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# FATIGUE LIFE OF STRESS-PEENED HELICAL COMPRESSION SPRINGS

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This project was accomplished as part of the U.S. Army's Manufacturing Methods and Technology Program. The primary objective of this program is to develop, on a timely basis, manufacturing processes, techniques, and equipment for use in production of Army materiel.

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

Stress peening variables (i.e., shot peening while under applied stress) are studied for their effect on fatigue life of helical compression springs. Comparisons are made with the effects of conventional shot peening. In terms of fatigue life, stress peening gives increases ranging from 1.4 to 5.8 times the life of conventionally peened springs. Increases were greatest for the smallest springs.

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## INTRODUCTION

The process of shot peening has been used for nearly fifty years to increase the resistance of surfaces to the formation and propagation of fatigue cracks. The process involves the impingement of shot upon the surface at sufficiently high velocity to cause permanent localized deformation of the surface. The result of the deformation caused by a large number of shot striking the surface is the introduction of a layer of compressive residual stress. This layer of compressive stress tends to keep closed any cracks which might be forming or propagating. This tendency is reflected in improved fatigue properties; i.e., longer life at a given cyclic stress condition or higher permissible cyclic stress conditions for a given required life. The bibliography lists a number of papers which discuss the shot-peening process and its benefits in greater detail.

The process of stress peening is similar to conventional shot peening except that the component being peened is subjected to an applied stress during the peening operation. This procedure can result in the creation of higher compressive residual stresses than would be obtained without the stressing. The subsequent superimposed service stresses then result in a lower total stress and, consequently, enhanced fatigue properties.

A more detailed description of the effect of stress peening is given in the body of this report.

The objective of the present study was the determination of any benefits of stress peening on the fatigue properties of helical compression springs. Any improvement in fatigue strength could be translated into increased service life or into savings in spring weight, size and, possibly, cost in their use in armament applications.

## THE STRESS-PEENING PROCESS

### Conventional Shot Peening

The conventional shot-peening process involves the impingement of high-velocity shot against a surface which is in a free stress state, except for any residual stresses which may be present. The resulting surface stress is biaxially uniform, as depicted in Figure 1. The magnitude and depth of this compressively stressed layer is dependent upon several variables, as will be discussed in the next section of this report. Immediately below the compressive layer, there must be a region of tensile stress in order that the requirements of force and moment balance be met. The magnitude and distribution of this tensile stress is dependent on the geometry of the part. The magnitude is generally small and is, therefore, not often of consequence with regard to the initiation of fatigue cracks. An example of a typical distribution of residual stress through the thickness of a plate which has been peened on both faces is shown in Figure 2.

The benefits of shot peening are depicted in Figure 3. This is the classical case of superposition of externally-imposed bending stress and the residual stress induced by shot peening. It is seen that the maximum surface tensile stress is lower as a result of the peening. Since most fatigue failures originate in regions of tensile stress, this reduction of the stress proves to be beneficial. The elevation of tensile stress in subsurface regions is not very detrimental in most cases because, first, the elevation is usually small and, second, subsurface regions are not as susceptible to crack formation as are surface regions. Another potentially detrimental aspect of the peening arises from the fact that externally applied compressive stresses superimpose on the compressive peening stresses, resulting in an increased maximum compressive stress. This increased compressive stress is not generally of serious consequence since failures do not commonly originate in regions of such stress. If the resultant compressive stress is sufficiently high, yielding will occur and the maximum value will be limited.

In the case of externally applied torsional loads, the resultant of peening stresses and applied stress is dependent on direction. Figure 4 shows this relationship. It is seen that, for a given direction of applied torque, there results a maximum principal (tensile) stress in one $45^{0}$ direction and a minimum principal (compressive) stress in the other $45^{0}$ direction. When equiaxial compressive residual stresses are superimposed at the surface, it is seen that the resultant tensile stress is decreased and the resultant compressive stress is increased. The degree of reduction of tensile stress depends upon the relative magnitude of the residual peening stress and the externally applied torsion stress. The resultant may be tensile, zero or even compressive. The compressive stress in the orthogonal direction will always remain compressive under such zero-to-maximum torque loading. If the torque loading reverses in direction, the two orthogonal planes of principal stress will be subjected to identical ranges of stress with compression-tension superposition during one portion of the cycle and compression- compression superposition during the opposite portion of the cycle.

Stress Peening

The stress-peening process differs from the conventional shot-peening process in that the part is subjected to an externally-applied load during the peening operation. In the case of a flat plate (such as a leaf spring) this load would be a bending load so that one face was in tension and the opposite face in compression. If the tensile face is peened, the resultant stress while under load will be about the same as if the load had not been applied. When the load is released, however, the stress will go even more compressive. Thus, if the service load is basically unidirectional, such as in a vehicle suspension, the net stress on the tension side will cycle from a high compression when unloaded to a lower compression when loaded. The mean stress of the cycle is more compressive than it would have been without the stress while being peened. Fatigue properties are thereby enhanced by the stress peening.

The benefits of peening the side which is prestressed in compression are not as clear. In fact, the mean stress actually ends up in less compression than for conventional peening and the superposition of the applied stress and the peening stress does not represent an improvement as it does on the tension side. Thus, stress peening of beams is usually limited to cases wherein the service loads are non- reversing.

The situation in a torsion bar is somewhat analogous to that of a beam in bending except that the conditions on the two opposite faces of a bending bar are found in the two orthogonal planes of principal stress in the torsion bar. If the bar is subjected to torque while being shot peened, the resultant stress will still be of essentially uniform compression in all directions. When the external torque is released, the original direction of applied tensile stress will exhibit the sum of the peening stress and the release of tension for a net value of high compression. The orthogonal plane will exhibit the sum of the peening stress and the release of applied compressive stress for a net value of small magnitude, compression or tension depending on the exact values. When an external uni-directional cyclic service torque is applied, the stress will cycle from high compression to moderate compression (depending on exact loads) in the one principal stress direction and will cycle from near zero to moderate compression in the orthogonal direction. Thus, the direction which was originally subjected to high tensile stress is now held to modest compression at the peak of the cycle. The orthogonal direction, which was originally subjected to high compressive stress is now alternated around a less compressive mean stress, being near zero or even in tension when unloaded.

Figure 5 shows, schematically, the above effects.

It should be noted that the preceding discussion assumes that stresses do not become sufficiently high to result in yielding of the material (except for the local yielding caused by shot peening). If stresses do exceed the yield strength, the resulting patterns will be modified to a degree dependent on the degree of yielding. The most likely place for the yielding to occur is at the point (or in the direction) where the applied load superimposes compressive stress on the peening stress. This phenomenon represents a limitation to any potential benefits of shot peening, whether conventional or stress peening.

It is seen that the net result of stress peening may be nearly the same as for conventional peening for a unidirectionally-applied torsion cycle, except that the roles of the two $45^{\circ}$ planes are reversed. Thus, the benefits of stress peening of torsion bars are not easily predictable. They depend on the details of the treatment and the response of the material to such treatment. Thus, it becomes necessary to conduct actual peening and fatigue tests to determine the benefits which might be attained with a particular set of variables.

The preceding discussion has been centered around a straight torsion bar. The present study is centered around helical compression springs. Stress analysis shows that the stress in a helical spring is predominantly torsion with only a small degree of superimposed bending and direct shear in most springs. Thus, any stress effects in a straight torsion bar are nearly the same in a helical spring.

One other aspect of stress in a torsion spring or a helical spring is that of setting. This term applies to the yielding or permanent set which results from the application of a high load. Any such setting in service usually occurs early in the life of the spring. In order that spring dimensions be maintained, it is sometimes the practice to impose this setting prior to installation (then known as presetting). Presetting was not one of the variables of study in this project although, as discussed later, it may have played a part in the performance of some of the springs. It is mentioned here since it does result in some modification of the residual stress pattern and may, consequently, have some effect on fatigue life.

