DYNAMIC ANALYSIS OF STEAM ISOLATION VALVE FOR CLOSURE UNDER FAULTED CONDITIONS

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SUMMARY

The paper is concerned with an evaluation of the structural response of a swing-disk steam-isolation valve in the event of an emergency closure, such as postulated in conjunction with a pipe break in a main steam line of a nuclear power station. Described are the analytical approach, the results obtained from the dynamic analysis, and the acceptance criteria adopted for evaluating the results.

A Lagrangian finite-difference computer code is employed to obtain an axisymmetric simulation of the impact problem associated with the sudden closure of the valve disk on the valve seat. Major dimensions that reflect the size of the valve considered are a disk diameter of 27.5 in. (70 cm), a disk thickness of 3.75 in. (9.5 cm), a body seat diameter of 25.0 in. (63.5 cm), and a body minimum wall thickness of 2.2 in. (5.6 cm). Stress-strain diagrams assumed for the disk material (SA-182, Type 304 austenitic steel) and the body material (SA-216, Type WCB ferritic steel) are bilinear, with an upper limit on stress. A solution is obtained for a disk impact velocity of 150 ft/s (46 m/s).

Pertinent analytical results cited in the paper are the following. The kinetic energy of $3.0 \times 10^6$ in. lb. ($3.4 \times 10^6$ J) is dissipated in 0.0013 s. Following impact, the center of the disk speeds up 20 percent before it slows down. The center deflection reaches a magnitude equivalent to half the disk thickness. Strain in the central portion of the disk is developed much later in the impact event than in the contact region of the disk and body. Effective strains as high as 15 percent, 17 percent, and 14 percent, are produced in the body seat region, the rim region of the disk, and the central region of the disk, respectively.

The gross geometric distortion and the large strains are considered acceptable because the distortion does not impede the proper functioning of the valve and the strains do not exceed specified strain criteria. These criteria are expressed in terms of the percent uniform elongation $e_u$ (true strain at maximum load in tension test) and the fracture strain $e_f$ (true strain derived from reduction of area at fracture location in tension specimen). The basis for the criteria is discussed in the paper.
I. INTRODUCTION

With reference to Section III on nuclear power plant components of the ASME Boiler and Pressure Vessel Code [1], faulted conditions constitute the most severe category of loading conditions to be considered for the safe design of such components. They pertain to postulated failure events, and combinations thereof, which are of extremely low probability. Engineered safety features of a nuclear power plant must be designed to withstand these events to ensure that public health and safety are not compromised by failure of a critical component in such a plant. It is important to stress here that faulted condition design procedures are not intended to assure operability of the system, either during or following the postulated event. Rather, they are intended to limit the consequences of the event by maintaining structural integrity and thereby assure a safe shutdown of the plant.

A type of failure in the faulted-condition category that has received much attention is the postulated guillotine pipe break in a main steam line and the potential pipe-whip problem connected with it. Analysis has shown that large amounts of stored energy may be released when such a break occurs and that, consequently, large forces may be transmitted to system components, unless the energy is dissipated in a safe manner. An effective way for coping with the problem is to absorb the energy through plastic deformation in the pipe-whip restraints. Restraints designed for such purpose were described in recent papers by Esswein [2] and by Palusamy, Patrick and Cloud [3].

Protection against pipe whip is just one major safety consideration toward limiting the consequences of a pipe break accident. Another is the isolation of the break from the steam supply system by means of steam isolation and check valves. These are often swing-disk valves, with the isolation valve stopping the flow of steam when a break occurs downstream from the valve, and the check valve preventing reversed flow in the event of a break upstream. Swing disk valves have the advantages of rapid closure, simplicity and reliability of operation, and low flow resistance. However, the rapid closure results in a high impact velocity for the disk. This is evident from recent papers by Gwinn [3] and by Leonard and O'Leary [5].

The rapid closure of the valve is again a problem of energy dissipation. That is, kinetic energy stored in the disk at the time of impact must be dissipated, for the most part, by plastic strain energy absorption in the valve components. In fact, the kinetic energy is of such magnitude that the valve must be designed with an outlet for this energy in mind. The geometry of disk and body must be shaped to allow plastic deformation of the structure without impedance to valve closure. Clearly, the use of ductile materials is essential.

