MASTER

LOFT TECHNICAL REPORT LTR 1111-55

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LOFT DOWNCOMER INSTRUMENTATION

ASSEMBLY ANALYSIS

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IDAHO NATIONAL ENGINEERING LABORATORY

DEPARTMENT OF ENERGY

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RESEARCH, ENGINEERING AND CONSTRUCTION REPORT ENGINEERING ANALYSIS DIVISION

APPLIED MECHANICS BRANCH

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LOFT DOWNCOMER INSTRUMENTATION ASSEMBLY ANALYSIS

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**LOFT TECHNICAL REPORT
LOFT PROGRAM**

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AUTHOR R. J. Beers, Jr.		GWA NO.
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for White for go Keeter
PSB Mgr.

ABSTRACT

A supplemental stress analysis of the LOFT Downcomer Instrumentation Assembly (DIA) has been completed in accordance with the stress requirements of Section III of the ASME Code [Ref. 1 of Appendix]. For the original analysis, see Reference 9 of the Appendix. Both thermal loads, due to LOCE and operating conditions, and fluid loads were considered. These loads were provided by the Thermal Analysis Branch. The fluid loads will be confirmed by a letter from W. C. Townsend. The following conclusions were made:

- (1) Inconel 600 is an acceptable material for the pressure tubes.
- (2) The flowmeter can be attached to the DIA with 1/4" diameter, SA-193 type B8M screws. They should be lubricated with two coats of Neolube and torqued to 1.25 ft-lbs. This applies to all of the screws attaching the flowmeter to the DIA. As an alternative, SA-453, grade 660 screws may be used. They should be coated with two coats of Neolube and torqued to 4.9 ft-lbs (4.5 ft-lbs is acceptable).
- (3) The method of mounting the instrument leads and the pressure tubes is adequate.

APPENDIX A

1.0) STRESS ANALYSIS OF THE PRESSURE TUBES (INCONEL 600)

THE ONLY SIGNIFICANT LOADS ON THE PRESSURE TUBES RESULT FROM TEMPERATURE DIFFERENCES BETWEEN THE INSTRUMENT STALK AND THE TUBING DURING LOCE, AND THE DIFFERENCES IN COEFFICIENTS OF THERMAL EXPANSION BETWEEN THE TWO MATERIALS. THE THERMAL ANALYSIS BRANCH PROVIDED THE OPERATING TEMPERATURES AND THE TEMPERATURE DIFFERENCES DURING LOCE. (SEE FIG. 1, PAGE A-2)

1.1) OPERATING CONDITIONS

STRESSES IN THE TUBES DURING OPERATING CONDITIONS RESULT FROM THE DIFFERENCE OF COEFFICIENTS OF THERMAL EXPANSION OF THE TUBE AND INSTRUMENT STALK. TUBE - INCONEL 600
STALK - SA-210 GR 304

OPERATING TEMPERATURE = 540°F

$$\alpha_s = 9.76(10)^{-6} \frac{\text{IN}}{\text{IN}^\circ\text{F}} @ 540^\circ\text{F}$$

$$\alpha_T = 7.85(10)^{-6} \frac{\text{IN}}{\text{IN}^\circ\text{F}} @ 540^\circ\text{F}$$

$$E_T = 29.2(10)^6 \text{ PSI} @ 540^\circ\text{F}$$

$$\delta_T = \alpha \Delta T L$$

$$\delta_T = 7.85(10^{-6})(540-70) L$$

$$\delta_T = 0.00369 L$$

$$\delta_s = \alpha \Delta T L$$

$$\delta_s = 9.76(10^{-6})(540-70) L$$

$$\delta_s = 0.00459 L$$

$$\Delta \delta = 0.00459 L - 0.00369 L$$

$$\Delta \delta = 0.000898 L$$

FIG. 1 [2]

$$\epsilon = \frac{\Delta S}{L} = 0.000898 \frac{L}{L}$$

$$\epsilon = 0.000898 \text{ IN/IN}$$

$$\sigma = \epsilon E$$

$$\sigma = 0.000898 (29.2)(10)^6$$

$$\sigma = 26,213 \text{ PSI} = 26.21 \text{ KSI}$$

FOR THERMAL EXPANSION, M.B-3222.3 APPLIES []

$$S_{ALL} = 3 S_m$$

$$S_m = 20.0 \text{ KSI}$$

$$3 S_m = 60.0 \text{ KSI}$$

SINCE,

$$3 S_m = 60.0 \text{ KSI} > \sigma = 26.21 \text{ KSI}$$

THE TUBING IS ADEQUATE FOR OPERATING CONDITIONS

1.2) LOLE

STRESSES IN THE TUBE DURING LOLE RESULT FROM THE TEMPERATURE DIFFERENCES AND THE DIFFERENCES OF COEFFICIENTS OF THERMAL EXPANSION BETWEEN THE INSTRUMENT STALK AND THE TUBE.

