Evaluation of Feedwater Containment
Isolation Check Valves for a
Hypothetical Pipe Rupture Condition

for

Limerick Generating Station

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CONTENTS

1. Summary

The Limerick Generating Station feedwater containment isolation check valves have been shown to retain their pressure containment and structural integrity for the conditions of a hypothetical pipe rupture assumed to occur upstream of the check valve outside of containment. A simplified fluid dynamics and stress analysis was performed to determine check valve closing time, disk closing velocity, and valve seat stresses. The piping and valve were shown to have the capacity to sustain the pressure surge from check valve closure.

2. Introduction

Early in the engineering design of the Limerick Generating Station (LGS), operational and faulted condition loads were included in the piping system design. One of the loads considered was pressure surges from valve closures in the piping systems. During evaluation of the feedwater piping system a faulted condition of pipe rupture outside of containment was chosen for evaluation of the containment isolation check valves and feedwater piping inside containment. This event is more severe with regard to check valve disk closing velocity and hence fluid pressure surge than other events postulated for evaluation purposes, such as a normal operating feedwater pump trip or an off-normal feedwater pump shaft seizure. The purpose of the evaluation was to determine that the containment pressure boundary integrity provided by the feedwater piping and check valves is maintained during and after the improbable, hypothetical pipe rupture event.

The evaluation was analytical and performed by mathematical modeling of portions of the feedwater (FW) piping system, from the piping upstream of the check valve located outside of the containment wall to the reactor pressure vessel FW inlet nozzles, and including the check valve closest to the pipe rupture, which was assumed to be located between the check valve and the junction of the two feedwater trains. The portions of the system modeled are shown in Figure 1. Pressures and stresses were calculated with the analysis model and compared to values that could be sustained by the feedwater system.

The LGS feedwater system consists of two trains entering the primary containment. The two trains are similar in configuration and equipment, therefore, only Train A was analyzed. Each train has two check valves, one inside and one outside the containment wall. The outside check valve was assumed to close first, since it is closer to the pipe rupture than the one inside and would be subjected to higher fluid velocities during the reverse flow, blowdown part of the analysis. The inside check valve would close also, but following in time the closure of the outboard valve.

The valve types are 24-inch, 900 lb, swing check/motor operated, manufactured by Atwood and Morrill Co.. Design conditions are for 2132 psig and 459°F and they were constructed in accordance with ASME Section III Code, 1971 Edition including 1972 Addenda, to include radiograph and magnetic particle examination. Seat material is Stellite-21. Although the valve design did not include stress requirements from the pipe rupture conditions postulated and analyzed here, the analysis demonstrated that the valve retained its pressure containing integrity throughout and after the hypothetical, imposed conditions.

The approach used for evaluation was to calculate: 1) the rate of reverse flow from the reactor vessel to the pipe rupture location, 2) the rate of check valve disk closure, 3) the energy observed from disk impact, 4) the fluid pressure surge from disk closure, and 5) the effects of the pressure surge on the associated piping. The criteria for success was that the mechanical impact of the disk and the pressure surge does not cause failure of the valve or piping to lose pressure containing capability. Material yielding for this faulted, pipe rupture hypothetical condition is permissible, but failure defined as exceeding the ultimate stress in a member is not permissible. Although the analytical method is relatively simplified, being performed before the widespread availability and use of sophisticated fluid dynamics and elastic-plastic stress analysis computer codes, it is believed to be conservative. For example, should recent versions of RELAP5 and ANSYS be used to reperform the analysis, it is expected that the results would be equal to or less than the conservatism used in the simplified approach described in the following sections of this report.

3. Valve Closing Analysis

This section describes the analysis performed to calculate the check valve disk closing velocity for conditions of reverse flow in the feedwater Train A piping from the reactor vessel to the pipe rupture location.

The mathematical model posed for the analysis is based upon the following assumptions and operation conditions:

- o RPV pressure is 1053 psia, corresponding to normal operating conditions.
- o FW temperature is 425°F.
- o Pipe rupture is assumed to be instantaneous with one pipe cross sectioned area available for flow from the rupture on the RPV side.
- o Flow reversal occurs after a relatively short time, on the order of a few milliseconds.
- o The check valve disc does not move until flow reversal starts.
- o The start of reverse flow is the zero time for the check valve closing transient.
- o Flow chokes immediately at the pipe rupture opening, limiting the blowdown flowrate (Moody maximum flow).
- o Flow choking at the rupture opening causes fluid between the RPV and rupture location to be pressurized above saturation pressure so that the flow is non-flashing in liquid phase.
- o The check valve inside containment remains full open.
- o Pipe and fitting fluid friction losses are included.
- o The check valve disk motion outside containment is based on a torque balance on the disk.

