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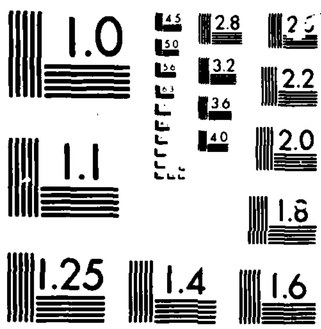

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TECHNICAL REPORT ARCCB-TR-86009

A DESIGN METHOD FOR AUTOFRETTAGED THICK-WALLED CYLINDERS WITH OUTSIDE DIAMETER DISCONTINUITIES

J. J. BUSUTTIL, JR.

J. A. KAPP

FEBRUARY 1986

US ARMY ARMAMENT RESEARCH AND DEVELOPMENT CENTER
CLOSE COMBAT ARMAMENTS CENTER
BENÉT WEAPONS LABORATORY
WATERVLIET, N.Y. 12189-4050

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A DESIGN METHOD FOR AUTOFRETTAGED THICK-WALLED CYLINDERS WITH OUTSIDE DIAMETER DISCONTINUITIES

by

J. J. BUSUTTIL, JR.

and

J. A. KAPP

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A design method has been developed to determine the fatigue life of thick-walled autofrettaged cylinders containing outside diameter (OD) notches which act as failure initiation sites. The method uses local strain analysis for fatigue crack initiation and fracture mechanics analysis for crack propagation to failure. The method is relatively easy to program and thus can be used to optimize cylinder designs with respect to OD initiated failures. The computer

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20. ABSTRACT (CONT'D)

program developed requires only minimal input to estimate life: cylinder dimensions, notch depth and root radius, internal pressure, and material yield strength. Other material properties (low cycle fatigue data, fracture toughness, and crack growth law) are permanently stored.

The program calculates the elastic stress concentration factor using a Neuber diagram as a default. If the elastic k_t is known from other sources, this feature of the program can be overridden. The stress concentration factor is used to calculate notch root stresses from which local strains are estimated. Once the local strains are known, crack initiation life is estimated using the stored low cycle fatigue data. A crack is then assumed to exist and a power law is integrated to determine crack propagation life to failure. The total life is the sum of the initiation and propagation lives.

The applicability of the design method is demonstrated by using it to predict the total fatigue lives of existing cylinder designs with measured fatigue lives. The method's predictive capability is very good to conservative. In no case was life substantially overestimated. In addition, a design example is presented.

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INTRODUCTION

Thick-walled cylinders designed to contain pressures in excess of 69 MPa (10,000 psi) are often autofrettaged to increase fatigue life and ensure dimensional stability. Autofrettage is an initial overpressurization that produces permanent residual stresses in the cylinder. The hoop residual stresses are compressive at the inner diameter (ID) and vary logarithmically to tension at the outer diameter (OD). The compressive residual stress at the ID both retards fatigue crack initiation at that surface and crack propagation through the wall. If no structural discontinuity exists at the OD, fatigue failure usually occurs from the ID out through the wall thickness. Certain pressure vessels such as modern large caliber cannon require some structural features on the OD of the cylinder. If these cylinders are autofrettaged and the pressure cycled, the combination of tensile residual stress and tensile operating stress plus stress concentration can often result in fatigue failure from the OD in through the wall thickness. This can be a much worse situation than the ID initiated failure since much smaller critical crack sizes are encountered in this case, increasing the possibility of catastrophic failures. In the case of ID initiated cracks, when modern materials and autofrettaged cylinders are considered, unstable fracture usually does not occur until cracks are very large and easy to detect. Since OD initiated failures are far more nasty than ID initiated failures, it is imperative that accurate or conservative design methods be developed to prevent their occurrence.

The approach taken here is to consider the fatigue failure of OD notched autofrettaged cylinders to be a two-step process: crack initiation followed by fatigue crack propagation to failure. To estimate the number of pressure

cycles to initiate a crack, the local strain method is used (ref 1). This method assumes that the element of material that exists at the root of the OD notch acts exactly as a small strain-controlled low cycle fatigue specimen. If the strain that occurs during pressure loading can be determined, the number of cycles to break the small strain-controlled specimen is directly obtained from the low cycle strain-life property of the material that is used to manufacture the cylinder. The cylinder is then considered to contain a small crack at the root of the OD notch. Further, it is assumed that this small crack will now act as a large crack that has depth equal to the notch depth and extends for the entire length of the cylinder. The growth of such a crack can be modeled using fracture mechanics (ref 2). If the stress intensity factors for the cracked cylinder, the crack growth behavior of the material, and the fracture toughness of the material are all known, the number of cycles necessary to grow the crack from its small length to critical size can be calculated. The total life of the cylinder is simply the sum of the crack initiation and the crack propagaton lives.