$$\alpha_s = 7.57(10^{-6}) \text{ IN/IN}^\circ\text{F @ } T=385^\circ\text{F}$$

$$\alpha_T = 7.56(10^{-6}) \text{ IN/IN}^\circ\text{F @ } T=300^\circ\text{F}$$

$$E_T = 30.5(10)^6 \text{ PSI @ } T=300^\circ\text{F}$$

$$\delta_T = \alpha \Delta T L$$

$$\delta_T = 7.56(10^{-6})(300-70)L$$

$$\delta_T = 0.00174 L$$

$$\delta_S = \alpha \Delta T L$$

$$\delta_S = 9.57(10^{-6})(385-70)L$$

$$\delta_S = 0.00302 L$$

$$\Delta \delta = 0.00302 L - 0.00174 L$$

$$\Delta \delta = 0.00128 L$$

$$\epsilon = \frac{\Delta \delta}{L} = 0.00128$$

$$\sigma = \epsilon E$$

$$\sigma = (0.00128) 30.5(10)^6$$

$$\sigma = 38,910 \text{ PSI} = 38.91 \text{ KSI}$$

FOR THERMAL EXPANSION, NB. 3222.3 APPLIES (1)

$$S_{ALL} = 3 S_m$$

$$S_m = 20 \text{ KSI}$$

$$3 S_m = 60 \text{ KSI}$$

SINCE,

$$\underline{3 S_m = 60 \text{ KSI} > \sigma = 38.91 \text{ KSI}}$$

THE TUBING IS ADEQUATE FOR LOCE

2.0) STRESS ANALYSIS OF FLOWMETER MOUNTING SCREWS

2.1) LOADS ON THE FLOWMETER

THE ONLY SIGNIFICANT LOAD ON THE FLOWMETER RESULTS FROM FLUID-INDUCED LIFT FORCE DUE TO THE VORTEX PROCESS. BASED ON A CONVERSATION WITH W. C. TOWNSEND, THE LOADS ON THE FLOW METERS OF THE DOT RAKE ASSEMBLY WOULD BE CONSERVATIVE TO USE FOR THE LOADS ON THE FLOWMETERS OF THE DOWNCOMER INSTRUMENTATION ASSEMBLY. HE ALSO SAID THE CROSS LOADS WOULD BE MINIMAL. [3]

DEAD WEIGHT AND THERMAL STRESSES ARE SMALL SINCE THE COMPONENTS HAVE THE SAME COEFFICIENTS OF THERMAL EXPANSION

FROM LOFT TECHNICAL REPORT LTR 141-62, [4] PAGE 7 AND LOFT TECHNICAL REPORT LTR 144-8, PAGE 1-A, [5] THE DRAG FORCE ON ONE FLOW METER, PARALLEL TO THE FLOWMETER IS:

$$D = 3216$$

2.2) STRESS IN THE MOUNTING SCREWS FROM DRAG

STRESS IN THE SCREWS RESULT FROM SHEAR FROM THE LOAD PLUS SHEAR FROM THE ECCENTRICITY OF THE LOAD.

SHEAR DUE TO THE LOAD

$$f_v = \frac{V}{A}$$

$$f_v = \frac{32/3}{0.032} \quad @ \quad 1/4" \text{ DIAMETER BOLTS}$$

$$f_v = 333 \text{ PSI}$$

SHEAR DUE TO ECCENTRICITY OF THE LOAD

$$\text{MAXIMUM MOMENT} = (32) 1.5$$

$$M_{\max} = 48 \text{ lb}$$

$$\bar{x} = \frac{(1.788)A_1 + (2.788)A_2 + (0.0)A_3}{3A}$$

$$\bar{x} = 1.525"$$

TO BE CONSERVATIVE AND SIMPLIFY CALCULATIONS, IT WILL BE ASSUMED THAT SCREWS ② & ③ WILL CARRY ALL THE MOMENT.

