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EFFECT OF RESIDUAL STRESSES INDUCED BY PRESTRESSING ON ROLLING-ELEMENT FATIGUE LIFE

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16. Abstract

A mechanical prestress cycle suitable to induce compressive residual stress beneath the surface of the inner race of radially loaded 207-size bearings was determined. Compressive residual stresses in excess of 0.69×10^9 N/m^2 (100 000 psi), as measured by X-ray diffraction, were induced at the depth of maximum shearing stress. The prestress cycle consisted of running the bearings for 25 hours at 2750 rpm at a radial load which produced a maximum Hertz stress of 3.3×10^9 N/m^2 (480 000 psi) at the contact of the inner race and the heaviest loaded ball. Bearings subjected to this prestress cycle and subsequently fatigue tested gave a 10-percent fatigue life greater than twice that of a group of baseline bearings.

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EFFECT OF RESIDUAL STRESSES INDUCED BY PRESTRESSING ON ROLLING-ELEMENT FATIGUE LIFE

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SUMMARY

A prestress cycle suitable for inducing compressive residual stress in the inner-race groove of a ball bearing was determined. Several bearings were run for different time periods at a load higher than typical application conditions, and residual stress measurements were made in the inner race beneath the running track. It was determined that compressive residual stresses in excess of 0.69×10^9 N/m^2 (100 000 psi) are induced at the depth of the maximum shearing stress after 25 hours at a maximum Hertz stress of 3.3×10^9 N/m^2 (480 000 psi) and a shaft speed of 2750 rpm.

Twenty-seven bearings were subjected to this prestress cycle and subsequently fatigue tested at a radial load which resulted in a maximum Hertz stress at the inner-race - ball contact of 2.4×10^9 N/m^2 (350 000 psi). The results of these tests were compared with results of baseline tests without a prestress cycle at identical test conditions. The 10-percent fatigue life of the prestressed ball bearings was greater than twice that of the baseline bearings. Additionally, it was determined that differences between the measured residual stress in the prestressed bearings and in the baseline bearings after 3000 to 4000 hours of testing are small.

INTRODUCTION

Compressive residual stresses induced beneath the surface of ball-bearing race grooves are beneficial to rolling-element fatigue life (refs. 1 and 2). In reference 1, ball-bearing lives were increased by a factor of 2 when metallurgically induced ("prenitrided") compressive residual stress was present in the inner rings. In reference 2, compressive residual stresses induced by unidentified "mechanical processing" operations were found to be beneficial to the fatigue life of ball bearings.

Compressive residual stresses are also induced as a result of the cyclic concentrated contact in rolling-element bearings (refs. 3 to 8). These stresses depend on plastic deformation of the microstructure. They tend to reach a maximum at a depth of several mils beneath the rolling surface, corresponding approximately to either the depth of the maximum shearing stress (refs. 3, 7, and 8) or the depth of the maximum orthogonal shearing stress (ref. 4). The magnitude of the residual stress tangential to the surface in the direction of rolling is dependent on both the applied load and the number of load cycles. Reference 3 indicates a threshold load below which significant residual stresses are not induced except for very long running times.

Changes in microstructure (phase transformations) have been reported to occur in the same areas as the maximum induced residual stress (refs. 3 and 7). Under some conditions of very high contact stresses, no microstructural alteration was apparent where significant residual stresses were induced in a few cycles (ref. 3). The correlation of induced residual stress with these microstructural alterations is not clear. In reference 8, they are proposed to be independent phenomena.

Reference 4 reports significant effects of residual stress on rolling-element fatigue life. In this work, maximum compressive residual stresses were induced when the ball hardness was 1 to 2 points Rockwell C greater than the race hardness. Bearings of AISI 52100 material with this hardness difference (ΔH = 1 to 2 points Rockwell C) produced longer rolling-element fatigue lives than bearings with ΔH values of greater or lesser magnitude. These residual stresses (ref. 4) were induced during fatigue testing of the bearings and apparently had an effect of extending the lives of those bearings where the induced stress was greatest. An identical effect is reported in reference 8.

The data of reference 9 show no effect on rolling-element fatigue life of a simulated compressive residual stress which was mechanically applied by interference fits of thin ring specimens. These results with the compressive stresses are contrary to all previously published test experience known to the authors and are not fully explained. Tensile stresses applied in a similar manner significantly reduced fatigue life.

