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DESCRIPTION:
Ltr notarized 11-7-73, trans the following... concern Reports PIO-02-03, and PIO-01-06 on Main Steam Check and Isolation Valves....

Same Dist: as 9-21-73 Ltr.....#7186

PLANT NAME: Kewaunee

ENCLOSURES:
Responses to question; consist of..... various statements, tables, and Figs.

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Mr. J. F. O'Leary Director
Directorate of Licensing
United States Atomic Energy Commission
Washington, D. C. 20545

Dear Mr. O'Leary:

Subject: AEC Docket 50-305
Kewaunee Nuclear Power Plant
Main Steam Check and Isolation Valve Analysis

Reference: Letter from Mr. K. Kniel to Mr. E. W. James dated October 18, 1973

We are submitting, under separate cover, forty (40) copies of our response to Mr. Karl Kniel's letter dated October 18, 1973. The letter requested additional information regarding supplemental reports PIO-02-03 and PIO-01-06 pertaining to analyses of the main steam check and isolation valves for the Kewaunee Plant.

Very truly yours,

C. W. Giesler
for

E. W. James, Senior Vice-President
Power Generation & Engineering

EWJ:sna

Subscribed and Sworn to
Before Me this 7th Day
of November, 1973

John J. Beckman
Notary Public, State of Wisconsin

My Commission Expires November 16, 1975

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REQUEST FOR ADDITIONAL INFORMATION

Report PIO-02-03, "Maximum Energy of Disc Impact, Main Steam Check and Isolation Valves for Kewaunee Unit 1", September 6, 1973.

1. Rough staff calculations of the amount of impact energy due to valve closure, based on a simplified model, indicate a possible energy level as much as 30% higher than predicted in the report. To enable the staff to adequately evaluate the report:

   a. Supplement the description on page 14 concerning the methods used to compute the pressure drop across the disc; and

   b. Verify the flow area used for calculating the choked flow as expressed in equation (13) on page 15.

Response to Part (a)

Using a simplified model where it is assumed that the back pressure, or pressure downstream of the valve disc, is zero would overestimate the amount of energy due to valve closure, as the actual back pressure is significantly higher than zero. Underestimating the back pressure would overestimate the pressure drop across the disc and, consequently, the torque acting to close it. As explained on page 14, the analysis reported in PIO-02-03 utilizes the actual back pressure for pressure drop calculation.

As the valve closes, the choke area is continuously reduced at a considerable rate. Due to the high speed of closure, the volume downstream of the valve disc does not have time to depressurize to the throat pressure value predicted by Moody's steady state model. In addition, the flow out of the volume downstream of the valve is limited due to choking at the break, at a time when there is a continuous mass flow into the volume from the up-stream side of the valve. This further affects the back pressure. As explained on page 14, the actual thermodynamic pressures in the volumes upstream and downstream of the disc were calculated using the state equations, at a conservative limiting flow rate predicted from the Moody model. All of this pressure drop across the valve between the volumes upstream and downstream of the disc was then conservatively assumed to act entirely across the disc, and in a direction perpendicular to the disc surface, thereby maximizing the predicted torque.

The average pressures calculated in the volumes upstream and downstream of the isolation valve disc during a postulated pipe break is shown as a function of time in Figure 6-17 of Report PIO-02-03. The resulting pressure drop as a function of time is shown in Figure 6-12, and the corresponding torque acting on the valve disc as a function of disc angular position during closure shown in Figure 6-7.

A simplified model can be used to check the results of the energy calculations. From the plot of calculated pressure drop across the disc as a function of disc angular position shown in Figure 1, attached,

$$E = \int T d\theta$$

$$= \int (\Delta p)\ (\text{Disc Area})\ (\text{Moment Arm})\ d\theta$$

Using the trapezoidal rule to perform the integration,

$$E = \sum_{i=I}^{IX} E_i$$

$$= \sum_{i=I}^{IX} (\Delta P_i)\ (\text{Disc Area})\ (\text{Moment Arm})\ \theta_i$$

$$= \sum_{i=I}^{IX} C\ \Delta P_i \theta_i$$

where:

Disc area = 438.18 in.^{2}

Moment Arm = 15.75 in.

