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LOFT TECHNICAL REPORT LTR 141-195

RE&C INTERNAL REPORT RE-A-78-211

JANUARY 23, 1979

P394

EGG/LTR--141-195

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LOFT DTT RAKE PIN STRESS ANALYSIS

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EG&G Idaho, Inc.

IDAHO NATIONAL ENGINEERING LABORATORY

DEPARTMENT OF ENERGY

IDAHO OPERATIONS OFFICE UNDER CONTRACT EY-76-C-07-1570

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LOFT Technical Report LTR 141-95

Report No. RE-A-78-211

Date: January 23, 1979

INTERNAL TECHNICAL REPORT

Title: LOFT DTT RAKE PIN STRESS ANALYSIS

Organization: APPLIED MECHANICS BRANCH

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LOFT TECHNICAL REPORT LOFT PROGRAM

FORM EG&G-229
(Rev. 12-78)

TITLE LOFT DTT RAKE PIN STRESS ANALYSIS		REPORT NO. LTR 141-95
		RE-A-78-211
AUTHOR W. R. Mosby		CWA NO.
PERFORMING ORGANIZATION Applied Mechanics LEPD Mgr.		DATE January 23, 1979
LOFT APPROVAL <i>[Signature]</i> 11-20-78 <i>[Signature]</i> 1-15-79 PSE LEPD Mgr. ORM Seismic PSB RSB RCSS Review Mgr. Mgr. Supv. P&CSB Mgr.		

ABSTRACT

A stress analysis of the 3/8-inch and 1/4-inch pins which hold the rake assembly to the flange was performed and shows stresses to be lower than the Class 1 allowables of the ASME Boiler and Pressure Vessel Code. The alternating pin stresses were found to be below the endurance limit and fatigue failure will not occur.

The rake assembly was assumed to be loaded by steady drag and lift forces and alternating vortex shedding forces.

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LOFT DTT RAKE PIN STRESS ANALYSIS

I. SCOPE OF INVESTIGATION

The LOFT DTT rake pins were analyzed for stresses caused by steady and alternating fluid loads.

II. METHOD OF ANALYSIS

The DTT rake was analyzed as a rigid body subjected to static loads^[1] representing steady and alternating fluid forces. An appropriate dynamic amplification factor was incorporated in the alternating force, assuming 7% damping^[2]. Resulting stresses in the 3/8-inch and 1/4-inch pins were compared to allowable stresses given in Reference 3.

III. RESULTS

The pins were found to be subjected to shear stress no greater than 4838 psi compared to an allowable shear stress of 16,140 psi (for SA-453, Gr. 660 material). The shear stresses in the flange and rake assembly adjacent to the pins is less than 4838 psi compared to an allowable of 11,100 psi. With 7% damping, the pin alternating stresses were found to be below the endurance limit. Fatigue failure will not occur.

If a 3/8-inch pin is substituted for the 1/4-inch pin assumed in this analysis, lower stresses will result.

IV. REFERENCES

1. R. G. Rahl, "Stress Analysis of LOFT Three DTT Rake Assembly", LOFT Technical Report LTR 144-8, February 16, 1977.
2. Division of Reactor Research and Development, United States Atomic Energy Commission, RDT Standard F9-2T, January 1974.

3. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section III, Division 1, "Nuclear Power Plant Components", 1977.

APPENDIX A
DTT RAKE PIN STRESS ANALYSIS

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CALCULATION WORK SHEET

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1. Geometry

The DTT rake is to be mounted to the flange with the aid of two $\frac{3}{8}$ " diameter pins and one $\frac{1}{4}$ " pin as shown in Figure 1.

Figure 1: DTT Rake Assembly

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2. Materials - ref. (3)

A. Pins - SA-453 Gr. 660

@ 650°F, $S_m = 26.9 \text{ ksi}$

$S_y > 26.9 \text{ ksi}$

$E = 25.1 \times 10^6 \text{ psi}$

@ 700°F, $S_y = 85 \text{ ksi}$

$S_v = 130 \text{ ksi}$

B. Flange - SA 479, 3/16 SST, BAR:

@ 650°F, $S_m = 16.7 \text{ ksi}$

$E = 25.1 \times 10^6 \text{ psi}$

$S_y = 18.5 \text{ ksi}$

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3. Fluid forces - ref. (1) with damping of 7%
assumed per ref. (2), table 1.

