Basic Crane Design and Stress Analysis

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SECT I.1: NOMENCLATURE

As------------------Area in Shear
C-------------------Distance from neutral axis to fiber stress S, in.
Fb------------------Allowable Bending Stress
Fbr-----------------Allowable Bearing Stress
Ft------------------Allowable Tension Stress
FTK---------------Foot-Kips
Fv-----------------Allowable Shear Stress
I--------------------Moment of Inertia
Mt-----------------Rotational Torque
R------------------Radius
T------------------Torque
Sb-----------------Calculated Bending Stress
Sbr----------------Calculated Bearing Stress
$\sigma_v, \sigma_x, \sigma_{br}, \sigma_b$ --- Calculated shear, x, bearing or bending stress
May also be seen
St------------------Calculated Tension Stress
Sv-----------------Calculated Shear Stress
TS----------------Tensile Strength
YS----------------Yield Strength
Z------------------Section Modulus

SECT I.2: COURSE DESCRIPTION

This course demonstrates the use of various stress analysis procedures and methods used to design cranes or other lifting components. The methods used could also be applied to other structures or lifting devices. Note that some of the problem examples in this course use simplifying assumptions to reduce the procedures to hand calculation methods. The assumptions in this course have been supported by FEA analysis where noted. It's possible that these assumptions may not apply to all the ranges of applications or types of cranes possible. Hand calculations alone are not always recommended when simplifying assumptions are used and should be backed up by a FEA analysis and code standards (where applicable). <u>On the other hand, FEA's alone should not be used unless supported by some hand calculations to support any boundary/support assumptions used</u>

The course goes through the steps required to design components fabricated from aluminum. Further design criteria for aluminum can be found in the Alum <u>association standards and alum design manual. This course is not meant to take the place of these standard design practices. It is a prime example of the use of these practices. Other applications may require a further study of these manuals and design codes. The procedures used here could also be used for steel components along with the AISC standards.</u> The components in these examples are designed to somewhat conservative allowable stresses and standards.

The crane used as an example in this course is designed to SOLAS (safety of life at sea) standards and you will see this referenced in some of the verbiage (see section 1.0). A lot of mobile cranes you see are telescopic. Telescopic crane design may be discussed in part 2 of this course, if there is enough interest.

It is assumed that the student is somewhat familiar with design codes such as the AISC and this should not be considered a replacement for these codes.

It is recommended that the figures be opened in the separate pdf file so that both the figures and text can be viewed side by side. As the figures are such an integral part of this course.

SECT 1.0: BASIC CRANE RATING AND THIS EXAMPLE CRANE

Most cranes are rated by both the reach and the elevation angle of the load lifted. A load chart is developed, and you can often see these on side the crane. This course focuses on a crane's max capacity at a o deg elevation angle, which is normally the worst case. Calculations for other angles are beyond the scope of this course. But these capacities will most always be less. Ratings at higher angles of elevation will be governed by bi axial bending of the boom acting as a column with combined axial and bending loads. The charts on the sides of mobile cranes will define these capacities at various angles of elevation and reach.

This course uses the example of a crane rated for SOLAS (Safety of life at Sea) standards. These standards are certified by a body such as ABS (American Bureau Of shipping), DNV (Det Norske Veritas, a Norwegian company), ECB (European conformity Bureau) or other regulatory body.

The Solas Rating for this application is for 3500# = 2300# tender + 6 people @ 175# (Solas rated person)+ 150# allocated added allowance. The crane must rotate this load and launch the tender as required with the boat at 20 degrees list and 10 degrees of trim with the crane boom elevated up to max position when rotating against the incline.

Most mobile cranes are designed to operate on an incline of 5 degrees but will rate them for 3 degrees.

Load Case #1) Crane basic rating (non-Solas) is for 10,000# @ 38ft reach. Dose not need to rotate this load up an incline.

Load case #2) Crane basic rating (Solas) is for 3,500# @ 38ft reach.

See the next section for which load case governs the design

Table GA lists allowable design stresses which are discussed in section G1. See the list of references for further study for a more in-depth study of designing cranes.

This crane was actually built and used on a research vessel. See the following pictures in the figures document. The design of the rotation test stand was a project onto itself.

- R0TATION TEST: Tested at 23degree incline angle
- ROTATION SLEW, WORM GEAR DRIVE
- FINAL ROAD TEST

Notice that it took 4 worm drives to rotate the load at the 23-degree incline and required speed!!

SECT G1: TENSILE AND ALLOWABLE STESSSES

Table displays the allowable design stresses used in the following design and other typical materials in KSI.

TABLE GA

	TS ksi/sq in	YS ksi/sq in	ABS (#2) Allowed Fb/Fv ksi/sq in	SOLAS (#3) Allowed Fb/Fv ksi/sq in	SOLAS Allowed Fbr ksi/sqin (#1)
Material					
5083-H116	44	31	15.5/9.77	9.77/5.98	17
5083-H116 by welds	40	24	14.4/8.88	8.88/5.3	N/A
5086-H116	40	28	14.4/8.6	9.33/5.57	12
6061-T6	45	40	16.2/9.7	10/5.98	19
6061-T6 by welds	29	17-20	10.4/6.26	6.44/3.87	N/A
					26
316 Stainless	75	30	15/9	16.66/9.9	21.6
17-4 PH Stainless	145	125	52/32	32/19.7	41.6
ASTM A-36 Steel	58	36	21.6/14.4	12.8/8.5	26

Note #1) SOLAS allowable bearing stress, from Ref 8; Aluminum Association Standards, Table 3 Engineering Data for Aluminum Structures. Then reduced to meet SOLAS standards

Note #2) Reference #1 was used to obtain the ABS values

Note #3) SOLAS

SOLAS stands for Safety of Life at Sea and applies to rescue craft aboard larger vessels. These higher safety factors may be considered for applications of cranes or davits in emergency situations where the vessel may be sinking and heeling over. The API standards Ref 7 could be consulted for higher dynamic and wind loads.

The ABS values could be used for most cranes in industrial settings where occasional or moderate daily use of well-defined loads is normal. Industrial safety factors can range from 2 to 3. For 6061-T6 that would be from 45/2=22.5 to 45/3=15. The ABS values are more on the conservative side of that spectrum.

Note the large difference in allowable stresses by welds for 5083 vs 6061-T6

The SOLAS standard safety factor is ultimate strength/4.5.

This example crane is dual rated. One rating for the SOLAS load of a 3500lb rescue Tender and 10,000lbs for a std crane capacity. Both at 38Ft of each

For this crane which load case requirement will govern the design?

10,000 x 38 (Load case 1) > 3500 x 38ft (Load Case 2) by 2.85 times

A STD crane design can handle 1.58 times more bending stress then a SOLAS crane. As the 15ksi ABS allowable bend stress for 5083 is 1.58 times more than the 9.77 ksi SOLAS allowable from Table GA, above.
As 1.58 x 3500=5530lb is < 10000lbs, the 10000lb capacity will still govern most all of the design and the ABS allowable stresses for a basic crane will be used.

SECT M: MAJOR MECHANICAL COMPONENTS

These components are generally purchased and have longer lead time items.

M1.0 Actuator for cranes on fairly level surface

A rotary actuator is often used to rotate the crane turntable bearing and is mounted at the centerline of the turntable bearing. Industrial cranes will generally use a pinion gear with a hydraulic motor. The rotational torque required would be the same for both types of rotation devices at the turntable bearing. If a pinion is used the gearbox torque required would be a function of the gear ratio between the pinion gear and pitch diameter of the turntable bearing. So, it would be less torque required at the pinion or the gearbox driving the pinion, But the pinion RPM would need to be more

Frictional torque on a rotary turntable bearing on a fairly level surface should be about 4.5 to 5% of the moment for a boom elevation of 0 degrees. As test measured on a 1500lb capacity crane in a shop I was in. Actuator torque values at this level will see periodic problems with rotating the load. Published values are available from manufacturers which may provide different variables. A 'fairly level" surface is generally below 1 to 1.5 degrees of list. In order to operate satisfactory, the min torque should be about 7-8% of the live load and dead load for cranes under 100FTK of moment. Standard Mobile truck mounted cranes are also designed to operate at +/- 5 degrees but are generally not allowed to operate above +/- 3 degrees. Above 100FTK the dead load of the crane must also be considered. Note that manufactures of pinion gear turntable bearings list larger recommended values for mobile cranes which are generally operated on inclined grade. Plus, additional safety factors depending on the application and how hard the crane is used. These catalogs will need to be consulted to finalize a turntable bearing selection

For grade inclines above 1.5 degrees, the torque required to rotate the crane arm up the incline will quickly become the dominate factor in selecting a rotation device. As will be seen in section M1.1

Mobile cranes will have a min incline that they must operate in. This could be 5 degrees or more. This Torque load must be added to the 7-8% for a level surface.