This paper gives a brief account of an analytical investigation of the dynamic response of a representative main-steam isolation valve subjected to the rapid closure triggered by a pipe break. The complete study had three parts: (1) a fluid dynamics analysis to determine the disk impact velocity, (2) a structural evaluation of the disk and valve body for impact loading, and (3) a structural evaluation of the disk support linkage. The linkage must be able to withstand the large inertia and centrifugal loads just prior to impact, as well as the impact loading. Only the second part of the study will be covered in the paper.

A sketch of the valve that was analyzed is depicted in Fig. 1. Its size is typified by a body seat opening of 25 in. (63.5 cm) in diameter. When the valve is open, the flow is from left to right. The disk is held in the fully-open position by an air-actuated cylinder. In the event of a pipe break, a signal will release the air and thereby trip the disk. Then, the
actuator spring force and gravity action will push the disk into the steam flow. Once the disk has entered the flow, a pressure differential across the disk will accelerate the disk motion until impact. The fluid dynamics analysis furnished the impact velocity as 104 rad/s and the kinetic energy in the disk at impact as $3.0 \times 10^5$ in. lb. ($3.4 \times 10^5$ J).

A dynamic solution for the impact problem was acquired with the aid of the PISCES computer program [6]. This Lagrangian finite-difference code permits the dynamic response of a structure to be analyzed as a function of time, under conditions of large plastic strain and gross geometric distortion. The solution proceeds in small incremental time steps, with a geometry update at each step. The theoretical basis for the program was laid by Wilkins [7]. A summary of the theory is provided in the PISCES manuals. The larger computer service centers provide access to the program. A computer code with a similar theoretical basis was described by Chang, et al. [8]. This is the REXCO-HEP code specially developed for analyzing the dynamic response of primary reactor containment systems to reactor excursions.

The PISCES code permits the solution of one- and two-dimensional problems, the axisymmetric problem included. The latter option was used to obtain an axisymmetric simulation of the valve impact problem. The axisymmetric approach provides a reasonable approximation of the three-dimensional problem, for two reasons. First, the body is much more massive than the disk, so that significant straining of the body can be assumed to be confined to the immediate surroundings of the valve seat, an assumption borne out by the solution acquired. Second, whereas the swinging disk does not have a uniform velocity at the instant of impact, it can be assumed that the initially nonuniform contact load along the circumference will even out early in the impact event. Clearance between the disk and its support arm, provided for disk seating flexibility, promotes such a load redistribution. Therefore, the axisymmetric analysis should give a sufficiently accurate picture of the strain distribution toward the end of the impact event.

A final introductory comment regarding the PISCES code relates to the formulation of a yield model for the material. A standard option is available for the use of a bilinear stress-strain diagram with a limit on maximum stress. A nonstandard yield model may be appended to the code in the form of a user-developed subroutine. It would be possible, therefore, to introduce strain-rate dependence into the yield model. As will be shown, the stress-strain curves for the disk and body materials were not very sensitive to strain rate. For this reason, the authors opted for the PISCES-supplied bilinear yield model.

Emphasis in the paper is placed on results that characterize the findings of the investigation, and on the evaluation of these results. In the absence of established acceptance criteria in terms of specific strain limits, special criteria were adopted. These are presented, along with their justification.

2. IMPACT ANALYSIS

Dimensions of the axisymmetric model are defined in Figs. 2 and 3. The valve body consists of two cylinders joined by a transition region that contains the valve seat. As indicated in Fig. 2, the model measures 100 inches (254 cm) in each direction from the valve seat. The long body extensions were added for the purpose of delaying the return of elastic stress waves reflected at the model ends until after the impact activity in the impact region and in the disk had subsided. The shape of the transition region was patterned after the body seat configuration at the lower end of the valve shown in Fig. 1. Here, the body seat opening is at
right angles to the seating surface, a feature representative of a significant portion of the valve body. As the results to be presented will confirm, precise modelling of the external surface in the transition region is not required by virtue of the concentration of strain toward the internal surface.