## TEST PROGRAM

### Test Parameters

In planning a program to assess the effects of stress peening on helical compression springs it is necessary to establish the various parameters and their ranges in order to insure that their effects are maximized and are correctly tested. The parameter values were selected on the basis of several factors such as the following:

Spring sizes of interest in armament applications.
Spring materials of interest in armament applications.
Shot sizes - Practical ranges for different part sizes.
Peening intensity - Experience from previous applications.
Peening coverage - Complete coverage in all cases.
Prestress levels - Practical range.
Fatigue test loads - Desired failure lives.
Fatigue test speeds - Available test equipment.

A test matrix was developed around the several parameters. The matrix is shown in Table 1. Details of various parameters in the matrix are discussed in succeeding sections of this report.

The study was based on the criterion of stress at a life of 100,000 cycles. In order to achieve good estimates of stress for that life, three specimens were tested for each set of parameter values. Test stresses were estimated with the intent of bracketing the 100,000-cycle failure life.

Springs

The basic wire sizes were selected to be representative of the range of springs in armament applications. The detailed design of the springs was based on various requirements of the test program. The various factors which were considered in the design of the springs are discussed in the following paragraphs.

Wire Size: The wire sizes were established as a contract requirement. They were selected as being representative of a reasonable range of sizes which are found in armament applications.

Wire Material: Although music wire was initially desired, such wire is not available in the larger sizes. Chrome-vanadium spring wire (ASTM A231) was used for the ½-in. size and chrome-silicon wire (ASTM A304, Grade 5160H) was used for the 1-in. size. The 1/8-inch diameter music wire was purchased to Federal Specification QQ-W-470b. Analysis showed 0.83% C, 0.49% Mn, 0.26% Si and other elements within maximum limits. The chrome-vanadium wire was purchased to ASTM A231. Analysis showed 0.50% C, 0.85% Mn, 0.91% Cr, 0.18% Va, 0.21% Si with other elements within maximum limits. Composition of the 1-in. 5160 H wire was 0.60% C, 0.87% Mn, 0.018% P, 0.016% S, 0.23% Si and 0.73% Cr.

Stress: A high-stress design was necessary in order that fatigue tests could be planned for failure at well less than 100,000 cycles. In anticipation of high fatigue strength resulting from the stress-peening process, allowance had to be made for higher-than-usual stress levels. Thus the springs were designed so that solid compression would give stresses well in excess of the yield strength. This design would then enable the development of high enough cyclic stresses to cause failure without the springs going solid. Stress calculations were based on the corrected Wahl formula.[1]

Gap Between Coils: In order that space be available for shot to pass between coils while the springs were compressed to the prestress levels, it was necessary to design open-coiled springs. This requirement was consistent with the high-stress requirement so it presented no problem.

Stiffness: The stiffness requirement was determined primarily by the need to keep test deflection low. The attainable cyclic rate of the fatigue testing is inversely related to the deflection required to achieve a given stress. The effect of this requirement was to keep the spring index (ratio of coil diameter to wire diameter) as low as possible and to keep the overall spring length to a minimum by limiting the number of turns.

Final Design: The result of these considerations was a design of spring which was nearly geometrically similar for the three wire sizes. The nominal design values are given as follows:

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1 A.M. Wahl, <u>Mechanical Springs</u>, Second edition, McGraw-Hill, New York, 1963, pp 229-235.

| | | | |
|---|---|---|---|
| Wire diameter (in.) | 0.125 | 0.5 | 1.0 |
| Spring index | 4 | 4 | 4 |
| Total coils | 5 | 5 | 5 |
| Active coils | 3 | 3 | 3 |
| Pitch (in.) | 0.25 | 0.875 | 1.625 |
| Free length (in.) | 1.0 | 3.625 | 6.875 |
| Outside diameter (in.) | 0.625 | 2.50 | 5.0 |
| Spring constant (lb/in.) | 940 | 3750 | 7500 |
| Solid stress (ksi)* | 320 | 240 | 200 |
| Load for 100 ksi (lb) | 110 | 1750 | 7000 |

* Corrected for helix angle.

The design allows for the expectation that the failure stresses would vary in an inverse relation to the wire size.

All springs were ground to final length.

Manufacture: The spring manufacture was subcontracted to Hardware Products Company, Inc., of Boston, MA. The smallest springs were cold wound on an automatic machine. They were given a stress relief of 30 minutes at 375F. The medium springs (0.5-in. wire) were cold wound on a spring lathe. They were given a stress relief of 550F. The large springs were hot-wound in the annealed condition, then oil-quenched and tempered.

Thirty extra springs of each size were manufactured. The 99 springs of each size in the test program were selected from the total of 120 on the basis of spring length. The group of 99 which represented the least spread in length was used. The range of length was small enough that the springs could be considered identical with regard to load-deflection-stress characteristics.

## Shot Peening

General: The shot-peening operations were performed by Metal Improvement Company at the Carlstadt, NJ plant. Materials and procedures were in compliance with MIL-S-13165 except where deviations were necessary in order to meet the needs of this study. Control of shot size and quality, measurement of intensity and measurement of coverage were performed according to the Standard. Shot size and intensity were, in some cases, outside of standard ranges in order that the effects of these variables could be tested.

Stress Peening Fixtures: In order to achieve the proper prestress in the springs, special fixtures were constructed. The fixtures were arranged as shown in Figure 6. Spacers were sized so that the springs would be compressed to stress values of 25, 50 or 100 ksi and held at that degree of compression during the peening operation. One fixture, with its accompanying set of spacers, was made for each spring size.

Peening Apparatus: A commercial peening cabinet with shot being accelerated through air nozzles, was used. Each spring was rotated about its axis so as to be exposed uniformly to the shot stream. To ensure coverage of the critical region on the inside surfaces of the coils, a separate lance-type nozzle was made. This arrangement allows shot to pass through a tube down the center of the spring until it strikes a 90° turn, whereupon it strikes the inner surfaces in a direction perpendicular to the spring axis. The strips for measurement of intensity are placed so that the shot strikes them normal to the surface.

Peening Control: The peening intensity is measured by Almen strips[2] which are thin steel strips which deform under the action of the shot. The peening machine parameters are adjusted at each setup to achieve the specified Almen strip deformation (arc height) prior to peening the springs. Intensity is checked periodically during the usage of each setup. The peening coverage is checked for completeness by means of observation of a fluorescent coating which is removed by the peening (trade name - Peenscan).

Peening Conditions: The actual conditions of shot peening were given in table 1. As mentioned earlier, these conditions were selected with the intent of bracketing the optimum conditions of shot size, intensity and prestress insofar as these optimum values could be estimated.

## FATIGUE TESTS

### Test Equipment

The cyclic tests were conducted in standard electro-hydraulic materials test machines. A schematic diagram of this type of test machine is given as figure 7. The test frame incorporates a hydraulic actuating cylinder which drives the lower test platen. Vertical posts hold an upper crosshead to which are attached a load cell and upper platen. The test springs are located between the upper and lower platens and are cyclically compressed to the desired degree by the action of the actuating cylinder. An electronic servo-control system allows a wide variety of test conditions wherein the load history or the displacement history, as selected, is constrained to follow an electrical input command signal. The machines are equipped with appropriate control features as well as with cycle counters, overload detectors, failure detectors and automatic shutoffs. Several machines were utilized during the course of this project, the particular machine being selected primarily on the basis of load requirements of the particular tests.

