SARGENT & LUNDY
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CHICAGO

EVALUATION OF FEEDWATER CHECK VALVES DUE TO POSTULATED PIPE RUPTURE

FOR

SUSQUEHANNA STEAM ELECTRIC STATION

UNITS 1 & 2

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TABLE OF CONTENTS

Evaluation of Susquehanna Steam Electric Station Feedwater Check Valves Due to Postulated Pipe Rupture

1.0 INTRODUCTION

The purpose of this report is to present the results of an analytical evaluation of the structural integrity of the three feedwater check valves subjected to impact loads resulting from the postulated rupture of the feedwater piping. The report concludes that the check valves are capable of withstanding impact loads due to postulated pipe rupture.

The three check valves evaluated are:

1. 24-inch, Anchor/Darling, 900 lb. tilting disk check valve (141F010A/B)

2. 24-inch, Anchor/Darling, 900 lb. air actuated swing check valve (HV14107A/B)

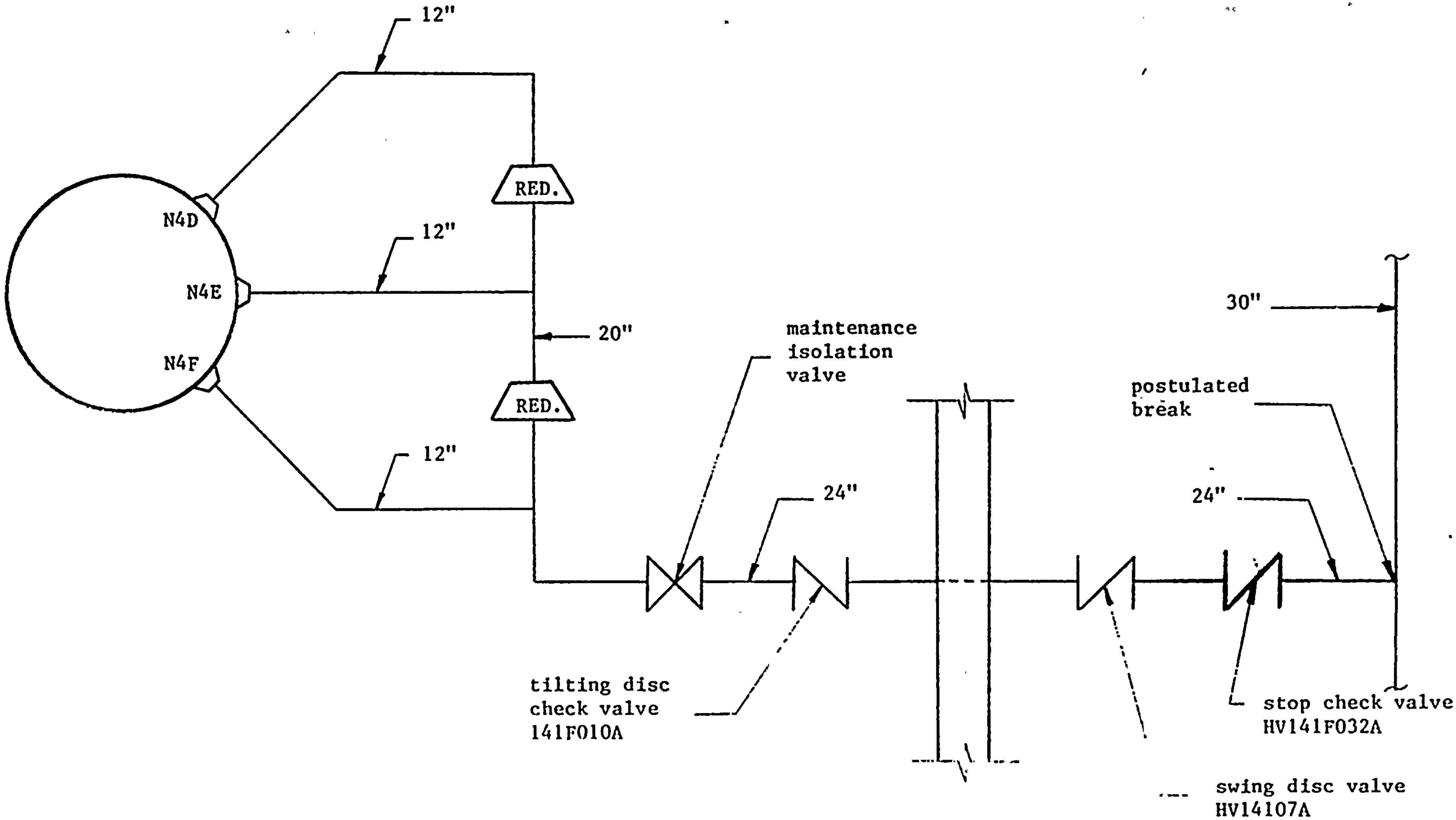
3. 24-inch, Atwood/Morrill, 900 lb. motor operated stop check valve (HV141F032A/B)

