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15 July 2012

JACKING POCKET DESIGN

Any time a jack cannot be placed under a structure to jack it up or down, then jacking pockets or brackets must be designed and either bolted or welded to the side of the structure. This holds true for a structure that has settled unevenly and has to be jacked back up to grade by welding jacking pockets to the sides of the columns or for rolling a vessel under a pipe rack where jacking pockets are required to be attached to the sides of the saddles for lowering the vessel down over the anchor bolts.

The following example is for rolling a horizontal 80 ton vessel under a 15’ pipe rack and into a structure.

Parameters for the rolling operation:

1. The 14’ overall height of the vessel, including the rollers, had to go under a pipe rack that had 15’ of clearance from the concrete paving up to the underside of the pipe rack.
2. Four 37.5 ton Hilman rollers were used, OT type with Accu-roller guides. After the roller size had been selected, Hilman recommended the channel size that should be used with the roller.
3. The Hilman’s ran in C 12 x 30 channel rails that were laid out along precise centerlines.
4. Where the rail sections were connected together, the ends were cut on a 45° angle to make crossing the joints with the rollers smooth and with little force to the joint.
5. The rail sections were held together using clips on each side of the joints. The clips were bolted to the sides of the channels.
6. Four 100 ton jacks were used to lower the vessel down over the anchor bolts because the structure did not have an opening for a crane hook.

Steps for the rolling and jacking operation:

7. Offload the vessel with a crane and set it on the four Hilman rollers positioned in the rails.
8. Bolt the top flange of the rollers to the underside of the saddles to keep them from falling out from under the saddle if the load is carried by three rollers due to the table top effect.
9. Push the vessel under and through the pipe rack with a push bar from the prime mover.
10. When the vessel is positioned over the anchor bolts, weld the pre-fabricated jacking pockets to each side of each saddle. They were left off for transportation.
11. Place 3” of hardwood cribbing under each jacking pocket.
12 Place the four jacks on top of the wood cribbing and extend the rams until the caps are tight up against the bottom of the jacking pockets. The overall height of the jacks should be about 18” at this point. This will leave about 2” of ram stroke for raising the saddles and freeing the rollers.
13. Extend the rams until the rollers are free.
14 Remove the rollers and channel rails.
15. Place 3” of hardwood cribbing under each corner of each saddle.
16. Lower the jacks until the saddles are resting on the cribbing.
17. Lower the jacks until they are free and then remove them and the wood cribbing.
18. Reinstall the jacks and jack up the saddles until they clear the cribbing.
19. Remove the cribbing and lower the saddles down over the anchor bolts.

20. Remove the jacks.

End Of Rolling & Jacking Procedure

Jacking Pocket Design:

ASSUMPTIONS AND DESIGN STEPS:

When designing a lifting attachment or weldment, the first thing that I usually do is to make a layout to scale, based on the actual dimensions involved. Then I do the calculations to make sure that the member sizes are correct and then size the welds. The jacking pocket design was no different. I completed the following steps in my design:

1. Made a layout of the jacking pocket based on the load, the vertical clearances, the jack dimensions and the jack stroke. See sheet 1 of 4 next page.
2. Elected to weld a ½" plate "e" to the saddle flange as a base for the jacking pocket. Assumed the ½" plate would be good by inspection.
3. As the saddles were designed with a single web, I had the choice to either use one vertical stiffener plate in line with the web or use two stiffener plates on either side of the web. The calculations show that a single stiffener plate with a thickness of 0.29" would be required. Therefore I used two ½" stiffener plates. Always be conservative, especially when it doesn't cost much more.
4. Because the stiffener plates were so close together, assumed by inspection that the 1" horizontal jack plate "f" was strong enough in bending from the force of the jack.
5. To determine the weld size between the saddle flange and the ½" base plate "e", I assumed that it was similar to a pad eye lug with two parallel welds 8" apart with an eccentricity of 7" and a load of 40 kips. The weld required was 0.22" on each side of the base plate. I elected to use a ¼" weld.
6. Knowing that a 1/4" weld would be good for welding plate "e" to the saddle flange, I know that a 1/4" weld would be twice as much, way more than enough, to hold the two vertical stiffeners to plate "e", ie, with four parallel welds.

CLARIFICATION:

1. In the 7th edition of AISC, table XIV , the sketch on the left is with the load in the plane of the welds, ie, say a lug plate that is being twisted off at the base by a load causing torque on the two parallel welds. An example might be when a beam is welded on both ends to a support by its web and then the beam is subjected to a twisting load. This also applies to table XIX in the 9th edition of AISC.

2. The sketch on the right shows a load that is out of plane of the welds, ie, like a pad eye lug with the load at 0 degrees and "al" equals the eccentricity of 7". The welds are then only subjected to bending and shear.

   Both the sketch for the special case on the right and my pad eye lug program are for a pad eye with two welds on either side of the lug.

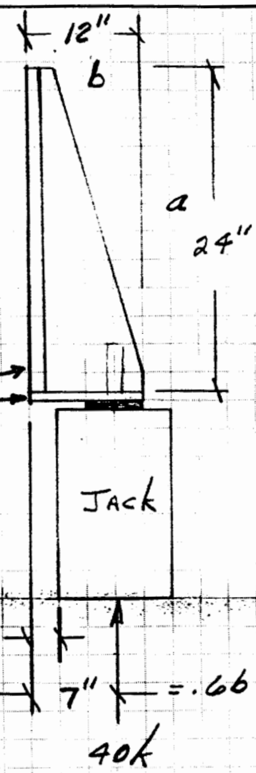
3. The comparison of the two tables and the pad eye lug program was just to show that the ¼" weld was very conservative from table XIV and kept getting more conservative for table XIX and the pad eye program, especially when a value of 1.0 was used for C1.

FLUOR 04-02-51

CONTRACT NO. 783500
SHEET NO. 1 OF 4
BY: KENT
DATE: 22 APR 87

ROLL IN OPERATION

DESIGN JACKING BRACKET
FOR 100 TON JACK (MAX.)
ENERPAC RC-1006



APPROX. JACK DIMENSIONS:
HEIGHT (COLLASPED) = 14 1/16"
" (EXTENDED) = 20 11/16
STROKE 6 5/8
CYL ROD ⌀ 4 1/8
HOUSING O.D. 6 7/8

USE 2 VERT STIFFENERS
OF THICKNESS t
WELDED TO 1/2" PL e &
1" PL f.
REF. SALMON & JOHNSON Pg 716

LET $P = 40k$ (1/4 VESSEL WT)
$a = 24"$
$b = 12"$

$b/a = .5$

$$Z = 1.39 - 2.2(.5) + 1.27(.5)^2 - .25(.5)^3$$
$$= .58$$

$$t \geq \frac{P}{bZ(.6F_y)} = \frac{40k \times 1.25}{12'' \times .58 \times .6 \times 36\text{ KSI} \times 2}$$

(2 PL's)

$$= .17''$$

CHECK STABILITY EQ 13.5.6

$$t \geq \frac{b\sqrt{F_y}}{250(b/a) \times 2} = \frac{12''\sqrt{36}}{250 \times .5 \times 2} = .29$$

(2 PL's)

∴ USE 2 - 1/2" PLATES.