Loading platens were built to support the springs and to transmit the machine loads. Figure 8 shows the design of the platens. They consist of simple plates with guide bosses to locate the ends of the springs. The guide bosses were moveable so that each set of platens could be set up to give balanced loading for one, two or three springs at a time.

Protective shields were located around the platen region in case of brittle fracture and flying pieces of broken springs.

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[2] See, for example, MIL-S-13165B Amendment 2, 1979.

Fatigue Test Procedures

The first step in setting up each test was the calculation of the test load or the first deflection. (Both modes were used, as discussed later.) These loads were estimated as nearly as possible to bracket a failure life of 100,000 cycles. The target short life was 50,000 cycles and the target long life was 200,000 cycles. Ideally, each group of three specimens of a set would span this range in order to yield the best estimate of stress for a life of 100,000 cycles. Springs from different sets were often tested together, the test platens having been designed for up to three springs.

The original plan to use the deflection-control mode of the test machine. In this mode, the calculated displacement which corresponds to the desired spring load is programmed into the machine. The load is monitored by the machine control system and, when the load begins to drop as a result of spring failure, the machine shutoff circuit is actuated. It turned out that, especially with the smallest springs, the shutoff occurred without any evidence of fatigue cracks. Instead, the springs were yielding slightly or, in spring-industry parlance, were setting. With constant deflection amplitude of the platens, this setting resulted in a reduced load amplitude and, on occasion, in separation of the springs from the platen surfaces. In order to avoid this situation, the remaining springs were tested under load control. The load was programmed to cycle between the maximum value for a particular test and a minimum value of 10% of the maximum. Failure was detected by an increase in the displacement amplitude under the influence of this constant load amplitude. It was still necessary to verify that actual fatigue cracking had occurred since setting of the spring would give a similar increase in deflection. In any case, the use of the load control mode gave assurance that the full amplitude was being applied to each spring throughout the tests.

Data Analysis Procedures

The first step in the analysis of data was to enter the data for each set of three springs into a small computer program. This program was set up to establish by linear regression the best fit to the data on a stress vs. log-cycles plot. A sample of the results is given in Figure 9. Since many of the aspects of the spring processing and testing are statistical in nature (as is the fatigue process itself) and since there were only three specimens per condition, the range of these curve fit values was rather wide. The most sensitive parameter was the slope of the curve. In cases where the three data points were grouped closely together, small deviations in data produced large and unrealistic changes in slope. In order to enable comparison of the various conditions at a life of 100,000 cycles, a single slope value was established by averaging the slopes of a number of cases wherein the data were well distributed. This slope was then fit to all the other sets of data and the stress at a life of 100,000 cycles was calculated. Figure 10 shows an example of this procedure. Data are presented in terms of the maximum nominal stress in the cycle, as calculated by the corrected Wahl formula.

# TEST RESULTS

## Data

Tables 2 through 6 show the results of the analysis. The stress to give 100,000 cycle life is given for each set of three specimens (one set of conditions). The best fit of the data from the nine non-peened specimens of each size is also included in Tables 2 through 4.

## Effects of Variables

### Average Effects

Table 5 shows the overall effect of each of the variables. The values shown in this table are obtained by averaging all data for each variable. For example, the effect of shot size on 1/8-in. springs is obtained by averaging all data for 1/8-in. springs for the smallest shot size, etc., for the other sizes. It is seen from table 5 that the average effects of the variables are the following:

- O Regardless of peening variables, the shot-peened springs show much higher fatigue strength than the non-peened springs, the improvement ranging from 25% to 52%.
- O Conventional peening (zero prestress) gives increases in fatigue strength varying from 18% to 48% over the non-peened condition.
- O The stress peening operation, as compared with conventional peening, gives average increases in fatigue strength ranging from 1% to 11%, the increase being greatest in the smallest springs.
- O The fatigue strength in terms of applied stress decreases with increasing spring wire size.
- O For the range of shot sizes used, the shot size was not a strongly-effective variable for the smallest and largest springs. The ½-in. springs did improve about 15% as shot size increased. These results can be interpreted as meaning that the selected shot sizes were about optimum for the smallest and largest springs and that the largest size of shot used on the ½-in. springs approached the optimum size.
- O In all cases, the average fatigue strength was greatest with the highest peening intensity. The effect was small, ranging from about 3% to 8%. This degree of effect indicates that the selected peening intensities were somewhere near the optimum values.

## Optimum Effect of Stress Peening

If the evaluation of stress peening is limited to those conditions where other variables are more nearly optimized, the stress-peening effect appears greater at least for the two smaller spring sizes. The trends are shown in tables 2 through 6 and figure 12. For the two smaller spring sizes, the optimum condition always occurred at the highest peening intensity. The optimum shot size was 170 for the small springs and 460 for the medium sized springs. The optimum peening conditions for the large springs were less consistent, varying with prestress condition. Examination of the data in table 4 shows that the peak values of fatigue strength with 25 ksi and 100 ksi prestress occurred with 550 shot while the 50 ksi prestress gave the best results using 460 shot size. The lowest peening intensity was best with the 100 ksi prestress whereas the highest intensity was best with the other prestress levels.

A prestress level of 50 ksi appears to be optimum for the two larger spring sizes. For the smallest size, a prestress of 25 ksi gave very slightly higher results. As a rule of thumb, 50 ksi appears to be a reasonable value of prestress regardless of spring size.

In summary, the conditions which give the absolute maximum benefit are given in the following table.

| Wire size (in.) | 1/8 | 1/2 | 1 |
|---|---|---|---|
| Intensity | high | high | high |
| Shot size | 170 | 460 | 460 |
| Prestress (ksi) | 25 | 50 | 50 |

The optimum results are compared in table 7, using the unpeened condition as a reference. Based on these optimum values of fatigue results, stress peening gives significant improvement over conventional peening in the two smaller spring sizes but gives relatively small improvement over conventional peening in the 1-inch size. If we use the conventionally peened condition as a baseline, the optimum stress peening gives 20%, 15% and 5% additional improvement for the 1/8-in. 1/2-in. and 1-in. sizes, respectively.

It should be noted that as a consequence of the slope of the fatigue curve (e.g., figure 9), a given increase in fatigue strength corresponds to much larger increase in life. The data obtained during this program indicate about a ten fold increase in life for a 55 ksi increase in fatigue strength. Thus, what may appear to be a relatively small improvement in fatigue strength can correspond to a significant increase in service life. Estimates of this increase in life for conventional peening and for stress peening are given in table 8.

An analysis of the cost benefit aspects of conventional peening and of stress peening is made in the Appendix. The results of this analysis show that as expected, the economic benefits of conventional peening are considerable but that the economic benefits of stress peening are greater except in the smallest spring size considered in this study.

## CONCLUSIONS

1. The fatigue strength under any of the peening conditions tested was markedly greater than that of non-peened springs.

2. The conventional shot peening treatment resulted in increases in fatigue strength of 18%, 22% and 51%, respectively, for springs having 1/8 in., 1/2 in. and 1 in. diameter wire. These increases correspond to increases in fatigue life by factors of 3.8, 4.2 and 10.4, respectively.

3. The stress peening process produced additional increases in fatigue strength of 20%, 15% and 5% over the conventionally peened conditions for the 1/8 in., 1/2 in. and 1 in. sizes, respectively, (based on the conventionally peened condition). These increases correspond to overall increases in fatigue life (relative to the unpeened condition) by factors of 22.2, 14.0 and 14.6, respectively.

4. A prestress level of 50,000 psi during shot peening resulted in increases in fatigue strength at or near the optimum values.

## RECOMMENDATIONS

1. The effect of presetting the springs before or after peening should be investigated. Presetting would minimize dimensional changes during high stress service. Any possible effects on the benefits of stress peening should however be determined.