C = 438.18 x 15.75 = 6901.3 in.^{3}

thus:

Area	ΔP_i, psi	θ_i, Radians	E_i, 10^6 in.-lb
I	1/2 (55+57	(11) π/180	0.074
II	1/2 (57+72	(15) π/180	0.117
III	1/2 (72+92)	(14) π/180	0.138
IV	1/2 (92+127)	(14) π/180	0.185
V	1/2 (127+154)	(7) π/180	0.118
VI	1/2 (132+170)	(5) π/180	0.091
VII	1/2 (170+218)	(4) π/180	0.093
VIII	1/2 (218+297)	(5) π/180	0.155
IX	1/2 (297+388)	(5) π/180	0.206

Total Impact Energy = $\sum E_i = 1.177 \times 10^6$ in.-lb

This result compares very well with the calculated value of 1.162×10^6 in-lb. given in Table 6-2 of Report PIO-02-03.

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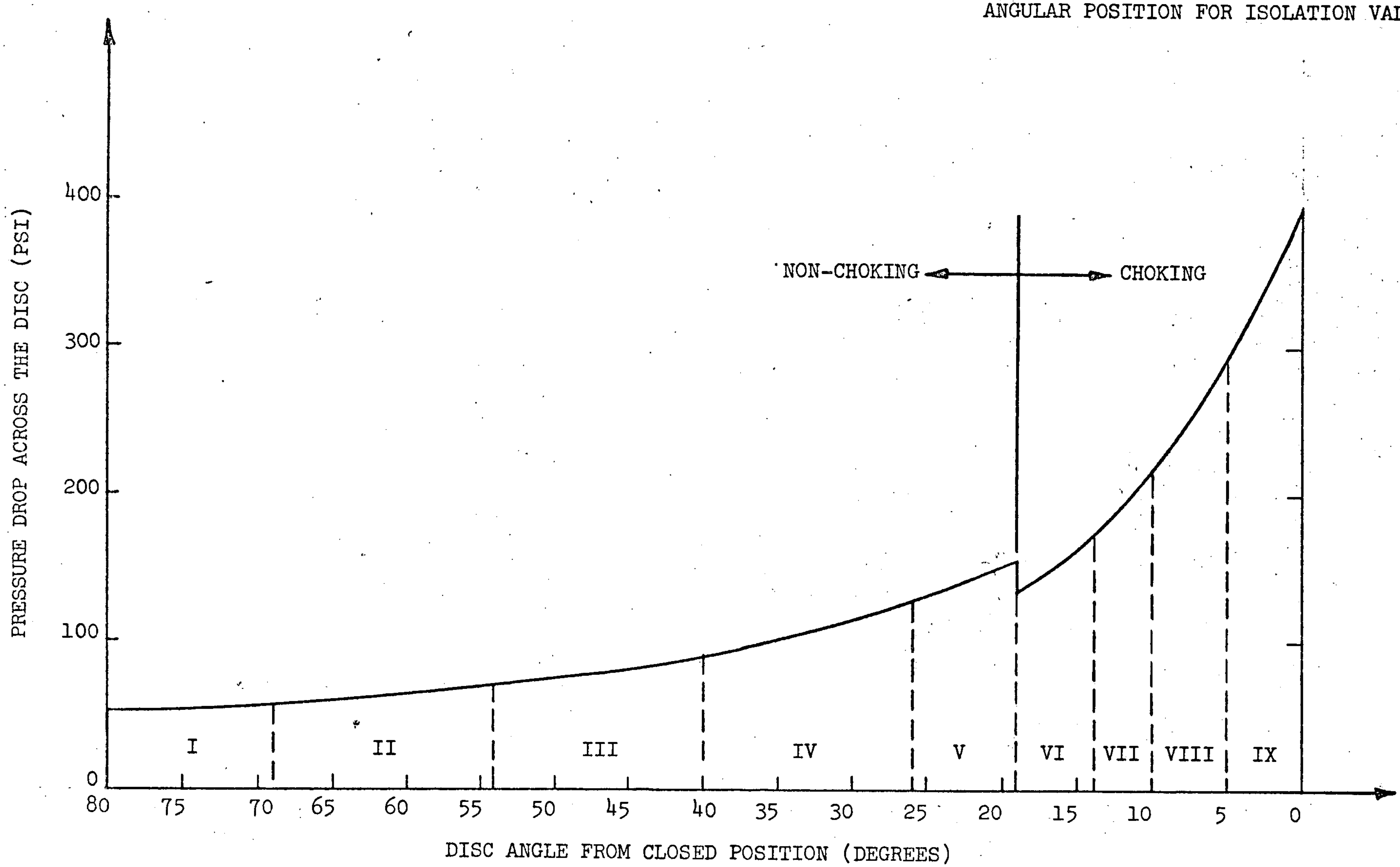
FIGURE 1: PRESSURE DROP ACROSS THE DISC VS DISC ANGULAR POSITION FOR ISOLATION VALVE ANALYSIS



