SAFETY ENGINEERING

[illegible]

MSIV 2HV8204 RAIL/SKIRT FAILURE
ROOT CAUSE EVALUATION
RCE-91-025
NOVEMBER 5, 1991

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EXECUTIVE SUMMARY

EVENT DESCRIPTION

On August 25, 1991 during the Unit 2, Cycle 6 refueling outage, SCE performed a scheduled boroscopic inspection of both Unit 2 Main Steam Isolation valves (MSIV's). The inspection of MSIV 2HV8205 did not show any apparent damage, but MSIV 2HV8204 exhibited damage to the valve disk locking mechanism. As a result of the boroscopic inspection, a decision was made to disassemble and further inspect the valve gate and segment locking mechanism of valve 2HV8204.

Valve 2HV8204 was disassembled and the following damage was noted:

1) Three broken guide rail to seat skirt capscrews on the north gate side rail and one broken capscrew on the south gate side rail.

2) The gate side seat skirt was severed at the location of the fourth capscrew from the top.

3) Both gate side rails had severe galling on the top chamfers.

4) Both lever lock arm shoes had severe galling on the flat bottom surface.

ROOT CAUSE

Design Deficiency:

The materials used for the sliding surfaces of the lever lock arm shoe and the seat skirt rail ramps are a poor design choice. The lever lock arm shoes have areas of stellite hardfacing, but the most highly loaded areas have 17-4 PH to 17-4PH stainless steel sliding surfaces. 17-4 PH stainless steel is a very poor choice of material for sliding surfaces that have appreciable contact stresses. The threshold for galling of 17-4 PH on 17-4 PH surfaces is approximately 2000 pounds per square inch (psi) which is well below the contact stresses between the lever lock arm shoes and the guide rail ramps.

The material problem was compounded by a modification of the rail ramp angles that was made during the Unit 2, mid-cycle outage prior to the Cycle 5 refueling outage. The ramp angles were hand machined from 45 degrees to approximately 30 degrees from the vertical with no change to the companion angle on the lever lock arm shoe. This mismatch of angles resulted in an increase in the loads and stresses between the shoes and the ramps.

CORRECTIVE ACTIONS

Field Change Notices (FCN's) were issued to modify the Unit 2 MSIV lever lock arm shoes and the guide rail ramps during the present refueling outage. The ramp angles were revised and the shoes were reconfigured to match a 30 degree ramp angle. The contact surfaces of both the shoes and the rail ramps were overlaid with stellite 6B. The Unit 3 MSIV's will be modified to the new shoe/ramp configuration at the next refueling outage.

BACKGROUND

Unit 3 Cycle 4 MSIV Inspection

In May of 1988, during the SONGS 3, Cycle 4 Refueling Outage, Southern California Edison was informed by Louisiana Power and Light (LP&L) that the two Main Steam Isolation Valves (MSIV's) installed at the Waterford-3 Nuclear Steam Station had suffered internal damage. The MSIV's currently installed at San Onofre Nuclear Generating Station (SONGS) Units 2 and 3 are of the same type as damaged at LP&L. A boroscopic inspection of the SONGS Unit 3 MSIV's and an interim Root Cause Analysis (Reference 1) of the LP&L failure and its applicability to San Onofre MSIV's were initiated to determine the failure mechanism for the LP&L valves and to determine if the SONGS MSIV's are subject to similar failures.

The results of the boroscopic inspection of the Unit 3 MSIV's are tabulated below:

(1) One broken capscrew was found in the body cavity of MSIV 3HV8205. The two guide rails (one of the two segment skirt rails and one of the two gate skirt rails) that could be inspected were in place. The rails were found firmly attached to the skirt plates. They did not exhibit any separation that was observed on the LP&L MSIV's. Only four capscrews could be inspected in place and they were found to exhibit no signs of elongation, deformation or looseness. The results of this inspection were documented in NCR 2-2043 Rev 0.

(2) No broken capscrews were found in the body cavity of 3HV8204. The two guide rails that could be inspected (again, one of the two segment skirt rails and one of the two gate skirt rails) were also in place.

(3) The chamfers of both the upstream and downstream guide rails for both valves were inspected for impact marks. Only the top chamfers on the downstream, or gate guide rails of 3HV-8205 were thought to have galling marks. The top edge of the chamfers seemed to have been rounded by impact and some metal had been rolled over. This metal roll over and a relatively high contact stress during impact was thought to be responsible for the observed galling marks.

(4) The shoes for the lever lock arms for both valves were inspected for impact or galling marks. No visible galling marks were observed on the surfaces inspected.

The conclusion of the Reference 1 Interim Root Cause Analysis was that the San Onofre MSIV's stroke more slowly than the LP&L valves and are not subject to the guide rail dislodgment failure as experienced at Waterford-3. A dynamic analysis of the impact energies of both the Waterford-3 and SONGS, Unit 2 and Unit 3 lever lock arm assemblies was performed. The Waterford-3 valves had been operated with a stroke time much faster than the SONGS valves. When the impact energies of the two valves were compared, the Waterford-3 impact energy was an order of magnitude higher than the SONGS MSIV's. Further analysis concluded that the impact energy of the SONGS valves was capable of breaking only two capscrews while the Waterford-3 impact energy was enough to shear all nine capscrews at once. The conclusion of this study was that the SONGS rails could not fail from guide rail dislodgment due to shearing of all of the capscrews.

No immediate action was required; however, as a long term measure an inspection plan for all SONGS MSIV's was recommended. A study to evaluate revisions to the chamfer angle on the rails from 45 to 30 degrees and the addition of stellite overlays on the contact surfaces of the lever lock arm shoes and the guide rails was also recommended. The addition of an orifice plate in the common dump piping of both valves was also recommended and implemented.

At the urging of the Nuclear Regulatory Commission, MSIV 3HV8205 was disassembled and inspected prior to the end of the outage. The results of the inspection were as follows and are documented in NCR 3-2046 and NCR 3-2056:

1. Three severed capscrews and one loose capscrew were found at the bottom of one of the segment side rails.

2. The bottom surface of the segment side rail chamfers were found to be galled. No galling was observed on the gate side rail chamfers, but some wear was noted. The boroscopic observation of galling was in error.

3. The lever lock arm shoes were galled where they contact the segment side rails.

All non-conforming conditions were returned to acceptable configurations and the sheared capscrews were replaced. The upper ramps on the gate side guide rails were also reworked at this time per the disposition of NCR 3-2046. The sharp corners were removed from both ends of the ramps to a 1/4 inch nominal radius and the ramps were machined/stone ground to an angle of 45 degrees +0 -5 degrees. The four damaged capscrews were replaced with screws made of 17-4 PH material for increased strength.

Reference 2, Inspection Results of SONGS 3MSIV8205, determined that the root cause of the failure of the guide rail capscrews was a self limiting galling process that builds up and removes the sharp corners of the guide rails. Once the sharp corners are removed, the contact area between the shoes and rails increases. With the increase in area, the loading is reduced, resulting in lower stresses and discontinuation of the galling and adhesion process.

On August 9, 1988, the NRC issued Information Notice 88-59 which discussed the main steam isolation valve guide rail failure at Waterford, Unit 3. The corrective actions noted in the report included the SONGS corrective actions from Reference 1 along with additional items from Waterford-3. These recommendations included changing the shoe and rail angles and adding stellite hardfacing to the wear surfaces, verifying bolt alignments and performing nondestructive testing of all new bolts, increasing the valve stroke time and performing periodic boroscopic inspections of the valve internals.

All corrective actions, with the exception of verification of bolt alignment and additional nondestructive testing of bolts, are being taken at SONGS. Since the valve stroke time at SONGS is much slower that at Waterford, the SONGS loadings are lower and the requirements for the bolting inspections were not considered necessary.

UNIT 2 MSIV INSPECTION

MSIV 2HV8204 was disassembled and inspected on January 19, 1989 during a Unit 2, mid-cycle 4 outage. The bottom chamfers on both segment skirt guide rails were found to be slightly galled. The top surfaces of the shoes were also slightly galled. All capscrews were found to be in place. The noted conditions were documented on NCR 2-2612.

All nonconforming conditions were repaired and all of the top guide rail chamfers were machined/stone ground to 45 degrees +0/-10 degrees (approximate). The sharp corners were also rounded to a 1/4 inch nominal radius per the disposition of NCR 2-2612.

UNIT 2 CYCLE 6 MSIV INSPECTION

During the SONGS 2, Cycle 6 refueling outage an internal inspection of Main Steam Isolation Valve 2HV8204 was performed with the use of a boroscope. The inspection revealed damage to the rails and a fracture in the material of the gate side skirt. Two of the countersunk machine screws that hold the rails to the skirts were observed to be protruding from the rail and appeared to either be backed out of the threaded holes or elongated due to high loading. One of the rails was observed to have a gap of approximately 1/8 inch between what should be the contact surface of the rail and the skirt in the area of the top three capscrews. The skirt plate material in the vicinity of the third capscrew appeared to be fractured, leaving a 1/2 inch gap in the skirt. The skirt and rail below this fracture did not show any indication of looseness or damage.

MSIV 2HV8204 was disassembled and further inspected to determine the full extent of damage to the locking mechanism. A summary of the damage is as follows:

1. The gate side rail had four broken capscrews. Three of the capscrews were from the north side rail and one was from the south side. The north side capscrews were the three top screws and the south side capscrew was the top screw on its side.

2. The gate side skirt plate was severed at the location of the fourth from the top cap screw on the north side. The bottom portion of the plate had been pushed down approximately 3/16 of an inch.

3. Both top rail chamfers had metallic deposits and severe galling.

4. Both lever lock arm shoes had galled areas on the flat bottom surface at the transition of the radiused portion of the shoe.

MSIV 2HV8205 was also inspected by boroscope during the refueling outage. The guide rail ramps appeared to be discolored, but it could not be determined if galling had occurred. The valve was then disassembled and no galling of the skirt or guiderails was found.

DESCRIPTION OF VALVE OPERATION

The Main Steam Isolation Valves are Cooper Industries, WKM Flow Control Division, Power Seal gate valves. The valves are 40X30X40 900 Pound Class double disk gate valves with pneumatic/hydraulic valve operators. The valve disk is a two piece assembly that consists of a gate that is attached to the valve stem and a free floating segment. The gate has upper and lower ramps that are followed by the segment during valve operation to expand the assembly and provide positive sealing of both the upstream and downstream valve seats at both ends of valve travel. This action results in the valve cavity being isolated from the flow stream when the valve is in either the full open or full closed position. The free floating segment is articulated by a lever lock assembly that prevents premature wedging of the gate during mid travel. The lever lock assembly is guided during its travel by rails that are attached to skirts that fit over the valve seats. At both ends of the valve stroke, the lever lock assembly disengages from the rails to allow the wedging action of the gate and segment to seal the valve disk to the valve seats.

The stem position of the MSIV's is controlled by an hydraulic system which exerts lifting pressure on the bottom of a 24 inch diameter piston actuator. Nitrogen, in a closed space above the piston, provides the primary closing force along with the weight of the piston/gate/stem assembly to close the valve. Dual pilot operated solenoid dump valves powered from separate Class 1E DC power sources open and dump hydraulic oil from the bottom of the actuator which allows oil to flow through the valve timing orifice and the nitrogen charge moves the valves to the closed position. The design stroke time for the San Onofre MSIV's was initially between 4 and 5 seconds. A design change to add flow restricting orifices to slow the valve closing stroke time down to approximately 7 seconds was initiated as a result of a failure of the MSIV's at Louisiana Power and Light, Waterford 3 Station.

FAILURE SCENARIO INVESTIGATION

o Possible Scenario 1 (Most Likely)

Wrong Material Used for Sliding Contact Areas

The base material of both the lever lock arm shoes and the rails is 17-4 PH stainless steel. The shoes are overlaid with stellite hardfacing on the radiused portion of the shoe that contacts the inside of the guide rails. The unhardfaced flat bottom of the shoe makes initial contact with the top of the rail as the valve begins closing. The shoe then rotates about the upper outboard edge of the ramp until the back surface of the shoe contacts the stop on the lever lock arm (the stop forms a 45 degree angle to the arm centerline). The shoe then slides across the 17-4 PH guide rail ramp until it clears the ramp and enters the space between the rails. (See figures 1 and 2 of Appendix 2). If the angle of the shoe matches the 45 degree angle of the guide rail ramp, there is a large contact area, as in figure 1. However, if the angles are different, as in figure 2, the contact area is a narrow line.

The 17-4 PH material, when sliding on 17-4 PH material, has a galling threshold stress of 2000 psi (Reference 4). This stress value will be exceeded when the shoe is in line contact with the ramp.

<u>Supporting Evidence</u>

Inspection of the lever lock arm shoes and the rails showed that galling had occurred on the 17-4 PH surfaces of both the shoes and the rails. The galling occurred at the top of the guide rail ramp. When the lever lock arm shoe is contacting this area, the shoe is in line contact with the ramp and the contact stresses are extremely high.

The rails of valve 2HV8204 had been modified from a 45 degree angle to a 30 degree from the vertical angle and the sharp transition had been smoothed to a 1/4 inch radius. Appendix 3, figure 2, calculated the contact stresses under these conditions to be in excess of 123,600 psi. This stress level is well in excess of the 2000 psi threshold stress for galling of 17-4 PH material.

Refuting Evidence

Valves 2HV8205 and 3HV8204 have not shown any galling damage to date.

The loading between the shoe and the ramp during valve closing with matching ramp angles can vary from between 514 pounds and 2961 pounds depending on the value of the coefficient of friction (see figure 1, Appendix 3). The contact stress for these loadings while the shoe is in the unlocked position at the beginning of the stroke, is between 987 and 3508 psi. Since the threshold stress for galling is 2000 psi, there is some probability of galling. If the coefficient of friction is lower than about .30 or the material has somewhat higher resistance to galling than the test threshold value, galling will not occur.

- Possible scenario 2 (Contributing Factor)

  Incorrect Ramp Angle

  The rail ramps on 2HV8204 were reworked from a 45 degree angle to a 30 degree to the vertical angle prior to the Cycle 5 refueling outage. No modifications were made to the shoes or lever lock arms to cause the shoe angle to match the ramp angle at this time. This mismatch resulted in line contact between the shoes and the ramp during unlocking of the lever lock assembly.

  The line contact increased the contact stress at the transition between the upper, flat rail surface and the rail ramp. After initiation of galling at the transition point, the friction forces increased which in turn increased the contact forces and stresses and the galled area moved down the ramp.

Supporting Evidence

Appendix 2, Figure 2 shows the path of the shoe across the rail with the rails cut back to a 30 degree angle. Once the shoe has rotated to position 2, line contact is maintained for the remainder of the travel. The radius at the transition point of the upper surface of the ramp to the rail is the most highly stressed point in the travel. Once the shoe has locked in the 45 degree position in the lever lock arm, the shoe will slide across the radius and galling is extremely likely. Appendix 3, Figure 2 calculated the contact stresses for this condition as over 123,000 psi compared to a threshold galling stress of 2000 psi for the 17-4 PH material.

Refuting Evidence

Valves 2HV8205 and 3HV8204 have not experienced similar damage.

Valve 3HV8205 had damage to the lower rails that was attributed to a combination of differing rail contact (the north arm was either shorter than the south or the valve skirt was not squarely installed). This valve has not shown any damage to the upper rails. When the valve was reworked to remove galling and wear on the lower rail ramps, the (upper) gate side rail ramps were inspected and showed signs of wear. The surfaces were stone ground to remove the wear marks, but there is no reason to believe that the ramp angle was appreciably changed from a 45 degree angle.

o Possible Failure Scenario 3 (Less Likely) (Appendix 1)

Gate/Segment Back Angle Friction Excessive

If the friction coefficient of the sliding surfaces between the back angles of the gate and segment assembly are appreciably higher than the friction coefficients between the seat surfaces, the gate/segment assembly can "stick" together and travel down the ramps locked together. Under these conditions, the force required to unlock the assembly would be increased by the frictional forces built up during back seating of the valve. This increased force would increase the contact stresses between the lever lock arm shoes and the guide rail ramps. The increased stresses would, in turn, result in a higher probability of galling and subsequent valve damage.

Supporting Evidence

The calculation in Appendix 1 indicates that the segment can fail to unlock from the gate and maintain the clamping force of backseating for friction values on the back seat about 2 to 3 times the valve seat friction. With the clamping force holding the segment ar. J gate together, there would be substantially more valve seat frictional force than if the weight of the segment had allowed the assembly to unlock. The increased seat frictional forces would increase the shoe to rail contact stresses and the probability of galling.

Refuting Evidence

Inspection of the valve seat and the back angles of the gate and segment assembly did not indicate any scuffing, scratching or galling. A coefficient of friction of between .4 and .8 would require that some damage had occurred.

o Possible Failure Scenario 4 (Less Likely)

Binding In Lever Lock Arm Journal

If galling or binding occurred on the bearing surface of the lever lock arm journal, the gate and segment can travel together until the lever lock arm shoes impact the guide rails. Under this condition, the force required to unlock the assembly will be increased by the frictional forces in the bushing. This increased force would increase the contact stresses between the shoes and guide rail ramps. This, in turn, would result in a higher probability of galling.

Supporting Evidence

None.

Refuting Evidence

Manual actuation of the lever lock arms of MSIV 2HV8204 did not indicate any unusual drag or binding that would be associated with journal galling or other material distress. The kinematic study in appendix 3 shows that galling of the guide rail ramps can result from stresses found during normal valve operation without additional loadings.