A somewhat similar experiment involving mechanically applied stresses superimposed on Hertzian contact stresses is described in reference 10. In a unique rotating-bending-rolling apparatus, a mechanically applied compressive stress markedly increased rolling-element fatigue life. Tensile stresses significantly reduced fatigue life.

All published test data with the exception of reference 9 indicate that superimposed compressive stresses (either residual or mechanically applied) of the proper magnitude and location can be beneficial to rolling-element fatigue life. An analysis presented in reference 5 indicates that a compressive residual stress that exists at the depth of the maximum shearing stress can decrease the maximum shearing stress. A similar analysis for superimposed stresses was subsequently reported in reference 11. Although observations (ref. 12) would indicate that the maximum orthogonal shearing stress is the critical stress in the initiation of fatigue cracks, there also is evidence (refs. 4 and 13 to 15) that the maximum shearing stress is the most significant stress in the fatigue process. Thus, if the maximum shearing stress for a given Hertz stress could be decreased by compressive residual stresses, rolling-element fatigue life could be increased (ref. 5).

It was hypothesized that such a beneficial compressive residual stress could be induced by prestressing a ball-bearing inner race; for example, by running the bearing at a load greater than the threshold load (ref. 3) for a prescribed number of cycles. The bearing, when subsequently run under more nominal service loads, would then be expected to experience a longer fatigue life. The test program presented herein was devised to determine the validity of this hypothesis. Radially loaded ball bearings were chosen for this program. The application of such a prestressing cycle to extend fatigue life of thrust-loaded ball bearings is complicated because of the variation of contact angle with load and speed conditions.

The specific objectives of this program were (1) to determine the number of cycles (prestress time) required to induce significant residual stress in the inner race of a ball bearing at a selected maximum Hertz stress, (2) to determine the effect of this mechanical prestress cycle on rolling-element fatigue life, and (3) to determine the effect of the number of cycles on residual stress at two levels of maximum Hertz stress.

In order to accomplish these objectives, 207-size deep-groove ball bearings of SAE 52100 steel were run for various times at a maximum Hertz stress of either 3.3×10^9 or 2.4×10^9 N/m^2 (480 000 or 350 000 psi). The induced residual stress was subsequently measured by X-ray diffraction as a function of depth below the inner-race surface. An optimum prestress cycle at a selected load was determined. Twenty-seven identical bearings were then prestressed by using this optimum cycle and were subsequently fatigue tested at 2.4×10^9 N/m^2 (350 000 psi). Another 27 bearings which had not been subjected to the prestress cycle were tested at identical conditions. The fatigue test results of the two groups of bearings were statistically compared to show the effect of this prestress cycle on rolling-element fatigue life. All bearing testing was conducted at TRW Inc., MRC Division, Jamestown, N. Y., under NASA contract NAS 3-10184. The residual stress measurements were made by TRW Inc., MRC Division, and by General Motors Research Laboratory, Warren, Michigan, under contracts NAS 3-15320 and NAS 3-12413, respectively.

TEST BEARINGS

All the test bearings were 207-size deep-groove ball bearings. The dimensions and specifications of the bearings are as follows:

Bearing bore, mm (in.)	35 (1.3780)
Bearing outside diameter, mm (in.)	72 (2.8346)
Number of balls	9
Ball diameter, cm (in.)	1.111 (0.4375)
Inner-race conformity, percent	51
Outer-race conformity, percent	52
Inner-race track diameter, cm (in.)	4.228 (1.6648)
Outer-race track diameter, cm (in.)	6.454 (2.5411)
Mounted radial clearance, cm (in.)	0.025 (0.010)
Bearing tolerance grade	ABEC-5
Ball specification	AFBMA-10
Inner-race hardness, Rockwell C	62.0 to 62.5
Outer-race hardness, Rockwell C	63.6 to 63.9
Ball hardness, Rockwell C	63.5 to 64.0

In order to obtain the maximum life from these bearings, the ball hardness minus inner-race hardness ΔH was controlled to be in the range from 1 to 2 points Rockwell C (ref. 5).