FORM E-045 . 1/76
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See Appendix A for the reference sheets for Salmon & Johnson

FLUOR

CONTRACT NO.
SHEET NO. 2 OF 4
BY:
DATE:

WELDS

GUSSET PLATES TO VERT 1/2" PL "e" &
VERT 1/2" PL & TO SADDLE.

TABLE XIV, SEVENTH EDITION, AISC

$$D = \frac{P}{C C_1 l}$$

$P = 40k * 1.25 = 50k$
$C_1 = .65$ FROM REM (FLUOR STD)
$al = 7"$ FROM SHEET 1
$l = 24"$
$a = .29 \Leftarrow 7/24$
$k = 0$
$C = .93$ FROM TABLE XIV

$$D_{REQ} = \frac{50k}{.93 * .65 * 24} = 3.45/16^{THS} = 0.22"$$

USE 1/4" WELD

WELD
— ALL EXPOSED JOINTS —

CHANGE $C_1 = 1.0$ PER ABOVE TABLE

$$D_{REQ} = \frac{50}{0.93 * 1 * 24} = 2.24/16^{THS} = 0.14"$$

TO SEE HOW THE AISC ALLOWABLES HAVE INCREASED, SEE SAME TABLE, NINTH EDITION
WHERE C = 1.17

$$D_{REQ} = \frac{50}{1.17 * 1 * 24} = 1.78/16THS = 0.11"$$

FORM E-04 . 1/76
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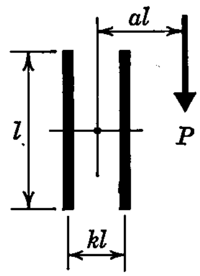
Notice that the C1 = 0.65 listed above was a Fluor standard at the time of this design. They now use C1 = 1.0 per the AISC table. See the next page. That is the reason for the re-calculation of D where it then = 0.14"

Also note that in the table from the Ninth Edition the weld gets even smaller. This is due to more testing over the years, better welding rod, better QC, etc. The fourteenth Edition is now out so the welds are probably smaller than above.

SEVENTH EDITION

ECCENTRIC LOADS ON WELD GROUPS

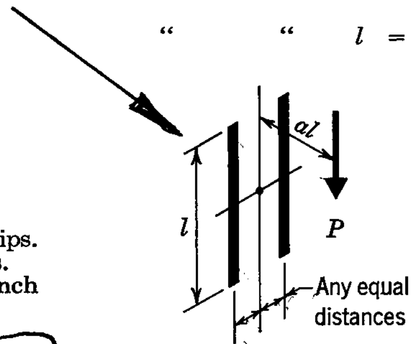
TABLE XIV Coefficients C



Required Minimum $C = \dfrac{P}{C_1 D l}$

" " $D = \dfrac{P}{C C_1 l}$

" " $l = \dfrac{P}{C C_1 D}$



P = Permissible eccentric load in kips.
l = Length of each weld in inches.
D = Number of sixteenths of an inch in fillet weld size.
C = Coefficients tabulated below.
C_1 = Coefficient for electrode used (see Table on page 4-67).
= 1.0 for E70XX electrodes.

$$P = C C_1 D l$$

Special Case*
(Load not in plane of weld group.)
Use C-values given in column headed k = 0.

0.29 0.93

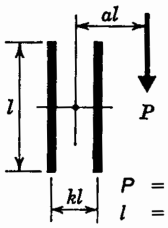
a	k = 0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	1.2	1.4	1.6	1.8	2.0
0.1	1.59	1.54	1.51	1.50	1.51	1.52	1.54	1.55	1.57	1.59	1.60	1.63	1.65	1.67	1.68	1.70
0.2	1.19	1.15	1.15	1.16	1.20	1.23	1.27	1.30	1.33	1.36	1.39	1.44	1.48	1.51	1.54	1.56
0.3	.901	.881	.892	.923	.966	1.01	1.06	1.11	1.15	1.19	1.22	1.28	1.33	1.38	1.41	1.45
0.4	.714	.704	.721	.756	.803	.854	.906	.956	1.00	1.05	1.09	1.16	1.21	1.26	1.31	1.34
0.5	.587	.583	.601	.637	.683	.734	.787	.839	.887	.933	.975	1.05	1.11	1.17	1.21	1.25
0.6	.497	.495	.514	.548	.593	.642	.694	.745	.794	.840	.883	.960	1.03	1.08	1.13	1.18
0.7	.430	.430	.449	.481	.523	.570	.620	.659	.717	.763	.806	.883	.951	1.01	1.06	1.11
0.8	.379	.380	.397	.428	.467	.512	.559	.607	.654	.698	.741	.812	.885	.945	.998	1.04
0.9	.338	.340	.357	.385	.422	.464	.509	.555	.600	.644	.685	.761	.828	.888	.941	.989
1.0	.305	.308	.323	.350	.384	.424	.467	.511	.554	.596	.637	.711	.778	.837	.891	.939
1.2	.255	.258	.272	.296	.326	.362	.401	.441	.481	.520	.558	.629	.693	.751	.804	.852
1.4	.219	.222	.235	.256	.283	.316	.351	.387	.424	.460	.496	.563	.625	.681	.732	.779
1.6	.192	.195	.206	.225	.250	.280	.312	.345	.379	.413	.446	.509	.568	.622	.672	.718
1.8	.171	.174	.184	.201	.224	.251	.280	.311	.343	.374	.405	.465	.521	.573	.621	.666
2.0	.154	.157	.166	.182	.203	.227	.255	.283	.313	.342	.371	.428	.481	.531	.577	.620
2.2	.140	.143	.152	.166	.185	.208	.233	.260	.287	.315	.342	.396	.447	.494	.539	.581
2.4	.129	.131	.139	.153	.170	.192	.215	.240	.266	.292	.318	.369	.417	.463	.505	.546
2.6	.119	.121	.129	.141	.158	.178	.200	.223	.247	.272	.296	.345	.391	.435	.476	.515
2.8	.110	.112	.120	.131	.147	.166	.186	.208	.231	.254	.278	.324	.368	.410	.449	.487
3.0	.103	.105	.112	.123	.138	.155	.174	.195	.217	.239	.261	.305	.347	.388	.426	.462

* Valid only when the connection material between the welds is solid and does not bend in the plane of the welds.

