THE STRENGTH OF SHOT PEENED PARTS
Design Calculations and Specifications

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ABSTRACT

This paper presents a method for predicting the fatigue strength of shot peened parts at any given fatigue life. The method considers the different criteria for crack initiation, arrest of small cracks, and change of the self-stress by yielding. Guides are given for choosing a good peening intensity and for specifying the desired result.

KEYWORDS

Shot peening; fatigue strength; self-stress; bi-axial stress; Haigh diagrams; fatigue strength prediction; crack arrest; yielding; Almen intensity.

INTRODUCTION

The savings obtainable by the use of shot peening are so great that designers must include its effects in their calculations if they want to remain competitive. In automobile suspension coil springs for instance, the permissible stress range with shot peening is 40% greater than without peening (SAE 1973). This means that shot peened springs weigh half as much as unpeened springs would weigh because the weights of springs of equal capacity are (approximately) inversely proportional to the squares of the permissible stresses. The saving achieved by shot peening is about 24kg per average car and worth perhaps $20 for suspension springs alone, without the savings on valve springs, gears, and shafts, and without the indirect savings produced by using smaller, lighter components.

Values of permissible stresses for peened and unpeened coil springs are given in the Manual on Helical Springs (SAE 1973) and in the Automotive Handbook (Bosch 1976). Similar data are available for only very few types of parts and materials. It is the purpose of this paper to provide the designer with a simple method for estimating the stresses permissible with shot peening for parts which he is designing.

OUTLINE OF THE METHOD

The method proposed here will give more realistic estimates than the often-used Goodman diagram. For designs which involve danger to life, or large losses in case of failure, the estimates achieved by this method should, of course, be verified by tests, or used with suitable safety factors, keeping in mind that usual S-N curves indicate median fatigue lives or 50% failure expectation.

Any useful method of calculation must be able to take account of three key observations about the effect of peening:

1. Peening has more effect at long fatigue lives than at shorter fatigue lives.
2. Peening has more effect on high strength materials than on low strength materials.
3. Peening has far more effect on notched parts or on parts with crack-prone surfaces (for instance decarburized or chromium plated) than on nicely finished smooth specimens. Peening can often overcome the notch effect, so that peened parts with notches can be as strong in fatigue as smooth, un-notched, parts. (Harris 1961).

These observations lead to four features of the calculations:

1. The fatigue strength at intermediate fatigue lives is obtained by interpolation between the strength at 10 million cycles, where peening has a great influence, and the strength at 1000 cycles where peening has very litte or no influence.
2. The effect of both steady and alternating stresses on yielding is included in the computations.
3. The strong effect of peening on notched parts is derived from the ability of compressive stresses to prevent the growth of cracks.
4. The effect of peening on smooth polished parts is related to the ability of compressive stresses to delay the appearance of cracks.

PREDICTION OF LONG LIFE FATIGUE STRENGTH

The prediction for 10 million cycles can be done on a Haigh diagram, on which alternating stresses S_a are plotted over mean stresses S_m (Haigh, 1930). Figure 1 shows the condition for zero crack propagation plotted on such a diagram. It represents the equation

$$S_a = 2S_{cat} - S_m, \text{ or } S_a = S_{cat}, \text{ whichever is greater.} \qquad (1)$$

For all combinations of mean stress and alternating stress below the line 3-4 there will be no crack propagation. All combinations of a maximum stress with zero minimum stress (R=0) are on line 1. All combinations of a minimum stress with zero maximum stress are on line 2. Line 3 indicates a constant maximum stress equal to $2S_{cat}$.

If a load fluctuates from zero to a maximum and the part (for instance a cantilever spring) is free of self-stresses, there can be crack propagation for any alternating stress greater than at A. If we impose a compressive self-stress P by peening, then the alternating stress can be as high as at B without producing crack propagation. Line 5 indicates the stress conditions produced by loads cycling from zero to any maximum when the self-stress P is present. The steep slope of line 3 corresponds to the strong effect of shotpeening on crack-prone parts.