All inner races for all bearings tested (both the prestress-cycle survey bearings and the fatigue-tested bearings) were from one heat of air-melted, vacuum-degassed AISI 52100 steel. All outer races used for the fatigue-tested bearings and the higher load prestress-cycle survey bearings were from the same heat of steel as the inner races. All balls for the fatigue-tested bearings and the higher load prestress-cycle survey bearings were from a single heat of air-melted, vacuum-degassed AISI 52100 steel. The outer races and balls from the remaining bearings were from separate heats of air-melted, vacuum-degassed AISI 52100 steel.

BEARING TEST APPARATUS

The bearings were run in test machines capable of testing four bearings simultaneously for both the prestressing and the fatigue testing. One pair of bearings is loaded radially against the other pair by a hydraulic load mechanism. The test spindle is coupled to the drive assembly with a spline and runs at 2750 rpm. Jet lubrication with a super-refined naphthenic mineral oil is provided for each test bearing. Oil-in temperature is maintained at 311 to 316 K (100° to 110° F). The bearing outer-race temperature is 339 to 344 K (150° to 160° F) at 5860 newtons (1320 lb) radial load and 358 to 366 K (185° to 200° F) at 13 800 newtons (3100 lb) radial load.

PROCEDURE

In order to determine an optimum prestress cycle, five bearings were run at 2750 rpm for time periods from 1 to 200 hours at a selected radial load of 13 800 newtons (3100 lb), or a maximum Hertz stress of 3.3×10^9 N/m^2 (480 000 psi) at the inner race. Eight bearings were also run at 2750 rpm for time periods from 2 to 500 hours at the fatigue-test radial load of 5860 newtons (1320 lb), producing a maximum Hertz stress at the inner race of 2.4×10^9 N/m^2 (350 000 psi).

These 13 bearings were then disassembled and the tangential residual stress was measured as a function of depth beneath the center of the inner-race groove. The X-ray diffraction technique of residual stress measurement (ref. 16) was used. Material was removed to varying depths (as much as 0.033 cm (0.013 in.)) beneath the surface by an electropolishing technique.

Baseline fatigue tests were conducted with 27 bearings at a radial load of 5860 newtons (1320 lb) and a shaft speed of 2750 rpm with the super-refined naphthenic mineral oil. Another 27 bearings were then subjected to the optimum prestress cycle of 25 hours at 3.3×10^9-N/m^2 (480 000-psi) maximum Hertz stress at 2750 rpm to induce the desired tangential compressive residual stress. These bearings were subsequently fatigue tested at conditions identical to those of the baseline tests. The fatigue-test data were statistically analyzed in accordance with the methods of reference 17. Several bearings from both series were disassembled after fatigue testing, and residual stress measurements were made to determine changes in the state of stress.

RESULTS AND DISCUSSION

Prestress-Cycle Determination

The radial load for the prestress cycle was selected based on the following criteria: It must (1) be greater than the desired load for fatigue testing, (2) be less than the static load capacity of the bearing (ref. 18), and (3) produce a maximum Hertz stress greater than the threshold stress quoted in reference 3. The selected load was 13 800 newtons (3100 lb), which produced a maximum Hertz stress at the heaviest-loaded-ball - inner-race contact of 3.3×10^9 N/m^2 (480 000 psi). The loads and stresses of interest are shown in table I.

The optimum prestress cycle time was determined from the five bearings that were run for time periods of 1 to 200 hours at the selected load. The tangential residual stress induced after each prestressing period is shown in figure 1 as a function of depth below the center of the inner-race groove. The profiles follow the familiar trends of

TABLE I. - PRESTRESS-CYCLE LOAD DETERMINATION

	Radial load		Maximum Hertz stress	
	N	lb	N/m^2	psi
Fatigue test load	5 860	1320	2.4×10^9	350 000
Static load capacity (ref. 18)	18 900	4260	3.7	540 000
Threshold stress (ref. 3)	------	----	3.25	470 000
Prestress load	13 800	3100	3.3	480 000