$$M = V(1.525 + .263)$$

$$48 = V(1.788)$$

$$V = 26.8 \text{ lbs}$$

$$f_{v_2} = V/A$$

$$f_{v_2} = 26.8 / .032$$

$$\underline{f_{v_2} = 840 \text{ PSI}}$$

TOTAL SHEARING STRESS

$$f_v = f_{v_1} + f_{v_2}$$

$$f_v = \sqrt{840^2 + 333^2}$$

$$\underline{f_v = 902 \text{ PSI}}$$

$$f_v = \text{ALLOWABLE SHEARING STRESS}$$

$$f_v = .6 S_m \quad \text{HG-3227.5 [1]}$$

$$S_m = 6.5 \text{ KSI @ } 540^\circ \text{F}$$

STAINLESS STEEL SA-193 88M

SINCE,

NOTE: S.S. BOLTS ARE WEAKER THAN SA-660

$$\underline{f_v = 902 \text{ PSI} < .6 S_m = 3,900 \text{ PSI}}$$

THE SCREWS ARE ADEQUATE FOR BOTH SA-193 & SA-660

2.3) ALLOWABLE CROSS FLOW LOAD

AT THE TIME THAT THIS ANALYSIS WAS MADE, THE CROSS FLOW LOADS WERE NOT KNOWN. BUT IT IS KNOWN THAT THESE LOADS ARE SMALL COMPARED WITH THE DRAG PARALLEL TO THE FLOWMETER. THEREFORE THE ALLOWABLE LOAD FOR CROSS LOAD WILL BE CALCULATED TO SEE IF IT IS REASONABLE AND THEN IT WILL BE CONFIRMED IN WRITING BY A LETTER FROM W. C. TOWNSEND [3]

CROSS FLOW LOAD

CROSS FLOW LOADS IN THE y DIRECTION, P_a & P_b ARE OUT OF PHASE SUCH THAT THEY CANCEL EACH OTHER. THE ONLY CROSS FLOW FORCE OF CONCERN IS THE FORCE P . [3]

THE FORCE P WILL STRESS THE SCREWS IN PURE SHEAR. THE ALLOWABLE LOAD P , WILL BE BASED UPON THE ALLOWABLE STRESS LEFTOVER AFTER CONSIDERING THE DRAG FORCE.

$$F_v = .6 S_m = \text{ALLOWABLE STRESS (SHEAR)}$$

$$.6 S_m = 3900 \text{ PSI}$$

$$\text{STRESS EXPENDED} = 1000 \text{ PSI}$$

by DRAG FORCE (CONSERVATIVE)

SEE PAGE A-7

$$\text{ALLOWABLE STRESS} = 3900 - 1000 = 2900 \text{ PSI}$$

REMAINING

$$F_v = \frac{P/3}{A} = \text{ALLOWABLE STRESS REMAINING.}$$

$$2900 = \frac{P/3}{0.032}$$

$$P = 270 \text{ lb}$$

THIS ALLOWABLE LOAD IS MANY TIMES HIGHER THAN THE EXPECTED CROSS LOADS. $\therefore$ THE SCREWS ARE ADEQUATE

2.4) FATIGUE IN THE MOUNTING SCREWS

SINCE THE STRESS IN THE SCREWS IS SO LOW, $f_u = 90\%$ PSI, FATIGUE NEED NOT BE CONSIDERED.

3.2) DETERMINATION OF PRELOAD (TORQUE (SA-193-B8M))

THE AMOUNT OF PRELOAD APPLIED TO THE SCREWS WILL BE GOVERNED BY THE THREAD AREA OR THE MAXIMUM SHEAR AND PRINCIPLE TENSILE STRESSES.

3.2.1) PRELOAD BASED ON THREAD AREA

BASED ON A CONVERSATION WITH T.O. CARLSON, ^[6] $\frac{1}{2}$ DIAMETERS ARE ENGAGED. THE FEMALE THREADS ARE STRONGER $\therefore$ PRELOAD BASED ON MALE THREADS.

$$f_u = \frac{P}{A}$$

P = PRELOAD

A = EFFECTIVE AREA

$$A = 2\pi r_{ave} \left(\frac{L}{Z} \right)$$

$$A = 2\pi \left(\frac{0.2175}{2} \right) \left(\frac{(0.5)(25)}{2} \right)$$

$$A = 0.128$$

$F_v = \text{ALLOWABLE SHEARING STRESS}$

$$F_v = .6 S_m \quad S_m = 13.0 \text{ KSI} \quad \text{HB 3227.5 [1]} \\ @ 540^\circ \text{F}$$

$$F_v = F_v = \frac{P}{A}$$

$$.6(13,000) = P/0.128$$

$$\underline{P = 1000 \text{ lbs}} = \text{MAXIMUM PRELOAD BASED ON THREAD AREA.} \quad \leftarrow$$