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Table 1. Test matrix

| Spring Size | Shot Size | Peening Intensity | Prestress (ksi) | | | |
|---|---|---|---|---|---|---|
| | | | 0 | 25 | 50 | 100 |
| 1/8-in. | | | | | | |
| | 110 | 9N | | + | + | + |
| | | 4-6A | | | | |
| | | 6-8A | + | | | |
| | 170 | 4-6A | | | | |
| | | 6-8A | | | | |
| | | 8-10A | + | | | |
| | 230 | 6-8A | | | | |
| | | 8-10A | | | | |
| | | 10-12A | + | | | |
| | None | None | | No peening | | |
| | | | 0 | 25 | 50 | 100 |
| 1/2-in. | | | | | | |
| | 230 | 8-10A | | + | + | + |
| | | 10-12A | | | | |
| | | 12-14A | + | | | |
| | 330 | 10-12A | | | | |
| | | 12-14A | | | | |
| | | 16-18A | + | | | |
| | 460 | 12-14A | | | | |
| | | 16-18A | | | | |
| | | 6-7C | + | | | |
| | None | None | | No peening | | |
| | | | 0 | 25 | 50 | 100 |
| 1-in. | | | | | | |
| | 460 | 16-18A | | + | + | + |
| | | 6-7C | | | | |
| | | 8-9C | + | | | |
| | 550 | 6-7C | | | | |
| | | 7-8C | | | | |
| | | 8-9C | + | | | |
| | 660 | 7-8C | | | | |
| | | 8-9C | | | | |
| | | 9-10C | + | | | |

Note: Three specimens per peening condition. Nine non-peened specimens per spring size. Total number of specimens in program = 297.

Table 2. Test results - 1/8-in. wire springs

| Shot size | Peening intensity | Prestress (ksi) | Stress for 100,000 cycle life (ksi) |
|---|---|---|---|
| No peening | | | 173 |
| 110 | 9N | 25 | 223 |
| | | 50 | 225 |
| | | 100 | 223 |
| | 4-6A | 25 | 229 |
| | | 50 | 223 |
| | | 100 | 228 |
| | 6-8A | 0 | 203 |
| | | 25 | 227 |
| | | 50 | 218 |
| | | 100 | 223 |
| 170 | 4-6A | 25 | 219 |
| | | 50 | 223 |
| | | 100 | 223 |
| | 6-8A | 25 | 224 |
| | | 50 | 218 |
| | | 100 | 222 |
| | 8-10A | 0 | 205 |
| | | 25 | 247 |
| | | 50 | 241 |
| | | 100 | 242 |
| 230 | 6-8A | 25 | 227 |
| | | 50 | 226 |
| | | 100 | 222 |
| | 8-10A | 25 | 233 |
| | | 50 | 235 |
| | | 100 | 239 |
| | 10-12A | 0 | 204 |
| | | 25 | 225 |
| | | 50 | 237 |
| | | 100 | 227 |

Table 3. Test results - 1/2-in. wire springs

| Shot size | Peening intensity | Prestress (ksi) | Stress for 100,000 cycle life (ksi) |
|---|---|---|---|
| No peening | | | 156 |
| 230 | 8-10A | 25 | 164 |
| | | 50 | 158 |
| | | 100 | 161 |
| | 10-12A | 25 | 174 |
| | | 50 | 184 |
| | | 100 | 183 |
| | 12-14A | 0 | 180 |
| | | 25 | 181 |
| | | 50 | 195 |
| | | 100 | 209 |
| 330 | 10-12A | 25 | 190 |
| | | 50 | 192 |
| | | 100 | 196 |
| | 12-14A | 25 | 186 |
| | | 50 | 183 |
| | | 100 | 197 |
| | 16-18A | 0 | 190 |
| | | 25 | 187 |
| | | 50 | 192 |
| | | 100 | 192 |
| 460 | 12-14A | 25 | 202 |
| | | 50 | 201 |
| | | 100 | 192 |
| | 16-18A | 25 | 201 |
| | | 50 | 204 |
| | | 100 | 210 |
| | 6-7C | 0 | 184 |
| | | 25 | 202 |
| | | 50 | 219 |
| | | 100 | 212 |

Table 4. Test results - 1-in. wire springs

| Shot size | Peening intensity | Prestress (ksi) | Stress for 100,000 cycle life (ksi) |
|---|---|---|---|
| No peening | | | 110 |
| 460 | 16-18A | 25 | 147 |
| | | 50 | 163 |
| | | 100 | 167 |
| | 6-7C | 25 | 159 |
| | | 50 | 160 |
| | | 100 | 170 |
| | 8-9C | 0 | 161 |
| | | 25 | 157 |
| | | 50 | 174 |
| | | 100 | 171 |
| 550 | 6-7C | 25 | 159 |
| | | 50 | 155 |
| | | 100 | 172 |
| | 7-8C | 25 | 154 |
| | | 50 | 155 |
| | | 100 | 165 |
| | 8-9C | 0 | 166 |
| | | 25 | 163 |
| | | 50 | 161 |
| | | 100 | 168 |
| 660 | 7-8C | 25 | 160 |
| | | 50 | 172 |
| | | 100 | 160 |
| | 8-9C | 25 | 159 |
| | | 50 | 153 |
| | | 100 | 161 |
| | 9-10C | 0 | 162 |
| | | 25 | 151 |
| | | 50 | 166 |
| | | 100 | 166 |

Table 5. Average effects of peening variables

$\sigma_{100}^{a}$ vs prestress (ksi)

| Spring size (in.) | No peening | 0 | 25 | 50 | 100 |
|---|---|---|---|---|---|
| 1/8 | 173 | 204 | 228 | 227 | 227 |
| 1/2 | 156 | 185 | 188 | 192 | 195 |
| 1 | 110 | 163 | 156 | 162 | 167 |

$\sigma_{100}$ vs spring size[b]

| | |
|---|---|
| 1/8 | 228 |
| 1/2 | 191 |
| 1 | 162 |

$\sigma_{100}$ vs shot size[b]

| | Small | Medium | Large |
|---|---|---|---|
| 1/8 | 224 | 229 | 230 |
| 1/2 | 179 | 191 | 205 |
| 1 | 163 | 161 | 160 |

$\sigma_{100}$ vs intensity[b]

| | Low | Medium | High |
|---|---|---|---|
| 1/8 | 223 | 228 | 232 |
| 1/2 | 184 | 191 | 199 |
| 1 | 162 | 160 | 164 |

---

a Stress to give 100,000-cycle life (ksi).

b Prestressed springs.

Table 6. Optimum effect of peening prestress

| | Maximum stress for 100,000-cycle life | | | |
|---|---|---|---|---|
| | Prestress level* | | | |
| Spring Wire Size (in.) | 0 | 25 | 50 | 100 |
| 1/8 | 205 | 247 | 241 | 242 |
| 1/2 | 190 | 202 | 219 | 212 |
| 1 | 166 | 163 | 174 | 172 |

* Stresses in ksi.

Table 7. Fatigue strength* improvement by conventional peening and by optimum stress peening

| Wire Size | 1/8 | 1/2 | 1 |
|---|---|---|---|
| No peening | 173 ksi | 156 ksi | 110 ksi |
| Conventional peening | 205 ksi | 190 ksi | 166 ksi |
| Improvement over unpeened | 18% | 22% | 51% |
| Stress peening | 247 ksi | 219 ksi | 174 ksi |
| Improvement over unpeened | 43% | 40% | 58% |
| Improvement over conventional | 21% | 15% | 5% |

---

* At 100,000 cycle life.

