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LOFT TECHNICAL REPORT LTR 1118-1
JUNE 6, 1978

DRY SCRAM EVALUATION

Prepared by

Todd Shipyards

per S-7133

EB

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
LOFT PROGRAM

TITLE Dry Scram Evaluation	REPORT NO. LTR 1118-1
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SUMMARY

The purpose of this LTR is to present the analysis performed by Todd Shipyards concerning the ability of the LOFT CRDMs to withstand a dry scram.

A "dry scram" could result in the CRDM components yielding; however, it would probably not render the CRDMs inoperable. It also concluded that a dry scram is highly unlikely based on a typical LOFT depressurization curve and the temperature of the fluid in the upper pressure housing. At the time of scram, the fluid in the upper pressure housing will not flash to steam owing to the pressure-temperature relationship existing during the scram cycle.

This analysis was originally Appendix C-5 to LOFT CDD 1.1.1.8B.

APPENDIX C-5

DRY SCRAM EVALUATION

1. INTRODUCTION

This appendix presents data relative to the selected CRDMs to withstand a dry scram.

2. CONCLUSIONS

Based on the analysis performed by Todd Shipyards, a "dry scram" could result in yielding of the CRDM components; however, it would probably not damage the CRDM to the extent that it would be inoperable. It is also concluded that a dry scram is highly unlikely based on a typical LOFT depressurization curve and the temperature of the fluid in the upper pressure housing at the time of scram. The fluid in the upper pressure housing will not flash to steam owing to its pressure-temperature relationship existing during the scram cycle.

TODD SHIPYARDS CORPORATION

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April 1, 1971

Idaho Nuclear Corporation
Post Office Box 1845
Idaho Falls, Idaho 84301

ATTENTION: Supplier Data Control - CF-689

ATTACHMENT: Analysis by Dr. C. D. Michaelopoulos of Dynamic Stresses During Dry Scram

SUBJECT: Report on Evaluation of Dry Scram - Subcontract S-7133

Todd Shipyards Corporation has evaluated the condition of scramming the control and without any water in the buffer piston area. This report is being submitted separately from the ASME Section III Code Analysis because it does not concern a pressure boundry.

The shock wave stresses have been calculated to be 150,000 psi in the leadscrew threaded joint that holds the buffer piston. During a "dry scram" this stress is equal to the yield stress for the 17-4 PH leadscrew.

Based on this analysis, it is concluded that the "dry scram" should be avoided. During the blowdown transient, the drives should be motored down in the same manner as the "capsize insertion" for the N.S. SAVANNAH.

Very truly yours,

E. L. Jackson, Jr.,
Project Manager

ELJ:llh:98:71
Attachment
cc: Mr. R. N. Moore
Dr. C. D. Michaelopoulos
Miss P. A. Jackson

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DYNAMIC STRESSES DEVELOPED DURING DRY-SCRAM CONDITIONS

In dry-scram conditions, the lead-screw assembly plus the control rod are in free fall. The maximum height, h, through which the assembly falls is 65 inches (5.41 ft). Thus, the impact velocity V_o is given by

$$V_o = \sqrt{2gh}$$

$$= \sqrt{2(32.2)(5.41)}$$

or

$$V_o \doteq 18.6 \text{ ft/sec.}$$

This is the velocity with which the buffer piston strikes the upper end of the motor tube. For the lead screw, $E = 28.5 \times 10^6$ psi, $\gamma = 0.28$ lb/in³. The acoustic velocity (velocity of propagation of a stress wave) in the lead screw, c, is

$$c = \sqrt{E/\rho} \quad , \quad \rho = \frac{\gamma}{g} = \left(\frac{0.28}{386}\right) \frac{\text{lb sec}^2}{\text{in}^4}$$

$$= \sqrt{\frac{28.5 \times 10^6}{(0.28)/(386)}}$$

$$= 198{,}000 \text{ in./sec}$$

or $\quad c = 16{,}500$ ft/sec.