ROOT CAUSE IDENTIFICATION

The root cause of the 2HV8204 failure was the use of 17-4 PH material that is subject to galling for the sliding surfaces between the lever lock arm shoes and the guide rail ramps. This inadequacy was compounded by a revision to the guide rail angle from 45 degrees to 30 degrees without a companion change to the lever lock arm shoe. The change in the ramp angle resulted in an increase in the contact stress between the surfaces by causing line contact between the lever lock arm shoe and ramp at the beginning of the slide down the ramp. The 45 degree ramp angle prior to the change allowed surface contact with a locked shoe while the segment was being lifted.

Damage to the lever lock mechanism can occur when the valve is in either the full open or full closed position and the valve is actuated. When the valve moves from the seated position toward midstroke, the lever arm shoes contact the guide rails and move the floating segment up and away from the valve seat, thus collapsing the gate assembly. The contact forces between the lever arm shoe and the guide rails, if excessive, can result in impact damage or galling of the shoe and rail surfaces and possible failure of the rails, mounting screws, or skirts.

Appendix 2, Figure 1 illustrates the normal travel path of the lever lock arm shoe across the rail ramps as the valve closes. The ramp angle is 45 degrees to vertical and the lever lock arm has a travel stop that limits rotation of the shoe to 45 degrees from vertical. When the shoe first contacts the ramp with its pivot at point 1, the shoe is probably in a horizontal position since the shoe pivot is above the center of gravity of the shoe and gravity will pull the shoe to this position. As the valve closes, the shoe will pivot about the outboard, upper, edge of the ramp and the shoe pivot centerline will move toward point 4. Points 2 and 3 illustrate the path assuming that the shoe contacts the ramp in a 45 degree from horizontal position which could occur due to fluid dynamic forces or if the pin friction is high. The face of the shoe will slide across the ramp, with full contact, until point 5 is reached. At point 5 the back surface of the shoe will contact the stop in the lever lock arm and relative rotation between the shoe and arm will cease. As the valve travels further, contact between the shoe and ramp decrease toward line contact, first on the outboard upper edge of the ramp and then as the outboard edge of the shoe travels past this point, on the ramp itself.

Appendix 2, Figure 2 illustrates the lever lock arm shoe travel path during valve closing with the rails reworked to a 30 degrees from vertical angle. Note that nothing was done to the lever lock arm or the shoe to change the locking point at a 45 degree angle. The shoe will contact the top surface of the rail, point 1, as before and will rotate about the outboard upper edge of the rail ramp until point 2 is reached and the shoe locks in the lever lock arm. At this point, the shoe will start to slide on a narrow line of contact with the edge of the rail ramp. The shoe will continue sliding on this line of contact, with increasing load until the edge of the shoe reaches the ramp. At this point the shoe material changes from 17-4 PH to a stellite overlay. The shoe continues down the ramp until it clears the rail at point 3.

The material of the rail ramp and the bottom of the shoe are both 17-4 PH stainless steel. This material combination is particularity susceptible to galling with a threshold galling stress as low as 2000 psi (Reference 4). The force between the shoe and the ramp is a function of the geometry, the weight of the segment and friction. As the friction coefficient between the shoe and the ramp increases, the resultant force vector moves closer to the center of rotation of the lever lock arm and requires more and more force to slide the shoe down the ramp. Once initial galling has occurred, the roughened surface increases the friction and the resultant force, which causes an increase in contact stress and increased galling. After several galling cycles, the forces can get high enough to sever the set screws attaching the rail to the skirt plate and skirt plate fracture can occur.

CORRECTIVE ACTIONS

The Unit 2 MSIV's have been modified to eliminate the possibility of galling of the guide rail ramp to lever lock arm shoe interfaces. The following changes have been made:

1. The guide rail ramps have been reconfigured to a 30 degree from the vertical angle.

2. The guide rail ramps have been hardfaced with Stellite 6B.

3. The Lever lock arm shoes have been reconfigured to match the 30 degree to vertical angle of the guide rail ramps.

4. The lever lock arm shoes have been hardfaced with Stellite 6B on all guide rail contact surfaces.

The Unit 3 MSIV's will be modified to the new shoe/rail configuration during the next refueling outage.

IDENTIFICATION OF OTHER SUSCEPTIBLE ITEMS

All WKM Pow-R-Seal gate valves use a similar mechanism for expanding the valve disks into the valve seats. As such, all WKM Pow-R-Seal valves should be examined. Engineering will review the designs of all WKM valves used in safety related applications and determine if they are subject to similar failure modes. Appropriate action will be taken if design changes are required.

10CFR21 REPORTABILITY

The generic problem of lever lock arm/guide rail ramp galling was prev reported to the NRC by Southern California Edison in July of 1988. The NRC issued Information Notice 88-59 in August of 1988 to inform the industry of the potential problem.

REFERENCES

1. INTERIM ROOT CAUSE ANALYSIS For The Guide Rail/Lev-R-Lock Interaction Problem For MSIVs SONGS 6-19-88

2. Inspection Results Of SONGS - 3MSIV - 8205 8-24-88

3. Advanced Mechanics of Materials, Boresi and Sidebottom

4. Armco product Data Bulletin No S-45

APPENDIX 1

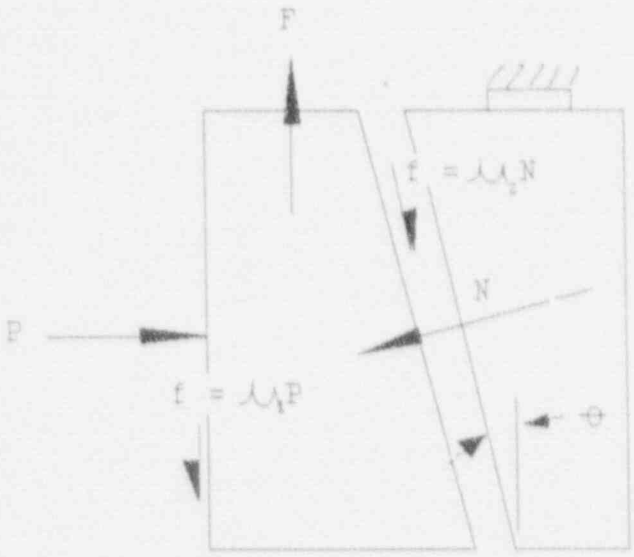
Analysis of Gate/Segment Back Angle Friction Forces
Possible Failure Scenario 3

ANALYSIS OF GATE/SEGMENT FORCES

This analysis examines the gate and segment forces with respect to geometry and friction. When the valve is backseated in the open position, the disk segment contacts a stop on the valve bonnet and the gate continues its upward travel. As the gate continues to move, the back angles of the gate segment assembly cause the disk to expand into the seats. The clamping force between the seats is a function of the applied stem force, geometry and the friction coefficients.

When the motion of the gate is reversed and the disk drives down, there is a possibility that the clamping forces will not relieve. In this case, the clamping force will increase the frictional drag on the segment and will add to the forces required to unlock the mechanism. This increased force can add to the contact stresses between the lever lock arm shoe and the ramp on the skirt rails.

FORCES WITH STEM DRIVING UP (IGNORING WEIGHT OF GATE AND SEGMENT)



$\sum_H = 0$

$P = N \, COS\,\theta - \mu_2 N \, SIN\,\theta$

$P = N \,(COS\,\theta - \mu_2 \, SIN\,\theta)$

$N = P/(COS\,\theta - \mu_2 SIN\,\theta)$

$\sum_V = 0$

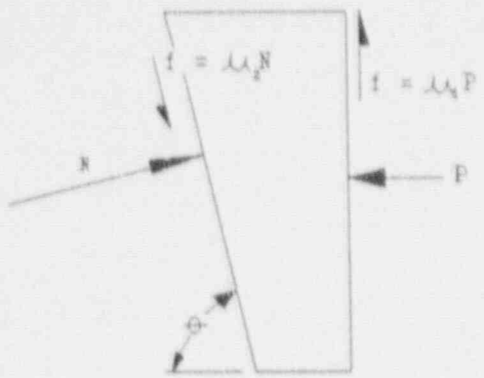
$F = \mu_1 P + \mu_2 N \, COS\,\theta$

COMBINING EQUATIONS YIELDS

$F = \mu_1 P + (\mu_2 P \, COS\,\theta)/(COS\,\theta - \mu_2 \, SIN\,\theta)$

$F = P[(\mu_1 + \mu_2 \, COS\,\theta)/(COS\,\theta - \mu_2 \, SIN\,\theta)]$

FORCES WITH STEM DRIVING DOWN (IGNORING WEIGHT OF SEGMENT)



$$\sum_H = 0$$
$$P = N\ COS\ \theta + \mu_2\ N\ SIN\ \theta$$
$$N = P/(COS\ \theta + \mu_2\ SIN\ \theta)$$
$$\sum_V = 0$$
$$\mu_1\ P + N\ SIN\ \theta = \mu_2\ N\ COS\ \theta$$

COMBINING EQUATIONS YIELDS

$$\mu_2 = (\mu_1\ COS\ \theta + SIN\ \theta)/(COS\ \theta - \mu_1\ SIN\ \theta)$$

BASIC PROGRAM FOR FRICTION VALUES

```
10 CLS
30 LPRINT
40 LPRINT
50 LPRINT
60 LPRINT "      SEAT FRICTION  VS. SEGMENT FRICTION
FOR SEGMENT NOT UNLOADING DURING TRAVEL"
70 LPRINT
80 LPRINT "  SEAT FRICTION U(1)","SEGMENT FRICTION U(2)"
90 LPRINT
100 A=15*3.1417/180
110 FOR U1=.01 TO 9.000001E-02  STEP .01
112 GOSUB 120
115 NEXT U1
116 GOTO 150
120 U2=(U1*COS(A)+SIN(A))/(COS(A)-U1*SIN(A))
130 LPRINT USING "         .##                         .##";(U1),(U2)
140 RETURN
150 FOR U1= .1 TO .5 STEP .05
151 GOSUB 120
155 NEXT U1
160 LPRINT::::
```

BASIC PROGRAM CONTINUED

```
170 LPRINT " BASED ON EQUATION        U1*COS (A) + SIN (A)

U2 = -------------------                              COS A - (U1)(SIN A)
180 LPRINT "  WHERE:   A  = BACK ANGLE
181 LPRINT "           U1 = COEFFICIENT OF FRICTION BETWEEN SEATS
182 LPRINT "           U2 = COEFFICIENT OF FRICTION BETWEEN GATE AND SEGMENT
190 END
```

RESULTS OF BASIC PROGRAM

SEAT FRICTION VS, SEGMENT FRICTION
FOR SEGMENT NOT UNLOADING DURING TRAVEL

SEAT FRICTION U(1)	SEGMENT FRICTION U(2)
.01	.28
.02	.29
.03	.30
.04	.31
.05	.32
.06	.33
.07	.34
.08	.36
.09	.37
.10	.38
.15	.44
.20	.49
.25	.56
.30	.62
.35	.68
.40	.75
.45	.82

With a seat coefficient of friction of .20 (which could be reasonably expected) the back angle friction would have to be .49 in order for the gate and segment to fail to unlock due to seat friction while the stem is driving down. Visual inspection of the back angles of the gate/segment assembly did not show any signs of galling, roughness or damage. Self locking of the gate/segment is, therefore, highly unlikely.

APPENDIX 2

Shoe Motion as it Travels Across Rail Ramp

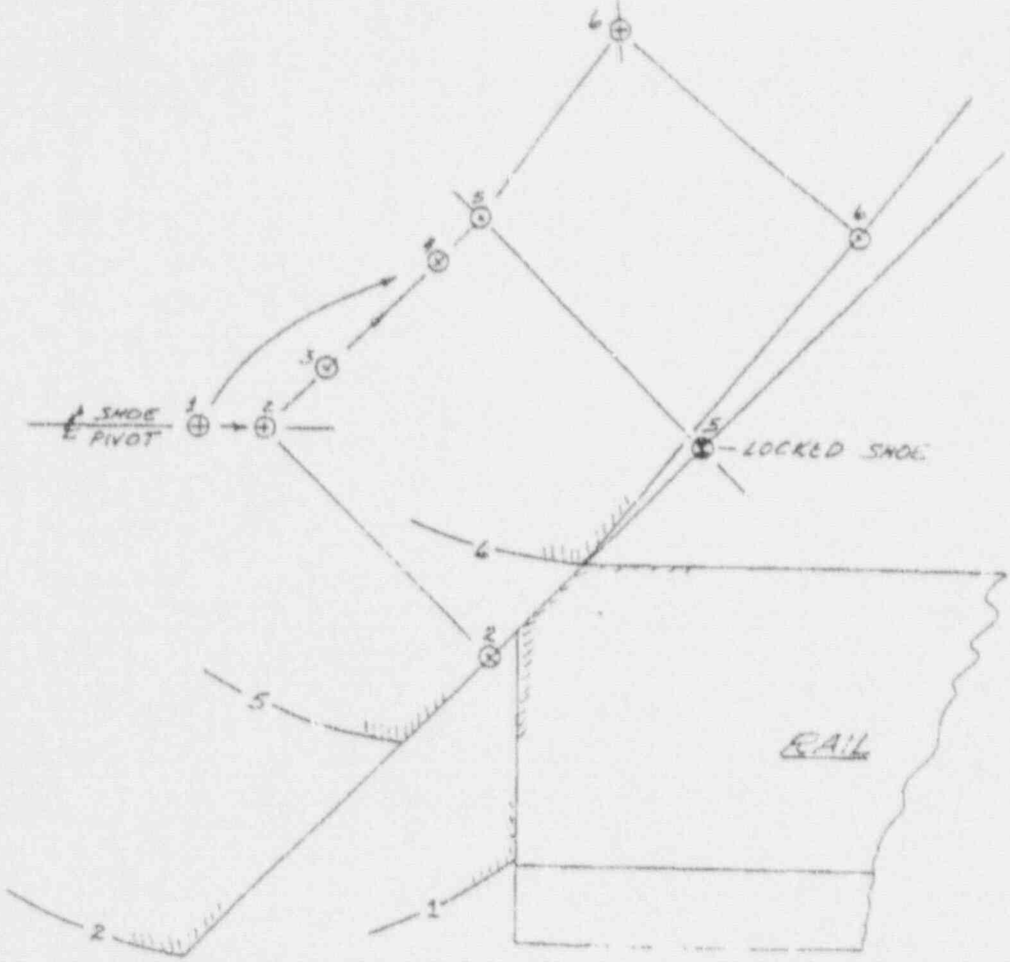


FIGURE 1 (ROTATED 90 DEGREES)
"AS DESIGNED" (45 DEGREE) RAIL RAMP CONFIGURATION

1) INITIAL CONTACT (1) Assumes that shoe is in a horizontal position when it contacts the top of the rail.

2) INITIAL CONTACT (2) Assumes that shoe is rotated to a 45 degree angle when contact occurs. Note that contact is a line between the rail corner and the flat surface of the shoe.

3) LOCKED SHOE (5) Shoe back angle contacts the stop in lever lock arm and shoe locks in position. Sliding starts.

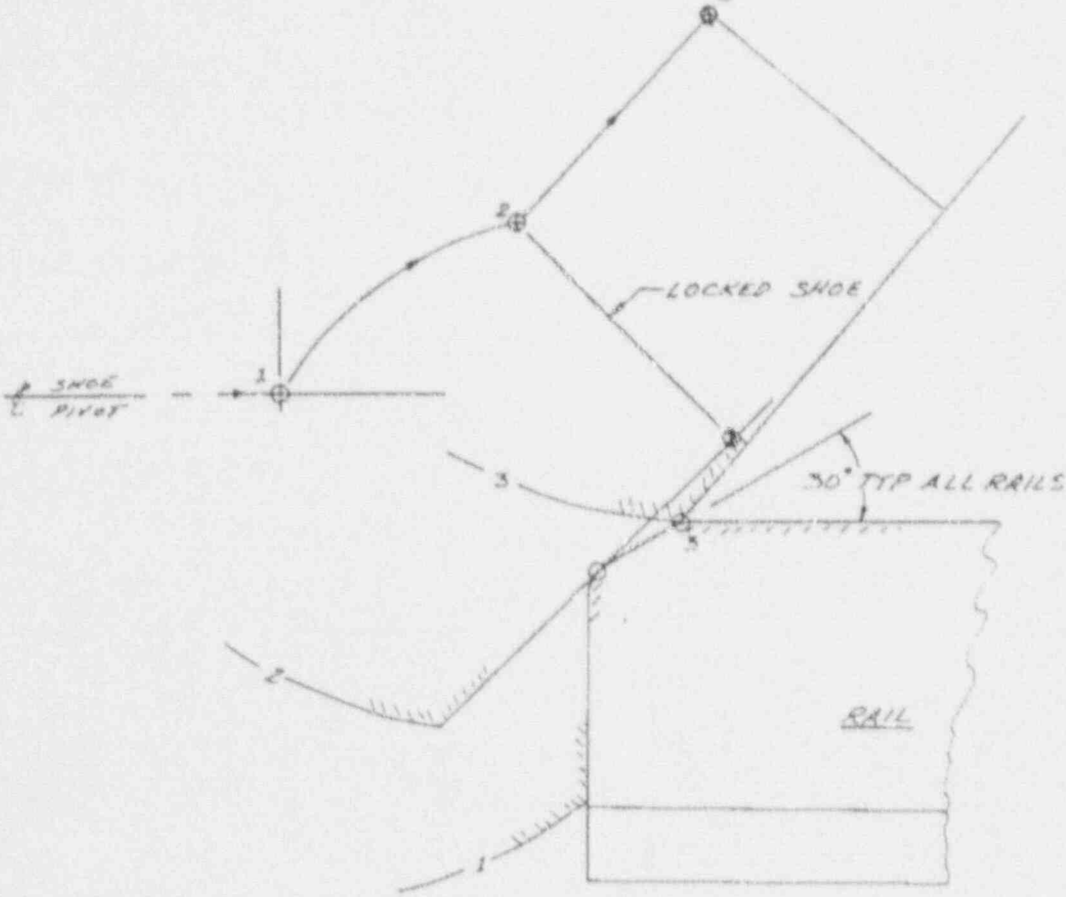


FIGURE 2

REVISED (30 DEGREE) RAIL RAMP CONFIGURATION
(ROTATED 90 DEGREES)

1) INITIAL CONTACT (1) Assumes that shoe is in a horizontal position when it contacts the top of the rail.

