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Factors Affecting the Service Life of Large-Diameter Wire Rope

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by
G. H. Beeman

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Pacific Northwest Laboratory
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FOREWORD

This project was initiated by the U.S. Department of Interior, the Bureau of Mines in July 1975, under Contract No. H0252060. On October 1, 1977 the contract was transferred to the Division of Solid Fuels Mining and Preparation, U.S. Department of Energy, and assigned a new number, ET-75-C-01-9099.

This project was administered under the technical direction of the Spokane Mining Research Center with Mr. Thomas Brady acting as the technical project officer.

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SUMMARY

Service life of large-diameter (>2-in) wire rope is a major concern in the surface coal mining industry. There is great diversity in the maintenance and retirement procedures used for large-diameter wire rope. Differences in rope construction may play an important role in service life and performance. Many of the factors affecting large rope service life could be economically studied and remedial proposals evaluated using small rope if results from small rope tests could be reliably projected to large rope behavior.

An experimental study to explore this possibility was begun in July of 1975. The title of this study was "Investigate the Feasibility of Improving Efficiency of Surfaces Mining Equipment by Extending Rope Life." Subsequently, a fatigue machine to test large-diameter wire rope was designed, constructed, and operated by Battelle, Pacific Northwest Laboratories through support from the U.S. Bureau of Mines, Spokane Mining Research Center under contract No. H0252060. A testing methodology was developed, and rope-over-sheave fatigue testing of 23 three-inch diameter ropes from two manfacturers was conducted at loads ranging from less than one-third to more than one-half the ropes' catalog breaking strength.

Fatigue life results were plotted and compared with results from small-diameter wire rope by employing the Drucker-Tachau Bearing Ratio.[1] The factor is a ratio of rope load to the product of wire strength, sheave diameter and rope diameter. If existing small rope data could be correlated with large rope test results based on this ratio, some confidence in extrapolating small rope development-testing results to large ropes would be generated. It was found that failure data for the two rope sizes do not correlate on the basis of curve shape or numerical agreement at cyclic lifetimes in the 10^3 to 10^4 curve range. Fatigue lives of the large rope were dramatically shorter at the higher loads. Lack of correlation may be due to basic differences in construction, but most probably are due to differences in core material. Metallurgical differences do not appear to be significant between large and small ropes. The large ropes tested were of independent wire rope core (IWRC)

construction, while the data used for comparison resulted from tests of small-diameter rope with fiber core construction. All larger ropes tested were of lang lay construction while smaller ropes were regular lay. These construction differences may, in turn, contribute to different rope-over-sheave damage dynamics at high fatigue or tensile loads.

Examination of fatigue-tested ropes disclosed broken wires, cracked wires and wear patterns. Relatively few cracked wires were found; microscopic fatigue striations could be identified in only a few of the broken wires. Most severe damage was in double-flex regions caused by wire notching between strands and the IWRC, not between adjacent strands.

Selected sections of field ropes from seven mines at scattered sites across the U.S. were also examined for broken wires, cracks and wear. From 29 hoist and 32 drag rope sections dissected, worst damage was found in the drag rope near the sockets. Wire notching between strands and the IWRC was seldom more severe than external abrasion. Hoist rope damage was generally worst due to rope on sheave abrasion at the point sheave.

Based on the markedly deleterious effects of high loads in the bending fatigue tests, and the observations of damage in both drag and hoist ropes, it is concluded that rope life in large-diameter surface coal mining draglines can be extended by reducing overloads on both hoist and drag, and by reduced drag rope whip, in addition to other good maintenance practices that would lessen external abrasion.

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FACTORS AFFECTING THE SERVICE LIFE OF LARGE-DIAMETER WIRE ROPE

INTRODUCTION

The use of large-diameter (>2 in.) wire rope has increased steadily since the end of World War II. Ropes up to 5 in. in diameter are currently in use in the surface coal mining industry. The replacement costs of the large-diameter wire ropes used on a dragline are substantial. Currently, drag ropes last about 10 weeks and hoist ropes about twice that long. However, rope lives may vary by as much as a factor of 2 from mine to mine. Ropes are usually retired because they are "worn out"; however, there is no standard procedure for determining safe service life of the ropes. In all probability, some ropes are retired that have many hours of good service remaining. If a method could be found to reliably predict safe service life of wire rope used on draglines, a savings could result in time and materials. Also, if a means could be found to improve the service life of these large-diameter wire ropes, substantial additional savings would result.

This report documents the findings of a research project conducted by Battelle, Pacific Northwest Laboratories for the U.S. Bureau of Mines. This project consisted of three phases:

- Phase I - Information Acquisition,
- Phase II - Acquisition of Baseline Experimental Data, and
- Phase III - Evaluation of Potential Improvements. Work was begun in July 1975 and completed in February 1978.

During Phase I, literature concerning large-diameter wire rope was reviewed. The literature review is found in Appendix A. A substantial number of field personnel in the surface coal mining industry were contacted. Data were gathered on the aspects of surface coal mining operations that involved wire rope. Results from Phase I show that service life and maintenance practices for large-diameter wire rope used on surface coal mining draglines differ widely.[a] Currently, there is no standard criterion for retiring large-diameter wire rope. Rather, rope retirement is based on experience and a "feel" by the operator that it is worn out.

During Phase II of the project, a machine to test 3-in. diameter wire rope in bend-over-sheave fatigue was designed and constructed. Actual testing of wire rope began in January 1976. Phase II testing was completed in May 1977. Also included in Phase II was the collection and examination of retired wire rope from several U.S. surface coal mines.

Work done during Phase III consisted of continued baseline evaluation of wire rope in bend-over-sheave fatigue. Because of the complex nature of wire rope, specific methods for improvement were not evaluated at this time; however, failure mechanisms were identified and methods to stop or slow these mechanisms were postulated. Phase III testing began in May 1977 and ended in January 1978.

When Phase III ended, a total of 23 ropes had been tested in bend-over-sheave fatigue. Of these, 17 were tested to failure.[b] All ropes tested were examined further by taking specimens from different areas and dismantling the rope down to individual wires. To date, 61 field ropes have also been examined in the above manner.

One of the major goals of the project has been to see if bend-over-sheave fatigue data from small-diameter wire rope could be compared with that of large-diameter wire rope via the Drucker-Tachau[1] ratio. This ratio has proven useful in predicting the service life of small-diameter wire rope in rope-over-sheave fatigue. The Drucker-Tachau ratio is a dimensionless bearing pressure ratio and is defined as

$$\beta = \frac{2T}{UDd}$$

(a) This is discussed in detail in Appendix B.
(b) Failure is defined as damage such as that causing the rope's retirement in the field.

where

T = tension in the rope, lb.

U = ultimate tensile strength of the wires, lb/in.^{2}

d = diameter of the rope, in.

D = diameter of the sheave, in.

Figure 1 shows the results from several laboratory studies of bend-over-sheave testing for several small-diameter wire ropes plotted as a function of β (Drucker-Tachau ratio) and cycles to failure. As shown, there is good agreement among ropes from 1/2 in. to 1 3/16 in. in diameter. Because of the relatively high cost of testing large-diameter wire rope, a method for predicting the service life of large-diameter wire rope from small rope tests would be most beneficial.

TESTING METHODS

A machine to test large-diameter wire rope in fatigue was designed, constructed and operated by Battelle, Pacific Northwest Laboratories for the U.S. Bureau of Mines.[a] The machine is capable of fatigue testing 3-in. diameter wire rope in tension under loads up to 850,000 lb. The machine was designed so that two ropes can be tested simultaneously. A load of 425,000 lb can be applied to each rope; this is ∿50% of the catalog breaking strength of 3-in. diameter wire ropes. The ropes are tensioned by a 24-in. diameter hydraulic cylinder to the desired load and cycled ∿20 ft over 90-in. diameter sheaves (see Figure 2) for a sheave-to-rope ratio of 30:1. After the ropes have been fatigue-tested, they can be tensile-tested to determine their remaining strength.

Procedures for socketing the rope are outlined in Appendix D. Wedge type sockets, the same type as used in the field, were used during all testing. After socketing, ropes are placed on the fatigue machine and connected at the top and bottom. The bottom connection links the rope to the cycling cyclinder. The top connection is made at a link that joins the two sockets together. Control of the load is obtained by adjusting the pressure on the load cylinder.

(a) See Appendix C.

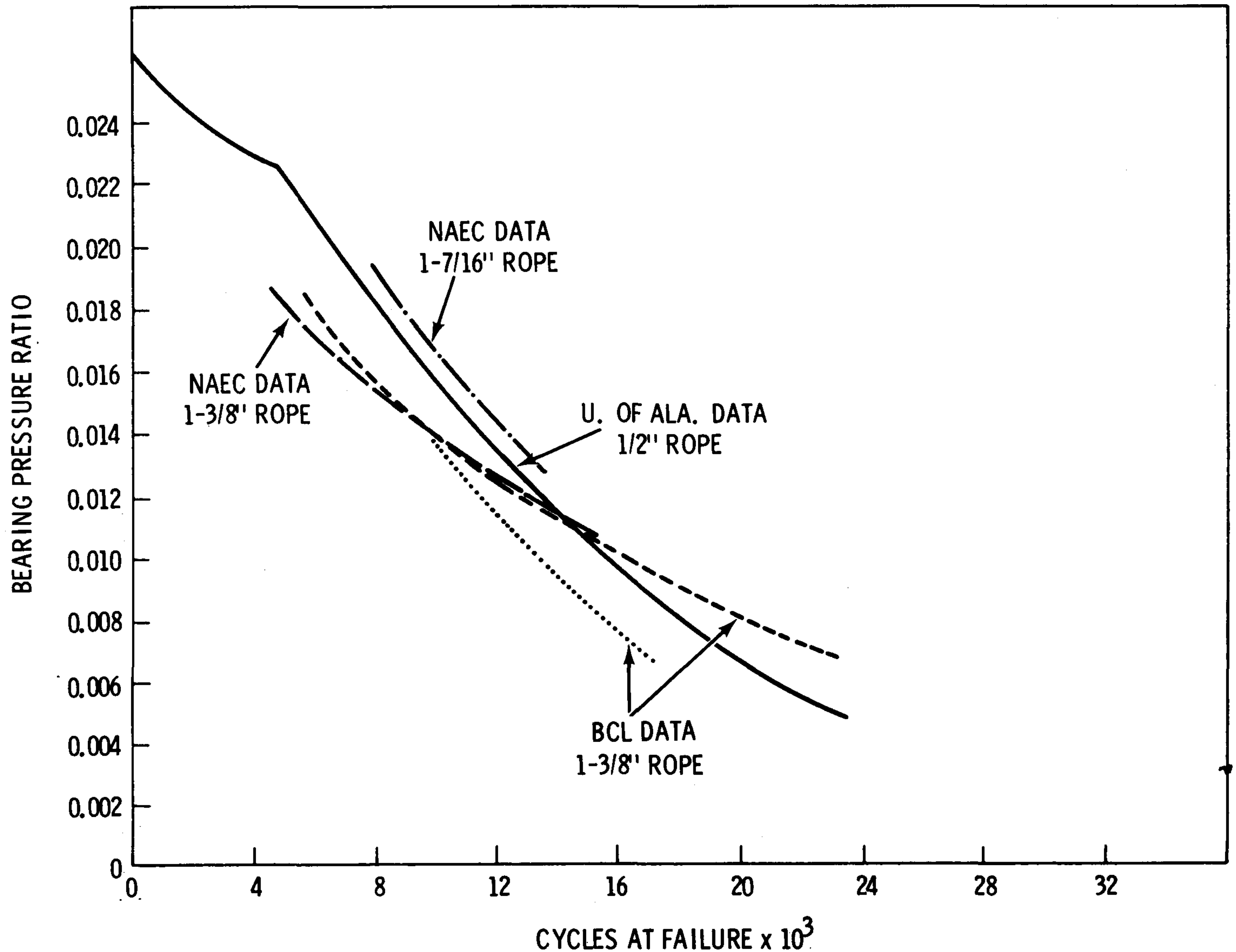
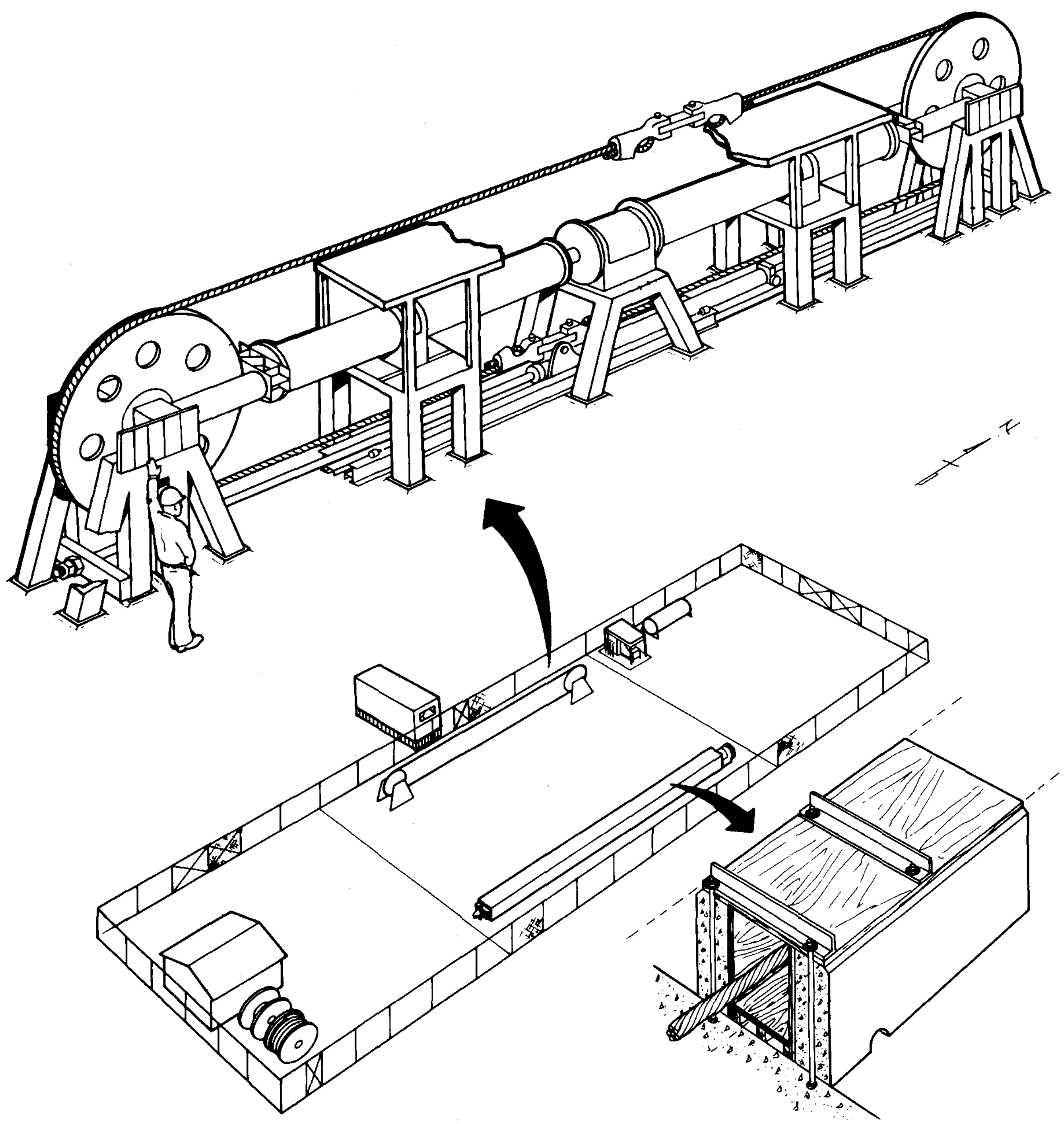


FIGURE 1. Small-Diameter Rope Test Results Using Drucker-Tachau Ratio



(Three-inch diameter capacity; machine owned by the U.S. Department of Energy and installed on Battelle-Northwest property)

FIGURE 2. Wire Rope Fatigue Test Machine and Tensile Test Facility

Traverse of the cycling cylinder from one end to the other and back constitutes one cycle. During Phases I and II, average elongation of the rope was obtained by measuring the extension of the load cylinder. During Phase III, extensometers were placed on each rope, so that the stretch of the rope was measured accurately (see Appendix C).

