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COMPARISON OF AUTOFRETTAGE CALCULATION METHODS

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13. ABSTRACT *(Maximum 200 words)*

High-pressure vessel designers and builders have used residual bore compression for about 150 years to extend the useful life of the vessel. Casting, winding (wire or tape), shrinking, swaging, and autofrettage have produced the compression. All of these methods are fundamentally nonlinear and present different levels of computational difficulty. The hydraulic autofrettage process has been in use for over 70 years, and has been the subject of many analysis methods. This report uses several of the more popular methods that are in current use and attempts to demonstrate the differences between them.

The study uses an open-end, thick-wall cylinder of wall ratio $W = 3$ that is fabricated from a steel with a "soft" knee in the stress-strain curve. This case presents a strong material nonlinearity under small deformation conditions, which are normal for autofrettage problems. The baseline case is a finite element solution using isotropic plasticity, which is then solved using two other material models. These solutions can then be compared with the classic Tresca and von Mises criteria along with the ASME Code solution. The curves for autofrettage pressures versus percent overstrain are presented, along with residual stress versus radius for an autofrettage pressure that produces the 56.5% overstrain condition of the baseline solution.

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Autofrettage, Residual Stress, Thick-Wall Cylinders, Finite Element Analysis, Tresca Yield Criterion, von Mises Yield Criterion

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INTRODUCTION

The use of residual bore compression to strengthen cannon barrels can be traced back to an 1856 test conducted by Captain Thomas Jefferson Rodman using ten-inch columbiads (ref 1). His residual stress was produced by a hollow casting method, and it is clear that Rodman did not understand the true reason for the nine-factor improvement in fatigue life over a solid cast gun. This problem had been corrected by 1880, when the Army was using the Rodman plan casting method and was measuring the residual stress by a slitting method performed on test rings (ref 2). By 1890, the use of steel for cannons was producing an active discussion of the relative merits between compound guns made by shrinking several hoops together and wire wound (square wire) guns. Both produced high residual hoop compression at the bore. Thus, the compound construction became the favored method for cannons, but wire-wound pressure vessels are still used for other applications. However, the autofrettage method was just around the corner and this was well established by 1930 (ref 3). The design and manufacture of medium-size monoblock cannons after 1960 was greatly aided by the development of the swage process for producing the residual bore compression. This is still the primary method used at Watervliet Arsenal, but the hydraulic method remains popular for most high-pressure applications.

Hydraulic autofrettage uses an internal hydraulic pressure to load a thick-wall cylinder well beyond the yield stress at the bore. Then the natural reduction of strain through the wall is used to contain this pressure and return the bore to a state of residual compression, when the autofrettage pressure has been removed. The autofrettage process uses the nonlinear material behavior of high-strength steels, but does not normally require large strain or involve large deformations. The analysis is nevertheless difficult and over the years has attracted many investigators. This is because of the importance of the problem and the simple geometry of the structure. In the past, many assumptions have been made about the material behavior such as the shape of the stress versus strain curve and the failure mechanism. The computer-based finite element method has eliminated many reasons for simplifying assumptions at the price of a somewhat longer solution time and the realization that all nonlinear analytical methods are, to some degree, approximations.

This work compares the results of six models for the analysis of an autofrettage system using a thick-wall cylinder with a large radius ratio ($W = 3$) and a material with a rather generous or soft knee in the stress-strain curve.

GEOMETRY LOADING AND MATERIAL

The geometry consists of a plane circular cylinder with an inner radius, $a = 1$, and an outer radius, $b = 3$, giving it a wall ratio of $b/a = W = 3$. The cylinder is loaded with a slowly increasing uniform internal pressure using the open-end condition where the axial load is not supported by the cylinder. The material is high-strength steel with a soft knee in the stress-strain curve. This is illustrated by the different yield strengths at different offset strains, 1069 MPa at 0.05%, 1172 MPa at 0.10%, and 1264 MPa at 0.20% offset. These material properties were determined from a set of seven tensile tests on HY-180 steel loaded to a maximum of 4% strain. The material is unlike the normal low carbon steel used in cannons and pressure vessels that have a rather sharp transition from elastic to plastic behavior, but serves to demonstrate the point of this study.

ANALYSIS METHODS

The study compares the elastic-plastic analysis using six different methods and/or material models. These include three solutions using finite element analysis (ref 4) with different material-hardening models and three solutions using the more classic analysis methods that use the assumption of perfect plasticity. Two different types of solutions were obtained. First the calculation of the autofrettage pressure versus percent overstrain curve tracked the onset of plasticity across the wall of the cylinder as the pressure was increased. Then the residual stress was calculated for a loading that corresponded to an overstrain of 56.5%.