2) INITIAL CONTACT (2) Assumes that shoe is rotated to a 45 degree angle when contact occurs. Note that contact is a line between the rail corner and the flat surface of the shoe.

3) END OF RAMP (3) Once the shoe passes this point it can rotate to a horizontal position and will remain between the rails.

APPENDIX 3

Kinematics and Contact Stresses

KINEMATICS AND CONTACT STRESSES

This appendix contains scale drawing layouts of the lever lock arms and rails. These layouts define the motion of the arm and shoe as the valve moves from the full open position and the lever lock arm shoes contact and move along the rail ramps. Once the motion is determined, it is possible to define scale vectors for forces and friction. These vectors can then be used to calculate the forces and moments that are imparted to the shoe and from these forces, the contact stresses.

Figure 1 is the kinematic diagram for the valve "as designed" condition. This diagram has a 45 degree rail ramp angle from the vertical and assumes that the transition from the flat top of the guide rail to the rail ramp is a sharp edge.

Figure 2 is the kinematic diagram for the "as found" condition of valve 2HV8204. This diagram has a 30 degree rail ramp angle from the vertical and has a 1/4 inch radius between the top of the rail and the ramp.

Both figures contain force vectors for a coefficient of friction of both .20 and .40. These friction coefficients were chosen as fair estimates of the upper and lower limits for normal friction of 17-4 PH stainless steel sliding on 17-4 PH stainless steel. The force vectors were drawn and used to scale the moment arms between the vectors and the lever lock arm pin. The moment arms were then used in the calculation of the forces on the ramps.

All dimensions used were obtained from measurements taken from valve 2HV8204. The segment weight of 2300 pounds was measured using a dynamometer.

CALCULATION OF CONTACT FORCES

The contact forces of the lever lock arm shoe on the guide rail ramp will be estimated by determining the force at the shoe that is required to lift the segment weight of 2300 pounds. The valve stem was assumed to move at a constant velocity of 7 inches per second and all acceleration of the mass was assumed to occur while one lever lock arm shoe was in contact with the guide rail ramp.

It was further assumed that the motion during the acceleration was simple, undamped, harmonic motion that could be modeled as a spring/mass system. The natural frequency of the system was calculated using the spring constant of the lever lock arm in bending and the mass of the segment. The mass of the arm, shoe and pins were ignored as being too small to affect the results of the analysis.

The spring constant of the lever lock arm was determined by modeling the arm in the ANSYS finite element program. The program determined that the deflection of the arm due to the 2300 pound weight of the segment was .00061 inch. This value was used to calculate the natural frequency of the spring mass system as follows:

$$f = 1/2\pi\sqrt{k/m}$$
$$f = 1/2\pi\sqrt{w/m\delta}$$
$$f = 1/2\pi\sqrt{g/\delta}$$
$$f = 3.13/\sqrt{\delta}$$
$$f = 3.13/\sqrt{.00061}$$
$$f = 126cps, P = .008sec$$

The natural period of the arm is .008 seconds. This can be used to calculate the average acceleration assuming that the deflection occurs in 1/4 period and the initial velocity of the arm is at .823 inches per second.

$$\textit{constant acceleration}$$
$$acc = 2(\delta \; -(-)V_0 \; t)/t^2$$
$$acc = 2(.00061 + .823(.002))/(.002)^2$$
$$acc = 1128 \; inches \; /sec$$
$$g = 2.9$$
$$\textit{simple harmonic motion}$$
$$acc = V_0 \; \omega$$
$$acc = .823 \; (\sqrt{384}/\sqrt{.00061})$$
$$g = 1.7$$

The acceleration due to the impact of the arm with the guide rail ramp was calculated to be 2.9 g's. This can be considered an upper bound value for the acceleration loading. Assuming simple harmonic motion, the maximum acceleration is 1.7 g's.

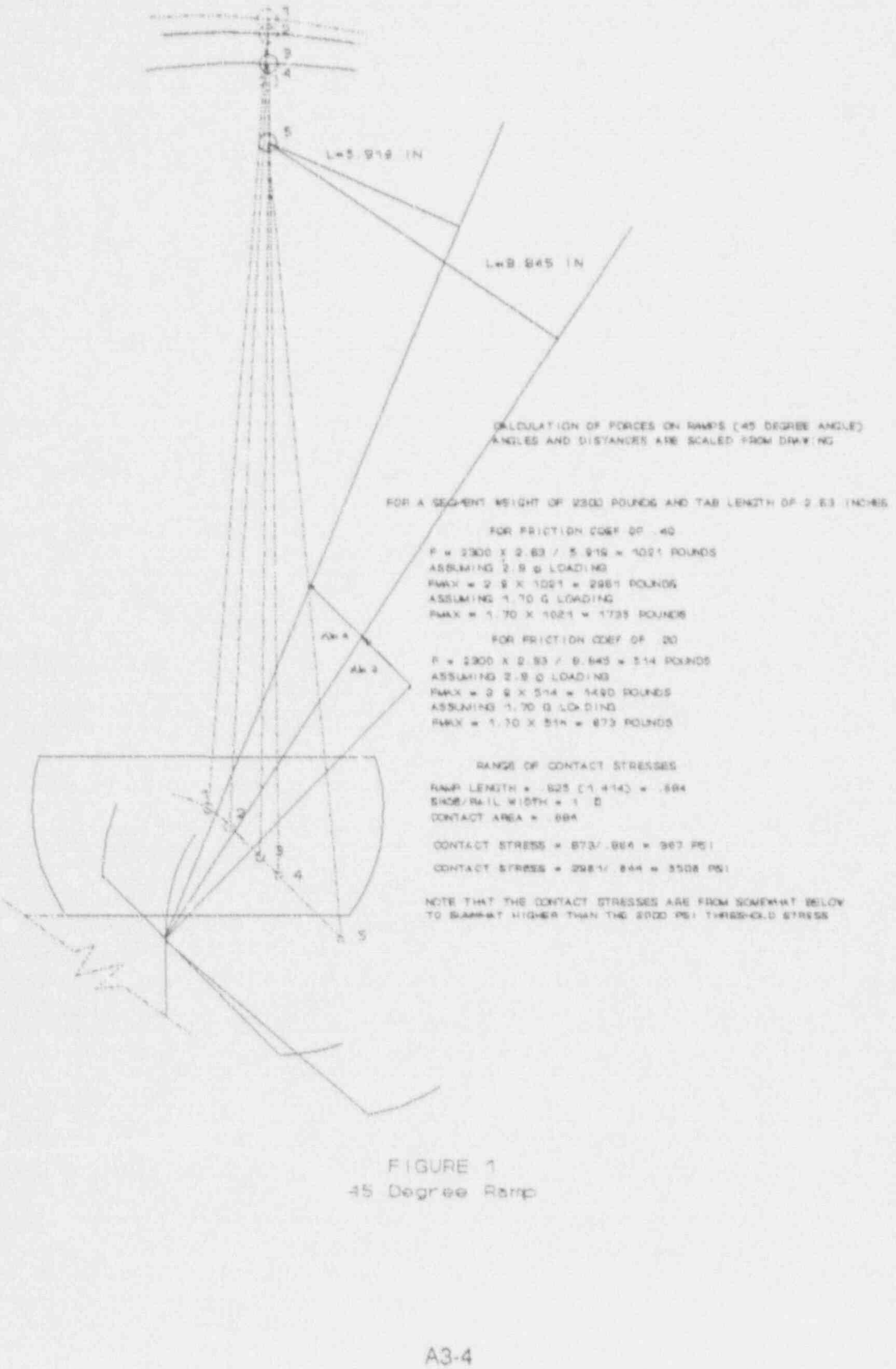
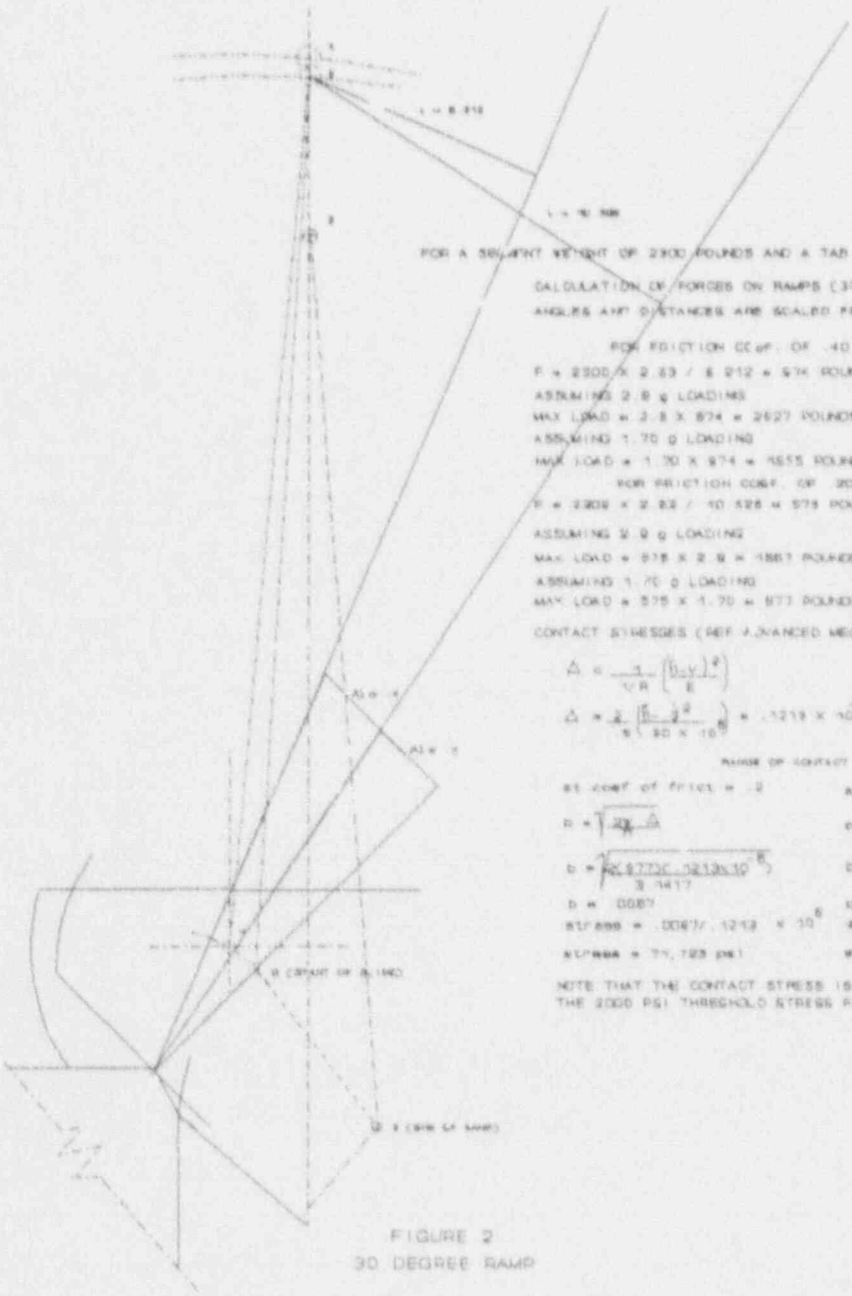


FIGURE 1
45 Degree Ramp



FOR A SEGMENT WEIGHT OF 2300 POUNDS AND A TAB LENGTH OF 2.93 INCHES

CALCULATION OF FORCES ON RAMPS (30 DEGREE ANGLE)

ANGLES AND DISTANCES ARE SCALED FROM DRAWING

FOR FRICTION COEF. OF .40

F = 2300 X 2.93 / 8.212 = 874 POUNDS

ASSUMING 2.9 g LOADING

MAX LOAD = 2.9 X 874 = 2527 POUNDS

ASSUMING 1.70 g LOADING

MAX LOAD = 1.70 X 874 = 1555 POUNDS

FOR FRICTION COEF. OF .20

F = 2300 X 2.93 / 10.528 = 575 POUNDS

ASSUMING 2.9 g LOADING

MAX LOAD = 575 X 2.9 = 1667 POUNDS

ASSUMING 1.70 g LOADING

MAX LOAD = 575 X 1.70 = 977 POUNDS

CONTACT STRESSES (REF ADVANCED MECH OF MAT. L - BORESI)

$$\Delta = \frac{1}{VR}\left(\frac{b \cdot v}{E}\right)^2$$

$$\Delta = \frac{2}{\pi}\left(\frac{(5 \cdot 3)^2}{30 \times 10^6}\right) = .1213 \times 10^{-6}$$

RANGE OF CONTACT STRESSES

at coef of frict = .2	at coef of frict = .4
$b = \sqrt{\frac{2K}{\pi}\Delta}$	$b = \sqrt{\frac{2K}{\pi}\Delta}$
$b = \sqrt{\frac{2(.977)(.1213 \times 10^{-6})}{3.1417}}$	$b = \sqrt{\frac{2(.2527)(.1213 \times 10^{-6})}{3.1417}}$
b = .0087	b = .015
stress = .0087/.1213 $\times 10^6$	stress = .015/.1213 $\times 10^6$
stress = 71,723 psi	stress = 123,660 psi

NOTE THAT THE CONTACT STRESS IS MUCH HIGHER THAN THE 2000 PSI THRESHOLD STRESS FOR GALLING OF 17-4 PH

FIGURE 2
30 DEGREE RAMP

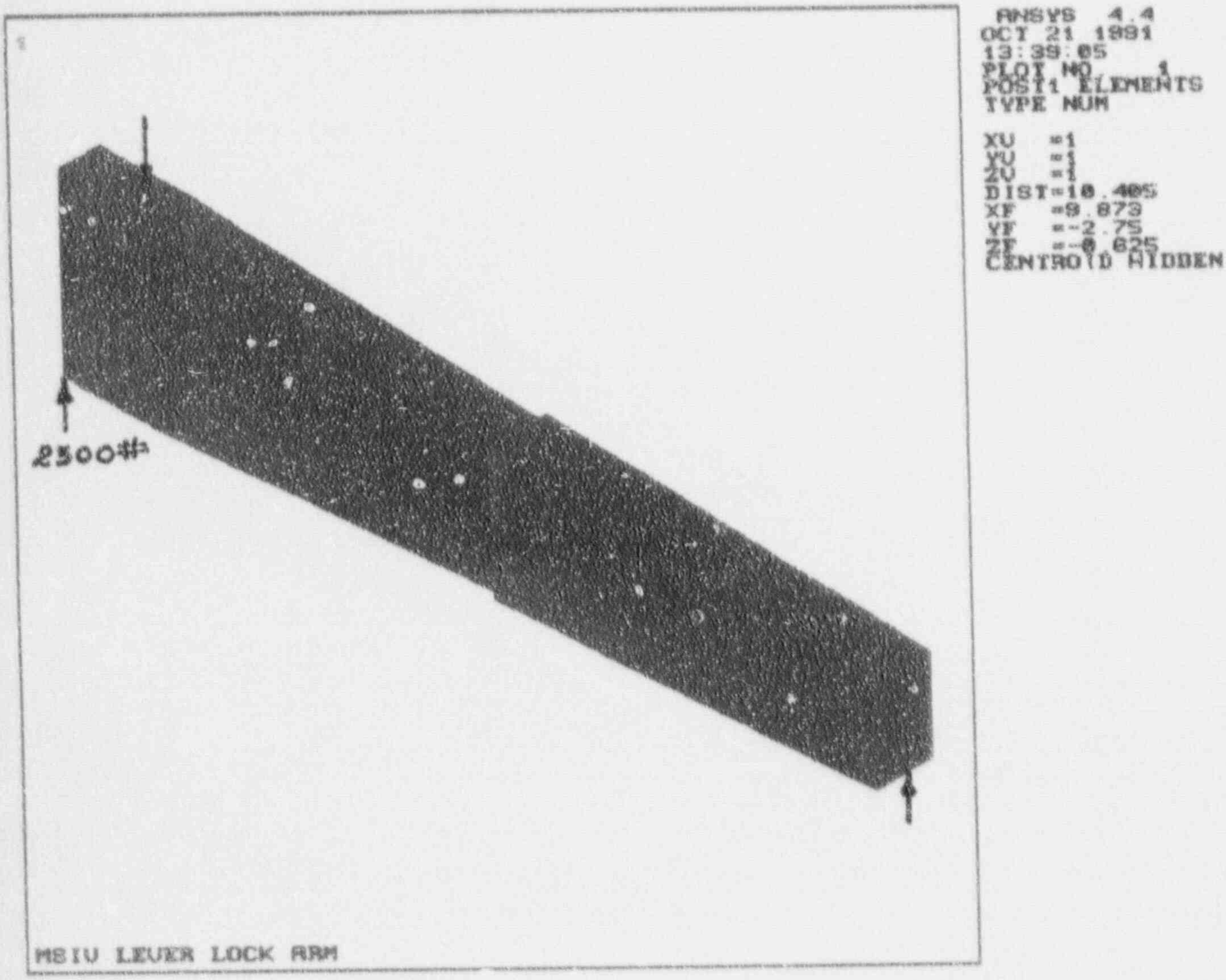


FIGURE 3

ANSYS MODEL OF LEVER LOCK ARM

ANSYS FINITE ELEMENT PROGRAM OUTPUT

The following table is a copy of the highest values of the lever lock arm deflection output that was used for calculation of the natural frequency of the arm. The maximum value at point 843 was used in the calculation. The complete results of the analysis are on file with G. Gartland.