FATIGUE TESTING

The fatigue machine subjects the rope to three different areas of strain, illustrated in Figure 3. In the no-flex regions, part of the rope is never cycled over a sheave. Part of the rope is cycled onto the sheave and back off in the single-flex regions. Part of the rope is cycled completely over the sheave in a double-flex mode. The double-flex region experiences two flexural cycles for each counted cycle, analogous to the double flexing of field rope during one complete load and dump cycle.

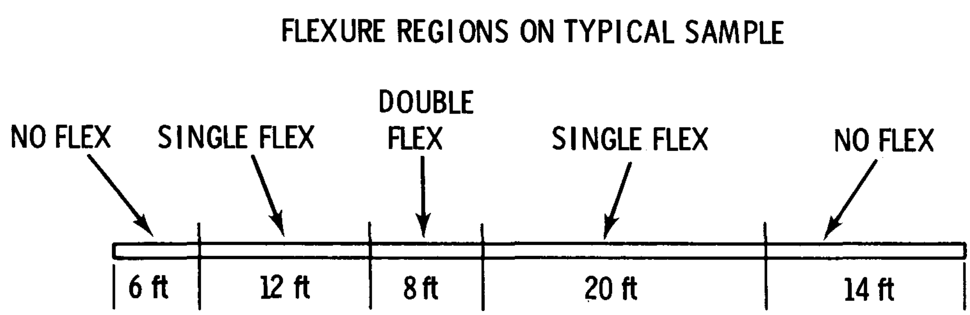


FIGURE 3. Flexure Regions on Typical Rope Sample

Fatigue testing was carried out on 23 ropes from two different manufacturers. Ropes tested were 3-in. diameter of the 6 x 37 class with an independent wire rope core (IWRC). Both right and left lang lay ropes were used. Tests were conducted at loads ranging from 249,000 lb to 424,100 lb. These loads ranged from about 30% to 60% of the catalog ultimate tensile strength (UTS)[a] of the rope. Ropes were cycled until one of the pair sustained

(a) UTS of the rope is taken as the catalog value for breaking strength. This may be low by as much as 30%. Specific ultimate breaking strength data are not available from the rope manufacturers at this time.

damage that would have caused its retirement in the field. In all cases, this was one or more broken strands. The number of cycles at failure was recorded. The failed rope was photographed and samples were taken to attempt to determine the cause of failure. Some nonfailed ropes were tensile-tested to determine remaining strength. Loads in the tensile tester were limited to 680,000 lb to prevent damage to the sockets.

Testing procedures varied somewhat from Phase II to Phase III. When a rope failed in Phase II, both ropes were removed and some of the ropes were tensile-tested. During Phase III, only the failed rope was replaced and the test continued. This allowed for generation of more failure data and also produced some insight into rope behavior at mixed loads.

TENSILE TESTING

In some instances after fatigue testing, one rope of the pair was tensile-tested on the apparatus shown in Figure 2. This was usually the rope with the least amount of visible damage. An hydraulic cylinder like that on the fatigue machine was used to tension the rope with a pressure transducer to monitor load on the cylinder. An extensometer was placed on the rope to measure deflection. Curves of load versus deflection were obtained for each tensile test (see Appendix E). As mentioned before, loads in the tensile tester were limited to 680,000 lb. This is about 80% of the catalog UTS of rope from manufacturer D. It is about 95% of the catalog UTS of rope from manufacturer B. Sockets used were a wedge type. Because of the expense of the sockets and long lead times for socket replacement, damage to the sockets during tensile testing could not be tolerated. For this reason, only one rope was tested to failure in the tensile tester.

ROPE EXAMINATION

After fatigue or tensile testing, rope samples one lay length long were taken from each of the three regions.[a] One strand was chosen from each

(a) Samples from each region were examined during Phase II of this project. However, because the no-flex region samples provided little or no wear data, they were not examined during Phase III.

sample for examination. The strands were dismantled into individual wires and the wires were cleaned. Each wire was then examined for cracks by magnetic particle detection techniques (see Appendix F). Cracks were broken open and characterized as to size, initiation and cause.

The entire testing procedure is sequentially outlined in Appendix G.

RESULTS

FATIGUE AND TENSILE TEST RESULTS

Since the project began, 23 ropes in all have been tested in bend-over-sheave fatigue. During Phase II, 13 ropes were tested; 7 of these were tested to failure. Of those 13 ropes, 5 were tensile-tested, and 1 rope was carried over into Phase III. Ten of the 11 ropes tested during Phase III were fatigue-tested to failure. The results of these tests are discussed on page 26.

Results for the fatigue and tensile tests are summarized in Tables 1 and 2, respectively. Data from the mixed load tests are summarized in Table 3. General operating procedure for the fatigue machine requires that it be run a few cycles at a load below the test load. Those cycles were recorded, and the adjusted number of cycles represent the cycles at the actual test load. During the second fatigue test, problems developed with the bearings on the fatigue machine. These problems prevented running at the test load for any extended length of time. This accounts for the large difference in total and adjusted cycles for this test. It is not known at this time what effect the large number of cycles run at low load had on the overall life of the rope.

Table 3 shows the results of those ropes tested under two different loads. In all cases, load 1 was the first load applied. Other than this, these ropes underwent no other special treatment.

Five ropes were tested in the tensile tester. A load limit of 680,000 lb was imposed to prevent damage to the sockets. Both ropes from fatigue test 3 were tensile-tested. The fatigue-failed rope had a single broken strand. It failed at 340,000 lb; the companion rope did not fail at twice the load.

TABLE 1. Fatigue Test Data Summary

Rope No.	Load, lb	Manufacturer	Cycles to Failure Total	Cycles to Failure Adjusted
1	249,000	D	30,805	25,535
1a	249,000	D	30,805	25,535[a]
2	249,000	B	76,002	42,356
2a	249,000	B	76,002	42,356[a]
3	330,000	D	15,408	14,673
3a	330,000	D	15,408	14,673[a]
4	330,000	B	21,933	21,880
4a	330,000	B	21,933	21,880[a]
5	413,000	D	1,387	1,341
5a	413,000	D	1,387	1,341[a]
6	413,000	B	307	254
7	413,000	B	784	609
8	348,300	B'	15,159	14,732
9	348,300	B'	13,333	13,104
10		See Table 3.		
11		See Table 3.		
12	424,100	B'	14,440	14,079
13	424,100	B'	1,590	1,462
14	424,100	B'	1,304	1,277
15		See Table 3.		
16	278,200	B'	22,116	22,043
17	278,200	B'	24,939	24,839
18	278,200	B'	21,645	21,557[a]

(a) Did not fail.

TABLE 2. Tensile Test Data Summary

Test	Manu-facturer	Fatigue Load, lb	Cycles	Applied Load, lb	Apparent Modulus of Elasticity x 10^6 psi
1	B	249,000	76,002	680,000	9.16
2	D	330,000	15,408	680,000	12.6
3	D	330,000	15,408	340,000[(a)]	9.6
4	B	330,000	21,933	680,000	10.8
5	D	413,000	1,387	680,000	13.9

(a) Initially, rope had one broken strand and failed in the tensile tester at the indicated load.

TABLE 3. Mixed Load Failure Data

Rope No.	Load 1	Load 2	Cycles at Load 1		Cycles at Load 2		Cycles at Failure	
			Total	Adjusted	Total	Adjusted	Total	Adjusted
10	413,000	348,300	477	408	33,185	32,575	33,662	32,983
11	348,300	424,100	4,693	4,670	12,358	12,128	17,051	16,798
15	424,100	278,200	789	777	25,410	25,325	26,199	26,102

Figure 4 shows data from the bend-over-sheave fatigue tests on a Drucker-Tachau plot along with those obtained from small-diameter wire rope. The values used to obtain Drucker-Tachau ratios for the rope are shown in Table 4. Table 5 shows the cycles at failure predicted from the Drucker-Tachau ratio using the curve for 1 3/8-in. rope and the actual cycles at failure. Very little agreement is found between data from large- and small-diameter wire rope tests. Few ropes failed within 10% of the cycles predicted from the small-diameter rope tests. At the highest test loads, ropes attained only 3% to 12% of their expected life. The reasons for this are discussed on page 28.

Figure 5 compares current data for 3-in. wire rope with that for 1 3/8-in. diameter wire rope. It appears that 3-in. diameter wire rope has a longer life at low loads and shorter life at high loads, when compared to small-diameter rope using the Drucker-Tachau ratio.

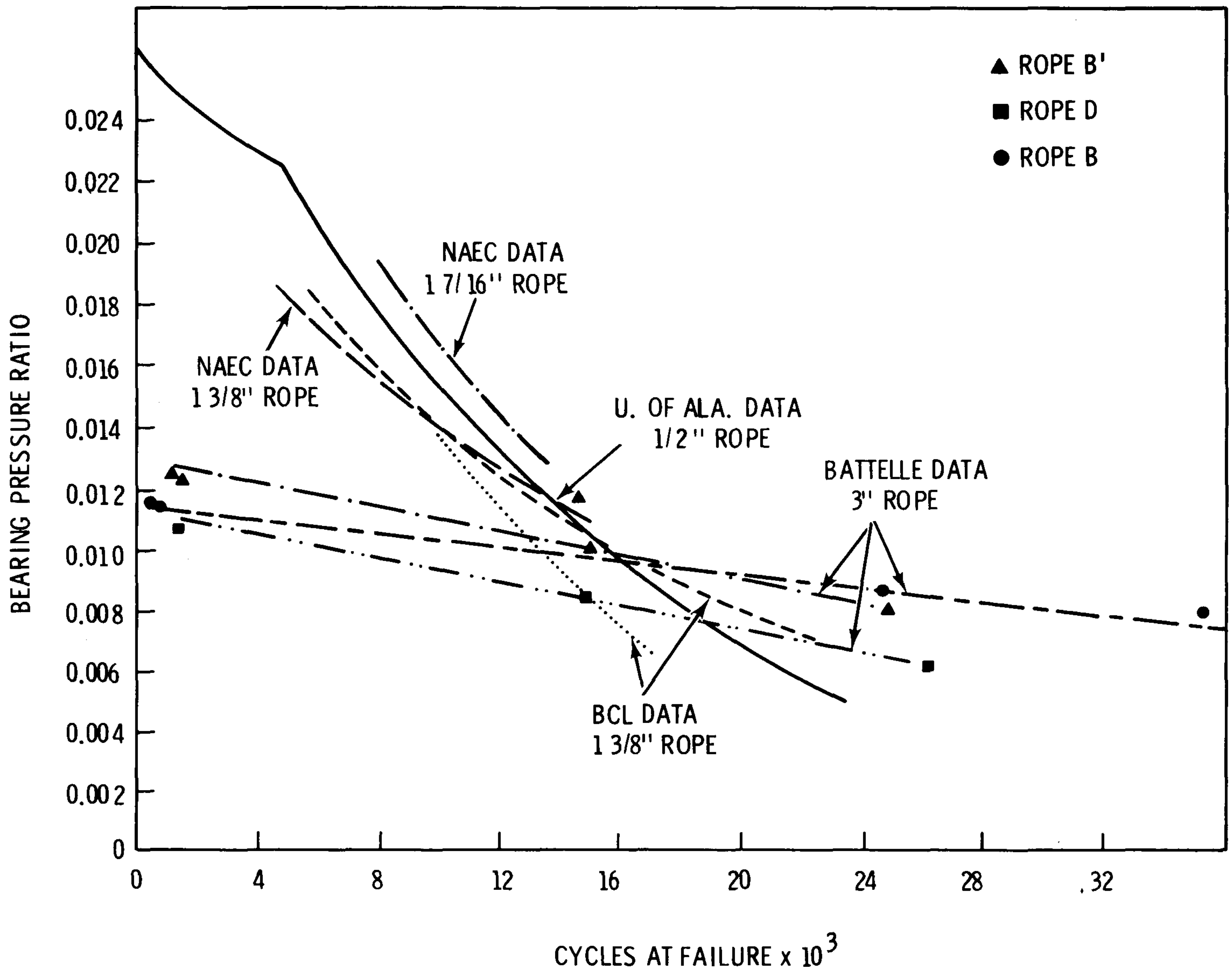


FIGURE 4. Small- and Large-Diameter Rope Data Compared Via the Drucker-Tachau Ratio

TABLE 4. Drucker-Tachau Ratio Values for Large-Diameter Test Ropes

$$\beta = \frac{2T}{UDd}$$

T = tension in the rope
U = UTS of the wires[a]
D = diameter of sheave
d = diameter of the rope

Rope Manufacturer B

U = 230 Ksi
D = 90 in.
d = 3 in.

T (kips)	β	Percent UTS of Rope (360 tons)
249	0.008019	34.6
330	0.010628	45.8
413	0.013301	57.4

Rope Manufacturer D

U = 260 Ksi
D = 90 in.
d = 3 in.

T (kips)	β	Percent UTS of Rope (425 tons)
249	0.007094	29.3
330	0.009402	38.8
413	0.011766	48.6

Rope Manufacturer B'

U = 240 Ksi
D = 90 in.
d = 3 in.

T (kips)	β	Percent UTS of Rope (398 tons)
278.2	0.008586	34.9
348.3	0.010750	45.8
424.1	0.013090	53.3

(a) Values of U were obtained by tensile test for each rope manufacturer. Results are reported in Appendix H.

TABLE 5. Predicted Cycles at Failure

Manu-facturer	Test Load	β	Actual Cycles[(a)]	Predicted Cycles	Percent Expected Life[(b)]
D	249,000	0.007094	30,805	21,900	117
B	249,000	0.008019	42,356	19,200	221
D	330,000	0.009402	14,673	16,500	89
B	330,000	0.010628	21,880	15,000	146
D	413,000	0.011766	1,341	13,500	10
B	413,000	0.013301	254	10,000	3
B	413,000	0.013301	609	10,000	6
B'	278,200	0.008586	14,732	18,400	80
B'	278,200	0.008586	13,104	18,400	71
B'	348,300	0.010750	22,043	15,100	146
B'	348,300	0.010750	24,839	15,100	164
B'	424,100	0.013090	1,462	12,500	12
B'	424,100	0.013090	1,277	12,500	10
B'	424,100	0.013090	14,079	12,500	113

(a) Adjusted for cycles not at load.
(b) As compared with expected life from Drucker-Tachau plot for 1 3/8-in. diameter rope.

Initial examination of the failed ropes revealed that the majority of broken wires in the rope's failed section failed in a brittle manner when the rope was tested at loads running between 30% and 46% of the rope catalog UTS. At higher loads, the dominant mode of wire failure was ductile. All ropes failed in the double-flex region. Photographs of some failures are contained in Appendix I.

Figure 6 is a plot of load versus cycles (S-N curve) for all ropes tested. More data are needed at the lower loads to complete the curve. If the current downward trend of the curve continues, there may be no practical endurance limit for large-diameter wire rope when tested in bend-over-sheave fatigue.