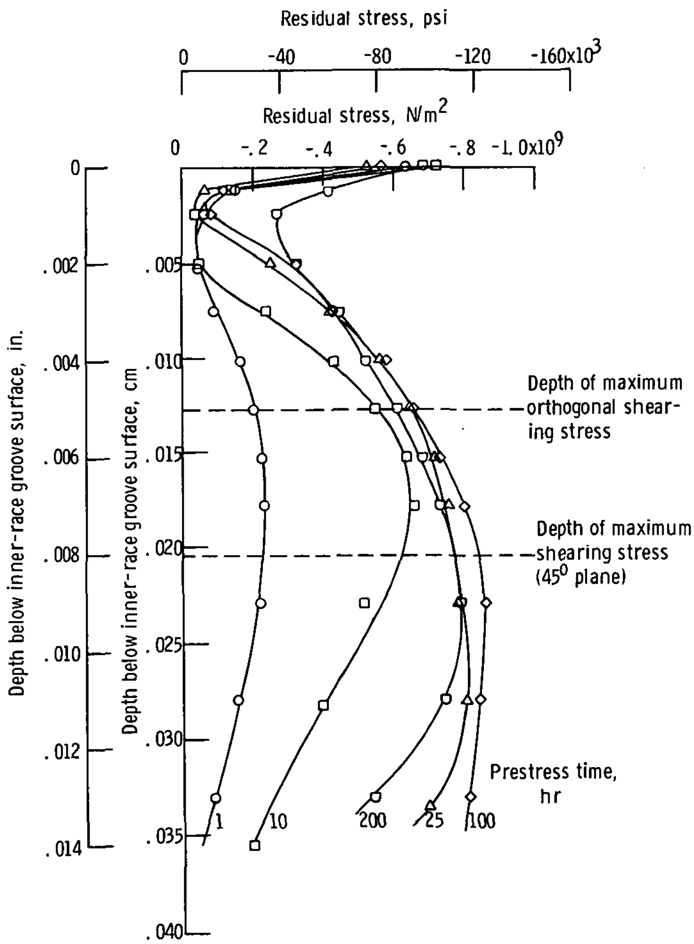


Figure 1. - Tangential residual compressive stress as a function of depth below center of inner-race groove for various prestress time cycles. Radial load, 13 800 N (3100 lb); shaft speed, 2750 rpm.

(1) high compressive stress at the surface due to race finishing processes, (2) a rapid decrease in compressive stress in the first mil depth, (3) an increase in compressive stress with a maximum at approximately the depth of the maximum shearing stress (0.020 cm (0.008 in.)), and (4) a gradual decrease in stress at greater depths. It is apparent that the compressive residual stress at the depth of the maximum shearing stress was near a maximum in the first 25 hours.

Based on these results, the prestress cycle for these bearings was chosen to be 3.3×10^{9} N/m^{2} (480 000 psi) for 25 hours at 2750 rpm.

Baseline Fatigue Tests

Twenty-seven bearings were fatigue tested at 5860 newtons (1320 lb) radial load at 2750 rpm to provide baseline data for the prestressed bearing tests. The results of these tests are shown in figure 2 and tabulated in table II. Ten of the 27 bearings experienced fatigue failures before the 4000-hour cutoff time was reached. Eight of these 10 failures were inner-race failures, and two were outer-race failures. Only the inner-race failures were considered in the statistical analysis since inner-race prestressing was the variable in these tests. The outer-race failures were treated as suspensions, in accordance with reference 17.

