frac{7}{8} \quad I := \frac{\pi}{4} (c)^4$$

$$\sigma_b := \left| M_{\max 4} \right| \frac{c}{I}$$

$$\sigma_b = 32705.52 \text{ psi}$$

Determine Bearing Stress on pin:

Area length of teeth on hinge $L := 7.0 \text{ in}$

Diameter of pin $D_p := .875 \text{ in}$

$$\text{Bearing stress } \sigma_{br} := \frac{F}{L \cdot D_p} \quad \sigma_{br} = 4897.96 \text{ psi}$$

Pin Allowable Stresses:

AISI 41L40 - 120,000 psi yield strength

Allowable including a 3:1 Factor of Safety for lifting

$$\sigma_{all} := \frac{1}{3} \cdot 120000 \quad \sigma_{all} = 40000 \text{ psi}$$

Shear allowable using maximum shear stress theory predicts yield stress for shear at $\frac{1}{2} \sigma_{all}$ (Shigley, 1989)

$$\tau_{all} := .5 \cdot \sigma_{all} \quad \tau_{all} = 20000 \text{ psi}$$

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Compare maximum stresses to allowables:

Largest shear stress at reaction points

$$\tau_1 = 6602.95 \text{ psi} \quad MS_a := \frac{\tau_{all}}{\tau_1} - 1 \quad MS_a = 2.03$$

Largest bending stress at reaction points and margin of safety

$$\sigma_5 = 9032.29 \text{ psi} \quad MS_b := \frac{\sigma_{all}}{\sigma_5} - 1 \quad MS_b = 3.43$$

Largest principal stresses at reaction points and margin of safety

$$\sigma_{p2} = 9870.97 \text{ psi} \quad MS_c := \frac{\sigma_{all}}{\sigma_{p2}} - 1 \quad MS_c = 3.05$$

Largest principal shear stress at reaction points and margin of safety

$$\tau_{p1} = 6602.95 \text{ psi} \quad MS_d := \frac{\tau_{all}}{\tau_{p1}} - 1 \quad MS_d = 0.22$$

Largest bending stress between reaction points and margin of safety

$$\sigma_b = 32705.52 \text{ psi} \quad MS_d := \frac{\sigma_{all}}{\sigma_b} - 1 \quad MS_d = 0.22$$

Bearing stress on pin and margin of safety

$$\sigma_{br} = 4897.96 \text{ psi} \quad MS_e := \frac{\tau_{all}}{\sigma_{br}} - 1 \quad MS_e = 3.08$$

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Evaluation of Lifting Bails

Both top and bottom lifting bails have the same total length of teeth of 7.0 in. Therefore, stresses will be equal. Loading for each bail is 30,000 lbf.

Width of bail: $W := 2.0 \cdot \text{in}$

Total length of teeth: $L := 7.0 \cdot \text{in}$

Diameter of shaft hole: $\phi h := .878 \cdot \text{in}$

Length of shear tear-out path:

$$L_s := \left(.90 \cdot \text{in} - \frac{\phi h}{2} \right) + \left(1.0 \cdot \text{in} - \frac{\phi h}{2} \right) \quad L_s = 1.02 \cdot \text{in}$$

Minimum tensile area: $A_t := L \cdot (W - \phi h) \quad A_t = 7.85 \cdot \text{in}^2$

Shear tear-out area: $A_s := L_s \cdot L \quad A_s = 7.15 \cdot \text{in}^2$

Bearing area: $A_b := L \cdot \phi h \quad A_b = 6.15 \cdot \text{in}^2$

Tensile Stress: $\sigma_t := \frac{F}{A_t} \quad \sigma_t = 3819.7 \cdot \text{psi}$

Shear Tear-Out Stress: $\tau_t := \frac{F}{A_s} \quad \tau_t = 4193.45 \cdot \text{psi}$

Bearing Stress: $\sigma_{bs} := \frac{F}{A_b} \quad \sigma_{bs} = 4881.22 \cdot \text{psi}$

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Lifting Bail Allowable Stresses:

AISI 1040 - 42,000 psi yield strength

Allowable including a 3:1 Factor of Safety for lifting

$$\sigma_{b\ all} := \frac{1}{3} \cdot 42000 \cdot \text{psi} \quad \sigma_{b\ all} = 14000 \cdot \text{psi}$$