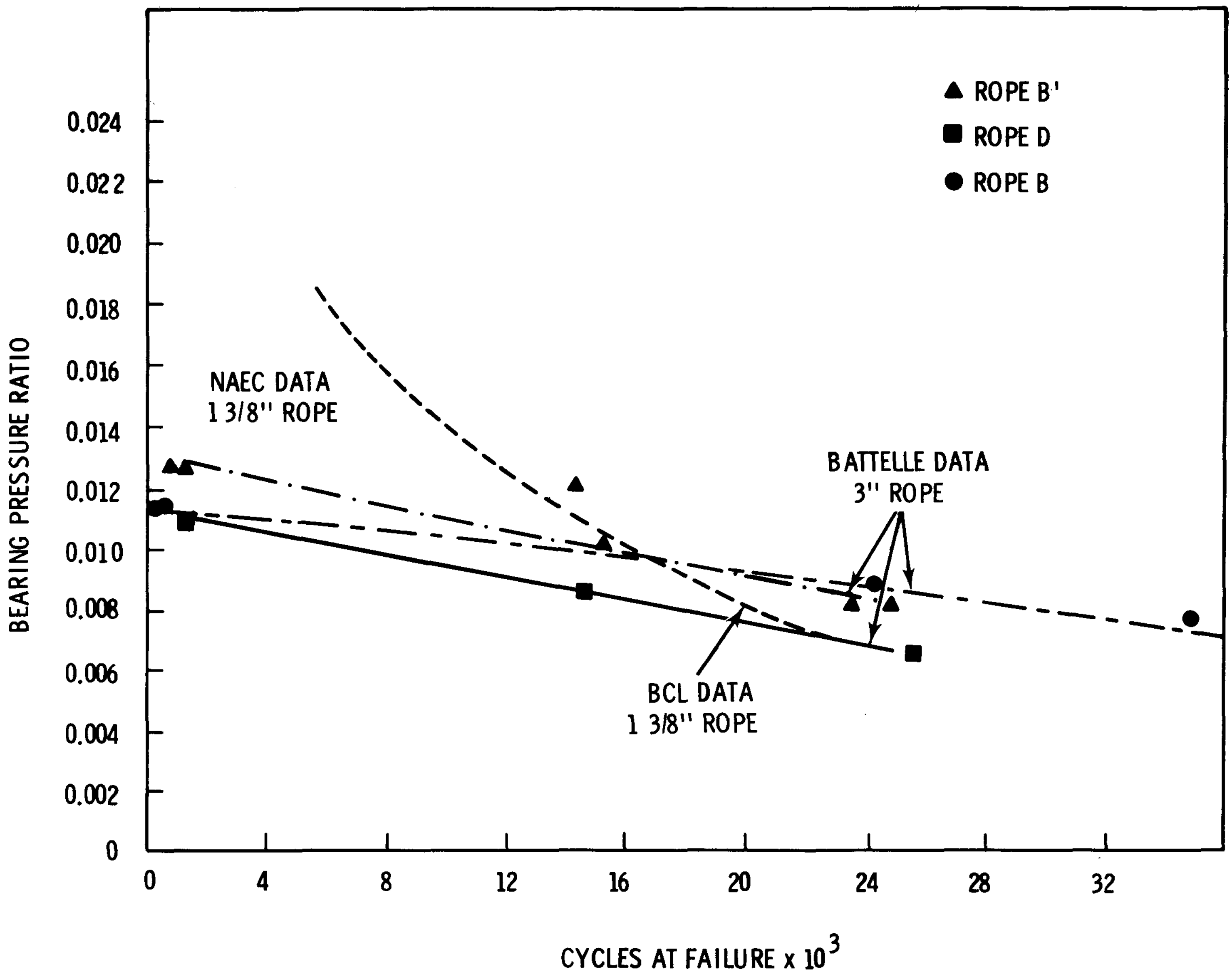


FIGURE 5. Comparison of 1 3/8-inch and 3-inch Rope Data Via Drucker-Tachau Ratio

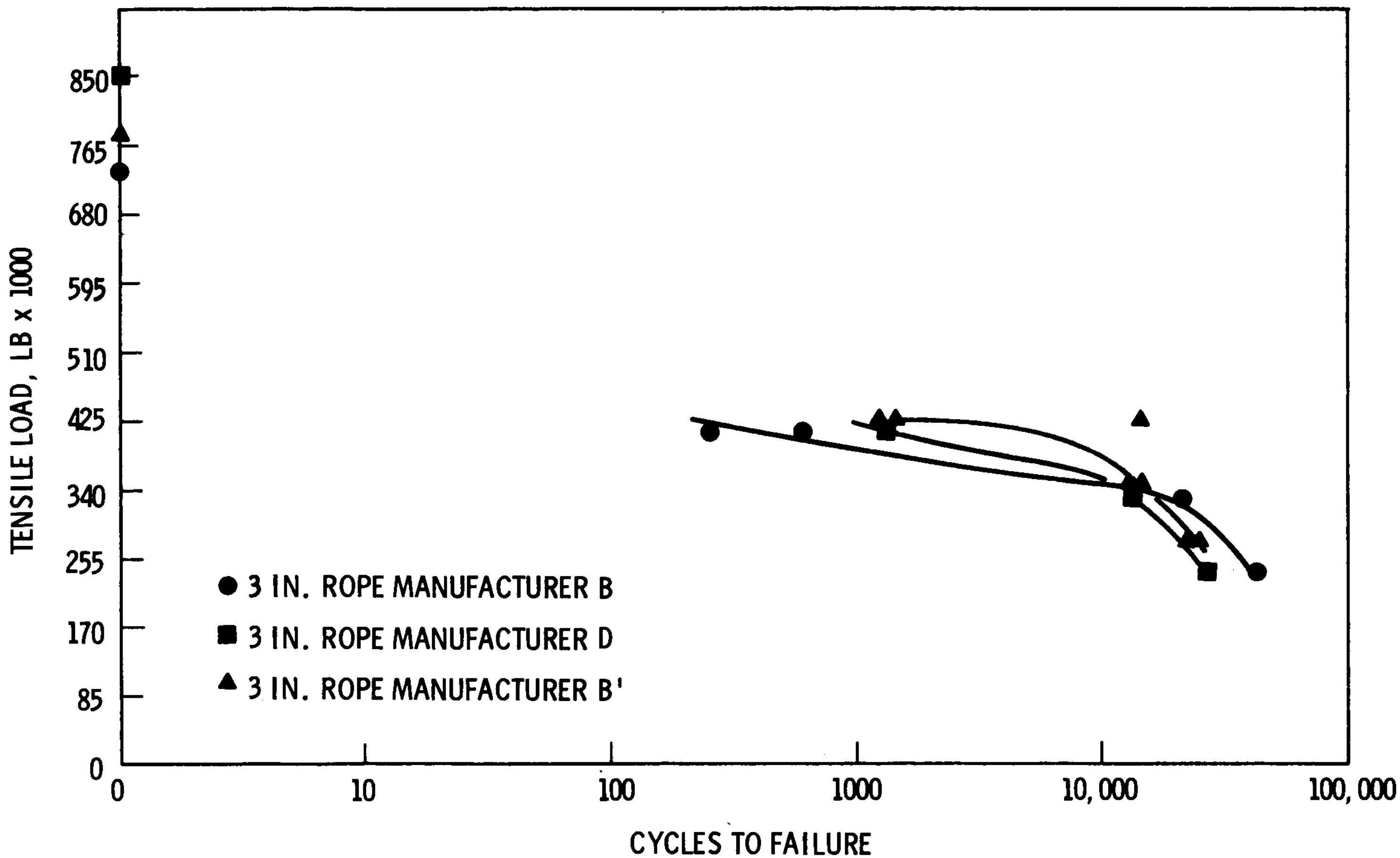


FIGURE 6. S-N Type Curve for 3-inch Diameter IWRC Wire Rope

Bending of the rope causes bending stresses that are related to the diameter of the rope, the diameter of the bend and the modulus of elasticity of the rope. When a rope is bent around a sheave, this stress may be above the endurance limit of the rope at a 30:1 sheave to rope ratio (D/d). Figure 7 shows additional data from small-diameter fiber core rope. These exhibit the same downward trend; D/d ratios ranged from 18:1 to 21:1.

ROPE ELONGATION

Figure 8 represents elongation of a rope as a function of the test cycle. Regardless of test load, all rope tested exhibited this type of elongation behavior. The only variation was a shortening of all three stages with increasing load as shown in Figure 8.

The steep slope of the curve seen during Stage 1 comes from the initial running in of the rope. A 6 X 37 class 3-in. diameter IWRC wire rope generally has more than 250 individual wires of varying sizes. Such composite nature of the rope encourages an initial elongation due to relative movement of the wires as they become more tightly packed, and leads to a reduction in rope diameter. This diameter change is greater in the double-flex region and less significant in those areas of the rope that are never cycled over a sheave. At high loads this initial compaction does not occur. The high loads lock the wires in place and prevent the wires from moving. This causes more severe notching at high loads.

The second stage of the elongation curve is the longest of the three. During this stage, elongation increases at a fairly constant rate. It is during this stage that most individual wire wear occurs. This wear will be discussed in more detail in a following section. Depending on the rope manufacturer, and probably the wire UTS, broken wires begin to appear during this stage. Wires usually break at some internal contact point and work out to the surface. In some tests, as many as 20 broken wires were found before failure. At the highest loads, no broken wires were found before failure.

The third and final stage has a steep slope and is of short duration. It may be from 10 to 100 cycles long, depending on the load, with the fewest number of cycles occurring at the highest loads. In some cases, particularly

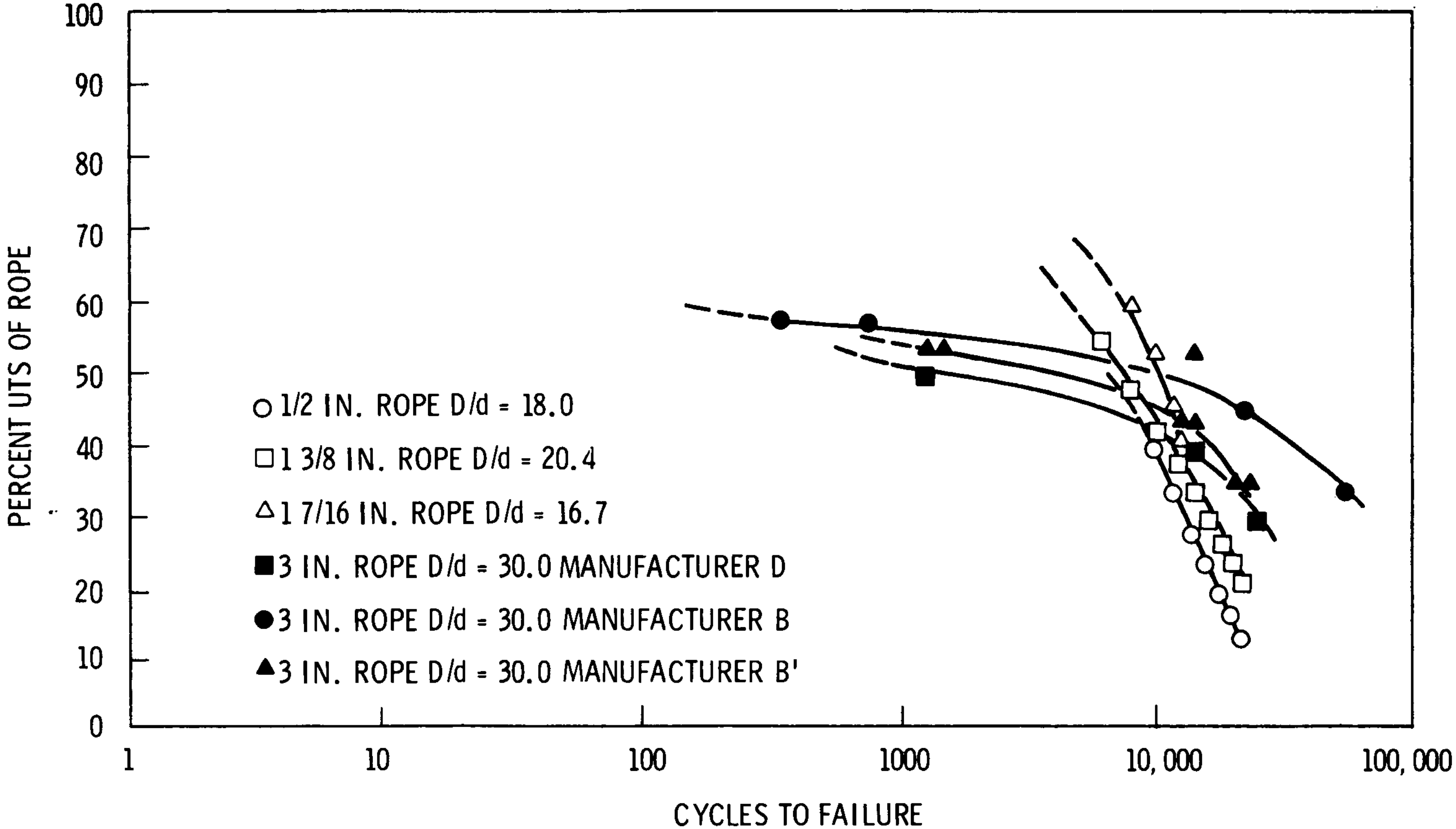


FIGURE 7. Comparison of Small- and Large-Diameter Rope Data

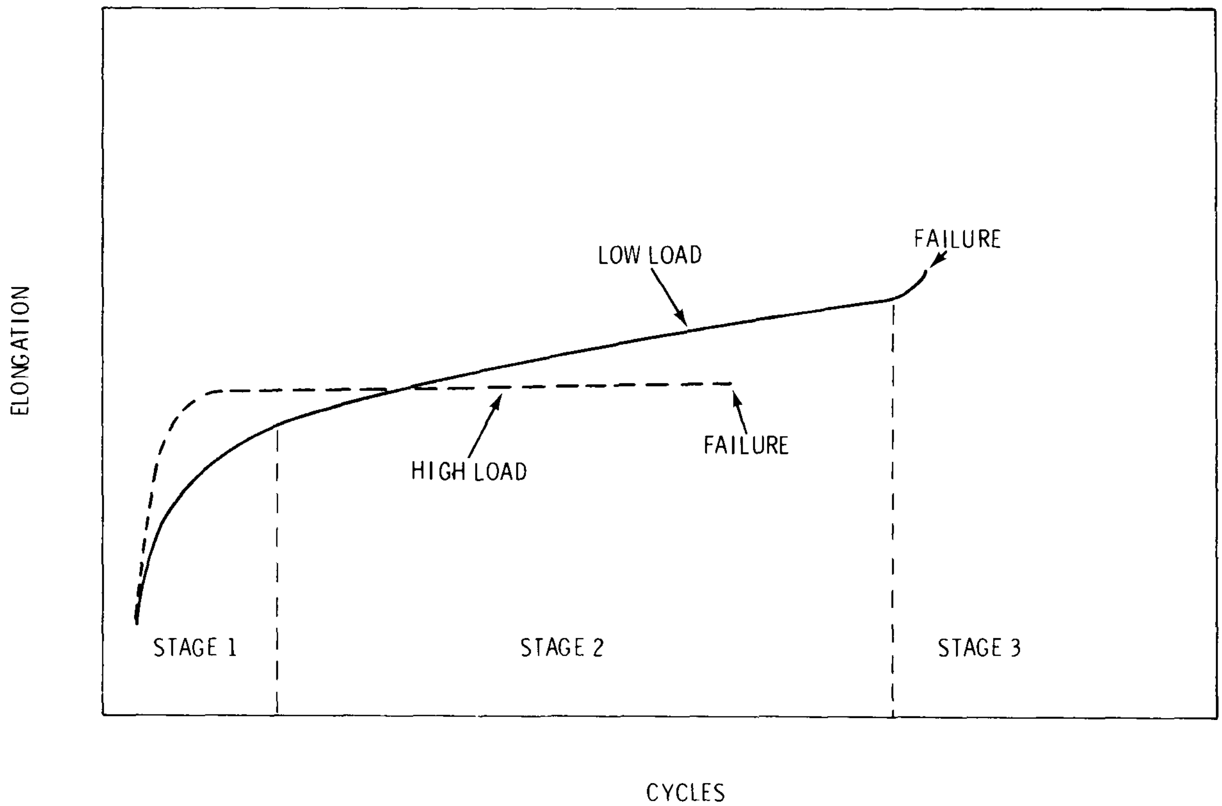


FIGURE 8. Rope Elongation Versus Test Cycles

at the higher loads, this final stage was not clearly defined. (Failure of wire rope usually occurs in a localized area. For this reason, any change in length at failure is small when compared to the overall length of the rope. The effect may obscure or completely eliminate the third stage of elongation.) During this final stage, more wires break; however, they may not become externally visible because of the short duration of this stage. Such wire breakage leads to a reduction in the load-carrying capacity and failure is imminent, Usually in one or two strands. At the lower loads, the majority of wires in the failed area have the appearance of brittle failure. At loads above approximately 40% of the catalog UTS for the rope, the majority of wires in the failed region failed in a ductile manner.