M1.1 SOLAS Rotational Torque on inclined surface

Rotating a crane on a larger incline must overcome the rotational torque of the incline which has a force of:

Pt= W x sin (theta),
where Pt = force in-lbs., W= Vertical load, theta= angle of incline.

R= Horizontal reach from the centerline of a vertical axis thru the center point of the turntable bearing to the vertical axis of the suspended load. The max torque will occur when boom is at the slew angle showed on the figure

see figure (M1.0-A Crane Rotational Torque) to visualize these distances

The Live load Moment on the base can be estimated to be about:
ML=live load moment=LL (Live Load) x Re,

The dead load moment is DL x R/2

R/2 is the approximate distance to the CG of the crane. For longer telescopic cranes the actual CG distance should be used

DL= dead load.

The actual moment on turntable bearing will be a function of the angle of incline and rotational slew angle of the crane boom and its angle of elevation. The moment load on the crane will actually be a function of W x cos theta, so the formula as shown is a bit conservative. This moment only contributes to the rotational torque required thru the friction in the turntable bearing, which can vary with lubrication and bearing design.

The example crane used in this course is a SOLAS crane and requires an incline of a 20 List and 10-degree trim, for a total incline = SQRT (20x20 + 10x10)=22.36 degree (say 23).

Live Load = 5380#, LL= 5380

DEAD LOAD, DL=5272#

Horizontal Reach R= The max required torque will occur when the slew angle is at zero degrees as shown in FIG. M1.0-A. At this angle the effective torque arm (Rt) is the longest

The General formula for total Maximum rotational torque required would be:

Rt = Pt x R_e + (0.07 x (ML + (DL x R/2))   (on a level incline)

For the general case of any incline
Rt= (W x SIN 23) Rt + DL x SIN 23 RPCG + (.07 W x Re) + (.07x DL x RPCG)

Values with .07 applied are for the friction loads on level surfaces accounts for an incline of about 3 degrees

Where
Rt= Torque arm of load W x sin(incline)
R= reach with Boom at 0 deg elevation
Re=reach with Boom at some elevation angle
Horizontal Reach R=38FT
DL= dead load of crane=5272lbs
W= live load lifted, 3500lbs for Rescue craft
Pt = Rotational torque at end of crane as show in (fig M1.0-A)
Pt= W x sin23.
23=angle of incline

RPCG= horizontal distance to CG of DL, assume Re/2.84=(38x12)/2.84=160”
0.07 approximate friction factor to turntable bearing

For this case assume Boom elevation is 0 deg, and slew angle is 0 degrees.
so Re=R=Rt.

So
Rt= (3500 x sin23 x 456) + (5272 x sin23 x 160”) + (.07 x 3500 x 456) friction + (.07 x 5272 x 160”)
Rt=623606 + 329589 +111720 + 59046

Rt=953195 + 170766 (from friction)
Rt=1123,961 in-lbs.

Note that friction Torque is only 15.1% of the total required torque. In practice the boom elevation needs only be about 45 degrees to rotate the rescue tender over the side of crane base to the tender cradle. So, the above is very conservative.

The friction torque would be the expected rotational torque required for a crane on a fairy level surface. Or a max incline of 1 to 2 degrees.

The rotational device required to deliver this torque can be seen in the figures document. This was made out of set of 4 worm gears.

M1.2 Rotational Torque for standard non-Solas crane

A standard crane designed for +/- 5 degrees would have this version of the formula described in the preceding section

Rt= (W x SIN 5) Rt + DL x (SIN 5)RPCG + (.07 W x Re) + (.07x DL x RPCG)

For this example, crane, the non-SOLAS rating is 10,000lbs

W=10,000lbs, Rt=38ft/456”, Re=Rt=456”, DL=5272, RPCG=160”

This gives

=397430 + 73517 + 319,200 + 59046
=849,193 in-lbs.

This is 1123,961/849,193=1.32 times less than the rotational torque of the above 3500lbs on a 23degree incline. Even with the load at 5 degrees being 10,000/3500=2.85 times less.
So, the SOLAS requirement dictates the design of the rotation system. In many cases the SOLAS crane boom can be at an elevation of 45 degrees and still perform its function. It does not have to rotate the load at a boom elevation of 0 degrees up the 23-degree incline. In this case the SOLAS rotational Torque would be

Sin 45 x 1123961=794760 in-lbs.

In this case the standard rotational torque is 794760/849193=0.935 0r 93% of the non-Solas rotation torque for this example. So, using a reach of Re=R (at 0 degrees) is conservative

<u>M2.0 Boom Cylinder (boom cylinder force)</u>

(SEE FIG M2.0-A):

This crane has a max capacity of 10,000lbs. which does not need to be rotated on a 23 deg incline as the 3500lbs rescue craft.

The boom cylinder force required;

Ph x Dh = Cap (Rp) + DL x Rp/2.56=M_{TOTAL}=M_{LL} + M_{DL}

Ph= the required hydraulic force
Dh= the hydraulic Arm, perpendicular distance from main boom pivot pin to axis of hydraulic cylinder
Dh= 35.5” (<u>See Fig M2.0-A</u>)
Rp= reach from cl of rotation plus distance from cl of rotation to main pin
Rp= 38ft x 12 + 21.5”
Rp= 477.5”

Ph x 35.5 = (10000 x 477.5) + (5272# x 477/2.56
Ph x 35.5= (397.9 FTK + 81.86 FTK) x 12=479.76 FT-Kips x 12in/ft
So M_{TOTAL} =479.8FT-Kips on level surface
Ph= 162.2 KIPS REQUIRED
(721.501.6 N)

A 9” bore cylinder has Ph=190.8 kips @ 3000 PSI
159 kips @ 2500 PSI
A 10” bore cylinder has Ph = 235 Kips @ 3000 PSI
Ph = 196 Kips @ 2500 PSI
(871,851.5 N @ 17,237.5 Kilopascals)

A 10” bore was chosen, in order to be on the safe side.

Hydraulic Power units HPU’s:
This would be a course in itself but a few simple guidelines are:
If a cylinder is to be rated for 3000psi, it is recommended to design at 2500psi for the max load, to allow for pressure losses and pressure used by any load holding counterbalance valves. These valves are most always used to hold the load if no pressure is applied to the extend side of the cylinder. The actual pressures measured are often close to the design pressure but can exceed this by a few 100psi. It would be a problem if when testing the crane, you had to increase the pressure of the system to 3100psi to meet the required crane capacity at 0 degrees of elevation. Or even

worse if you bought a 3000psi hydraulic power unit and could not increase the pressure. A power unit should deliver abut 3300 to 3500psi for a 3000psi system. The pressure relief needs to be set about 300psi above any pressure compensator setting on the HPU. The compensator would be set for 3000psi and the relief for 3300psi. Unless it's critical the 300psi never be exceeded.