The finite-difference grid of the model is presented in Fig. 4. Only the central portion is shown. Just outside this portion, the number of zones across the body wall changes from six to two. Also, the length of the zones gradually increases to 10 inches (25.4 cm). The first five rows of zones adjacent to the centerline of the disk were given a higher mass density than that of the actual material. This was done to simulate the impact energy contribution imparted by the disk support mechanism. In the impact region, zones on opposite sides of the contact plane can slide relative to each other and they can also separate. The coefficient of friction was assumed to be zero for the analysis.

Properties selected for the disk material (SA-182, Type 304 austenitic steel) and the body material (SA-216, Type WC8 ferritic steel) were those available for a temperature of 600°F, a temperature slightly higher than under actual service conditions (approximately 550°F). Using the notation defined in the Appendix, the properties used for the disk were: $E = 25.4 \times 10^6$ psi = 1.75 x 10^5 MPa, $\nu = 0.3$, $\rho = 7.85$, $\sigma_y = 25.0 \times 10^3$ psi = 1.72 x 10^2 MPa, $\sigma_u = 80.0 \times 10^3$ psi = 5.52 x 10^2 MPa, $\varepsilon_y = 0.314 \times 10^6$ psi = 2.17 x 10^3 MPa. The properties for the body were: $E = 25.7 \times 10^6$ psi = 1.77 x 10^5 MPa, $\nu = 0.3$, $\rho = 7.85$, $\sigma_y = 30.0 \times 10^3$ psi = 2.07 x 10^2 MPa, $\sigma_u = 62.0 \times 10^3$ psi = 4.28 x 10^2 MPa, $\varepsilon_y = 0.320 \times 10^6$ psi = 2.21 x 10^3 MPa.

A comparison of the assumed true stress-strain diagrams with available experimental results is presented in Fig. 5 for the disk material* and in Fig. 6 for the body material. Figure 5 shows curves for different strain rates, as obtained with a formula provided in the LMFB Materials Handbook [5]. This formula, derived from experimental data reported by Steichen [10-12], pertains to strain rates in the range from 0.00001 s^{-1} to 100 s^{-1}. Some strain hardening with increased strain rate is noted. (The curve in Fig. 5 for a strain rate of 1000 s^{-1} is dashed to accentuate the fact that it was obtained by extrapolation beyond the applicable range of the formula.) The bilinear stress-strain diagram in Fig. 5 was constructed such that it approximates high-strain-rate behavior at large strain and low-strain-rate behavior at small strain. Because the yield strength at the low strain rate of 10^{-5} s^{-1} was less than the specified minimum value in Reference 1, no downward adjustment of the curve to account for minimum properties was necessary.

No published high-strain-rate test data were available for the body material. For this reason, tension tests were conducted on a blank of the casting steel. A dozen tests were made in three orthogonal directions, at nominal strain rates of 0.1, 1 and 10 s^{-1}. A set of curves obtained for one direction is shown in Fig. 6. Very similar results were obtained for the other directions. Slight softening with increase in strain rate was observed. The yield strength of the material, as determined from a conventional tension test, was above that specified as a minimum in Reference 1. Therefore, the bilinear stress-strain diagram was adjusted slightly downward.

Not modelled was a Stellite overlay on the contact surfaces of the body and disk, inclusion of which would have necessitated a finer grid. Treating the overlay as having the same

*Figure 5 is a plot of true stress versus true plastic strain. Total true strain is obtained by adding the elastic strain $\varepsilon_t/E$ to the plastic strain.
properties as the underlying base material was considered a justifiable approximation in that it would tend to increase local strain levels. This leads to a more conservative evaluation, when based on strain limits.

The dynamic solution was performed for a disk impacting the body with a velocity of 150 ft/s (46 m/s). This velocity was determined from the angular impact velocity of 104 rad/s by matching the kinetic energy of translation to the kinetic energy of rotation. Axial motion of the body was constrained at the upstream end of the model. No such constraint was introduced at the downstream end. An internal pressure of 705 psi (4.86 MPa) was assumed to exist on the upstream side of the model for the duration of the impact event.

A PISCES solution is started with a small specified initial time step. Subsequent time steps are generated within the program by means of a numerical stability criterion. The valve solution was started with an initial time step of 0.05 μs. A stable time step of 0.38 μs was reached in about 15 steps. The time-marching solution was executed for 3000 time steps, for a total elapsed time of 1150 μs. In addition to standard output provided by the PISCES program, extra output was generated by means of a specially written subroutine appended to the program (via the EXOUT option provided for such purpose). The extra output consisted of the true strain components \( e_{rr}, e_{θθ}, e_{zz}, e_{rz}, \) the effective true strain \( \bar{e} \) and the effective true stress \( \bar{σ} \). The definitions of \( \bar{e} \) and \( \bar{σ} \) are given in the Appendix.