MIXED LOAD TESTS

Three ropes were tested at two different loads. The were tested initially for periods ranging from 33% to 75% of their expected life at that load. The load was then changed and the test continued until failure. These data are summarized in Table 3. Figure 9 is a schematic representation of the load history of the three ropes. Expected and actual life are compared in Table 6. Expected cycles are obtained by estimating from final loading only and using the average number of cycles from previously obtained data.

FIELD ROPE EXAMINATION

Fairly early in the program, it was realized that if the bend-over-sheave fatigue tests were to be representative of actual field service, they must produce the same types of rope damage seen in the field. Toward this end, retired field rope specimens were obtained from seven U.S. surface coal mines. A total of 61 rope samples from various locations were examined, taken from 29 hoist ropes and 32 drag ropes. Rope diameters varied from 2 7/8 in. to 4 1/2 in.

Each rope was cut into sections one lay length long. These samples were then dismantled to strands, one strand dismantled further into individual wires. The number of broken wires and any other findings of interest were recorded. Wear of outer wires at the three wear points was also noted.[a] Table 7 is a summary of the findings from the field rope examination.

In hoist ropes, wear from contact with the point sheave generally accounted for the most wear. In some cases, outer wires were worn to nearly 50% of their original cross-sectional area. Sheave groove marks and some amount of sheave overspin may cause the external damage. Wear between the strands and the IWRC was not as severe as abrasion from the point sheave, but it was greater than that from adjacent strands. The IWRCs are severely worn in this area and may contain numerous breaks and cracks.

(a) Wear points were at outer, strand to strand, and strand to IWRC locations; see Figure 10.

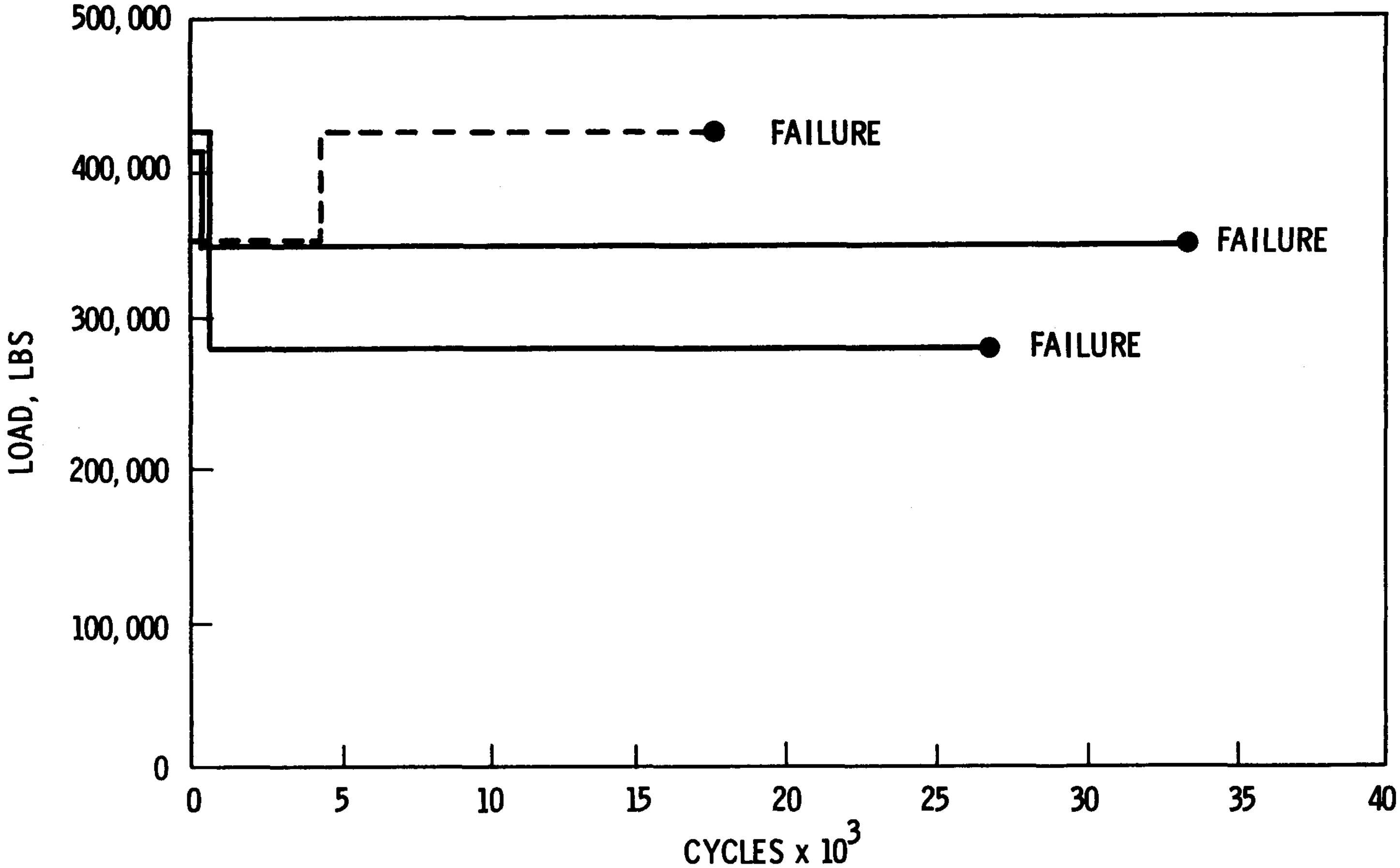


FIGURE 9. Mixed Load Testing

TABLE 6. Mixed Load Life Data

Rope No.	Final Load (lb)	Expected Cycles	Actual Cycles	Percent Increase in Life
10	384,300	14,250	32,983	131
11	424,100	1,440	16,798	1067
16	278,200	23,550	26,102	11

This is discussed in detail on page 34.

Near the socket, the outer wear of the hoist ropes probably results from being run on and off the drum before the ends are switched. Rope that was near the socket when finally removed may have a severely damaged core. This could be due to shock loadings from hitting rocks and may also be a result of bucket overloading. In this area, wear between the strands and the IWRC is greater than wear between adjacent strands.

Wear of the hoist rope near the drum is minimal, if the ends have never been switched. In some cases, the rope appeared to be brand new; the original lubrication was still intact.

The drag rope usually sees more severe service conditions than does the hoist. As far as outer wear is concerned, the portion of the rope running over the fairleads and that out near the socket encounter the most wear. This is a result of dragging the rope through the roll and from sheave over-spin. Sheave overspin appears to be a major contributor to external damage on drag ropes.

Near the socket, wear between the strands and the IWRC is greater than anywhere else (including this type of wear in the hoist rope). The IWRC in this area is usually severely damaged. This appears to be the result of loadings unique to their location. Standing waves develop in the drag ropes at certain times during the digging cycle. These waves produce bending and significant strain in the rope. When they reflect off the bucket jewelry, impulse loads and kink strains may be imparted into that part of the rope within 20 ft of the socket, adding to the damage caused by hitting rocks and overloading the bucket.

TABLE 7. Examination of Field Ropes

Rope Size	Classification	Mine No.	Sample	Location	Bucket Size, yd^3	Yards Removed	Wear Outer	Wear IWRC	Wear Strand	Broken Wires
3 in.	6x57	11	Drag	Near Socket	58	2,702,279	19%	10%	14%	0
3 in.	6x57	11	Drag	Over Fair Leads	58	2,702,279	40%	19%	12%	0
3 in.	6x57	11	Drag	Drum End	58	2,702,279	8%	5%	5%	0
3 in.	6x57	11	Hoist	Point Sheave	58	2,702,279	5%	7%	3%	0
3 in.	6x57	11	Hoist	Drum End	58	2,702,279	8%	3%	1%	0
3 in.	6x57	11	Hoist	Near Socket	58	2,702,279	3%	7%	2%	0
2 7/8 in.	6x39	6	Hoist	Drum End	41	3,598,016	9%	13%	1%	0
3 3/4 in.	6x41	6	Hoist	Near Socket	70	1,539,033	20%	3%	1%	0
3 in.	6x57	11	Hoist	Near Drum	58	--	5%	3%	1%	0
3 in.	6x57	11	Hoist	Over Point Sheave	58	--	16%	8%	3%	0
3 in.	6x57	11	Hoist	Over Point Sheave	58	--	19%	7%	3%	0
3 1/2 in.	6x57	11	Drag	Drum End	58	--	1%	1%	0%	0
3 1/4 in.	6x57	11	Drag	Over Fair Leads	58	--	24%	13%	3%	0
3 in.	6x41	3	Hoist	Near Socket	60	3,618,300	10%	6%	4%	0
3 in.	6x41	3	Hoist	Drum End	60	3,618,300	16%	12%	3%	0
3 in.	6x41	3	Drag	Over Fair Leads	60	3,618,300	12%	5%	2%	0
3 in.	6x41	3	Drag	Drum End	60	3,618,300	15%	11%	4%	0
3 1/4 in.	6x57	11	Drag	Near Socket	58	--	18%	8%	2%	3
3 1/4 in.	6x41	4	Hoist	Drum End	56	1,954,298	2%	1%	0%	0
3 in.	6x41	3	Hoist	Over Point Sheave	60	3,618,300	15%	7%	4%	0
3 in.	6x41	3	Drag	Near Socket	60	3,618,300	10%	8%	4%	0
3 1/4 in.	6x41	4	Hoist	Near Socket	56	1,954,298	8%	9%	3%	0
3 3/4 in.	6x41	6	Hoist	Drum End	70	1,539,033	25%	11%	1%	0
2 7/8 in.	6x39	6	Hoist	Near Socket	41	3,598,016	10%	4%	2%	0
3 1/4 in.	6x41	4	Hoist	Over Point Sheave	56	1,954,298	4%	10%	4%	0
3 1/4 in.	6x41	4	Drag	Drum End	56	1,954,298	3%	3%	1%	0

TABLE 7. (contd)

Rope Size	Classification	Mine No.	Sample	Location	Bucket Size, yd^3	Yards Removed	Wear: Outer	Wear: IWRC	Wear: Strand	Broken Wires
3 1/4 in.	6x41	4	Drag	Over Fair Leads	56	1,954,298	6%	8%	3%	0
3 3/4 in.	6x41	6	Hoist	Over Point Sheave	70	1,539,033	32%	9%	2%	3
3 1/4 in.	6x41	4	Drag	Drum End	56	1,954,298	6%	6%	2%	0
4 1/2 in.	6x64	11	Drag	Drum End	100	1,954,530	21%	6%	3%	1
4 1/2 in.	6x64	11	Drag	Over Fair Leads	100	1,954,530	31%	8%	2%	0
4 1/2 in.	6x64	11	Drag	Near Socket	100	1,954,530	13%	13%	4%	1
4 1/2 in.	6x64	11	Drag	Drum End	100	1,954,530	2%	2%	1%	0
4 1/2 in.	6x64	11	Drag	Near Socket	100	1,954,530	19%	17%	4%	0
4 1/2 in.	6x64	11	Drag	Drum End	100	1,954,530	3%	2%	1%	5
3 1/2 in.[a]	6x57	5	Drag	100 ft from Bucket	70	3,715,248	50%	Severe	Moderate	0
3 1/2 in.[a]	6x57	5	Hoist	200 ft from Bucket	70	4,456,991	30%	Severe	Moderate	8
3 1/2 in.[a]	6x57	5	Hoist	20 ft from Bucket	70	4,456,991	5%	Moderate	Moderate	0
3 1/2 in.[a]	6x57	5	Hoist	125 ft from Bucket	70	4,456,991	50%	Severe	Severe	3
3 1/2 in.[a]	6x57	5	Drag	10 ft from Bucket	70	3,715,248	20%	Moderate	Moderate	1
3 1/2 in.[a]	6x57	5	Drag	10 ft from Bucket	70	3,949,000	20%	Moderate	Moderate	0
3 1/2 in.[a]	6x49	5	Drag	100 ft from Bucket	70	3,949,000	20%	Moderate	Moderate	0
3 1/2 in.[a]	6x49	5	Hoist	20 ft from Bucket	70	3,620,000	0%	Minor	Minor	2
3 1/2 in.[a]	6x49	5	Hoist	200 ft from Bucket	70	3,620,000	5%	Minor	Minor	0
3 7/8 in.[a]	6x49	7	Drag	10 ft from Bucket	78	1,850,000	30%	Severe	Severe	2
3 7/8 in.[a]	6x49	7	Drag	100 ft from Bucket	78	1,850,000	--	Severe	Severe	3
3 7/8 in.[a]	6x49	7	Hoist	20 ft from Bucket	78	6,000,000	5%	Minor	Minor	0
3 7/8 in.[a]	6x49	7	Hoist	200 ft from Bucket	78	6,000,000	10%	Moderate	Moderate	0
3 7/8 in.[a]	6x49	7	Hoist	200 ft from Bucket	78	6,000,000	10%	Moderate	Moderate	0
3 1/2 in.[a]	6x57	5	Drag	10 ft from Bucket	70	3,214,000	20%	Moderate	Moderate	0
3 1/2 in.[a]	6x57	5	Drag	100 ft from Bucket	70	3,214,000	0%	Moderate	Moderate	1
3 1/2 in.[a]	6x57	5	Hoist	20 ft from Bucket	70	5,341,000	25%	Severe	Severe	0
3 1/2 in.[a]	6x57	5	Hoist	200 ft from Bucket	70	5,341,000	15%	Moderate	Moderate	0

(a) These examinations done by Dr. Sam Gambrell, University of Alabama, as a consultant to this program. Wear of individual wires was estimated, not calculated as before.

As with the hoist rope, wear near the drum is not significant and will not be discussed further.

In general, there is little evidence of fatigue in the field ropes examined. Wires break due to abrasion and notching. Because of the notch sensitivity of the wires, any cracks that form grow for only a few cycles before fracture is complete. In excess of 1750 wires were examined for cracks by magnetic particle detection and none were found. Methods to reduce the abrasion and the notching will lead to increased rope life.

TEST ROPE EXAMINATION

Examination of the test ropes was done so that damage could be compared with that found on field ropes. There also was an attempt to correlate damage with the test load.

All 23 of the test ropes were examined for damage. A total of 57 rope specimens from fatigue tested ropes were examined for cracks, broken wires and other visible damage.

Wear patterns on the test ropes were very similar to those found in the field ropes (see Figure 10). The only major difference was that the outer wear of the test ropes was not as great as that of the field ropes. The reason for this is twofold. The sheaves on the test machine are made of mild steel and this causes little wear of the rope. Also, test methods and machine design prevent any relative sliding between the rope and the sheave.

Wear of outer wires from contact with the sheave was from 5% to 15% of the original diameter, depending on the number of cycles. Wear at points of contact with the IWRC ranged from 4% to 16% as the number of cycles increased. Interstrand contact wear varied from 1% to 5% as loads decreased and cycles increased.

CRACKING OF WIRE

Magnetic particle detection was used to examine individual wires for cracks. Over 3000 wires were examined and only 7 cracks were discovered.