NINTH EDITION

ECCENTRIC LOADS ON WELD GROUPS

TABLE XIX Coefficients C



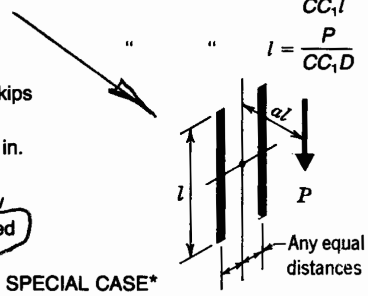
Required Minimum $C = \dfrac{P}{C_1 D l}$

" " $D = \dfrac{P}{C C_1 l}$

" " $l = \dfrac{P}{C C_1 D}$

P = Allowable eccentric load in kips
l = Length of each weld in in.
D = Number of sixteenths of an in. in fillet weld size
C = Coefficients tabulated below
C_1 = Coefficient for electrode used (see Table on p. 4-72).
= 1.0 for E70XX electrodes.

$P = CC_1Dl$



SPECIAL CASE*
(Load not in plane of weld group)
Use C-values given in column headed $k = 0$

.29 1.17

a \ k	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	1.2	1.4	1.6	1.8	2.0
.06	1.67	1.67	1.68	1.68	1.68	1.69	1.69	1.69	1.69	1.70	1.70	1.70	1.71	1.71	1.71	1.71
.08	1.64	1.65	1.65	1.65	1.66	1.66	1.66	1.66	1.67	1.67	1.67	1.67	1.68	1.68	1.69	1.69
.10	1.61	1.61	1.62	1.62	1.62	1.63	1.63	1.63	1.63	1.64	1.64	1.65	1.65	1.66	1.66	1.67
.15	1.51	1.51	1.52	1.52	1.53	1.53	1.54	1.54	1.55	1.56	1.56	1.57	1.58	1.59	1.60	1.61
.20	1.39	1.39	1.40	1.41	1.42	1.43	1.44	1.45	1.46	1.47	1.48	1.50	1.52	1.53	1.54	1.56
.25	1.26	1.27	1.28	1.30	1.31	1.33	1.35	1.36	1.38	1.39	1.41	1.43	1.45	1.47	1.49	1.50
.30	1.14	1.15	1.17	1.19	1.21	1.24	1.26	1.28	1.30	1.32	1.33	1.36	1.39	1.41	1.43	1.45
.40	.939	.951	.976	1.01	1.04	1.07	1.10	1.13	1.16	1.18	1.20	1.24	1.28	1.31	1.33	1.36
.50	.787	.792	.813	.865	.903	.941	.976	1.01	1.04	1.07	1.09	1.14	1.18	1.21	1.25	1.27
.60	.673	.679	.701	.734	.795	.834	.872	.907	.940	.970	.998	1.05	1.09	1.13	1.17	1.20
.70	.585	.592	.615	.647	.708	.748	.787	.823	.857	.888	.918	.971	1.02	1.06	1.10	1.13
.80	.517	.524	.546	.579	.636	.676	.714	.751	.786	.818	.848	.903	.952	.995	1.03	1.07
.90	.463	.469	.491	.524	.576	.615	.654	.690	.725	.757	.788	.844	.893	.938	.978	1.02
1.00	.419	.425	.446	.478	.527	.565	.602	.638	.672	.704	.735	.791	.842	.887	.928	.965
1.20	.351	.357	.377	.406	.448	.484	.519	.553	.586	.617	.647	.702	.752	.798	.840	.878
1.40	.302	.307	.326	.352	.390	.423	.455	.488	.519	.548	.577	.631	.680	.725	.766	.805
1.60	.265	.270	.287	.311	.344	.375	.405	.435	.465	.493	.520	.572	.619	.664	.704	.743
1.80	.236	.241	.256	.278	.308	.336	.365	.393	.421	.448	.474	.523	.569	.612	.652	.689
2.00	.213	.217	.231	.251	.279	.305	.331	.358	.384	.410	.434	.481	.526	.567	.606	.642
2.20	.193	.198	.211	.229	.254	.279	.303	.328	.353	.377	.401	.446	.488	.528	.566	.602
2.40	.177	.181	.194	.211	.234	.256	.280	.303	.327	.350	.372	.415	.456	.495	.531	.566
2.60	.164	.168	.179	.195	.216	.237	.259	.282	.304	.326	.347	.388	.428	.465	.500	.534
2.80	.152	.156	.166	.181	.201	.221	.242	.263	.284	.305	.325	.365	.402	.438	.472	.505
3.00	.142	.145	.155	.169	.188	.207	.226	.246	.266	.286	.306	.344	.380	.415	.448	.479

*Valid only when the connection material between the welds is solid and does not bend in the plane of the welds.

To show that the weld for the jacking pocked plate to the saddle could have been calculated on the Padeye Lug program on my website, see the printout sheet below which shows a slightly larger weld size than AISC. The welds calculated on both my lug programs using the weld treated as a line by Blodgett are always a little larger than those calculated by AISC.

PROGRAM TO DESIGN A PAD EYE TYPE LIFTING LUG v.02

COMPANY: Maximum Reach **PROJECT:** Design Of Jacking Pockets 2 a
ITEM NUMBER: 04-02--51

Crosby G2130x35		Select a metric shackle from the lookup table based on the force on the lug or click the SHACKLE button to enter your own	
3.25	in	Shackle Inside Width at Pin	
4.81	in	Shackle Eye Diameter	
2.30	in	Shackle Pin Diameter	
2.43	in	Lug Pin Hole Diameter	Recommend hole be 0.13" or > than shackle pin dia.
4.00	in	Lug Radius	
2.25	in	Lug Plate Thickness	
24.00	in	Lug Plate Width at Base	Minimum value of 2*radius of lug
0.00	in	Lug Pad Thickness	Input zero if pads are not required
0.00	in	Lug Pad Radius	Input zero if pads are not required
7.00	in	Lug Eccentricity	
40.00	kips	Force on the Lug	
0.00	deg	Angle of the Force on the Lug	Measured from the horizontal
36.00	ksi	Yield Stress of the Lug Material	Fy
14.85	kips/in	Allowable Force on the Weld	Use 10.91 for LH60 or 14.85 for LH70
1.80		Impact factor, IF	Recommend that a minimum 1.8 impact factor be used

OUTPUT:

Checking combined stress of the lug plate

54.00	in^2	Area of Lug Plate at Base
216.00	in^3	Section modulus of the lug plate at the base
2.33	ksi	Bending stress of the lug plate fb, actual
0.00	ksi	Tension stress of the lug plate ft, actual
21.60	ksi	Allowable bending and tension stress, Fb & Ft
0.11		Combined stress of the lug plate. Must be less than 1.0