To predict the fatigue life of OD notched cylinders all that is needed are three material properties (low cycle strain-life behavior, fatigue crack growth behavior, and fracture toughness) and two calculated quantities (the local strains at the notch and the stress intensity factors). Since the material properties for the pressure vessel steel considered are known and the stress intensity factors for pressurized thick-walled cylinders with OD cracks

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[1]H. S. Reemsnyder, "Constant Amplitude Fatigue Life Assessment Models," SAE Technical Paper Series 820688, Proceedings of the SAE Fatigue Conference, April 1982, pp. 119-132.

[2]P. C. Paris and F. Erdogan, Journal of Basic Engineering, Vol. 85, No. 4, 1963, pp. 528-534.

are also available (ref 3), as are the stress intensity factors for autofrettaged thick-walled cylinders with OD cracks (ref 4), all that is needed for life prediction is to calculate the strains at the root of the notch. A stress analysis of a proposed design would probably be performed in any case. Therefore, no additional analysis is necessary to estimate OD initiated fatigue failure as outlined above. Furthermore, since the method is reasonably accurate as will be demonstrated, and is easily programmed, it can be very useful in attempting to optimize the design of an OD notched cylinder because all that is needed is a calculated quantity and not something that needs to be experimentally measured.

PROCEDURE

The first quantity needed is an estimate of the strains at the root of the notch. These can be calculated from an elastic stress analysis of the notch geometry being considered. If we know the elastic stress concentration factor k_t, the local strain can be estimated as:

$$\epsilon = k_f \, \sigma / E \tag{1}$$

where ϵ is the strain, k_f is the fatigue strength reduction factor, σ is the nominal stress, and E is the elastic modulus. The maximum strain is obtained from Eq. (1) with the nominal stress as the maximum stress, and the minimum strain is similarly calculated with the nominal stress as the minimum stress. All other quantities describing the loading cycle (strain range $\Delta\epsilon$, stress

[3] J. A. Kapp and R. Eisenstadt, "Crack Growth in Externally Flawed, Autofrettaged Thick-Walled Cylinders and Rings," ASTM STP 677, 1979, pp. 746-756.

[4] J. A. Kapp and S. L. Pu, "Considering the OD as a Failure Initiation Site in the Design of Thick-Walled Cylinders," ARRADCOM Technical Report ARLCB-TR-82029, Benet Weapons Laboratory, Watervliet, NY, April 1982.

ratio or strain ratio R, mean strain, etc.) are determined in the usual manner from the minimum and maximum loading values.

In order to determine k_f, the elastic stress concentration factor k_t must first be determined. This can be accomplished by either determining k_t by an independent stress analysis or with a Neuber diagram (ref 5). For convenience, the Neuber diagram approach was included as a default in the computer program developed. To use this method, three critical notch dimensons are required: the notch root radius, the notch depth, and the wall thickness. Knowing these parameters, k_t is determined by assuming that the notch in a cylinder is approximated as a single edge notch specimen subjected to pure bending. This is considered a conservative assumption. The Neuber diagram in Reference 5 for the pure bending case was fit with an algebraic expression, from which k_t is easily determined.

The fatigue strength reduction factor k_f is determined in the usual manner:

$$k_f = \alpha(k_t - 1) + 1 \tag{2}$$

where q is the notch sensitivity which is a function of the notch root radius ρ and the ultimate tensile strength σ_{UTS} of the material (ref 1).

$$q = 1/(1 + a'/\rho) \tag{3}$$

$$a' = 0.001(300/\sigma_{UTS})^{1.8} \tag{4}$$

Equation (4) holds for most high strength low alloy steels with the ultimate tensile strength in units of Ksi, and with a' and ρ in units of inches.

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[1]H. S. Reemsnyder, "Constant Amplitude Fatigue Life Assessment Models," SAE Technical Paper Series 820688, Proceedings of the SAE Fatigue Conference, April 1982, pp. 119-132.

[5]A. C. Ugural and S. K. Fenster, Advanced Strength and Applied Elasticity, American Elsevier Publishing Company, New York, 1975, p. 83.

When using SI units, the following equation holds, with the ultimate strength in MPa and a' and ρ in mm:

$$a' = 0.0254(2068/\sigma_{UTS})^{1.8} \quad (5)$$

Once the local strains are known, we estimate the crack initiation life using the low cycle fatigue strain-life data for the material given in the next section. As stated above, once crack initiation has occurred, we assume that a crack is present which has depth equal to the notch depth and extends over the entire length of the cylinder. It is further assumed that this crack will grow according to the power law given below which accounts for the effects of stress intensity factor ratio ($R = K_{min}/K_{max}$) due to the presence of the autofrettaged residual stresses.