The first finite element solution used a kinematic, strain-hardening model, which required a bilinear material approximation. This condition was established by fitting the experimental data with two linear segments that met at the yield point. The fitting method produced the yield point with the minimum root mean square (RMS) error between a strain of zero and 4%. In this case, the yield point was at 1309 MPa.

The second ABAQUS solution used the isotropic-hardening condition, which may be defined with any number of linear segments in the stress versus plastic-strain table. In this case, four points were used to define the three linear segments. Again, the position of the initial yield point was varied to produce a solution with a minimum RMS error, which resulted in a yield stress value of 1090 MPa. This solution was used to establish the 56.5% overstrain condition. The overstrain produced a residual von Mises stress at the bore of 1090 MPa, which is just at the point of reyield.

The third finite element solution used the combined hardening model from ABAQUS, which links the ability to define a nonlinear stress-strain model with a kinematic, strain-hardening condition. Here the model was calibrated from eight data points starting at the yield point of 968 MPa.

The last three solutions used classic closed-form methods that use only a yield stress and assume perfect (flat) plasticity. These methods used the same yield strength at the 0.20% offset point, which was 1264 MPa.

The fourth solution was taken from the ASME Boiler and Pressure Vessel Code (ref 5), which is a modified Tresca yield criterion. The fifth solution used the von Mises yield criterion as defined in the classic paper of Davidson et al. (ref 6). The sixth solution is the older Tresca yield criterion usually attributed to Hill (ref 7).

RESULTS

Figure 1 is a single plot of the autofrettage pressure versus percent overstrain for each of the six methods. The figure was produced using a strict definition of percent overstrain, where percent overstrain defines the position in the wall of the elastic-plastic interface. In ABAQUS, this becomes the existence of a nonzero value for the equivalent plastic strain. The other type of presentation is a plot of the three principal stresses versus radius for the standard overstrain of 56.5%. In this case, there is a separate plot for each solution, shown in Figures 2 through 7. Note that Figures 2 through 4 show three principal stresses—radial, axial, and hoop—as these solutions are for the open-end condition that would result when the autofrettage process is done on a mandrel or in an external frame. Figures 5 through 7 show no axial stress because all of these solutions assume the plane-stress condition, which has no axial stress effects.

Also note that Figures 2 through 7 have a notation of the autofrettage pressure required to produce the data shown. One interesting point is that these pressures vary about 10% around the mean value of 1417 MPa.

DISCUSSION

Figure 1 shows that the six solutions tend to be in two groups—the three finite element solutions and the three closed-form solutions. All six start in the same range at initial plasticity (0% overstrain). However, the three finite element solutions contain some form of strain hardening, which requires a greater pressure to achieve the 100% overstrain condition.

Figure 2, using the bilinear kinematic, strain-hardening condition, demonstrates some reyield at the bore. The effect is rather pronounced in the hoop stress curve, with a much smaller effect in the axial stress curve. Also note the well-defined break in the hoop stress at the elastic-plastic interface.

Figure 3 shows the isotropic strain-hardening condition with a multi-linear stress-strain approximation. Each break in the stress-strain approximation is evident in the hoop stress curve. There is no evidence of reyield in this curve because this is a master solution that defined the 56.5% overstrain with the maximum elastic von Mises stress at the bore.

Figure 4 demonstrates results from the combined isotropic/kinematic-hardening model. In this case, the curves are much smoother because the experimental stress-strain data have been calibrated to an analytical function that was used for the solution. This method does not show the sharp change in the hoop stress curve, as shown by the kinematic solution. The break in this curve at the elastic-plastic interface is not very well defined.

Figure 5 includes data resulting from application of the solution given in the ASME Boiler and Pressure Vessel Code, Division 3 (ref 5). The figure predicts the lowest value for the residual hoop stress of any of the six solutions, and it also predicts a smooth hoop stress curve similar to that shown for the better finite element model in Figure 4. The low value for the residual hoop stress may be the result of the code materials, which show a large Bauschinger effect.

Figure 6 depicts an analysis using the von Mises yield criterion and an elastic-perfectly plastic stress-strain curve. The residual hoop stress at the bore was off the chart at 1838 MPa; thus this method cannot account for plastic reyield. This is an unfortunate result because the von Mises yield criterion is generally considered to be the preferred method, and it is the method used for all the finite element results.

Figure 7 shows the standard solution for the Tresca yield criterion, which produces rather reasonable results despite the fact that it is less accurate. Also, it shares the problem of the last three solutions in that it ignores all axial stress effects.