Response to Part (b)

There are three possible limiting flow areas for calculating choked flow at the valve:

(1) the port area

(2) the area between the disc rim and the inside surface of the valve body

(3) the area between the disc and the port opening

Of the first two, it can be seen from inspection of the valve drawings that the port area is always smaller. During the initial phase of closure, the port area is smaller than area (3). However, it is obvious that at the smaller angles just before the valve reaches the seat, area (3) will be less than the port area and must be considered. An exact calculation of this complicated geometric area at the various disc angular positions is difficult to determine. For the analyses of Report PIO-02-03, a simplified conservative correlation of this area was determined, equation 13 described on page 15 of the report. An expression for determining the flow area between the disc and the port opening, area (3), is derived below and compared with equation 13.

This geometric flow area is illustrated in Figure 2a, attached. The flow area is given by:

$$A_{flow} = \int_{o}^{2\pi} S dr$$

$$= \int_{o}^{2\pi} S(\theta, \alpha) \; R_p \; d\alpha$$

$$= 2\int_{o}^{\pi} S(\theta,\alpha)\ R_p\ d\alpha$$

Assuming that for small angles Sin $\theta \approx \theta$, the function $S(\theta,\alpha)$ at α values of 0, $\pi/2$ and π is given by:

$$S_1 = S(\theta,\ \alpha=0) = (L - R_D)\theta$$

$$S_2 = S(\theta,\ \alpha=\pi/2) = L\theta$$

$$S_3 = S(\theta,\ \alpha=\pi) = (L+R_D)\theta$$

From above, the function $S(\theta,\ \alpha)$ can be empirically described as

$$S(\theta,\ \alpha) = (L - R_D \cos \alpha)\theta$$

Therefore,

$$A_{flow} = 2\int_{o}^{\pi} (L - R_D \cos \alpha)\theta\ R_p\ d\alpha$$

$$= 2\ R_p\ \theta\ \pi\ L$$

In the derivation of equation 13, a simplified conservative assumption was made in using the area of the curved surface of a cylinder of port radius and cut by a plane at angle θ.

Therefore, from Figure 2b,

$$A_{flow} = 1/2 \text{ (area of curved surface of cylinder)}$$

$$= 1/2 \left[(2\pi) R_p (2R_p \tan\theta) \right]$$

$$= 2\pi R_p^2 \tan\theta$$

A comparison of the two expressions is given below for several values of θ.

θ	A_{flow}, equation 13 ($2\pi R_p^2 \tan\theta$), in^2	A_{flow} between disc and port opening ($2\pi R_p \theta L$), in^2
5°	64.24	93.35
10°	129.46	186.71
15°	196.74	280.06
20°	267.24	373.42

It is clear from above that the simplified expression, equation 13 of Report PIO-02-03, yields the minimum area and thus is the most conservative.