Checking the lug weld size, with the weld treated as a line

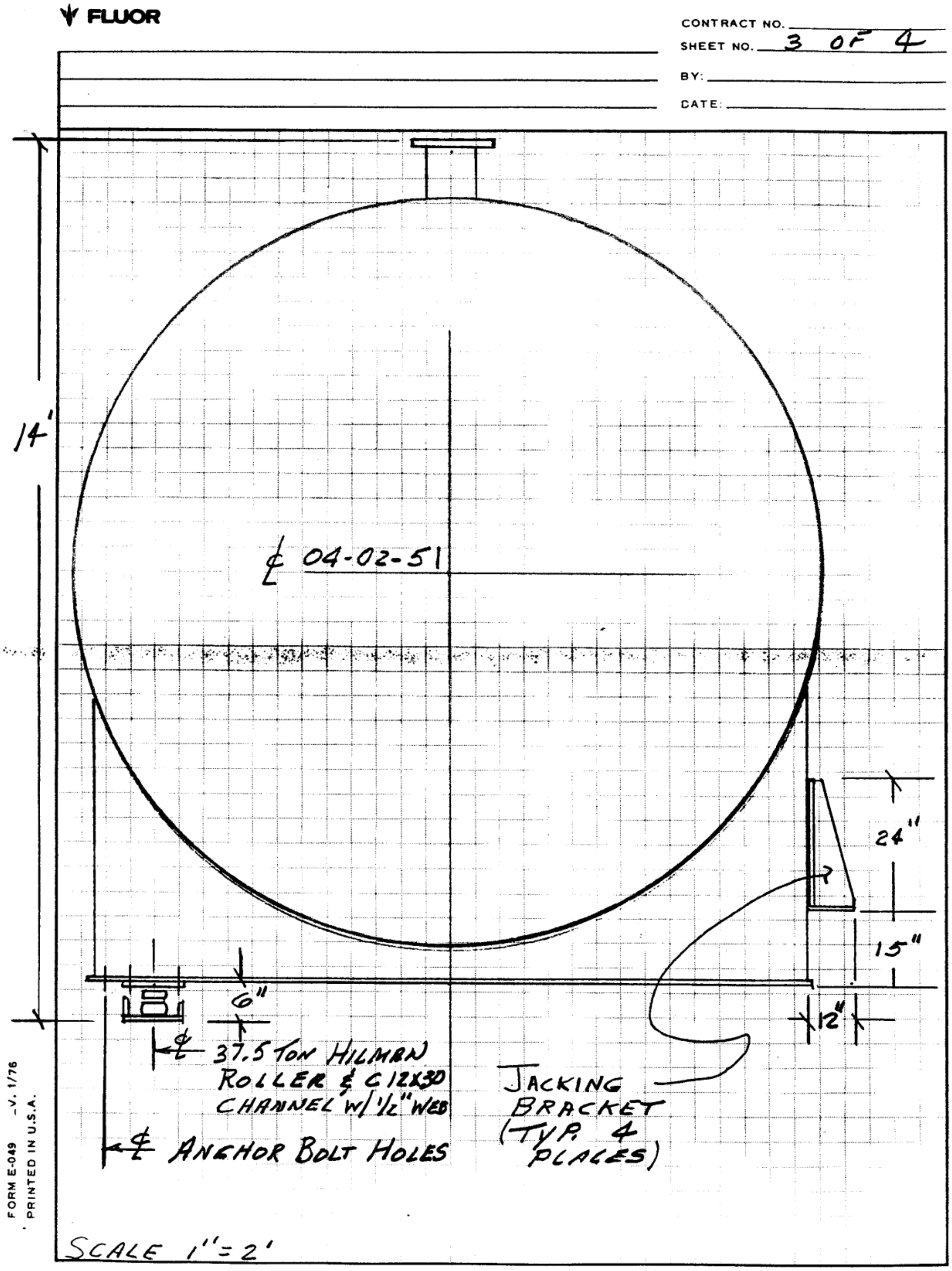
48.00	in	Area of the weld
192.00	in^2	Section modulus of the weld
3.02	kips/in	Resultant Force on the weld
0.20	in D	Minimum weld size

Checking bearing at the pin hole

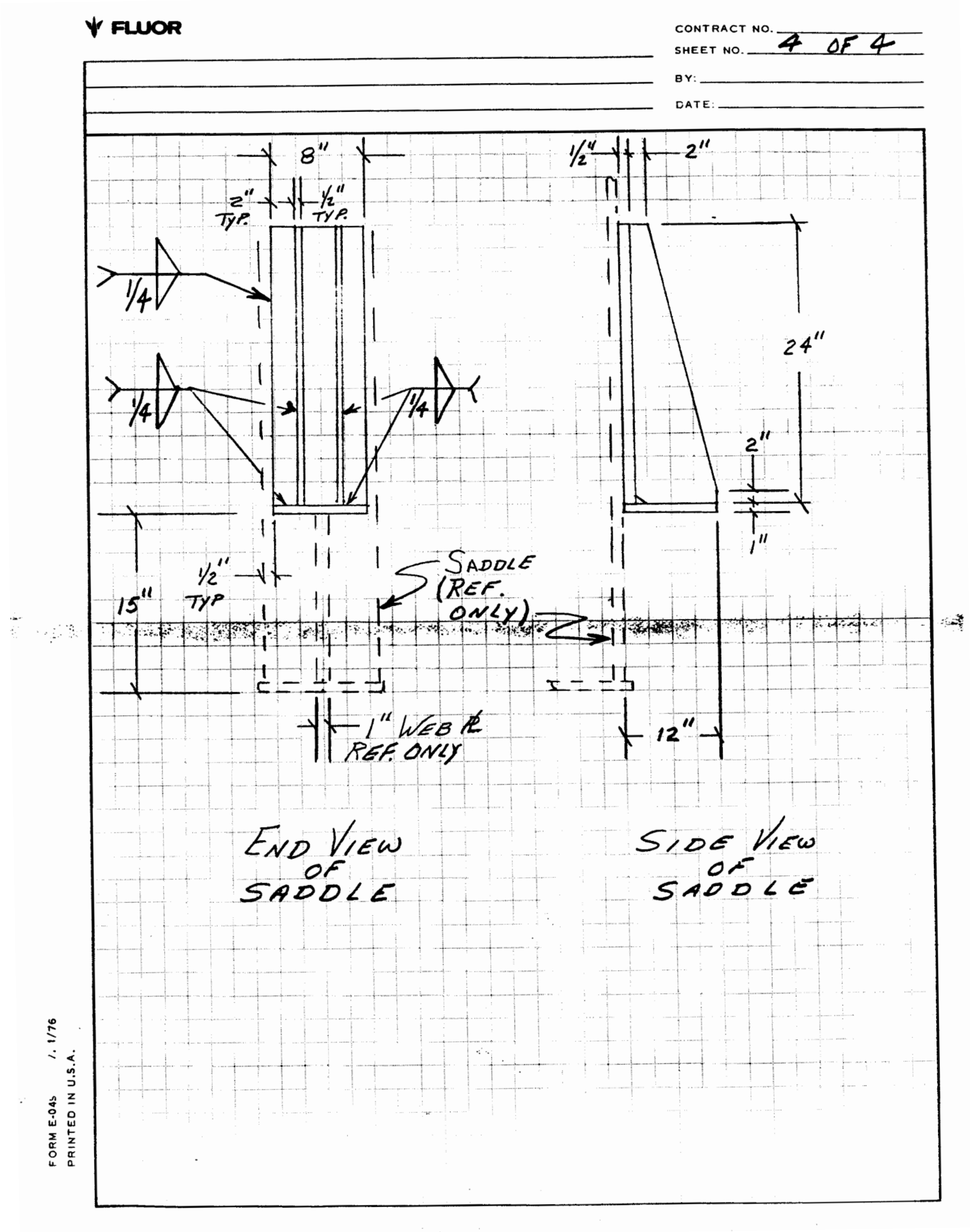
TO CHANGE THE IF
FROM 1.8 TO 1.25

$$D = \frac{0.2 * 1.25}{1.8} = 0.14''$$

See Appendix B for information from Blodgett's "Design Of Welded Structures", where he treats the weld as a line. See Part 7, section 7.4 "Determining Weld Size".