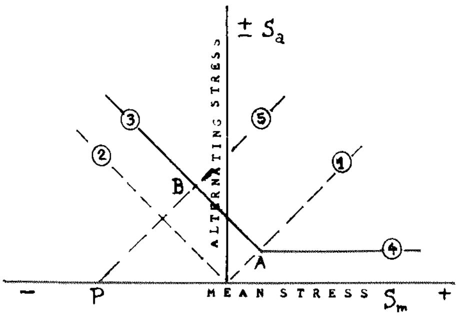


Fig. 1. Haigh diagram for resistance to propagation of small cracks

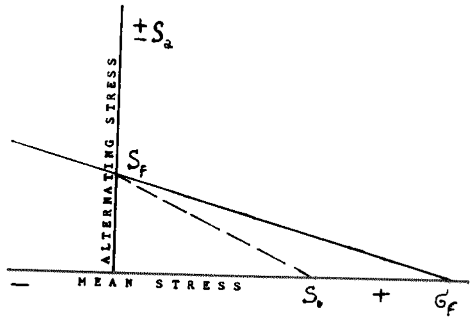


Fig. 2. Haigh diagram for resistance to crack initiation

Nicely finished smooth specimens almost always require alternating stresses above the line 3-4 to start cracks. Under those conditions cracks which have been started will always propagate to failure. The effect of peening on such parts can be predicted for long fatigue lives by the relation of the fully reversed fatigue limit S_f to the true fracture strength σ_f (Morrow 1968).

$$S_a = S_f\ (1 - S_m/\sigma_f) \qquad (2)$$

On a Haigh diagram this relation is shown in Fig. 2. It shows, for comparison, also the well known modified Goodman relation as a dashed line from S_f to the ultimate tensile strength S_u. The slope of the Morrow line is less steep than the slope of the crack propagation line in the region of compressive mean stress.

The two relations explained above - for crack propagation and for crack initiation - must be modified if the yield strength is exceeded. In particular, the self-stresses may be changed by yielding. The monotonic tensile yield strength is generally known. The compressive yield strength is assumed to be the same. The cyclic yield strength, for fully reversed loading, is available for some materials (SAE 1981a). One assumes that the cylic yield strength for cycles which are not fully reversed can be interpolated as shown on the Haigh diagram Fig. 3.

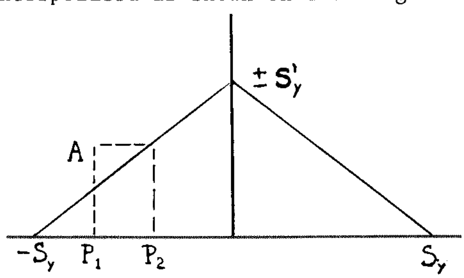


Fig. 3. Alternating stress ± A changes self-stress from P_1 to P_2

In the region outside the triangle - S_y, S_y', S_y the yield strength is exceeded by the combination of mean stress and alternating stress. If this happens, the mean stress will relax until

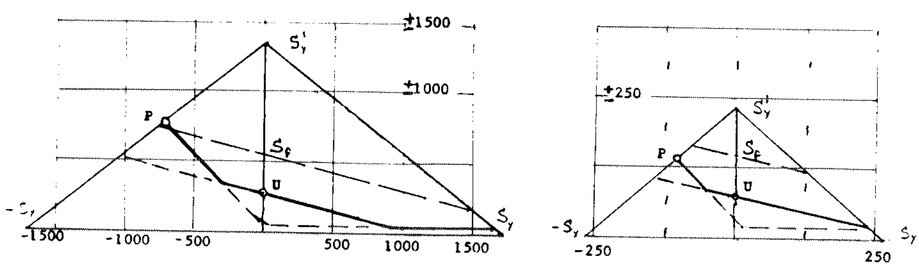
$$|S_m| + |S_a| = |S''| \qquad (4)$$

where S'' is the yield strength given by a point on the straight line from $-S_y$ to S_y' or from S_y to S_y'. In Fig. 3 the alternating stress ± A causes the self-stresses to relax from P_1 to P_2.