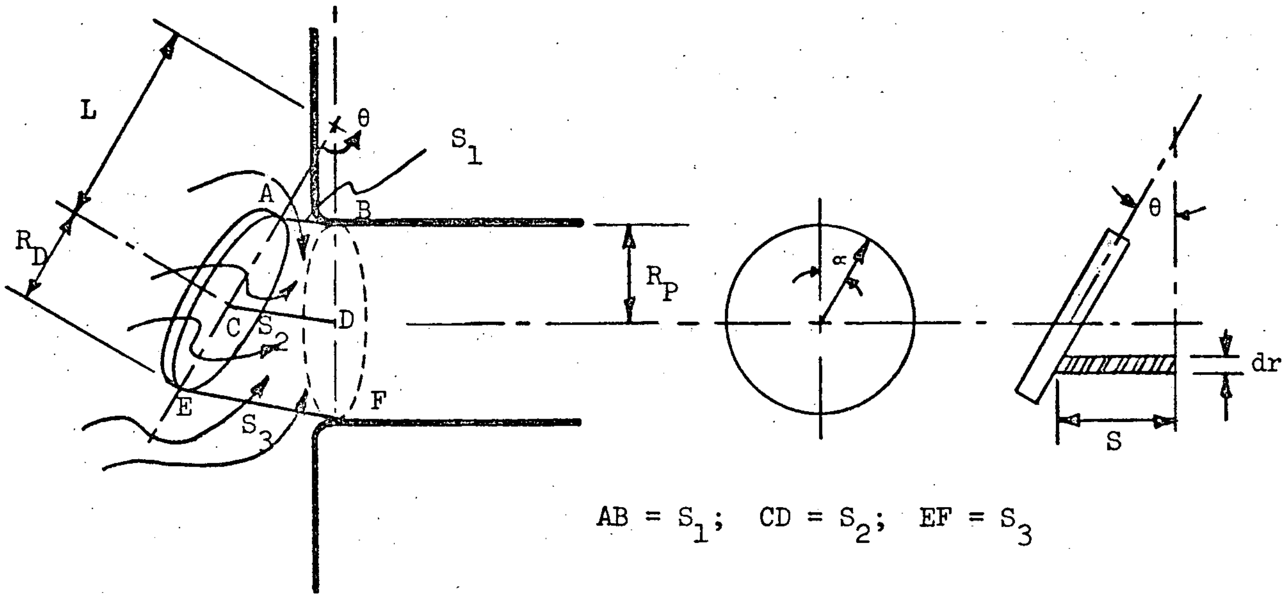


FIGURE 2a = FLOW AREA BETWEEN DISC AND PORT OPENING

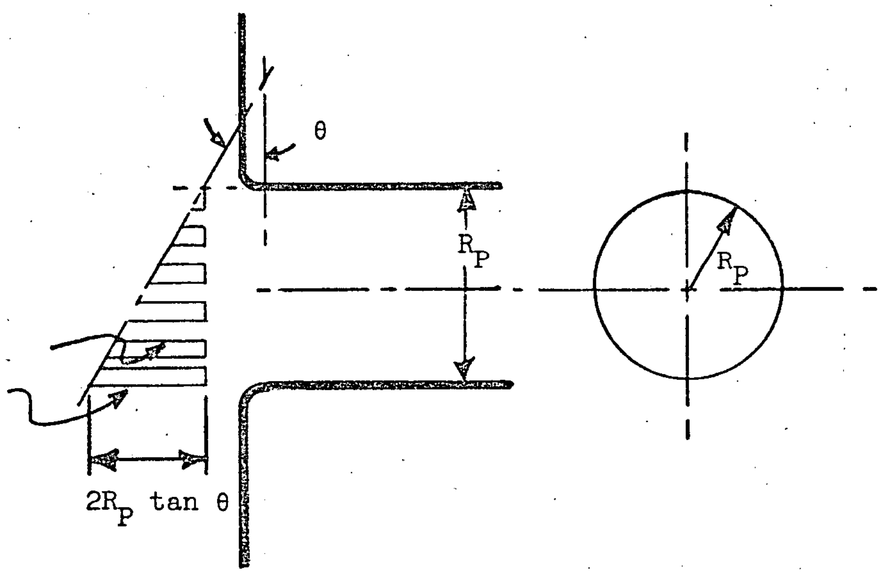


FIGURE 2b - FLOW AREA USED TO DERIVE EQUATION 13 IN REPORT PIO-02-03

REPORT PIO-01-06, "Structural Analyses of Main Steam Check & Isolation Valves for Prairie Island, Unit 1", September 14, 1973.

1. Axisymmetric impact loadings as a result of disc closure are assumed. Since the distribution of impact energy is related to the distance of each mass unit to the center of rotation, the loading along one side of the disc may be substantially higher than the computer value. Provide information to substantiate the existence of adequate design margins to cope with the axisymmetry approximation used.