M2.01 Rod Load Boom Cyl

(IF AT 72 DEG BOOM ELEV. AND 0 DEG HEEL)

Max column load @ 72 deg is (**Ref FIG M2.0-B**):

$$Ph \times 42.60 = 10000\# \times 128 + 5272 \times 64$$
$$Ph = 38.0 KIPS$$

169,032.4 N

For 4" Dia rod in cyl, $R = (I/A)^{1/2} = 0.7088$

$$I = \tfrac{1}{4} \times \pi \times r^3 = 6.28; A = 12.5 in^2$$

$K \times (L/R) = 1 \times (137.13/.7088) = 193.5$; (137.13" PIN TO PIN EXTENDED)

Sallow=3y.99 KSI, (Table 1-36 AISC, Fy=36 STL ROD)

P allow=S allow$\times$12.5=49,875# > 38 KIPS

Other Boom elevation angles could also be investigated for rod column loads. The allowable load is a function of L (Pin to PIN) squared and Reach x load. L squared is generally the predominate factor. You can prove this, if desired.
Note that most cranes are rated for higher capacities at higher angles of elevation. A crane load chart will usually be seen on the side of a mobile crane. In this case the crane is rated 0 degrees of elevation, for the sake of simplicity.

To apply a higher rating the Boom would need to be analyzed for combined column and bending per AISC section 1.6 for use of the interaction formulas. In order to properly use the AISC manual I highly recommend "Basic Steel Design" by Bruce G. Johnston and others. This book breaks the AISC guidelines into easy-to-follow flow charts. This could be added to section B1.0-B 5.0 but was not at this time.

M2.1 Boom Cylinder Tube, Rod End & Cap End

TUBE END ROD SIDE (SEE FIG M2.1-A, & -B, & -C)

- Sb1 by pipe to rod (for 5.25ODx3"ID pipe end) Near Pt 1.

$$Sb1 = (1.07 in \times 81100 lb \times 2.62 in)/33.31 in^3 = 6.825 KSI < 15; \underline{\text{OK}}$$

Where Sb1=MC/I

C = 2.62 & 33.31 are shown in Fig M2.1-B

<u>TUBE ON CAP END</u> (<u>SEE FIG M2.1-D</u>)

$$Sb = (2.00in \times 81100lb \times 2.62in)/33.31in^3 = 12.8KSI < 15; \quad \underline{\text{OK}}$$

Where Sb=MC/I

C = 2.62 & I=33.31 are shown in Fig M2.1-B, (This section modulus is same as on Rod End.)

<u>M2.2 Boom Cylinder Pins, Rod End & Cap End</u>

(SEE FIG M2.1-D & -E & -F)

<u>3” pin</u> 17-4PH stainless

Stress for Boom Cylinder Pin Cap End.
(See Figure M2.1-E)

Note: Figure M2.1-E represents FEA performed for 50,000 lb. loading
However, results will be linear.

Therefore, Pin stress = (81,000/50,000) x 18,870 = 30,570 psi. < 52 KSI (TABLE GA, Fb Allow) There will be similar allowable for 4340 case hardened pins. Thru hardened pins may have higher tensile stress but do not handle the impact loads as well and are not normally used. A Cold drawn rod of 4340 will have a higher Ultimate stress and still have good impact strength.

Stress for Boom Cylinder Pin Rod End.
(See Figure M2.1-F)

Note: Figure M2.1-F represents FEA performed for 50,000 lb. loading
However, results will be linear.

Therefore, Pin stress = (81,000/50,000) x 27,431 = 44,438 psi. < 52,000 psi TABLE G2 Fb Allow)

<u>M2.3 Cylinder Shell Pressure Stress (Boom Cyl.)</u>

In general, the manufacturer of the cylinder will work out these parameters out. But in some cases, these values will need to be checked. Such as in this case, which requires a SOLAS certification or comparing manufacturers specifications. The ABS documents are available for free on line and can be downloaded to review these calculations. The below is a summary of these cylinder checks

Using ABS STEEL VESSELS PART4
2003 PART4-4-1A1/3.1
EQ1. (AS GUIDE)
For W< $0.385 \times SE$

$3000\text{PSI} < 0.385 \times (18{,}888) \times 1 = 7271$

Where W=operating psi
SE=allowable stress
T=thickness of metal
R=d/2
d=Cylinder bore

For t<R/2 use the formula below
t<5/2=2.5, so should definitely be less than 2.5 and below formula can be used.

$$T = [W \times R/(\delta \times S \times E - (1 - Y) \times W)] + C$$

δ=1, W=3000 PSI, S=85,000/4.5 (SOLAS)
Y=.4, C=0, E=1 SEAMLESS, R=5.0
C=0 was used as our S.F. is already 4.5 vs ABS or 3.5 and ABS issued as a guide.

$T = [3000 \times 5.0/(18888 - ((1 - .4) \times 3000))] + C$
T=.878 Use .1.00" to .878" wall (22.3 mm)

<u>HEAD THICKNESS</u> (CAP END)
Per EQ(4) sect 4-4-1A/sect 5.7.3

$$T = d \times ((K \times W)/(\delta \times S))^{0.5} + C$$

FOR d=10, K=.33

δ =1, S=18,888=85000/4 C=N/a

$T = 10 \times ((0.33 \times 3000)/(1 \times 18{,}888))^{0.5}$
$T = 2.29$ in. req. 58.17 mm.

<u>HEAD THICKNESS</u> (ROD END)

Per ABS 4-4-1A1/7.3.2, can use same formula, but with k=.7.
For case N, k=.7

$T = 10 \times (0.7 \times 3000/1 \times 18{,}888)^{0.5}$
$T = 3.33$ in. req.

M3.0 Cable Cylinder

(Also see Fig M2.0-A)

Most cranes will use a winch, which can be purchased. A few cranes will use a winch cylinder which has 3 sheaves mounted on its rod end and 3 sheaves mounted in the crane head. See Fig M2.0-A. The wire rope is wrapped around these sheaves, so there are 6 lines between the sheaves and on the line holding the lifted load. As the rod extends 1ft the load lowers 6Ft. But the load on the cylinder is 6+ times more than the lifted load.

8"BR x2.5" ROD
Phydrl@3000 PSI=150,769# PUSH /136,070# PULL

Rod Load (due to pull) is

Pull=10000#× [$6^{(1.05)}$+cable head/track frict]
=10000#×{6^1.05+2.12]
Pull Req=87000# @1918PSI (See note #1, below)
(386,995 N @ 13,224.6 Kilo-pascals)
The 1.05 factor account for friction

M3.1 Rod Stress (Cable Cylinder)

Operating Stress @ rated load of 10,000lbs
(Least Dia. Of Rod = 1.75 Dia. At the nuts for 2.5" DIA ROD)

St= 87,000/2.4 in^2 = 36,250 PSI < 52 TABLE GA

Solas load = 3500#

Rod stress:

Pull=3,500#× [$6^{(1.05)}$+cable head/track frict]
=3,500#×{6^1.05+2.12]
Pull Req=30,388# @670 PSI (See note #1)
135,172.6 N @ 4619.7 Kilo-pascals)

Operating stress @ rated
St=30,388/2.4 in^2=12661PSI < 32 KSI allowable Table GA 17-4

NOTE #1
Cable cylinder pulls in Sect M3.0 of 87,000# is conservative as this value includes hydraulic press losses in piping. Std procedure is to use 5% per sheave per Lloyds "provisional rules for launch & recovery appliances for survival boats & rescue craft" this procedure gives.
$6^{(1.05+0.05)} \times 10000 = 71{,}773.9\#$.

M3.2 Hydraulic Cable Cyld Ear

CABLE CYLD EAR SHEAR STRESS (SEE FIG M3.2-A) and M3.0 above

Sv=87000 max/2)/1.375 x 1.938 x 2)=8162 PSI (SHEAR STRESS) < 75/4.5=16,666 OK

Sbr=43,500/(1.375 x 2.5)=12.7 KSI OK

St (tension/hoop) ≈ Sv

FIG M3.2-A 2D FEA supports above hand calcs for Cylinder ears

M3.21 Cable Cylinder Alum Mount Plate

\`$S_v = {}^{87000}/_{(4x2x2)} = 54300$
The shear value in 2D FEA Fig M3.2-B supports this.

The S_y Stress has additional bending causing close to the 8000ksi stress shown in Fig M3.2-B.

The bend moment in front of the pin is about:

$\frac{wl^2}{8\ to\ 12}$ =34.8 for 10

W=87,000/4=21.75Kips
l=4,
say 10 for denominator as is semi-fixed across the 4" length

$$S_b = \frac{M}{S} = \frac{21.75}{2.66} = 8176psi$$

S=1/6 b x t^2=1/6 x 4 x 2^2=2.66

This agrees with the fig and shows that bending stresses around pin holes cannot be ignored

M3.3 Shell Pressure Stress (Cable Cyl.)