The stress intensity factor (K) solutions for a cylinder with radius ratio (k = b/a) of two for the case of internal pressure (ref 3), and for the cases of 100 percent and 50 percent overstrain (ref 4) have been developed. Both of these solutions were developed using finite element analysis, and polynomial expressions have been fit to them to explain the variation in K as the crack grows through the wall thickness (ref 4). Although published in Reference 4, these expressions are reprinted here as Table I. For overstrain conditions between 50 and 100 percent, the stress intensity factor is determined by interpolation at constant crack depth. To determine the crack propagation life, the power law given below is integrated using Simpson's Rule from the assumed initial crack depth to that crack depth at whick K_{max} is equal to

[3] J. A. Kapp and R. Eisenstadt, "Crack Growth in Externally Flawed, Autofrettaged Thick-Walled Cylinders and Rings," ASTM STP 677, 1979, pp. 746-756.

[4] J. A. Kapp and S. L. Pu, "Considering the OD as a Failure Initiation Site in the Design of Thick-Walled Cylinders," ARRADCOM Technical Report ARLCB-TR-82029, Benet Weapons Laboratory, Watervliet, NY, April 1982.

TABLE I. EXPRESSIONS FOR STRESS INTENSITY FACTORS FOR THICK-WALLED CYLINDERS WITH OD CRACKS

$$K = \sigma_{OD}\sqrt{\pi c}\{1.12 + A_1(c/t) + A_2(c/t)^2 + A_3(c/t)^3 + A_4(c/t)^4\}$$

c = Crack Length

t = Wall Thickness

σ_{OD} = The Hoop Stress at the OD

Loading	A_1	A_2	A_3	A_4
Internal Pressure	0.31	6.85	-12.1	10.0
50% Overstrain	0.00	0.00	16.9	-21.0
100% Overstrain	-1.44	4.44	- 4.31	0.0

For Internal Pressure:

$$\sigma_{OD} = \frac{2p}{k^2-1}$$

For 50% Overstrain:

$$\sigma_{OD} = \frac{2}{\sqrt{3}} \sigma_y \left\{\frac{1+2k+k^2}{4k^2} + \frac{1}{k^2-1} \left[\frac{1+2k+k^2}{4k^2} - 1 - 2 \ln\left(\frac{1+k}{2}\right)\right]\right\}$$

For 100% Overstrain:

$$\sigma_{OD} = \frac{2}{\sqrt{3}} \sigma_y \left[1 - \frac{2 \ln k}{k^2-1}\right]$$

Where:

p = Internal Pressure

k = Radius Ratio

σ_y = Yield Strength

the fracture toughness (K_{Ic}) of the material. At this point, unstable crack propagation is assumed to occur and the life of the cylinder is complete. The total number of pressure cycles of the cylinder is then the sum of the cycles to cause crack initiation and the number of cycles to propagate the crack to unstable size.

MATERIAL

The material that the cylinders are to be manufactured from is Grade 1, Class 4, ASTM A723 Alloy Steel Forgings for High-Strength Pressure Component Application. The yield strength of the material is about 160 Ksi (1105 MPa) and has ultimate strength of 175 Ksi (1240 MPa). The fracture toughness of such material is nominally about 130 Ksi$\sqrt{in.}$ (143 MPa$\sqrt{m}$). The low cycle fatigue data including the effects of strain ratio for the material have been measured (ref 6). For the purposes of the design method, a mathematical expression has been fit to the lower bound of these data. This assumes that there is no effect of mean stress on crack initiation which is incorrect; but for our purposes, this assumption is conservative. The data were fit with a power law using least squares analysis:

$$N = 0.90\Delta\epsilon^{-2.42} \qquad (6)$$

where N is the number of cycles and $\Delta\epsilon$ is the strain range ($\Delta\epsilon = \epsilon_{max} - \epsilon_{min}$) with units in percent. Equation (6) is compared in Figure 1 to the measured data from Reference 6. This comparison shows that the equation is an adequate lower limit.

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[6] J. A. Kapp, "Predicting Catastrophic OD Initiated Fatigue Failure of Thick-Walled Cylinders Using Low Cycle Fatigue Data," Mat. Res. Soc. Symp. Proc., Elsevier Science Publishing Co., Inc., New York, Vol. 22, 1984, pp. 243-248.

Since fatigue cracks, once initiated, will be growing through a residual stress field, the effects of these residual stresses must be accounted for. It is assumed that the residual stresses will affect the K ratio R throughout the entire fatigue propagation life of the cylinder. In Reference 4 several crack growth laws were applied to the problem of OD cracks growing through autofrettage residual stress fields. It was stated in that report that a simple power law that assumes that the crack growth rate is proportional to the maximum stress intensity factor raised to some power is the most conservative representation of the measured results. This power law was used as a first attempt in this study. The results were considered to be overly conservative. To give a better representation of the crack growth behavior, Forman's power law was used (ref 4). For the material of concern here, the power law is:

$$\frac{dc}{dN} = \frac{4.5\times10^{-8}\Delta K^{2}}{K_{Ic}(1-R) - \Delta K} \quad (7)$$

where dc/dN is the crack growth rate in m/cycle and ΔK is the stress intensity factor range ($\Delta K = K_{max} - K_{min}$).