3.22) PRELOAD BASED ON MAXIMUM SHEAR AND PRINCIPLE TENSILE STRESS

THESE STRESSES ARE CALCULATED BY FORMULAS AND DIMENSIONS IN UNIFIED SCREW THREADS [7]

TENSION

$$S_T = \frac{P}{A_s} \quad @ \frac{1}{4}'' \text{ DIAMETER}$$

$$S_T = \frac{P}{0.032} = 31.25P$$

TORSION

$$S_s = \frac{T_1}{Z_p}$$

$$Z_p = 0.1963 (K_{sm})^3$$

$$Z_p = 0.1963 (.1887)^3$$

$$Z_p = 1.319 (10)^{-3}$$

$$T_1 = T/2 \quad T = \text{TORQUE}$$

$$S_s = \frac{T/2}{1.319 (10)^{-3}}$$

$$S_s = 379 T$$

MAXIMUM SHEAR

$$S'_s = \sqrt{\frac{S_T^2}{2} + S_s^2}$$

$$S'_s = \sqrt{\frac{(31.25P)^2}{2} + (379T)^2}$$

$$T = 0.0618P \text{ (SEE PAGE A-12 OF THIS REPORT)}$$

$$F_v = .6 S_m = \text{ALLOWABLE SHEAR NB-3227.5 [1]}$$

$$.6 S_m = (.6)(13,000) \text{ PSI SA-193, B8M BOLTS @ 540°F}$$

$$F_v = 7800 \text{ PSI}$$

NA PAGE 86 [1]
FOOTNOTE 1

$$S'_s = 7800 = \sqrt{\frac{(31.25P)^2}{2} + (379(.0618)P)^2}$$

$$\underline{P = 242 \text{ lb}}$$

MAX. PRELOAD BASED
ON MAX SHEAR

MAXIMUM PRINCIPLE TENSILE STRESS

$$S'_t = \frac{S_T}{2} + S'_s$$

$$F_T = \text{ALLOWABLE TENSION}$$

$$F_T = 2 S_m \text{ NB 3232.2 [1]}$$

$$F_T = 2(6.5) @ 540°F, SA-193, B8M$$

$$F_T = 13,000 \text{ PSI}$$

$$13,000 = \frac{31.25P}{2} + \sqrt{\frac{(31.25)^2 P^2}{2} + (379(0.0618)P)^2}$$

$$\underline{P = 272 \text{ lb}}$$

MAX PRELOAD BASED ON
MAX PRINCIPLE TENSION

REVIEWING THE PRELOAD CALCULATIONS, THE MAXIMUM PRELOAD IS GOVERNED BY MAXIMUM SHEAR. SINCE THE THREAD SHEAR DOES NOT GOVERN THE PRELOAD, THE TORQUE WILL BE THE SAME FOR ALL 4 SCREWS.

$$\therefore \underline{P = 242 \text{ lb}} \quad (\text{SA-193-B8M})$$

3.1.3) DETERMINATION OF TORQUE

THE TORQUE WILL BE CALCULATED ACCORDING TO "TORQUE-LOAD RELATIONSHIP FOR STAINLESS STEEL FASTENERS" [8]

$$T = C D L (\text{in-lb})$$

C = COEFFICIENT

D = NOMINAL DIAMETER

L = DEVELOPED AXIAL LOAD = P

STAINLESS STEEL BOLT WITH TWO COATS OF NEOLUBE

$$C = 0.247$$

$$D = 0.250$$

$$L = P = 242 \text{ lb}$$

$$T = (.247)(.250) P$$

$$T = 0.0618 P$$

$$T = 0.0618 (242)$$

$$T = 16.0 \text{ IN-LB}$$

$$T = 1.25 \text{ FT-LB} \quad \text{WITH TWO COATS OF NEOLUBE}$$

THE TORQUE WILL BE THE SAME FOR ALL 4 SCREWS FOR SA-193-B8M

3.2) DETERMINATION OF PRELOAD & TORQUE (SA-453-660)

THE AMOUNT OF PRELOAD APPLIED TO THE SCREWS WILL BE GOVERNED BY THE THREAD AREA OR THE MAXIMUM SHEAR AND PRINCIPLE TENSILE STRESSES.