Table 8. Estimated increases in fatigue life

| Wire size (in.) | Factor of increase in fatigue life* Conventional peening | Stress peening | Stress peen/ convent. peen |
|---|---|---|---|
| 1/8 | 3.8 | 22.2 | 5.8 |
| 1/2 | 4.2 | 14.0 | 3.3 |
| 1 | 10.4 | 14.6 | 1.4 |

* Relative to the unpeened condition.

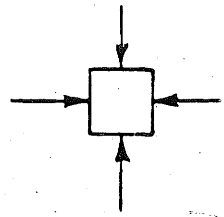


Figure 1. Equiaxial surface stress

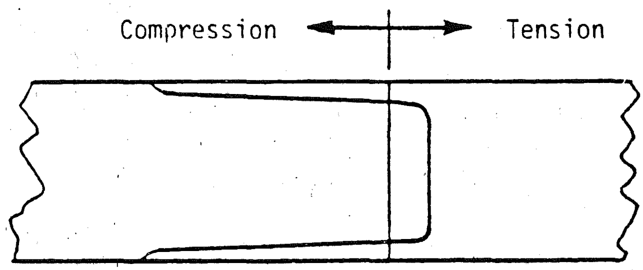



Figure 2. Distribution of peening stress through a plate

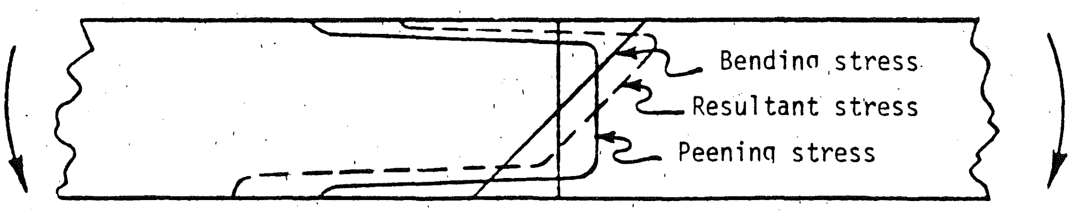
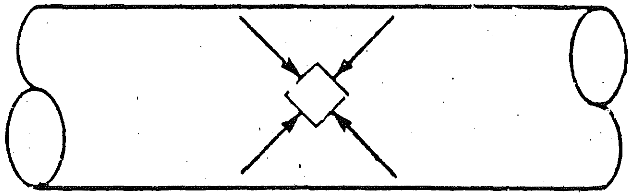
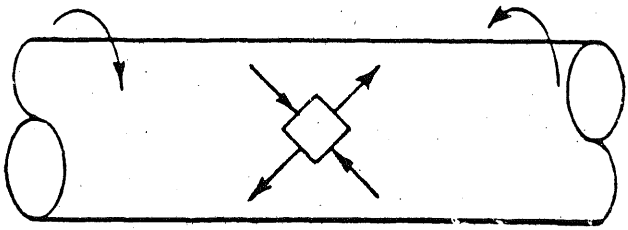


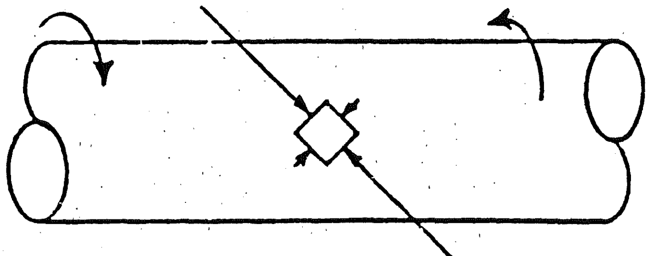

Figure 3. Benefit of shot peening in a beam in bending



a. Peening stress

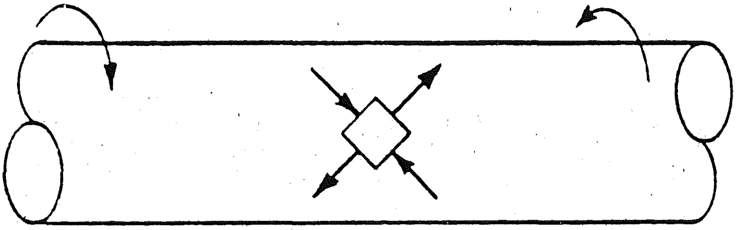


b. Torsional stress

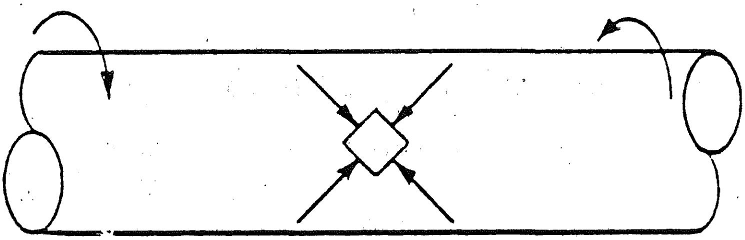


c. Sum of Stresses

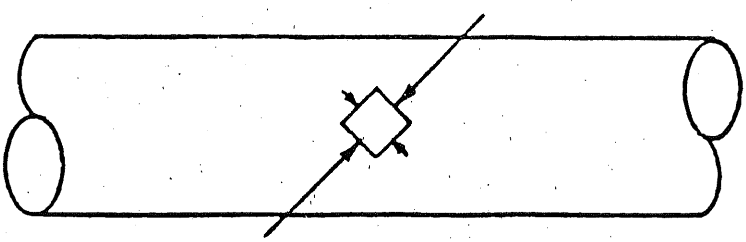
Figure 4. Superposition of conventional peening stress and torsional stress