The tilting disk check valve and motor operated stop check valves serve as the primary containment isolation valves per General Design Criteria 55 (GDC 55) for the feedwater lines penetrating primary containment. The motor operated stop check valve also prevents reverse flow into the feedwater system from the RCIC, HPCI and RWCU lines connecting to the feedwater system between the swing check valve and the stop check valve. Figure 1 is a simplified schematic diagram showing the relationship of the three feedwater check valves to the reactor pressure vessel and the primary containment.

Manufacturer's seismic reports demonstrate the acceptability of the feedwater check valves for design loads other than postulated pipe rupture. The manufacturer bases acceptance upon a worst case struc-

FIGURE 1

LOOP A FEEDWATER PIPING SCHEMATIC



tural analysis considering faulted plant conditions. Maximum stresses of the pressure retaining components satisfied the allowable stress limits prescribed for normal plant conditions at design temperatures. Loads for the faulted plant condition included weight, seismic and operating loads.

The first step in the evaluation of the feedwater check valves for postulated pipe rupture loads is to perform a hydraulic transient analysis of the affected portions of the feedwater piping. This analysis predicts the behavior of each valve as a function of time during the pipe rupture transient. This analysis provides valve closure velocities and closure times which form the input for the valve structural integrity evaluation.

The next step in the evaluation is to determine the kinetic energy of impact of the valve disk onto the valve seat using the valve closure velocities from the hydraulic transient analysis. Next, the plastic work capacity of the valve is computed. Then the stresses in the valve hinge pins and 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 and arms are less than allowables it can be concluded that the valve will maintain structural integrity during a postulated pipe rupture event.

2.0 HYDRAULIC TRANSIENT ANALYSIS

This section describes the hydraulic transient analysis performed to estimate the final closure velocities of the three check valves resulting from a postulated feedwater pipe line break.

2.1 Assumptions

The following assumptions are made for the hydraulic transient analysis:

1. Constant wave velocity

2. Single phase flow

3. Constant fluid density and temperature during the postulated event.

2.2 Analysis Methodology

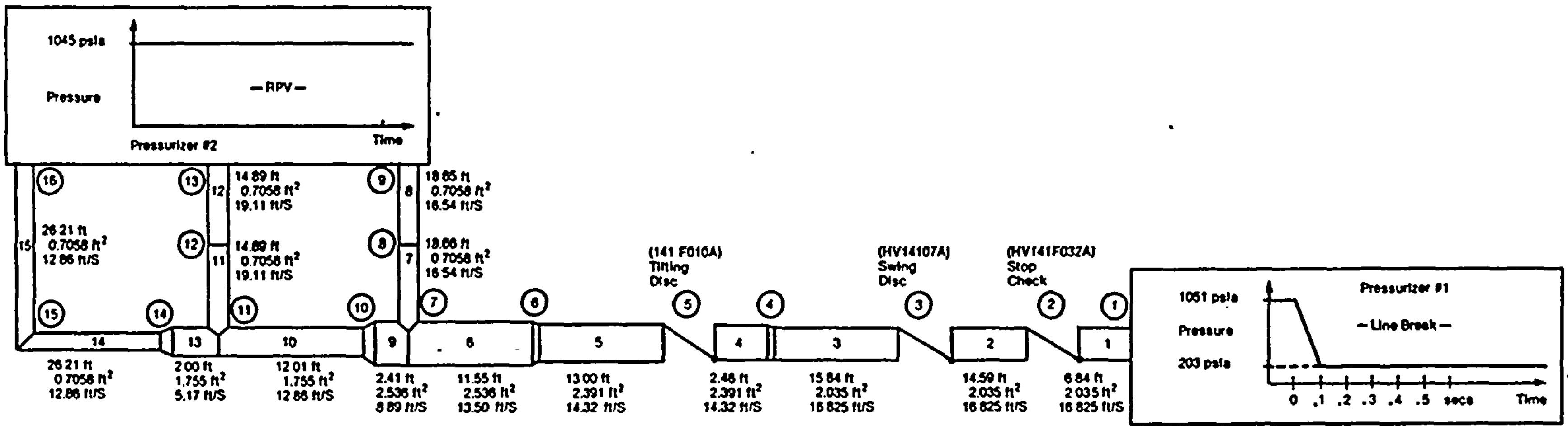
The S&L in-house computer program HYTRAN (Ref. 1) is used to determine the three check valves' behavior during the pipe rupture hydraulic transient. The HYTRAN program, validated in accordance with S&L QA requirements, uses the standard Method of Characteristics solution to fluid transient problems, requiring an assumption of constant wave velocity and fluid density. The HYTRAN modeling of the check valves and of the feedwater line from the postulated break location to the RPV is addressed in detail in Reference 2, and summarized below.