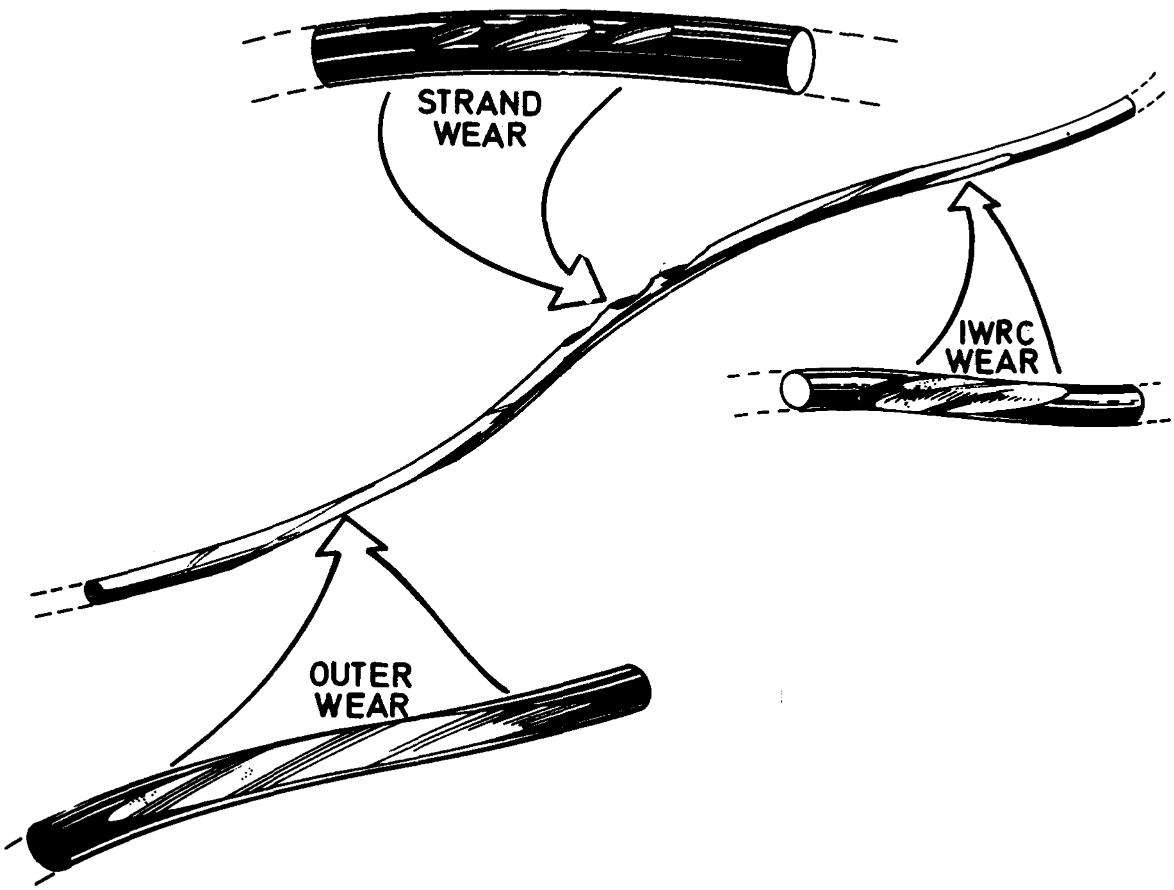


FIGURE 10. Wire Wear Patterns

A 6 x 37 class IWRC wire rope has over 250 individual wires of varying diameter, tensile strength, and carbon content. The wires are made from plain carbon steel in the range of AISI 1030 to 1080. The wires are severly cold-worked during drawing and have an ultimate tensile strength in the range of 220 to 280 ksi. In general, the higher strengths are associated with the greater carbon content; however, this may vary with different amounts of cold working. Usually core wires and filler wires are made from the lower carbon steel and therefore are softer and more ductile. The core and filler wires wear sacrificially with respect to the outer wires of the strand. Strand wires are made from the higher carbon steels and will fail in a ductile manner under normal conditions; however if a crack is present the wire fails in a brittle fashion. When a crack is present very little energy is required to propagate the crack through the wire.

DISCUSSION

Test results are discussed in the following sections.

FATIGUE TESTING

In most cases, there is total separation of at least two strands under a fatigue failure. Separation of four strands and the IWRC occurred in some test ropes. Initially, it appeared that this was only a function of the test load, but it has also been seen at the lower loads. Part of the reason for the dramatic failures at high load (four strands and the IWRC) is probably the placement of the overtravel safety circuit. If this is placed too far from the sheave bearings, load is maintained on the rope until this circuit is tripped.

Failure of ropes at a very low number of cycles under loads above 45% of the catalog UTS was not expected. As shown in Table 5, ropes would have been expected to fail between 10,000 to 13,000 cycles when compared via the Drucker-Tachau ratio with small-diameter rope data. At these higher loads, ropes lasted fewer than 1500 cycles. It appears that repeated cycling of a rope loaded at this high level is extremely detrimental. More information is needed about actual field loads; however, overloading of the rope due to digging conditions or bucket overloading could be a major cause of early failure of large-diameter wire rope on surface coal mining draglines.

WEAR OF WIRE ROPE

As shown in Figure 10 there are three types of wire wear. They are strand wear due to contact and sliding between adjacent strands, IWRC wear due to sliding and contact between the strands and the core, and outer wear caused by abrasion from sheaves, rocks and dirt. Externally visible damage[a] to wire rope is basically limited to broken wires and flattening of the outer

(a) There are three distinct types of wear found in wire rope that has been retired from service. This wear has been found in the field ropes and also the ropes from bend-over-sheave fatigue testing. Figure 10 schematically represents these three types of wear.

wires due to abrasion from digging and sheave overspin. Seldom are any wires other than those of the outer strand found to have visible damage when the rope is inspected before being dismantled. Because wear and breakage of outer strand wires are one of the operators' common gages of rope integrity, only wear of these wires will be discussed.

Strand wear results from interstrand contact. Wear of this type is transverse at $\sim 10°$ to the wire axis and comes from contact with wires in adjacent strands. This wear is usually the least extensive of the three types; however, the helical nature of the wire causes this wear to occur twice as often along the wire as the other two wear types. Breakage of the wire at this point does not occur as often as at the other points.

Wear between the rope and the sheave results in a flattening of the outer wires. Some wear here is also due to abrasion from digging but it appears to represent a small portion of the total wear. This outer wear is mostly caused by relative sliding between the rope and the sheave, or sheave overspin, resulting from the inertia of the sheave. For this reason, larger, more massive sheaves would be detrimental to rope life. The larger sheave would decrease the bending stresses in the rope somewhat, but the increased inertia of the sheave would lead to more overspin and increased wear. Broken wires resulting from this outer wear are usually called "crown" failures. The failures can also be caused by a sheave that is too hard. A balance must be maintained between rope and sheave wear. If the sheave is too soft, the rope will wear grooves in the sheave, necessitating its repair or replacement. These grooves can also be detrimental to rope life.

Wear between the IWRC and the strand is the third type of wear. The IWRC is of regular lay construction while the strands are lang lay. This produces a longitudinal wear pattern on the outer wires of the strand. The IWRC is made of a softer material than the strands. The reason for this difference in hardness is so that the IWRC is sacrificially worn away. In areas of high flexure and/or shock loadings, this results in severe damage to the IWRC. Because contact pressures and relative motion between the strands and the IWRC are high, wear at this point is higher than the wear between adjacent strands. This area is the main initiator of internally broken strand wires.

WIRE CRACKING AND FAILURES

The notch sensitivity of the wires may explain why so few cracks were found. Cracks when formed may last only a few cycles before fracture is complete. Also, because of the lower stress levels, one would expect to find more and deeper cracks at the lower loads. At high loads, more broken wires but fewer and shallower cracks would be expected. At low loads, the wires are able to adjust somewhat and reduce notching by adjacent wires. At high loads, the wires are locked in place. This locking-in produces more notching and thus more broken wires.

In the ropes tested at high loads, the majority of failures are ductile. The failure surfaces are cup and cone types, as shown in Figure 11, and there is some evidence of necking before failure. Brittle type fractures are in the majority for ropes tested at less than approximately 40% of the UTS of the rope.

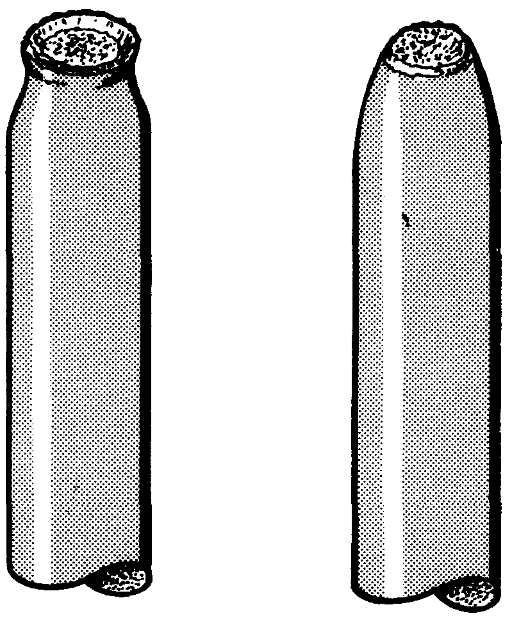


FIGURE 11. Cup and Cone Failure

More broken wires were found in the ropes tested at high loads. Those cracks that were detected were superficial. The depth of the cracks was never more than 10% of the wire cross section. Ropes tested at lower loads had fewer broken wires and deepter cracks (up to 60% of the wire cross section).

Scanning electron microscopy was conducted on some wires. Very little evidence of fatigue was found; none was found in the large-diameter wires making up the outer wires of a strand. One filler wire showed some evidence of fatigue; however, the filler wires have a lower UTS than do outer wires and are less notch-sensitive than the large-diameter wires and some fatigue striations would be expected. The lack of fatigue marks on the outer (large-diameter) wires of the strands tends to support the hypothesis that cracks are relatively short-lived.

In general, as with the field ropes, very little evidence of individual wire fatigue was found. Wires broke at points of IWRC or strand contact, with the former being more prevalent. There were no crown failures observed in the test ropes.

Most available small-diameter rope data are from ropes having a fiber core. Table 8 compares fiber core and IWRC ropes of the same size tested at the same Drucker-Tachau factor. Fiber core ropes had over 12 times the life of IWRC ropes at this test load (approximately 55% of actual breaking strength). Figure 12[2,3] shows some data on 1/2-in. and 3/4-in. IWRC ropes compared with data from 3-in. IWRC ropes. It should be noted that actual breaking strength is known for the small-diameter ropes and estimated for the large-diameter ropes. Also, the small rope is regular lay and the large rope is lang lay.

It is apparent from Figures 4 and 5 that data from small-diameter wire rope bend-over-sheave fatigue tests cannot be used to predict the life of large-diameter wire rope, under similar test conditions, using the Drucker-Tachau ratio. Figure 8 indicates that small- and large-diameter rope data do not correlate as a function of the rated rope strength. It should be noted that data available on small-diameter ropes tested in bend-over-sheave fatigue

TABLE 8. Cycles to Failure for 1/2-in. Fiber Core and IWRC Ropes[(a)]

Test No.	6x25 FW, IPS LL, FC	6x25 FW, IPS LL, IWRC
1	4,769	328
2	5,140	429
3	5,125	348
4	4,748	420
5	4,279	
6	4,728	
7	5,050	
8	4,300	
Average	4,767	376
UTS Wire	279,000 psi	276,000 psi
Load	13,750 lb	13,600 lb

$\beta = 0.0219$

(a) Data provided by Dr. Sam Gambrell, University of Alabama.

comes from ropes having a fiber core, not an IWRC, as in the case of the 3-in. diameter ropes that were tested. It is not known at this time exactly what effect this difference in core has on wire rope fatigue life, because the available data on small-diameter wire rope having an IWRC are limited. However, an educated guess can be made.

It is reasonable to expect that the differences in core materials will change the failure mechanisms in the rope. Wire breaks in a small-diameter rope before failure are generally a result of rope on sheave contact. Breaks in large ropes are predominantly at points of internal contact with the IWRC. Wear also occurs in the two ropes somewhat differently. In a fiber core rope, one type of wire wear is completely eliminated. The major portion of the wear on a fiber core rope is between the rope and the sheave; the Drucker-Tachau ratio takes this into account. It relates the bearing area of the rope on the

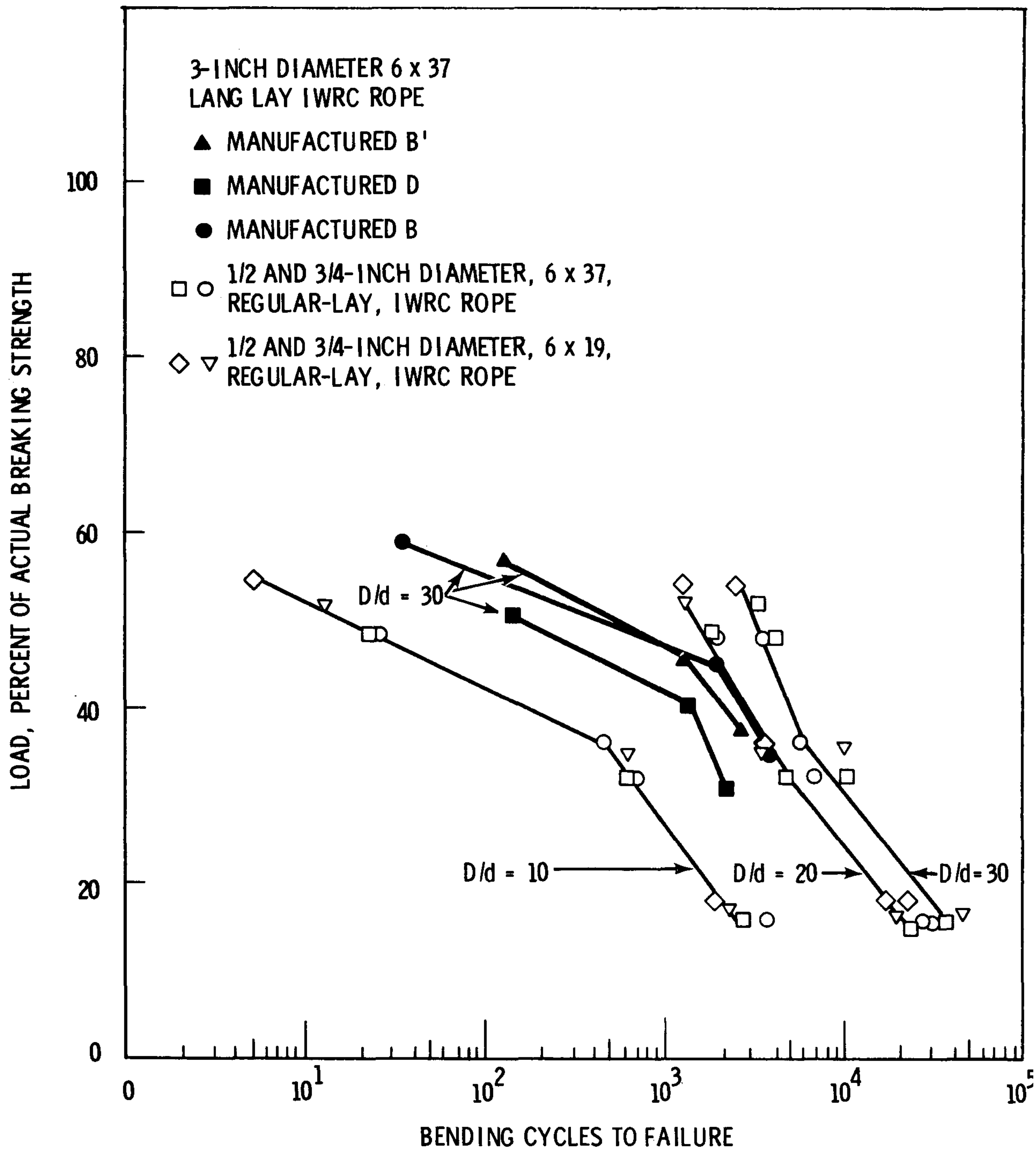


FIGURE 12. Comparison of Small- and Large-Diameter IWRC Rope Data(2,3)

sheave (Dxd) to the applied force (2T). This bearing pressure is then normalized by using the ultimate tensile strength of the wires, producing a dimensionless number. This can be used to compare various rope sizes. Results have been good for ropes from 1/2-in. diameter to 1 7/16-in. diameter.