$$F_x = 0$$

$$F_y = -22 \pm 168 \text{ lb}$$

$$F_z = -200 \pm 26 \text{ lb}$$

$$M_x = 0$$

$$M_y = 304 \pm 168 \text{ in-lb}$$

$$M_z = -306 \pm 1223 \text{ in-lb}$$

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4. Pin stresses

A. Pin Forces

For conservatism, assume the pins take only shear forces. Then (see fig. 2)

$$\Sigma F_x = 0$$

$$(1) F_{x1} + F_{x2} + F_{x3} = 0$$

$$\Sigma F_y = 0$$

$$(2) F_y + F_2 - F_1 \sin 30^\circ + F_3 \sin 30^\circ = 0$$

$$\Sigma F_z = 0$$

$$(3) -F_2 - F_1 \cos 30^\circ - F_3 \cos 30^\circ = 0$$

Figure 2: Pin Reaction Force System

$$\Sigma M_x = 0$$

$$(4) -F_1 - F_2 + F_3 = 0$$

$$\Sigma M_y = 0$$

$$(5) M_y + (F_{x1} + F_{x3})(1.005) - F_{x2}(2.01) = 0$$

$$\Sigma M_z = 0$$

$$(6) -M_z + (F_{x3} - F_{x1})(1.74) = 0$$

CALCULATION WORK SHEET

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Solving sets of equations 2, 3, & 4 and 1, 5, & 6,

$$F_{x_1} = -.166 M_y - .287 M_z = 37 \pm 379 \text{ lb}$$

$$F_{x_2} = .332 M_y = 102 \pm 56 \text{ lb}$$

$$F_{x_3} = -.166 M_y + .287 M_z = -138 \pm 379 \text{ lb}$$

$$F_1 = .333 F_y - .577 F_z = 108 \pm 71 \text{ lb}$$

$$F_2 = -.667 F_y = 15 \pm 112 \text{ lb}$$

$$F_3 = -.333 F_y - .577 F_z = 123 \pm 71 \text{ lb}$$

B. Pin shear areas

(1) $\frac{3}{8}$ - pins #1 and 3

$$A = \frac{\pi (\frac{3}{8})^2}{4} = \underline{\underline{.1104 \text{ in}^2}}$$

(2) $\frac{1}{4}$ - pin #2

$$A = \frac{\pi (\frac{1}{4})^2}{4} = \underline{\underline{.0491 \text{ in}^2}}$$

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C. Pin stresses

$$(1) \tau = \frac{1}{A} \sqrt{F^2 + F_x^2} = \frac{1}{1104} \sqrt{(108 \pm 71)^2 + (37 \pm 379)^2}$$

$$\tau = \underline{\underline{493 \pm 3609 \text{ psi}}}$$

$$(2) \tau = \frac{1}{.0491} \sqrt{(15 \pm 112)^2 + (102 \pm 56)^2}$$

$$\tau = \underline{\underline{1326 \pm 3512 \text{ psi}}}$$

$$(3) \tau = \frac{1}{.1104} \sqrt{(123 \pm 71)^2 + (-138 \pm 379)^2}$$

$$\tau = \underline{\underline{-952 \pm 3754 \text{ psi}}}$$

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D. Allowable stresses

Direct shear (NB-3227.2, ref. 3)

Average primary shear stress $\leq .6 S_y$

$$.6 S_y > .6 (26.9 \text{ ksi}) > \underline{16.14 \text{ ksi}}, \text{ pins}$$

$$= .6 (18.5 \text{ ksi}) = \underline{11.1 \text{ ksi}}, \text{ flange}$$

From c above, the maximum shear stress
in the pins is

$$\tau_{\text{pin}} = \underline{4838 \text{ psi}} < 16.14 \text{ ksi} \quad \therefore \text{good}$$

The highest flange shear stress is less than
this due to greater area, so

$$\tau_{\text{flange}} < \underline{4838 \text{ psi}} < 11.1 \text{ ksi} \quad \therefore \text{good}$$

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E. Fatigue Analysis

$$\text{Max stress difference} = 2(3754) \text{ psi}$$

$$S_{ALT} = 7508 \text{ psi}$$

$$S_{ALT} \times \frac{30}{25.1} = \underline{\underline{8974 \text{ psi}}}$$

The endurance limit for the pin material
(Fig. I.9-4, upper curve) is $S_{ALT} \times \frac{30}{25.1} = 13,700 \text{ psi}$,

∴ The pins are good for more than 10^6 cycles.

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