LIFE PREDICTIONS FOR CYLINDERS WITH MEASURED FATIGUE LIVES

Several OD notch geometries have been studied (ref 6). These generally have consisted of two general types shown in Figure 2. The sectored geometry is for quick-connect end enclosures, and the single notch type is for a cylinder with an applied torque. Both geometries can be addressed using the

[4] J. A. Kapp and S. L. Pu, "Considering the OD as a Failure Initiation Site in the Design of Thick-Walled Cylinders," ARRADCOM Technical Report ARLCB-TR-82029, Benet Weapons Laboratory, Watervliet, NY, April 1982.

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life prediction scheme described herein. In all cases, the material was assumed to be the nominal material described above, and life predictions were made assuming worst case conditions. These conditions included the minimum allowed root radius, the maximum allowed notch depth, and the minimum allowed wall thickness, etc.

The predicted fatigue lives for these geometries are summarized in Table II. As can be seen, the prediction method gives results which vary from relatively accurate to conservative. In the case of the single notch cylinder with 100 percent overstrain and 241 MPa internal pressure, the lower limit of the measured life is predicted quite accurately. The same cylinder with less autofrettage (75 and 62 percent) has a substantially longer measured fatigue life. The prediction reflects a longer life, but not to the extent observed. Since both of these measured lives are based upon only one specimen, it is difficult to draw many conclusions except that the prediction is quite conservative. For a cylinder with a single notch, 100 percent overstrain, and higher pressure (386 MPa), the predicted life is within the range of the observed results, but is near the high end. We do not consider the method to be nonconservative in this case. This result suggests that the method may need some improvement for cases of relatively short life. The final comparison with a sectored cylinder reveals the method is again quite conservative. In all, we can say that the predictive capabilities of the combined local strain-fracture mechanics approach to the problem is adequate for design purposes. In no case is the actual fatigue life substantially overestimated. On the contrary, for the most part it is conservative. Only in the instance of relatively short life is there reason to be concerned. Thus, if the

predicted life is less than about 5,000 cycles, we would suggest that safety factors be used to ensure a safe design.

TABLE II. COMPARISON OF PREDICTED FATIGUE LIVES AND ACTUALLY MEASURED LIVES

Notch Type	Notch Depth (mm)	Root Radius (mm)	ID (cm)	OD (cm)	O.S. (%)	p (MPa)	Prediction (kcycles)	Actual (kcycles)
A	10	1.5	15.5	27.0	100	241	14.9	14.3-34.4
A	10	1.5	15.5	27.0	100	386	4.1	2.7-4.5
A	10	1.5	15.5	27.0	75	241	16.2	32.0
A	10	1.5	15.5	27.0	62	241	17.1	39.5
B	5	0.8	16.5	30.2	100	372	8.5	18.0-24.8

Notch Type A = Single Notch
Notch Type B = Sectored Notches

A DESIGN EXAMPLE

The predictive method is now used to suggest design changes to alter the fatigue life of one of the cylinder designs examined in Table II. The case of the shortest life cylinder is chosen. It is desired to increase the life of this design to about 12,000 cycles and still provide the same service. As stated above, the notch in designs such as this reacts to an applied torque. This is accomplished by fitting a key inside the notch. It is required that the improved design be able to react to the same torque. In other words, the bearing area of the notch must remain constant or increase.

The very short life is due to two factors, the depth of the notch and the amount of autofrettage. The depth of the notch is necessary for load bearing capability, but results in a high stress concentration factor and leaves relatively little material for subcritical cracks to propagate through. The large

amount of autofrettage overstrain results in higher R ratios and thus increased crack growth rates. It also increases K_{max} during each loading cycle which decreases the critical flaw size. To increase the fatigue life, two changes should be made. The notch depth should be decreased to reduce stress concentration and thus increase crack initiation life. This will also allow for more stable crack growth as the fatigue crack that initiates from the notch will have farther to grow before fast fracture occurs. Secondly, the amount of autofrettage should be decreased. This will have little effect on crack initiation but will decrease the R ratio, slowing crack propagation and will also increase the critical flaw depth as K_{max} during pressure loading will be decreased.

The fact that the bearing area must remain constant or increase remains a troubling problem. The solution suggested above will only decrease the load bearing area. This can be overcome by introducing a second notch. Both of these notches will be shallower than the original, but their load bearing capability taken in aggregate will be greater than that of the single deeper notch. In addition, the amount of autofrettage overstrain was reduced from 100 to 60 percent. This level of overstrain was chosen arbitrarily. As will be demonstrated, this level of overstrain is not optimum. The proposed solution is shown in Figure 3.