Response

The question concerning the symmetry of disc loading in view of the non-symmetrical velocity distribution at impact was addressed with the analyses described in Sections 5.3 and A.4.2 of Report PIO-01-06. The results were discussed in the last paragraph on page 18. The analyses were conducted with the aid of the computer program PIPERUP (Reference 7 of PIO-01-06). PIPERUP performs nonlinear dynamic analyses of three dimensional systems.

The analyses were based upon the simplifying assumption of concentrating all mass at the center of the disc. This is considered to be a reasonable representation of the true mass distribution, which is illustrated in Figure A.1-2 of the report, PIO-01-06. Figure A.1-2 shows the proportional magnitude of mass distribution for a unit section through the disc, including a representative mass for the contributing portion of the tail link applied at the disc center. It can be seen that most of the mass is actually located at the center, as represented in the analyses.

2. Table 6-1, page 20, indicated that the valve body and valve seat weld will experience a significant amount of local plastic strain due to spurious valve closure. Such deformation may introduce a certain amount of configuration change and a loss of material ductility. Provide additional information to demonstrate that after the assumed three spurious closures, the seat body of the isolation valve will still meet the initial design criteria and perform the intended safety function.

Response

In order to evaluate the effects of a reduction in the ductility of the valve seat due to repeated spurious trips, it is appropriate to consider an energy balance which is dependent on three quantities: (1) total available strain energy, (2) kinetic energy absorbed by the seat for the design trip, and (3) the potential energy lost to strain hardening as a result of the spurious trips.

To establish a value for the available strain energy of the two valve seat components, body and weld, the force-deflection conditions at maximum allowable strain levels must be determined from the analyses provided in Report PIO-01-06. However, as the allowable strains of 12 and 20 percent (Table 6-2) occur at loads which are beyond the range of the available analyses, it is necessary to extrapolate the data to the points of allowable strain. By employing an extrapolation of the curves shown in Figures A.2-2 and A.2-3 of Report PIO-01-06, the available strain energies associated with the allowable strains are found to be 0.192×10^6 in-lb. and 0.440×10^6 in-lb. for the body and weld respectively. These extrapolated curves are shown in Figures 3 and 4 attached. Upon comparing these energy values, it is apparent that the valve body is the critical component for establishing the spurious trip limit.

For the design trip (kinetic energy = 1.35×10^6 in-lb.), the amount of energy absorbed by the valve seat is dependent on the characteristics of the seat materials at the time of impact. These characteristics for a particular time are governed in part by the spurious trip history of the valve. In order to define material properties for this analysis, it is conservatively assumed that the valve seat is of "virgin" material and will thus absorb a greater portion of the input energy than would a seat of "hardened" material. Moreover, the contribution of the weld is neglected, so that the body is assumed to absorb all of the energy input to the valve seat. Upon applying the forgoing assumptions in conjunction with Figures A.2-2 (Total Force Versus Equivalent Deflection) and A.3-2, (Impact Force Versus Input Energy), the energy absorbed by the seat during a design trip is found to be 0.082×10^6 in-lb.

By postulating repeated occurences of the worst case spurious trip (130% full load, kinetic energy = 0.15×10^6 in-lb.) and applying the principle of isotropic strain hardening, the potential energy loss is given by

$$U_L(n) = \left[U_L(1) + (n-1)\ u\right]$$

where

$U_L(1)$ = loss from first trip = 0.0423×10^6 in-lb. (From Figures 4 and 5).

u = average loss per trip after first trip = 0.0006×10^6 in-lb. (from Figures 4 and 5).

n = number of spurious trips

The allowable number of worst case spurious trips prior to a design trip is determined by considering the energy constraint:

$$U_L \leq U_A - E_A$$

where

U_L = total energy loss in seat

U_A = total available strain energy in seat

E_A = energy absorbed by valve seat during design trip

When the above relationship is solved for the appropriate energy values, a lower bound limit for the number of spurious trips is established. For the valve body the energy balance is given by:

$$\left[.0423 + .0006\,(n-1)\right] \times 10^6 \leq 0.192 \times 10^6 - 0.082 \times 10^6$$

which is satisfied when n is equal to 114. Thus, it is concluded that the seat portion of the valve exhibits sufficient ductility to withstand at least 114 spurious trips prior to a design trip while maintaining the capability to meet the initial design criteria. Furthermore, after three spurious closures, when the valve disc has reached the critical condition, it is apparent that the valve seat has a considerable amount of unused strain energy.