When an IWRC rope is tested in bend-over-sheave fatigue, wear between the IWRC and the strands is as great or greater than wear between the strands and the sheave. The Drucker-Tachau ratio makes no allowances for this wear and should predict longer service life than what actually occurs. At loads above about 35% of the catalog strength of the large-diameter IWRC, rope service life is less than that predicted by the Drucker-Tachau ratio. At loads below this, the rope lasts longer than predicted. This is most probably due to the size difference in the outer wires of large- and small-diameter ropes. The ratio of outer wire diameter to rope circumference is smaller for large-diameter rope so large numbers of cycles are not as damaging to large-diameter rope as they are to small-diameter rope from the standpoint of wear to outer wires. Small-diameter ropes contain fewer wires than do large-diameter ropes. Therefore, a single wire break in a small rope reduces the load carrying capability by a larger amount than a single wire break in a large rope. For these reasons, large-diameter wire rope will have a greater service life at lower loads than small-diameter rope.

Another reason for the behavior difference in IWRC and fiber core ropes when tested in bend-over-sheave fatigue is lubrication. The fiber core is saturated with lubrication and is a source for internal lubrication. If this core dries out, the rope can degrade very quickly; however, it is easier to resaturate the fiber core than it is the metallic core. IWRC ropes are also lubricated during fabrication and this initial lubrication is usually lost fairly early in use. Attempts to replace this lubrication achieve only mild success. Usually oils that penetrate to the core are used to provide the necessary corrosion protection, however, high contact pressures between individual wires preclude the continued formation of full film lubrication. This allows metal-to-metal contact, resulting in an increase in wear. The extra metal-to-metal contact in IWRC ropes leads to their easy failure at high loads.

From the standpoint of flexibility, IWRC ropes exhibit one-half the elongation under load that fiber core ropes do. This reduced flexibility makes the IWRC rope more susceptible to damage at high loads. The wires are not as likely to move, which reduces these stresses. These higher stresses lead to early failure. Ropes having an IWRC are used almost exclusively in the surface coal mining industry. The metalic core provides about a 7% increase in ultimate tensile strength and also provides an increase in resistance to damage from dynamic loads.

Ropes from two different manufacturers were tested; however, ropes from manufacturer B, having two different wire strengths, were also tested. The rope from manufacturer D had the highest outer wire UTS (260 ksi). This rope had the lowest fatigue life at all test loads; however, it gave more warning of failure by exhibiting more broken wires. Rope from manufacturer B having the least outer wire UTS (230 ksi) had the best fatigue life at loads below 330,000 lb. Failures were not preceded by visible broken wires. The remaining rope from manufacturer B (identified as B') had an outer wire UTS of 240 ksi. It displayed some broken wires before failure and had a better fatigue life at loads above 340,000 lb than did the other rope from manufacturer B. The increased wire strength gave better high load performance, but the associated reduction in ductility decreased the low load performance.

In general, rope from manufacturer B gave less warning of impending failure, visible broken wires, but had better fatigue life than that from manufacturer D. Aside from the aforementioned differences in wire strengths, the rope from manufacturer B contained more wires. These extra wires increased the flexibility and probably played a major role in the increased fatigue life.

The testing methodology of Phase III yielded three ropes that saw two different loads. The results of these tests were presented in Table 3 and Figure 9. More data are needed, but it appears that running a rope at a load less than the final working load greatly increases the life. This is consistent with recommendations by rope manufacturers that ropes should be run before use.

Running in the ropes gives the wires a chance to adjust and place themselves in positions that minimize their stresses. If this is not done, the wires are not able to adjust and fail prematurely.

An increase in life was also exhibited by the two ropes that were initially cycled at a load higher than the final test load. This loading did not produce as dramatic an increase in life as the reverse loading. A possible explanation is that any existing cracks were arrested and did not lead to wire breakage. These data are incomplete and need to be investigated further.

KEY FINDINGS

As seen in Figure 4, the Drucker-Tachau factor is unsuitable for comparing bend-over-sheave fatigue behavior of large-diameter IWRC and small-diameter fiber core ropes. This is due to different wear mechanics because of dissimilar core material and large differences in outer wire diameters.

Large-diameter IWRC rope has an extremely short fatigue life at continuous loads above 45% of its rated UTS. This drastic reduction in fatigue life at the higher loads is of major importance. Overloading of wire rope is probably one of the major factors in early failure of surface coal mining dragline ropes. One rope that was run in at a lower load and then tested at the high load showed a dramatic increase in life. This is only one data point, but it correlates well with manufacturers' recommendations that rope be run in before use. At loads below 35% of rated UTS, large-diameter IWRC rope has better fatigue life than small-diameter fiber core rope.

Classical fatigue of individual wires in large-diameter ropes was not present. The notch-sensitivity of the material precluded sustained crack growth. Major wire damage resulted from external wear from dirt, rocks, and sheave, and internal contact with the IWRC and adjacent wires. The majority of wire breaks initiated at internal contact points except in the areas experiencing extreme wire wear due to rope on sheave contact.

Stall loads on draglines are generally set at around 30% of the catalog UTS of the rope to prevent damage to the rope and the dragline. Damage to the drag ropes near the socket indicated that the response of this safety system is too slow when compared to the dynamic load imparted to the rope. Sudden loads in the rope caused by hitting rocks, hard soil, inertial loads from dumping and swinging and operator inexperience are most probably much greater than the 30% stall load. A means of reducing the dynamic load or of obtaining better load control is needed.

A variation in rope bend-over-fatigue life was found between the ropes of the two manufacturers. The combination of different construction and wire tensile strength most certainly account for this. At this time, there are insufficient data to determine the exact effect of these differences.

CONCLUSIONS

Wire notching and rope on sheave abrasion were found to be the major failure mechanisms in large-diameter wire rope. Classical fatigue damage of individual wires was not present. In field rope, abrasive wear from rope or sheave is a major concern as is internal wire wear in those parts of the rope near the bucket. Test ropes exhibited the same internal wear as those parts of field ropes run over a sheave. External wear on test ropes was less than field ropes because relative sliding between the rope and sheave did not occur. In all cases, damage between the strand and IWRC was the largest contributor to internal wear.

Data from small-diameter fiber core ropes and large-diameter IWRC ropes did not correlate when compared via the Drucker-Tachau ratio. Large-diameter IWRC ropes failed at a much smaller number of cycles when tested under continuous loads above 45% of their catalog UTS. Data from the three different ropes tested were compatible.

RECOMMENDATIONS

Current criteria for rope retirement are still applicable. Rope wear, broken wires and changes in lay length or rope diameter are currently the best indicators of rope conditions. Some improvements in rope service life can be achieved through improved maintenance and operation procedures:

- Run in rope for several cycles before filling the bucket. Do not completely fill the bucket for the first few passes.
- Inspect the entire length of the drag rope once a day. Note and remove any broken wires.
- Use a lubricant that can pentrate to the core yet gives full film lubrication and good corrosion protection.
- Minimize wear between rope and sheave by maintaining sheaves in good order.
- Minimize rope overloading caused from hitting rocks, rope whip and bucket overfilling.
- Refrain from pulling the drag rope through the roll.

FUTURE RESEARCH

Actual loads on wire ropes used on draglines are not known. To make use of the fatigue data being generated, these loads must be known. Some type of load sensor should be developed toward this end.

More data on large- and small-diameter IWRC ropes are needed to correlate their behavior. Such a correlation would yield savings in time and money by rendering the results of small-diameter rope developmental work applicable to large-diameter ropes.

Additional information on wire wear is needed. This will facilitate development of better wire materials and increased rope performance.

Continued liaison between field personnel and rope researchers is needed so that existing problems can be identified and the resulting solutions can be instigated in the least amount of time.

REFERENCES

1. D. C. Drucker and H. Tachau, "A New Design Criterion for Wire Rope." Journal of Applied Mechanics. pp. A-33 - A-38, March 1945.

2. Gibson, P. T., F. G. White, L. A. Schalit, R. E. Thomas, R. W. Cole, and H. A. Cress, "A Study of Parameters that Influence Wire Rope Life," Final Report prepared by Battelle's Columbus Laboratories for the Naval Ship Systems Command under Contract No. N00024-72-C-5427 (October, 1974).

3. Gibson, P. T., C. H. Larsen, and H. A. Cress, "Determination of the Effect of Various Parameters on Wear and Fatigue of Wire Rope Used in Navy Rigging Systems," Final Report prepared by Battelle's Columbus Laboratories for the Naval Ship Research and Development Center under Contract No. N00600-70-C-1045, AD-776 993 (March, 1972).

APPENDIX A

THE USE OF LARGE-DIAMETER WIRE ROPE IN SURFACE-MINING MACHINERY[a]

(a) This chapter was written by R. L. Jentgen and R. C. Rice of Battelle Columbus Laboratories, Columbus, Ohio, May 1976.

FOREWORD

This report was prepared by Battelle's Columbus Laboratories, Columbus, Ohio, as a task under Bureau of Mines Contract No. H0252060. Compiled as a literature review and technical appraisal of large-diameter wire ropes in surface-mining machinery, the report presents some of the primary considerations in the construction, installation and uses of large-diameter wire ropes. As such, it provides a more complete discussion than is available elsewhere of some of the factors affecting wire-rope service life in surface-mining machinery.

Areas not discussed in this report, which should be considered at a later date, include nondestructive inspection of wire rope, the effects of vibration and rope dynamics on rope life and the analysis of internal stresses in ropes.

SUMMARY

This paper discusses some of the basic considerations of using wire rope in large draglines and shovels and presents a review of the related literature. The effect of specific system and operational factors on wire rope life is discussed, and recommendations are made based on available information and personal experience.

INTRODUCTION

Since about 1960, the size of surface-mining shovels and draglines has increased dramatically; the wire ropes required for these systems have concurrently increased in size at about the same rate.(1) Before 1960, 2-inch diameter ropes were considered large. Today, large draglines and shovels routinely use ropes 3 to 4 inches in diameter; one system, the 220-cubic yard, 4250-W, Bucyrus-Erie machine, uses four 5-inch diameter hoist and drag ropes simultaneously.

Currently, 5-inch diameter wire ropes are the largest used in surface-mining systems, but even larger ropes may be required in the future. Some ropes are already so large that no laboratory in the world has completely simulated field loads and conditions on full-size rope samples to determine the acceptability of one rope design versus another, or to identify the effects on rope life of a particular system design or operating procedure.

The economic importance of these ropes is significant, in terms of both new rope cost and downtime expense during replacement. For example, one of the largest draglines currently employed--the 150-cubic yard Marion 8950--uses four 4-inch diameter drag and hoist ropes, each of which is 480 and 760 feet long, respectively. In 1976 dollars, a 4-inch diameter rope sufficiently long to replace all ropes at once would cost more than $100,000. In addition, the effective cost of downtime for rope replacement would, in most cases, exceed $30,000. Considering that drag ropes survive an average of 10 weeks in this system, and hoist ropes about twice that long, it is easy to see that yearly expenses for wire rope for this machine are substantial.

All of these factors lead to the question--what can be done to extend the useful life of large-diameter wire ropes and minimize unscheduled downtime?

This report identifies some of the basic characteristics of large-diameter wire ropes, describes how those ropes are used in surface-mining machinery, and discusses some of the primary system design factors affecting rope life. A detailed list of information sources, categorized by specific subject, is included at the end of this report.

LARGE-DIAMETER WIRE ROPES

The methods and materials used in manufacturing a wire rope determine the adequacy of rope performance--regardless of rope size. The following paragraphs review the common constituents of a wire rope and the role of each in developing specific rope characteristics.

MATERIALS, MANUFACTURING AND DESIGN CONSIDERATIONS

Large-diameter (greater than 2-inch) wire ropes, are constructed of a large number of individual steel wires formed through a process that includes patenting, cleaning and drawing.[2] Patenting, a heating and quenching process, results in uniform wire or rod microstructure before drawing. Cleaning involves submerging the wire or rod coils in a dilute acid solution, rinsing them in water, neutralizing them in lime or phosphate and then baking them until dry. Drawing commonly involves the mechanical reduction in diameter (cold working) of wire or rod stock through from 6 to 8 consecutive dyes with a resultant 70 to 90 percent total reduction in cross-sectional area. The corresponding tensile strength of the cold-worked wire may increase to approximately 180 to 350 ksi, depending on the carbon content of the wire material.[3] Carbon contents between 0.40 and 0.90 percent are commonly used in rope wire, with the highest carbon percentages resulting in the highest wire strengths. With current technology, 0.90 percent approaches the practical upper limit in carbon content, because higher levels cause decreased wire ductility and increased wire breakage during drawing.[4]

Carbon steel wire strengths are normally categorized in the industry by specific trade names. Nearly all of these names are based on the various grades of "plow steel" used in the wire-rope industry for many years. In order of increasing wire strengths, they are as shown in Figure 1--mild plow, plow, improved plow, and extra-improved plow. The mild plow is relatively low in strength and is used infrequently. Plow steel is somewhat higher in strength, but it, too, is decreasing in popularity in favor of the higher-strength grades. Improved plow steel wire is used quite frequently because of its relatively high strength and good abrasion resistance. Extra-improved plow steel is even higher in strength and is reported to be superior in abrasion resistance. It is commonly used in severe loading conditions such as those characteristic of surface-mining excavation machinery.

The final diameter of a drawn steel wire is important in determining a wire's strength level for a particular grade. Smaller wire diameters can be drawn to higher tensile strengths while still maintaining adequate ductility. For this reason, standard strength levels for different rope wire grades vary

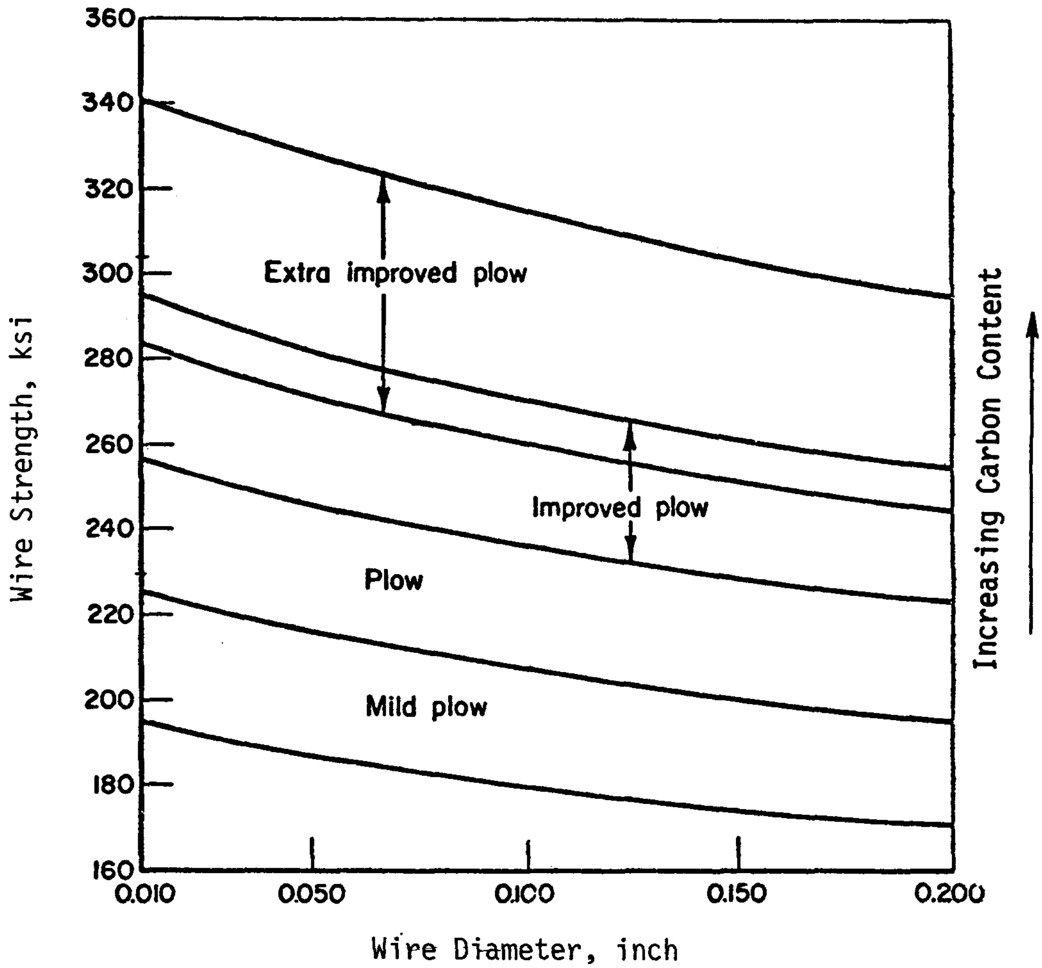


FIGURE 1. Carbon Steel Wire Breaking Strengths for Various Grades and Diameters

with wire size.[5] As Figure 1 illustrates, an 0.010-inch diameter improved plow steel wire should have an average tensile strength of 275 ksi, whereas that same grade wire in an 0.200-inch diameter has an average strength of 240 ksi.