THE FOLLOWING X,Y,Z DISPLACEMENTS ARE IN GLOBAL COORDINATES

***** POST1 NODAL DISPLACEMENT LISTING *****

LOAD STEP 1 ITERATION = 1 SECTION = 1
TIME = 0.00000E+00 LOAD CASE = 1

THE FOLLOWING X,Y,Z DISPLACEMENTS ARE IN GLOBAL COORDINATES

NODE	UX	UY	UZ
843	0.70484475E-05	-0.61042662E-03	0.49095384E-06
848	0.70921406E-05	-0.37221059E-03	0.49736207E-06
838	0.72524740E-05	-0.37205007E-03	0.48804520E-06
738	0.21954611E-03	-0.29013213E-03	-0.15824420E-05
736	0.21971333E-03	-0.28993812E-03	0.49808936E-05
729	0.21632150E-03	-0.27437074E-03	0.21143691E-05
687	0.21681921E-03	-0.27378469E-03	0.12844332E-05
739	0.21749919E-03	-0.27278684E-03	0.20477262E-05
802	-0.19765024E-03	-0.28744497E-03	0.28863970E-05
735	0.21783288E-03	-0.27240034E-03	0.13507744E-05
804	-0.19741483E-03	-0.28729217E-03	-0.36258814E-05
845	0.15357781E-03	-0.30719584E-03	0.12627243E-05

MAXIMUMS

NODE	762	843	695
VALUE	0.22038163E-03	0.61042662E-03	0.20153209E-04

***** RUN COMPLETED ***** CP = 594.8400 TIME = 10.0685

APPENDIX 4

Valve Drawing and Photographs

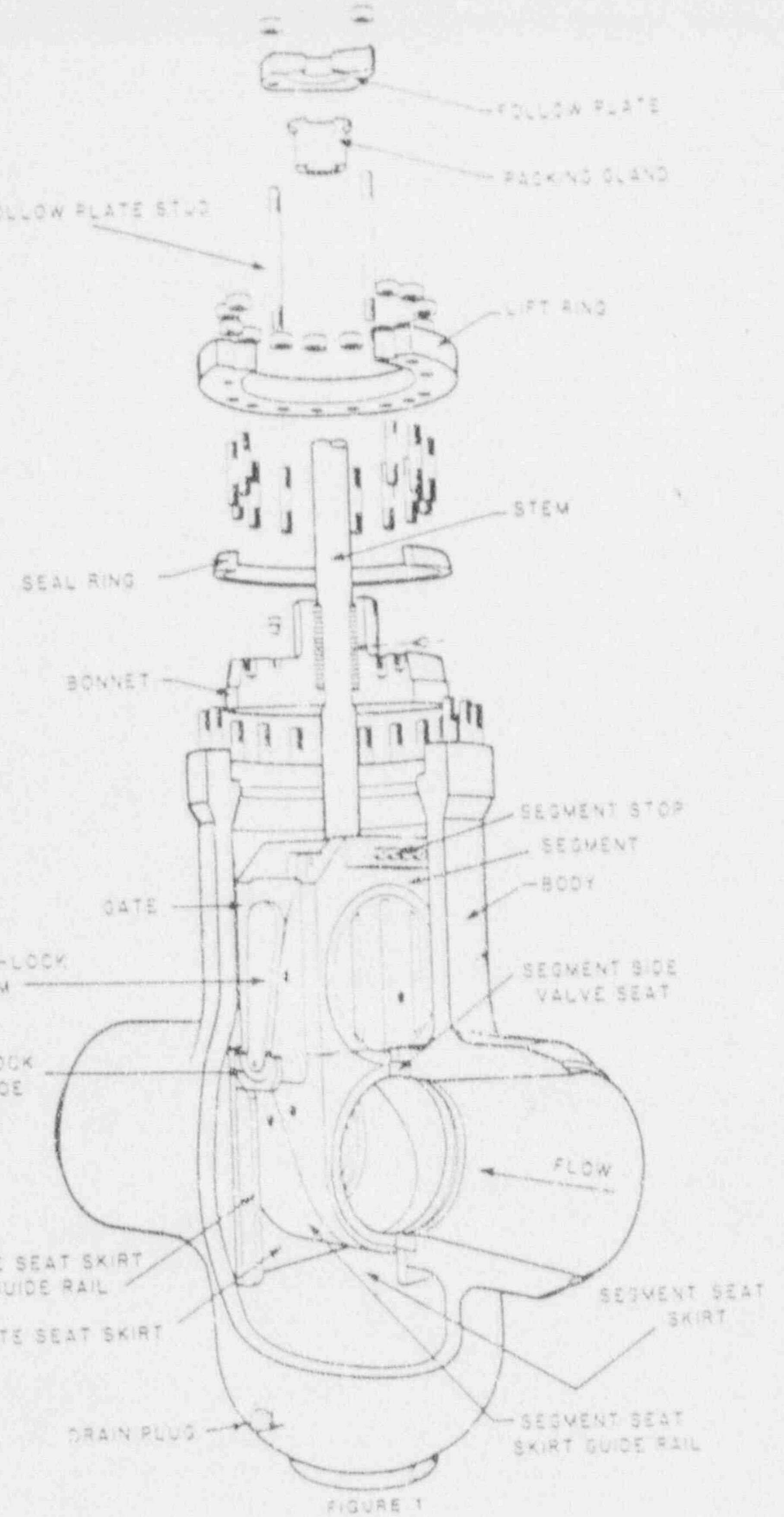
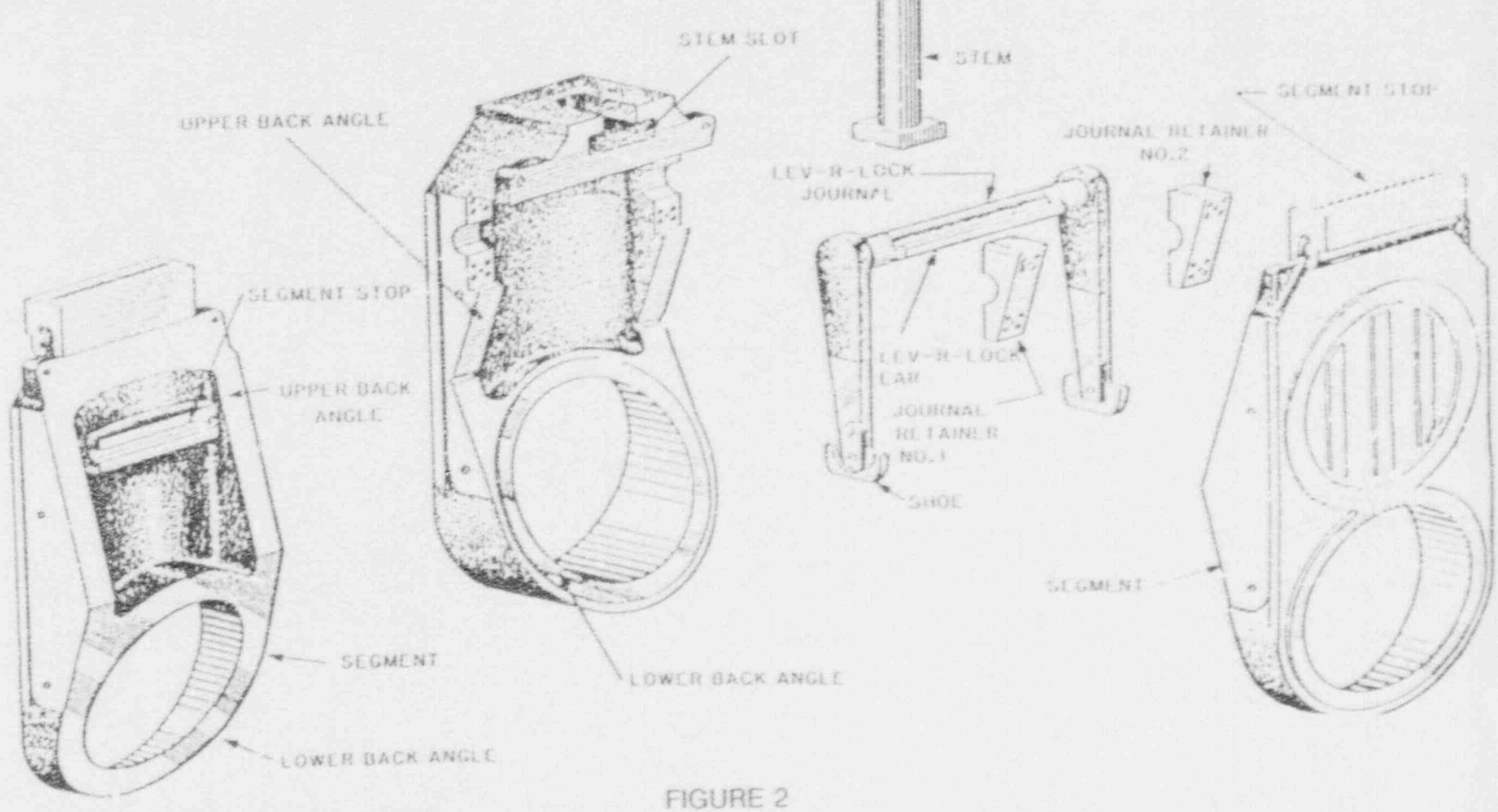


