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Failure of a heavy-duty hydraulic cylinder and its fatigue re-design

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ABSTRACT

The unexpected in-service failure of a heavy-duty hydraulic cylinder motivated the present investigation. The combined use of fracture mechanics concepts and of the finite element method demonstrated that part failure was due to the specific weld joint solution between cylinder and end-cap and the fatigue life predictions correlated with the estimated service life before crack detection. Alternative designs involving modified end cap geometry were developed and demonstrated to achieve a considerably longer operational life.

1. Introduction and motivation

A hydraulic cylinder (also called a linear hydraulic motor) is a mechanical cylinder that is used to give a linear force through a linear stroke. Hydraulic cylinders get their power from pressurized oil. Hydraulic cylinders are frequently found in equipments and machinery, such as construction equipment (excavators, bull-dozers, and road graders) and material handling equipment (fork lift trucks, telescopic handlers, and lift gates).

The relative product simplicity, long industrial experience with its use and the large number of manufacturing companies with strong competition reduce the design phase to some standard considerations and previous service experience is often the indirect validation of the design solution.

In some instances, however, a combination of unexpected factors may reveal a potential criticality of the product that requires quick action to overcome the crisis and solve the problem. Such a situation was dealt with by the authors and is summarized in this contribution. A company producing heavy-duty cylinders was called upon by a customer to explain an unexpected and premature cylinder failure by fatigue. Since many identical parts are currently in operation worldwide, the objectives of the activity summarized in this paper were: (i) explanation of the unexpected failure and evaluation of probability for additional failures; (ii) demonstration that the part failure could be predicted and (iii) development of improved and alternative designs to achieve a considerably longer operational life.

The paper is organized as follows: initially the hydraulic cylinder under investigation is presented in terms of structure, function, geometry, material, service load, fabrication, and design details that are critical under fatigue loading. The current design is assessed and the motivation for criticality demonstrated by calculation. Alternative designs are proposed that maintain the critical detail but achieve a considerably longer service life.

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2. Fatigue failure investigation

2.1. Structure and service loads

The scheme of the hydraulic cylinder is shown in Fig. 1a. The hydraulic cylinder consists of a cylinder barrel, in which a piston connected to a piston rod moves back and forth. The barrel is closed on each end by the cylinder bottom (also called the cap end) and by the cylinder head where the piston rod comes out of the cylinder. The piston divides the inside of the cylinder in two chambers. The hydraulic pressure acts on the piston to do linear work and motion. The service load is alternate pressurization in the two chambers of the cylinder separated by the piston. The force developed by pressurization of the chamber is applied by the actuating rod through a mounting attachment connecting it to the machine part that it operates. A trunnion is mounted to the cylinder body to connect it to the machine frame.

The cylinder barrel is a seamless thick-walled forged steel pipe (i.e. cylinder bore D = mm and cylinder thickness t = mm) that is machined internally (i.e. ground and honed). In most hydraulic cylinders and in the present case as well, the steel barrel and the steel end cap are welded together. Welded cylinders have a number of advantages. Welded cylinders have a narrower body and often a shorter overall length enabling them to fit better into the tight confines of machinery. The welded design also lends itself to customization.

Cyclic pressurization was the main service load seen by the multi-pass welded joint depicted in Fig. 1b. The present application was characterized by two such cylinders operating in parallel a scrap steel press machine 24 h/day at the full design pressure of 280 bars. An oil leak was unexpectedly found at the welded joint shown in Fig. 1b by the operator and reported to the manufacturer leading to the present investigation. The estimated service life was approximately 40,000 cycles (i.e. 1000 service hours and 40 cycles/h).

2.2. Material and reference data

The material of the cylinder and of the end cap is a low carbon pearlitic steel, E355 and S355 JR EN 10297-1 respectively, commonly used for this application. Typical static properties for such steel are reported in Table 1. Fatigue fracture mechanics was used to assess structural integrity in this work. Therefore, reference fatigue crack growth material constants for pearlitic steel within the framework provided by the Paris law.

$$\frac{da}{dN} = C\Delta K^m \tag{1}$$

were also found in the literature, [1], and are reported in Table 1. The threshold stress intensity factor ΔK_{th} was also considered for residual strength assessments. The threshold stress intensity factor is known to depend on different parameters in addition to material strength, namely load ratio and crack length, [2]. Crack closure concepts are often invoked to explain the local mechanisms that hinder crack propagation by shielding the crack tip from full load effect. A rather conservative value for $R = 0$ for constructions steel was taken from the literature [3], and is given in Table 1.