A stress analysis of both the original design and the proposed design was conducted using finite element analysis. The meshes used to generate the stresses are pictured in Figures 4 and 5. Quadratic isoparametric elements were used throughout. The structural model for the proposed design included 174 elements and 536 nodes. Such a model is equivalent to a model which has about 2,500 degrees of freedom using constant strain elements. From previous

experience, we expect that this representation of the problem will give elastic stress concentration factors to an accuracy of no worse than about five percent. The finite element model of the present design included fewer elements and nodes than the proposed design. Although it did not include as many nodes or elements, it is considered to be of comparable accuracy, as the geometry is somewhat less complex. The analysis was conducted only for the case of internal pressure. The calculated stress concentration factors are used only to predict crack initiation. As stated above, the effects of mean stress or mean strain have been ignored by using the shortest life low cycle fatigue data measured in Reference 6.

The results of the stress analysis and fatigue life predictions are summarized in Table III. It is clear that the requirements of the design problem have been met. The elastic stress concentration factor has been reduced substantially. This resulted in a predicted initiation life that is roughly two and one-half times greater than the present design. Also, the critical crack size was increased such that more than double the amount of crack extension is allowed. The predicted crack propagation life is accordingly increased by a factor about six and one-half. All together, the predicted total life was increased by a factor of 2.98 which is very nearly the required amount of three.

[6] J. A. Kapp, "Predicting Catastrophic OD Initiated Fatigue Failure of Thick-Walled Cylinders Using Low Cycle Fatigue Data," Mat. Res. Soc. Symp. Proc., Elsevier Science Publishing Co., Inc., New York, Vol. 22, 1984, pp. 243-248.

TABLE III. COMPARISON OF FATIGUE LIFE PREDICTIONS FOR THE PRESENT DESIGN AND THE PROPOSED DESIGN

Design	k_t	Allowable Crack Growth (mm)	Initiation Life (cycles)	Propagation Life (cycles)	Total Life (cycles)	Actual Life (cycles)
Present	3.8	6.6	3767	384	4151	2752-4547
Proposed	2.8	14.1	9828	2550	12378	10813

A cylinder was manufactured to the new design dimensions and was cycled to failure. The measured life was somewhat less than the predicted life. This also occurred with the predicted life for the current design. However, in the case of the proposed design, the cause was not an inability of the prediction scheme. The specimen failed from a crack that initiated at the ID and not from the OD as had been assumed. This is the result of reducing the amount of overstrain from 100 to 60 percent. This result suggests that the design method outlined here must be used in conjunction with some method of determining the fatigue life of a cylinder from cracks initiating at the ID. The marriage of two such techniques would allow for optimization of designs considering both ID and OD initiated failures. A further comment on the OD initiated design method is warranted. It is clear that the prediction method described is adequate for the proposed design. Had a crack initiated at the OD notch, the cylinder would have had a total life of at least 10,813 cycles. If it were less than that, the cylinder would have failed from the OD rather than the ID as it did. Although the cylinder failed from the ID, the proposed design should be acceptable, since the life was increased by a factor between 2.4, by comparing with the longest life measured in the present design cylinders, and 3.9, by comparing with the shortest life measured in the present

design.

CONCLUSIONS

A method for predicting total fatigue life of autofrettaged thick-walled cylinders with OD notches has been developed and can be applied to the design of such cylinders. The technique is accurate to quite conservative when applied to cylinders with measured fatigue lives. All that is needed to apply this method is a stress analysis of the OD notch design. The properties of the material that the cylinder will be manufactured from are ASTM A723 Class 1, Grade 4. If a different material is to be used, low cycle fatigue data, crack propagation data, and fracture toughness must be measured. The design method was applied to a specific problem and proved adequate in predicting the fatigue life of the new design.