FIGURE 3

VALVE SEAT STRAINS VERSUS TOTAL APPLIED LOAD

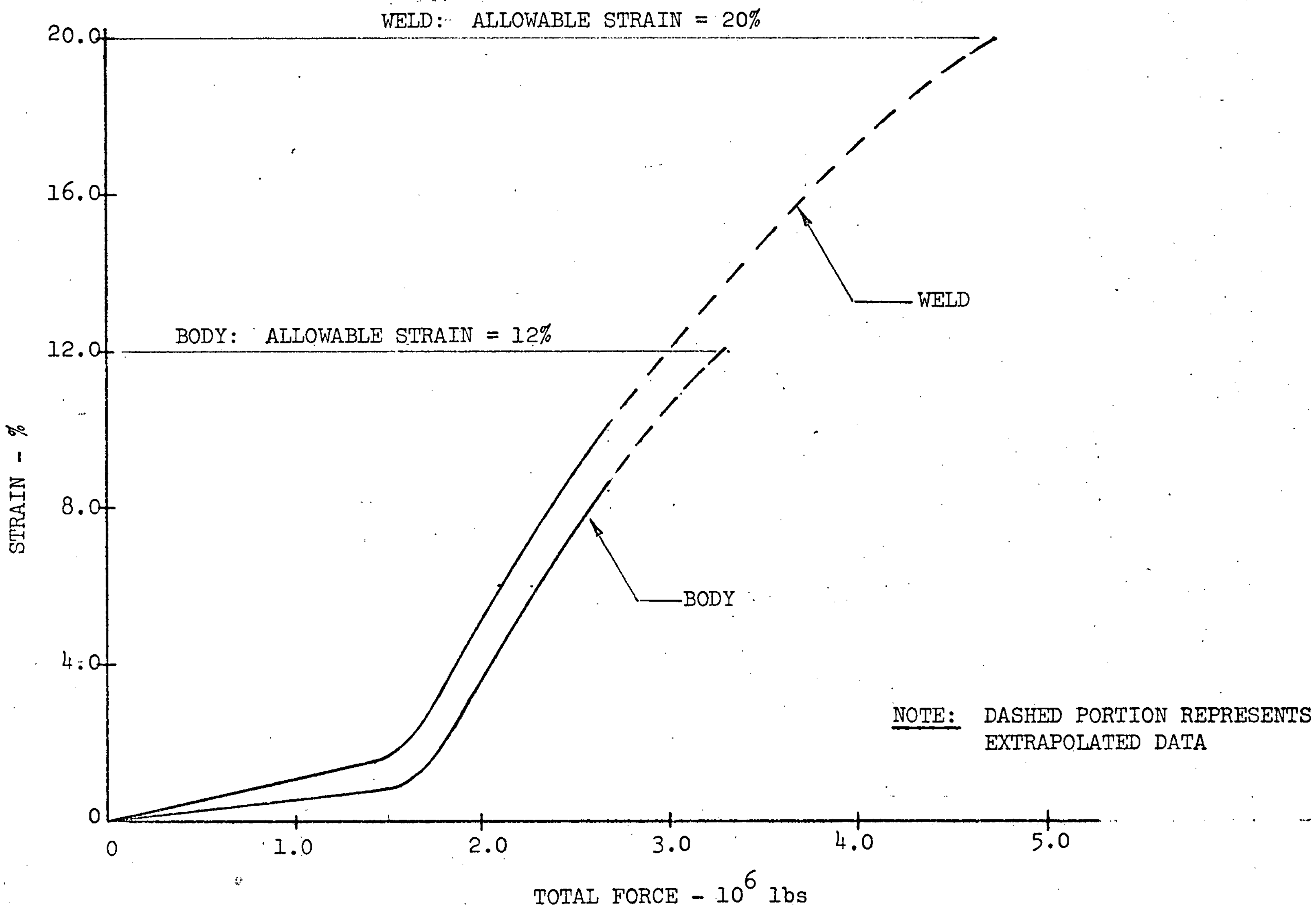


FIGURE 4

TOTAL FORCE VERSUS EQUIVALENT DEFLECTION FOR VALVE SEAT