2.3. Welded construction

The critical detail is the cylinder-to-end cap welded connection shown in Fig. 1b, which is subjected to fatigue due to cyclic pressurization. The detail of the welded joint including important dimensions of the welded joint is shown in Fig. 2. The chamfered end of the cylinder is positioned axially with respect to the chamfered end of the end cap via a step shoulder (i.e. dimensions d and c in Fig. 2 define it). It is a standard weld design favoring easy barrel-cap relative positioning and a strong connection via multi-pass weld deposition. The designer prescribes the dimensions c and d.

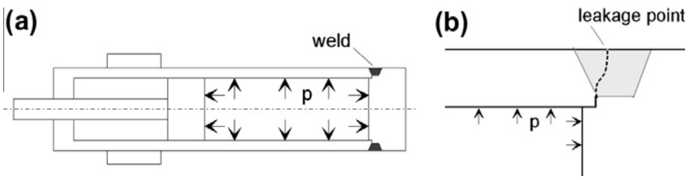


Fig. 1. (a) Scheme of the hydraulic cylinder; (b) detail of circumferential multi-pass weld and unexpected leakage location.

Table 1
Mechanical properties of the cylinder steel.

Material	Yield stress R_e (MPa)	Ultimate stress R_m (MPa)	Elongation (A%)	C	m	ΔK_{th} (MPa $\sqrt{m}$)
Construction steel	383	573	20	1.3×10^{-11}	3	8

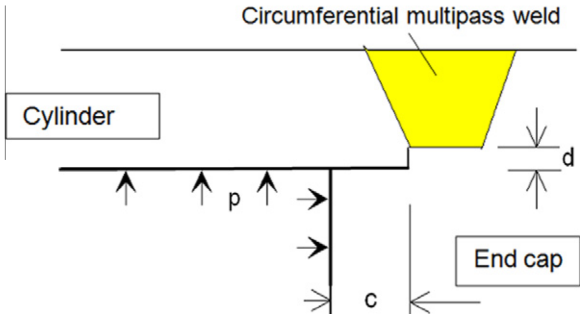


Fig. 2. Detail of the welded joint.

Since unexpected losses of oil in several cylinders were reportedly found on the outer joint surface by users after some estimated 40,000 duty cycles, the material discontinuity due to the fabrication process was considered as fabrication defect, which under unfortunate conditions could propagate as in the case discussed here. The present hypothesis was therefore investigated using fracture mechanics calculations with the aim of demonstrating that indeed such a failure and useful life could be predicted.

2.4. Finite element modeling and SIF determination

The geometry under study, however, presented an initial crack configuration for which no stress intensity factor solution was available. The finite element method was therefore applied to develop a structural model of the different crack configurations of interest using axisymmetric plane elements. The elastic material assumption is appropriate since the small scale yielding condition applies to fatigue loading level. Mixed-mode fracture mechanics concepts were required as the discontinuity was intrinsic to the welded joint.

Two cases were assumed: the first (Type H) that the discontinuity has the d and c dimensions given in the part drawing, the second that weld deposition reduced $d = 0$ (Type V). The two types of crack of Fig. 3 were investigated by FEM using the discontinuity dimensions and the corresponding stress intensity factors computed. A

The Mises stress distribution for the two crack configurations shown in Fig. 3 reveals a local singularity and a tilt of the iso-stress lines with respect to the discontinuity plane. Fracture mechanics identifies a mixed-mode loading condition, which