3.2.1) PRELOAD BASED ON THREAD AREA

BASED ON A CONVERSATION WITH J.O. CARLSON, ^[6]
 $1\frac{1}{2}$ DIAMETERS ARE ENGAGED. THE MALE THREADS ARE STRONGER $\therefore$ PRELOAD BASED FEMALE THREADS.

$$f_v = \frac{P}{A}$$

P = PRELOAD

A = EFFECTIVE AREA

$$A = 2\pi r_{ave} \left(\frac{L}{2} \right)$$

$$A = 2\pi \left(\frac{0.2175}{2} \right) \left(\frac{(1.5)(25)}{2} \right)$$

$$A = 0.128$$

F_v = ALLOWABLE SHEARING STRESS

$$F_v = .6 S_m$$

$$S_m = 17.1 \text{ KSI}$$

NB 3227.5 [1]

SA-240-304

@ 540°F

(CHANNEL)

$$F_v = F_v = P/A$$

$$.6(17,100) = P/0.128$$

$$\underline{P = 1313 \text{ lbs}} = \text{MAXIMUM PRELOAD BASED ON THREAD AREA.}$$

3.2.2) PRELOAD BASED ON MAXIMUM SHEAR AND PRINCIPLE TENSILE STRESS

THESE STRESSES ARE CALCULATED BY FORMULAS AND DIMENSIONS IN UNIFIED SCREW THREADS [7]

TENSION

$$S_T = \frac{P}{A_s} \quad @ \frac{1}{4}'' \text{ DIAMETER}$$

$$S_T = \frac{P}{0.032} = 31.25 P$$

TORSION

$$S_s = \frac{T_1}{Z_p}$$

$$Z_p = 0.1963 (K_{smsh})^3$$

$$Z_p = 0.1963 (.1887)^3$$

$$Z_p = 1.319 (10)^{-3}$$

$$T_1 = T/2 \quad T = \text{TORQUE}$$

$$S_s = \frac{T/2}{1.319 (10)^{-3}}$$

$$S_s = 379 T$$

MAXIMUM SHEAR

$$S'_s = \sqrt{\frac{S_I^2}{2} + S_s^2}$$

$$S'_s = \sqrt{\frac{(31.25P)^2}{2} + (379T)^2}$$

$$T = 0.0557P \text{ (SEE PAGE A-16 OF THIS REPORT)}$$

$$F_v = .6 S_m = \text{ALLOWABLE SHEAR NB-3227.5 [1]}$$

$$.6 S_m = (6)(54,000) \text{ PSI SA-453-660 BOLTS @ 540°F}$$

$$F_v = 32,400 \text{ PSI}$$

HA PAGE 86 [1]
FOOTNOTE 1

$$S'_s = 32,400 = \sqrt{\frac{(31.25P)^2}{2} + (379(0.0557P))^2}$$

$$P = 1060 \text{ lb}$$

MAX. PRELOAD BASED ON MAX SHEAR

MAXIMUM PRINCIPLE TENSILE STRESS

$$S'_t = \frac{S_I}{2} + S'_s$$

$$F_t = \text{ALLOWABLE TENSION}$$

$$F_t = 2 S_m \text{ NB 3232.2 [1]}$$

$$F_t = 2(27.05) @ 540°F, SA-193, B8M$$

$$F_t = 54,100 \text{ PSI}$$

$$54,100 = \frac{31.25P}{2} + \sqrt{\frac{(31.25P)^2}{2} + (379(0.0557P))^2}$$

$$P = 1170 \text{ lb}$$

MAX PRELOAD BASED ON MAX PRINCIPLE TENSION

REVIEWING THE PRELOAD CALCULATIONS,
THE MAXIMUM PRELOAD IS GOVERNED BY MAXIMUM
SHEAR.

$$\therefore \underline{P = 1060} \quad (SA-453-660)$$

3.2.3) DETERMINATION OF TORQUE

THE TORQUE WILL BE CALCULATED ACCORDING TO
"TORQUE-LOAD RELATIONSHIP FOR STAINLESS STEEL
FASTENERS" [8].

$$T = C D L = 1.66 f D L$$

C = COEFFICIENT f = FRICTION FACTOR

D = NOMINAL DIAMETER

L = DEVELOPED AXIAL LOAD = P

STAINLESS STEEL BOLT WITH TWO COATS OF
NEOLUBE.