a. Torsional stress



b. Stress after peening but still under torsional load



c. Torsional load released

Figure 5. Effect of stress-peening a torsion bar

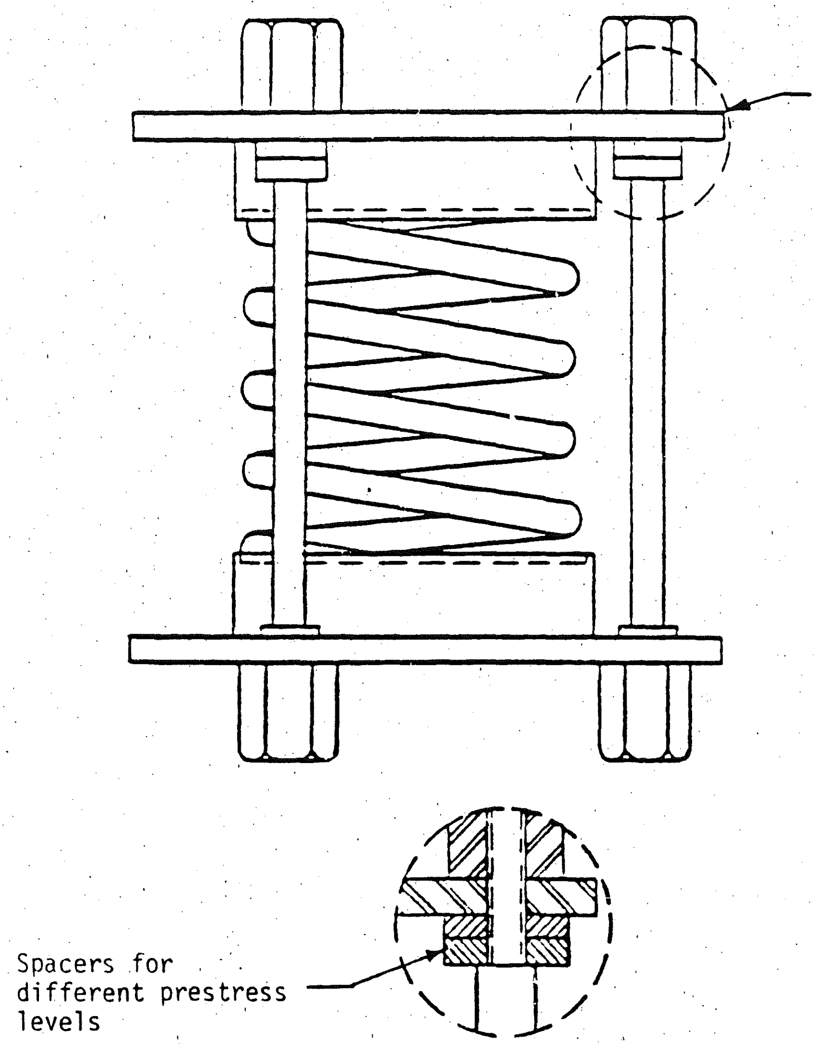



Figure 6. Stress peening fixture

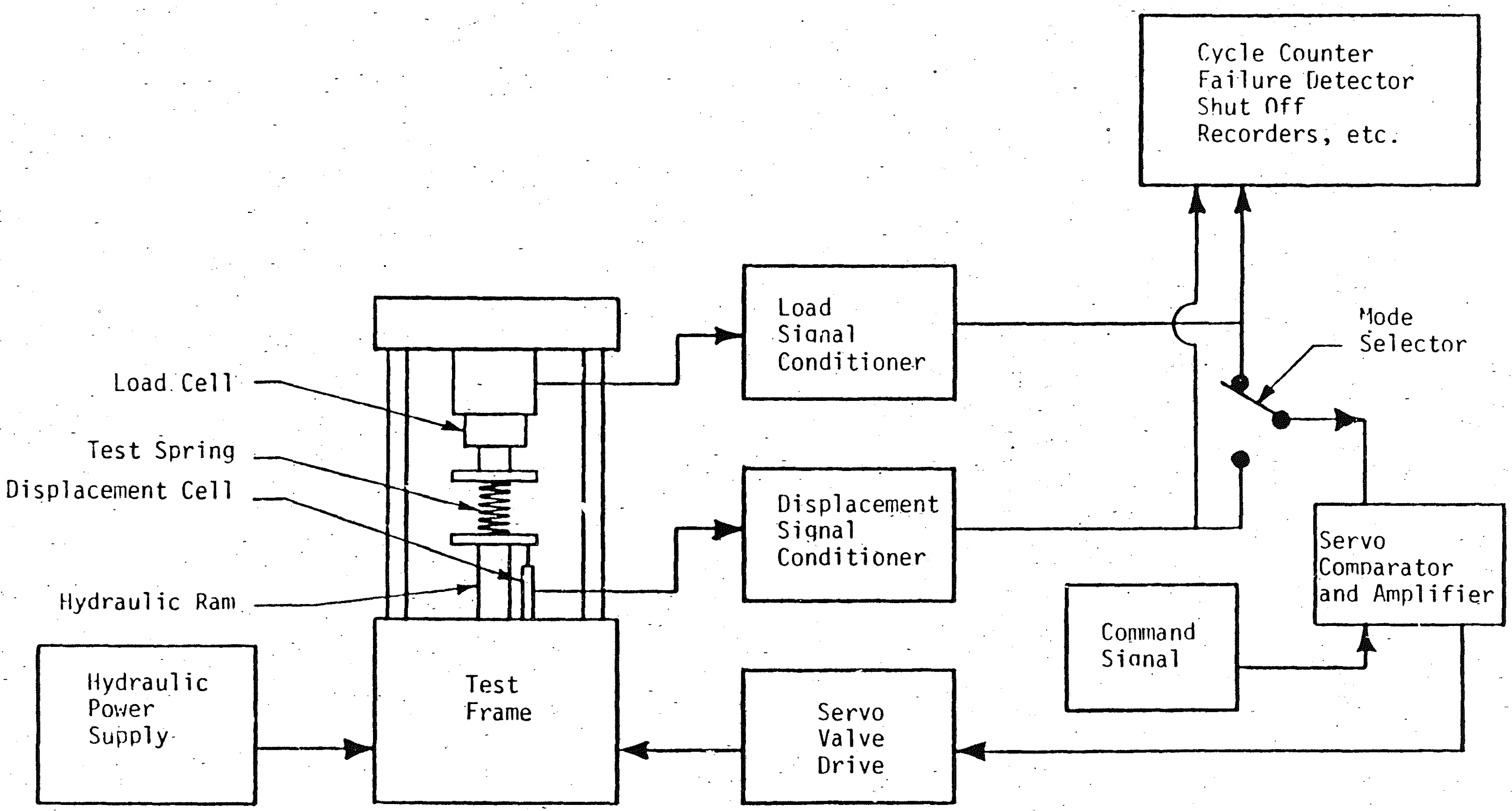



Figure 7. Fatigue test system

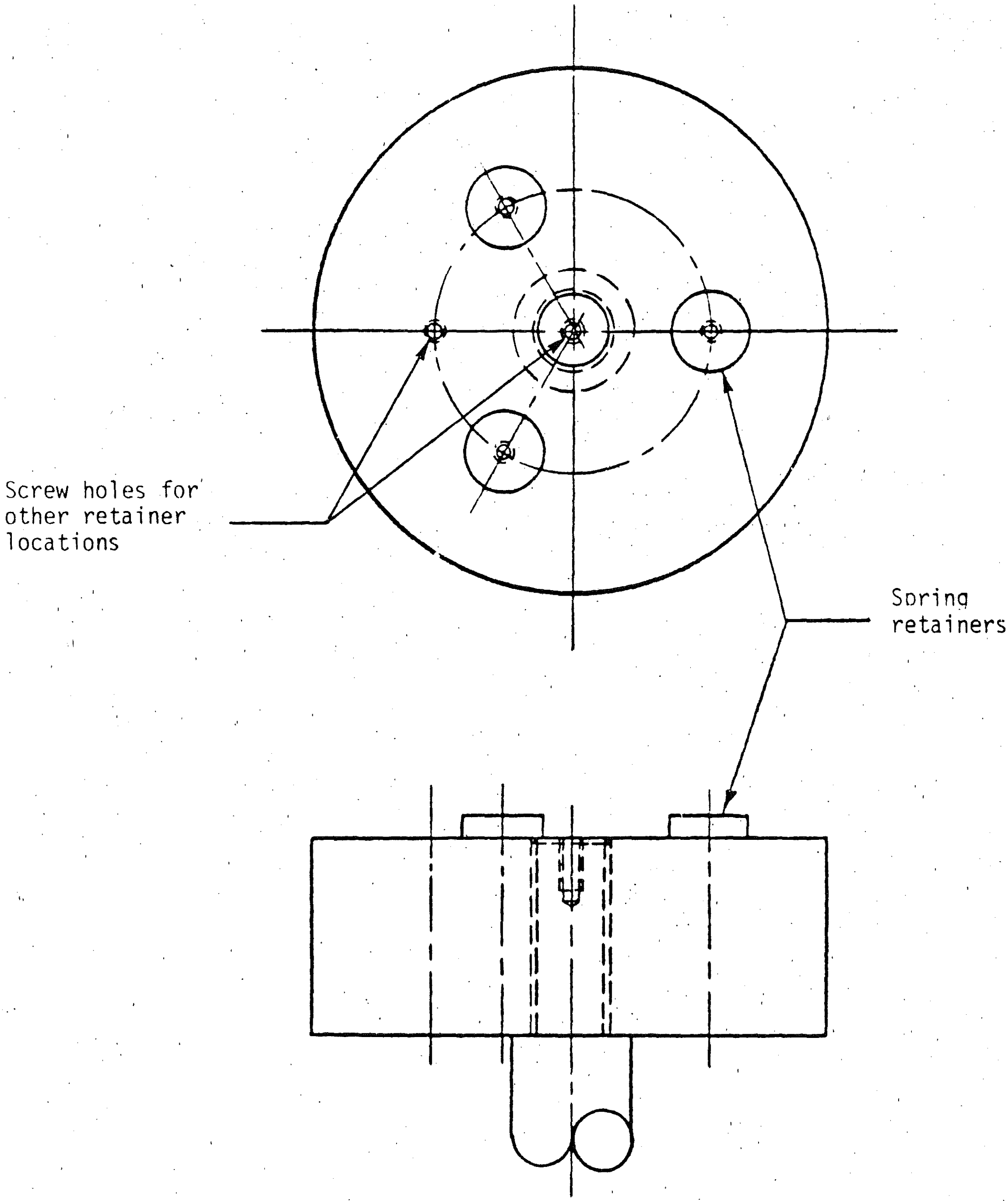



Figure 8. Fatigue Loading Platen (Typical)

