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

for

Limerick Generating Station

September 1984

1SS021

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 outermost containment isolation valves (41-1F032 A/B). The computer code RELAP5 was used to determine check valve closing time, disk closing angular speed, and acceleration time histories and the pressure surge in the FW line due to check valve closures following line break. A stress analysis was then performed to evaluate the structural integrity of the 24" swing check valves 41-1F074 A/B. Check valves 41-1F074 A/B were chosen because they are structurally weaker than the inboard check valves 41-1F010 A/B. Check valves 41-1F074 A/B and 41-1F010 A/B perform the isolation function for a feedwater line break outside of containment. The piping and valves were shown to have the capacity to sustain the pressure surge and slamming from check valve closure.

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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 41-1F074 A/B 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. 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 original evaluation was analytical and performed by mathematical modeling of portions of the feedwater (FW) piping system from the reactor pressure vessel FW inlet nozzles to the hypothetical break location between check valve 41-1F074 and the FW pump. The portions of the FW system modeled are shown in Figure 1. Recently, a confirmatory analysis using the RELAP5 computer code to obtain check valve angular disc speeds and accelerations was performed. From these the analytically calculated stresses were compared to values that could be sustained by the feedwater system. This report discusses this confirmatory analysis.

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 three check valves, one inside and two outside the containment wall. The two valves closest to the RPV, 41-1F010A and 41-1F074A, perform the isolation function for feedwater line breaks outside containment. The structural aspects of these valves were compared and it was found that the outboard valve 41-1F074 is structurally weaker than the inboard. Therefore, the analysis considers the closing of the outboard valve only. For the analysis, the inboard valve 41-1F010 and the outermost valve 41-1F032 were kept fully open throughout the transients.