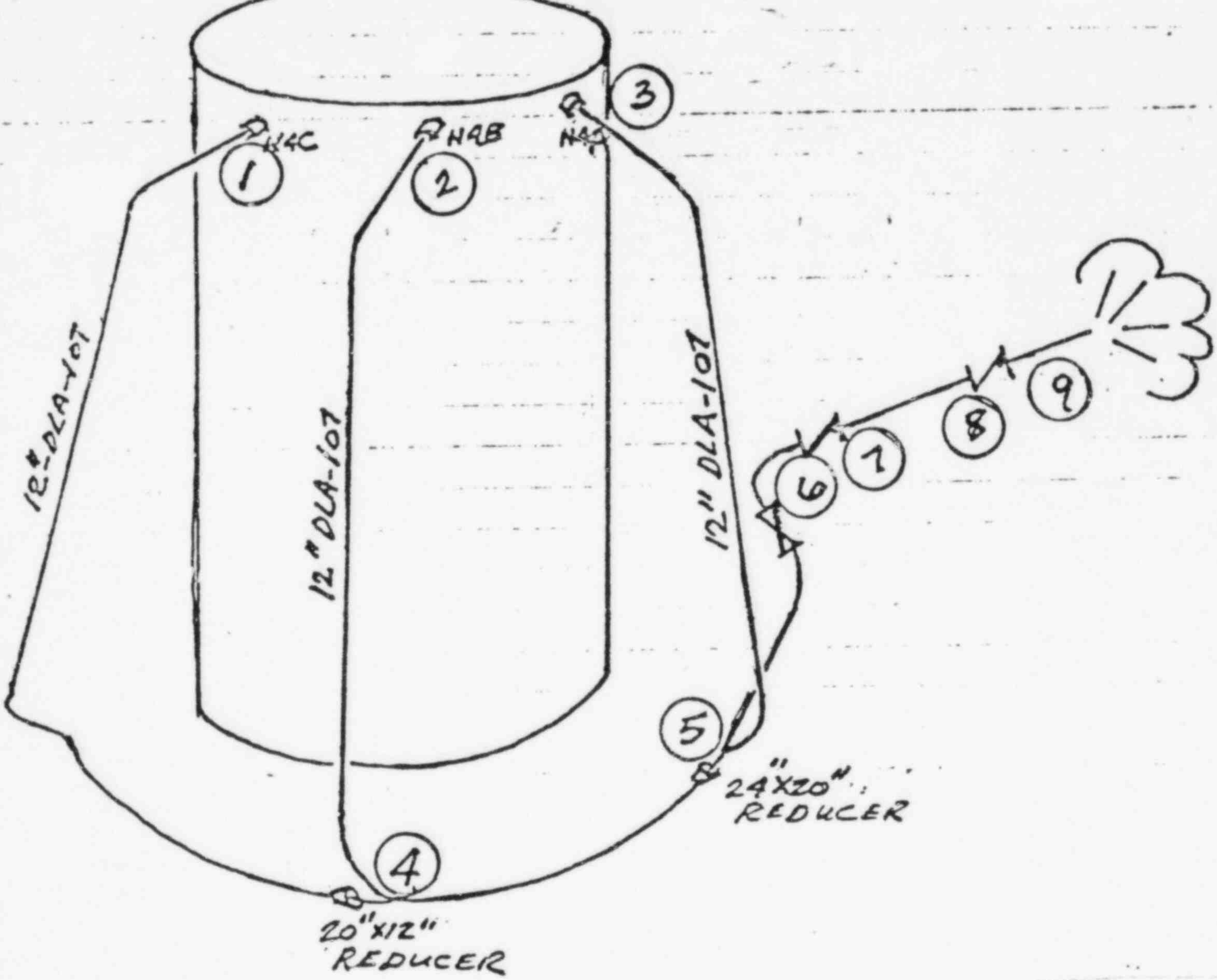


Figure 1. Sketch of FW Lines, RPV to CV

- The disk is closed by fluid drag forces, with the disk flow coefficient varying inversely with disk position.