The HYTRAN model (Figure 2) includes all significant (>6" dia.) loop A feedwater piping from the postulated break location, at the junction of lines 24" DBD-101 and 30" DBD-101, through the stop, swing and tilting disk check valves to the three RPV nozzles (because of the mirror symmetry of feedwater loops A and B, only the former is analyzed). All relevant flow discontinuities are included in the model. Initial flow velocities, temperatures and pressures are determined using rated feedwater operating characteristics (Reference 13). Check valve parameters are determined from vendor information (References 5 through 8) and PPL letters PLE-2457 (10/29/82) and PLE-2465 (11/1/82).

The pipe rupture is modeled as a linear pressure drop from rated feedwater operating pressure to saturation pressure. Two-phase flow is assumed to exist at the pipe rupture location, the second phase being steam at saturation pressure. As a result, the pressure at the rupture location will be maintained at saturation pressure during the transient.

A pipe rupture linear pressure drop duration of 100 milliseconds is used which is considered a conservative break representation based on NUREG/CR-1319 (Ref. 3).

Figure 2
Hytran Model



Water Temperature: 383°F
Water Density: 54.6 lb/ft³
Wave Velocity: 3950 ft/sec

NUREG/CR-1319 assesses the margin of safety against a large break in the primary coolant loop piping of a pressurized water reactor. Typical pipe sizes considered in NUREG/CR-1319 are 32" diameter, 2.5" wall thickness, experiencing temperatures and pressures exceeding 500°F and 2000 psig, respectively; correspondingly, the Susquehanna feedwater piping at the break location has a 24" diameter and a 2.343" wall thickness, operating temperatures and pressures of 383°F and 1045 psig, respectively. In NUREG/CR-1319, both axial and circumferential cracks are postulated at critical areas within the cold leg piping system. It is determined that the expected minimum time between a detectable leak and the development of a large break is on the order of a few hours.

The piping system characteristics and operating conditions reviewed in NUREG/CR-1319 bound those of the Susquehanna feedwater piping system. Therefore, it can be concluded that NUREG/CR-1319 applies to the feedwater piping. The postulation of full guillotine feedwater pipe break in 100 milliseconds is thus conservative.

HYTRAN output provides the check valve disk angular velocities at one millisecond time intervals, along with fluid pressures and velocities at various points in the system. The results of the analysis are detailed in the following section.

2.3 Results

The following table provides the disk angular velocities at the time of valve closure. It also provides the time from pipe rupture initiation to valve closure for each valve.

Check Valve Type	Valve Number	Maximum Closure Velocity (rad/sec)	Time of Closure (msec)
Tilting Disk	141F010A	36.5	297
Swing	HV14107A	33.1	189
Stop	HV141F032A	41.0	178

The pipe break analysis predicts that the stop check valve is the first valve to close. Since insignificant pressure is acting on the break side of the stop check valve disk after it closes, reactor pressure will hold the stop check valve closed for the remainder of the transient.

The analysis predicts the swing check valve closes after the stop valve closure. Because of the pressure surge created by the rapid closure of the stop check valve, subsequent reopening of the swing check valve is possible. However, with the closure of the stop check valve there can no longer be any significant net reverse flow through the swing check valve. This is also true of the tilting disk check valve, which at the time of the initial swing check closure is not yet closed. Therefore, the tilting disk check valve and the swing check valve can experience only a rapidly attenuated flow oscillation through the remainder of their closure strokes. This is unlike the steadily increasing reverse flow velocities (and corresponding disk - accelerating forces) experienced by the valves during most of their initial closure strokes.

The hydraulic transient analysis predicts a pressure surge to 4694 psia between the tilting disk and the stop check valves.

3.0 Valve Integrity Evaluation

3.1 Assumptions

The following assumptions are made for this analysis:

1. The kinetic energy of the disk is transformed into the following components on impact:

   a. Plastic work of deformation in the localized region near the interface of the valve disk and the seat.

   b. Strain energy in the disk, valve body and hinge pin.

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

3.2 Method of Analysis

Stresses and deformation in the critical sections of the three valves were computed using simplified but conservative upper-bound approaches.

3.2.1 Stop Check Valve (Fig. 3)

The evaluation of the stop check valve is divided into two parts as follows:

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

2. Evaluation of the impact-related plastic deformations of the valve weir/disk interface for assurance of structural integrity.