For yielding and for the propagation of cracks we consider only the nominal stresses (Fuchs, 1972). In a notch it is quite possible that there will be some yielding even at loads low enough to permit ten million cycles or more on a shot peened part. But the small cracks produced by the repeated yielding will not be able to propagate through compressed material below the notch root where the applied stress is lower. For this reason we do not use a stress concentration factor for the yield condition. For the arrest of small cracks we follow the practice of fracture mechanics in using nominal stresses. For crack initiation, however, we must allow for stress concentration. We do this by dividing the nominal stresses, both S_a and S_m, by the notch factor K. This leads to combined diagrams such as Fig. 4.

Examples

Figure 4 and Table 1 show the effect of peening on two parts with notches which produce a fatigue strength reduction factor K = 2, one made of hard steel, the other of mild steel. The diagrams are constructed from the data given in the first six lines of Table 1. The last lines list the strength obtained from the diagrams for 10 million cycles of fully reversed loading.



4a) 4142 steel at 1930 MPa UTS 4b) 1020 steel at 440 MPa UTS

Fig. 4. Haigh diagrams for steel parts with notch factors K = 2, peened (P) and unpeened (U), for two different materials. Note different scales.

TABLE 1 Estimates for Fatigue Limits of Peened and Unpeened Parts of Two Materials

Line #			4142 MPa	1020 MPa	4142 % of S_u	1020 % of S_u
1	S_u	Ultimate tensile strength	1930	440	100	100
2	σ_f	True fracture strength	2170	710	112	161
3	S_y	Tensile yield strength	1725	260	89	59
4	S'_y	Cyclic yield strength	1345	240	70	55
5	S_f	Fatigue strength at 10^7 cycles	570	150	30	35
6	S_{cat}	Critical alternating tensile stress	58	13	3	3
7	U	Unpeened notched fatigue strength	285	75	15	17
8	P	Peened notched fatigue strength	780	135	40	31

Some Conclusions From This Method

Figure 4 and Table 1 show several relations which correspond to the experience of long life fatigue testing.

The shot peened part of hard steel will survive alternating stress cycles of $\pm$ 780 MPa (Point P). The unpeened part can survive only $\pm$ 285 MPa (Point U). The improvement is 170% or 495 MPa. A smooth specimen of the hard steel would survive stress cycles of $\pm$ 570 MPa (Point S_f). Peening could increase this also to $\pm$ 780 MPa. The improvement is only 37% or 210 MPa.

The peened notched part has fatigue strength equal to that of an un-notched specimen.

For the same part made of mild carbon steel the improvement obtainable by shot peening is less in relative terms (80%) because the yield strength at point P is only 165% of the fatigue strength S_f while for the hardened 4142 steel it was 270% of the fatigue strength S_f. The improvement is far less in absolute terms, only 60 MPa for the mild steel part instead of 495 MPa for the part made of 4142. For parts made of mild steel more improvement than by peening can be obtained by changing to harder steel.

Another Method

Diagrams such as Fig. 4 give excellent insight of the effects produced by peening and are easy to construct. The same results can, of course, be obtained by electronic computation based on the same relations between yielding, mean stress and alternating stress. A somewhat simpler computation (Smith, Topper and Watson 1970) is available which leads to similar results for K > 2 (Fuchs and Stephens 1980 p. 154). The formula for that method is

$$S_f^2 = K^2 \; (S_m + S_a) \; S_a \text{ or } S_a = (S_f/K)^2/S_{max} \tag{4}$$

The limitation imposed by yielding must be separately checked, as before. For small values of K equation 4 tends to overestimate the effect of mean stress.