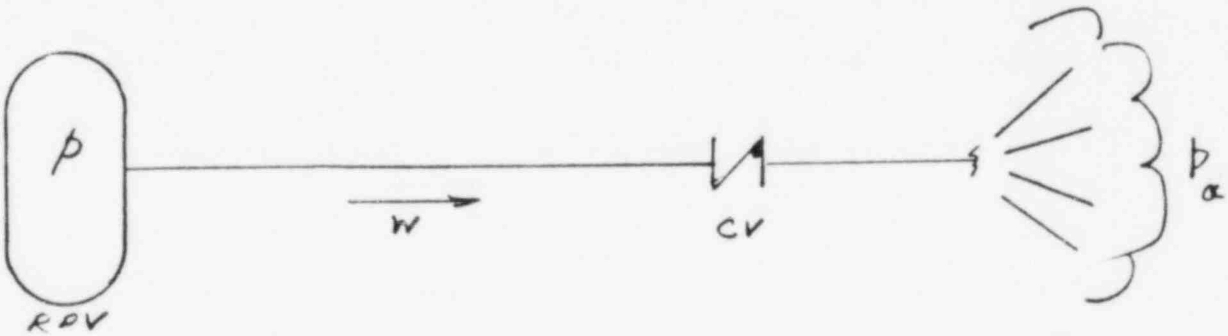
- Feedwater sparger (located just inside RPV) flow resistance is assumed to be zero.

- Flow resistances are assumed constant (square-law friction).

- Flow inertia is included.

- The flow calculation is based on a spatially averaged momentum equation.

Based on the foregoing assumptions, the equation of motion for flow is described as follows:



Using a lumped parameter spatial approximation the momentum equation is

$$I_w \frac{dw}{dt} = p - p_a - R_w W^2$$
$$= \Delta p - R_w W^2 \qquad (1)$$

where $Iw = L/Ag_c$, flow inertia, $\frac{psf}{lbm/sec^2}$

L = length of pipe, ft

A = flow area, ft^2

g_c = constant (32.2 lbm ft/lbf-sec^2)

$R_w = (fL/D)/2g_c A^2$, $\frac{psf}{lbm^2/sec^2}$

ρ = density, lbm/ft^3

f = friction factor (Blasies)

w = flowrate, lbm/sec

p = pressure, psf (psi if divided by 144)

t = time, sec

A torque balance on the disk give

$$I_c R_m \frac{d^2\theta}{dt} = A_c R_c W^2 R_h - M_g R_m / g_c \tag{2}$$

where θ = opening angle of CV, radians

I_c = disk inertia, lbf-sec^2

R_m = mass center radius, ft

A_c = disk area, ft^2

R_h = hydraulic center radius, ft

M = disk mass, lbm

g = acceleration of gravity, ft/sec^2

w = flowrate through CV, lbm/sec and

R_c = CV flow resistance

which is assumed to vary as $1/\theta$ such that there is infinite flow resistance for $\theta = 0$, or as $R_c \to \infty$, then $W \to 0$. Therefore

$$R_c \simeq R_c^o(\theta^o/\theta)$$

where R_c^o = flow resistance at $\theta = \theta^o$

θ^o = full open position

Results of using the foregoing simplified model are as follows:

o The check valve disk closes in .070 sec.

o The closing velocity at time of impact is 65 rad/sec.

4. Valve Stress Analysis

For evaluation of check valve pressure containing integrity for a high velocity disk impact condition, the basic approach was to state that the total strain energy absorbed by the valve seat at the instant of maximum deformation will equal the kinetic energy of the valve disk at the instant of impact with the valve seat. The valve seat absorbs all the energy at impact.

For conservatism, no credit was taken for the following energy absorbing mechanism:

- o disk deformation
- o hinge deformation
- o valve body deformation
- o hinge friction
- o strain hardening or rate of strain effects for the disk seat material