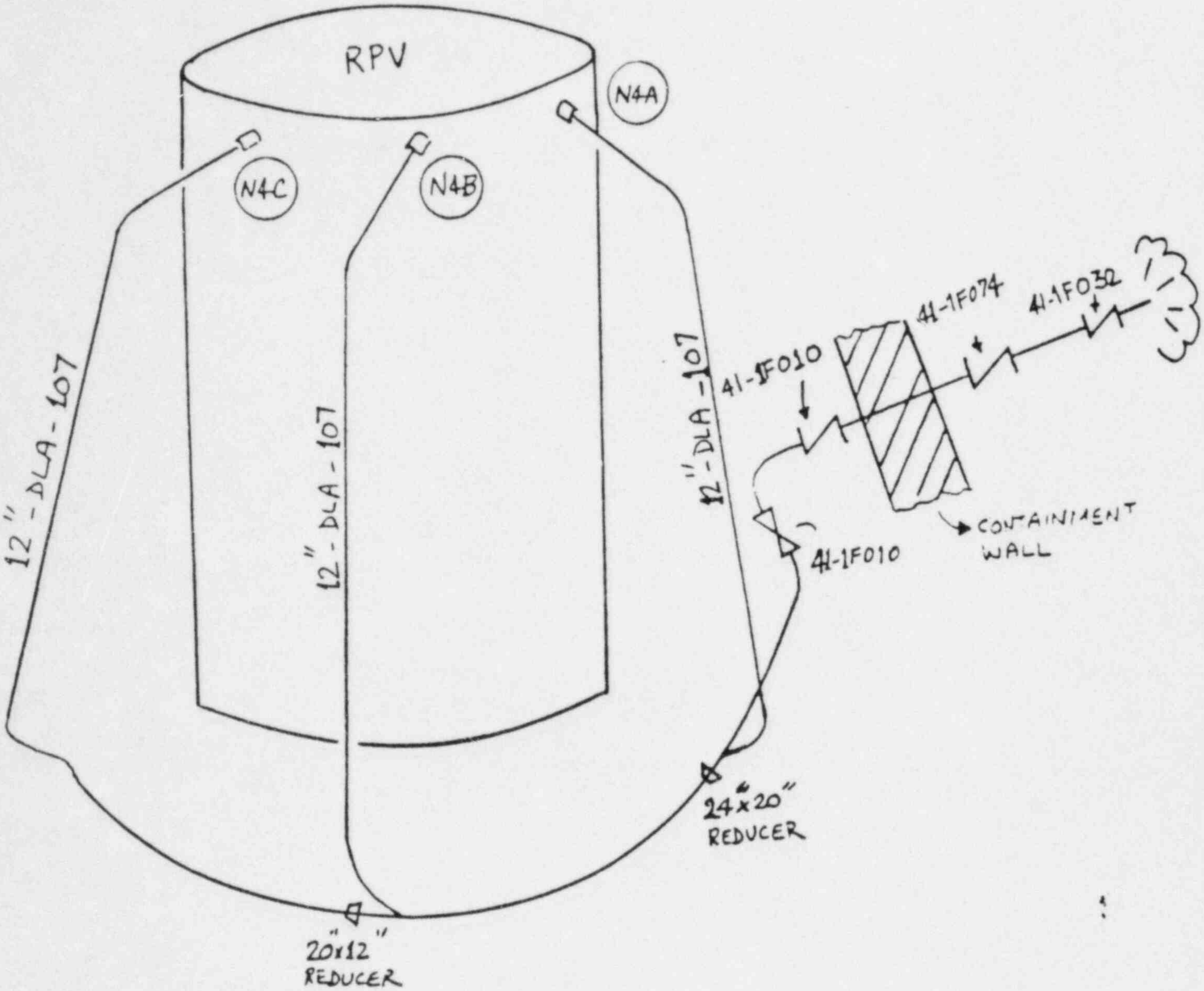

Figure 1. Sketch of FW Lines, RPV to CV

The valves 41-1F074 A/B are 24-inch swing check manufactured by Atwood and Morrill Co.. These outboard valves are pneumatically assisted (Reference 1). However, for the feedwater pipe break transient the pneumatic assist does not come into play during the valve closure (see Section 3.1). 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. 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 the following:

1. The check valve closure based on time dependent fluid conditions at the valve location and the physical flow and geometrical characteristics of the 24 inch check valve in question. The piping from the reactor nozzle to the break location was modeled on the RELAP5/MOD1 computer code. The check valve dynamics equations were incorporated into the RELAP5 input data using the control system options allowed by RELAP5. This part of the evaluation gave the surge pressures in the pipe, the check valve disc angular speed and acceleration time histories.

2. The energy absorbed from disk impact.

3. The plastic work capacity of the valve and the stresses in the valve hinge arm.

4. The effects of the pressure surge on the associated piping inside containment.

The valve structural integrity during a pipe rupture event can then be considered satisfactory if the plastic work capacity of the valve is greater than the kinetic energy of impact and the stresses in the valve hinge pins less than the allowables (50% of the yield strength).

3. Valve Closing Analysis

3.1 Methodology

This section describes the analysis performed to calculate the check valve disk closing angular speed 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 feedwater inlet pressure is 1045 psia, corresponding to normal operating conditions.
- o FW temperature is 420°F.
- o Pipe rupture is assumed to be in one millisecond with one pipe cross sectioned area available for flow from the rupture on the RPV side.
- o The start of the pipe rupture is the zero time for the check valve closing transient.
- o The check valve inside containment and the outermost check valve outside containment remain full open.
- o Pipe and fitting fluid friction losses are included from the reactor to the first check valve outside containment. Only pipe friction losses are assumed from the check valve to the break.
- o The check valve disk motion outside containment is based on a torque balance on the disk. The torques acting on the disk are those due to pressure differential across the disk, weight of the disk and hinge assembly, friction in the disk hinge and any spring that helps or hinders closure.

- The disk is closed by the net closing torque with the disk flow resistance varying with disk position as specified by Atwood Morrill Co..

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

- Full FW flow (14.260×10^6 lbm/hr) is assumed into the reactor from the feed pumps before the break (Reference 2).

- Flow inertia is included.

- The flow calculations in the RELAP5 code are based on the solution of mass, energy, and momentum conservation equations for one dimensional flow. RELAP5 code is expressly written for the analysis of loss of coolant accident.

- The effect of the spring pneumatic assist is ignored because the check valve should be closed due to pipe break before the spring comes into play according to Atwood & Morrill Co.

Details of RELAP5 calculation methodology can be found in Reference 3. The RELAP5 code allows users to model a simple flow device or control system through the use of the control system options. The equations describing the dynamics of the check valve were modeled using this approach. With reference to Figure 1a, the formulation of these equations is given below:

Newton's second law of motion in angular form is

$$\Sigma T = I\ddot{\theta} \tag{1}$$

where ΣT = sum of torques exerted on the check valve disk.

I = valve disk assembly moment of inertia about the hinge. Some equivalent water mass could be allowed for in the calculation of I. However, this was conservatively neglected.