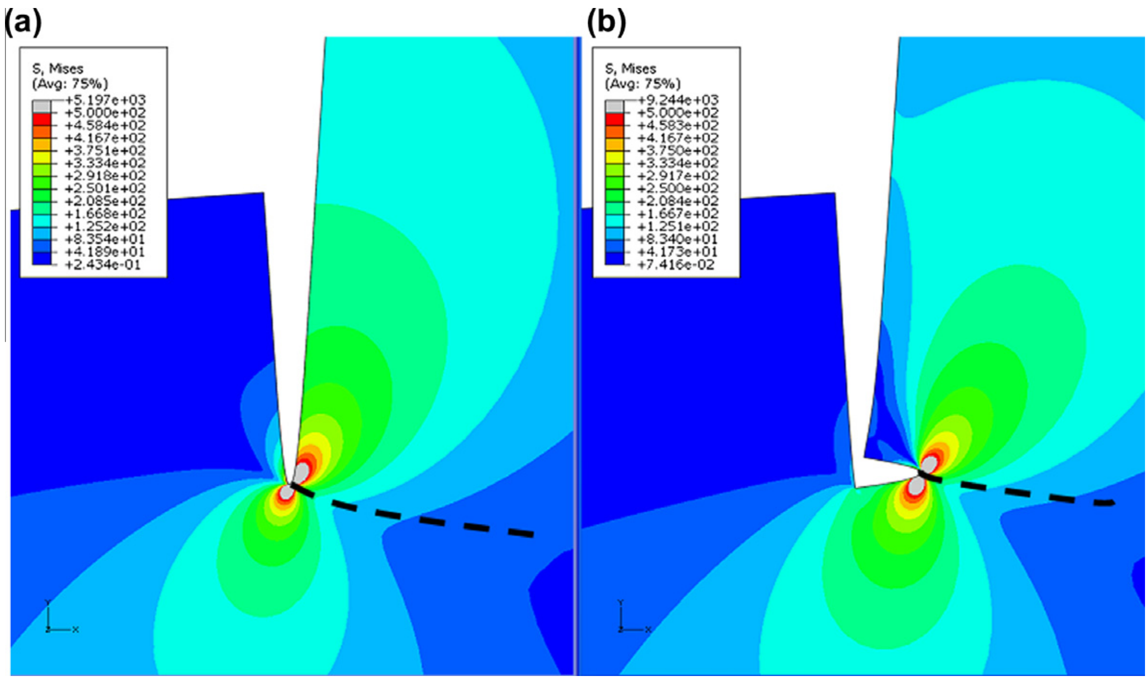
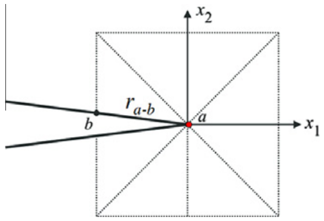


Fig. 3. Local deformation, elastic stress distribution and crack path for (a) V crack; (b) H crack. Scale ×100 (note that models are rotated 90° clockwise respect to Fig. 2).



$$K_I = \frac{\mu\sqrt{2\pi}(v_b - v_a)}{\sqrt{r_{a-b}}(2-2\nu)}$$

$$K_{II} = \frac{\mu\sqrt{2\pi}(u_b - u_a)}{\sqrt{r_{a-b}}(2-2\nu)}$$

$$K_{III} = \frac{\mu\sqrt{\pi}(w_b - w_a)}{\sqrt{2r_{a-b}}}$$

Fig. 4. (a) Node referred to the crack tip; (b) Mixed-mode stress intensity factors as a function of relative nodal displacements.

affects crack initiation and fatigue crack path. Mode I crack loading is actually affecting crack advance, while the other two modes have a strong influence on crack direction. The Mode II loading is expected to mainly steer the crack out of plane [4], as shown schematically in Fig. 3.

Determination of K at the crack tip can be obtained using either special singular elements or matching crack surface displacements to the elastic displacement solution for a mixed mode loaded crack [5]. The strategy used here is explained using the scheme of Fig. 4 where K is determined on the basis of the relative displacement components u, v, w in the reference coordinate system $x1$, $x2$ e $x3$ centered at the crack tip given by LEFM asymptotic field equations.