3. RESULTS

It is conservative to assume that all the kinetic energy stored in the disk and its support linkage must upon impact be dissipated by plastic strain energy absorption in the disk and body. The rate of energy dissipation is shown in Fig. 7, where the kinetic energy in percent of its initial value is plotted as a function of time. It is seen that at solution termination time (\( t = 1150 \) μs), only 2.8 percent of the kinetic energy remained to be absorbed. Graphical extrapolation of the curve indicated that the duration of the impact event, defined as the end of plastic strain energy absorption, is approximately 1300 μs. Similar extrapolations of other results from 1150 μs to 1300 μs produced minor changes, so that continuation of the solution beyond 1150 μs was unnecessary.

Recalling that the impact velocity was 150 ft/s (46 m/s), it is of interest to note, from the results presented in Fig. 8, that when the disk motion is arrested at the rim, the velocity at the center of the disk initially increases. At the back surface, a peak velocity of 180 ft/s (55 m/s) is reached 350 μs after impact. A slightly lower value is reached at the front surface, which indicates compression of the disk due to inertia loading. The "whiplash" phenomenon reflected in Fig. 8 was also reported by Leonard and O'Leary [5].

Evidence of gross permanent distortion of the valve resulting from the impact loading is presented in Fig. 9, where the shape of the model before and after the impact event is shown. The center deflection of the disk is 1.77 in. (4.50 cm) at the front surface and 1.84 in. (4.67 cm) at the back surface, or almost equal to half the thickness of the disk. Such a large displacement necessitates a disk linkage design that will allow disk closure to take place without mechanical interference. Inspection of the deformed shape shows that the contact surfaces have undergone rotation and that the adjacent disk fillet has flattened out. It is also noted that the external radius of the disk contact surface increased a small amount and that the internal radius of the body contact surface decreased a small amount.
In Figs. 10 and 11, the distribution of the effective strain (\( \varepsilon \)) is shown as obtained at solution termination time (t = 1150 \( \mu s \)). Strain values have been rounded off to the nearest full percentage point. The three zones denoted A, B, and C indicate maximum values of 15 percent in the body, 17 percent in the rim region of the disk, and 14 percent in the central region of the disk. It is observed that the maximum strain in the contact region of both disk and body occurs at some distance from the contact surface, a phenomenon not uncommon to contact problems. It is noteworthy also that large strains in the body are confined to the immediate vicinity of the body seat. This result shows that precise modelling can be limited to the body seat region, with coarser modelling allowed for other parts of the body.

As apparent from Fig. 11, the largest strains in the central region of the disk occur at the centerline. The distribution of strain components along this line is shown in Fig. 12. At the forward surface, \( \varepsilon_{rr} = \varepsilon_{bb} = 8.5 \) percent, and \( \varepsilon_{zz} = -17 \) percent. These values yield for \( \varepsilon \) the peak value of 17 percent at the surface.

Figures 10 through 12 pertain to the distribution of strains at the end of the impact problem. It is also of interest, however, to examine how the strains at different locations accumulated with time. Strain accumulation for the three locations of maximum strain is shown in Fig. 13. It is apparent from this diagram that strain in the rim region of the disk is accumulated much faster than in the central region. At t = 300 \( \mu s \), strain accumulation is essentially complete at the rim, while strain accumulation at the center has barely begun. Strain accumulation in the body is somewhat slower than in the disk rim. To provide a yardstick for the rate of straining, two straight lines are drawn in Fig. 13 that correspond to linear strain accumulation at rates of 100 s\(^{-1}\) and 1000 s\(^{-1}\). The computer solution furnished maximum strain rates at A, B, and C of 370 s\(^{-1}\), 790 s\(^{-1}\), and 270 s\(^{-1}\), respectively.

4. ACCEPTANCE CRITERIA

To ensure that the valve will survive the dynamic loading resulting from a pipe break, it is necessary to establish that two conditions are satisfied. One is that the valve will perform its function of closing off the steam flow, despite geometric distortion that may require the valve to be repaired thereafter. The other is that the structural integrity of the valve components is preserved, so that the valve remains intact.