A Quick Method

A more rapid method of obtaining a quick approximate estimate of the long life fatigue strength obtainable with good shot peening is based on the location of point P in Fig. 4 almost exactly in the middle of the line which joins $-S_y$ and S_y'. This leads to

$$S_a \simeq 0.25 \; (S_y + S_y') - 0.125|S_y - S_y'| \simeq 0.5 \; S_y \tag{5}$$

where S_a is the highest nominal alternating stress which a properly peened notched part can survive for 10 million cycles.

PREDICTION FOR SHORTER FATIGUE LIVES

Fatigue strength at any number of cycles can easily be estimated if the strengths at a long life and at a short life are known. It can be done graphically by connecting the 2 known points by a straight line on a log S - log N diagram or numerically by

$$S_x/S_y = (N_y/N_x)^p \quad (6)$$

and

$$p = \log (S_k/S_f)/\log (N_f/N_k) \quad (7)$$

where S_i is the fatigue strength at N_i cycles and S_k the known or estimated fatigue strength at a small number of cycles N_k. The exponent b (= - p) given in the literature (SAE 1981a) applies only to smooth specimens. For notched parts p is always greater than for smooth specimens of the same material.

We find the exponent for notched parts by assuming that for ductile materials the fatigue strength at 1000 cycles is the same for smooth specimens and for notched parts, whether peened or not peened. The fatigue strength of smooth specimens at 1000 cycles (S_{1000}) can be estimated in several ways:

For the materials listed by SAE (1981a)

$$S_{1000} = \sigma_f'/2000^p \quad (8a)$$

where the fatigue strength coefficient σ_f' is the extrapolated fatigue strength at 1/2 cycle and p is the absolute value of the listed exponent b.

Shigley (1977) approximates

$$S_{1000} = 0.9\ S_u \quad (8b)$$

If a cyclic stress-strain curve is available one can estimate

$$S_{1000} = \sigma \text{ for } \varepsilon = \pm 1\% \quad (8c)$$

Estimates for the fatigue strength at 1000 cycles, obtained from these equations for the materials of the example in Fig. 4 and Table 1, are listed in Table 2.

Table 2 Estimates of Fatigue Strengths at 1000 Cycles for Two Materials

Material		4142	1020	
S_u	Ultimate tensile str.	1930	440	MPa
S_f	Fatigue str. at 10^7 cycles	570	150	MPa
σ_f'	Coefficient	2170	895	MPa
b	Exponent	-0.081	-0.12	
S_{1000}'	$= \sigma_f'/2000^{-b}$	1172	396	MPa
S_{1000}	$= 0.9S_u$	1737	396	MPa
S_{1000}	$\varepsilon = \pm 1\%$	1406	312	MPa

With the fatigue strenghts at 10 million cycles and at 1000 cycles estimated in Table 1 and Table 2 one can construct the estimated S-N curves for median lives. The need to design for more than 50% probability of survival and the differences in the estimates in Table 2 remind us of the need to choose a design strength lower than the calculated median strength.

BI-AXIAL LOADING

The method explained above applies to uni-axial stresses as observed in parts subject to bending or tension. For bi-axial stresses such as torsion or combined torsion and bending one must use an expanded set of assumptions, illustrated in Fig. 5.

Figure 5 is drawn with the two principal stresses as coordinate axes. Any point represents a state of stress. Uni-axial stresses are located on the coordinate axes. Pure torsional stress superimposed on bi-axial compression is represented by the arrows OQ which have their center at the bi-axial compressive stress O. The solid line ellipse represents the end points of all alternating stress arrows starting at 0 which correspond to the same equivalent alternating stress as OQ according to the octahedral shear stress theory. The dash line ellipse represents the yield strength according to the same theory. The dash-dot lines C-C represent the limits of crack propagation based on a critical alternating tension stress.

In such a diagram the dash-line yield ellipse stays fixed, but the center of the solid line ellipse moves to the point representing the mean stress.

The size of a fatigue strength ellipse is given by those points which represent uni-axial fatigue loadings; they are located on the horizontals and verticals through the origins of the ellipses and are determined as explained for uni-axial loading in Fig. 2 except that the sum of the mean stress components σ_1 and σ_2 is used as mean stress.