CONCLUSION

The six methods discussed here produce substantially different results, for this material and wall ratio, but the importance of these differences is left to the reader. A study using more conventional low carbon steel may produce smaller differences, but greater care may be necessary when the materials are changed. Comparisons should be done from a carefully performed experimental and theoretical study in order to verify the material models, however these authors know of no definitive experimental data set that could be used as the basis for this study.

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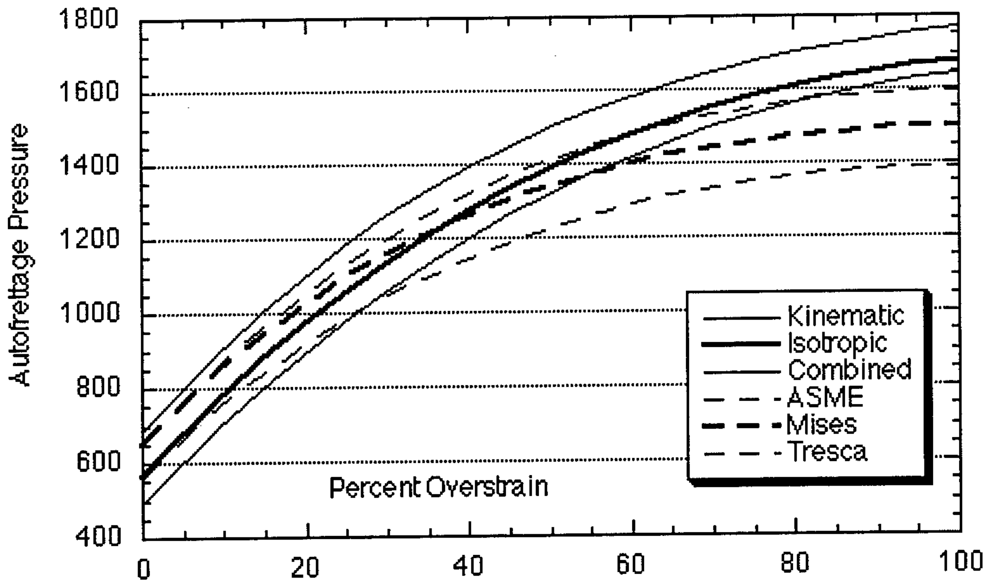

Figure 1. Autofrettage pressure versus percent overstrain for all six solution methods.

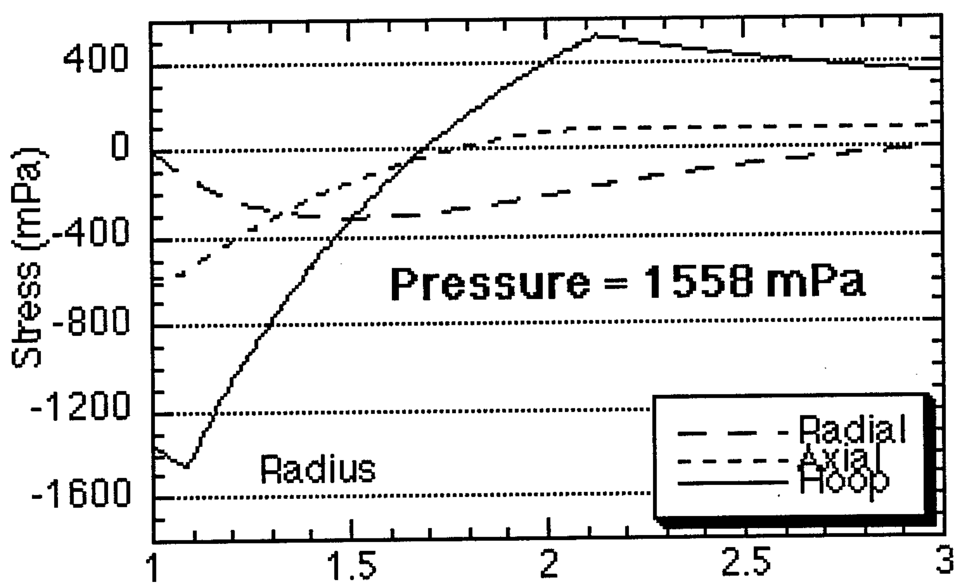

Figure 2. Finite element analysis using isotropic strain hardening.