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1. H. S. Reemsnyder, "Constant Amplitude Fatigue Life Assessment Models," SAE Technical Paper Series 820688, Proceedings of the SAE Fatigue Conference, April 1982, pp. 119-132.
2. P. C. Paris and F. Erdogan, Journal of Basic Engineering, Vol. 85, No. 4, 1963, pp. 528-534.
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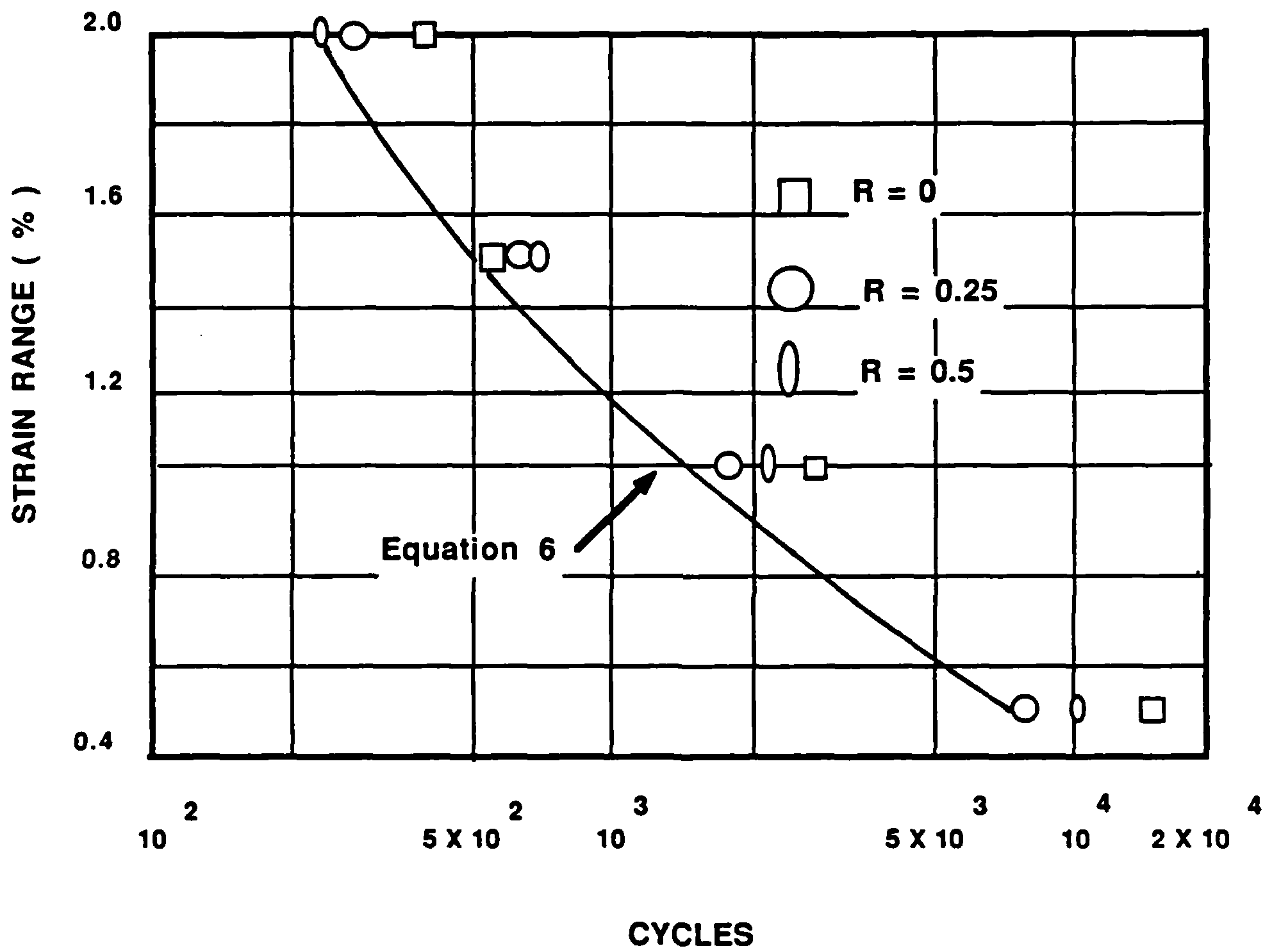
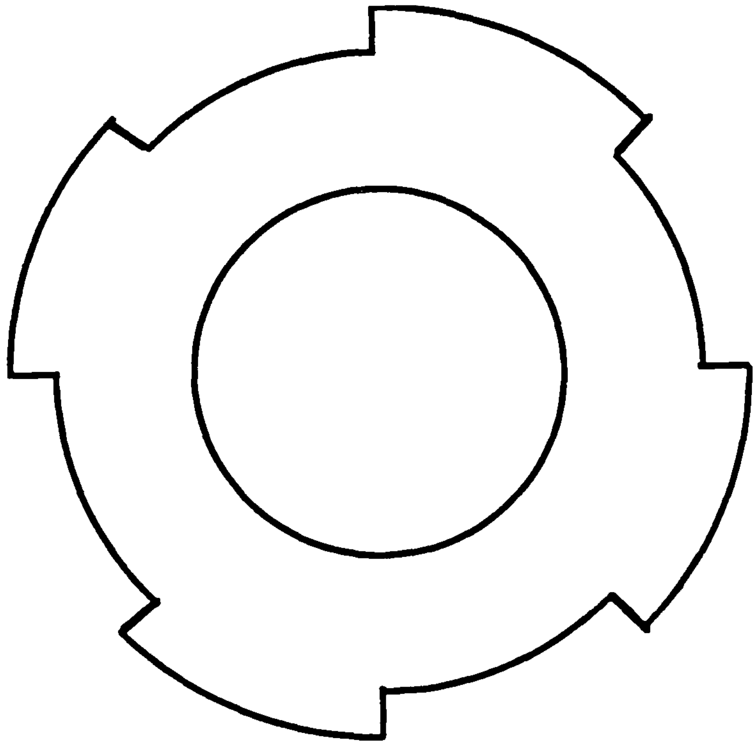
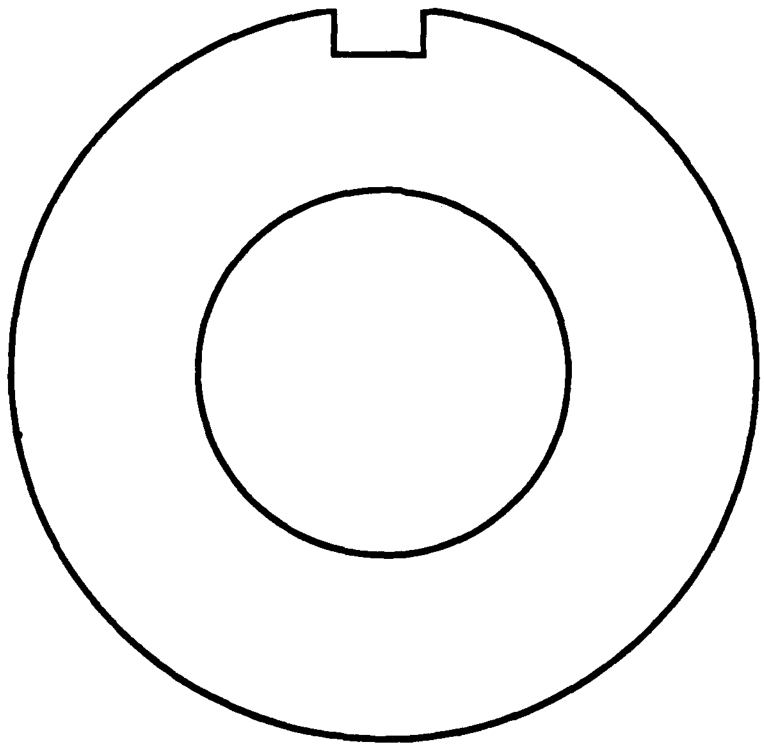