Hinge Pin Analysis

The maximum centrifugal force (F_c) on the hinge pin occurs just before impact. This is given by:

$F_c = m\omega^2 r = 22{,}133$ lbs., where [Ref. 10]

m = mass of the disk

ω = angular velocity of the disk = 41 rad/sec

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

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

$$= \frac{F_c/2}{\text{Area of the pin}} = 6.14 \text{ ksi} \qquad \text{[Ref. 10]}$$

The allowable shear stress for the pin is 40% of the yield strength of 36 ksi, or 14.4 ksi [Reference 12]. Since the allowable stress exceeds the calculated stress, the hinge 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 since the disk wedges-in on impact. Thus the stresses in the pin will be negligible on impact, and are bounded by the stresses before impact.

Plastic Deformations Analysis for the Valve Seat/ Disk Interface

The amount of impact energy transferred to the disk, valve weir and hinge pin is dependent on the stiffness of each of these components. The stiffness of the disk is very high and as seen from Figure 3, the local plastic deformations will occur primarily by shear only. 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 disk as illustrated below.

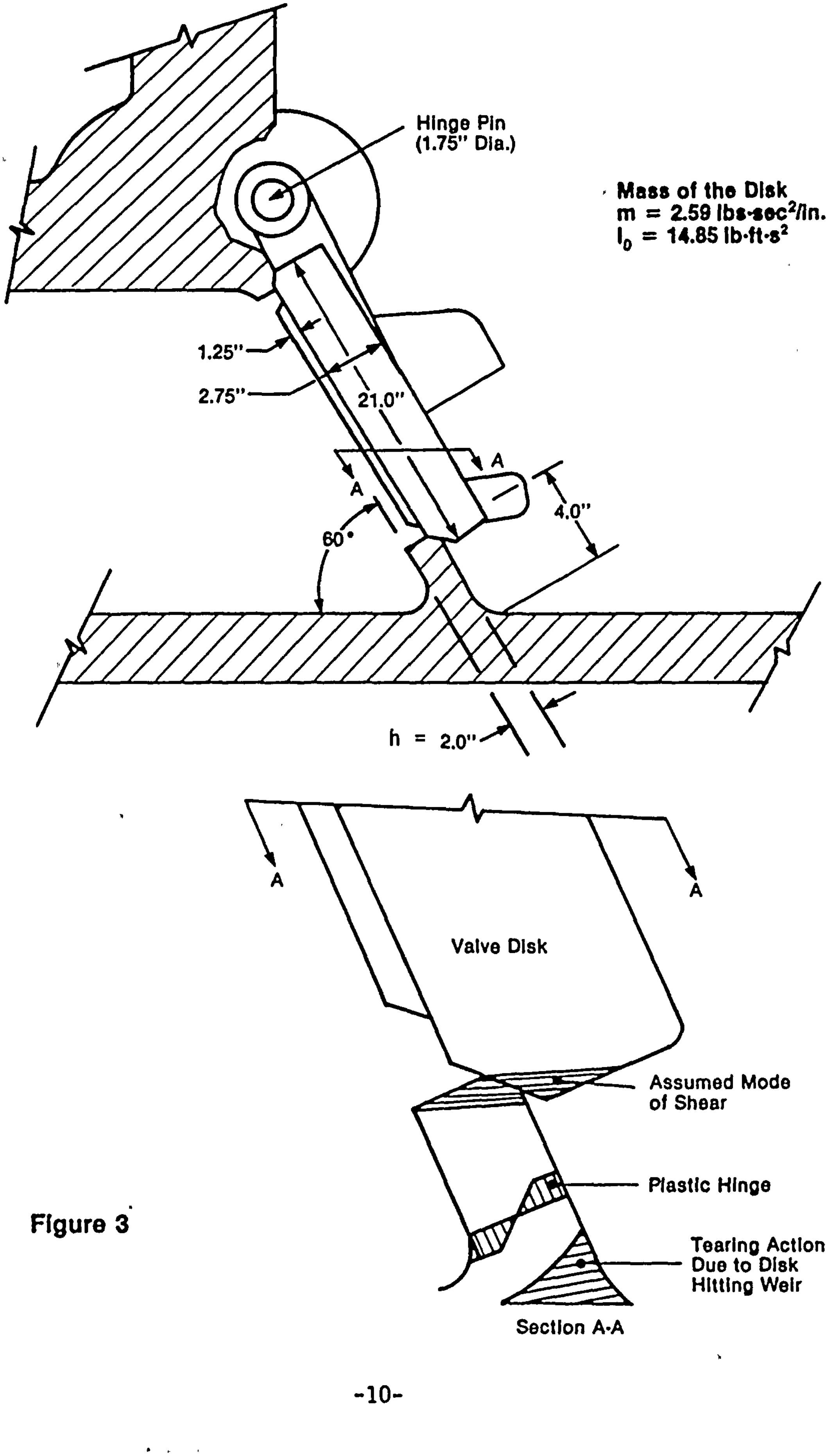

Hinge Pin
(1.75" Dia.)
Mass of the Disk
m = 2.59 lbs-sec²/in.
I₀ = 14.85 lb-ft-s²
1.25"
2.75"
21.0"
A
A
4.0"
60°
h = 2.0"
A
A
Valve Disk
Assumed Mode
of Shear
Plastic Hinge
Tearing Action
Due to Disk
Hitting Weir
Section A-A