See Appendix C for information on Hilman Rollers.



END OF DESIGN:

APPENDIX A: Salmon & Johnson

The Intext *Series in*
Civil Engineering
Series Editor
Russell C. Brinker
New Mexico State University

Bouchard and Moffitt—**Surveying, 5th ed.**
Brinker—**Elementary Surveying, 5th ed.**
Clark, Viessman, and Hammer—**Water Supply and Pollution Control, 2d ed.**
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Ural—**Matrix Operations and Use of Computers in Structural Engineering**
Wang—**Matrix Methods of Structural Analysis, 2d ed.**
Wang and Salmon—**Reinforced Concrete Design**
Winfrey—**Economic Analysis for Highways**

STEEL STRUCTURES
Design and Behavior

CHARLES G. SALMON
Professor of Civil Engineering
The University of Wisconsin
Madison, Wisconsin

JOHN E. JOHNSON
Professor of Civil Engineering
The University of Wisconsin
Madison, Wisconsin

INTEXT EDUCATIONAL PUBLISHERS
College Division of Intext

except the overstress in view of the fact that using $e_s = 0.8W$ is believed to overestimate the moment arm.

Use $^5/_{16}$ in. weld, $L = 20$ in. Use stiffener plate, $^9/_{16} \times 7 \times 1'$-$8''$; and seat plate $^5/_8 \times 7 \times 1'$-$0''$. The seat plate width equals the flange width (10.46 in.) plus enough to easily make the welds (approx. 4 times the weld size is often used). The final design is shown in Fig. 13.4.5.

13.5. TRIANGULAR BRACKET PLATES

When the stiffener for a bracket is cut into a triangular shape, as in Fig. 13.4.3b, the plate behaves in a different manner than when the free edge is parallel to the direction of applied load in the region where the greatest stress occurs, as in Fig. 13.4.5. The triangular bracket plate arrangement and notation is shown in Fig. 13.5.1.

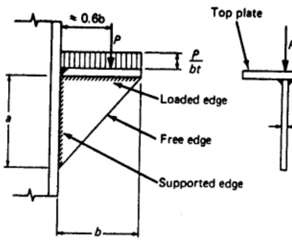

Fig. 13.5.1. Triangular bracket plate.

The behavior of triangular bracket plates has been studied analytically[9] and experimentally[10] and design suggestions have been proposed.[11] For small stiffened plates to support beam reactions there is little danger of buckling or failure of the stiffener if cut into a triangular shape. In general, it provides a stiffer support when so cut than if left with a rectangular shape.

Most Exact Analysis and Design Recommendations. For many years design of such brackets was either empirical without benefit of theory or tests, or when in doubt, angle or plate stiffeners were used along the diagonal edge. The recommendations presented here are based on certain assumptions: (a) the top plate is solidly attached to the supporting column; (b) the load P is distributed (though not necessarily uniformly) and has its centroid at approximately $0.6b$ from the support; and (c) the ratio loaded edge to supported edge, lies between 0.50 and 2.0.

The original theoretical analysis was concerned with elastic buckling; however, the experimental work showed that triangular bracket plates have considerable post-buckling strength. Yielding along the free edge frequently occurs prior to buckling, at which point redistribution of stresses occurs. A considerable margin of safety against collapse was observed indicating the ultimate capacity may be expected to be at least 1.6 times the buckling load.

The maximum stress was found to occur at the free edge; however, because of the complex nature of the stress distribution, the stress on the free edge is not obtainable by any simple process. Because of this difficulty, a ratio z was established between the average stress, P/bt, on the loaded edge to the maximum stress f_{max} on the free edge. The original theoretical expression for z[11] was revised as a result of the tests which conformed closely to what one could realistically expect in practice. The relationship is given[10] as

$$z = \frac{P/bt}{f_{max}} = 1.39 - 2.2\left(\frac{b}{a}\right) + 1.27\left(\frac{b}{a}\right)^2 - 0.25\left(\frac{b}{a}\right)^3 \qquad (13.5.1)$$

which for practical purposes may be obtained from Fig. 13.5.2.

If yielding controls strength, and $0.60F_y$ is considered a safe allowable

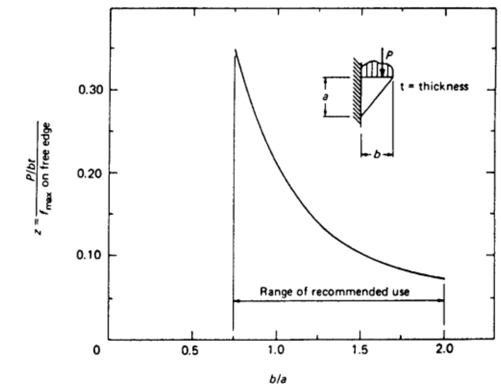

Fig. 13.5.2. Coefficient used to obtain maximum stress on free edge.

stress, the safety criterion becomes

$$f_{max} = \frac{P/bt}{z} \le 0.60F_y \qquad (13.5.2)$$

For stability controlling, the plate buckling equation, as discussed in Chapter 6, is

$$f_{cr} = \frac{k\pi^2 E}{12(1-\mu^2)(b/t)^2} \qquad (13.5.3)$$

where f_{cr} is the principal stress along the diagonal free edge. In order to guarantee that yielding is achieved without buckling, let $f_{cr} = F_y$ and

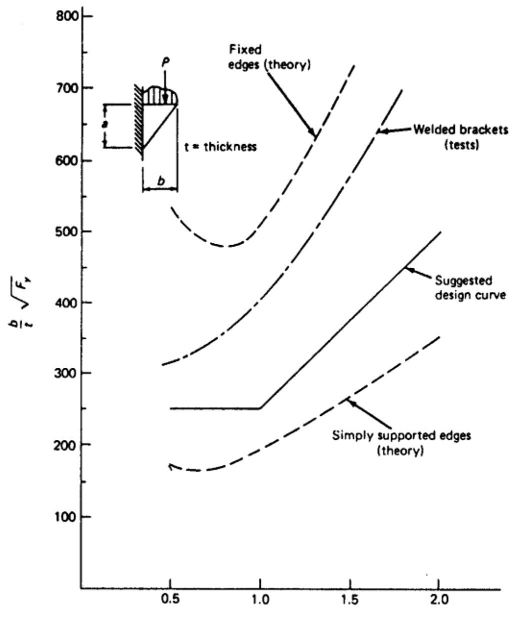

Fig. 13.5.3. Critical b/t values so that yield stress is reached along diagonal free-edge without buckling.

solve Eq. 13.5.3 for maximum b/t,

$$\frac{b}{t} \le \sqrt{\frac{k\pi^2 E}{12(1-\mu^2)F_y}} = 162\sqrt{\frac{k}{F_y}} \qquad (13.5.4)$$

where $E = 29{,}000$ ksi and $\mu = 0.3$.