With reference to Appendix F in Section III of the ASME Boiler and Pressure Vessel Code [1], various approaches can be taken to the formulation of acceptance criteria. Listed among the approved procedures in conjunction with a plastic analysis is the use of a strain limit. This approach was adopted for the valve analysis in view of the large strains produced in the valve. In doing so, it was considered essential that the strain limit be expressed in terms of material ductility. The acceptance criteria that were used follow below, where \( \varepsilon_u \) is the percent uniform elongation (true strain at maximum load in tension test), \( \varepsilon_f \) is the fracture strain (true strain at fracture location in tension specimen), and \( \sigma_m = (\sigma_{rr} + \sigma_{bb} + \sigma_{zz})/3 \). (Stress \( \sigma_m \) is variously called the average stress, mean stress, spherical stress, hydrostatic stress, or octahedral normal stress.)

Criterion 1: Geometric distortion shall not impede proper closure and leaktightness of the valve.

Criterion 2: Where the average stress \( \sigma_m \) is tensile, local peak strains shall be controlled by limiting the magnitude of the true effective strain \( \varepsilon \) to the smaller of \( \varepsilon_u \) and \( \varepsilon_f / 3 \).
Criterion 3: Where the average stress $\sigma_m$ is compressive, local peak strains shall be controlled by limiting the magnitude of the true effective strain $\bar{\varepsilon}$ to the larger of $\varepsilon_u$ and $\varepsilon_f/3$.

Criterion 4: Gross strains, such as composed largely of membrane and bending components, shall be controlled by limiting the magnitude of the true effective strain $\bar{\varepsilon}$ to $0.6 \varepsilon_u$.

A brief discussion of the basis for these rules follows. It is obvious that the intent of Criterion 1 is to ensure, by inspection of the analytically determined deformed shape, that no path for unacceptable steam leakage will be created past the disk. This inspection also encompasses analytical verification that the disk support linkage will not interfere with proper disk action.

Criterion 2 provides protection against a crack originating at peak strain locations. The choice of limits was aided by the results of burst tests on special disk specimens, reported by Cooper, Kottcamp and Spiering [13], and subsequently interpreted analytically by Riccardelli [14]. In the tests, the axisymmetric specimens were rigidly clamped along the rim and subjected to pressure on one face. Three materials of different ductility were used. Depending on the material and the disk geometry, the disks failed in tension either at the rim fillet or in the center. The rim failures resulted from strain concentration, the center failures from strain instability. Applying Criterion 2 to the maximum effective strain in the rim fillet, it was ascertained that no rim failures would have occurred, had the tests been interrupted when the true strain reached the lesser of $\varepsilon_u$ and $\varepsilon_f/3$ of the materials involved.

Criterion 3 is less stringent than Criterion 2 because it is known that when $\sigma_m$ is compressive, materials have more ductility than when $\sigma_m$ is tensile [15,16]. Consequently, cracks have difficulty forming and propagating when the former stress condition prevails.

Criterion 4 is a conservative rule introduced to provide protection against collapse through excessive plastic deformation. Its formulation is based on the observation gleaned from the work of Cooper [17] that in an internally pressurized thin-walled sphere, the circumferential strain at the maximum pressure attainable in such a sphere will be $\varepsilon_u (e^{n/3}-1)/(e^n-1)$, or not less than approximately $0.3 \varepsilon_u$ for most steels, $n$ being the strain hardening exponent. By neglecting elastic strains as being very small, it follows that $\bar{\varepsilon}$ is equal to twice the circumferential strain. Therefore, limiting $\bar{\varepsilon}$ to $0.6 \varepsilon_u$ will prevent plastic instability in the sphere. Invoking Criterion 4 for the evaluation of gross strains produced at the center of the disk is a conservative procedure, for pressure loading of a thin-walled sphere produces only membrane strains, while the straining mode of the disk is primarily bending. In applying Criterion 4 to the aforementioned disk experiments, it was found that the rule would have been effective in preventing strain instability failures at the center of the disk specimens. That is, had the tests been stopped when the true strain at the center reached the value of $0.6 \varepsilon_u$, no failures would have occurred.