FIGURE 1

Configuration of Main Steam
Isolation Valve



REVERSE SEGMENT - SHOWN
ROTATED 180 DEGREES

FIGURE 2

GATE AND SEGMENT ASSEMBLY DETAIL

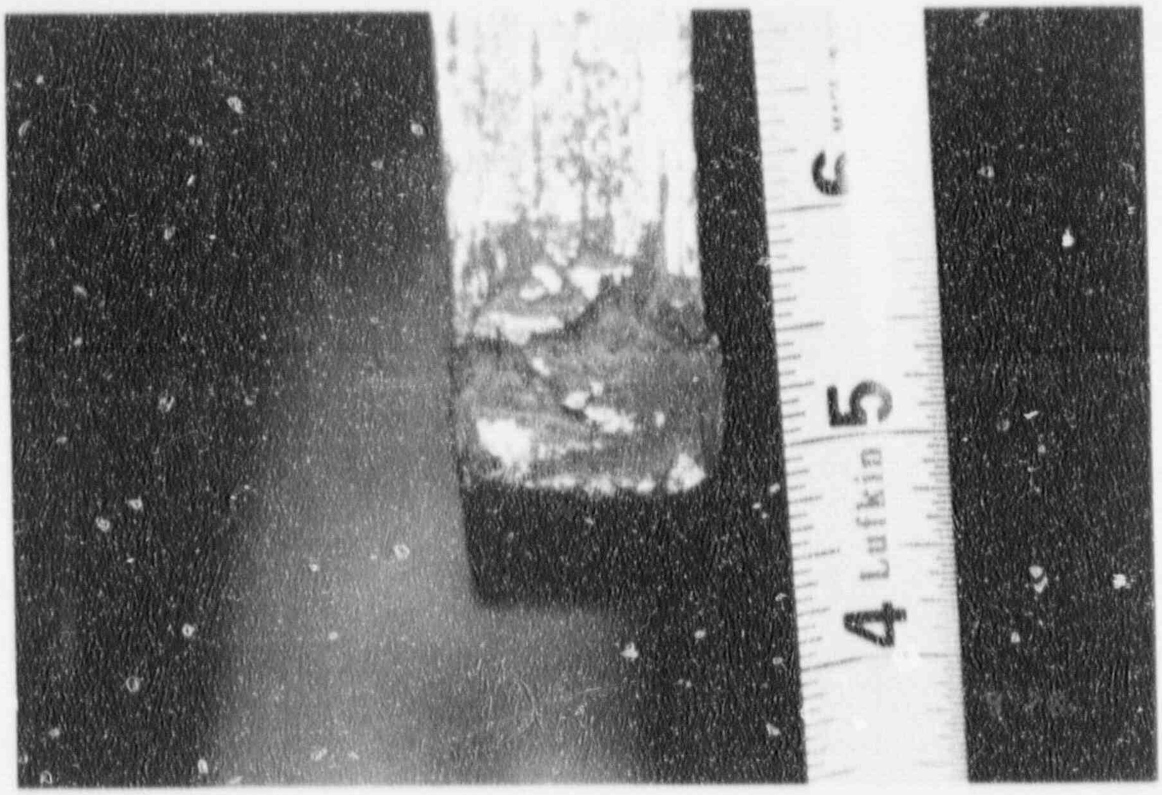


PHOTO #1
NORTH SIDE LEVER LOCK ARM SHOE DAMAGE

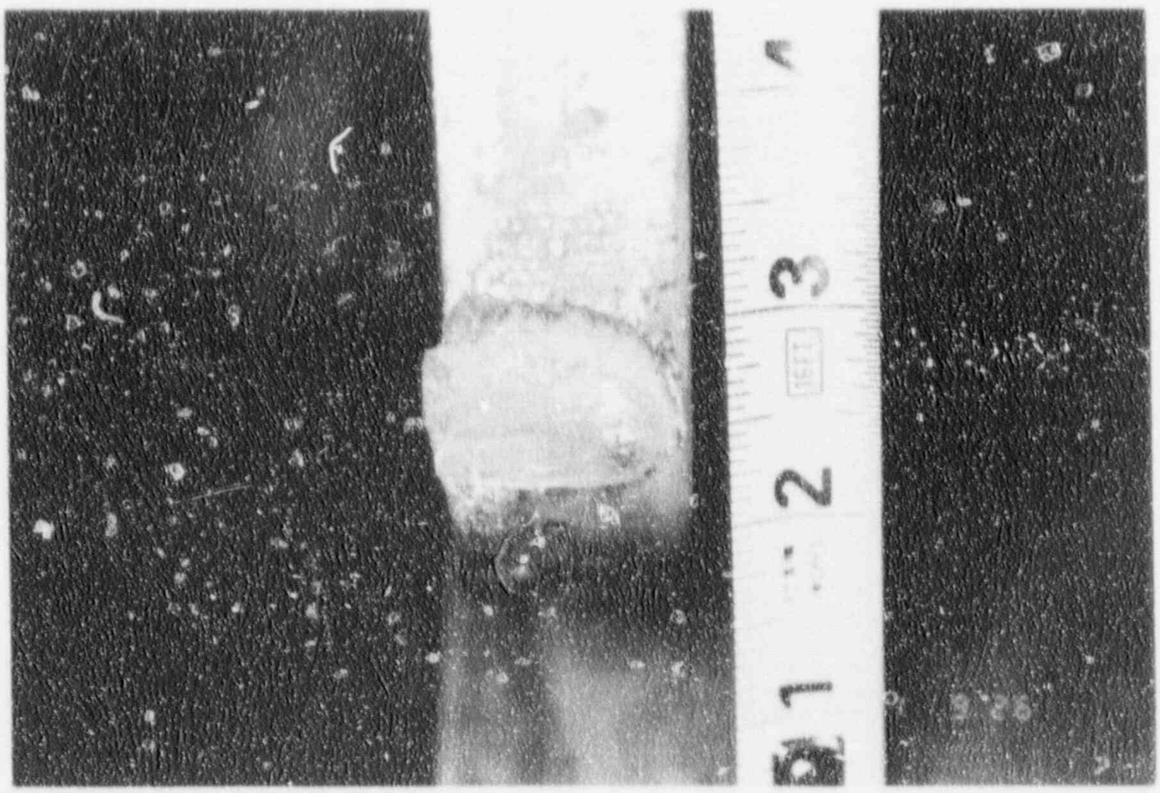


PHOTO #2
SOUTH SIDE LEVER LOCK ARM SHOE DAMAGE

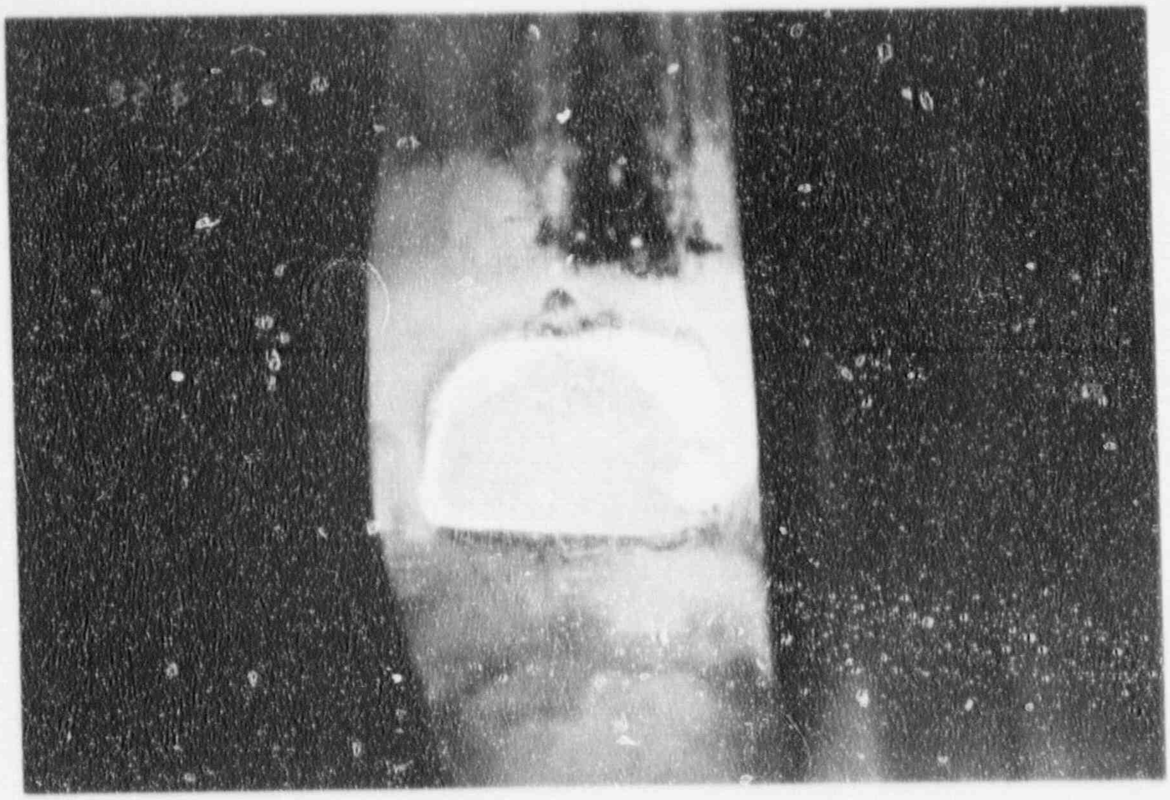


PHOTO #3
NORTH SIDE SKIRT RAIL DAMAGE

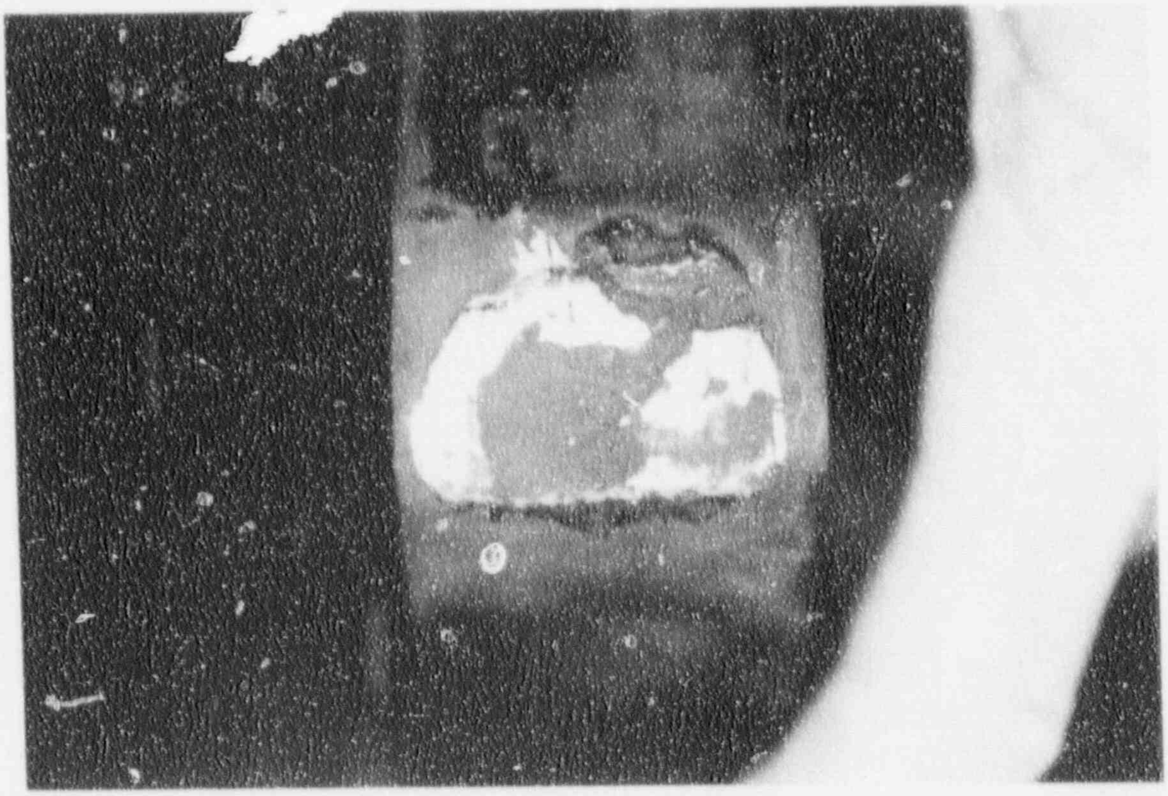


PHOTO #4
SOUTH SIDE SKIRT RAIL DAMAGE

PHOTO #5
CRACK IN DISK SKIRT (END VIEW)

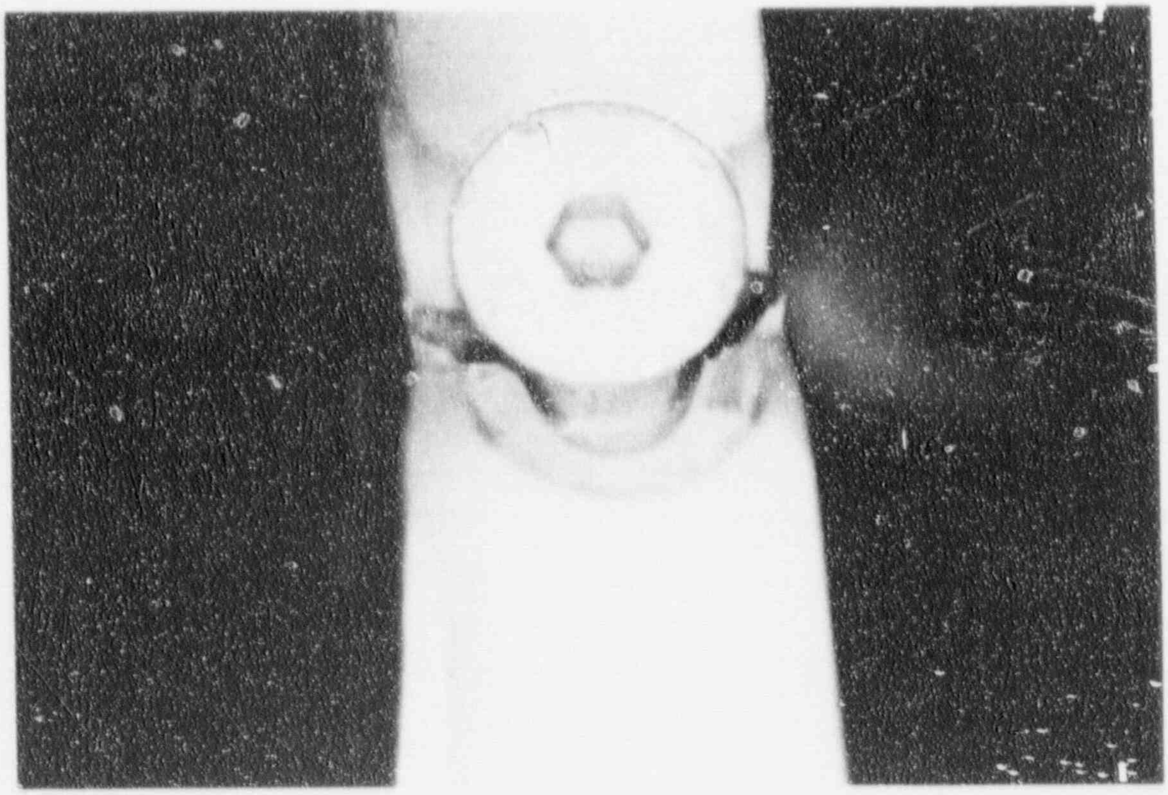


PHOTO #6
CRACK IN DISK SKIRT (PLAN VIEW)

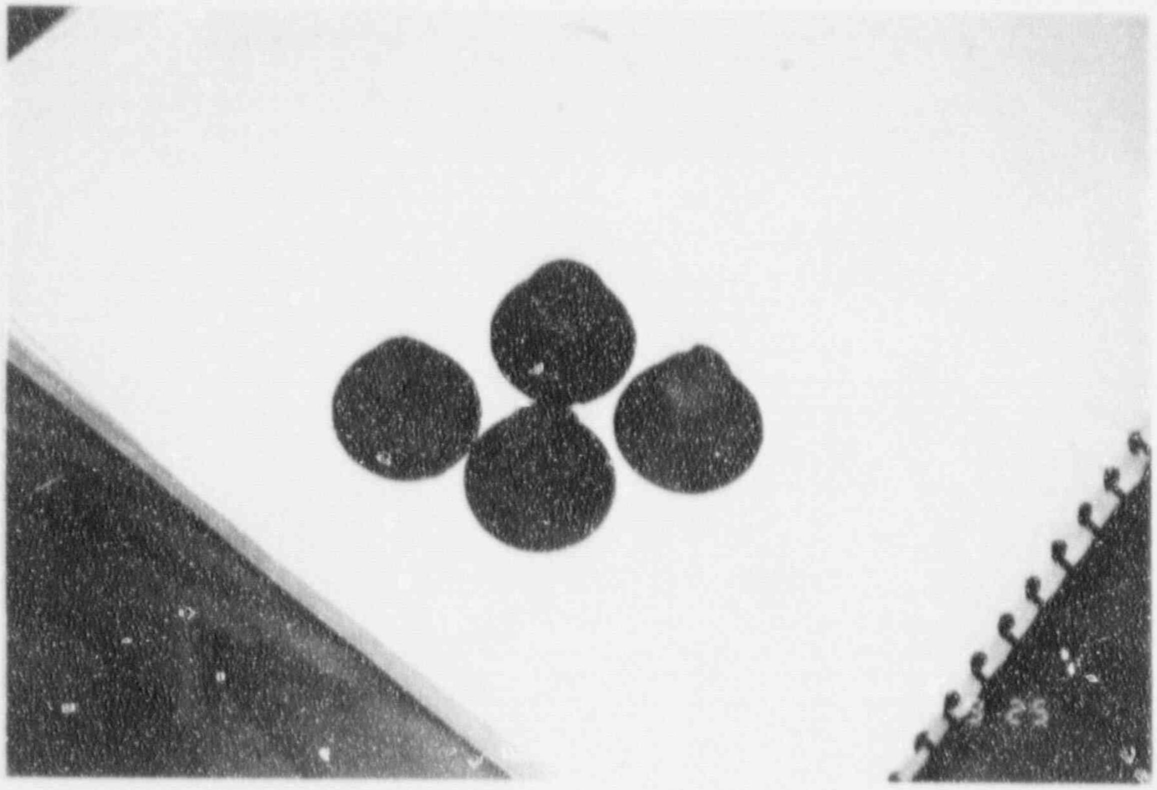


PHOTO #7
BROKEN CAP SCREWS

APPENDIX 5

KALSI Engineering Analysis Report

ANALYSIS FOR DESIGN MODIFICATIONS OF MSIV GATE GUIDE

DOCUMENT NO. 1719, Rev. 1
OCTOBER 24, 1991

Prepared for

SOUTHERN CALIFORNIA EDISON COMPANY
SONGS UNIT 2
SAN ONOFRE, CALIFORNIA

Prepared by

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KEI File No. 3.25

KALSI ENGINEERING, INC.
MECHANICAL DESIGN & ANALYSIS

TABLE OF CONTENTS

ANALYSIS FOR DESIGN MODIFICATIONS OF MSIV GATE GUIDE

1. INTRODUCTION

This report summarizes the results of the gate guide analysis performed for the Main Steam Isolation Valve (MSIV) to support Southern California Edison's design modification effort for SONGS Unit 2. These main steam isolation valves are designed and manufactured by W-K-M Product Division of Cooper Industries. The basic design consists of a through-conduit, parallel expanding gate valve of 40" x 30" x 40" size which can provide the wedging action in the closed as well as the open position. Figure 1 shows the details of the internal components and the associated nomenclature. In addition, a description of W-K-M gate and segment assembly operation and its wedging/expanding action in the open and closed position is given in Section 2 immediately following the introduction.

The failures of the guide rail capscrews and skirt were discovered during the August 1991 refueling outage inspection of 2HV-8204 MSIV using an optical boroscope. The MSIV was subsequently disassembled and inspected to determine the cause of failure and the design modifications necessary to correct it.

Inspection of the disassembled gate and segment assembly showed that the failures occurred at these gate skirt locations:

1. The skirt fractured at the fourth capscrew from the top location. This is the narrowest skirt width under tensile load in the stem axis direction. The fracture appears to be tensile overload with some necking at the fractured cross-section. Only the left-hand side of the skirt was fractured (looking from downstream).

2. Three capscrews above the fractured skirt section were broken.

3. The top capscrew on the right-hand side of the skirt was also broken (looking from downstream).

In addition to the obviously fractured components listed above, the following damage or abnormalities were observed during inspection of the gate and segment assembly:

1. The gate rail chamfers, where the lever-lock arms and guide rails interact during valve closing, show severe signs of galling. Galling resulted in metal removal from the bottom of the shoe surfaces and deposited it onto the guide rail chamfers. The segment side guide rails and the top shoe surfaces are in good condition, indicating that the valve did not experience problems during valve opening

2. Sliding marks exist on both gate and segment seating surfaces. The back angle contact surfaces appear to be in good condition.

3. Sliding marks also exist on all four guide rail sliding surfaces.

4. Indentation marks exist on the matching shoes and lever-lock arms at the interaction points where the shoe articulation is stopped by a slot in the arm.

Kalsi Engineering, Inc. was engaged by SCE to perform a detailed review of the current MSIV problems and provide recommendations for corrective actions. The current effort, as documented in this report, is focussed on reviewing the current design deficiency and ensuring that the proposed design modification will eliminate the problems and provide the required safe operation of the MSIVs. More detailed analysis on the effects of operating pressure, operating temperature, local contact stresses, and gate assembly dimensional tolerance stack-up will be performed and documented in a later report.

The proposed design modifications for the lever-lock arm shoes and guide rails, as shown in Figures 4 and 5, are basically the same design modifications used at Louisiana Power & Light, Waterford SES Unit 3, for their MSIVs with the identical valve design. The modifications were successfully implemented in 1988 with no reported problems to date. The main difference between SCE and LP&L's MSIVs is in the operating requirements: the LP&L valves are required to close in a shorter time (as low as 1.6 seconds) while SCE requires the MSIVs to close within 6.9 seconds (normally operated within 3 to 6 seconds).

A mathematical model that simulates impact forces and motions of the gate and segment assembly during closing action was used to conduct the review for the current design and proposed modifications. The model takes into account the velocity of the shoe at impact, the coefficient of friction between the shoe and guide rail, and the local contact geometry, including the contact angle that the shoe can acquire within the clevis of the lever-lock arm. The math model provides a way of analyzing key contributing factors to the guide rail impact force during a valve closing. It quantifies the current design deficiency as well as determines the magnitude of design improvement that can be achieved under the proposed modification. A comparison of the two designs is presented in this report which shows the advantages of the new design in reducing impact force and in the ability to operate under a high coefficient of friction. The detailed dynamic impact force analysis is described in Appendix A.

Problems with the valve opening stroke on the segment guide rails are analyzed in Appendix B. The available stem thrust, the effect of frictional coefficient, and the load on the segment rail are also discussed. Inspections of gate and segment assembly are suggested to assure that the MSIV will operate within the design range.

Section 3 summarizes the analysis results and key conclusions.

2. DESCRIPTION OF W-K-M GATE AND SEGMENT ASSEMBLY PARALLEL EXPANDING OPERATION

Figure 1 shows the details of the internal components of the W-K-M parallel expanding gate valve and the nomenclature used in this report. Basically this valve is a wedge gate valve design in which two wedge pieces are employed which expand in a lateral direction whenever relative axial motion (in the direction of the stem) is allowed to occur between them. This design relies upon the two pieces to travel together as an assembly without any relative axial motion (or lateral expansion) during the entire stroke except at the very ends when the assembly approaches the fully open or fully closed position. To keep the two wedge pieces called "gate" and "segment" from moving relative to each other, a special mechanism is employed which provides the necessary kinematic restraint to the entire assembly. This mechanism is comprised of a "lever lock" assembly, which consists of a lever lock arm, a lever lock shoe, a cam assembly between the gate and segment, and guide rails which are fastened to a seat skirt on each seat. During the mid-travel position, the lever lock shoe is guided between the guide rails which prevents any relative axial motion between gate and segment thereby preventing any tendency for the two pieces to expand laterally which can result in an increase in the force required to move the assembly.

The guide rails are terminated at either end of the stroke of this valve to permit the lever lock shoe to move outside the parallel restraint provided between the guide rails. When going toward the fully open position, this in turn permits the gate to continue to move upwards after the segment hits the bottom of the bonnet which stops the upward movement of the segment. This relative motion between the gate and segment is transmitted through a special cam mechanism (not visible in this sketch) through the lever lock arm which kicks the shoe to the left as shown in Figure 1. This allows a wedging/climbing action between the gate and segment to take place which is accompanied by an increase in the dimension between the parallel faces of the gate and segment.