Fig. 13.5.3 gives the variation in $(b/t)\sqrt{F_y}$ with b/a for the theoretical studies[9] (fixed and simply supported), the welded bracket tests result,[10] and the authors' suggested design curve. The design requirement may be expressed as

$$\text{For } 0.5 \le b/a \le 1.0; \quad \frac{b}{t} \le \frac{250}{\sqrt{F_y}} \qquad (13.5.5)$$

$$\text{For } 1.0 \le b/a \le 2.0; \quad \frac{b}{t} \le \frac{250(b/a)}{\sqrt{F_y}} \qquad (13.5.6)$$

Satisfying the above limits means that yielding along the *diagonal free edge* will occur prior to buckling.

It is noted that the b/t limits suggested here are higher than those of Ref. 11 (p. 552), which were based solely on the theoretical studies.[9] Reference 11 suggests a coefficient of 180 instead of 250 in Eq. 13.5.5 and reaches a maximum of 300 instead of 500 as indicated by Eq. 13.5.6 for $b/a = 2.0$. The reason for the higher values is found in the test results which showed the principal stress along the diagonal free edge to be lower relative to the stress on the loaded edge than had been established by the theoretical study. In other words, the z value, Eq. 13.5.1, as determined by tests is substantially smaller than assumed for the design suggestion of Ref. 11.

Approximate Beam Analysis for Stress. For many years stress on triangular, as well as other shaped, bracket plates has been based on a beam analysis (see Ref. 8) using the relationships from Fig. 13.5.4.

The combined stress at the free edge on the critical section is computed as follows:

$$f_{max} = \frac{P}{A} + \frac{Mc}{I} = \frac{P/\sin\phi}{bt\sin\phi} + \frac{(P/\sin\phi)(e\sin\phi)(b\sin\phi/2)}{(b\sin\phi)^3 t/12}$$

$$= \frac{P}{bt\sin^2\phi}\left[1 + \frac{6e}{b}\right] \qquad (13.5.7)$$

or, in terms of e_s, Eq. 13.5.7 becomes

$$f_{max} = \frac{P}{bt\sin^2\phi}\left(\frac{6e_s}{b} - 2\right) \qquad (13.5.8)$$

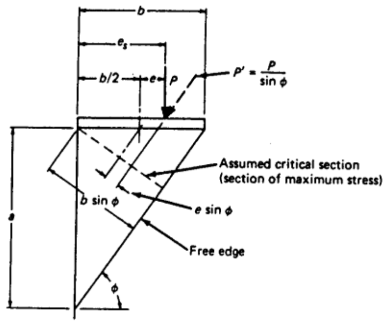

Fig. 13.5.4. Beam analysis for bracket plates.

If the maximum stress is limited to $0.6F_y$, the required thickness for stress is

$$t \geq \frac{P}{b\,(0.60F_y)\sin^2\phi}\left(\frac{6e_s}{b} - 2\right) \tag{13.5.9}$$

When the beam-analysis approach is used, one may imagine as suggested in Ref. 11, that a strip of width $(b \sin \phi)/4$ acts as a compression element. The problem then arises of what slenderness ratio limitation, if any, should be used. Ref. 11 gives no suggestion. The authors suggest that one may imagine a strip of plate acting as a column and use Eq. 13.5.4 with $k = 1.0$ for the pin-end plate; letting $b = L_d$, the length of the diagonal,

$$\frac{L_d}{t} \leq \frac{162}{\sqrt{F_y}} \tag{13.5.10}$$

Plastic Strength of Bracket Plates. Reference 11 suggests that to develop the full plastic strength of brackets the b/t ratios should be restricted to about $\frac{1}{3}$ of those limitations for achieving first yield on the free edge. The test results[10] indicated that ultimate strengths of at least 1.6 times buckling strengths could be achieved due to post-buckling strength. Probably to be certain of developing the plastic capacity of the bracket, it would be realistic to use half of the limitations of Eqs. 13.5.5 and 13.5.6.

To establish the plastic strength of a bracket plate used in rigid frame structures, one may follow the approach of Ref. 11, as shown in Fig. 13.5.5. This method assumes that plastic strength develops on the critical section. Equating the sum of the stresses normal to the critical section

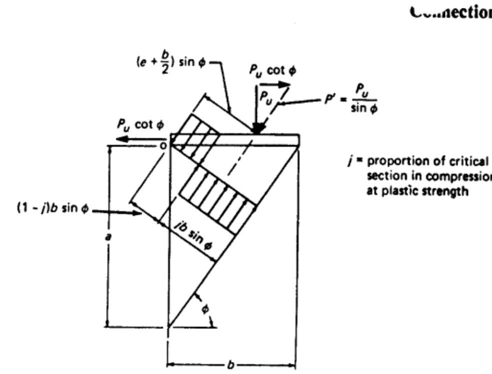

Fig. 13.5.5. Plastic-strength analysis.

to P' gives for $\Sigma F = 0$,

$$\frac{P_u}{\sin\phi} = jb\sin\phi\, F_y t - (1 - j)\,b\sin\phi\, F_y t \tag{13.5.11}$$

which upon solving for j gives

$$j = \frac{P_u}{2\sin^2\phi\, btF_y} + \frac{1}{2} \tag{13.5.12}$$

Moment equilibrium about point O gives

$$P_u\left(\frac{b + 2e}{2}\right) = \frac{b^2 t \sin^2\phi\, F_y}{2}\,[-2j^2 + 4j - 1] \tag{13.5.13}$$

Letting $A = P_u/(\sin^2\phi\, btF_y)$, Eq. 13.5.12 becomes

$$j = \tfrac{1}{2}(A + 1) \tag{13.5.14}$$

and Eq. 13.5.13 becomes

$$A\left(\frac{b + 2e}{2}\right) = -2j^2 + 4j - 1 \tag{13.5.15}$$

Substitution of Eq. 13.5.14 into Eq. 13.5.15 gives

$$A^2 + A\left(\frac{2e}{b}\right) - 1 = 0 \tag{13.5.16}$$

wherein

$$A = \frac{1}{b}[\sqrt{4e^2 + b^2} - 2e] \tag{13.5.17}$$

and using the definition of A gives for the ultimate load

$$P_u = F_y t \sin^2\phi\left(\sqrt{4e^2 + b^2} - 2e\right) \tag{13.5.18}$$

In addition to the triangular plate being adequate, the top plate must carry the ultimate force $P_u \cot \phi$.