Using ABS STEEL VESSELS PART4
2003 PART4-4-1A1/3.1
EQ1. (AS GUIDE)
For W< $0.385 \times SE$
3000PSI<$0.385 \times (18{,}888) \times 1 = 7271$

Or for t<R/2

t<5/2=2.5, so should definitely be less than 2.5.

$$T = [W \times R/(\delta \times S \times E - (1 - Y) \times W)] + C$$

δ=1, W=3000 PSI, S=85,000/4.5 (SOLAS)
Y=.4, C=0, E=1 SEAMLESS, R=5.0
C=0 was used as our S.F. is already 4.5 vs ABS or 3.5 and ABS issued as a guide.

$$T = [3000 \times 4.0/(18888 - ((1 - .4) \times 3000))] + C$$

T=.702 Use .75 WALL (19.05 mm)

<u>HEAD THICKNESS</u> (CAP END)
Per EQ(4) sect 4-4-1A/sect 5.7.3

$$T = d \times ((K \times W)/(\delta \times S))^{0.5} + C$$

FOR d=8, K=.33

δ =1, S=18,888=85000/4 C=N/a

$$T = 8 \times ((0.33 \times 3000)/(1 \times 18{,}888))^{0.5}$$

$T = 1.83$in. req. 46.48 mm.

<u>HEAD THICKNESS</u> (ROD END)

Per ABS 4-4-1A1/7.3.2, can use same formula, but with 2*k for 2k less than .7.
For case N, k=.3; 2k=.7

$$T = 8 \times (0.7 \times 3000/1 \times 18{,}888)^{0.5}$$

$T = 2.67$in. req.
67.81 mm.

<u>M4.0 Main Turn Table Bearing</u>

The moment capacity of the main turntable bearing raceway and bolt pattern is normally specified by the bearing MFG. This section shows how to check the size and number of bolts used by the MFG.

Nomenclature:

D--------------------------Diameter of Bolt Pattern
H--------------------------Dead Load + Live Load (5272+10,000)
M(calc)-------------------Calculated Moment
M(allow)-----------------Max. Moment load allowed
N-------------------------Number of Bolts
P(max)-----------------Max. Bolt Load Allowable

Pb-----------------------Bolt load max operating
SF-----------------------Safety Factor = 3.5
DL------------------------Dead Load
DLCG-------------------Dead Load CG apox= R/2.56
R-------------------------Reach 38ft

MOMENT on main bearing is:
M=Capacity x R + DL x DLCG
M(calc)=(10,000 lb. x 12 in x 38 Ft) + ((5272 lb. x 12 in x 38 ft)/2.56)
M(calc) = 458.3 ft-kips (621,363 N*M) (Max. Operating)

MAX ALLOWABLE BOLT LOAD for 7/8 Bolt, .77" DIA at thread:

P(max)/SF=Max bolt load allowable for 7/8 Bolt

P(max.)/3.5=((.469 in^2) x 140 KSI)/3.5=18,760 lbs.

P(max) = 18,760 lbs. (Allowable)

Bolt load operating is Pb=3M/(D*N)

--For Top Pattern to Starship, the outer bearing pattern:

M(allow)=(Pmax + H/N)x(ND/3)
M(allow)=(18,760+15,272/24) x (24x41.25/3) = 6,400,790 in-lb.
533.4 ft-kips >458.3 ft-kips

-As M(allow)/M(oper) = 533.4/458
Max Bolt Load Operating = 458/533.4 *(18.7) = 16 Kips

Bolt Stress:
Sa = 16/.465 = 32 Ksi < 45 Ksi per Table G2

--For Bottom Inner Pattern to Base--

M(allow)=(18,760+15,272/30) x (30x33.67/3) = 540.6 Ft-K. (732,955 N*M)

Max Bolt Load Operating = 458/540 *(18.7) = 15.91 Kips (on Base Bolt Pattern) (70,771 N)

These are approximate formulas for sizing bolts that can be found in various catalogs. The manufacturers catalogs will have load charts for moments and various bolt sizes and number of bolts. The above calcs are intended to serve as a checks and the manufacture's catalogs should always be consulted. The length of the bolt will also influence the load sharing between bolts on the tension side of the bearing. Some uses will use cylindrical spacers to increase the length of the bolts and improve load sharing.

It is important to use the application factors recommended by the MFG in sizing bolts and raceway size and diameters. To account for impact loads, vibration, inclined loads, winds and other factors. A crane rated for 4000lbs in a fabrication shop would not be rated for 4000lbs on an oil rig

SECT ST: STARSHIP SECTION

ST1.0 Main Starship Ears Profile & Thickness

BENDING STRESS

The stress at the rear of the starship ear is the most critical stress area in the Starship. Welds must be perfect in this area, or they will crack.

Load on each ear is Boom CYLD load/2= 162.5/2=81.25KIPS (see pg. xx) and (FIG ST1.0 STARSHIP, MAIN EAR PROFILE)
Approx. max tensile stress
Sb= M C/I see (FIG ST1.0) for values and dimensions
S_b=((33.5"× 81.25KIPS)18.59)/8562=5.9 KSI <14.4ksi for 5083 by welds

This value is somewhat confirmed by (fig ST1.0-A) with the AutoCAD Mech 2D FEA from. S_B in Y axis≈8000-8200 PSI, away from the lower rear edge of the ear.

This is why FEA's should be done on main ears and other critical areas. The simplified hand calc is not enough to estimate the 3D stresses happening in this area. There will be doublers in this area to compensate for the stress concentration factors in this area. Even this 2D FEA is not fully accurate and too conservative. See section ST3.0 for the Horizontal Plate and doublers in this area

ST1.1 Shear Stress By Main Pin Hole Dblr in Main Large Ears

$v_{s.}$=81.25KIPS/(2×W×t)=81.25/(2×3.7"× 2")=5489 PSI < 15.5KSI Table GA

(FIG. ST1.0-A) confirms Shear Stress is ok, with shear stress less than 7ksi Across the assumed shear plane distance of 3.7" and 2" thick metal. Note that 81.25 is the load per ear.

BEARING STRESS in main pin hole of Main large ears

S_{br}=81.25/((2")x 4.0))=12ksi < 17ksi bearing allowable on aluminum from able GA

S_{br}=81.25/((2")x 3.5))=15.5ksi <25 ksi allowable for composite CJ of 3.5ID/4.0OD

See (fig ST 1.0) for the diameters of the hole in this area

BEARING STRESS in main pin hole of Boom

S_{br}=81.25/((1.25+0.5)x 4.0))=13.8ksi < 17ksi bearing allowable on aluminum from able GA

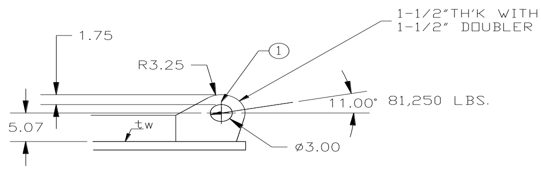
S_{br}=81.25/((1.25+0.5)x 3.5))=17.8ksi <25 ksi allowable for composite CJ of 3.5ID/4.0OD

The diameters of the hole in this area are the same as in the large ears

ST2.0 Stainless Boom Cyld Mount Ears

BEARING STRESS in Boom Mount Pin hole

S_{br}=81,250/(3×3)=9027.78 PSI<21.6 Table GA for stainless 316 See (Fig ST2.0), here and in figures secton



ST2.0
BOOM CYLINDER MOUNT

SHEAR STRESS by 3.0 PIN HOLE IN STAINLESS EAR

S_{br}=81,250/(3×3 X 2)=4.513 KSI<21.6 Table GA for stainless 316

See (Fig ST2.0-A1,FEA). This FEA of 8900ksi is only over 1.5", over a 3" thickness it would give 4.45ksi and be close to the above hand calc

ST2.1 Bending in Boom Cylinder Mount stainless Ear

BENDING IN STL 1.5" thick EARS

See (Fig ST2.1`-A2 BENDING STRESSES IN BOOM CYLD EAR, FEA). This 2D FEA shows that bending stresses in ear can approach a Sb of +8/-15 at section A-A, which has the min section modulus. Tension being the positive stress at the bottom of section A-A

Stresses to the right of section A-A do not account for the additional metal of the base plate. A 3D FEA would need to be done to get an estimate of stress in this area. This will be updated at a later date.