Thus, the basic equation to be solved for a simplified approach is

$$K.E. = I\dot{\theta}^2/2 = \int_0^{X^1} F dx \qquad (3)$$

where θ = disk position

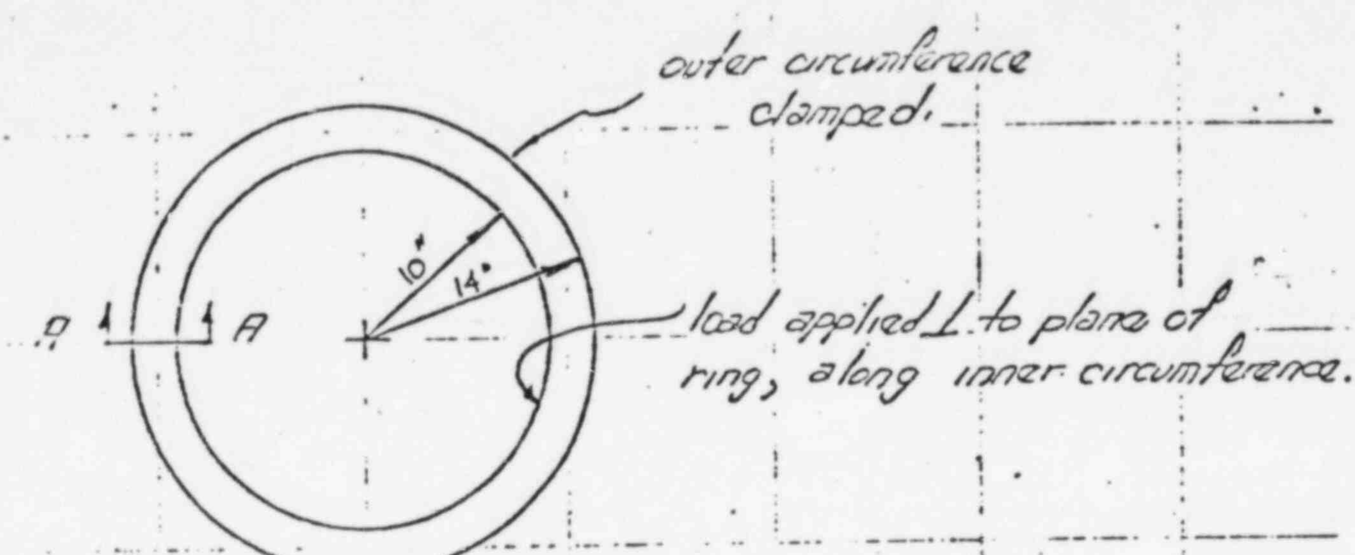
I = disk inertia

F = seat resistance to deformation

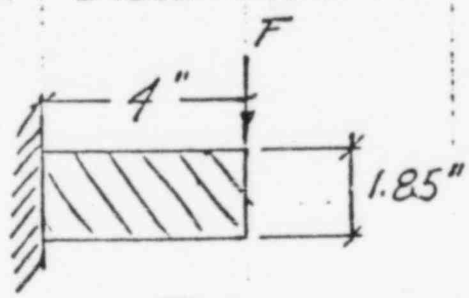
x = seat plastic displacement

In the foregoing section the disk closing velocity was calculated to be 65 rad/sec. For conservatism an impact velocity of 100 rad/sec was used to evaluate the valve seat plastic deformation. For 100 rad/sec the kinetic energy is 70,335 ft-lbf, which was used or evaluation.

The valve seat is idealized as a uniform circular ring, 1.85" thick, with 14-inch outer radius and 10-inch inner radius.



IDEALIZATION OF VALVE SEAT



The valve seat was modeled as an assemblage of six discrete, bilinear, elastic-plastic elements (see Figure 2).

The valve seat was assumed to be stressed to 50% of yield under the design pressure of 2132 psi.

The elastic-plastic elements were assumed to have a plastic section modulus of 1.5, and were assumed to fail at a ductility factor of 30.

The valve seat was found to yield around it entire circumference. No elements failed.

For seat element strength assume valve seat stressed to 50% of yield @ design pressure of 2132 psi and assume elements have plastic section modulus of 1.5 (i.e. same as for rectangular cross-section beam), then the yield load capacity of each element is