During normal valve operation, when the valve is given a signal to go closed, the gate and segment assembly starts to move down with the shoe still in the left position until it hits the top of the guide rail (as shown in Figure 1). In the basic design, there is a 45-degree chamfer provided at this location of the guide rail which tends to guide, or force, the shoe from this extreme left position to a central position between the guide rails. It should be noted that the edges of the 45-degree chamfer on the guide rails are relatively sharp with no significant radii. These are potential sites for initiating galling. In the current SCE design for 2HV-8204, the chamfer angle was modified to a smaller angle (from stem axis) and the corner radius at the chamfer was enlarged. The gate and segment assembly travels together through the entire stroke until at the very end when the segment-stop hits the stop pad provided in the body upstream conduit. After the segment motion is stopped and the gate continues to be pushed down by the stem, the relative motion between the gate and segment now engages the other set of inclined planes and starts the wedging action. This is accompanied by a motion of the lever lock arm and the shoe assembly which is kicked to the right in the closed position. In the fully closed position, both the gate and the stem wedge pieces are laterally expanded to firmly contact their respective seats and provide the desired seating force.

3. RESULTS AND CONCLUSIONS

3.1. Valve Closing Action

1. Inspection of the failed MSIV skirt and guide rails (2HV-8204) and review of the detailed valve internal component designs indicate that the current lever-lock arm assembly and guide rail designs have two major deficiencies:

   a. The high contact angle (45 degrees measured from the stem axis) at the shoe/guide rail interaction point is ineffective in guiding the lever-lock shoe toward the space between the guide rails during a valve closing stroke. It results in a high impact force on the guide rail chamfer.

   b. The absence of overlay at the contact surfaces and lack of radius at the guide rail chamfer cause accelerated surface deterioration, high coefficient of friction, reduced torque to rotate the lever-lock arm into its correct position, and excessively high impact force.

2. A mathematical model capable of simulating the dynamic impact action of the assembly was used to analyze the key design factors affecting the gate closing (detailed in Appendix A). The model was used to investigate the effect of coefficient of friction on the guide rail impact force, the effect of shoe/guide rail contact angle for the existing and proposed designs, and the effect of the valve closing speed on the guide rail impact force within the valve operating range. The results of this analysis are summarized in Figures 2 and 3.

3. *Effect of coefficient of friction.* As shown in Figure 2, the current design (45-degree contact angle) has low impact force for the normal range of coefficient of friction (0.2 to 0.4 range). As the contact surfaces deteriorate (μ = 0.6 or higher), there is an asymptotic rise in the magnitude of impact force delivered to the guide rails. Theoretically, as the coefficient of friction reaches a magnitude of 0.72 (for the 45-degree contact angle design), no torque is generated at the shoe-to-guide rail contact to cause rotation of the lever-lock arm and force it between the guide rails. Under this condition, all the kinetic energy of the moving gate assembly is absorbed by the load-resisting components and converted into strain energy. When this occurs, the magnitude of the forces transmitted to the guide rail becomes very high, and can easily cause shearing of the guide rail-to-skirt bolting or the weakest section of the

skirt itself, as observed in the failed valves. The apparent coefficient of friction of 0.72 is possible with the galling contact surfaces and sharp corner indentation.

4. *Effect of shoe-to-guide rail contact angle.* For the WKM proposed design modifications (30 degrees of contact angle WITH RESPECT TO STEM AXIS), the impact force is much lower than the current design under the normal range of coefficient of friction. But more importantly the new design can tolerate a high coefficient of friction, up to 1.0, without significantly increasing the contact force. The advantage of a shallow contact angle that can effectively convert contact force to lever-lock arm torque to rotate the arm toward the center of the guide rails is clearly illustrated in Figure 2. This shallow angle keeps the impact force low and avoids surface degradation.

   It should be noted that the normal contact force at the shoe/guide rail interface is perpendicular to the bottom shoe surface, as shown in Appendix A force diagram, while the frictional force is parallel to the shoe surface. Therefore, the bottom shoe surface angle during the closing contact determines the magnitude of the rotating torque to the lever-lock arm instead of the guide rail chamfer angle.

   Figure 3 shows the time history plot of the impact forces for the old and new designs. As the coefficient of friction changes from 0.6 to 0.67, the old 45-degree angle design increases the impact force significantly from 7,050 to 11,190 pounds while the new design keeps the impact force at less than 2,160 pounds of impact force (for closing speed of 10 in/sec).

5. *Effect of valve closing speed.* The SCE valve closing speed is estimated in the range of 5 to 15 in/sec. Figure 2 plots three closing speeds of 5, 10, and 15 in/sec for the MSIV closing speed range. As shown in the figure, the impact force increases rapidly as the closing speed increases. Comparing the current SCE design of 10 in/sec versus the old LP&L design of 25 in/sec shows that the impact forces at $\mu = 0.4$ are 4,252 pounds for SCE and 10,630 pounds for LP&L. Therefore, the SCE design should be able to perform better if other factors remain the same. For the proposed design, similar relationships exist with generally lower impact loads over a wider coefficient of friction range, as shown in Figure 2. However, as the coefficient of friction increases (contact surface degradation), both SCE and LP&L designs

approach an asymptotic rise in the magnitude of impact force delivered to the guide rails at $\mu = 0.72$ for the current designs and $\mu = 1.2$ for the proposed design modification *regardless* of the gate closing speed.

6. In addition to the improvement in contact angle to a more favorable orientation, the new shoe and guide rail designs provide the following design improvements:

   a. Stellite overlay on both shoe and guide rail chamfer surfaces;

   b. Nitro-hardening of the shoe surface;

   c. Generous corner radii in both lever-lock shoe (R = 1") and guide rail (R = 5/8").

   Therefore, based on the above combined design improvements, the new design should have a sufficient safety margin to keep the contact surfaces in good working condition after repeated valve closing.

3.2. Valve Opening Action

As shown in Appendix B calculations, the valve actuator has sufficient hydraulic force (649 kips) to pull the gate assembly upward if the resistance exists due to the shoe and rail sticking.

If the gate and segment back angle surfaces are in good condition, the gate and segment should be in a collapsed configuration with very little load applied to the segment rails. If the back angle surfaces deteriorate much faster than the seat surfaces to reach a significant difference in coefficient of friction as tabulated in Appendix B, then an upward wedging movement may occur, resulting in high contact load on the rails. Another possible shoe and segment rail chamfer contact may occur during the final stage of a valve closing stroke if the dimensional stack-up is outside of the design limits (discussed in Appendix B).

Inspection of the 2HV-8204 MSIV showed this valve to be free from both of these potential problems.

To ensure freedom from these potential problem areas related to the opening stroke operation in other MSIVs, the following inspection should be performed at disassembly:

1. The back angles of the gate and segment should be inspected to ensure that they are in good working condition.
2. The gate and segment assembly, including the segment skirt, should be inspected to ensure that the top lever-lock shoe surface is not contacting the segment rail chamfer during the final wedging action of the gate and segment in a closing stroke.

The force equilibrium analysis in the opening direction shows that, in order to prevent the lever-lock arm from rotating to the center of the guide rails, the coefficient of friction would have to be as high as 1.28. This magnitude of friction is not encountered under normal contact surface conditions even with some degree of surface damage. The new improved shoe and guide rail design with Stellite overlay, nitro-hardening, and ample radii should provide sufficient margin to avoid contact surface galling and high segment rail load.

3.2. Overall Conclusions

The analysis has identified the current design deficiencies. The proposed design modifications should be able to eliminate current design problems. A review of the modifications shows no new problems are introduced. Therefore, it is recommended that the proposed design should be implemented for the SONGS Units 2 and 3 MSIVs.

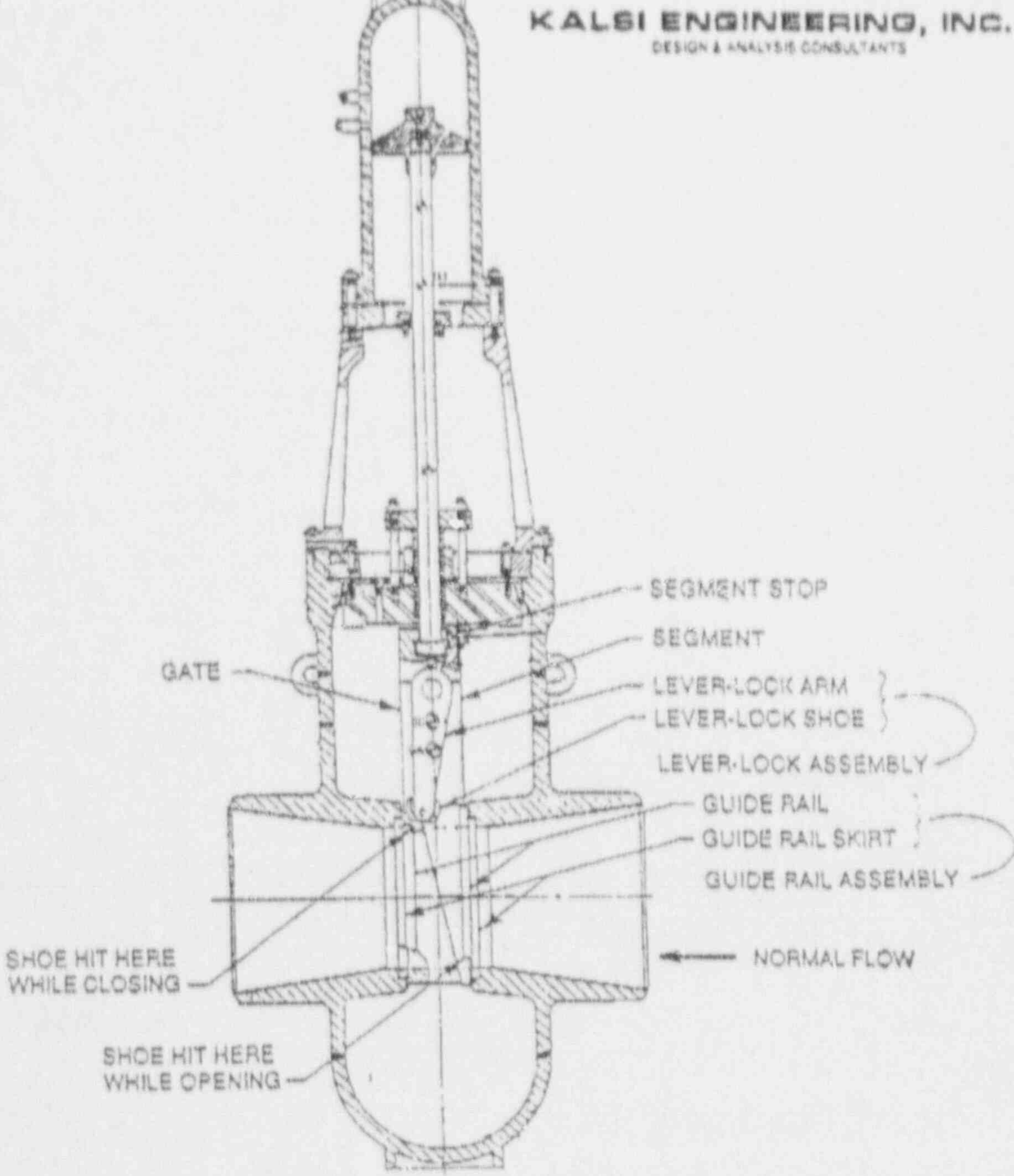

Figure 1

Nomenclature of Components Used in WKM MSIV Gate Galve

KALSI ENGINEERING, INC.
MECHANICAL DESIGN & ANALYSIS

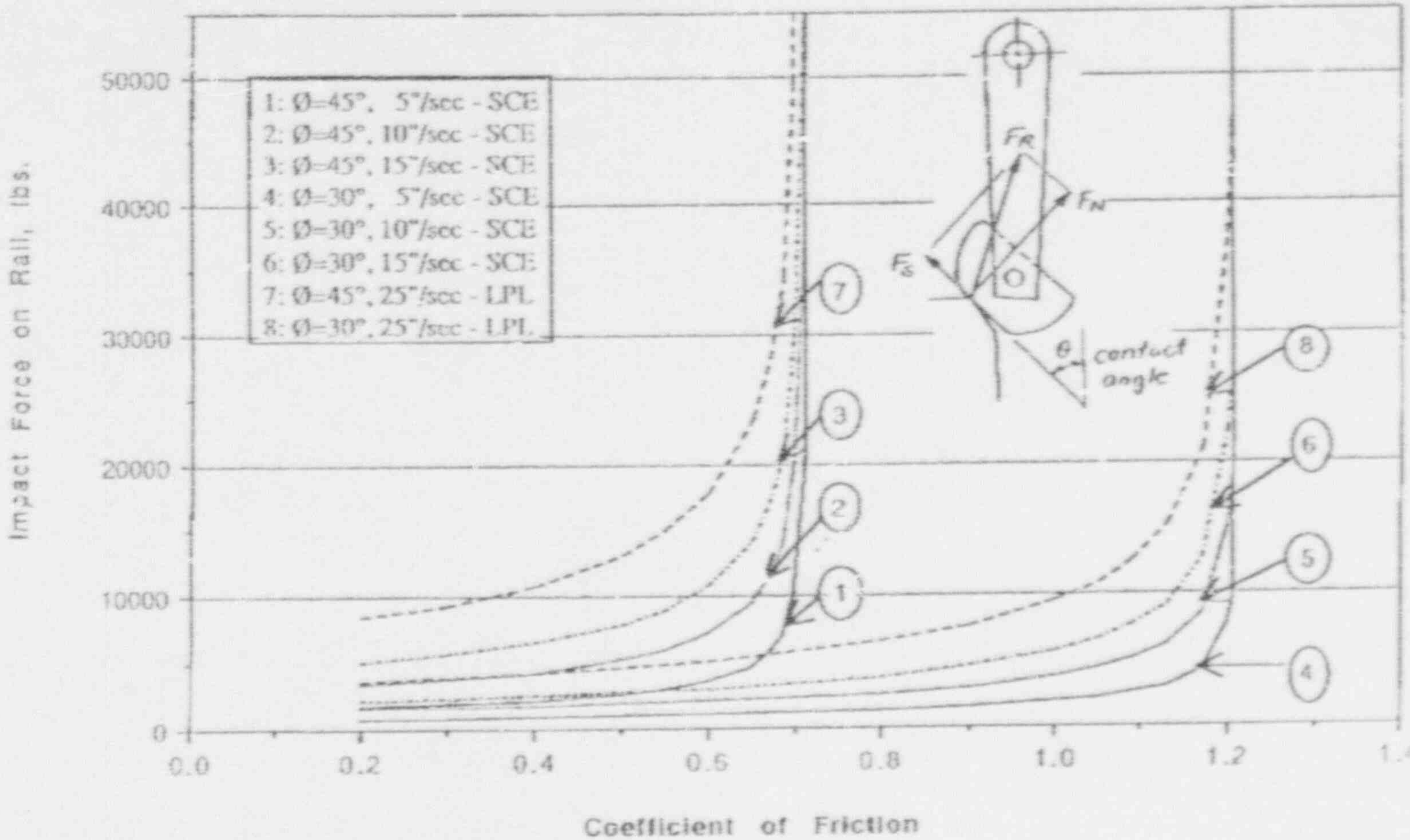

Figure 2
Comparing Impact Forces on Gate Rail for the Old and New Guide Designs

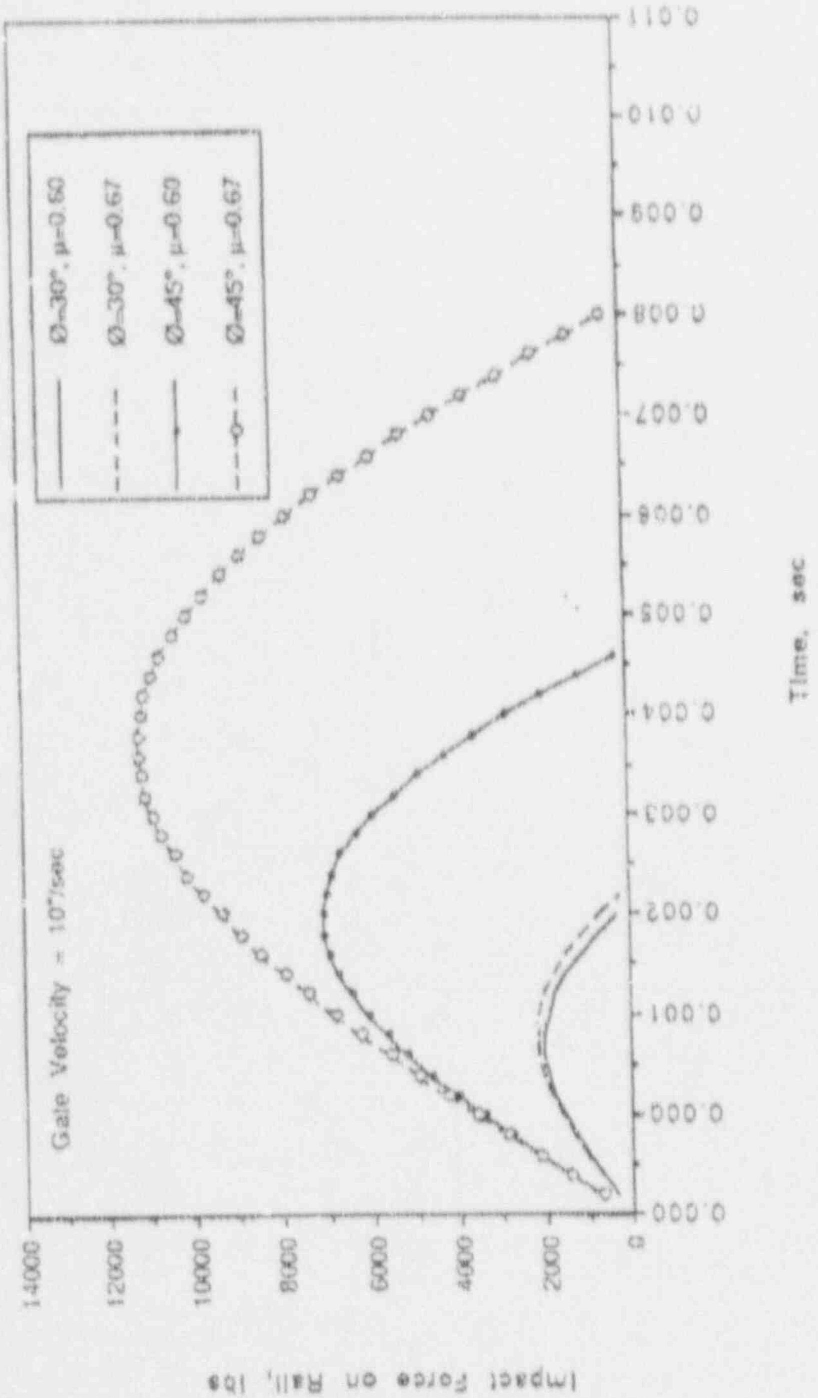

Figure 3
Time History Impact Force Plot for New and Old Rail Designs
(Coefficient of Friction = 0.6 and 0.67)