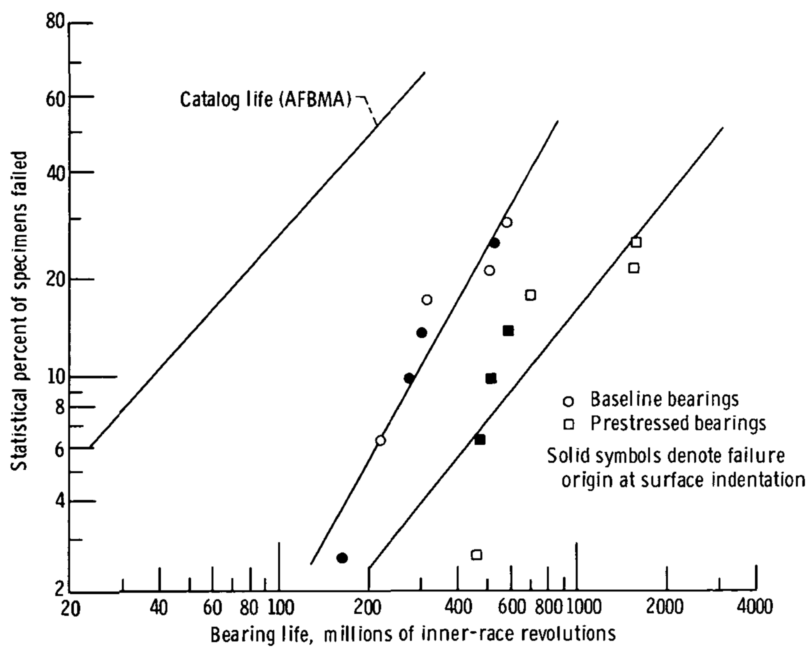


Figure 2. - Results of fatigue tests with 207-size ball bearings tested at a radial load of 5860 newtons (1320 lb) and a shaft speed of 2750 rpm with a super-refined naphthenic mineral oil.

TABLE II. - RESULTS OF FATIGUE TESTS WITH 207-SIZE BALL BEARINGS

[Radial load, 5860 N (1320 lb); shaft speed, 2750 rpm; lubricant, super-refined naphthenic mineral oil.]

	Baseline bearings	Prestressed[a] bearings
Ten-percent life, millions of inner-race revolutions	290	663
Fifty-percent life, millions of inner-race revolutions	826	3095
Weibull slope	1.80	1.22
Failure index[b]	8 out of 27	7 out of 27
Confidence number[c], percent:		
Ten-percent life level	--	76
Mean life level	--	99

[a]Prestress cycle: 25 hours at 13 800-N (3100-lb) radial load and 2750 rpm.
[b]Inner-race failures only.
[c]Probability that the prestressed bearings are superior to the baseline bearings.

At these test conditions, the Anti-Friction Bearing Manufacturers Association (AFBMA) catalog life is 230 hours (38×10^6 inner-race revolutions). The 10-percent life of these baseline bearings was 1800 hours (290×10^6 inner-race revolutions), or more than seven times AFBMA life. This relatively good performance of these air-melted, vacuum-degassed AISI 52100 bearings is partially due to the beneficial ΔH effect (ΔH = 1 to 2 points Rockwell C, ref. 5).

Prestressed-Bearing Fatigue Tests

Another group of 27 bearings was prestressed for 25 hours at 2750 rpm and 3.3×10^9-N/m^2 (480 000-psi) maximum Hertz stress. These bearings were subsequently fatigue tested at test conditions identical to those of the baseline bearings. The fatigue tests were conducted so that each bearing was run in the same test machine in the same bearing position and direction of rotation as it was for the prestress cycle. The results of these tests are shown in figure 2 and tabulated in table II. Eight of the 27 bearings experienced fatigue failures before the 10 000-hour cutoff time was reached. Seven of these failures were inner-race failures; the other was a ball failure. Only the inner-race failures were considered as failures in the statistical analysis. The ball failure was treated as a suspension.

At the 10-percent-life level, the prestressed bearings show a life improvement over the baseline tests of approximately 2.3 times. Considering the small number of failures, the significance of this life difference is not great. The confidence number, calculated by the methods of reference 17, is only 76 percent. This number indicates that there is a 76-percent probability that the prestressed bearings are superior to the baseline bearings. At the mean life level, where the life ratio is greater than 4, the confidence that the prestressed-bearing life exceeds the baseline-bearing life is greater than 99 percent.

Failure Analysis

From the appearance of the fatigue spalls on the 15 failed inner races, two types of fatigue failure initiation were apparent. As shown in table III, only two races (from the baseline tests) had failures initiated at inclusions in the subsurface region, as determined by metallurgical sectioning. Seven races (four from the baseline tests and three from the prestressed tests) had definite evidence of fatigue spall initiation from surface indentations. This type of spall initiation has also been reported in references 19 to 21. The origin of the other six failures could not be ascertained.

TABLE III. - FAILURE INITIATION ON BEARING

INNER RACES

	Baseline	Prestressed
Total number of spalled inner races	8	7
Origin of failure:		
Subsurface inclusion[a]	2	0
Surface indentation	4	3
Unknown[b]	2	4

[a]As determined by metallurgical sectioning.
[b]Origin unknown, multiple spalls or extensive spall propagation.