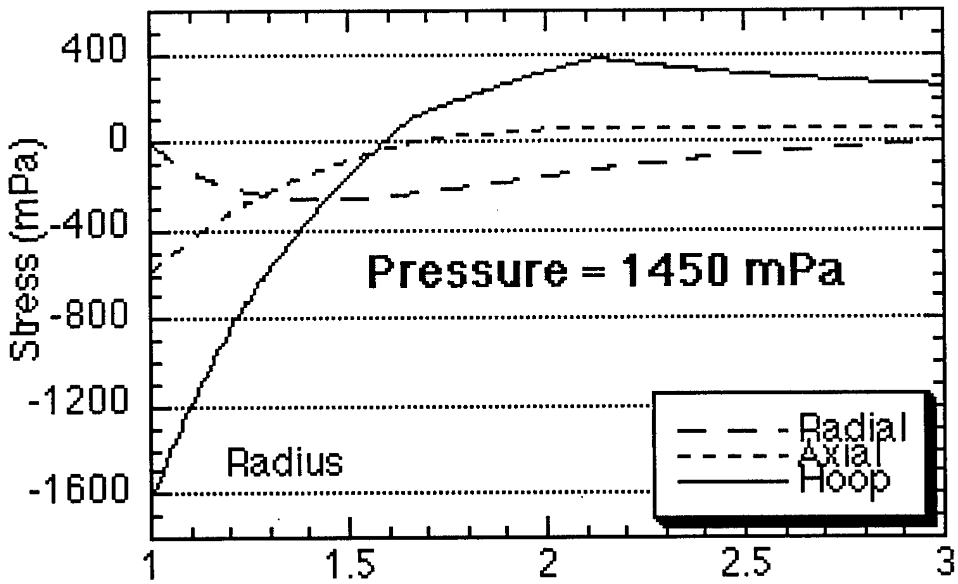

Figure 3. Finite element analysis using kinematic strain hardening.

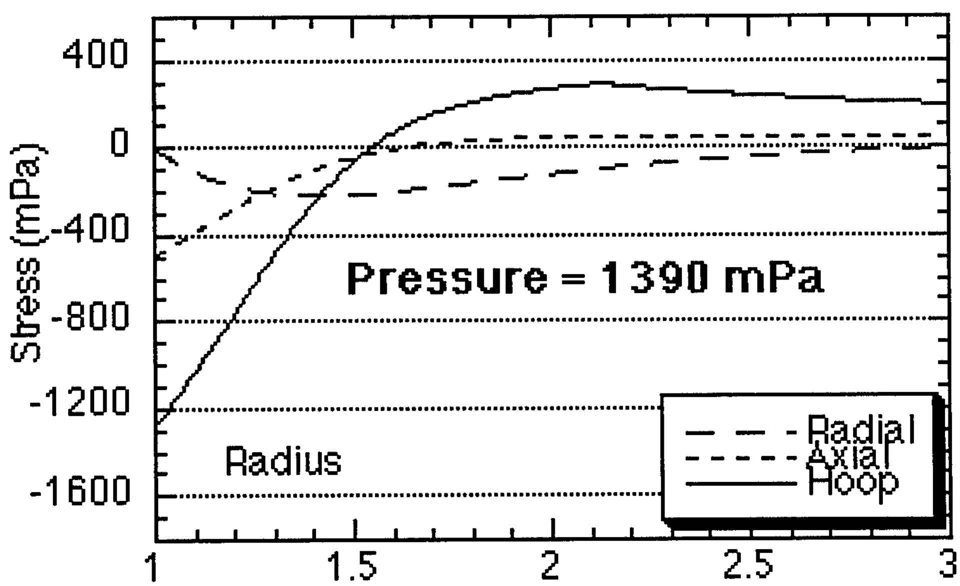

Figure 4. Finite element analysis using combined strain hardening.