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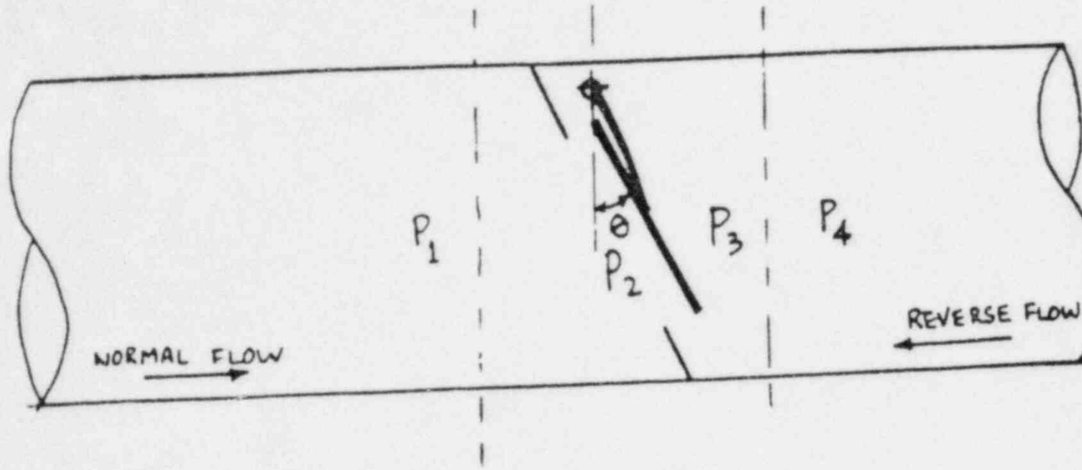