The large-diameter wire ropes used in surface mining require large diameter, relatively high strength wires. This combination of size and strength is difficult to attain without very large and carefully controlled wire-drawing equipment.

Constructions

Wire ropes are manufactured in a large number of different constructions. Some estimates suggest over 90,000 different variations in wire-rope size and construction,[6] although less than 1 percent of them are used with great regularity.

The large-diameter wire ropes used in overburden removal machinery are almost always of 6-strand construction with an independent wire rope core (IWRC). The specific 6-strand construction used is based largely on the function and size of the rope. Ropes with 8 strands are made but they lack the flexibility needed for surface mining applications.

The typical stripping shovel includes hoist ropes, crowd and retract ropes, dump ropes for emptying the bucket, and boom suspension pendants. The hoist ropes and crowd and retract ropes are the most critical, large-diameter, running (moving) ropes within the shovel rigging system.

In comparison, the average large dragline involves hoist ropes, drag ropes, dump ropes, boom and mast suspension pendants, boom hoist ropes and intermediate suspension lines. In these systems, the hoist and drag ropes are the most critical, large-diameter running ropes.

Large hoist ropes for both shovels and draglines are normally made with from 37 to 61 wires per strand in a preformed lang-lay construction.[7-14] Typical constructions used are shown in Figure 2. The following are typical constructions:

- Seale
  A number of large wires are laid around an equal number of inner and smaller wires. This is done in such a manner that each outer wire lies in the interstice or valley of the two underlying wires. This is the best laid rope where flexibility is not improtant.
- Filler
  An even number of wires is laid around an inner layer of half that number, and each valley is filled with a small wire. The number of valleys is thus doubled and each outer wire lies in the valley formed by

one main wire and one of the filler wires. This rope is more flexible than Seale and has smaller outside wires and a less solid cross section.

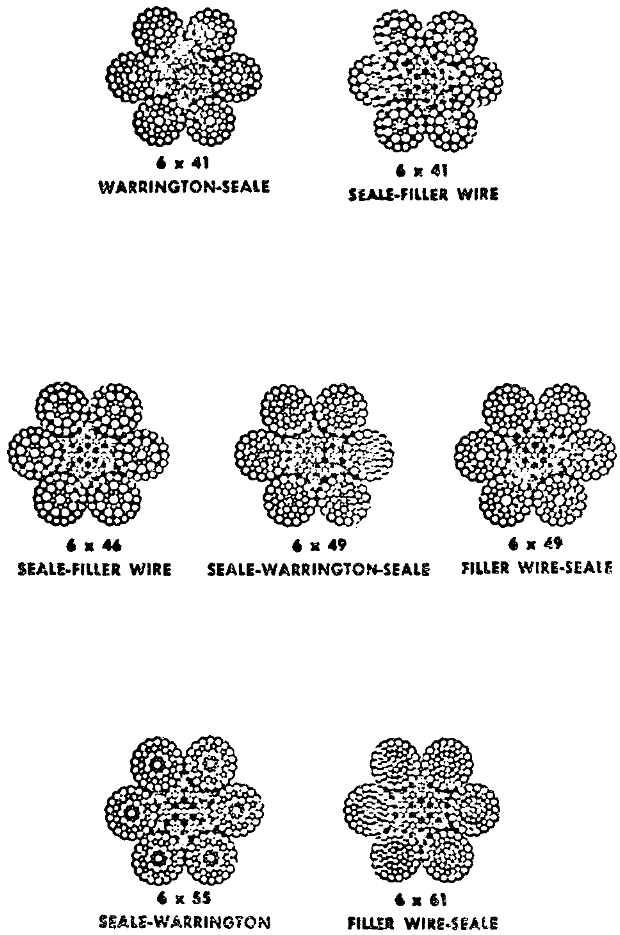


FIGURE 2. Large-Diameter Wire Rope Constructions Commonly Used in Surface Mining

- Warrington-Seale

  An outer layer of wires is laid over a second layer of pairs of wires (each comprising a larger and a smaller one). This, in turn, is laid over a third layer comprising a number of wires equal to the number of pairs in the second layer. Wires are positioned to seat in the valleys produced by aclaiming layers. This rope is flexible yet has a very solid cross section.

All rope constructions are based on the above variations. The number of wires and the size of the outer wires are the most common variations.

The ropes wires are normally improved plow steel grade (medium strength) or extra-improved plow steel grade (high strength). Current standards for wire strength and grade,[5] suggest that a 3-inch diameter 6 x 37 construction wire rope, having approximately 1-inch diameter strands and 0.15-inch diameter outside wires, should have outside wires with ultimate strengths from 230 to 265 ksi for improved plow steel grade, and from 255 to 305 ksi for extra-improved plow steel grade.

The drag ropes on large draglines are manufactured in some of the same constructions used for hoist ropes and with similar wire strengths. However, ropes with larger diameter outer wires which can best accommodate the severe abrasion seen with drag ropes, are used more frequently. For this reason, rope strands with as few as 25 wires per strand are sometimes used for drag ropes as large as 2-3/4 inches in diameter.[8]

The crowd and retract ropes on shovels are normally somewhat smaller in diameter than are the hoist ropes for the same system. The rope constructions used in this application are chosen for flexibility and maximum bending-fatigue resistance, because these ropes are commonly used for longer periods of time at lower loads and over somewhat smaller sheaves than are the hoist ropes.[15]

Core Materials

As mentioned earlier, the material for large-diameter wire rope cores is normally an independent wire rope core (IWRC) as opposed to a fiber or a wire strand. An IWRC is normally used because it offers a combination of strength and support for the outer strands. To enhance its ability to withstand the extreme radial pressures inside the rope, the wire materials of the rope core are sometimes made in lower-strength, higher-ductility grades. The lay lengths of the rope core are also adjusted to match those of the outer strands so that more uniform load distribution between the core and strands can occur. The IWRC is of regular lay construction to minimize wear between the core and the strands.

Independent wire rope cores increase breaking strength 7-1/2 percent when compared to that of fiber-core wire rope. This increased strength may be relatively short-lived, especially under conditions involving fluctuating axial loads, because the IWRC can break up quite significantly--well before the overall rope is sufficiently degraded to require replacement.[2]

Lubricants and Coatings

Lubricants and coatings are used to inhibit wire rope corrosion and wear.

Lubricants

Wire-rope lubricant serves to reduce internal friction so that the rope runs smoothly and to coat the wires so that corrosion is inhibited. In some instances it is also used to seal the rope to keep the dirt out. Ideally, the lubricant should penetrate the rope core easily and adhere strongly to the wire surfaces. To be most effective, the lubricant should maintain these characteristics for the full range of operating temperatures. In many applications, it is desirable that the lubricant be transparent to permit easy visual inspection for broken wires.

In most cases, lubricant selection is based on a compromise between desired properties. There are two basic kinds of lubricants for wire rope: (1) heavy compounds (asphaltic residua or petrolatums) and (2) light oils.

Most heavy lubricants provide adequate corrosion protection for moderate rope usage conditions, but internal penetration is almost always poor unless the lubricant is well heated during application. Heavy lubricants may also become ineffective at low temperatures and may cake and fall off as the rope is flexed during operation. They also tend to be messy and opaque. Thorough rope cleaning is necessary before inspections can be made, because broken wires may be obscured on a rope covered with a heavy lubricant. Heavy lubricants used on fiber core ropes may actually shorten rope life; their poor internal penetration can cause the core to dry out, increasing the likelihood of rope corrosion. Lubricant breakup or moisture absorption may also occur.

Light oil lubricants are thinner, clearer, more penetrating, and less prone to solidify. In addition, they are effective when applied in light coatings and are readily adaptable to automatic lubrication systems. The corrosion protection provided by light oils is sufficient for most operations and can be enhanced through the use of appropriate additives.

A list of specific references describing lubrication practices and equipment is included at the end of this report.

Coatings

Two types of coatings--metallic and organic--are available on wire rope to provide corrosion resistance. Zinc is the most widely used metallic coating, while polypropylene is the most commonly used organic coating. Coatings are used mostly on boom pendants.

Metallic Coatings. Coating wires with zinc, aluminum, tin or copper retards the onset of corrosion, so long as the coating remains on the wire. This is true because aluminum and zinc are more corrosion-resistant and anodic than steel. A rope wire made with a steel center and an aluminum or zinc coating will resist initial corrosive action. The steel wire center will remain essentially uncorroded even after corrosive action on the nonferrous surface metal has taken place because the corrosion process preferentially removes the zinc or aluminum before steel.

The galvanizing process coats steel with zinc and is the most common practice used to retard rusting in wire rope. Tests have indicated that the rate of corrosion of zinc under average atmospheric conditions is about 5 to 10 percent that of mild unprotected steel.

Organic Coatings. Plastic-coated wire rope has been available in small sizes for some time. Some plastic-coated and plastic-filled ropes have recently been introduced in larger sizes, but no published laboratory fatigue data are available for these sizes. An unbroken coating effectively prevents corrosion. However, if pierced, the protection is lost and the coating may actually serve as a retainer for the corrosive medium. The type of plastic used depends on specific requirements of each application.

Coating wires with low-friction organic films before the rope is manufactured has been suggested for replacing the conventional lubricant/corrosion-preventive liquids. Permanent, built-in lubrication and corrosion protection through the application of organic coatings holds great promise and is well within the scope of the technological development today. Such ropes would be very attractive for application to the surface-mining industry.

Terminations

As with rope constructions, the type of rope termination used depends to a great extent on the specific application of the rope within the shovel or dragline system. Almost all drag and hoist ropes are terminated with wedge sockets because they are among the easiest terminations to install in the field.

Some hoist ropes and almost all boom pendants are terminated with zinc sockets, although there is increasing interest in swage sockets and combination zinc-epoxy poured sockets. The primary concern with zinc sockets has not been strength, but axial fatigue resistance--since the standard zinc socket restricts interwire movement near the nose of the termination, which, in turn, causes stress concentrations that may lead to wire breakage.(15) Swage sockets, if properly applied, have been shown through laboratory experimentation(2) to provide axial fatigue resistance superior to that for zinc sockets. Combination zinc-epoxy sockets are also said to improve axial fatigue resistance.(16) A modified zinc socket with an elongated nose is also reported to reduce stress concentrations at the socket, thereby increasing axial fatigue resistance.(17) No comparative data are available on these "vibration-resistant" terminations, but their basic design concept--distributing the radial compressive stresses within the socket over a longer rope section to minimize stress concentrations at the socket nose--appears to be quite reasonable. It would be expected that a somewhat greater axial and vibration fatigue resistance than has been observed with standard zinc sockets would result from these terminations.

Mechanical Characteristics

Before discussing some of the specific factors that influence rope life in surface-mining machinery, some of the mechanical characteristics of these large-diameters ropes will be described.

First, they are extremely strong. Figure 3 shows the rated breaking strengths of 2- to 5-inch wire rope for several wire strength grades. As illustrated, a 3-1/4-inch diameter improved plow steel rope has a rated breaking strength of about 1 million pounds, while a 4-3/4-inch diameter rope of the same grade has a rated breaking strength of nearly 2 million pounds. The difference in rated strength between extra-improved and improved plow steel grades is also significant, but even the lower-strength grade has very high breaking strengths in the largest diameters.

Large-diameter wire ropes are also very heavy and present unique transport problems. For example, a 4-inch diameter wire rope weighs 29.6 pounds per foot, and a 5-inch diameter rope weighs 46.2 pounds per foot.

The modulus of elasticity of these ropes is relatively high. For example, 6 x 37 classification rope with an IWRC normally has a modulus of approximately 12 million pounds per square inch. Therefore, a 3-inch diameter, improved plow steel, 6 x 37 classification wire rope (with a metallic cross-sectional area of approximately 3.8 square inches) would be expected to elongate elastically approximately 5 inches in a 100-foot length when subjected to a 216,000-pound load (30 percent of rated breaking strength).

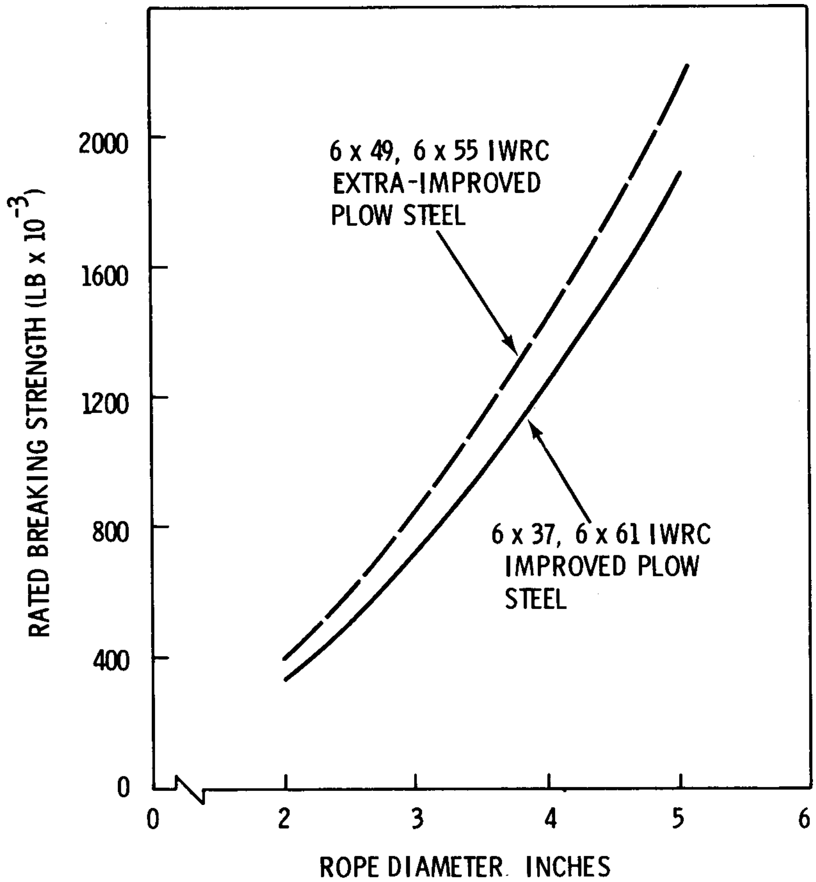


FIGURE 3. Wire Rope Rated Breaking Strength as a Function of Diameter and Wire Grade

PREUSAGE WIRE-ROPE CONSIDERATIONS

The previous section described some of the characteristics of wire ropes in general, with some specific references to the large-diameter wire ropes used in surface mining. The next three sections discuss the field use of these ropes and the effects of various factors on ultimate service life. First, the influences of storage and handling on rope performance are reviewed.

STORAGE

The available literature provides few details about storage of wire rope used in surface-mine applications. Usually, it is stored uncovered in the equipment yard and the reels are delivered in advance of need to the relative locations of the shovels and draglines. There, too, they are stored uncovered. Because of the recent (1975) shortages of wire rope caused by increased mining activity and decreased availability of steel products, the mining companies have made larger volume buys where possible; therefore, some of the reels are exposed to the elements for a longer than usual period. The authors have observed at a number of mine sites that rope stored for long periods (even when adequately lubricated during manufacturing) can become noticeably corroded. Even boom-pendant strand, which is normally galvanized, can become significantly corroded when stored outdoors for long periods. As a result, storage under cover is recommended where possible. Good storage practices are presented in Volume III of the Wire-Rope Handbook(18) and the storage care that should be accorded wire rope in general is discussed by Russell and Flowers.(19) The latter article recommends that large ropes which cannot be stored indoors should be raised off the ground or, preferably, placed on a wooden platform.