Figure 1. Low Cycle Fatigue Data.

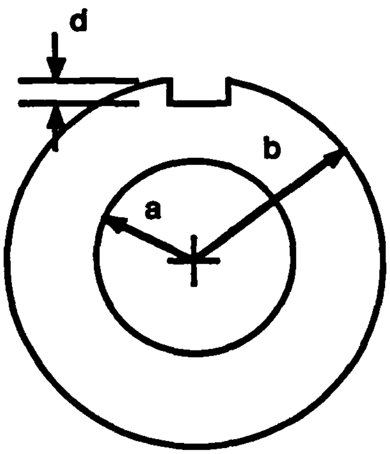


(a) Sectored Notches.

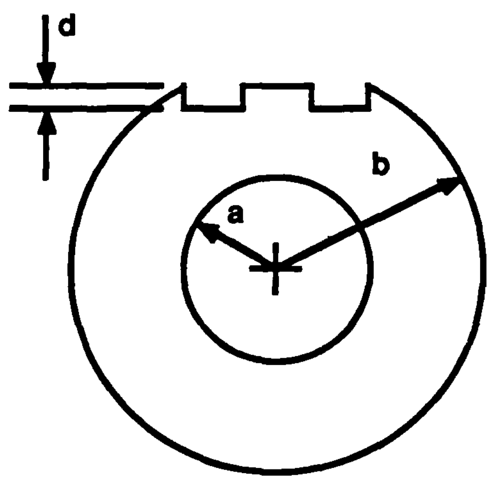


(b) Single Notch.

Figure 2. OD Notch Schematics.



(a) Present Design.



(b) Proposed Design.

Figure 3. Design Examples.