FIGURE 1a

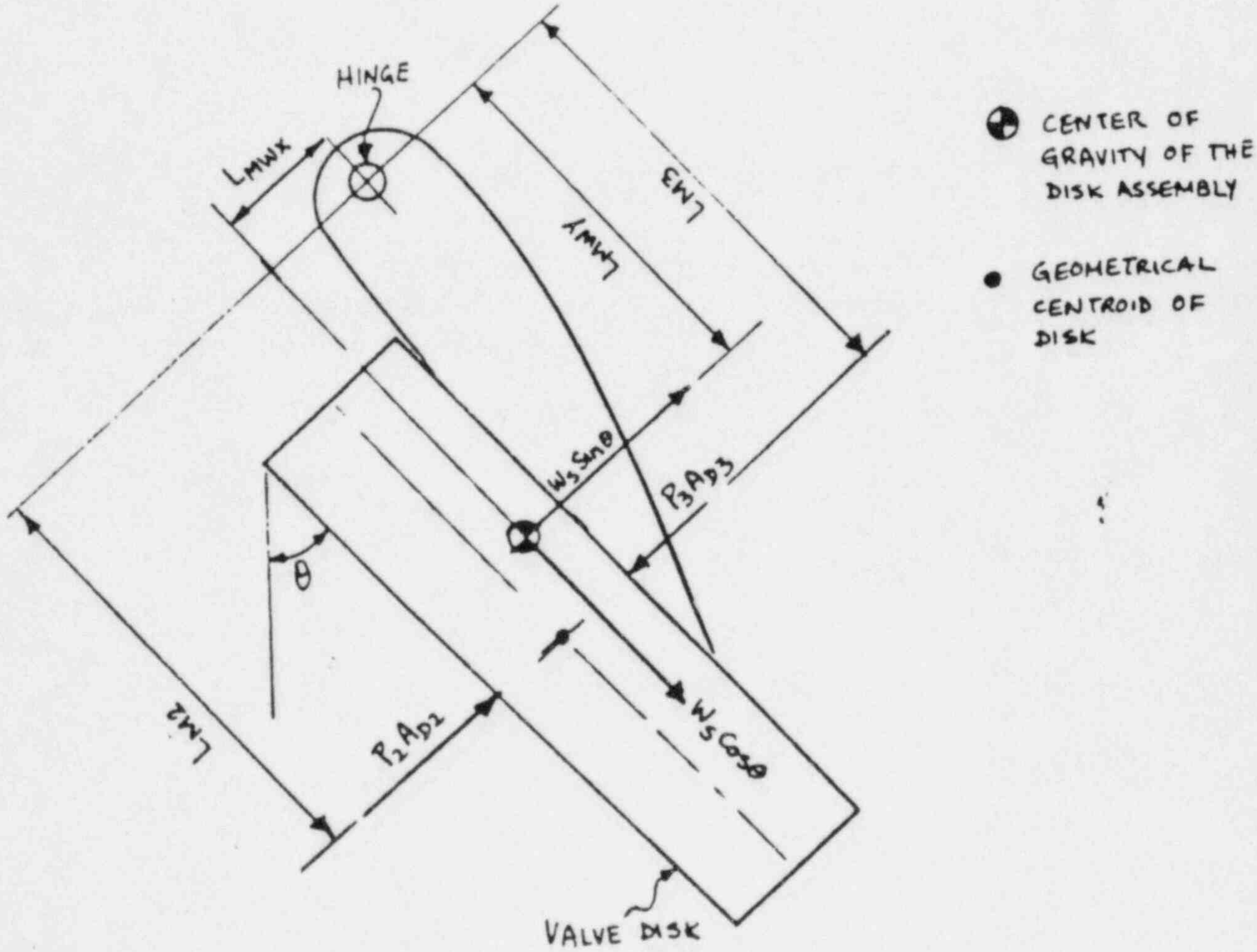

FIGURE 1b

$\ddot{\theta}$ = angular acceleration of the disk.

Using equation (1) and allowing for a fluid damping term that is proportional to the square of the disk angular velocity, the valve disk motion governing equation becomes:

$$\Sigma T = I\ddot{\theta} + \beta\dot{\theta}|\dot{\theta}| \quad (2)$$

where β = damping coefficient

$\dot{\theta}$ = disk angular velocity.

Note that this damping term accounts for the retardation of the disk motion as it moves through the water.

Assuming that $\dot{\theta}$ does not change significantly during any time step (Δt), equation (2) can be written as

$$\ddot{\theta}(t) = \frac{\Sigma T - \beta\dot{\theta}(t - \Delta t)|\dot{\theta}(t - \Delta t)|}{I} \quad (3)$$

where t = current computational time

Δt = computational time step.

Integrating (3) yields the angular velocity

$$\dot{\theta}(t) = \int \ddot{\theta}(t)dt \quad (4)$$

Integration of (4) yields the valve disk angle

$$\theta(t) = \int\dot{\theta}(t)dt \quad (5)$$

The sum of the torques T acting on the valve disk is defined as

$$\Sigma T = T_p + T_w + T_f + T_s \quad (6)$$

where T_p = torque due to pressure differential across valve disk

T_w = torque due to weight of the valve disk assembly

T_f = torque due to hinge friction

T_s = torque due to any spring that assists or hinders valve closure.

These torques are given by

$$T_p(t) = P_2(t) \cdot A_{D2} \cdot L_{M2} - P_3(t) A_{D3} L_{M3} \tag{7}$$

$$T_w(t) = -W_s L_{Mwy} \text{Sin } (\theta(t - \Delta t)) + W_s L_{Mwx} \text{Cos } (\theta(t - \Delta t)) \tag{8}$$

$T_f(t)$ = Provided by user (176.33 ft-lbs; Reference 4).

$T_s(t)$ = Provided by user (= 0, see analysis assumptions).

Where (see Figure 1b)

A_{D2} = disk surface area on Region 2 side of disk

A_{D3} = disk surface area on Region 3 side of disk

L_{M2} = moment arm for A_{D2} (disk center to hinge)

L_{M3} = moment arm for A_{D3} (disk center to hinge)

W_s = submerged weight of valve disk

L_{Mwx} = moment arm for $W_s\text{Cos}\theta$ portion of weight

L_{Mwy} = moment arm for $W_s\text{Sin}\theta$ portion of weight

In equation (7) the following approximations are utilized to obtain the pressure differential.

$$P_1(t) = P_2(t) \tag{9}$$

and $P_3(t) = P_4(t)$ (10)

3.2 Results

The RELAP5 analysis for feedwater check valve closure following pipe break outside containment was done for several break locations (2 feet to 70 feet from the check valve, though a no break zone extends upstream of the check valve for a distance of about 25 feet) and using check valve data from Reference 4. It was found that the maximum check valve disk angular speed at closure occurs with the break about 25 feet from the valve. Using conservative check valve loss coefficient assumptions, this angular speed is calculated to be about 65 rad/sec. The valve closure occurred at about 80 msec after break initiation. The peak pressure at the closed valve has been calculated to be 2800 psia.