A typical spall showing evidence of surface indentation initiation is presented in figure 3. The indentation is seen at the leading edge of the spall. These indentations apparently occurred during fatigue testing and were caused by foreign particles in the test

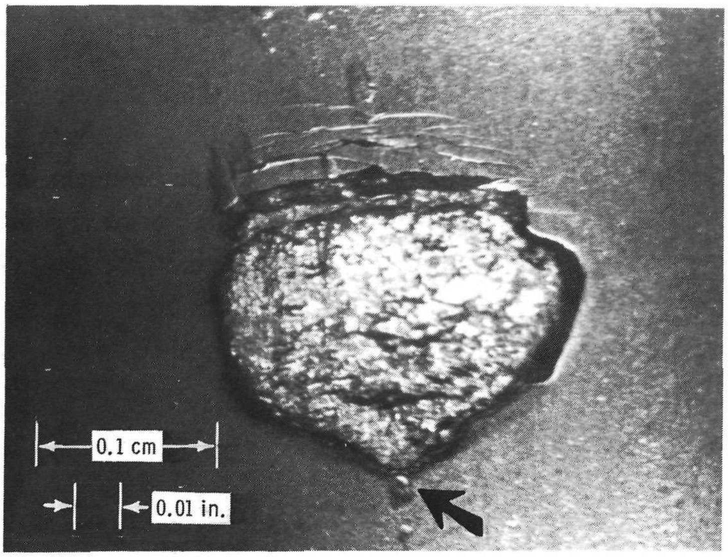


Figure 3. - Typical surface-originated fatigue spall showing evidence of surface indentation initiation (arrow).

apparatus or lubrication system. The indentations were definitely not present prior to testing.

This type of failure, being surface initiated, would be less affected by residual stress in the subsurface region than would failures originating in the subsurface region. For this reason, it is possible that the full advantage of prestressing was not realized. The only two failures that could be definitely attributed to subsurface initiation were on baseline bearings. No failures in the prestressed tests could be definitely attributed to subsurface initiation, although the origin of four failures could not be established.

Prior to fatigue testing, these bearing inner races were examined by a magnetic perturbation technique (ref. 22). This nondestructive technique was used to detect non-metallic inclusions beneath the surface of the ball track in the bearing race groove. It has the capability of determining the size, depth, and location of such inclusions. The results of this work are reported in reference 23. Inclusions were detected in the vicinity of the running track in 26 of the bearing inner races, including both the pre-stressed and baseline bearings. Subsequent to fatigue testing, metallurgical sectioning and examination confirmed the size, depth, and position of these inclusions. Both failures on the baseline bearings that were attributed to subsurface initiation were at non-metallic inclusions located by the magnetic perturbation method. Inclusions of similar size and location on two of the prestressed bearings did not cause failures even though the bearings were run to more than five times the life of these failed bearings. This result, which was presumably due to compressive residual stress in the region of the inclusions, reinforces the beneficial effect of the prestressing process.

Effect of Running Time on Residual Stress

Figure 1 shows that the maximum residual stress is induced in the first 25 hours at the prestressing load which produces a maximum Hertz stress of 3.3×10^9 N/m^2 (480 000 psi). After 25 hours and up to 200 hours, there is no increase in residual stress. A maximum was also reached in the residual stress data of reference 7 for similar test conditions (i.e., 40-mm-bore bearings of AISI 52100 steel, a radial load giving maximum Hertz stress of 3.25×10^9 N/m^2 (470 000 psi), and a shaft speed of 2000 rpm). The maximum Hertz stress in both cases was at or above the threshold stress of 3.25×10^9 N/m^2 (470 000 psi) suggested in reference 3 for inducing residual stress in rolling contact.

The maximum induced compressive residual stress in both reference 7 and figure 1 is in the range from 0.55×10^9 to 0.83×10^9 N/m^2 (80 000 to 120 000 psi). Compressive residual stresses of lesser magnitude have been observed in bearings run at stresses lower than this threshold stress (refs. 3, 4, and 8).