The maximum hoop stress criterion was applied to determine the deflection angle θ of the propagating crack and the effective stress intensity factor ΔK_{eff} was estimated using to the following formula [6],

$$\Delta K_{\mathrm{eff}} = \frac{1}{2}\cos\frac{\theta}{2}[\Delta K_{\mathrm{I}}(1+\cos\theta) - 3\Delta K_{\mathrm{II}}\sin\theta] \tag{2}$$

The results are reported in Table 2 for the service operating pressure (i.e. 280 bars). The H type crack has a higher K_{I} than type V crack while the K_{II} response is the opposite for the two crack configurations. In both crack cases, ΔK_{eff} is significantly larger than the ΔK_{th}, therefore crack propagation and early failure of the cylinder could be expected and predicted.

2.5. Estimation of useful life of the original design

An estimate of the residual life of the cylinders containing the present discontinuities was obtained by determining the K evolution through the cylinder wall by FEM and integrating the Paris' law for the structural steel, Eq. (1). The calculated residual life for type H crack was 41,000 cycles, which is in reasonable accord with estimated service before failure detection. The residual life of the Type V crack is expected to be considerably longer as the re-orientation phase adds a significant number of cycles to the Type H residual life.

The present elastic analysis could be readily used to determine the influence of a reduction of the maximum operating pressure that would eliminate the possibility of unexpected early failure by either reducing ΔK_{eq} below ΔK_{th} or, more reasonably, accepting a ΔK_{eq} low enough to significantly extend the operating life. However, this approach would be applicable only for cylinders in operation. The development of an improved cylinder design was then undertaken and I reported in the next section.

3. Proposed cylinder re-design

The application of fracture mechanics concepts in the previous section demonstrated that the weld joint design of the cylinder was prone to fatigue failure due to crack propagation from the local discontinuity. Therefore, the same concepts were used in proposing a re-design of the cylinder-end cap connections.

The finite element method and mixed-mode fracture mechanics were used in a damage tolerant approach to modify the local geometry (i.e. size of the discontinuity, scarf type, end cap geometry) reducing the local stress intensity below the threshold value for fatigue crack propagation thus making the present weld fabrication solution still viable.

The proposed re-design still assumes the weld joint fabrication approach depicted in Fig. 2 because it is convenient technologically but modifies the end cap geometry according to the scheme of Fig. 5a and b shows that the increased flexibility of the weld connection results in a reduced bending stress component. Fig. 6 shows the local deformation of the original and modified discontinuities under same magnification of 200X.

3.1. SIF determination for improved designs

As Mode I stress intensity factors are obtained from relative crack face opening displacements, the crack profiles of different end cap geometries shown in Fig. 7 immediately reveal the degree of improvement that can be obtained. Inspection of Table 3 reveals that a series of alternative proposals all reduce drastically local stress intensities 3–4 times those of the original design summarized in Table 2. The $K_{\mathrm{I}}/K_{\mathrm{II}}$ ratios indicate the relative mixed-mode loading contribution.