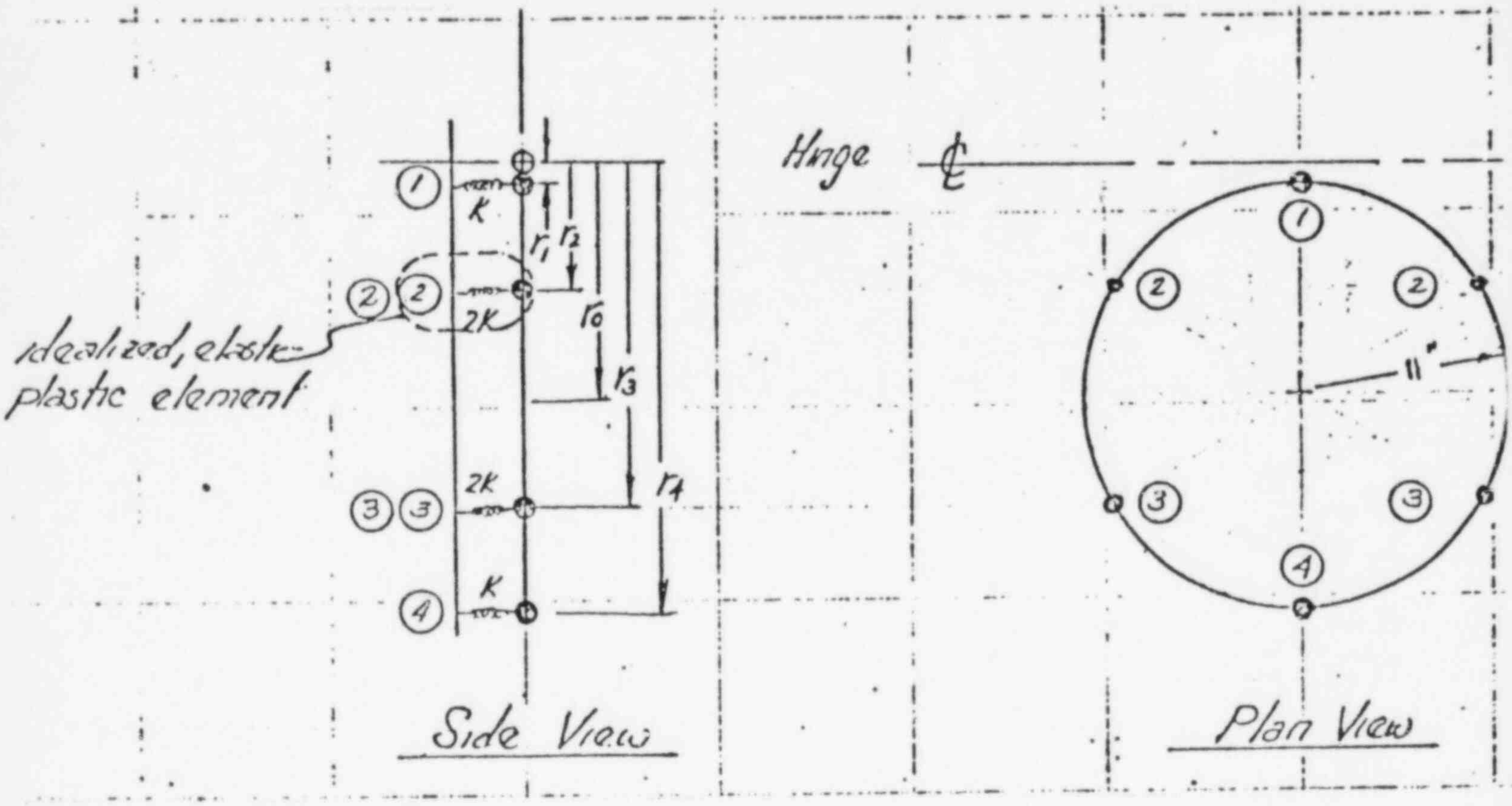


Figure 2. Discrete-Element Model of
Valve Seat and Disk

$$Fy = (1/6)(380\ in^2)(2132\ psi)(1/.5) = 270,000\ lbs$$

and plastic load capacity of each element is

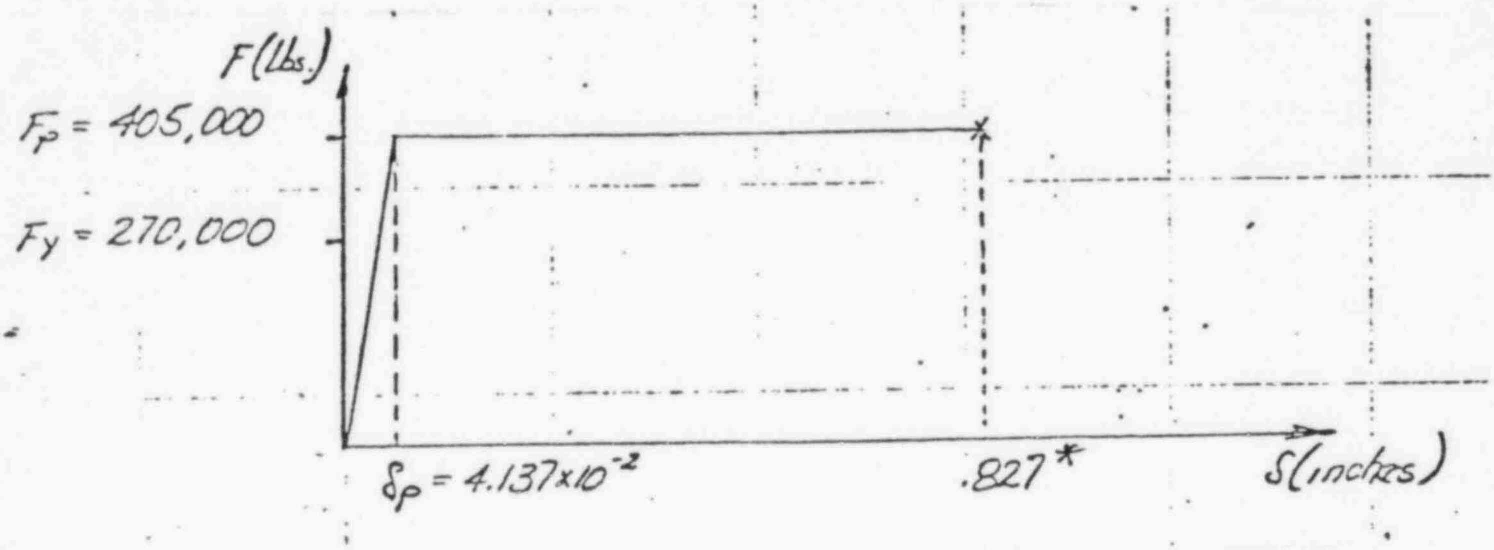
$$Fp = 1.5\ Fy = 405,000\ lbs$$

For element stiffness, the elastic stiffness of each element is taken at 1/6 the stiffness of the valve seat "ring" idealization.