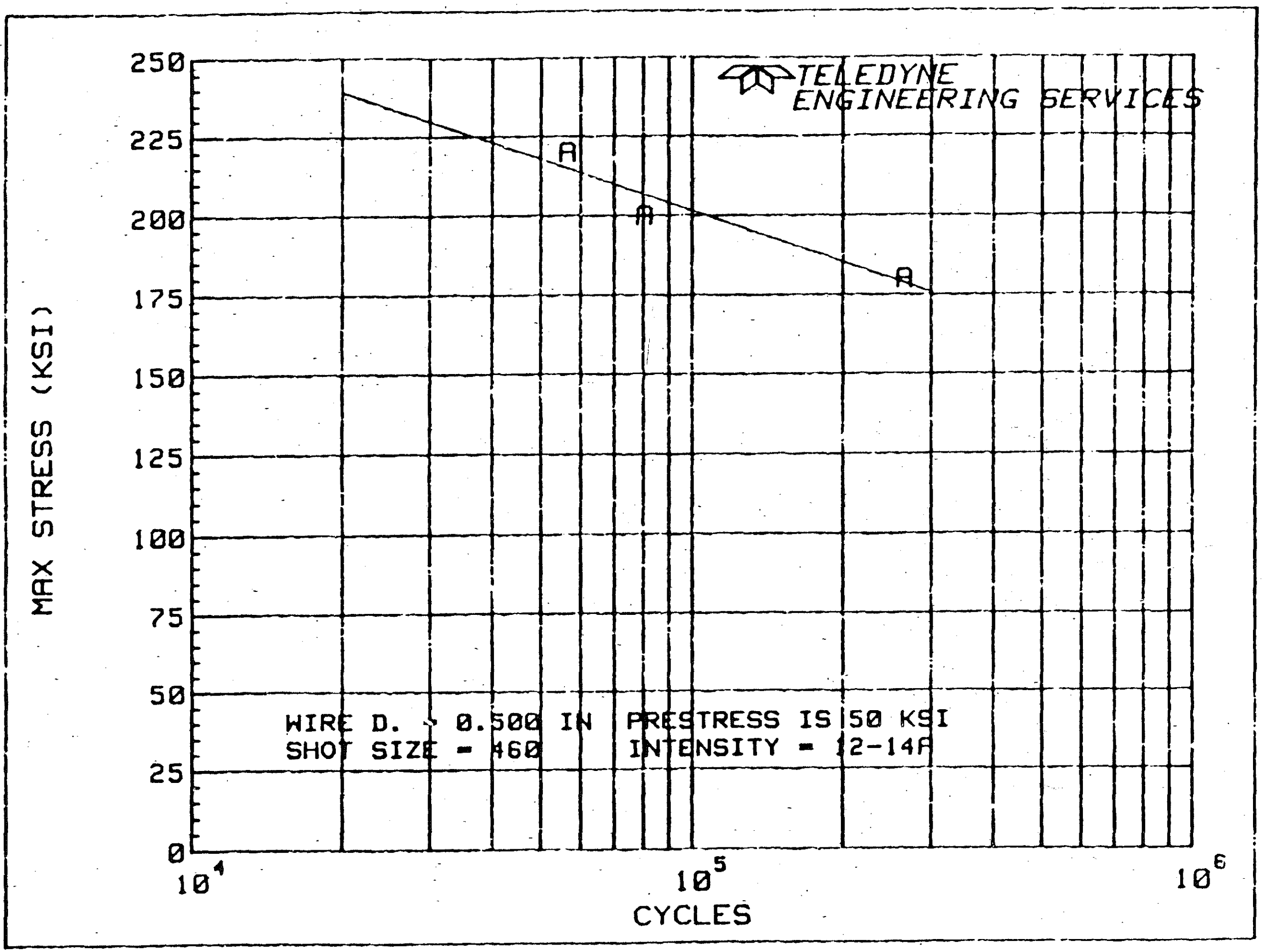



Figure 9. Sample linear regression fit of data

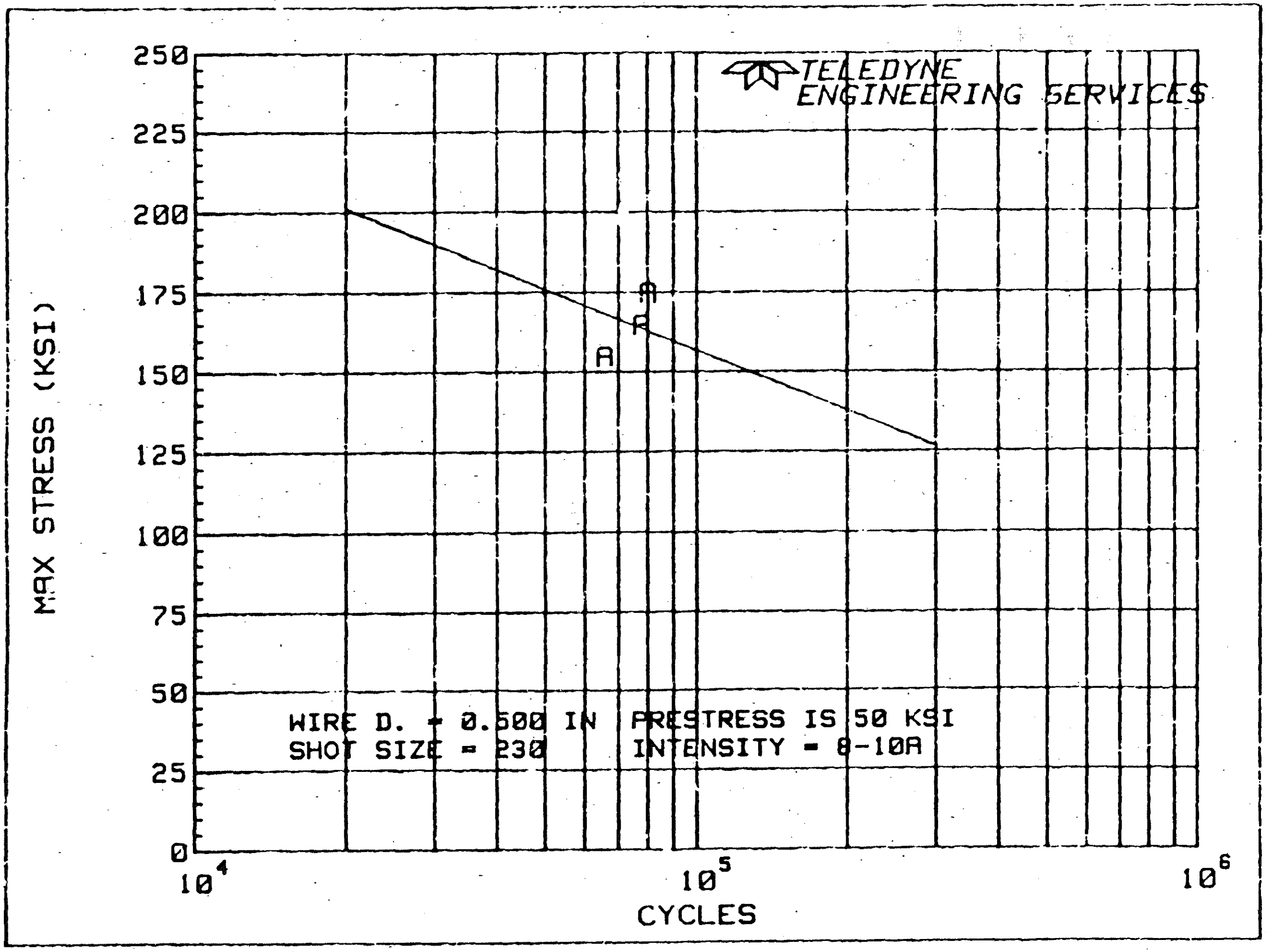



Figure 10. Average slope fit through data

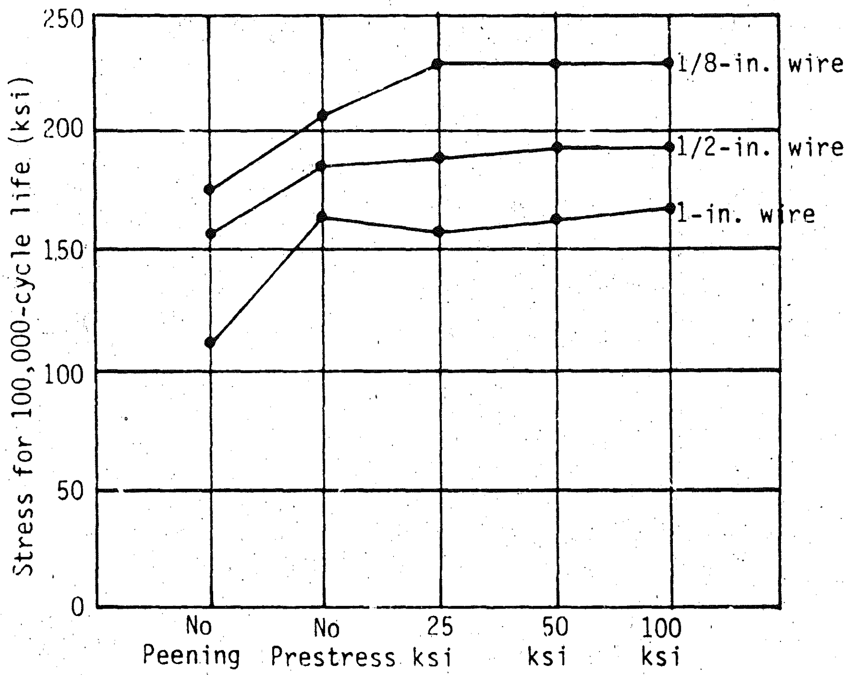



Figure 11. Effect of prestress: average of all peening conditions

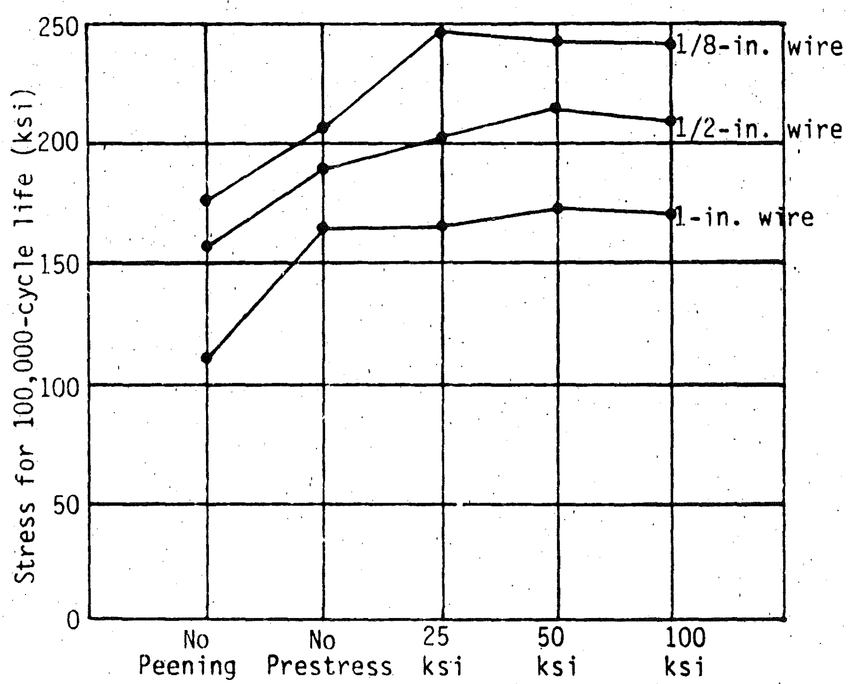



Figure 12. Effect of prestress: optimum peening conditions

# APPENDIX
# COST OF STRESS PEENING

The primary effort in this project has been directed toward the technical aspects and benefits of the stress peening process. In order to enable an assessment of the cost effectiveness of the process, inquiries were made of suppliers of peening services and estimated costs were assembled. These estimates are based on the spring designs which were used in this test program. Some variation could be anticipated if the designs were to differ appreciably. The tabulated costs do, however, serve as guidelines for use in any cost benefit study.

Table A-1 gives the estimates of conventional peening costs and of stress peening costs for the various spring sizes. The base prices of the springs are also included.

It is seen in table A-1 that the cost of stress peening is very much higher than the cost of conventional peening. This great difference arises from the fact that stress peening requires individual handling and loading of each spring whereas conventional peening can be performed in batches. It is conceivable that mechanized loading equipment could be devised for some cost saving if very large quantities of springs were to be processed.