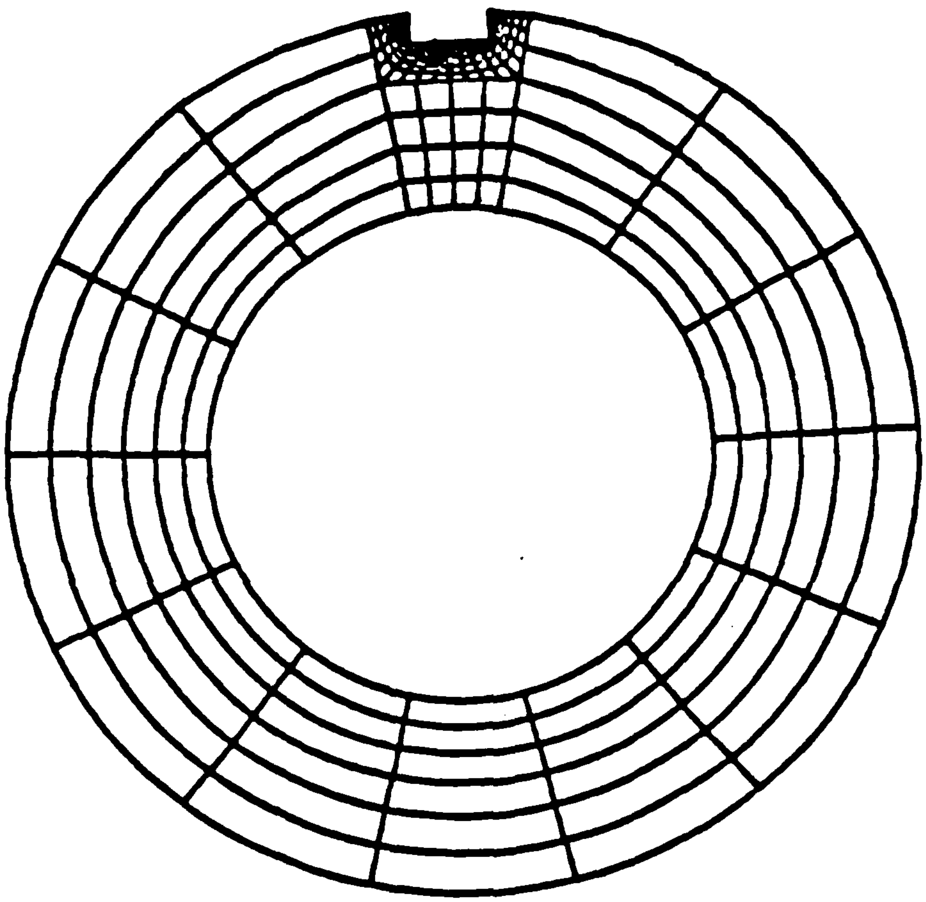


Figure 4. Finite Element Mesh for the Present Design.

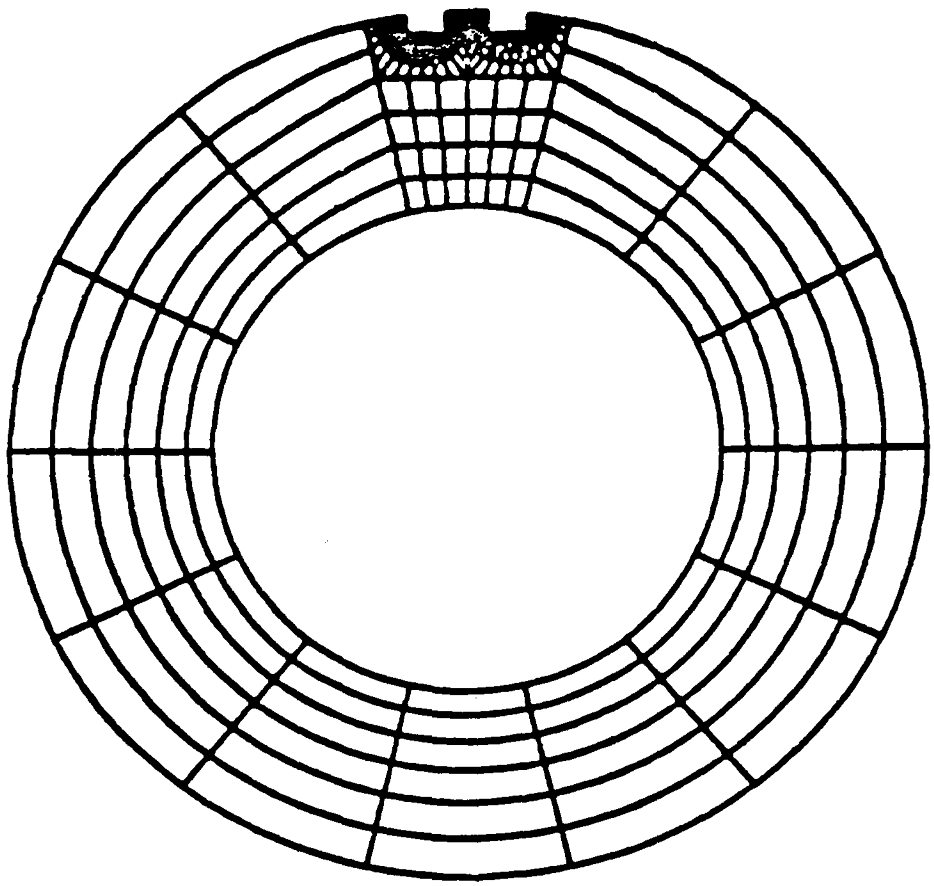


Figure 5. Finite Element Mesh for the Proposed Design.

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