4. Valve Stress Analysis

The structural aspects of containment isolation valves 41-1F074 A/B, outboard, and 41-1F010 A/B, inboard, were compared, and it was determined that the outboard check valve is structurally weaker than the inboard check valve. Therefore, only the outboard check valve was analyzed to verify that it could withstand the worst-case disk closing velocity that it would experience.

For evaluation of the structural integrity of the 24 inch feedwater swing check valve (41-1F074A/B), Reference Drawing 21357-H, the first step is to determine the kinetic energy of impact of the valve disk onto the valve seat using the valve velocity from the hydraulic transient analysis. Next, the plastic work capacity of the valve is computed. Then the stresses in the valve hinge arms are calculated and compared with allowable stresses. Finally, if the plastic work capacity of the valve is greater than the kinetic energy of impact, and the stresses in the valve hinge pins are less than allowables (50% of the yield strength), it can be concluded that the valve will maintain structural integrity during a postulated pipe rupture event.

4.1 Assumptions

The following assumptions are made for this analysis

a. The kinetic energy of the disk is transformed into the following manner upon impact:

   o plastic work of deformation in the localized region near the interfaces of the valve disk and the seat.

b. No credit is taken for the following energy absorbing mechanisms

   o Hinge deformation

   o valve body deformation.

c. No credit is taken for the energy losses due to flow resistance and interface friction on impact.

4.2 Method of Analysis

Stresses and deformation in the critical sections of the swing check valve (see Figure 2) were computed using simplified but conservative upper bound approaches. The evaluation of the swing check valve is divided into two parts as follows:

i. Evaluation of the stresses in the hinge pin before impact for assurance of the valve closure.

ii. impact-related plastic deformations of the valve disk/body ring interface for assurance of structural integrity.

4.2.1 Hinge Pin Stresses Analysis

The stress in the hinge pin before impact must be evaluated in order to ensure valve closure. The maximum centrifugal force (Fc) on the hinge pin occurs just before impact. This is given by:

$$F_c = m\omega^2 r = 56507 \text{ lb}$$

where m = mass of disk

$$= \frac{w}{g} = \frac{444}{32.2 \times 12} = 1.149 \text{ lb-sec}^2\text{/in}$$

w = weight of disk

= 444 lb (Reference 4)

ω = angular velocity of the disk

= 65 rad/sec

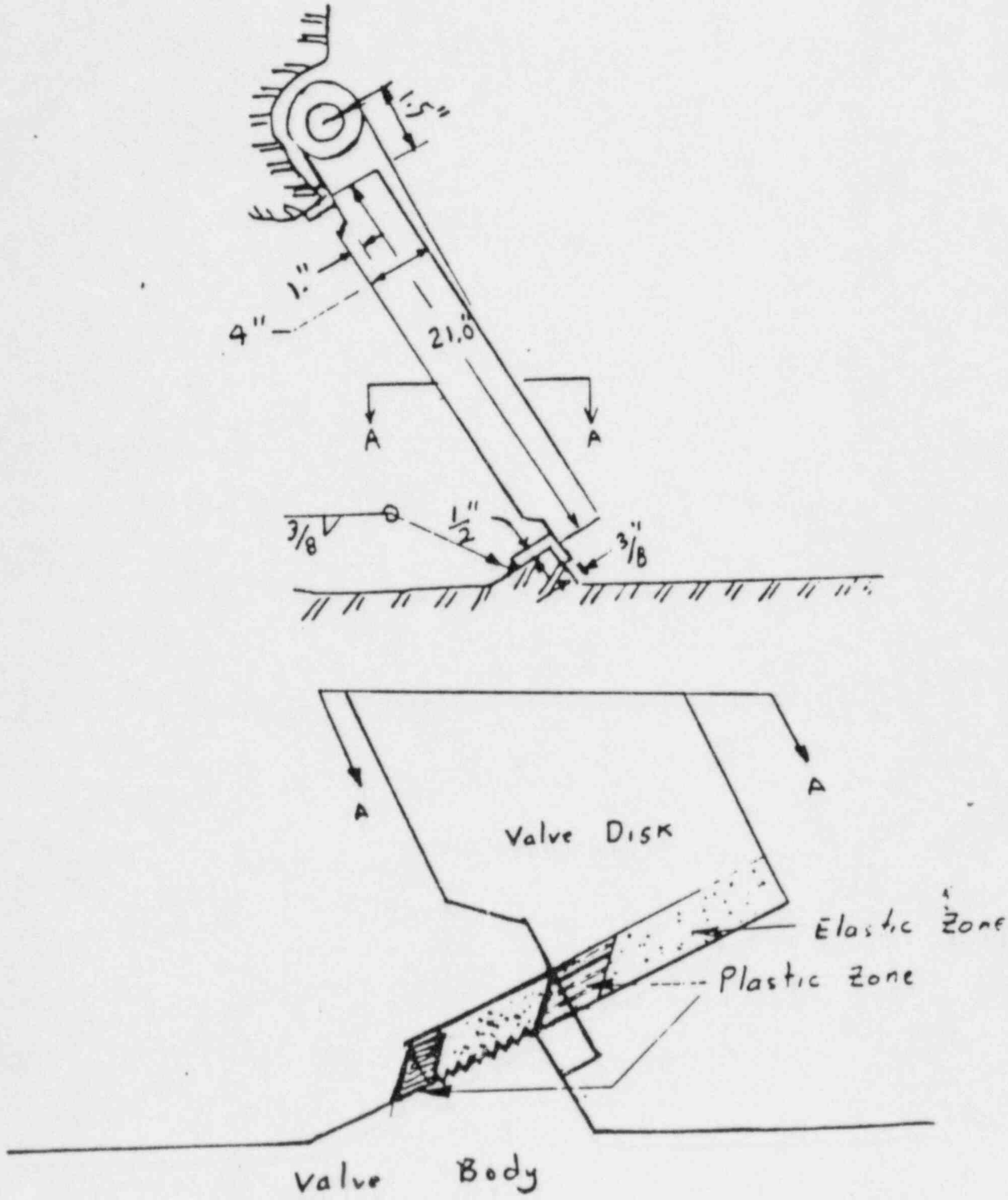
1.5"
1"
4"
21.0"
A
A
3/8
1/2"
3/8"
A
A
Valve Disk
Elastic Zone
Plastic Zone
Valve Body