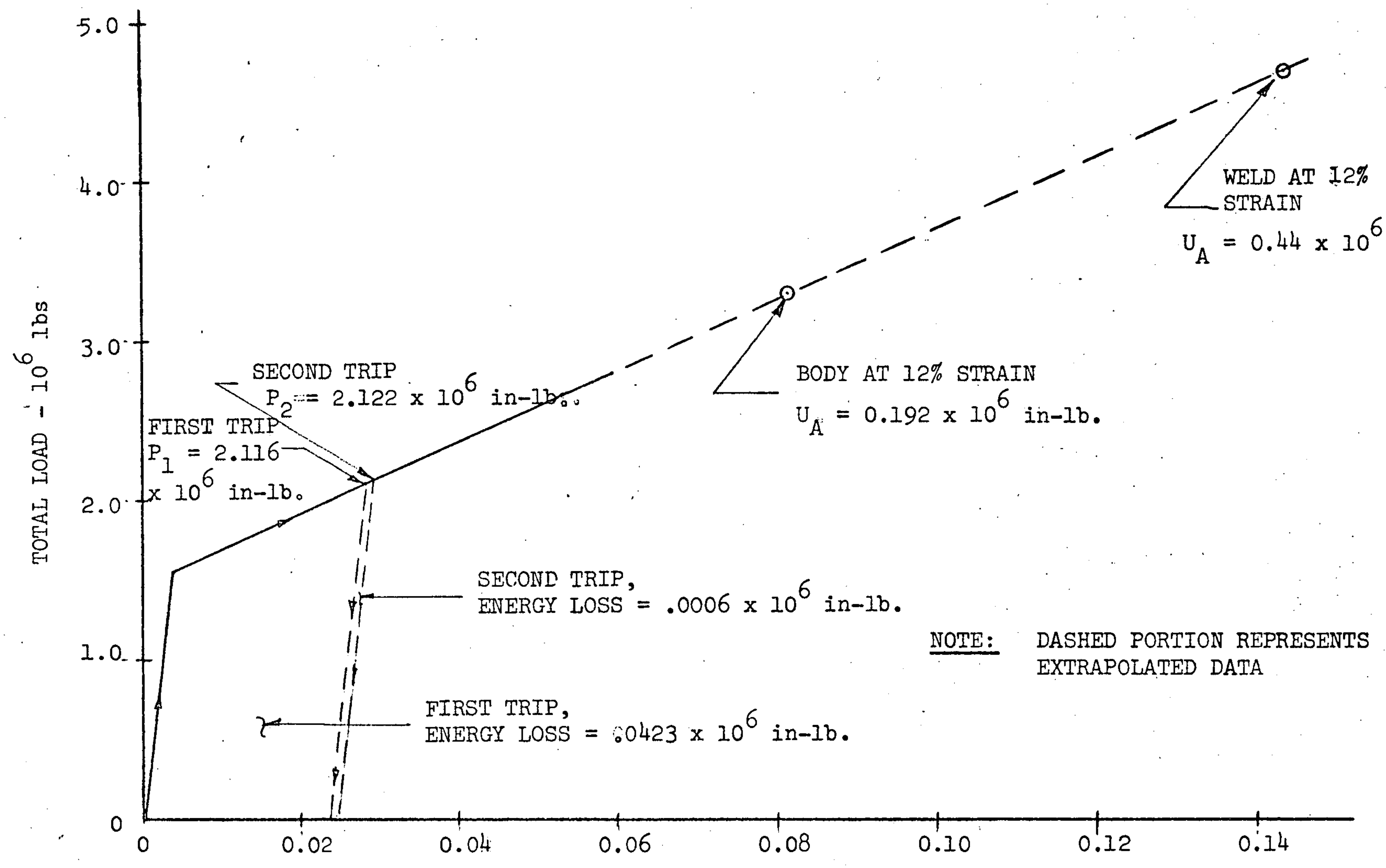


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FIGURE 5

TOTAL FORCE VERSUS EQUIVALENT DEFLECTION
DISC AND VALVE SEAT SYSTEM