HAND CALC TO CHECK 2D FEA: (See FIG ST2.1-A1)

∑Moments about R2:

(40.6 x R1y) – (5.1" x R1x) =(81.2K x Cos11°) x 8.6"=685kip-in

Indeterminate free body, assume R1x=46.7K per (Fig ST2.1-A1)

40,6R1y=685.48-238.17

R1y=11K (somewhat close to 13.14K)

Sb at section A-A is Sb=M/S= ((11k x 21.4) –(46.7x5.08))/12.7=-+164.28/10.47=+12.9ksi tension at Bottom of A-A. See S=12.47 in (fig. ST2.1-A1)

Per table GA any stress more than 15ksi in bending for 316 stainless may make it necessary to change to 17-4 stainless to handle the increased stress.
In this case a 3D FEA may be justified to confirm that 316 stainless could be used.

The hand calc for Compression at A-A:
Sb=35.6/3.4=10.47.
Sb=(164.28)/10.47= -15.6ksi compression, close to that shown in the 2D FEA

SHEAR STRESS WELDS OF STL EAR TO 1.5" THICK BASE STL PLATE

- Min. approx. weld size t_w (conservative, as neglects stop alum black)

(81,250 COS 11 DEG) / (.707×.75" WELDS((38.8X 2 welds) + 2" on ends))=

79757lbs/41.3in^2=1928ksi

COMBINED SHEAR & TENSION in Welds to 1,5" stainless base plate

This is not specifically addressed in AISC 1.5.3 for welds, but an approximation can be obtained by limiting the tension stress per formula from AISC.

F_{rt} (allow)=[1-(Sv/F_t)2]× F_t AISC Sup#3

Sv=1.9 ksi from preceding sect
Ft=15 the allowable tensile stress from table GA allowable
Frt= reduced allowable tensile stress

$(1-(Sv/Ft)^2$ x Ft=Frt

$(1-(1.9/15)^2)$ x 15=13.1>> 12.9 KSI (from above tension calc in red). This passes the allowable but a FEA of this part is justified.

The crane has worked fine for over 10 years

ST2.2 Stainless Boom Cylinder Mount Bolts

(ref Fig ST3.0-A2 and ST2.1-A2)

STAINLESS CYLD MOUNT (For Boom Cylinder)
(Ref Fig ST2.1-A2)
BOLT SHEAR capacity per bolt
Shear cap per/.875" bolt=16.6 KSI×Abolt×.8=7985#/bolt (316 Stainless)

OR 32 KSI×Abolt×.8=12025#/bolt (for 17-4 bolts)
WELDED ALUM 2" THICK STOP:
If allowable $C_{compressive}$ = 8000
Stop cap=8 KSI× (1.5"× 1.5")× 2 (Stl contact area, very conservative estimate) =36,000#

This is a conservative estimate of the shear load taken by the stop for the purposes of estimating the bolt loads only.

SHEAR LOAD=162.2×Cos11°=159.2 KIPS (Pg M2.0)

Min # of shear bolts req= (159.2-36)/7.98/bolt=15.44 bolts if 316 Stainless. There are 19 bolts

Or 15.44 x 7985/12025=9.98 bolts if 17-4 stainless. About 19 bolts were used, and all will be 17-4 bolts.

(BEARING/SHEAR LOAD)/BOLT=(162.2-36)/19bolts=6642.1# BOLT (conservative)
36K is conservative shear load taken by the stop.

S_{br}=6642/(1.50"×.875)=5061<21 (Ref Table GA) OK (In Stainless Mount)

S_{br}=6642/ (2.00×.875)=3795 PSI (Bolt to 2" Thk Alum Plate)

<u>MOST STRESSED BOLT (4 BOLTS TO BEARING)</u>
$S_{a\ BOLT}$ = 32,000 PSI (See Sect M4.0)
Sa=axial stress

Shear stress per bolt:
S_v=6642lbs/(A_{bolt}×.8)=6642/(.6 in^2×.8)=13,840 PSI

These are the bolts that hold the bearing in addition to handling the shear loads of the cylinder. So, 3 methods will be used to check this critical area for stress: the actual thread root diameter of 7/8 bolt is .749 with an area of .44 in ^2 vs the .48in^2 here. An approximating rule of thumb was used.

<u>By Mohr formula (Bolt)</u>

$\sigma_{x=}$ axial stress=32ksi

Sv= Shear stress= 13.8 ksi

$\sigma_1, \sigma_2 = (\sigma_x + \sigma_y)/2 \pm \{[(\sigma_x + \sigma_y)/2]^2 + Sv^2\}^{0.5}$
$= (32 + 0)/2 \pm ((32/2)^2 + (13.8)^2)^{0.5}$
$= 16 \pm 21.13$
σ_1, σ_2=37.13, -5.13 (ok for Non-Solas condition)

VON MISES (BOLT)

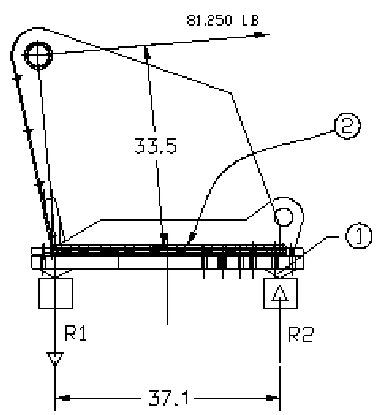
$\sigma_{VM}= (\sigma_1{}^2+ \sigma_2{}^2- \sigma_1\times \sigma_2)^{0.5}$
$=(37.13^2+(-5.13)^2-(37.13)\times (-5.13))^{0.5}$
$=(1405+190.5)^{1/2}=39.9$
17-4 bolts will be made to min T.S= 155KSI H.T. H1025

Method 3 by AISC 1.6.3
Since TS of 17-4 is >> than a A490 Bolt (TS=about 108-130ksi)
The below formula can be used for reduced tension
Ft=70-1.6Sv<54
Sv=13.84 ksi
Ft=47Ksi which is > 32ksi from (sect M4.0)…. OK

ST3.0 Starship Main Horizontal Pl 2.0", & Main Ear Reinforcement

The high-tension stresses in the main ear are distributed to the Bolts/Bearing by the half Moon doublers, ear stiffener and alum stop for the stainless boom cylinder mount. As shown in (Fig ST3.0-A1 and ST3.0). See section ST1.0 for main Ear general stresses.

Starship main horizontal plate and starship reinforcement.



ST3.0
STARSHIP HORZ PL. AND
REINFORCEMENT

Tension stresses under main ear only by R1:

R1≈R2≈(81.25 X 33.5)/37.1" =73.37 Kips. This is distributed over (ref sketch above and FIG.M2.1-A AND FIG M2.1-B)

AREA=20.8 in^2 (PER FIG M2.1-B)

$S_{ty} \approx Sry \cong 73.4/20.8 = 3528\, psi.$

Per fig ST1.0-A, the bend stress Sy in the main ear just above the re-enforcement is 8000 psi. The hand calc does not account for stress concentrations in this area. The 3.5ksi looks low and a 3D FEA would be needed to get a better estimate of stress. The allowable bend stress for 5083 by welds is 14.4ksi >8ksi. So, it should be OK but a 3D FEA is still recommended. I have done several in this area on other starships. Note that both the 2D FEA and the hand calc do not fully represent the 2" ear stiffener between ears.

Compressive stresses under main ear only by R2:
The R2 compressive load will be distributed to the bearing thru the 2" half-moon doublers, 2.0" main horizontal alum plate, then the chock fast , then the 3/8 alum ring spacer.
This area is approximated by the 20.8in^2 area shown in fig M2.1-B. At this stage we are only looking at the compressive stresses under the main ear by R2 as shown in Fig ST3.0 and not the reaction loads under the stainless mount shown in fig ST2.2-B

R_2/Area= 73.4/20.8=3528ksi.