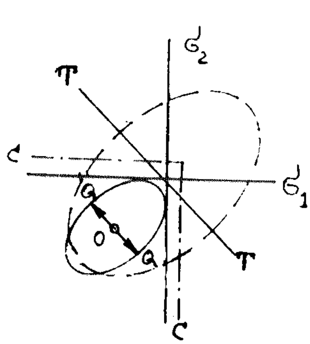


Fig. 5. Fatigue Criteria in a Bi-axial Stress Field

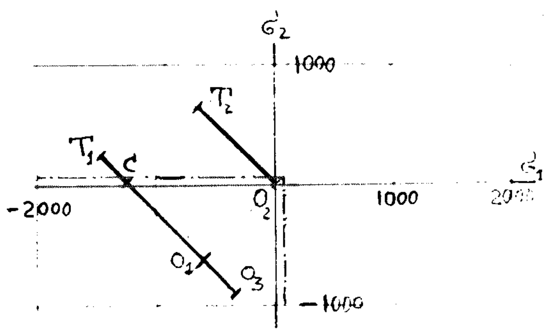


Fig. 6. Fatigue Strength of a Torsion Bar With and Without Shot Peening and Presetting

An Example

Figure 6 shows the application of this method to a torsion bar of 4142 steel, heat treated to 1930 MPa ultimate tensile strength, and shot peened to have a biaxial compressive stress of 600 MPa. With this compressive stress the uni-axial fatigue strength would be 720 MPa as shown on Fig. 4a. (It was 570 MPa without shot peening). The bar will be loaded from zero to a maximum load. We want to estimate the maximum permissible torsional load.

Point 0_1 in Fig. 6 is the stress condition for zero load. The uniaxial fatigue limit is 720 MPa; the corresponding torsional stress is 0.577 x 720 = $\pm$ 415 MPa or a range of 830 MPa, corresponding to the range from point 0_1 to point T_1. This would be the strength of a bar as smooth and free of decarburization as a laboratory specimen. As we cannot expect this, we estimate the smaller stress range of 620 MPa slightly below point C, the intersection with the critical alternating tensile stress which just permits propagation.

Without shot peening a bar with perfect surface would have been limited to a torsional stress range of 660 MPa from 0_2 to T_2. A less perfect bar would have no protection against crack propagation and would require a considerable reduction in stress range to avoid the initiation of fatigue cracks from surface imperfections or from decarburization. An increase in stress range beyond that of the shot peened bar can be obtained by torsional presetting, shifting the origin, at zero load stress, to point 0_3. The stress range of 920 MPa from 0_3 to T_1 would initiate cracks, but they would not propagate.

DEPTH OF PEENING

The discussion so far was concerned with surface stresses. Protection of the surface is most important because it is exposed to harmful influences of corrosion and of scratches from machining or rough handling; the surface may also be less resistant to fatigue because the elements of material on the surface are not completely surrounded by other elements which help to restrain and strengthen them. In addition, the load stresses are highest at a surface in almost all parts. However, failures on shot peened parts may occur below the surface, at stresses higher than those which would produce failures on unpeened surfaces. It is therefore desirable to shot peen at intensities high enough to protect sub-surface material against fatigue failures.

Figure 7 shows the relation between peening intensity and depth of compressive self-stress for several materials.

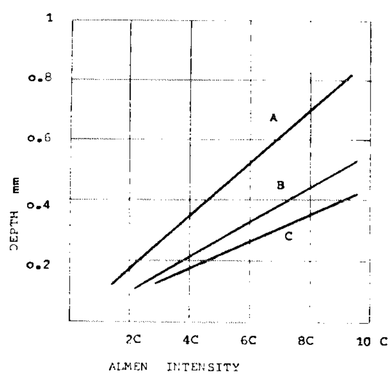


Fig. 7. Depth of Compressive Stress as a Function of Almen Intensity.