EXAMPLE 13.5.1

Determine the thickness required for a triangular bracket plate 25 in. by 20 in. to carry a load of 40 kips. Assume the load is located 15 in. from the face of support as shown in Fig. 13.5.6, and that A36 material is used.

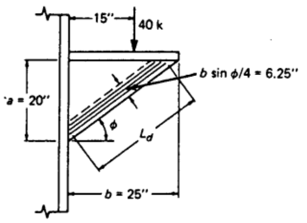

Fig. 13.5.6. Bracket for Example 13.5.1.

Solution

(a) Use the more exact method; working stress design. Since the load is approximately at the 0.6 point along the loaded edge, the bracket fits the assumption of this method. Using Eq. 13.5.2,

$$f_{avg} = \frac{P}{btz} = 0.60F_y$$

where from Fig. 13.5.2, for $b/a = 25/20 = 1.25$, find $z = 0.135$; for the yield criterion

$$t \geq \frac{P}{bz(0.60F_y)} = \frac{40}{25(0.135)22} = 0.54 \text{ in.}$$

For stability, using Eq. 13.5.6,

$$t \geq \frac{b\sqrt{F_y}}{250(b/a)} = \frac{25\sqrt{36}}{250(1.25)} = 0.48 \text{ in.}$$

Use $^9/_{16}$-in. plate.

(b) Use the approximate beam analysis; working stress method. Using Eq. 13.5.9,

$$t \geq \frac{P}{b(0.60F_y)\sin^2\phi}\left(\frac{6e_s}{b} - 2\right)$$

where $\sin^2\phi = (0.625)^2 = 0.39$

$$t \geq \frac{40}{25(22)0.39}\left(\frac{6(15)}{25} - 2\right) = 0.30 \text{ in.}$$

which by comparison with the previous method is not conservative.

Considering stability according to Eq. 13.5.10 gives

$$t \geq \frac{L_d\sqrt{F_y}}{162} = \frac{(\approx 27)(6)}{162} = 1 \text{ in.}$$

where L_d was taken along the center of the edge strip. The authors conclude that the beam analysis does not provide realistic answers; the stress requirement is low and the stability requirement is high, though past practice may well have used either a 1-in. plate, or an edge stiffener on a thinner plate.

Use 1 in. plate.

(c) Plastic-strength method. Use a load factor of 1.7. Using Eq. 13.5.18 for the stress requirement,

$$t \geq \frac{P_u}{F_y \sin^2\phi\left[\sqrt{4e^2 + b^2} - 2e\right]}$$

Using $e = 15 - 25/2 = 2.5$ in.

$$t \geq \frac{1.7(40)}{36(0.39)\left[\sqrt{4(2.5)^2 + (25)^2} - 2(25)\right]}$$

$$= \frac{1.7(40)}{36(0.39)(20.5)} = 0.24 \text{ in.}$$

For stability, using *one-half* of Eq. 13.5.6

$$t \geq \frac{b\sqrt{F_y}}{125(b/a)} = \frac{25\sqrt{36}}{125(1.25)} = 0.96 \text{ in.}$$

Use 1-in. plate as a conservative practice to assure deformation well beyond first yield along the free edge.

The authors note that if a 1-in. plate is adequate to develop the full plastic strength of the bracket, the resulting plastic strength would be about 4 times (1.0/0.24) the required P_u.

13.6. CONTINUOUS BEAM-TO-COLUMN CONNECTIONS

The connections are AISC Type 1—rigid frame connections. It is the design intent to have full transfer of moment and little or no relative rotation of members within the joint. Since the flanges of a beam carry most of the moment as a couple consisting of tension and compression flange

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BY

Omer W. Blodgett

THE JAMES F. LINCOLN ARC WELDING FOUNDATION
CLEVELAND, OHIO

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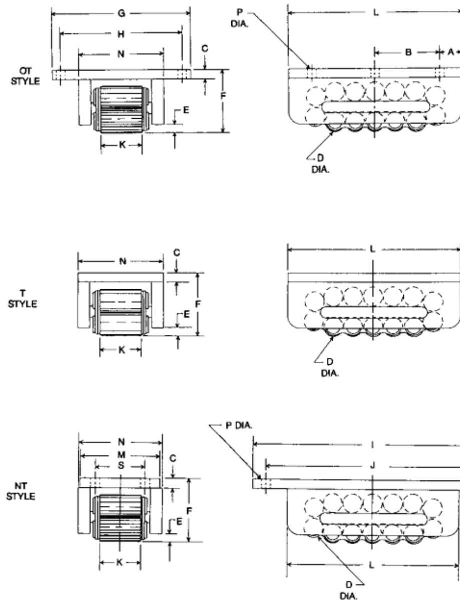
APPENDIX C: Hilman Rollers

OT, NT, T SERIES

• INDIVIDUAL ROLLERS .75-75 TONS

T Style NT Style OT Style



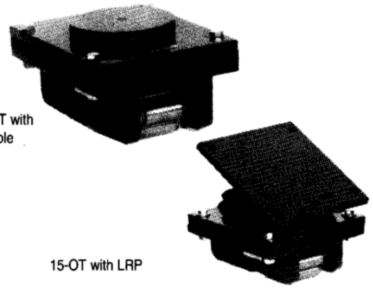
A variety of top plate styles, exceptionally strong materials of construction, and a wide range of modification possibilities make OT, T, and NT Rollers the true OEM leaders of the Hilman line! Rollers in this series are built for long life and a wide range of operating conditions.

For basic linear motion, these rollers are used as bearing slides, conveyors, guides, casters or dollies. Typical applications include the straight motion required for thermal expansion under a furnace or heat exchanger, as slides under nuclear shield doors, upside down as a conveyor in heavy die handling carts, in concrete forming for the casting and curing process as well in the launching of heavy segments, attached to a cradle for the launching of huge ships, or as a heavy duty wheel-like device to mobilize grandstand seats.

The OT type has an oversized top, the T type has a flush top, and the NT type has a long narrow top. The top plates add extra strength to the frame and variety to mounting capabilities, making one type ideal for cavity mount, bolt-on or weld mount. OT and NT Rollers are available with standard top plate sizes and mounting hole patterns, which can be modified to customer specifications. Hilman Elastomeric Preload Pads are available in either fabric impregnated or neoprene form for each roller model.