In order to apply the strain criteria, $\varepsilon_u$ and $\varepsilon_f$ must be available for the temperature of interest. The strain rate dependence of these ductility parameters must also be considered for the materials involved. To retain conservatism, minimum values of $\varepsilon_u$ and $\varepsilon_f$ must be used. For the problem at hand, $\varepsilon_u$ and $\varepsilon_f$ were 30 percent and 105 percent for the disk, 11 percent and 88 percent for the body. The values for the disk material were obtained from work of Steichen [10,11], those for the body material were determined as part of the tension test program conducted for the purpose of acquiring stress-strain diagrams, which was discussed earlier.
5. **EVALUATION**

Testing of the results of the dynamic solution against the acceptance criteria leads to the conclusion that the body and disk are both structurally adequate to withstand the severe dynamic loading caused by emergency closure of the valve due to a pipe break in the main steam system. This favorable conclusion was reached on the basis of the following findings:

(1) Deformation of the valve, while large, does not result in the creation of a steam path past the disk.

(2) In the highly-strained contact region, the peak value of \( \varepsilon \) is 15 percent in the body and 17 percent in the disk. Furthermore, \( \sigma_m \) remains compressive in the region during the impact event, so that the limit on \( \varepsilon \) is 29 percent for the body and 35 percent for the disk. These limits on \( \varepsilon \) are not exceeded.

(3) In the central region of the disk, the maximum value of \( \varepsilon \) is 17 percent. This is below the strain limit of 18 percent for gross strain.

6. **DISCUSSION**

From the foregoing description of the impact analysis, it is clear that the faulted condition was treated as an isolated event. In reality, such a condition may follow many operational cycles, with attendant fatigue damage accumulation. So the question may be raised as to whether the evaluation procedure is still valid if the faulted condition is postulated to occur near the end of life of the plant. To provide an answer to this question, it is useful to examine the residual ductility concept.

The residual ductility concept was reviewed by Ohji, Miller and Marin [18]. Several formulas for relating residual ductility to fatigue damage were tested against experimental data published by Sessler and Weiss [19] and by Weiss, Sessler and Packman [20]. The following formula was found to best fit the data \( \varepsilon_{fr} = \varepsilon_f (1 - D)^{1/2} \), where \( \varepsilon_f \) is the initial fracture ductility, \( \varepsilon_{fr} \) the residual fracture ductility, and \( D \) the cumulative fatigue damage \( D = \Delta n_i / N_i, 1 = 1, 2, 3, \ldots \). When \( D \) approaches 1, \( \varepsilon_{fr} \) goes to zero, thus indicating complete ductility exhaustion when the fatigue damage reaches its limiting value. However, in applying the formula to faulted conditions, it must be remembered that when the fatigue design curves were constructed for inclusion in the ASME Boiler and Pressure Vessel Code [1], a reduction factor of 20 was applied to the measured number of cycles. Therefore, the formula for the residual ductility must be adjusted accordingly to be meaningful, or \( \varepsilon_{fr} = \varepsilon_f (1 - 0.05 D')^{1/2} \), where \( D' \) is the cumulative fatigue damage obtained via the code fatigue curve. Then, when \( D' = 1 \), \( \varepsilon_{fr} = 0.975 \varepsilon_f \). Clearly, the difference between \( \varepsilon_f \) and \( \varepsilon_{fr} \) is now too small to be of practical significance when the conservative fatigue evaluation procedures of Reference 1 are followed. The authors believe that on the same grounds, any reduction of \( \varepsilon_u \) due to prior fatigue usage can also be dismissed as insignificant.

Another question that may be raised is in regard to possible effects of any cyclic strain hardening or softening caused by cyclic plasticity during normal operational cycles of the plant. Here, the authors must assume that neither the gross deformation nor the controlling strain levels are affected, simply because the strain amplitudes permitted within the conservative procedures of Reference 1 are small compared to the large strains experienced during a faulted condition.