Roark, Formulas for Stress and Strain, gives a formula by which the stiffness of the ring may be determined (see pg. 216, Case 18). (4)

$$K_{elem} = \frac{(1/6)(30 \times 10^6)(1.85)^3}{(.0165)(14.0)^2} = 9.79 \times 10^6\ lb/in$$

For element force-displacement relationship assume a plastic section modulus = 1.5 to give



*Note: It assumed that if any element is forced to deform more than 0.827 inches (ductility factor 30) it will cease to carry load (as if it has been sheared off).

$K_B = K_{elem} (\phi_1{}^2 + 2\phi_2{}^2 + 2\phi_3{}^2)$

$KB = 46.78 \times 10^6$ lb/in (element 4 has yielded)

$K_C = K_{elem} (\phi_1{}^2 + 2\phi_2{}^2)$

$K_C = 6.375 \times 10^6$ lb/in (elements 3 and 4 have yielded)

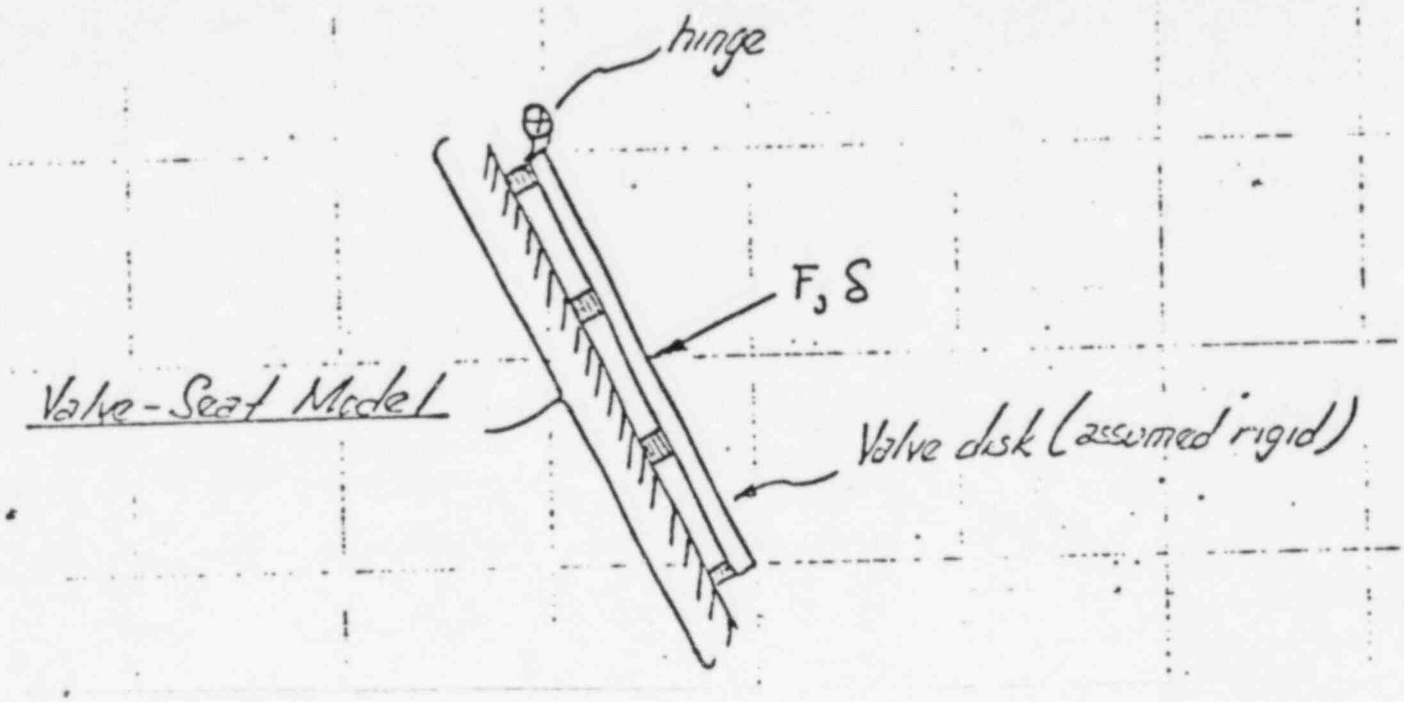
$K_D = K_{elem} (\phi_1{}^2)$

$K_D = .1576 \times 10^6$ lb/in (elements 2, 3, and 4 have yielded)

In Figure 3 is presented the force-displacement curve for the valve-seat and disk assembly.