As with the tension stress under R1 I believe this hand calc underestimates the stresses under R2 in this area and a 3D FEA is recommended for cases like this. Fig ST1.0-A shows a compressive stress of about 6.2ksi in the main ear before the load is spread over the lower 2" horizontal plates and re-enforcement in this area, so 3.5ksi is conceivable.

$\sigma_{C2} = R_2/A \approx {}^{73.4}/_{2 \times 6.2} = 5.919 PSI$ (as a hand calc.)
also supports the 6.2ksi in fig ST1.0-A.

MAIN HORIZONTAL 2.00" PL

The highest bending stress should occur just under the upward vertical reaction load of the STL/BOOM cylinder mount. See PT5 in the (fig. ST 3.0-A2,ST3.0-A1, ST2.1-A2).
The R1y tension loads from the main ear (figST3.0-A2) mostly go directly to the Bolts. The stress can be estimated as shown in this figure as less than 2000psi. This figure is an isolated view of just the 2" horizontal plate. The 6.6k value comes from Fig ST2.1-A2. 13142/2=6571lbs

Note that Roarke's formulas for stress and strain (Ref #5) is a great ref for doing hand calcs to estimate FEA stresses. In this situation case 2d on flat plates table 11.4 could have been used, with some extrapolations.

The approximate thickness of plate under the bolt should be checked with a bearing catalog such as SKF Slew bearing (ref 6), pg. 24 fig3. For a raceway diameter of 38.3"/972mm > than 500mm, the Thickness should be > than .04x38.3"=1.53". 1.53 is < than existing 2" so is Ok.

Horizontal Shear stress is caused by the Cylinder mount load on the 2" horizontal Plate. (ref fig ST3.0-A2) is distributed over the 15" x 2" thickness, thru the bolts in this area. This shear load was estimated in sect 2.1 as 162.2K-36K=126K. 36K being the estimated shear load taken by the stop and the end of the stainless Boom cylinder mount.

(126K)/AREA≈(126K)/15" X 2" X 2 = 2100 PSI

Note that a 3D FEA of the starship is recommended in this area.

SECT B: MAIN BOOM SECTION

B1.0 Main Girder Stresses

Refer to (Fig. B1.0-A) to see Boom major cross sections.
Max Moment occurs @ CS6 at the PIN

$$M_{CS6} = \left(10000^{\#} \times 356.85\right) + \left(Wboom \times 77.35\right)$$
$$M_{CS6} = (10000 \times 356.85) + (5272 \times 77.35)$$
$$M_{CS6} \approx 3568500\, LIVE + 407789\, DL$$
$$M_{CS6} \approx 3{,}976{,}289\, IN^{\#}$$

MAIN GIRDER STRESSES @ CS6 (DOUBLERS NOT USED TO CALCULATE I)
From (fig B1.0-A1), $I_{c/s}$= 10000, and Z_{BOT}=10000/19.833=504.21
Z is section modulus. All from this fig.

$$S_{bZ@CS6} = \frac{M_{CS6}}{Z_{CS6}} = \frac{3{,}976{,}289}{504.21} = 7886.18$$
Psi COMPRESSIVE (BOT PL. OF GIRDER)

$$S_{bzTOPPL} = {}^{3{,}976{,}289 \times 12.93}\!\big/_{10000} = 5141.34$$ PSI TENSION (TOP PLATE)

- Lateral Load at $3500^{\#}$ from lifting rescues craft on a 23deg incline is:

$$S_{MAXLATERAL} = \frac{(356.85'' \times 3500^{\#}(\sin 23°) + 77.35'' \times 5272^{\#}(\sin 23^{\circ})) \times 12.047}{6239} = \frac{M_y \times C}{I}$$
@ SC6 (See Fig. B1.0-A)

$S_{maxlateral}$= 1250 PSI ± TENSION or COMPRESSION (Live Load and Dead Load)
(Refer to Fig B1.0-A For Locations of Loads)
(See Fig B1.0-A1 for Properties at "CS6")
(Refer to Sect B4.1 For Lateral Load)

MAIN GIRDER STRESSES @ CS2 (no side DBLRS here)

- Conservatively say same moment as $M_{Z\,CS6}$, see (Fig B1.0-A2) for properties
- $S_{bz} = {}^{M_Z \times C}\!\big/_{I}$

$$S_{bz} = {}^{3{,}976{,}289 \times 15.462}\!\big/_{12084} = 5087.84\, PSI$$ Bottom PL – Compressive

And

$$S_{Sbz} = {M_Z \times C}/{I}$$

$$S_{bz} = {3{,}976{,}289 \times 15.564}/{12084} = 5121.4 PSI$$ Top PL- Tension

All within allowable of table GA

<u>B2.0 Boom Cylinder Pin Doublers</u>

See (<u>fig B2.0-A1)</u>
`
P_{CYLD}=164,580[#]
Per sect M2.0

S_{br} - BEARING LOAD

$$S_{br} = {\left(164580/2\right)}/{3.25 \times \left[t_{BO} + t_{BI}\right]}$$

For t_{BO}=.5, t_{BI}=1.0

$$S_{br} = 16{,}880 PSI < 17{,}000$$

$S_{BR=}$ 16880<allowable Fbr for alum

<u>B 3.0 Combined Stress Under Boom Cylinder Pin</u>

A high stress area would be at PT2 ref. (<u>fig B2.0-A1)</u>. See (<u>Fig. B4.0</u>) both below and in the figures section, for stress direction notation.

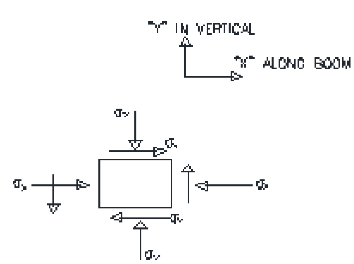


FIG. B4.0
BOOM MAIN PIN
DOUBLERS STRESSES

$\sigma_Y =$ Neglect, as Minor

$\sigma_{X2} =$ Local compression=general girder stress + local from pin hole

General Main Girder stress with doublers

$$\sigma_{x2} = {M_{cs6} x\, C}/_{I} = {3{,}976{,}289\ x\ 7.6}/_{15865}\ = 1904ksi$$

Compressive Stress See (fig and sect B1.0-A1) for cross section CS6 and M_{cs6} Moment values above. The 15865 is from this fig.

Local compressive stress in front of pin hole is about the same as the bearing stressing stress above:

$$\sigma'_{2x} = S_{br} = 16880psi$$

Local shear stress at point 2 is approximately:
σ_{2V} = Shear stress just forward of PIN Hole is approximately:

$$\sigma_{2V} = {(81.25/2)}/_{5.8"\ \mathrm{x}\ 2"}\ = 3500psi$$

The shear value is close to that shown in (Fig B2.0-A2).
The compressive stress in X axis in this figure is less than 16880 as it assumes the load is over all 3 plates, where it is only distributed to the t_{bo} and t_{bi}, not t_{bs}, as shown in (Fig B2.0-A1)

16880+1904=18784ksi in X

$$\sigma_{MAX} = \frac{\sigma_X + \sigma_Y}{2} \pm \left(\left(\frac{\sigma_X - \sigma_Y}{2} \right)^2 + \sigma_r^2 \right)^{1/2}$$

$$\sigma_{\max} \quad = {(18784+0)}/_{2} \pm \left(\left({18784-0}\right)/_{2})^2 +\ 3500^2 \right) \Big)^{1/2}$$

$\sigma_{\mathrm{max}= +19414.9, -630.9} = \sigma_1, \sigma_2$

The Von mises stress is:

$$\sigma_{vm} = \left((\sigma_1^2 +\ \sigma_2^2 - (\sigma_1 *\ \sigma_2) \right)^{\frac{1}{2}}$$

$\sigma_{vm} = 19737psi$

Safety factor =yield/19737=31/19.737=1.57 for 5083-H116 9 (table GA)
This is conservative as the bearing stress on the hole was added to the σ_{x2}compressive stress, which is why it is higher than the max von mises shown in fig B2.0-A2 of 12ksi

B4.0 Boom Main Pin Doublers (Star Ship to Boom)

BEARING STRESS (For 10,000# Tender)

$$S_{br} = \frac{82.3}{(1.25 + 0.50) \times 3.50}$$

$= 13.4$psi on CJ

$= 13.4 \frac{3.50}{3.75} = 12.51$ on Alum (For 3-1/2" PIN)

Doubler thickness and shape shown in (fig B1.0-A).