In conclusion there is the question as to the engineering validity of replacing the dynamic solution by a less expensive quasi-static analysis in which the disk is monotonically
loaded by an equivalent static load until the plastic strain energy in the disk and body is equal to the kinetic energy to be dissipated. Leonard and O'Leary [5] investigated both methods of analysis and concluded in their paper that the quasi-static approach was satisfactory for the valve geometry considered in their investigation, but cautioned against a generalization of this conclusion for other valve geometries. The authors concur that until more valve geometries have been analyzed by both methods, it would be premature to conclude that a quasi-static analysis will generally be sufficient. For the present, the authors believe that a quasi-static approach will show more energy absorption in the central region of the disk and less in the contact region than with the dynamic approach. Therefore, strain magnitudes would tend to be computed too large at the center of the disk and too small in the rim region of the disk and the seat region of the body. Further studies will be required to determine how large these deviations may be. Such studies should preferably consist of solving the identical valve problem by both dynamic and quasi-static analysis. Regrettfully, the authors were not in a position to extend their work beyond the dynamic approach.

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REFERENCES

APPENDIX. DEFINITION OF SYMBOLS

\begin{align*}
\tau & \quad - \text{time} \\
r, \theta, z & \quad - \text{cylindrical coordinates} \\
\varepsilon_{rr}, \varepsilon_{\theta\theta}, \varepsilon_{zz}, \varepsilon_{rz} & \quad - \text{true strain components} \\
\sigma_{rr}, \sigma_{\theta\theta}, \sigma_{zz}, \sigma_{rz} & \quad - \text{true stress components} \\
\sigma_{m} & \quad - \text{average stress} = (\sigma_{rr} + \sigma_{\theta\theta} + \sigma_{zz})/3 \\
\sigma_{e}, \sigma_{\varepsilon} & \quad - \text{effective strain, effective stress (see formulas below)} \\
\sigma_{t}, \sigma_{\varepsilon_{t}} & \quad - \text{true strain, true stress (tension test)} \\
\sigma_{p} & \quad - \text{true plastic strain (tension test)} \\
\sigma_{y}, \sigma_{\varepsilon_{u}} & \quad - \text{yield stress, ultimate stress (tension test)} \\
\varepsilon_{e} & \quad - \text{true uniform elongation (tension test)} \\
\varepsilon_{f} & \quad - \text{true fracture strain} = \ln[1/(1-RA)] \\
RA & \quad - \text{reduction of area (tension test)} \\
E & \quad - \text{modulus of elasticity} \\
E_{t} & \quad - \text{tangent modulus (plastic range)} \\
\nu & \quad - \text{Poisson's ratio} \\
\rho & \quad - \text{specific mass} \\
\end{align*}

Formulas:

\begin{align*}
\bar{\varepsilon} & = (\sqrt{2}/3)[(\varepsilon_{rr}-\varepsilon_{\theta\theta})^{2} + (\varepsilon_{\theta\theta}-\varepsilon_{zz})^{2} + (\varepsilon_{zz}-\varepsilon_{rr})^{2} + 6\varepsilon_{rz}^{2}]^{1/2} \\
\bar{\sigma} & = (1/\sqrt{2})[(\sigma_{rr}-\sigma_{\theta\theta})^{2} + (\sigma_{\theta\theta}-\sigma_{zz})^{2} + (\sigma_{zz}-\sigma_{rr})^{2} + 6\sigma_{rz}^{2}]^{1/2}
\end{align*}
Fig. 1 - Sketch of isolation valve in closed position

Fig. 2 - Model dimensions of body (in inches and in centimeters)
Fig. 3 - Model dimensions of disk (in inches and in centimeters)

Fig. 4 - Finite-difference model (body shown in part)

Fig. 5 - Stress-strain diagrams for disk material, \( \sigma_t \) versus \( \varepsilon_p \)

(SA-182, Type 304 austenitic steel)
Fig. 6 - Stress-strain diagrams for body material, $\sigma_t$ versus $\epsilon_t$ (SA-216, Type WCB ferritic steel)

Fig. 7 - Kinetic energy dissipation versus time

Fig. 8 - Axial velocity at centerline of disk versus time (I = front surface, II = back surface)

Fig. 9 - Shape of model before and after impact
Fig. 10 - Distribution of effective strain after impact (t = 1150 μs)

Fig. 11 - Distribution of effective strain after impact (t = 1150 μs)

Fig. 12 - Distribution of effective strain along centerline (t = 1150 μs)

Fig. 13 - Strain history at Zones A, B and C defined in Figs. 10 and 11.