The area under the curve represents absorbed energy or "amount of kinetic energy expended," in deforming the seat.

	in-lbs	ft-lbs
1. (.022088)(.5)(1791779)	19788	1649
2. (.006711)(.5)(1791779 + 2105720)	13078	1090
3. (.044617)(.5)(2105720 + 2390200)	100297	8358
4. (.252332)(.5)(2390200 + 2430000)	608145	50679
5. (.115790)(2430000)	281370	23447
6. (.134167)(1672650)	224414	18701
7. (.891895)(506250)	451522	37627
8. (5.0442)(52650)	265577	22131
		163682 ft lbs



The "force-displacement characteristics" of the valve-seat and disk "assembly" will be defined in terms of Force and Displacement at Center of Valve Disk.

The contribution to the stiffness of the "assembly" made by an individual element depends upon the location of the element, relative to the hinge axis and the center of the disk.

For any element "i", this geometric parameter will be defined as

$$\phi_i = \frac{r_i}{r_c}$$

where r_i = distance from hinge axis to element
r_c = distance from hinge axis to center of disk.

The values of "assembly" stiffness for the various stages of yielding of the valve seat model are as follows:

$$K_A = K_{elem} (\phi_1{}^2 + 2\phi_2{}^2 + 2\phi_3{}^2 + \phi_4{}^2)$$

K_A = 81.12 x 10^6 lb/in (all elements elastic)

7. References

1. Limerick Generating Station, Final Safety Analysis Report.

2. Atwood and Morrill Co., Design Drawings for 24-inch 900 lb Swing Check Valve, 1972.

3. V. L. Streeter, E. B. Wylie, "Fluid Mechanics," McGraw-Hill Book Co., 1971.

4. R. J. Roark, "Formulas for Stress and Strain," McGraw-Hill Book Co., Fourth Edition, 1965.

6. Conclusions

A simplified analysis has been conducted to demonstrate pressure containment integrity of the feedwater isolation check valves under the adverse conditions of a hypothetical pipe rupture in the feedwater piping just upstream of the check valve. Although the analysis predicts a high closing velocity of 65 rad/sec, the valve seat was shown to be able to absorb the energy by a factor of over two before failure for a closing velocity of 100 rad/sec. The pressure surge for a finite valve disk closing time of .070 sec, which was predicted by the analysis, was estimated to be approximately 2157 psi which is easily within the pressure capability of the check valve and feedwater piping pressure design. The conclusion was reached that containment isolation is achieved during this hypothetical pipe rupture event.

The 24 inch pipe adjacent to the check valve is schedule 120 and the 12 inch pipe adjacent to the RPV is schedule 80. These pipes can sustain 2157 psig for a statically applied internal pressure loading within the constraints of pipe rupture loading conditions.

5. Pipe Pressure Surge

When the check valve closes under pipe rupture, reverse flow conditions the relatively fast closure causes a pressure surge to propagate from the closed check valve to the RPV and return. For a finite closure time the pressure surge can be estimated as

$$\Delta p = \frac{\rho u}{g_c}\left(\frac{2\ell}{\tau}\right) = \frac{G}{g_c}\left(\frac{2\ell}{\tau}\right) \qquad (4)$$

where Δp = pressure surge, psi

G = mass flux, lbm/ft^2-sec

ℓ = average pipe length, ft

τ = valve closing time, sec

The valve closing time was calculated to be .070 sec for a mass of 2200 lbm/ft^2-sec. The average length pipe to the RPV is approximately 70 ft. From Eq. (4) above, a pressure surge of 2157 psi was calculated. Note that in Eq. (4) the $2\ell/\tau$ part of the expression takes the place of the sonic velocity term for the acoustic water hammer equation for instantaneous valve closure. (If an instantaneous closure were to be assumed with a sonic velocity of 4500 ft/sec, a pressure surge of over 4800 psi could be calculated.)

For finite valve closure, the pressure surge should take the form approximately of a sine wave function for the first half cycle. The base is for the first half circle, π radians (there is no second half which would represent a negative surge of the same amplitude). The dominant frequency of this pressure surge forcing function is $1/(2 \times .070) = 7.14$ cps = 44.6 rad/sec. This frequency is much lower than the natural frequency of the check valve major components and the piping hoop stress frequency. Therefore, the dynamic load factor is unity for application of the maximum surge pressure to the check valves or feedwater piping.

The check valve pressure design is 2132 psig, so that it can withstand a pressure of 2157 psig. (The 1% difference between these two pressures is within the uncertainty of either the check valve fabrication process or the pressure surge calculation.)