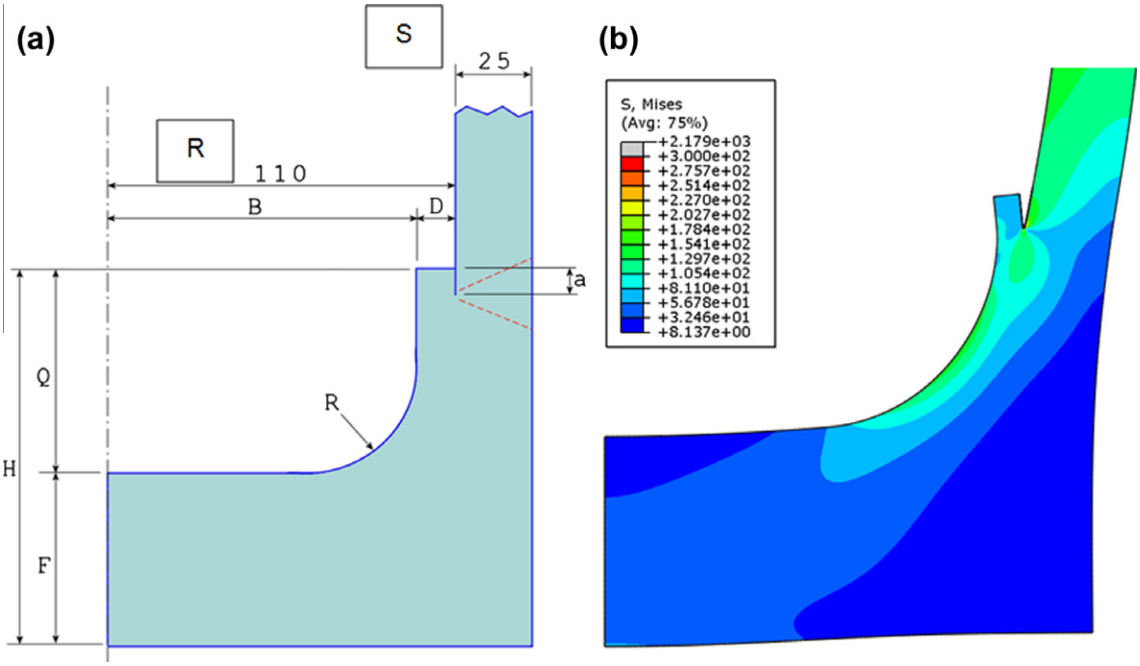


Fig. 5. (a) Modified end cap geometry with dimensions for parametric study; (b) elastic stress distribution for a configuration.

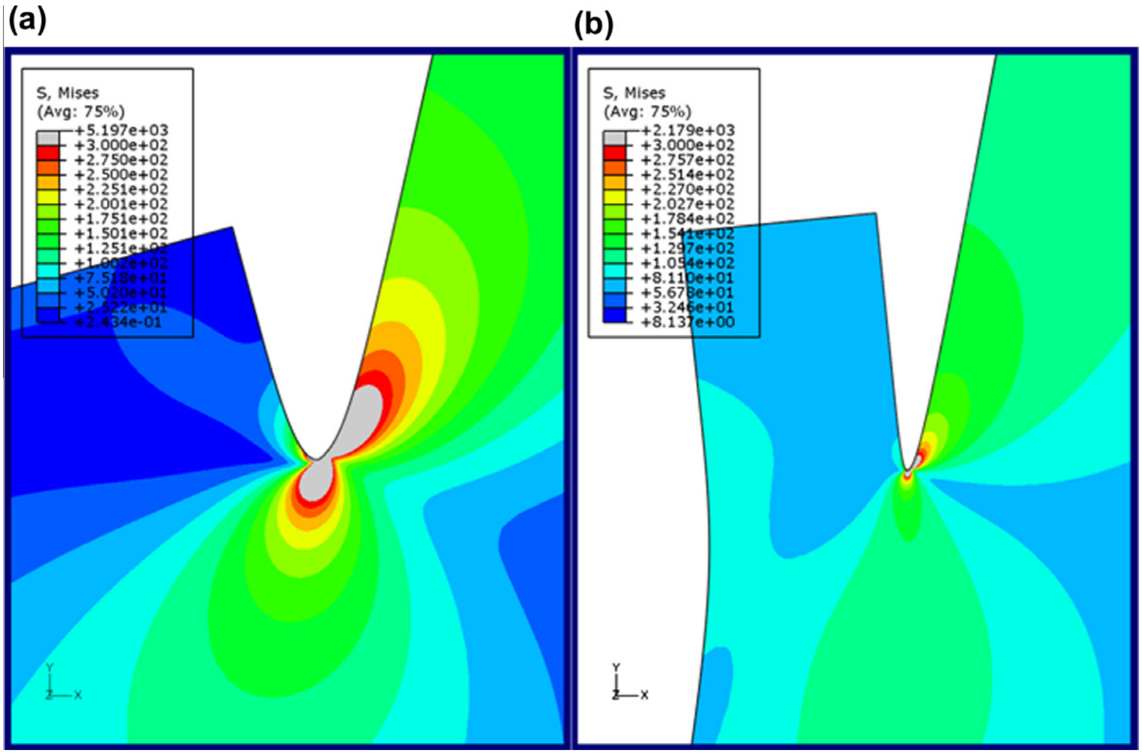


Fig. 6. Local stresses and deformation of (a) original cap design; (b) improved end cap design.