Figure 3

Area of the seat at intersection with the valve disk (Figure 3) = $(2\pi)(r)(h) = 132 \text{ in}^2$ - (1)

Plastic work of deformation =

$= Wp = (\tau_y)\ A\ (2/3)(h)$ - (2) [Ref. 10]

τ_y = yield strength in shear ≈ 2/3 x yield strength in tension. (Ref. 12)

The yield strength in tension is 30,300 psi per ASME Section III, Appendices Table I-2.1, P. 44, 1980 Edition for SA-216 WCB material at 383°F.

Hence $\tau_y = 20{,}533$ psi - (3)

From (1), (2) and (3), $W_p = 3.56 \times 10^6$ in-lb. - (4)

The kinetic energy (E) of the valve disk on impact, for an angular velocity ω of 41 rad/sec is given by:

$E = \frac{1}{2}\ I_m \times \omega^2 = 1.57 \times 10^5$ in-lb. - (5) [Ref. 10]

where I_m = Mass moment of inertia of the disk.

From equations (4) and (5) 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.

3.2.2 Swing Check Valve

The evaluation of the swing check valve is divided into three parts as follows:

1. evaluation of the stresses in the hinge arm and hinge pin before impact
2. evaluation of the stresses in the hinge arm and hinge pin during and after impact

3. assessment of impact-related plastic deformations in the seat disk interface area by comparison with similar valve analysis.

Stresses Before Impact

The stresses in hinge arm and hinge pin before impact must be evaluated in order to ensure valve closure. The bending and axial stresses in the hinge arm and shear stress in the hinge pin are calculated as follows:

The centrifugal force, F_c, developed for an angular velocity of 33 rad/sec and a distance from the axis of rotation to the center of mass of 14.85" is (Fig. 4).

$$F_c = m\omega^2 r = 8.74 \times 10^3 \text{ lbs} \qquad \text{[Ref. 10]}$$

The moment M_e, due to eccentricity e, is

$$e = 2.74''$$

$$M_e = F_c e = 2.4 \times 10^4 \text{ in-lb} \qquad \text{[Ref. 10]}$$

The maximum bending stress in the hinge arm is

I = moment of inertia

Z = distance from the neutral axis to the outer fiber

$$\sigma_b = \frac{M_e Z}{I} = 25.67 \text{ ksi} \qquad \text{[Ref. 10]}$$

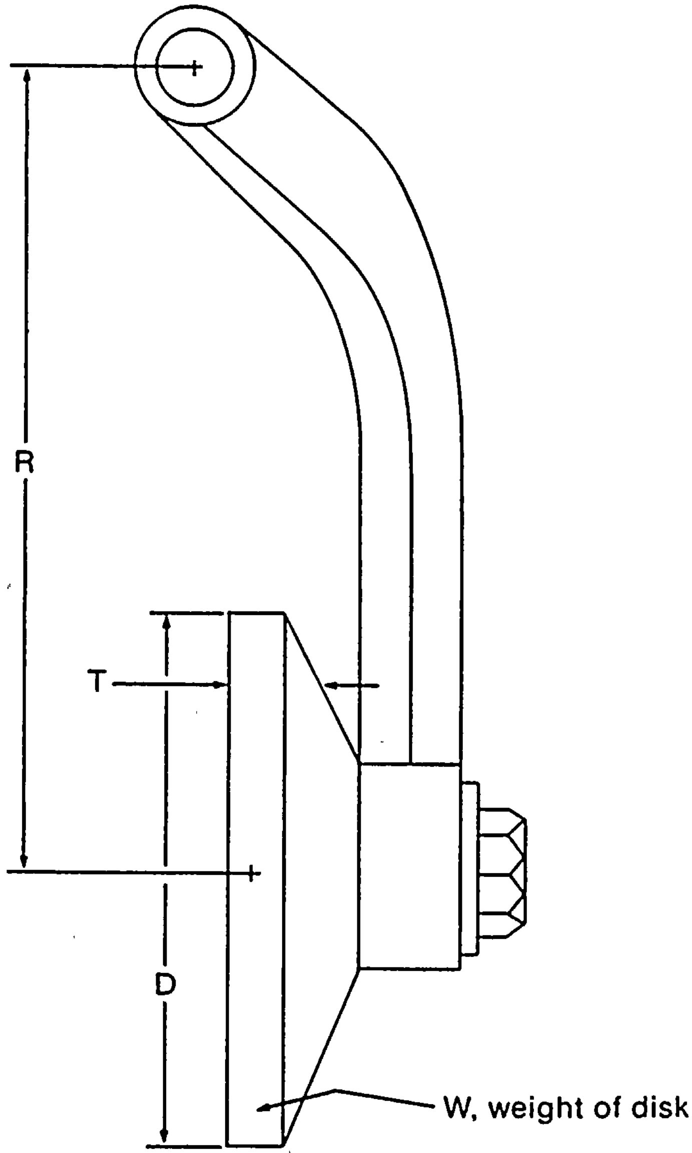
The maximum axial stress in the hinge arm is:

A = cross sectional area = 4.99 in^2 [Ref. 5]

$$\sigma_t = \frac{F_c}{A} = 1.75 \text{ ksi} \qquad \text{[Ref. 10)}$$

The combined stress is obtained by superimposing the bending and the axial stress components:

$$\sigma_T = \sigma_b + \sigma_t = 27.42 \text{ ksi}$$



Parameters	Susquehanna Valve	Reference Valve
D (in)	13.88	27.25
T(in)	3.16	3.5
R in)	14.85	20
W (lb)	209	468.4

Figure 4

The allowable stress in the hinge arm is 90% of the yield strength or 61.9 ksi [Ref. 12]. (The yield strength of the hinge arm and pin is 68.8 ksi per Reference 5). Since the allowable stress exceeds the calculated stress, the hinge arm is acceptable.

The maximum shear stress in the hinge pin is:

A_{pin} = cross sectional area of the pin = 3.2 in^2
(Since the pin is in double shear, total area = 2A)

$$\tau = \frac{F_c}{2A_{pin}} = 1.37 \text{ ksi} \qquad \text{[Ref. 10]}$$

The allowable stress in the hingepin is 40% of the yield strength for shear stress (Ref. 12),or 27.52 ksi. Since the allowable stress exceeds the calculated stress the hinge pin is acceptable.

Stresses After Impact

Stresses in the hinge arm and hinge pin are computed using the conservative assumption that the entire kinetic energy is absorbed by the disk in the form of strain energy. The deflection at the center of the disk due to this absorbed strain energy is computed. Using this deflection the stresses are computed in the hinge arm and hinge pin, the hinge arm is assumed to be fixed at the hinge pin beyond the five degrees closure angle. The following equations were used in performing the calculations:

The strain energy (U_s) in the disk is given by

$$U_s \approx \frac{W_{max}}{2} A\, q \text{ in-lb} \qquad \text{[Ref. 9]} \qquad - (1)$$

where

A = Area of the disk

q = equivalent uniform pressure corresponding to deformation of the disk after impact.

W_{max} = maximum transverse deflection of the disk

Also, W_{max} is given by

$$W_{max} = \frac{q\ a^4}{64D} \ \frac{5+\nu}{1+\nu} \quad \text{[Ref. 9, p. 363]} \qquad - (2)$$

where

D = Plate constant = $\dfrac{Et^3}{12(1-\nu^2)}$

E = Modulus of elasticity

ν = Poisson's ratio

t = thickness of the plate

a = radius of the disk

From (1) and (2)

$$U_s = 1.29 \times 10^{-4}\ q^2 \qquad - (3)$$

The kinetic energy of the disk is given by

$$K.E. = \tfrac{1}{2}mv^2 = 6.49 \times 10^4 \text{ lb-in [Ref. 10]} \qquad - (4)$$

where

m = mass of the disk

v = velocity of the c.g. of the disk

Equating the strain energy U_s to the K.E., from (3) and (4) we get q and substituting q in (2) we get the maximum displacement as

$$W_{max} = .0384"$$

Based on this deflection (W_{max}), the equivalent load (P) on the hinge arm is computed using the formula

$$P = \frac{3EI_s W_{max}}{R^3}$$

$$P = 1.28 \times 10^3 \text{ lbs}$$

(Note: I_s = Section Moment of Inertia of the hinge arm)

The corresponding maximum bending stress in the hinge arm is $\sigma_{bending}$ = 20.36 ksi. This stress is less than the allowable stress of 61.9 ksi.

The shear stress (combined torsion & direct shear) in the hinge pin due to the above load of 1.28×10^3 lbs is computed to be τ_{max} = 12.17 ksi which is less than the allowable value of 27.52 ksi.

Hence, both the hinge pin and the hinge arm maintain their structural integrity after the impact.