Figure 2

r = distance of c.g. of the disk from c.g. of the hinge pin

= 11.64"

The force F_c will primarily produce shear stress in the pin, which is given by:

$$= \frac{F_c/2}{\text{Area of the pin}} = \frac{56507/2}{x\ 1.75^2/4}.$$

= 11.75 ksi

The allowable stress of the pin is 50% of the yield strength, Sy (Reference 5). The pin material is A 276 type 410 steel which is equivalent in strength to SA 479 type 410 steel. The yield strength at 500°F of SA 479 type 410 steel is 34.7 ksi (Reference 6). Therefore the allowable stress is 0.5 x 34.7 = 17.35 ksi which exceeds the calculated stress of 11.75 ksi. The pin is acceptable.

The design of this valve is such that the hinge pin stress is relieved on impact. Impact loads are primarily carried by the seat and disc on impact. Thus the stresses in the pin will be negligible on impact and are bounded by the stresses before impact.

4.2.2 Plastic Deformation Analysis for the Valve Seat/Disc Interface

The amount of impact energy transferred to the disc, valve weir body ring and hinge pin is dependent on the stiffness of each of these components. The stiffness of the disc is very high and as can be seen from Figure 2, the local plastic deformation will occur primarily by shear only and the bending deformations are assumed to be negligible. The plastic work capacity is computed and is shown to be higher than the kinetic energy of the disc as illustrated below:

Plastic work of deformation to rupture = Wp

$= U_R\, V = \frac{1}{3} e_u\, (Sy + (2 \times S_u)) \times V$ (Reference 7)

where U_R = ultimate resilience, is measured by the area of the stress deformation diagram for rupture.

e_u = the total strain at rupture

Sy = minimum yield stress

S_u = minimum tensile strength

V = Volume of plastic zone

Plastic Work in Body Ring $(W_p)_B$

Body Ring Material: A182 Gr F304

$e_u = 30\%$ (Reference 8)

$(Sy)\ 500°F = 19.4$ ksi (Reference 6)

$(Su)\ 500°F = 63.5$ ksi (Reference 6)

$$V = \pi \times (10.5^2 - 10.125^2) \times \frac{3}{8} + \frac{1}{2} \times \pi \times (10.5^2 - 10.0^2) \times \frac{3}{8}$$
$$= 15.15 \text{ Cu in}$$