3.2. Service life estimate for selected design

The service life was estimated with reference to the design solution #7 in Table 3. The simplifying assumptions are that Mode I crack propagation through the cylinder wall of a circumferential crack described by the Paris law. A Type V

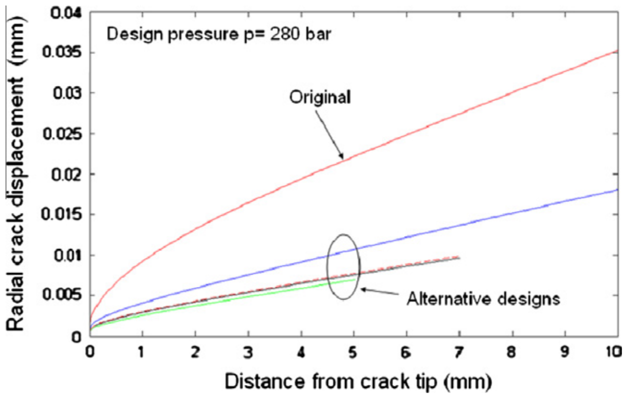


Fig. 7. Comparison of the opening displacement component of the original design and of alternative end cap designs.

Table 2
Stress intensities for the two crack configurations, see Fig. 3.

Pressure (bar)	Crack type	K_I (MPa $\sqrt{m}$)	K_{II} (MPa $\sqrt{m}$)	$\|K_I/K_{II}\|$	θ (deg)
280 bar	V	15.4	−9.5	1.6	43.8
	H	23.2	6.6	3.5	−27.9

Table 3
Parametric study of influence of the modified end cap geometry on local stress intensity factors (design pressure p = 280 bar, R = 110 mm and S = 25 mm).

Design case	H (mm)	a (mm)	Q (mm)	B (mm)	R (mm)	K_I (MPa *sqrtm*)	K_{II} (MPa *sqrtm*)	$\|K_I/K_{II}\|$	θ (deg)
1	120	10	50	100	40	8.21	−5.67	1.45	46.7
2	120	10	60	100	40	6.99	−4.75	1.47	46.4
3	120	10	60	105	40	4.95	−2.97	1.67	44.0
4	120	5	60	105	40	3.67	−2.99	1.23	49.7
5	120	5	65	105	45	3.61	−2.90	1.24	49.5
6	120	5	60	102.5	40	4.74	−3.87	1.23	49.8
7	120	10	60	102.5	40	6.05	−3.99	1.52	45.9

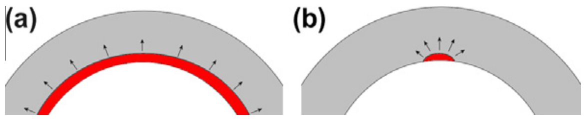


Fig. 8. (a) assumed crack propagation in cylinder; (b) semi-elliptical crack propagation.

discontinuity is assumed as appropriate counter measures are taken to avoid the formation of a Type H discontinuity. The re-orientation phase is assumed to add life cycles to the lower-bound life estimate associated to circumferential crack growth through the cylinder wall. Therefore, FE determination of SIF for a crack of increasing length is obtained using the approach described in the previous sections and used in combination with the fatigue crack propagation law for the present steel. In this way, design #7 is determined to have a service life of approximately 300,000 cycles, almost an order of magnitude longer than the original design.

As a final comment, the crack propagation in FE models and calculations was assumed to occur radially along the entire circumference, see scheme of Fig. 8a. Oil leaking however was observed on a limited ark of the outer cylinder perimeter, an indication that possibly crack growth patters resembled the semi-elliptical surface crack propagation schematically shown in Fig. 8b.

4. Conclusions

A case of unexpected service failure of a heavy-duty hydraulic cylinder motivated the present investigation and subsequent re-design activity. Fatigue fracture mechanics concepts supported by finite element analysis were used to demonstrate that the in-service failure could have been predicted although the required methodologies are not widespread among designers in industry. The same concepts and tools were then successfully used to develop and propose a re-design of the hydraulic cylinder requiring limited modification to the original solution that increased the predicted service life of almost an order of magnitude.

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