The assessment of impact-related damage can be made from the results of a stress analysis performed for the disk and seat region of a 30" main steam check valve installed at a nuclear power station. The design of this swing check valve is similar to the Susquehanna swing check valve [Fig. 4]. This reference valve is larger in size and is heavier and hence can be considered as bounding case for the Susquehanna valve analysis.

A detailed elasto-plastic finite difference analysis [Ref. 4] of this valve indicated that the structural integrity of the valve was maintained for 68 rad/sec closure velocity. Plastic deformations of the valve/seat/disk interface were small and localized. Valve integrity and operability were not affected. Based on this, it can be concluded that the local plastic deformations for the Susquehanna swing check valve due to impact will be very small and the operability and the structural integrity of the valve will not be impaired.

3.2.3 Tilting Disk Check Valve

The evaluation of the tilting disk check valve is divided into two parts as follows:

1. Evaluation of the stresses in the hinge pin before impact for assurance of valve closure.
2. Evaluation of the impact-related plastic deformations of the valve weir/disk interface for assurance of structural integrity.

Hinge Pin Analysis

The maximum centrifugal force F_c on the hinge pin occurs just prior to impact. This is given by:

$$F_c = m\omega^2 r = 17{,}500 \text{ lbs} \qquad \text{[Ref. 10]}$$

where

m = mass of the disk = 2.59 lbs sec^2/in
ω = angular velocity of the disk = 36.5 rad/sec
r = distance of the c.g. of the disk from the c.g. of the hinge pin

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

$$\tau = \frac{F_c}{2A} = 5375 \text{ psi};$$ A = Area of the pin, (2A since there are two pins - one on each side)

The allowable shear stress for the pin is 40% of the yield strength of 36 ksi or 14,400 psi. [Ref. 12] Since the allowable stress exceeds the calculated stress, the hinge 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 since the disk wedges-in on impact. Thus, the stresses in the pin will be negligible on impact, and bounded by those before impact.

<u>Plastic Deformation Analysis for the Valve Seat/Disk Interface</u>

The kinetic energy of the disk at impact is given by

$$K.E. = \frac{1}{2} I_o \omega^2 = 118{,}704 \text{ lb-in} \quad \text{[Ref. 10]} \qquad - (1)$$

where

I_o = Mass moment of inertia = 14.85 lb-ft-s^2

ω = angular velocity = 36.5 rad/sec

For simplicity and conservatism, assume a one-dimensional state of stress in the plastic zone. This state of stress is shown in the figure below.

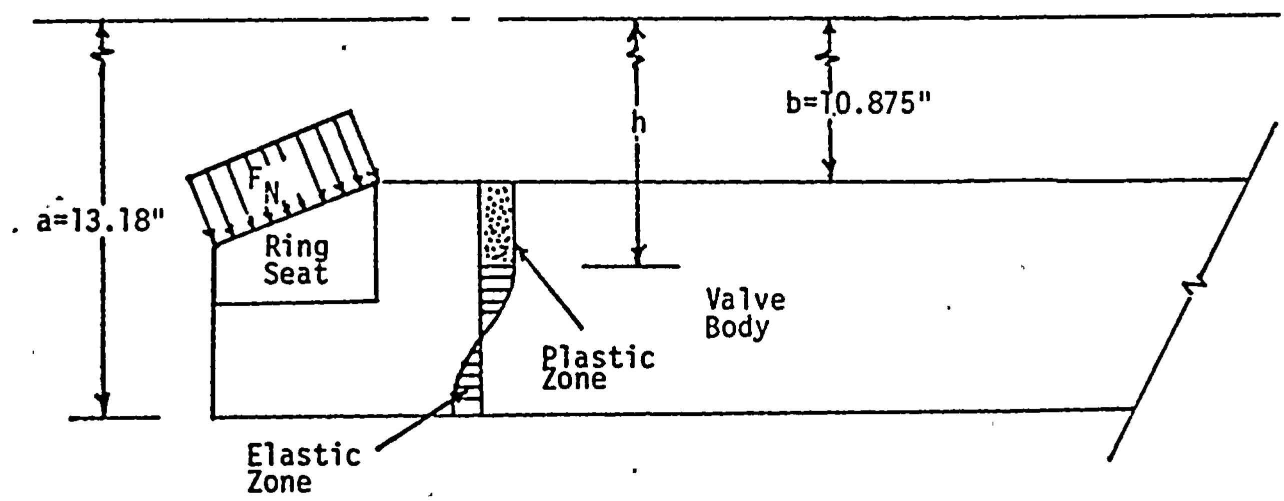


FIGURE 5

The plastic work of deformation (W_p) is given by:

$$W_p = \pi(h^2-b^2)(\ell)(\sigma_y) \qquad - (2)$$

where

b = 10.875 (see Fig. 5)

σ_y = 28300 psi [Ref. 7]

$\ell \approx .1$ = axial length of plastic zone (a conservatively assumed valve)

h is to be determined from conservatively assumed value of circumferential strain of .1

Thus,

$$.1 = \frac{2\pi h - 2\pi b}{2\pi b}$$

or

$$h = 11.9625 \text{ in} \qquad - (3)$$

From (2) and (3)

$$W_p = 220{,}808 \text{ in-lb} \qquad - (4)$$

From equations (1) and (4) 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 impact.