Plastic Work in Body Ring = $(W_p)_B$ =

$$= \frac{1}{3} \times \frac{30}{100} \times (19.4 + 2 \times 63.5) \times 15.15 = 222 \text{ kip-in}$$

$$= 2.22 \times 10^5 \text{ lb-in}$$

Plastic Work in Disc $(W_p)_D$

Disc Material: SA 352 LCB

$e_u = 24\%$ (Reference 9)

$(S_y)\ 500°F = 28.3$ ksi (Reference 6)

$(S_u)\ 500°F = 65$ ksi (Reference 6)

$V = 15.15$ cu-in

Plastic work in disc = $(W_p)_D$ =

$$= \frac{1}{3} \times \frac{24}{100} (28.3 + 2 \times 65) \times 15.15 = 191.9 \text{ kip/in}$$

$$= 1.91 \times 10^5 \text{ lb-in}$$

Total plastic work = $W_p = (W_p)_B + (W_p)_D$ =

$$= 2.22 \times 10^5 + 1.91 \times 10^5 = 4.13 \times 10^5 \text{ lb-in} \qquad (11)$$

The kinetic energy, E, of the valve disk on impact for an angular velocity of 65 rad/sec is given by:

$$E = \frac{1}{2} \times I_m \times \omega^2 = \quad \times 187.3 \times 65^2 = 3.96 \times 10^5 \text{ lb-in} \qquad (12)$$

where I_m = mass moment of inertia of the disk

$$= \frac{1}{4} m\, r_o^2 + mr^2$$

$$= \frac{1}{4}\, 1.149 \times 10.5^2 + 1.149 \times 11.64^2$$

$$= 187.3 \text{ lb-sec}^2\text{-in}$$

From equations (11) and (12) it is evident that the kinetic energy of impact is less than the work capacity of plastic deformation. Therefore, it can be concluded that the valve integrity is maintained during and after the impact.

4.3 Results

The swing check valve 41-1F074 was analyzed for failure of the hinge pin and integrity of the valve body ring/disc boundary. The disc was subjected to a maximum angular velocity of 65 rad/sec giving a kinetic energy on impact of approximately 3.96×10^5 in-lb. The maximum stress in the hinge pin was calculated as 11.75 ksi. The calculated maximum impact energy that can be absorbed by the valve body ring/disc region before gross failure can occur was 4.13×10^5 in-lb. Hence, the integrity of the valve is assured.

5. Pipe Pressure Surge

The maximum pressure calculated from the RELAP5 run is about 2800 psia next to the check valve following a pipe break and subsequent check valve closure. The check valve design pressure is 2132 psig. The check valve stresses will stay within ASME Section III code allowables with a pressure of 2800 psia under faulted conditions. 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 2800 psia for a statically applied internal pressure loading within the constraints of pipe rupture loading conditions.

6. Conclusions

An 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. The analysis modeled the outboard check valve closest to the containment wall. The disk maximum angular velocity at the nearly closed position of the disk is calculated to be about 65 rad/sec using the RELAP5 code. The pressure surge for a finite valve disk closing time of 80 msec, which was predicted by the analysis, was estimated to be approximately 2800 psia which is within the pressure capability of the check valve and feedwater piping. The conclusion was reached that containment isolation is achieved during this hypothetical pipe rupture event.

7. References

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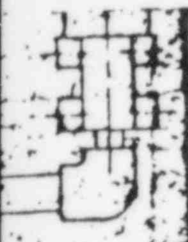

404	HEX HD CAP SCREW	STEEL	ABOT
406	HEX NUT	STEEL	[illegible]
406	HEX SOC [illegible]	ALLOY STEEL	
212 307	WASHER	STEEL	
711	LEVER	CAST STEEL	A216 WCB
712	KEY	STEEL	AISI 4140
401	MOUNTING PLATE	STEEL	[illegible]
202 401	SWITCH PLATE	STEEL	[illegible]
407	LOCK WASHER	STEEL	
407	SWITCH BRKT SPACER	STEEL TUBING	[illegible]
[illegible]	SET SCREW	ALLOY STEEL	

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LIMERICK GENERATING STATION UNIT 1 & 2

PHILADELPHIA ELECTRIC COMPANY

2F074B
1F074B
1F074A
2F074A

REVISION NO.

1962

CYLINDER LOCATED ON RIGHT HAND SIDE WHEN FACING INLET

MK 24x20x24-DLA-CK-E-V-AD-1F-

1F074A
1F074B
2F074A
2F074B

WEIGHT-5335

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24" FEEDWATER CHK

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