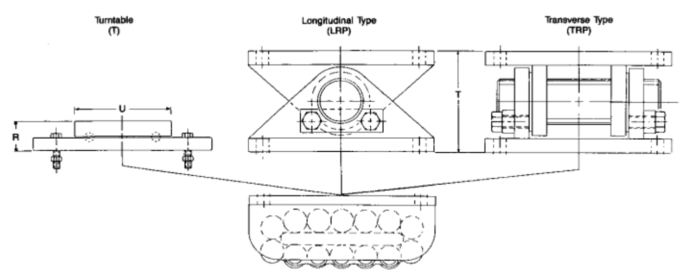
PRODUCT NUMBERS				DIMENSIONS (INCHES/MM)																		WEIGHT LBS / KGS.		
OT STYLE	NT STYLE	T STYLE	CAPACITY (TONS)	A	B	C	D	E	F	G	H	I	J	K	L	M	N	P	S	CONTACT ROLLS		OT	NT	T
75-OT	.75-NT	.75-T	.75	3/4 19	2-1/2 64	3/8 10	11/16 17	3/8 10	3-1/8 79	7 178	5-1/2 140	8 203	7-1/4 184	2 51	6-1/2 165	3-1/2 89	3-5/8 92	5/16 8	2 51	6		12 5	12 5	11 5
2.5-OT	2.5-NT	2.5-T	2.5	3/4 19	2-1/2 64	3/8 10	15/16 24	3/8 10	3-5/8 92	7 178	5-1/2 140	9-1/2 241	7-1/4 184	2-7/16 62	6-1/2 165	4 102	4-1/8 105	9/16 14	2 51	4		21 10	19 9	17 8
5-OT	5-NT	5-T	5	3/4 19	3-1/4 83	3/8 10	15/16 24	7/16 11	3-1/8 79	8 203	6-3/4 171	11-3/4 298	10-1/8 257	2-7/16 62	8 203	4-1/2 114	4-5/8 117	9/16 14	3 76	5		22 10	21 10	19 9
8-OT	8-NT	8-T	8	3/4 19	3-1/4 83	3/8 10	15/16 24	7/16 11	3-1/8 79	8 203	6-3/4 171	11-3/4 298	10-1/8 257	3-5/16 84	8 203	5-1/4 133	5-1/2 140	9/16 14	3 76	5		25 11	23 10	23 10
15-OT	15-NT	15-T	15	1-5/8 41	3-11/16 94	5/8 16	1-3/16 30	1/2 13	3-7/8 98	10 254	8-1/2 216	14-3/4 375	12-11/16 322	2-3/4 70	10-5/8 270	5 127	5-1/4 133	11/16 17	2-5/16 59	5		46 21	44 20	40 18
20-OT	20-NT	20-T	20	1-5/8 41	3-11/16 94	5/8 16	1-3/16 30	1/2 13	3-7/8 98	10 254	8-1/2 216	14-3/4 375	12-11/16 322	4 102	10-5/8 270	6-1/2 165	6-1/2 165	11/16 17	3-1/2 89	5		48 22	49 22	49 22
37.5-OT	37.5-NT	37.5-T	37.5	2 51	5-1/2 140	3/4 19	1-5/8 41	5/8 16	5-1/2 140	12 305	10-1/2 267	21 533	18-3/4 476	3-1/2 89	15 381	7 178	7-1/4 184	13/16 21	4-1/4 108	6		121 55	114 52	105 48
75-OT	75-NT	75-T	75	1-1/4 32	9-1/4 235	1 25	1-15/16 49	1/2 13	6-3/4 171	14 356	11-1/2 292	27 686	24 610	3-5/8 92	21 533	7-1/2 191	7-1/2 191	1-1/16 27	5 127	7		241 109	227 103	213 97

• OT, NT SERIES TURNTABLES
• OT SERIES ROCKER TOPS



37.5-OT with Turntable

15-OT with LRP



Turntables are available for OT and NT Series rollers in capacities from 5 tons to 75 tons. Turntables bolt to the base roller unit and allow the roller to swivel or provide some freedom of lateral motion in Accu-Roll applications where tracks aren't perfectly parallel. Connection hardware is included.

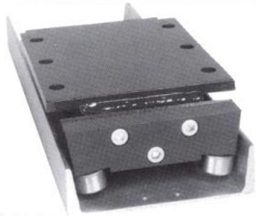
Transverse (TRP) and Longitudinal (LRP) Rocker tops for the OT Series contain two attachment plates connected by a pivot pin assembly. Rockers are designed to be bolted to the base roller unit, then attached to either the load or the surface, enabling the roller to follow an incline, or to allow a load to be tilted, launched, or more accurately positioned. Connection hardware is included.

PRODUCT NUMBERS TURNTABLES / ROCKERS				FITS ROLLER	DIMENSIONS IN (MM)						WEIGHTS TURNTABLES / ROCKERS			
T-OT	T-NT	LRP	TRP	MODEL	R		U		T		LBS.	KGS.	LBS.	KGS.
T-5-OT	T-5-NT	5-LRP	5-TRP	5-OT/NT	1-1/2	38	6	152	4-3/8	111	15	7	36	17
T-8-OT	T-8-NT	8-LRP	8-TRP	8-OT/NT	1-1/2	38	6	152	4-3/8	111	16	8	43	20
T-15-OT	T-15-NT	15-LRP	15-TRP	15-OT/NT	1-1/2	38	6	152	5	127	23	11	101	46
T-20-OT	T-20-NT	20-LRP	20-TRP	20-OT/NT	1-1/2	38	6	152	5	127	23	11	120	55
T-37.5-OT	T-37.5-NT	37.5-LRP	37.5-TRP	37.5-OT/NT	2-3/8	60	8	203	7	178	54	25	196	89
T-75-OT	T-75-NT	75-LRP	75-TRP	75-OT/NT	2-3/8	60	8	203	9-1/4	235	85	39	380	173

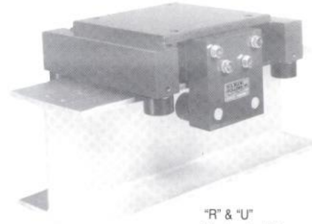
ACCU-ROLL SYSTEMS

- **EXTERNAL GUIDANCE**
- **UPLIFT PREVENTION**
- **ALIGNMENT INSURANCE**

Hilman's Accu-Roll guidance system, a positive external alignment option, can be added to any Hilman Roller as either a removable bolt-on or permanent weld-on system. In applications where load alignment is critical, or close tolerances, repetitive movement or uplift prevention are required, Accu-Roll often becomes an ideal solution to these heavy motion problems. Depending on the configuration selected, Accu-Roll will allow the roller to track precisely on a flat bar, I-Beam, rail, or in a channel or trench. Accu-Roll on an OT or NT Roller combined with a turntable can allow the roller to follow a somewhat skewed track. Accu-Roll Systems are custom made and entirely flexible; consult factory for recommendations.



"C" System for a channel



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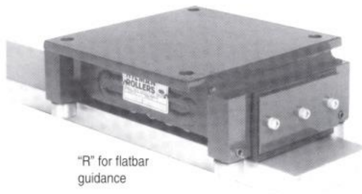
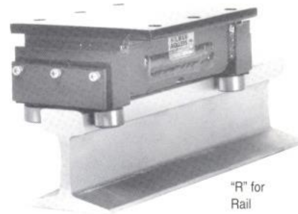


"R" for flatbar guidance



"R" for Rail

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