3.3 Results

3.3.1 Stop Check Valve

The stop check valve was analyzed for failure of the hinge pin and integrity of the valve weir/disc boundary. The disc was subjected to a maximum angular velocity of 41 rad/sec giving a kinetic energy on impact of approximately 1.57×10^5 in-lb.

The maximum stress in the hinge pin was calculated as 6.14 ksi. The calculated maximum impact energy that can be absorbed by the valve weir/disk region before gross failure can occur was 3.56×10^6 in-lb. Hence, the integrity of the valve is assured.

3.3.2 Swing Check Valve

The analysis for swing check valve indicated that the valve is acceptable for an angular velocity of 33 rad/sec. At this velocity the maximum stresses in the various valve components are [Ref. 3]:

Hinge arm = σ_{max} = 27.42 ksi

Hinge Pin = σ_{max} = 12.17 ksi

3.3.3 Tilting Disk Check Valve

The analysis for the tilting disk check valve indicated that the valve is acceptable for an angular velocity of 36.5 rad/sec. The maximum stress in the hinge pin was calculated as 5375 psi. This stress is well below the allowable value of 14,400 psi.

The maximum kinetic energy on impact was calculated to be 118,704 in-lb. This energy is small compared to the plastic work capacity of 220,808 in-lb. Hence, the integrity and operability of the valve is assured.

The following table summarizes the results for all the three valves.

SUMMARY OF THE RESULTS

Valve Type	Angular Velocity rad/sec	Valve Component Analyzed	Calculated Parameter	Calculated Value	Allowable Value	Remarks
Stop Check Valve	41	Hinge Pin	Maximum Shear Stress	6.14 ksi	14.4 ksi	Operability and structural integrity assured
		Valve Seat/ Disk Interface	Impact Energy	1.57×10^5 in-lb	3.56×10^6 in-lb	
Swing Check Valve	33	Hinge Arm	Maximum Combined Stress	27.42 ksi	61.9 ksi	"
		Hinge Pin	Maximum Shear Stress	12.17 ksi	27.52 ksi	
Tilting Disk Check Valve	36.5	Hinge Pin	Maximum Shear Stress	5.375 ksi	14.40 ksi	"
		Valve Seat/ Disk Interface	Impact Energy	118,704 in-lb	220,808 in-lb	

4.0 Pipe Pressure Integrity Evaluation

Based on hydraulic transient analysis, the maximum pressure surge calculated in the portion of pipe between the stop check valve and the tilt disk check valve was found to be 4694 psi [Ref. 2]. The allowable faulted pressure can be calculated using the following equations from ASME Section III Code paragraphs NB-3640 and F-1360.

$$\text{Allowable faulted pressure} = 2\ Pa = 2X\frac{2S_m T}{D_o - 2yt} = 5377\ \text{psi}$$

where Pa = Calculated maximum internal pressure

Sm = Maximum allowable stress intensity
= 20,000 at 400°F

t = Specified or actual wall thickness = 1.531 in.

D_o = Outer pipe diameter = 24 in.

y = 0.4

The allowable faulted pressure exceeds the surge pressure, therefore, the pressure integrity is demonstrated.

5.0 CONCLUSIONS

This report demonstrates that for a guillotine pipe rupture of the 24" Feedwater pipe upstream of the outer containment isolation valve (HV141F032A,B), the impact loads resulting from rapid valve closures will not jeopardize the structural integrity of the valves. Therefore, the valves' ability to perform their containment isolation function is assured.

The pipe rupture time of 100 milliseconds is determined to be conservative, based on the analogy made to information contained in NUREG/CR-1319. Simplified but conservative stress analyses and impact energy to plastic work capacity comparisons of the valve internals show the stresses experienced by each valve due to the impact loads following a postulated 100 milliseoncds pipe rupture are well within the allowable values for their respective materials. Therefore, it was concluded that the motor operated stop/check valve and the tilting disk check valve which serve as the containment isolation valves are capable of withstanding impact loads resulting from rapid valve closures, and containment isolation is assured. By determining analytically that for a pipe rupture time which is conservative in comparison to published values the valve internals are not overstressed, it can be concluded that the impact loads are not too great to prevent the valves from performing their intended containment isolation function in accordance with General Design Criterion 55 of 10CF50, Appendix A.

6.0 REFERENCES

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