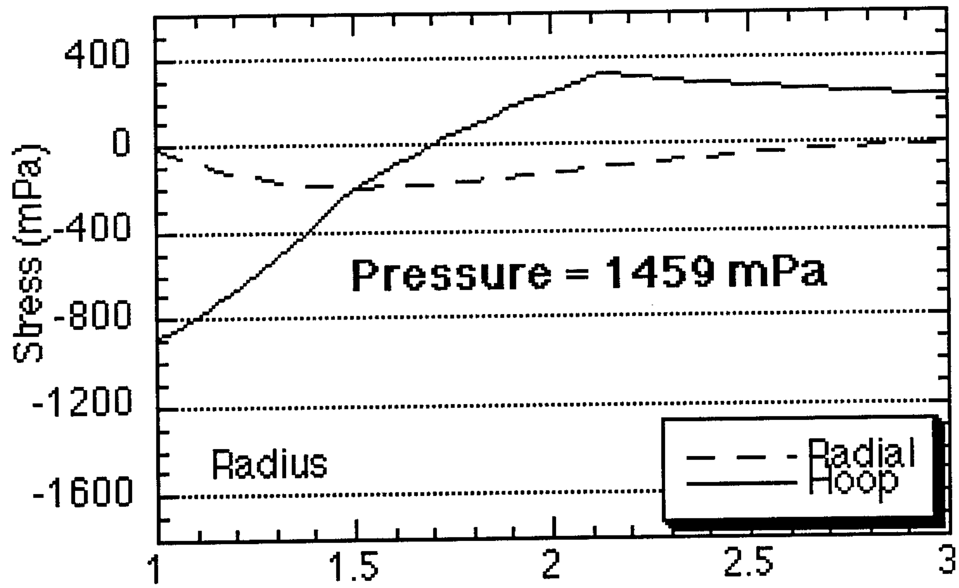

Figure 5. Analysis from the ASME Code, Division 3.

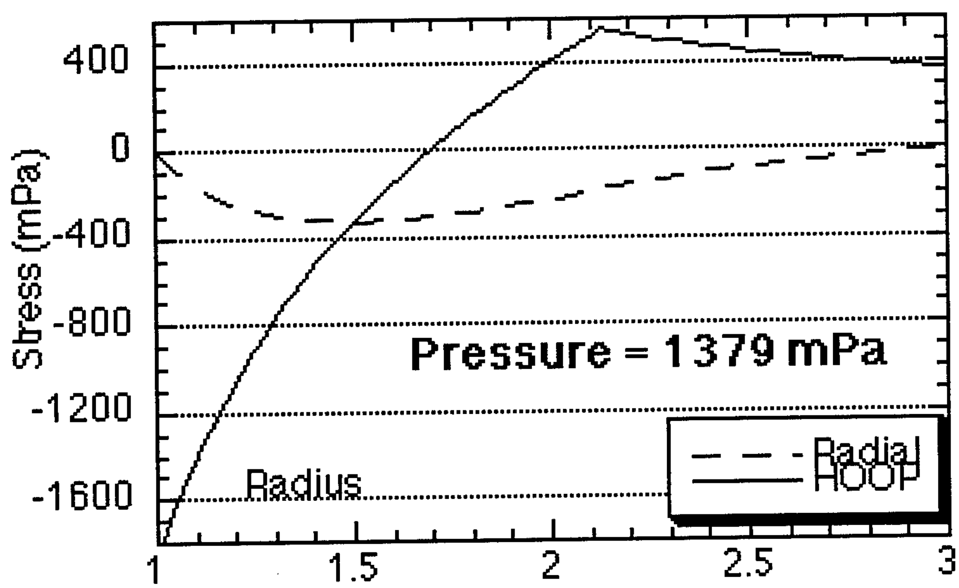

Figure 6. Analysis using the von Mises failure criterion and perfect plasticity.

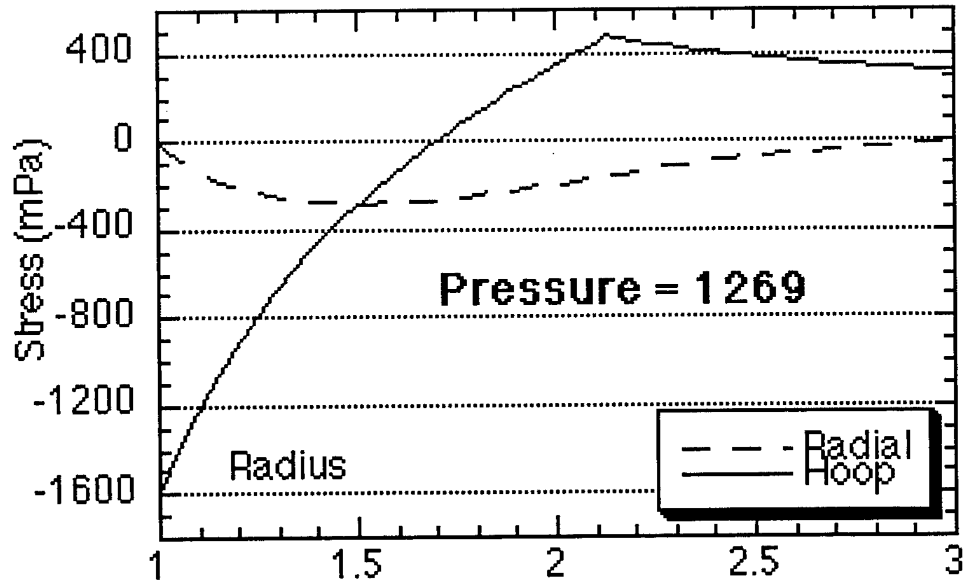

Figure 7. Analysis using the Tresca failure criterion and perfect plasticity.

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