The effect of running time on induced residual stress at the fatigue test load which gives a maximum Hertz stress of 2.4×10^9 N/m^2 (350 000 psi) was determined from bearings run from 2 to 500 hours at this load. The residual stresses induced in these bearings are shown in table IV.

The induced tangential compressive residual stress was less than 1.4×10^8 N/m^2 (20 000 psi) and showed no significant or consistent increases between 2 and 500 hours.

Several bearings from both the baseline and the prestressed groups were chosen for residual stress measurements to determine the presence and magnitude of residual stress after longer running periods. Inner races from two bearings for each group that had run to the cutoff time without failure (4000 or 10 000 hours) were measured for residual stress beneath the groove surface.

As shown in figure 4(a) for the baseline bearings, significant compressive residual stresses are present after 4000 hours. This indicates that even at lower contact stresses, significant residual stresses may be induced after a sufficient number of stress cycles. However, data presented in reference 4 for identical bearings tested under identical conditions have shown no clear effects on induced residual stress of running times as long as 1800 hours.

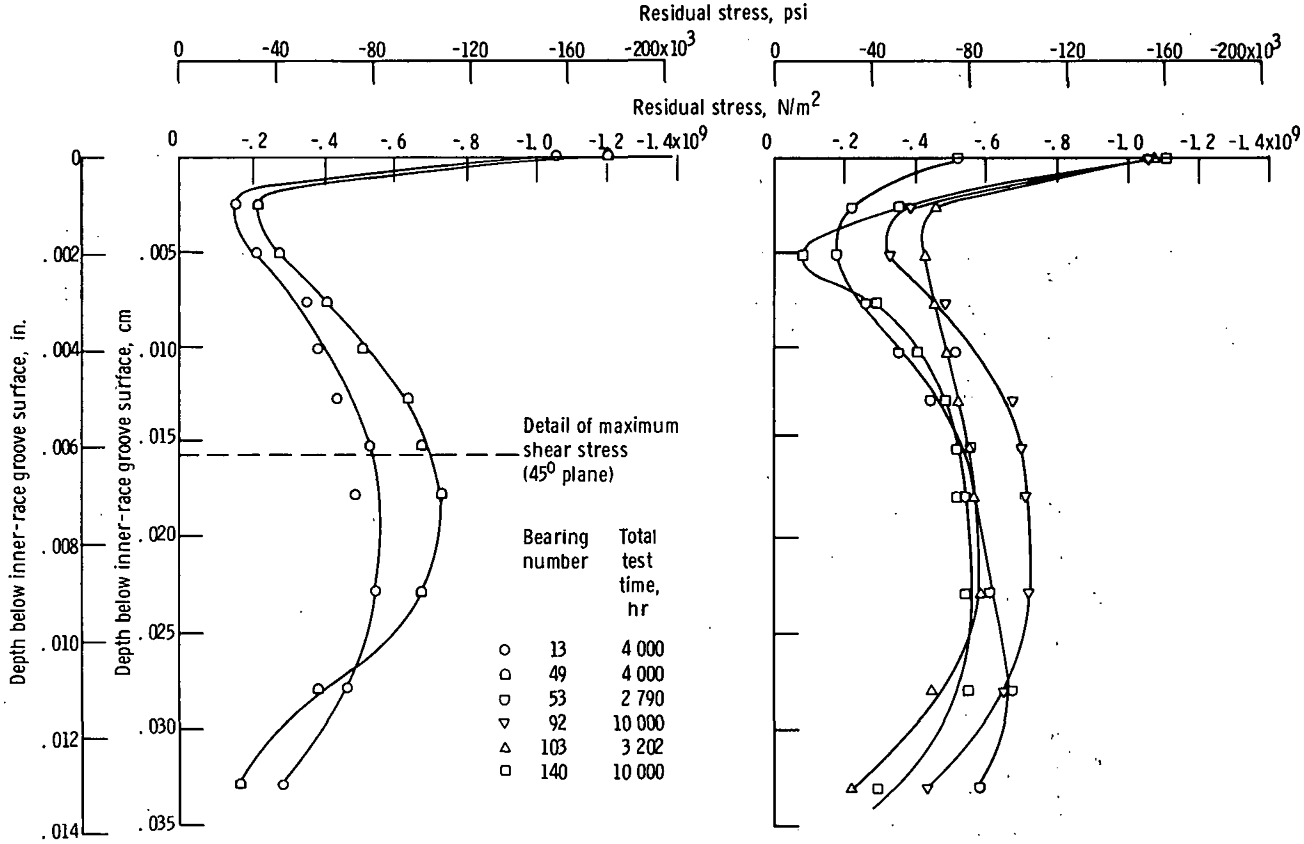
Residual stresses measured on prestressed bearings after 3000 hours and after 10 000 hours are shown in figure 4(b). Little difference exists between the stress profiles after these running times and the initial prestress profile as represented by the 25-hour data in figure 1.