SHEAR STRESS BY PIN HOLE:
81K/(4.2 X 1.75 X 2)=5.14ksi. Close to value on (FIG B5.1-A BOOM MAIN PIN DBLRS)

B4.1 Lateral Loads On Crane When Rotating On 23deg Incline

REACTION LOAD FOR 3500# with boom at zero degrees of elevation and rotating up against the steepest part of the incline

- Lateral Load at 3500#, causes loads in Z direction (Longitudinal to boom) on main ears:

$$(3500(sin\,2\,3°)\overset{*1}{480}/_{24.09}) + ((5272\#\,sin\,2\,3)x199.38/24 = 44.36\,KIPS/EAR$$

See fig B1.0-A for the 24" between main ears and other dimensions
Loads contributed by Boom CYLD load of 162 Kips cause more loads in Z direction: $162\left({}^{3500}/_{10000}\right) = 56.7$KIPS/EAR
Horizontal Load=28.35 Kips in Z per ear.

- Vertical Load = $(3500\# + 5200\#)/2 = 4350$
  4.35 KIPS/EAR

Rp_z=44.36 + 28.35 = 73.11 KIPS/EAR for ear where Z loads add

R1pz=-44.36 + 28.35 = -16 KIPS/EAR For other ear

Rp=$\sum(73.11^2 + 4.35^2)^{1/2}$

Rpz= 73.24 KIPS/EAR max bearing load on one ear

$S_{br} = {}^{73.24}\!/\!(1.25 + 0.50) \times 3..50 = 11.96KSI$ On CJ`

$= 11.96 \times {}^{3.50}\!/_{4.0} = 10.46KSI$ On Alum.

(For 3 1/2" PIN) is OK

Note#1) *This Lateral load is not required by SOLAS as the boom is required to be elevated at 60-74 degrees when rotating through the steepest (highest Torque) of incline. Also 73.4< 81.25K*

Conclusion: A 3 ½" PIN will be used with 1.25" DBLR R1. See Sect P1.0 for bending in PIN.

<u>BOOM PIN DBLR'S on Boom</u> (Boom to Starship)

<u>SHEAR STRESS from Lateral Load, Rpz</u>

$S_v = \frac{73.24}{3.875 \times t \times 2} = 5.4KSI$

For t = 1.750 = 0.500" PL
+ 1.25" DBLR

Ref. Fig. B5.1A support the above calc. shear stress. The stress from the 10,000 loads on a level base is more than from the 3500lb on an incline

SECT P: MAJOR PINS

P1.0 Boom/Starship Pin

PINS

Also see M2.1-A,B,C,D,E and F for Boom Cylinder Pins as a ref for similar calcs

MAIN BOOM / STAR SHIP PIN (See Fig. P1.0)

$$M \approx [1/2 \times (0.50 + 1.25) + 0.32 + (1/2 \times (0.75 + 1.25))] \times 81KIP$$

$M \approx 177{,}795IN^{\#}$, $S = \frac{1}{4} \times \pi \times r^3 = 4.209$ (3.5” PIN)

$S_{b1} \approx M/S = 42.2KSI$ < 52 PER ABS TABLE GA

Note: Actual distribution of load is more trapezoidal (fig P1.0 MAIN BOOM/STRASHIP PIN). S_{b1} would be approx. 80% or less of above values. This can be verified with a FEA.

Conclusion: The 3.5” PIN will be made from 17-4 will be fine.

P2.0 Cable Cyld Pin

(See Fig. P2.0 CABLE CYLINDER PIN)

$$M = 43.5\left[\left(1.375 \times 1/2 + 0.0625\right) + \left(1 \times 1/2\right)\right]$$

$$M = 54.38KIP - IN$$

$$S_{2.5PIN} = 1/4\,\pi \times r^3 = 1.5339$$

$S_{bPIN} = M / S = 35{,}452PSI$ < 45 KSI Ok

$S_V = \frac{87}{2.5^2 \times \pi/4} = 17.723KSI$ < 41 KSI, Table GA

$\underset{BEARING}{S_{br}} = \frac{87}{2 \times 2.5} = 17.400 < 24KSI$ OK

The 35ksi bend stress shows that bending stresses in pins can not be ignored as is often done

P3.0 Boom Head Pulley Pins

(See Fig. P3.0 and P4.0)

These calcs would be similar to a block and tackle with sheaves that leads to a hook. Except with this set of sheaves the tension load in cable is about the same as load lifted. Normally, with a block a tackle the load on the cable is the load lifted/# of line parts

$$P_{MAX} = 1.05^6 \times 10000 \times 2 = \underset{MAX}{26{,}802^{\#}} \Big/ PER.PULLEY \text{ (maximum)}$$

$$M \approx P/2 \times \left[0.375 \times 1/2 + 0.065 + 0.50 \times 1/2\right]$$

$$M = 6{,}734\#\text{–}IN$$

$$S_{bp} = M/Z_P = \frac{6734}{1/4 \times \pi \times r_P^3} = 2540\,PSI < 45{,}000 \text{ PSI}$$

The Pin size Req. = 3.0" is driven by the .375 Aluminum Fish plates. (see section H1.1)

The bitter end Pin for the composite rope has a max load of $P = 26802 \times 1.05 = 28.4KIPS$ after going over the pulleys, as friction load is additive. (see Fig. P4.0).

M=19.4 lbs. x 1.12 in. = 21.72 in-lbs.

$$S_b = \frac{M}{Z} = \frac{21.72}{1.0^3 \times \pi / 4} = 27.6KSI$$

SECT BA: BASE SECTION

<u>BA1.0 Main Cross Section Stress</u>

Max Moment at Base is (Ref. Sect 1.1 and Sect M2.0)

$$MOMENT = LL \times 456'' + \underset{BOOM}{DL} \times 178.13''$$

$$MOMENT = 10000 \times 456'' + 5272 \times 456 / 2.56 = 458 FT - KIPS$$

A bit less then M_{T0TAL} at the main pin as done for Hydraulic cylinder. As dimensions go to CL of rotation not the main hinge pin. See (<u>FIG B1.0-A SECTION B2-B2</u>)

<u>BA1.1 Stress at Section B-2</u>

Stress at Section B2 is; Ref. (<u>Fig. BA1.0-A</u>.)

$$S_b = {(458FTK\ x\ 12000) * 25.3}/{21170}$$

$S_b = 5582 PSI$ OK, could be compressive or tensile

Load for Direct Compression axial Y direction Stress = 5272+1200+625+10,000(live load) = 17,097 lbs. Includes actuator and base:

$$S_A = {17{,}100^{\#}} \Big/ \left(AREA = 144.17 IN^2\right) = 118.6 PSI \text{ minimal}$$

<u>BA2.0 Base Top Horizontal plate</u>

- Tensile Stress in Top PL from Bolts
- Max Bolt Load from sect M4.0 is = 16 KIPS on inner Bolt Pattern at the quadrant of the bolt Pattern.
  (See Fig.BA2.0)
- Required thickness by turntable bearing manufacturer

Turntable manufactures list criteria for min Horizontal plate thickness. SKF has tables for this.

The max bolt load = 16 Kips per Sect M4.0

See (<u>Fig BA2.0 TOP PLATE BOLT LOADS</u>) for view of how the bolts load the horizontal plate. See Table 11.2 Case #2e Roarke Formulas for Stress and Strain Ref #5.
Or can use a FEA.