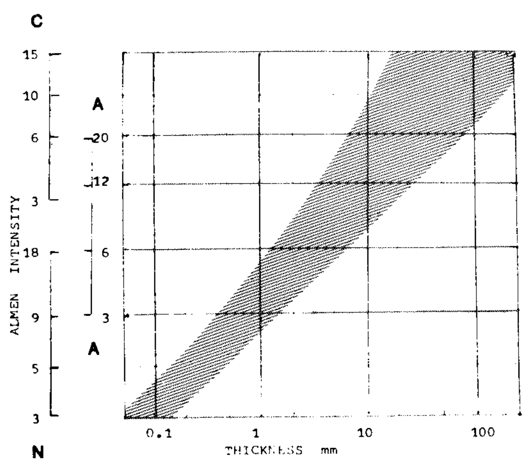


Fig. 8. Almen Intensities with Steel Parts of Different Thickness.

(Courtesy of Metal Improvement Co.)

CHOICE OF INTENSITY

The choice of intensity depends on several factors:

Cost. - Higher intensity usually requires more time to achieve full saturation.

Tensile stress. - The compressive stress in material near the surface is in equilibrium with tensile stress in the core. If 5% of the area has a self-stress which averages 1000 MPa, then the remaining 95% will be stressed in tension to an average 53 MPa before any load is applied. This is harmless. If the ratio of areas were 30/70 the average tensile self-stress would be 430 MPa which would be cause for concern. The part would be overpeened. Sharp edges are easily overpeened and should be avoided on peened parts.

Stress gradients. - Less depth of peening will be required if the load stresses decrease rapidly below the surface, as they do for instance in sharp notches.

Surface composition. - If a steel part has a shallow surface layer depleted of carbon and therefore lower in hardness, then the peening intesity must be high enough to produce compressive residual stress in the harder material below the decarburized skin.

If all factors are known, one should be able to optimize the peening intensity. The optimum is obtained when failures below the surface are just as likely as failures at the surface of the peened part. A rough guide from experience with hard steel parts is shown in Fig. 8.*

SPECIFICATION OF PEENING PARAMETERS

When a designer knows that shot peening is desirable he must specify what he wants. He will want his specification to be as complete, clear, and concise as possible. The Almen intensity (SAE 1981b) is the accepted and the best means of specification, but may not always be sufficient, as discussed below. The Almen intensity specification consists of a number and a letter, A, C, or N, for instance 10C. (This is a convention, like the hardness callout R_c 45, not a measure convertible from inches to millimeters.)

The Almen intensity alone is sufficient specification for steel parts of hardness between R_c 35 and R_c 55. For parts made of other materials it may be advisable to specify also the size and nature of the shot. The Almen strip itself is made of steel of hardness R_c 47 so that the effect of peening on the strip duplicates the effect on steel parts of similar hardness within practical limits. On parts which are harder or softer, however, the size and hardness of the shot may produce differences.

On harder steel, for instance, shot of greater hardness produces more effect than softer shot even when both produce the same intensity measured by the Almen strip. On aluminum parts large shot produces more effect than small shot at the same Almen intensity. (Fuchs 1957).

These specifications will produce a desired depth of compressive stress. The magnitude and the exact distribution of the compressive stress are not subject to control by specification but determined by the response of the material to the treatment which produces the desired depth.

Surface roughness can be controlled to some extent. Larger shot or softer shot will produce less surface roughness for the same depth of compressive stress, but at the expense of longer time required for saturation.

*Distortion may become a factor at small thickness or high intensity.

CONCLUSION

Although the analysis presented in this paper is only approximate and does not cover everything observed in shot peening (for instance the effects of peening after 100% coverage has been achieved), it explains why the compressive stresses produced by peening are so highly effective on notched parts of hard material at long fatigue lives and it permits good esimates of the effect of shorter lives, and for less severely notched parts made of softer material. Use of such a strength prediction method in design will result in more confident application of shot peening to produce stronger parts.

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