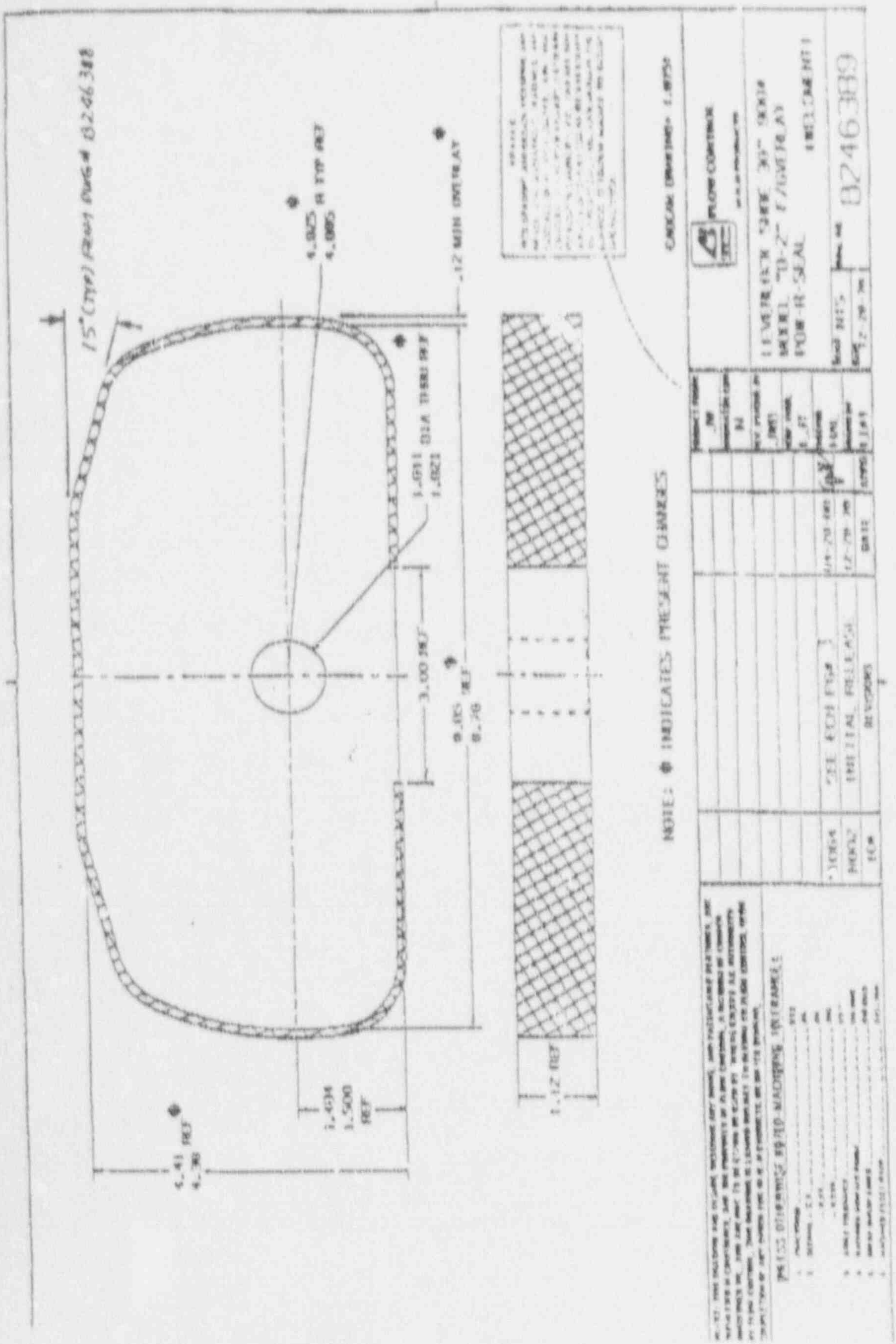

Figure 4
New Lever-Lock Shoe Design with Stellite Overlay

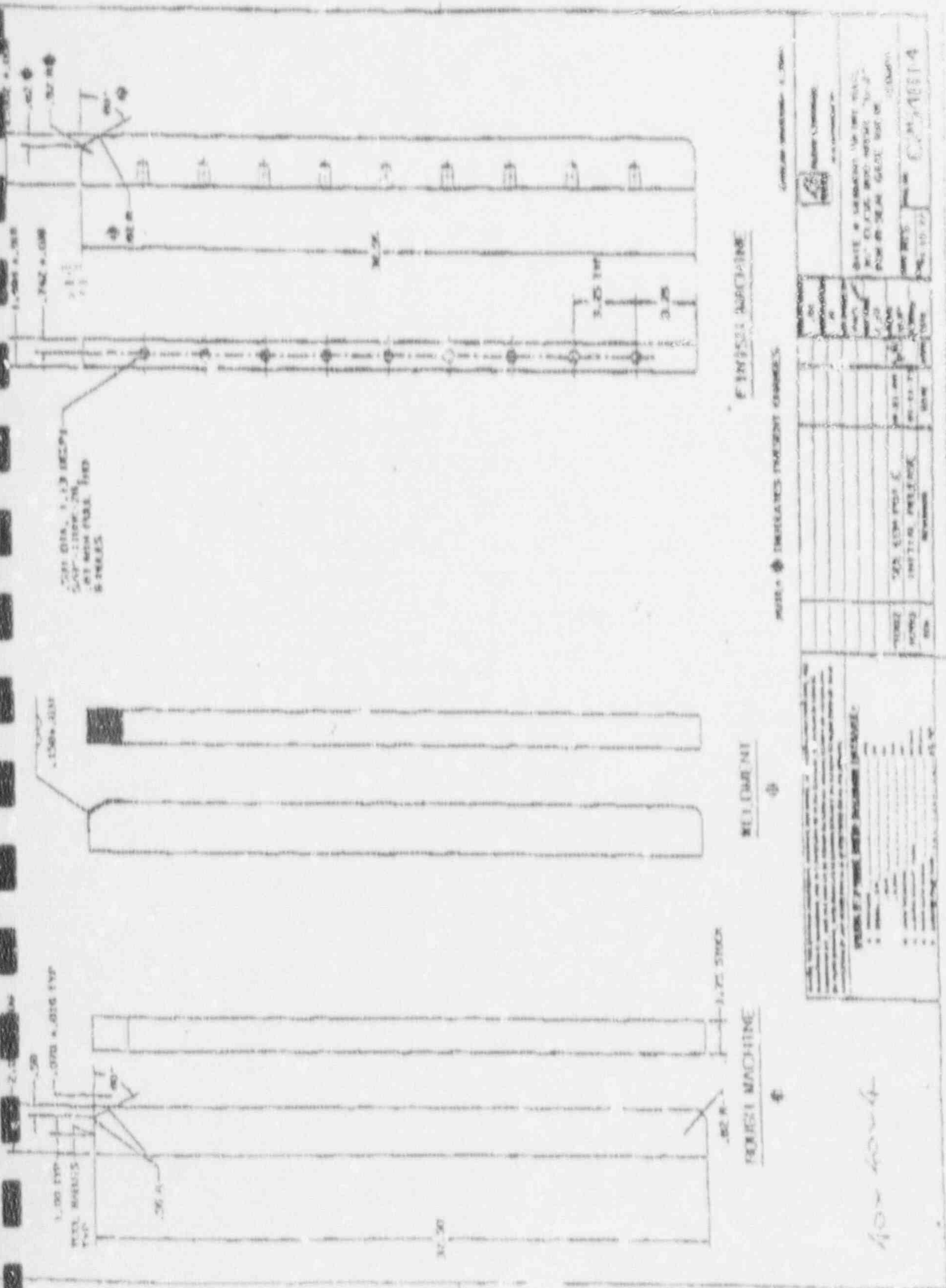

Figure 5
New Guide Rail Design with Stellite Overlay

REFERENCES

1. **Friction and Wear of Materials** by E. Rabinowicz, John Wiley & Sons, 1965.

2. *Root Cause Analysis of MSIV Gate Guide Failure*, Kalsi Engineering report to Louisiana Power & Light Company, KEI Document No. 1560C, Rev. 1, June 1988.

Appendix A
Transient Dynamic Impact Analysis During Gate Closing Action

Appendix A
Transient Dynamic Impact Analysis
During Gate Closing Action

In order to take into account the significant factors that affect closing impact load, a simplified math model was developed based on the closing mechanism, relative stiffness, and some assumptions. The significant factors that affect closing impact loads are the contact angle, coefficient of friction during impact, valve closing speed, segment weight, and lever-lock arm geometry and stiffness. The math model was developed to enable us to simulate the closing impact from initial contact of the shoe and rail till their separation. It provided us a way to analyze the contribution of these significant factors and their performance ranges. The results of this analysis enabled us to identify the current design deficiency and assure the performance of the proposed new design.

A.1. CLOSING IMPACT ANALYSIS

During early valve closing, the bottom of the lever-lock shoe contacts the rail chamfer. The resultant reaction force from the rail normally provides a necessary torque to push the lever-lock assembly to the center of the guide rails. This mechanism is designed to prevent gate and segment wedging during closing.

The closing impact analysis was developed to calculate the transient impact load using incremental time steps to simulate the impact action from initial contact to finish. Velocity, force, torque, and acceleration relations were developed based on some simplified assumptions to provide a reasonable size of the math model.

The following conservative assumptions were used in developing the math model:

1. The lever-lock arm is the only flexible member under impact load (compared with the other gate and segment components). If additional flexibility exists in the assembly, it will make the system softer and the impact load lower.
2. The gate and segment assembly travels at a constant speed before and after the impact load (unless a very high impact load occurs, the closing down speed will not be significantly changed from the assumption).

3. Segment weight is the only weight used in the lever-lock arm acceleration calculations. The lever-lock arm weight, which is relatively small compared with the segment weight, is neglected. Impact simulation is performed by the following steps:

   a. Select a small time step.

   b. Calculate the downward travel and lateral deflection.

   c. Calculate the force components, resultant force, and torque due to this added travel and deflection.

   d. Calculate the acceleration, velocity, and displacement of the segment.

   e. Modify the lateral deflection due to segment movement and go to the next time step.

A.2. EQUATIONS FOR THE IMPACT ANALYSIS

The following mathematical equations were derived for the impact simulations:

1. Lever-lock arm flexibility

   The lever-lock arm flexibility is calculated based on the following simplified math model:

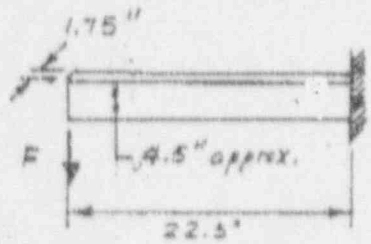

Figure A.1
Lever-Lock Arm Math Model

$$\text{Deflection} = \frac{F\ell^3}{3EI}$$

where
$\ell = 22.5"$
$E = 3E7$
$I = \frac{1.75 \times 4.5^3}{12}$

$\therefore$ Deflection $= 9.524 \times 10^{-6}$ F ←

2. Vertical and lateral displacements at the contact chamfer

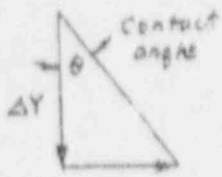

Figure A.2
Shoe Displacement

$$\Delta Y = V \times \Delta t$$

where V = closing velocity, in/sec

Δt = time increment, sec

$$\Delta X = \tan\theta\,\Delta Y + \Delta X'$$

where θ = contact angle

$\Delta X'$ = ΔX movement due to segment ypward movement (as derived below)

3. Force and torque relations

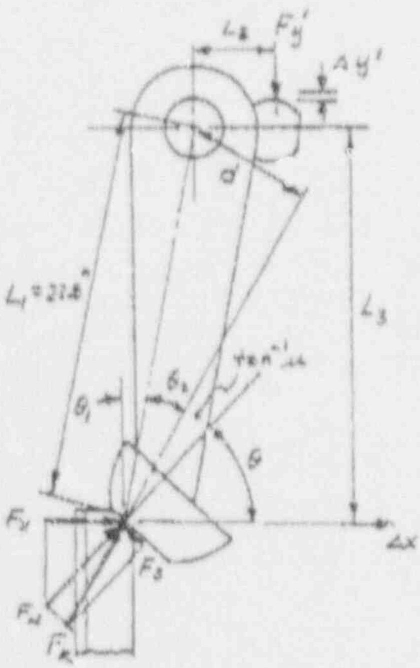

Figure A.3
Lever-Lock Arm Force and Torque Relationships

Lateral force increment for a given lateral deflection increment:

$$\Delta F_X = \frac{\Delta X}{9.524 \times 10^{-6}}$$

Normal force increment:

$$\Delta F_N = \frac{\Delta F_X}{\cos\theta}$$

Resultant force increment:

$$\Delta F_R = \frac{\Delta F_N}{\cos\left(\tan^{-1}\mu\right)}$$

where μ = coefficient of friction at the contact surfaces

Moment arm distance, d:

$$d = L_1 \sin\theta_2 = L_1 \sin\left(\frac{\pi}{2} - \theta_1 - \tan^{-1}\mu - \theta\right)$$

Vertical displacement of lever-lock arm's ear, $\Delta Y'$, due to lateral deflection, ΔX:

$$\Delta Y' = \frac{L_2}{L_3}\,\Delta X = \frac{2.63}{22.5}\,\Delta X$$

Torque increment, Δ Torque:

$$\Delta \text{Torque} = \Delta F_R \times d$$

Force increment at ear, $\Delta F_Y'$:

$$\Delta F_Y' = \frac{\Delta \text{Torque}}{L_2}$$

Acceleration increment for the segment:

$$\Delta \text{Acc} = \frac{\Delta F_Y'}{m}$$

where m = segment mass = 2,500/386.4

Segment vertical displacement for Δt increment:

$$\Delta Y' = V_o \Delta t + 1/2\ a_o \Delta t^2$$

where V_o = Previous iteration's velocity, in/sec
a_o = Previous iteration's acceleration, in/sec^2

Segment vertical velocity change for Δt increment:

$$V = V_o + a_o \Delta t$$

Undeflected ΔX movement due to ΔY' movement, ΔX':

$$\Delta X' = \frac{L_3}{L_2} \Delta Y' = \frac{22.5}{2.63} \Delta Y'$$

Based on the above derived relationships, the parametric study of these significant factors can easily be performed using a FORTRAN computer program. The results of this study were plotted as shown in Figures 2 and 3.

Figure 2 shows the overall trend of the lever-lock assembly performance as functions of the gate closing velocity, the chamfer angle, and the coefficient of friction. The 45-degree angle represents the old design while the 30-degree angle is the new chamfer design. For the old design, as the coefficient of friction increases, the impact force increases slowly; but the impact force picks up quickly after the coefficient of friction exceeds 0.6. It reaches a maximum of 0.72 where no torque can be generated under this high coefficient of friction. At this point the actual impact force is beyond the capability of this math model to calculate. (It may be estimated from the bounding forces.) However, the trend clearly shows that high impact forces occur after the coefficient of friction exceeds 0.6 due to the fast-disappearing torque as the resultant force approaches the lever-lock axis (d = 0). For smooth steel-on-steel surfaces, the coefficient of friction is estimated to be in the 0.2 to 0.4 range [1]. For a galled surface, the coefficient of friction can quickly go beyond 0.6, resulting in high impact forces that cause severe guide rail and gate skirt damages.

The gate closing speed at the lever-lock shoe and guide rail contact is critical to the performance of the lever-lock mechanism. It not only affects the magnitude of the impact load (as shown in Figure 2 plots), but also causes the initiation of surface degradation/galling, as observed on the guide rail chamfers and the bottom of the lever-lock shoes. The gate closing speed at the shoe and guide rail contact is estimated to be in the range of 5 to 15 in/sec for SCE's MSIV design. The impact forces, as calculated from the math model, are in the range of 2,126 to 6,378 pounds before contact surface degradation (μ = 0.4 used). The impact force for the SCE design is significantly lower than the estimated impact force of 10,630 pounds for the LPL design with the maximum closing speed of 25 in/sec, as shown in Figure 2 comparison [2]. However, as the coefficient of friction increases (contact surface degradation), both SCE and LPL designs approach an asymptotic rise in the magnitude of the impact force delivered to the guide rails at μ = 0.72 regardless of the gate closing speed.

The new shoe and guide rail designs, as shown in Figures 4 and 5, use a shallower contact angle of 30 degrees (measured from the stem axis). This new design angle can more effectively generate high lever-lock arm torque to push the shoe toward the center of the guide rails. Figure 2 shows that the impact force for the new design is in the range of 825 to 2,474 pounds (same as LP&L's modified shoe and guide rail angle design), which is only 38.8 percent of original design impact forces with 45 degrees contact angle.