In table A-2, the benefits of both conventional peening and stress peening, as determined from the factors in table 8, are compared with the cost factors in table A-1. The resulting cost/life factors are a measure of the benefits to be gained from the processes. It is seen that the conventional peening, as expected, exhibits very favorable cost benefit factors for all spring sizes. Comparing the cost benefit factors for the stress peening versus the conventional peening, it is seen that the stress peening factors are better in the two larger spring sizes but less in the case of the small springs.

Based on these values, the stress peening process would be economically beneficial for the two larger spring sizes but not for the smallest size.

Table A-1. Estimated peening costs*

| Wire size (in.) | Basic spring cost (each) | Conventional peening (each) | Increase over basic (%) | Stress peening (each) | Increase over basic (%) |
|---|---|---|---|---|---|
| 1/8 | $ 0.88 | $0.045 | 5.1 | $ 6.79 | 772 |
| 1/2 | 11.40 | 0.17 | 1.5 | 8.29 | 73 |
| 1 | 39.22 | 0.60 | 1.5 | 10.54 | 27 |

* For lot sizes of 1000 to 5000. Slight variations of cost may occur within this range.

Table A-2. Cost/Benefit Factors

| Wire Size (in.) | 1/8 | 1/2 | 1 |
|---|---|---|---|
| Conventional peening* | | | |
| Cost factor | 1.051 | 1.015 | 1.015 |
| Life factor | 3.8 | 4.2 | 10.4 |
| Cost/life | .28 | .24 | .10 |
| Stress peening* | | | |
| Cost factor | 8.72 | 1.73 | 1.27 |
| Life factor | 22.2 | 14.0 | 14.6 |
| Cost/life | .39 | .12 | .09 |

* Factors relative to non-peened condition.

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