Shear allowable using maximum shear stress theory predicts yield stress for shear at $\frac{1}{2} \sigma_{b\ all}$ (Shigley, 1989)

$$\tau_{b\ all} := .5 \sigma_{b\ all} \quad \tau_{b\ all} = 7000 \cdot \text{psi}$$

Compare maximum stresses to allowables:

Tensile stress and margin of safety

$$\sigma_t = 3819.7 \cdot \text{psi} \quad MS_f := \frac{\sigma_{b\ all}}{\sigma_t} - 1 \quad MS_f = 2.67$$

Shear tear-out stress and margin of safety

$$\tau_t = 4193.45 \cdot \text{psi} \quad MS_g := \frac{\tau_{b\ all}}{\tau_t} - 1 \quad MS_g = 0.67$$

Bearing stress and margin of safety

$$\sigma_{bs} = 4881.22 \cdot \text{psi} \quad MS_h := \frac{\sigma_{b\ all}}{\sigma_{bs}} - 1 \quad MS_h = 1.87$$

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Evaluation of Torsional Spring Force

The torsional spring for this application was analyzed with spring information provided by Diamond Wire Spring Co. The spring has been designed by Diamond Wire Spring Co. to satisfy all of the requirements of the application. The material chosen is stainless steel. The model/part number assigned by Diamond Wire Spring Co. is $\phi 2.0 \times 3.86 \times \phi .374$ 17-7 (DWS 1999).

Angle between spring legs at the relaxed state: $\theta = 55$ deg

Width of the spring from end to end: $w = 3.86$ in.

Number of turns in the spring: $n = 9.35$

Diameter of the spring wire used: $d = .37$ in.

Outer diameter of the spring: $D = 2.00$ in.

Volume of top lifting bail (derived by AutoCAD): $V := 404.47 \cdot \text{in}^3$

Density of steel (Shigley, 1989): $\rho := .283 \cdot \frac{\text{lb}}{\text{in}^3}$

Weight of top lifting bail: $W := \rho \cdot V$ $W = 114.06$ lb

Centroid location (derived by AutoCAD): $Y_{\text{bail}} := 7.21$ in

Maximum torque to overcome (at 0 deg):

$$T_0 := W \cdot Y_{\text{bail}} \cos(0 \cdot \text{deg}) \quad T_0 = 825.29 \cdot \text{in} \cdot \text{lb}$$

Torque at 45 deg:

$$T_{45} := W \cdot Y_{\text{bail}} \cos(45 \cdot \text{deg}) \quad T_{45} = 583.57 \cdot \text{in} \cdot \text{lb}$$

Torque at 60 deg:

$$T_{60} := W \cdot Y_{\text{bail}} \cos(60 \cdot \text{deg}) \quad T_{60} = 412.65 \cdot \text{in} \cdot \text{lb}$$

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Determine the steady state angle of the lifting bail.

Spring rate $S_r := 9.7067 \cdot \frac{\text{in} \cdot \text{lb}}{\text{deg}}$

Weight of top lifting bail $W = 114.47 \text{ lb}$

Centroid location bail $Y_{\text{bail}} := 7.21 \text{ in}$

Steady state angle

$$T_a := S_r \cdot \phi \cdot \text{deg}$$

$$T_b := W \cdot Y_{\text{bail}} \cos(\phi \cdot \text{deg})$$

$$T_a := T_b$$

$$\phi := 52.16$$

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*AutoCAD output for top lifting bail

Volume: 404.4687
 Bounding box: X: -9.5450 - 9.5450
 Y: -1.0000 - 16.1000
 Z: -1.0000 - 1.0000
 Centroid: X: 0.0000
 Y: 7.2082
 Z: 0.0000
 Moments of inertia: X: 30125.8997
 Y: 9383.4752
 Z: 39237.3121
 Products of inertia: XY: 0.0000
 YZ: 0.0000
 ZX: 0.0000
 Radii of gyration: X: 8.6303
 Y: 4.8166
 Z: 9.8493
 Principal moments and X-Y-Z directions about centroid:
 I: 9110.7225 along [1.0000 0.0000 0.0000]
 J: 9383.4752 along [0.0000 1.0000 0.0000]
 K: 18222.1349 along [0.0000 0.0000 1.0000]

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