From one-dimensional wave propagation theory:

if a prismatic bar moving with a uniform (rigid body) velocity V_o is suddenly stopped at one end, a stress wave develops at that end and travels down the bar with velocity c. The magnitude of this stress wave σ_o is given by

$$\sigma_o = c \rho V_o = \frac{V_o}{c} E$$

For $V_o = 18.6$ ft/sec,

$$\sigma_o = \frac{18.6}{16{,}500}(28.5 \times 10^6)$$

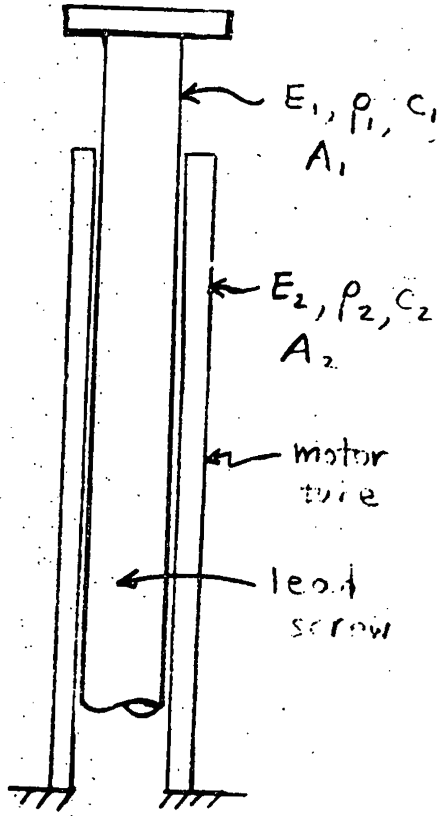
$$\underline{\underline{\sigma_o = 32{,}200 \text{ psi}}}$$

This would be the stress (tensile) if the buffer piston hit a rigid stop and it represents the maximum possible stress that can be developed in the lead screw (if it is assumed uniform — no threads). Note that the lower end is "free" that is the tensile wave will reflect as a compressive stress wave.

Using a stress concentration factor of 4.5 for the threaded connection to the buffer piston, the maximum stress in the lead screw during dry-scram conditions is:

$$\underline{\underline{\sigma_{max} \cong 150{,}000 \text{ psi}}}$$

This is a conservative answer, since the motor tube is not rigid. Taking into account the elasticity of the motor tube, the system can be idealized as shown in the accompanying figure.



The magnitude of the stress waves (tensile in the lead screw and compressive in the motor tube) are given by*

$$\sigma_1 = \frac{c_1 \rho_1 V_o}{KQ + 1}$$

$$\sigma_2 = \frac{KQ\, c_2 \rho_2 V_o}{KQ + 1}$$

where the subscripts 1 and 2 refer to the lead screw and motor tube, respectively, and

$$K = A_1 / A_2 \; , \; Q = c_1 \rho_1 / c_2 \rho_2 \, .$$

For the motor tube, $E = 29.2 \times 10^6$ psi and $\rho_2 = \rho_1 = (0.280/386) \frac{\text{lb sec}^2}{\text{in}^2}$. Thus,

$$c_2 = \sqrt{\frac{29.2 \times 10^6}{(.280)/(386)}} \doteq 201{,}000 \text{ in/sec}$$

or

$$c_2 = 16{,}800 \text{ ft/sec}.$$

Since $Q = \dfrac{c_1 \cancel{\rho_1}}{c_2 \cancel{\rho_2}} = \dfrac{16{,}500}{16{,}800} = 0.98 \cong 1.$

the above formulas simplify to

$$\sigma_1 \cong \frac{A_2}{A_1 + A_2}(c_1 \rho_1 V_0)$$

$$\sigma_2 \cong \frac{A_1}{A_1 + A_2}(c_2 \rho_2 V_0)$$

The cross-sectional areas are

$$A_1 = \frac{\pi}{4}\left[(1.576)^2 - (1.092)^2\right]$$

$$\cong 1.005 \text{ in}^2$$

$$A_2 = \frac{\pi}{4}\left[(4.375)^2 - (2.455)^2\right]$$

$$\cong 10.4 \text{ in}^2$$

Substituting in the above expressions for the stresses,

$$\sigma_1 = \frac{10.4}{11.4}(32,200)$$

or

$$\underline{\underline{\sigma_1 = 29,400 \text{ psi}}}$$

and

$$\sigma_2 \cong \frac{1.00}{11.4}(32,200)$$

or

$$\underline{\underline{\sigma_2 = 2,800 \text{ psi}}}$$

Note that the motor tube was considered fixed at the lower end since it is attached to the reactor. Actuall the *stress in the tube is about $2\sigma_2 = 5,600$ psi due to the reflected compressive wave

* M. F. Spotts, "Mechanical Design Analysis, Prentice-Hall, Inc., 2nd Printing, pp 357-360.