$$f = .134 \quad (10\% \text{ LESS THAN STAINLESS STEEL})$$

$$D = 0.250$$

$$L = P = 1060 \text{ lb}$$

$$T = (1.66)(.134)(.250) P$$

$$T = 0.0557 P$$

$$T = 0.0557 (1060)$$

$T = 59 \text{ IN-LB}$	SA-453-660
$T = 4.9 \text{ FT-LB}$	WITH TWO COATS OF NEOLUBE

4.0) STRESS ANALYSIS OF THE INSTRUMENT LEAD AND TUBING CLIPS (METHOD OF MOUNTING ONLY)

4.1) TUBING CLAMP (ITEM 19) (DRAW 205050)

ALLOWABLE LOAD ON WELD

$$A_{\text{EFFECTIVE}} = 2(.5)(.06)(.707)$$

2 REDUCTION FOR EFFECTIVE THROAT AREA

$$A_{\text{EFFECTIVE}} = 0.0124$$

$$F = 21.0 \text{ KSI} = \text{ALLOWABLE STRESS FOR E 70 XX ELEC.}$$

$$P = FA = 0.0124(21,000)$$

$$P = 890 \text{ lb} = \text{ALLOWABLE LOAD ON WELDS (SHEAR OR TENSION)}$$

ALLOWABLE LOAD ON CLAMP

$$F_u = 65 \text{ ksi NB-3227.5 [1]}$$

$$S_m = 17.1 \text{ SA 240 GR 304}$$

$$P = F_u A = \text{ALLOWABLE LOAD}$$

$$P = [.6](17.1)[(.5)(.05)]$$

$$P = 260 \text{ lb} = \text{ALLOWABLE LOAD ON CLAMP. (SHEAR)}$$

LOADS ON THE WELD AND CLAMP

SINCE THE CLAMP HAS A GREATER COEFFICIENT OF EXPANSION THAN THE TUBING, THE TUBING WILL NOT BE DAMAGED NOR WILL STRESS BE IMPOSED ON THE CLAMP BY THE TUBING WHEN HEATED. THE CALCULATIONS ARE SHOWN BELOW. OPERATING CONDITIONS ARE MOST CRITICAL.

EFFECTIVE LENGTHS OF EXPANSION

$$\text{TUBE: } L = 0.25''$$

$$\text{CLAMP: } L = B + A = 0.31''$$

EXPANSION-TUBE: $\delta_T = L \alpha \Delta T$

$$\delta_T = (0.25)(7.85 \times 10^{-6})(540-70)$$

WHERE T = 540 = OPERATING TEMP

$$\delta_T = 0.0009 \text{ in}$$

EXPANSION-CLAMP: $\delta_C = L \alpha \Delta T$

$$\delta_C = (0.31)(9.76 \times 10^{-5})(540-70)$$

$$\delta_C = 0.00142$$

CONCLUSION

1) SINCE THE EXPANSION OF TUBE IS LESS THAN THE EXPANSION IN THE CLAMP, NO STRESSES WILL BE IMPOSED ON THE CLAMP OR THE TUBE

2) STRESSES DUE TO THERMAL EXPANSION OF THE CLAMP BY ITSELF WERE CALCULATED AND ARE NEGLIGIBLE. (FOR LOLE)

3) SINCE THE CLAMPS CAN CARRY 260 lbs IN SHEAR OR TENSION AND THERE ARE NO SIGNIFICANT LOADS ON THE CLAMP, THE CLAMP AND WELDS ARE ADEQUATE.

4.2) TUBING CLAMPS (ITEMS 46, 47, 52, 62, 64, 26) (DRAW. 205030)

ITEMS 46, 47, 52, 62, 64 AND 26 ARE CONSTRUCTED OUT OF THE SAME MATERIAL AND BUILT IN A SIMILAR FASHION AS ITEM 48 (SEE SECTION 3.1). SINCE THEY HAVE THE SAME COEFFICIENT OF EXPANSION AS ITEM 48, BY ANALOGY, NO SIGNIFICANT STRESS WILL BE IMPOSED ON THE TUBING OR CLAMPS.

EACH OF CLAMPS CAN CARRY AT LEAST 260 lbs IN SHEAR OR TENSION, SINCE THE CLAMPS CAN CARRY THIS LOAD AND THERE ARE NO SIGNIFICANT THERMAL STRESSES, THE CLAMPS ARE ADEQUATE.

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