The differences between the measured residual stresses in the prestressed and baseline bearings after 3000 or 4000 hours are not great. It is expected, therefore, that the major contribution to extended fatigue life of prestressing is to prevent or at least delay the early subsurface-initiated fatigue failures.

TABLE IV. - RESIDUAL STRESS MEASUREMENTS AS A FUNCTION OF DEPTH FOR 207-SIZE INNER RACES RUN FOR VARIOUS TIMES AT 2.4×10^9 N/m^2 (350 000 psi) AND 2750 rpm

Depth, cm	Running time, hr							
	2	5	10	25	50	100	200	500
	Tangential residual stress, N/m^2							
0	[a]-390×10^6	-530×10^6	-620×10^6	-510×10^6	-700×10^6	-900×10^6	-800×10^6	-690×10^6
.0025	-21	-90	55	7	-41	62	69	90
.0051	41	0	-7	35	-90	-41	-111	55
.0075	-21	21	-7	-28	-7	-28	14	0
.0102	28	-48	-62	62	-90	-28	-41	-35
.0127	21	-62	-7	-21	0	-41	69	-21
.0150	-76	-83	-69	-21	-130	-14	-7	41
.0180	-130	-130	-41	0	-140	-41	21	76
.0230	-90	-140	-120	-76	7	21	21	111
.0280	-90	-130	62	-35	62	0	-21	28
.0330	-55	-55	-21	-76	-55	-35	21	28
Depth, in.	Tangential residual stress, psi							
0	[a]-56×10^3	-77×10^3	-90×10^3	-74×10^3	-101×10^3	-131×10^3	-116×10^3	-98×10^3
.001	-3	-13	8	1	-6	9	10	13
.002	6	0	-1	5	-13	-6	-16	8
.003	-3	3	-1	-4	-1	-4	2	0
.004	4	-7	-9	9	-13	-4	-6	-5
.005	3	-9	-1	-3	0	-6	10	-3
.006	-11	-12	-10	-3	-19	-2	-1	6
.007	-19	-19	-6	0	-20	-6	3	11
.009	-13	-20	-18	-11	1	3	3	16
.011	-13	-19	9	-5	9	0	-3	4
.013	-8	-8	-3	-11	-8	-5	3	4

[a]A minus sign denotes compressive stress.



(a) Baseline bearings.

(b) Prestressed bearings.

Figure 4. - Tangential residual compressive stress as a function of depth below center of inner-race groove measured after bearings were fatigue tested.

SUMMARY OF RESULTS

Residual stress measurements were made on several bearings that were run for different time periods to determine a prestress cycle suitable for inducing significant compressive residual stresses in the inner-race - ball groove. Twenty-seven ball bearings were prestressed at an optimum prestress cycle and then fatigue tested. The results were compared with results from a group of baseline bearings that were fatigue tested under identical conditions.

The following results were obtained:

1. Compressive residual stresses in excess of 0.69×10^9 N/m^2 (100 000 psi) are induced in the region of the maximum shearing stress in the inner race of ball bearings run for 25 hours at a maximum Hertz stress of 3.3×10^9 N/m^2 (480 000 psi) and a shaft speed of 2750 rpm.

2. The 10-percent rolling-element fatigue life of ball bearings that were prestressed for 25 hours at 3.3×10^9 N/m^2 (480 000 psi) and 2750 rpm was greater than twice the 10-percent life of baseline bearings tested under identical conditions.

3. The differences between the measured residual stresses in the prestressed and baseline bearings after 3000 or 4000 hours of testing are small.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, July 27, 1972,
501-24.

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