Energy absorption capacity = 163,700 lbs-ft

Energy to be absorbed = 70,300 ft lbs

Maximum deflection of center of disk:

$$.32575 \text{ in} + .11579 \left(\frac{70300 - 50680}{23447}\right)$$

$$= .32575 + .09689$$

$$= .42264 \text{ in}$$

$$\theta = \arcsin\left(\frac{.42264}{12.6}\right) = 1.9° \equiv \text{Angular displacement of valve disk beyond first contact with valve seat.}$$

The results of the valve seat stress analysis indicates that the seat does not fail, with 43% of its energy absorbing capacity used and 57% remaining. This gives a margin of over two before failure would occur.

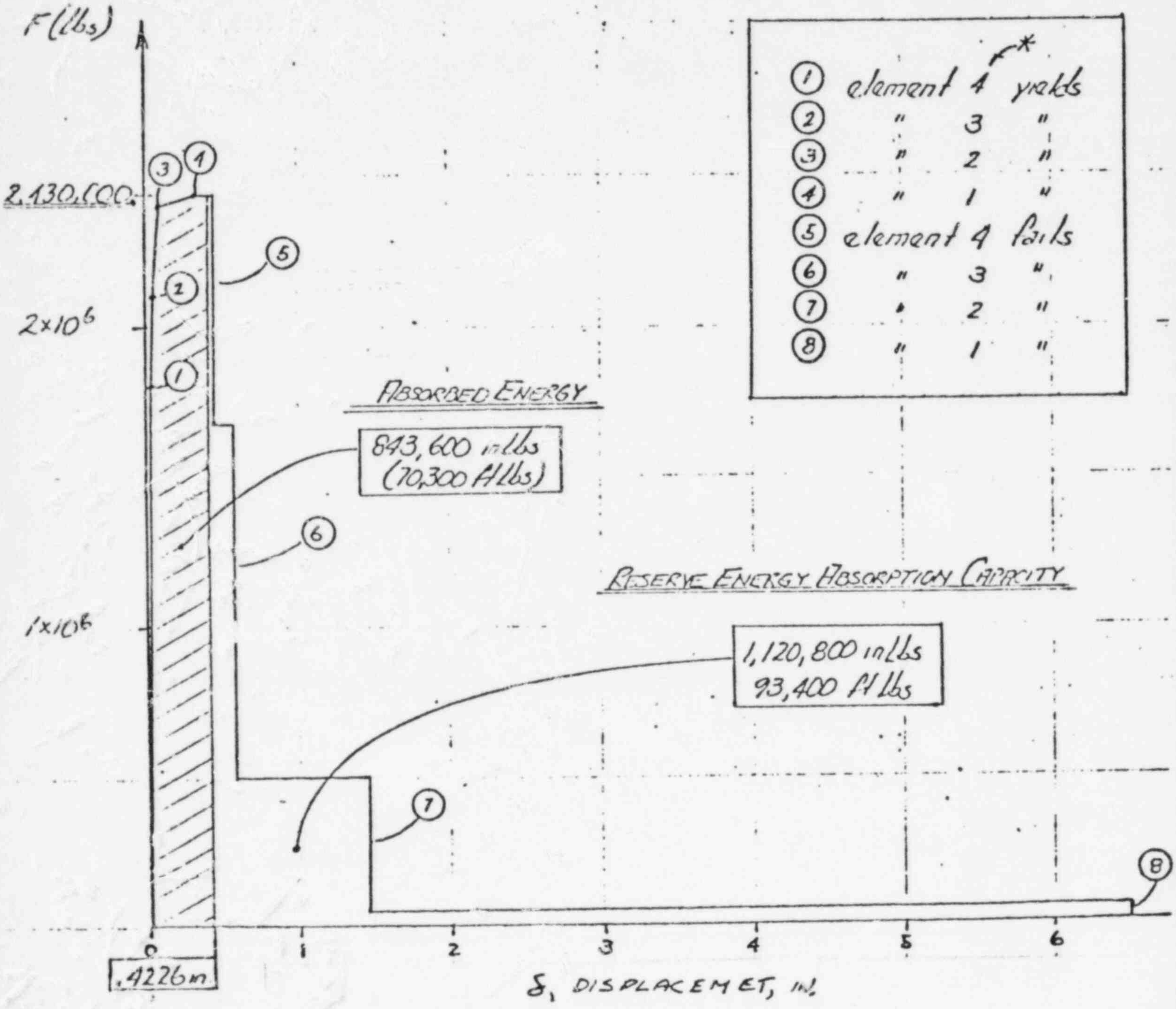


Figure 3. Force-Displacement Curve

F = force @ center of disk,

δ = displ. @ center of disk