ω=16 KIPS / 3.5 in. = 4.57 KIPS/IN.

a=18.9 IN.
b=11.0 IN.
r_0=16.74 IN. TO BOLT
b/a=.58

q=16 KIPS/(3.5”x4.13”)
q=1.11 KIPS/in^2

For b/a = .58

M_{ra}= -qa^2(L_{17} – (C_7/C_4) L_{14}) case 2e

$$L_{17} = \frac{\frac{1}{4}\left[1 - \frac{1-.3}{4}(1-(\frac{14.77}{18.9})^4)\right]}{.627} - \left(\frac{14.7}{18.9}\right)^2\left[1+(1+.3)Ln\frac{18.9}{14.7}\right]$$

$$L_{17} = .022$$

$$L_{14} = \frac{1}{16}\left(1-\left(\frac{14.77}{18.9}\right)^4 - 4\frac{\left(\frac{16.7}{18.9}\right)^2 Ln\frac{18.9}{16.7}}{.3864}\right)$$

$$L_{14} = .0151$$

$$C_7 = \frac{1}{2}\left(1-.3^2\right)\left(\frac{18.9}{11.0} - \frac{11}{18.9}\right)$$

$$C_7 = .5169$$

$$C_4 = \frac{1}{2}\left[(1+.3)\frac{11.0}{18.9} + (1-.3)\frac{18.9}{11}\right]$$

$$C_4 = .887$$

$$M_{ra} = -1.11(18.9)^2\left(.022 - \frac{.5169}{.887}(.0151)\right)$$

$$M_{ra} = 5.23 kip - in/in.$$

$$S_b = \frac{6(M_{ra})}{t^2}$$

$$S_b = \frac{6(5.23)}{2^2} = 7.85 KSI$$ < 14 KSI per Table GA for Non-Solas Load.

A FEA would be easier. But this would be a check.

(See Fig.BA2.1 TOP PLATE) bolt loads are:

# HOLE	BL
0	16
1	15.3
2	13.93
3	11.95

BL_{max} = 16 per Sect M4.0

$$BL \approx BL_{MAX} \times R / R_{MAX}$$

$R_{max.}$ = 16.7 Per Fig. BA2.1

BL#1=(16.0110”/16.7428”) * 16 = 15.3

BL#2=(14.5795”/16.7428”) * 16 = 13.93

BL#3=(12.5108”/16.7428”) * 16 = 11.95

BA3.0 Base Interior Vertical Plates Gussets Or Annular Shells

Local tensile and compressive stress at inner annular ring is caused by the tension loads from the bolts in this area(see BA2.0, BA2.1 & BA3.1) is:

Pt (see fig BA3.1) from the bolts is distributed over about 13.5 inches. (FIG BA2.1) = (2 x 16) + (2 x 15.3) = 62.6 KIPS. HL 0=16kips and HL 1=15.3kips

Sum of moments about R_1

R1 = (62.6 x (2.1 + 2.1))/2.1 = 125.2 KIPS (ref Fig BA3.1) approx. tensile load on gusset is R_1=125.2kips

St (gusset #1) = 125.2 KIPS / (13.5 x .5 th’k) = 18.5 KSI < 21.6 KSI for A36 per Table GA

Based on this assumption the Local stress in the outer shell by R_2 caused by Pt is compressive, but will quickly radiate to a tension stress over the general cross section. Load R2 on Horz plate is up and +, The reaction load on outside shell is down causing the local compression

BA 4.1 Check On Local Buckling of Inner Annular Ring

(see FIG BA 4.1 data for localized buckling for values below)

L=13.50 effective length of plate exposed to Column loads

L/3 = 4.50 This is a standard assumption for effective plate width to length ratio

I = .05069

$$R = \sqrt{I/A} = \sqrt{.05069/2.255} = .1499$$

R = .1499
K = 1

Kl/R = (1 * 13.50)/.1499 = 90

Sallow = 15.36 for 90 (per AISC Steel Manual Table 1-36) this is less than the 5582 psi from section BA1.1

SECT H: HEAD SECTION

H1.0 Stainless Cable Head

(See Fig.H1.0, & Fig H1.0-1)

P=87000 per sect M3.0 and note#1 for the 71,773lbs

(See Fig.H1.01)

MIN t_F Ref (fig h.101)

Usually Driven by Bearing Stress

$R_2 = \frac{71{,}773}{3} x2 = 23924.lb$ Use 23,924.33 LB. (Ref. Sect M3.1 for the 71.7kips)

$R_{1=}$ (71773/(3x2)) =11962lbs

$S_{BR} = {23{,}924.33}\Big/{t_F \times \phi PIN}$ (See Below)

H1.1 Plate Size Based On Bearing Stress

Pin and Plate sizes for STL head on Cyld rod and Alum Head Fins on Boom Ext Head are:

R_1	t_F	ϕP	$\varsigma_{bR} = \frac{R_2}{\phi P \times t_F}$
23,925 lb.	0.375 STL	3.0	21,267 PSI < 21.6KSI OK.
23,925 lb.	0.375 ALUM	3.0	21.2<.40/1.65=24

-The Solas allowable for alum per table GA is 19, but does not need to be met. The Ref 9. Aluminum Association Design Manual has Fy/1.65 for the allowable bearing strength. -For stainless the allowable bearing stress is 21.6 per table GA, but since SOLAS is not required for the 10000lbs the allowable bearing stress is about .9 x fy=.9 x 30=27ksi.

The stress in the Aluminum plates in the boom head (at the end of the boom) have about the same bearing stress at the pin holes and are included above.

H1.2 Plate Stresses in Stl. Head

(at end of Boom cyld rod)

SHOWN IN (FIG H1.0). 5.5” IS

- $S_{r3} \cong \dfrac{23{,}925}{5.5 \times 0.375 \times 2} = 5{,}800\,PSI$ Is Less than 16.6KSI OK.

- FEA IN (Fig.H1.2) confirms this stress is acceptable. Note hot spots are ignored due to 23.9K load as point loads and not fully distributed.

- Note that in this case the hoop stress is higher as shown by the 11189psi in the 2d FEA for stress in Y axis is:

$$S_{ty} = \frac{23.9}{5.5\ x\ .375} = 11.58ksi$$

SHEAR STRESS S_{v1} Gussets to Cyld Rod
(See Fig.H1.2-1)

SHEAR AREA $= 4.20 \times 8 gussets \times 0.25 \times 0.707 = 5.94 IN^2$

$S_{v1} = {71{,}773^{\#}}\big/{5.94} = 12{,}083$ PSI < 16.6 OK

Shear Stresses at Base of Fish Plate is:

$$S_{3vx} = {23925lbs}\big/{(.375\ x\ 11")} = 5800psi$$

STAINLESS HEAD SOLAS 6K/ COMP ROPE
(See Fig.H1.0-1 CABLE HEAD TOP VIEW and H1.0 CABLE HEAD SIDE VIEW)

P=86800=2 x R_1 + 2 x R_2 = 2 x R_1 + (2 x (2 x R_1)= 6R_1

The 87,000lb load is used here

$R_1 = 14{,}500LB.$
$R_2 = 2 \times R_1 = 29{,}000LB.$

Gen axial stress in fish plates is:
S_{2ax} = 29,00/(.375 x 11)= 7.03ksi < 21.6 KSI

This is close to the same as shear stress in welds.

$^{87,000}/_{7030} \geq 4.5 \quad OK$ Safety factor for non-Solas <15ksi Table GA

Shear Stress SvR2 (see fig H1.2-1) in plate R2:
29000/(4.47 x .375)2=8650ksi
Close to FEA value shown in (Fig H1.2 FEA in STAINLESS 3/8 PL)

BENDING STRESS AT SY2:
SECTION MODULUS=1/6 t x h^3
SECTION 1/6 x .375 x 3.8^2=.94in
APROX MOMENT=wl^2/8=(29/3.5)X3.5^2=12.6
Fb=M/S=12.6/.94=13,5ksi close to that shown in FIG H1.2 and close to the hoop stress of 11.58ksi

THE END FOR NOW

REFERENCES

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3. AWS (Previously sent CD as part of T27 Documentation on Last Job.)
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5. Roark’s Formulas for stress and strain
6. SKF Slew Bearing catalog PU BU/P2 06115/3E
7. API 2C “specification for offshore pedestal mounted cranes” from the American Petroleum Institute
8. Aluminum Association Standards
9. Aluminum Association Design Manual