It should be noted that the normal contact force at the shoe/guide rail interface is perpendicular to the bottom shoe surface, as shown in the force diagram, while the friction force is parallel to the shoe surface. Therefore, the shoe bottom surface angle instead of the guide rail chamfer angle will determine the effectiveness of the shoe-to-guide rail design in diverting the contact force to the rotating torque for the lever-lock arm.

The new shoe and guide rail designs also provide significant improvement in contact surface galling resistance with Stellite overlay on both shoe and chamfer surfaces and with nitro-hardened surface on the shoes, as shown in Figures 4 and 5. The rail chamfer angle is also 30 degrees (from stem axis) to match the shoe angle to provide a large flat surface contact area. Generous corner radii are provided in both lever-lock shoes (R = 1") and guide rails (R = 5/8") to keep the contact surfaces from degrading. Based on the math model calculations, the new design contact force will remain low until the apparent coefficient of friction reaches the extremely high value of 1.2. Therefore, the new design has sufficient safety margin to avoid damage to the contact surfaces after repeated closing.

Figure 3 shows the time history plot of the impact forces for the old and new designs. As the coefficient of friction changes from 0.6 to 0.67, the old 45-degree angle design increases the impact force significantly from 7,050 to 11,192 pounds while the new design keeps the impact force at less than 2,160 pounds of impact force (for closing speed of 10 in/sec).

Appendix B
Analysis of Forces and Stresses
During Gate Opening Action

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During Gate Opening Action

B.1. VALVE OPENING FORCE ESTIMATION

During gate opening, the gate speed is slow. Therefore, the load on the segment rails can be treated as a static load. The available force from the hydraulic pressure can be estimated as:

$$
\begin{aligned}
F_{net} &= \text{net upward force} \\
&= \text{hydraulic force} - \text{nitrogen force} \\
&\quad - \text{actuator piston drag} \\
&\quad - \text{actuator packing drag} \\
&\quad - \text{valve packing drag} \\
&\quad - \text{moving component weight} \\
&= 2{,}750 \times 0.7854\,(24^2 - 4^2) - 1{,}150 \times 0.7854 \times 24^2 \\
&\quad - 15{,}922 \\
&\quad - 12{,}983 \\
&\quad - 5{,}369 \\
&\quad - 6{,}124 \\
&= 648{,}869 \text{ lbs}
\end{aligned}
$$

Therefore, the actuator can provide sufficient force to overcome the segment rail resistance (shearing the bolts) if the shoe and rail are stuck together.

B.2. EFFECT OF FRICTIONAL COEFFICIENTS

During valve opening, the stem slowly pulls the gate upward. If the gate and segment are already in a collapsed position, then the lever-lock arm will stay between the gate and segment rails; i.e., no significant contact force is applied to the segment rails. For the gate and segment to not collapse, and remain wedged during opening, the coefficients of friction at seat surface and segment back angle have to be significantly different. (Note: This is an undesirable mode of operation.) The force equilibrium relationships for opening direction can be expressed as:

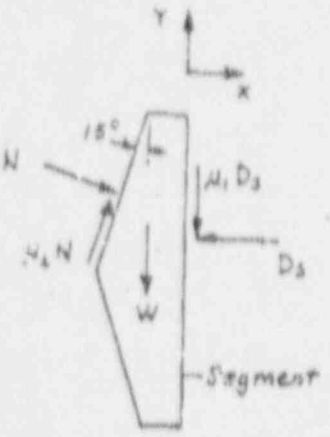

Figure B.1
Segment Free Body Diagram

$$\Sigma F_X = 0$$
$$N \cos 15° + \mu_2 N \sin 15° = D_S$$
$$\Sigma F_Y = 0$$
$$\mu_2 N \cos 15° - N \sin 15° - W - \mu_1 D_S = 0$$
$$\therefore \quad N = \frac{D_S}{(\cos 15° + \mu_2 \sin 15°)}$$
$$\therefore \quad D_S \frac{(\mu_2 \cos 15° - \sin 15°)}{(\cos 15° + \mu_2 \sin 15°)} = W + \mu_1 D_S$$

If DS >> W, then

$$\mu_{2\,min} = \frac{\mu_1 \cos 15° + \sin 15°}{\cos 15° - \mu_1 \sin 15°}$$

Therefore, we have the following relations for causing the undesirable mode of gate/segment wedging while opening:

μ_1	$\mu_{2\,min}$
0.20	0.494
0.25	0.555
0.30	0.618
0.35	0.682
0.40	0.748
0.45	0.816
0.50	0.887

The calculated μ_2 is the minimum coefficient of friction required to move gate and segment upward together in a wedged (expanded) position with segment weight assumed to be small compared with the seat contact force, D_S. The above calculated μ_1 and μ_2 indicate that the differences between μ_1 and μ_2 are higher than the usual values for seat and back angle surfaces that are in good condition (example: μ_1 = 0.25 and μ_2 = 0.4 are commonly used in design calculations). Therefore, in an ideal design condition where both contact surfaces are in good condition, the gate and segment should be in a collapsed condition with very little load applied to the segment rails. If the back angle surfaces deteriorate faster than the seat surfaces to reach the above tabulated ranges, then the wedged upward movement can occur, which may result in high contact load at the rails (if a high coefficient of friction also exists at the segment rail chamfers).

Another possible shoe and segment rail chamfer contact may occur during the final stage of a valve closing stroke when the gate and segment are wedging to make upstream and downstream seat contacts and the lever-lock arm is swinging toward the segment rail. For a normal gate and segment assembly, the lever-lock shoe should go under the segment rail chamfer without contacting it. Should dimensional tolerances stack-up outside the design limits (a short lever-lock arm and/or a long segment rail), then shoe and segment rail chamfer contact may occur while closing. The contact load under this condition may be high due to the insufficient lever-lock arm swing that prevents complete wedging and expanding of the gate and segment. Shoe and rail chamfer surface damage may result in high apparent coefficient of friction in the subsequent opening stroke.

The inspection of 2HV-8204 MSIV showed this valve to be free from this potential problem. Whenever disassembly of any MSIV is done, this area should be examined to ensure proper operation.

B.3. LOAD ON SEGMENT RAIL

Section B.1 discusses the condition that can stick the lever-lock arm shoe at the segment rail chamfer. In this section, we assume that the shoe sticks at the chamfer and estimate the maximum load on the rail.

Assume as the worst case that F_R passes through the lever-lock axis. If all the load-carrying components are strong enough, then the load applied to both rails can reach 649 kips, as calculated in Section B.1. From the design layout, we have the required coefficient of friction as:

$$\mu = \tan\theta$$

$$\theta = 180 - 90 - 38 = 52^\circ$$

$$\therefore \quad \mu = 1.28$$

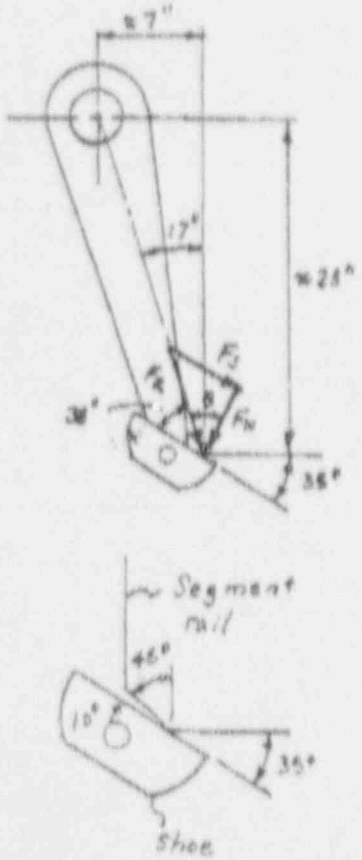

Figure B.2
Load on Segment Rail

It is not possible for this high coefficient of friction to occur on usual contact surfaces even with some galling. However, the actual contact, based on the old MSIV layout, occurs at the lower corner of the rail chamfer as shown in Figure B.2.

The indentation of the lower rail chamfer corner on the top shoe surface may cause the initiation of material galling, especially for the old 17-4 PH material without surface hardening on both sides. The apparent coefficient of friction under galling and corner indentation may be increased to a level that is close to the above calculated value.

The new shoe and segment rail designs, as shown in Figures 4 and 5, provide significant improvement to contact surface galling resistance with Stellite overlay on both shoe and rail contact surfaces. The rail chamfer corner radius changes from an originally sharp corner to a generous 5/8-inch radius to avoid surface degradation. The shoe surface is also nitro-hardened to provide additional galling resistance.

To assure the MSIV opening stroke operated within the design range, two areas are recommended for inspection after the valve is disassembled:

1. The back angles of the gate and segment contact surfaces should be inspected to assure that they are in good condition. Therefore, the coefficient of friction on the back angles will not be exceedingly higher than the seat surfaces, as discussed in Section B.2.

2. After shoe and guide rail modifications, the gate and segment assembly, including the segment skirt, should be inspected to assure that the top lever-lock shoe surface is not contacting the segment rail chamfer during the final wedging action of the gate and segment in a closing stroke.

APPENDIX C

VERIFICATION OF COMPUTER PROGRAM for MSIV GATE GUIDE IMPACT LOAD ANALYSIS

CLIENT: SCE
PROJECT: ANALYSIS FOR DESIGN MODIFICATIONS OF MSIV GATE GUIDE
PREPARED BY: J.K. WANG DATE: 10-23-81 PROJ NO.: 825
REVIEWED BY: M.S. KALSI DATE: 10-26-81 QA ______ DATE: ______

Based on the math model and equations derived in Appendix A of this report, a simple FORTRAN computer program was developed to perform the numerical calculations for the MSIV gate guide impact load analysis. This appendix provides the computer program validation/verification by comparing the analysis results from both the hand calculations and computer output using a typical case as plotted in Figure 3 of the main text.

C.1 TEST CASE DATA

K = lever-lock arm flexibility (one side) = $\frac{1}{9.524 \times 10^{-6}}$ = 104,998 lb/in

Δt = time increment = 0.0002 sec

θ = Shoe and rail contact angle = 30° (new design case)

θ_1 = contact point to lever-lock arm axis angle (Fig. A.2) = 9.5°

L_1 = Lever-lock arm length = 22.8 in.

L_2 = distance from lever-lock arm axis to its ear contact pt. = 2.63 in

L_3 = Vertical lever-lock arm length = 22.5 in

μ = coefficient of friction = 0.6

V = gate velocity at impact = 10 in/sec

w = segment weight = 2500 lb

C.2 SIMPLIFIED EQUATIONS FOR THE TEST CASE

Based on the equations derived in Appendix A, they can be simplified for the specific test case for the convenience of hand calculations.

Y = Shoe position in Y-axis (stem axis) $= Y_0 + V \times \Delta t$

$= Y_0 + 0.002$ (in)

where Y_0 = previous time step position

ΔX = net lever-lock arm deflection $= \tan\beta\ \Delta Y - \Delta X'$

$= 0.0011547 - \Delta X'$ (in)

where $\Delta X'$ = ΔX movement due to segment upward movement

ΔF_R = resultant impact force (2 sides) $= 2 \times \dfrac{\Delta F_N}{\cos(\tan^{-1}\mu)}$

$= 2 \times \dfrac{\Delta F_X}{\cos\theta\ \cos(\tan^{-1}\mu)}$

$= 2 \times \dfrac{\Delta X}{9.524 \times 10^{-6} \cos\theta \cos(\tan^{-1}\mu)}$

$= 282780\ \Delta X$ (lb)

ΔTorque = Torque on lever-lock arm $= \Delta F_R \times d$

$= \Delta F_R \times L_1 \sin(\frac{\pi}{2} - \theta_1 - \tan^{-1}\mu - \theta)$

$= 0.635366\ \Delta F_R$ (ft-lb)

$\Delta F_y'$ = Force at lever-lock arm ear $= \dfrac{\Delta \text{Torque}}{L_2}$

$= 4.56274\ \Delta$Torque (lb)

ΔAcc = Segment acceleration $= \frac{\Delta F_y'}{m}$

$= 0.15436\, \Delta F_y'$ (in/sec²)

$\Delta Y'$ = segment vertical displacement for Δt increment

$= V_0 \Delta t + \frac{1}{2} a_0 \Delta t^2$

$= 0.0002 V_0 + 2 \times 10^{-8} a_0$ (in)

where V_0 = previous iteration's velocity, in/sec

a_0 = previous iteration's acceleration, in/sec²

$\Delta X' = \Delta X$ movement due to segment upward movement

$= \frac{L_3}{L_2} \Delta Y' = 8.555133\, \Delta Y'$ (in)

V = Segment velocity

$= V_0 + a_0 \Delta t$

$= V_0 + 0.0002\, a_0$ (in/sec)

C3 COMPUTER OUTPUT VERIFICATION

The following table compares the computer output with the hand calculations to verify the accuracy of the developed FORTRAN program.

KALSI ENGINEERING, INC.

DESIGN & ANALYSIS CONSULTANTS

DOC. NO. ______
REV. NO. ______
PAGE 56 OF 56

CLIENT: SCE

PROJECT: ANALYSIS FOR DESIGN MODIFICATIONS OF MSIV GATE GUIDE

PREPARED BY: J. K. WANG DATE 10-22-91 PROJ. NO. 3.25

REVIEWED BY: M. S. KALSI DATE 12-24-91 QA ______ DATE ______

Time, Sec.	Computer/ hand cal.	Y (in)	ΔX (in)	ΔFR (lb)	ΔTorq (ft-lb)	ΔFy (lb)	ΔAcc (in/sec²)	ΔY' (in)	ΔX' (in)	V (in/sec)
0.0002	Comp	0.002	0.001155	327	208	947	146	0	0	0
	hand	0.002	0.001155	327	207	947	146	0	0	0
0.0004	Comp.	0.004	0.002309	653	415	1893	293	2.926×10^{-6}	2.503×10^{-5}	0.02926
	hand	0.004	0.002309	653	415	1893	293	2.926×10^{-6}	2.503×10^{-5}	0.02926
0.0006	Comp	0.006	0.003439	973	618	2819	436	1.170×10^{-5}	1.001×10^{-4}	0.08778
	hand	0.006	0.003439	973	618	2819	436	1.170×10^{-5}	1.001×10^{-4}	0.08778
0.0008	Comp	0.008	0.004494	1271	807	3684	569	2.627×10^{-5}	2.248×10^{-4}	0.1749
	hand	0.008	0.004494	1271	807	3684	569	2.627×10^{-5}	2.248×10^{-4}	0.1749
0.0010	Comp	0.010	0.005424	1534	975	4446	687	4.638×10^{-5}	3.967×10^{-4}	0.2888
	hand	0.010	0.005424	1534	975	4446	687	4.637×10^{-5}	3.967×10^{-4}	0.2888
0.0012	Comp.	0.012	0.006182	1748	1111	5066	783	7.151×10^{-5}	6.118×10^{-4}	0.4263
	hand	0.012	0.006182	1748	1111	5066	783	7.151×10^{-5}	6.117×10^{-4}	0.4263
0.0014	Comp.	0.014	0.006724	1902	1208	5513	852	1.009×10^{-4}	8.634×10^{-4}	0.5829
	hand	0.014	0.006725	1902	1208	5513	852	1.009×10^{-4}	8.633×10^{-4}	0.5829
0.0016	Comp	0.016	0.007016	1984	1261	5752	889	1.336×10^{-4}	1.143×10^{-3}	0.7533
	hand	0.016	0.007016	1984	1261	5752	889	1.336×10^{-4}	1.143×10^{-3}	0.7533
0.0018	Comp.	0.018	0.007027	1987	1263	5761	890	1.684×10^{-4}	1.441×10^{-3}	0.9311
	hand	0.018	0.007028	1987	1263	5761	890	1.684×10^{-4}	1.441×10^{-3}	0.9311
0.0020	Comp.	0.020	0.006741	1906	1211	5526	854	2.040×10^{-4}	1.746×10^{-3}	1.109
	hand	0.020	0.006741	1906	1211	5526	854	2.040×10^{-4}	1.746×10^{-3}	1.109
0.0022	Comp.	0.022	0.006150	1739	1105	5042	779	2.389×10^{-4}	2.044×10^{-3}	1.280
	hand	0.022	0.006150	1739	1105	5042	779	2.389×10^{-4}	2.044×10^{-3}	1.280
0.0024	Comp.	0.024	0.005261	1488	945	4313	667	2.716×10^{-4}	2.323×10^{-3}	1.436
	hand	0.024	0.005261	1488	945	4313	667	2.716×10^{-4}	2.324×10^{-3}	1.436
0.0026	Comp.	0.026	0.004092	1157	735	3355	519	3.005×10^{-4}	2.571×10^{-3}	1.569
	hand.	0.026	0.004092	1157	735	3355	519	3.005×10^{-4}	2.571×10^{-3}	1.569
0.0028	Comp.	0.028	0.002676	757	481	2194	339	3.242×10^{-4}	2.774×10^{-3}	1.673
	hand	0.028	0.002676	757	481	2194	339	3.242×10^{-4}	2.774×10^{-3}	1.673
0.0030	Comp.	0.030	.001057	299	190	867	134	3.414×10^{-4}	2.920×10^{-3}	1.741
	hand	0.030	.001057	299	190	867	134	3.414×10^{-4}	2.920×10^{-3}	1.741

The above table shows that the results from both computer output and hand calculation are identical except a few data points with the fourth digit round-off error from hand calculations. Therefore, the accuracy of the program is verified.