INSTALLATION AND HANDLING IN THE FIELD

A wire rope may quite possibly be damaged due to improper handling and installation procedures. Therefore, to protect the large investment and to yield maximum life, considerable care should be taken not to damage the rope when transporting it on reels, and during unreeling or installing on the machines. Russell and Flowers[19] have cautioned the mining industry about careful installation and handling. However, only Meger[20] specifically addresses the field handling of large wire ropes for surface mining machinery. In his article covering the fabrication and handling of the (then) world's largest wire rope, Meger discusses the care and techniques required to install and handle the 5-inch diameter wire rope used on American Electric Power Company's 220-cubic yard dragline. Meger stresses that, when working with large-diameter ropes, the machinery should provide the power. Taking short cuts can be disastrous before one has determined, based on analysis and experience, what short cuts can be taken that do not adversely influence rope performance and life. Rope changing time on every shovel and dragline currently in operation has been reduced to a practical minimum as a result of careful study, planning and the use of machinery capable of handling the heavy loads.

The investigators generally observed good handling practice in the field; however, some was less than good. At one mine, the rope was moved on the ground using a choker sling and crawler resulting in too tight a bend and subsequent kinking. Each wire rope company produces a handbook containing warnings about bad installation and handling practices as well as many good recommendations for care and handling to ensure that improper installation and handling will not attenuate the design life of the rope.

IN-SERVICE WIRE-ROPE CONSIDERATIONS

After a wire rope is installed in a surface-mining machine, it is subjected to a wide range of degrading factors that cannot be avoided or eliminated (at least with present system designs). How some of these factors influence rope life is discussed in this section.

CONTAMINATION AND DIRT

In surface-mining applications wire rope must operate in a dirty environment. Much of this dirt is soil which becomes airborne as the excavator uncovers the coal; this soil is deposited by gravity on the exposed areas of wire rope or machine elements where the rope contacts the sheaves or drums, etc. The dirt is held in contact to the rope by the lubricant or is rubbed into the rope by the contacting machine elements. Where relatively high-viscosity lubricants are used, the investigators have observed ropes that appear to be solid bars; the helical grooves of the rope are literally filled with a semisolid mixture of dirt and lubricant.

In addition to exposure to airborne dirt, service for certain rope uses assumes intimate contact with the overburden being excavated. In particular, the drag ropes on dragline excavators are frequently dragged through the lip or "roll" of the overburden. Even when conditions are such that the operator tries to produce little roll, and at some mines a crawler knocks the roll down after each pass, the length of rope nearest the bucket is still frequently dragged through the dirt and rocks.

The literature on surface-mining machinery wire rope discusses the influence of a dirty, abrasive environment on rope life. Sherwood(21) believes that rock damage is the greatest single cause of short rope life: "falling rocks and even small hard pebbles hit the rope hard enough to cut or nick individual wires or distort them out of their proper position." This initial damage may not be noticed at first, but it will set off the chain of events leading eventually to rope failure. Such damage can occur even when the initial trouble spot is a nicked wire rather than a complete cut, for this nick will be the first to yield under fatigue as the ropes run over sheaves and drums subjects it to a continual flexing action. Sherwood's beliefs are not consistent with the results of field rope examinations done during this study as regards wear of strand wires from external abrasion.

Earlier, Meger[22] cited the same problem. In his experience with both live and dead ropes on buckets, rock damage is by far the greatest cause of short rope life, if the operator grows the least bit careless. Falling rocks can hit the leading side hard enough to distort or rip wires out of position. Small hard pebbles dropped between the sheave and the rope can severely damage one or two strands and eventually cause rope removal.

Myers[23] and Reynolds[24] report that ropes in operation tend to pick up and become impregnated with grit and dust, etc. Relubrication under these conditions is difficult to achieve because the lubricant gets sopped up in the dirt-lubricant mixture already present on the rope. Therefore, any hard-particle contaminants remain in contact with or between the wires to cause abrasive wear or nick damage as the rope is used. Layton[25] indicates that when checking for abrasion damage caused by dirt ingestion, wear on the unseen wire must be considered. He recommends that frequent lubrication is the best way to allay corrosion and abrasion. It is difficult to mention the abrasive environment without offering some suggestions for negating or ameliorating its adverse effects. In an old article, Meals[26] tried to offer promising suggestions when he said, "Where the rope must function under excessive abrasive conditions, such as in sand and gravel quarries (and in surface mining), select a lubricant having high surface-protective ability, in addition to lubricating, adhering and penetrating qualities." If such a lubricant could be specified, it should be used for wire rope, regardless of the presence of dirt in the environment.

Layton[27] suggests inspection for grit penetrating between strands and wires. Abrasive wear is generally influenced by the size of the individual outer wires, the physical properties of the wire, and the distribution of the wearing surfaces. Layton presents rope "cutoffs" and/or reversal to handle severe and unavoidable abrasive wear.

Concurring with Sherwood[21] and Meger,[22] Barden and Klaus[28] state a key cause of shortened drag rope life is rock damage resulting from the operator pulling the rope through the bank. They indicate that "well located drag sheaves are one way of minimizing this effect." They also recommend cutoffs and reversals as a way to treat the unavoiable wear damage.

Reference 29 reports:

> Sand or other abrasive material embedded in the lubricant can cause rapid wear of both sheaves and wire rope. ....If severely contaminated is known or suspected, the rope should be cleaned as thoroughly as possible and fresh lubricant applied.

It is, of course, impractical to clean a hoist rope. Part of the secret in keeping a rope well lubricated resides in the physical properties of the lubricant. As pointed out in Reference 30, the lubricant "must not tend to cake, gum, or ball up, especially if contaminated with an excess of dust, dirt or metallic particles."

CORROSION

Corrosion from exposure to precipitation and (sometimes) acidic mine water is another detriment to rope life. As mentioned earlier, when the rope is stored outdoors corrosion can begin if the lubricant or coating ineffectively prevents rust of the rope wire materials. Then, even after use, the process can proceed until it influences or produces wire breakage. Corrosion can be insidious because it very often takes place unseen within the interstices of the rope. It can be "glossed over" by the lubricant and lubricant/dirt mixtures as it does damage to interior wires.

Several papers discuss the role of corrosion in reducing the metallic area and subsequently causing wire damage resulting in a wire rope's removal from service. Russell and Flowers(19) provide inspection recommendations. Kasten(31) describes a "rust-bound" condition in which a rope given a sudden jerk cannot absorb the shock along its length; instead, the brunt of the shock is confined to one spot, resulting in breakage of one or more strands and, occasionally, the entire rope.

Sherwood(21) indicates that regardless of service conditions, corrosion is promoted by inadequate lubrication. When the rope becomes dry, the wires can no longer move freely past each other. Then, as the rope is bent, the wires at the outside curvature (carrying the greatest stress) are no longer free to adjust to a position of minimum loading. These wires remain overloaded and will break. As rust builds up on or between the wires, it further hinders free movement within the rope, resulting in even earlier breakage.

The causes of rope failure were listed by Myers[(23)] and corrosion was indicted as being wire-rope's worst enemy. Internal corrosion is described as the most common and serious form of rope deterioration; it is unobservable and may result in serious damage to both life and property. On the other hand, external corrosion is easily detected during inspection and can be arrested. According to Myers, mining ropes operate under conditions ideal for corrosion: humidity, water, acids, dust, mud, etc. Despite measures to control these unfavorable conditions, ropes nevertheless become contaminated. These corrosive elements eventually find their way into the rope interstices and unseen destructive activity continues. It is far more difficult to check corrosion once it has started than it is to prevent it, and, when corrosion has occurred, lubricant effectiveness is greatly reduced.

Layton describes corrosion as a "deadly enemy."[(27)] Corroded wire rope has neither its original strength nor its original ability to resist bending fatigue. Further, its ability to withstand abrasion is reduced. Fine particles of corroded steel work themselves into the rope's internal structure and contribute to the abrasion.

Grimwood[(32)] discusses criteria for rope removal based on corrosion damage. He reports that marked reduction in rope diameter is often due to excessive abrasion or corrosion of the outside wires, but, in some instances, it is due to excessive corrosion of the inside wires, which is not indicated by a surface inspection of the rope. Therefore, internal deterioration is a serious and dangerous condition, and any significant reduction in the rope diameter should be investigated carefully and its cause determined. He concludes that, when corrosion is present, the remaining strength cannot be accurately calculated and there is no reasonable way to judge whether or not the rope is safe for further service. When corrosion is present, all of the known methods for estimating remaining strength of wire rope becomes useless.

ABRASION AND WEAR

Murphy[33] lists resistance to abrasive wear as the most important selection criterion for ropes used in construction equipment. He suggests that a coarse wire construction be considered for ropes exposed to abrasive wear, but offers little else (not even lubrication) in the way of practical suggestions to counteract abrasive wear. Murphy does, however, discuss uneven drum winding which causes adjacent wraps of rope to slide against each other. In shovels and draglines where grooved drums are used, the grooves are supposed to separate the rope, precluding intimate contact between adjacent wraps. Shovel drums with badly worn grooves can cause one wrap of rope to ride atop and pinch the adjacent wrap, resulting in severe wear of these wraps. This effect was observed by the authors during several mine visits.

Murphy mentioned one other wear phenomenon, that connected with the misalignment between sheaves, and between the hoist drum and the first fixed sheave with which the rope makes contact. The maximum fleet angle occurs at the extreme point of traverse of the rope on the drum. Experience shows that fleet angles should not exceed 2 degrees on grooved drums, when they do, severe wear occurs to the deflection sheave and the drum groove.

Russell and Flowers[19] report that drag ropes wear out from abrasion near the wedge socket at the bucket end due to abrasion from digging and hitting rocks. Field observations noted that wear by abrasion does take place as the terminal end of the drag rope is pulled through the roll. Therefore, rope longer than that required for attachment to the drum is stored on the drum, when abrasion ruins the terminal 5- to 6-feet of rope, this end is cut off, rope is payed off the drum and the terminal end is resocketed. In addition to obviating the threat of rope breakage due to loss of strength from abrasive wear at the terminal end, cuts or cutoffs also change the location of the load-induced stresses near the drum and at the sheaves. This reduces the possibility of wear and/or fatigue in those locations. After one or more cutoffs, the drag rope sometimes is reversed to accomplish the same purposes.

The fairlead (or deflection) sheave on a drag-rope system should be positioned with respect to the drum so that the fleet angle is divided by their location, thereby minimizing wear at the groove flanges. If the fleet angle is all one way, wear might be serious.

Drag ropes usually are placed on the drum in only one layer. This is good, as rope life may be reduced by 50 percent for multiple-layer wrappings.

Improperly sized sheave and drum grooves can also cause, or be caused by, wear.

A fluctuating load on the hoist rope of large shovels or draglines causes the rope to slap violently against other ropes or against the boom. Supporting sheaves and rollers wear seriously when exposed to this treatment and so does the rope. Rope life can be lengthened under fluctuating loads by using tire casings, wooden beams, or other such sacrificial wear surfaces against which the rope can slap and slide. In some machines, sister sheaves are used to handle rope passage down the boom and in the house from the drum to the fairlead sheave. Reports indicate good performance in many cases; however, in others, ropes have been damaged and broken in the areas of the sister sheaves themselves.

As indicated earlier, all large-diameter ropes on large surface-mining machines are lang-lay. When wires break in a lang-lay rope, the protruding ends should be broken or cut off with a pair of pliers at the line of contact between strands. If this isn't done, the broken wire will lie across adjacent wires; when the rope passes over a drum or a sheave, the protruding wires will plastically notch the adjacent wires, inducing damage and eventual breaking.

Sherwood[(21)] calls for a program of preventive maintenance that includes rectification of any equipment condition which causes excessive rope wear. Examples include rough spots on drums, worn sheave grooves into which part of the rope might be wedged or pinched, and unevenly worn sheave grooves that do not match the replacement rope. Sherwood also discusses the need to cut off broken wires immediately when they occur, as well as the importance of anticipating rope wear-caused problems in advance so that reversing ends or cutting off abraded portions near the terminations (which also changes the points of stress contact) will preclude breakage.

The importance of preventive maintenance was illustrated by a visit to one coal company having an excellent ongoing program including periodic sheave inspection. During the visit, a padlock sheave on a shovel was being replaced because of a cracked flange that opened up to a sharp-edged surface at the flange periphery that was obviously cutting the rope. Had this damaged sheave remained unnoticed, it could have caused shortened rope life (or possibly even breakage) over a number of rope changes. Such an inspection/maintenance program probably pays for itself many times over when savings in downtime are considered.

Meger(22) considers the influences of interaction between sheave/drum design and wire rope wear. Tool marks, cracks or pits on sheaves can "machine off" steel from wire rope much as a coarse file effects the same kind of material removal. Consequently, sheaves should be carefully inspected for rock damage and other surface irregularities. Tough alloy materials provide the best drum surfaces for contact with the rope. These, too, should be carefully machined, polished to size and spaced or pitched to give adequate clearance between wraps to avoid side scrubbing and wear damage to the rope wires.

Meger reports that end fittings are subject to wear; therefore, they should not be neglected or thought of as "nonwearing." The exterior surfaces of terminations used for drag ropes are especially subject to wear. Even the interior surfaces of some others can be worn. Therefore, reuse, without inspection, can create a condition that can easily lead to rope failure.

In addition to wear due to abrasion of the rope itself, Myers(23) cited worn sheave bearings as one of the principal causes of rope failure. When bearing wear occurs, the sheave can wobble and the rope will scrub the sheave flange.

During a mine visit, the Battelle investigators observed a shovel being operated with a bad sheave bearing. This bearing had produced a sheave motion, wearing the sheave groove so markedly that the rope fit was very sloppy. Undoubtedly, rope wear and reduced rope life due to crushing were concurrent with the sheave wear/bearing wear condition.

Reynolds(24) discusses two types of rope wear: (1) external, caused by contact with the rope's surroundings, e.g., rollers, sheaves, drums. and (2) internal, caused by the components of the rope which move relative to one another under load. Although he appears to make wire-rope lubrication an easy task ("More frequent application (of apparently any nondescript lubricant) may be simpler and more economical than ordering, storing, handling, and dispensing a special lubricant"), he does appreciate the rigor of some of the contact conditions that produce wear.

Layton(25) cautions that, when checking for wear, internal wear changes must be considered. By gaging the rope diameter at points suspected of greatest wear, changes in the condition of internal wires can be assessed to some extent.

Poor sheave alignment increases the probability that the rope will jump out of the sheave groove; wear on the rope and sheave groove will increase. Layton(27) discussed this situation, as well as the rope-wear problems created by unbalanced sheaves running at high speeds, worn sheave bearings, and intolerable fleet angles. Layton also mentioned several other wear problems:

- internal rope wear caused by grit penetration
- kinking
- improperly attached end fittings
- dragging rope over obstacles
- sheaves or drums with improperly fitting grooves or broken flanges.

As a precaution against wear, Barden and Klaus(28) recommend arranging sheaves so as to produce very small fleet angles. They also recommended that sheaves, particularly the bottom drag sheave, should be constructed from a manganese-containing alloy, because these alloys work-harden to very tough, abrasion-resistant surfaces, resulting in minimal wear for the sheave and the rope. Drum hardness also can be related to rope life. The authors state that a flame-hardened groove will add to rope life much the same as does the manganese alloy sheave.

Barden and Klaus argue that "the harder the sheave, the longer the rope and sheave hold their contour and remain serviceable." Caution should be used in applying this rule of thumb. Provided the hardness differences between the rope steel and the sheave steel are not too great, the rope and sheave can "remain serviceable." However, extreme hardness of the sheave relative to the wire materials of the rope can result in undesirable rope wear when inertial effects induced by the sheave mass and normal engagement of the rope with the flange cause sliding contact between the rope and sheave. Barden and Klaus cited evidence of such sliding contact in the same paper where they stated:

> Experienced operators can determine when to lubricate wire rope by sight and by sound. At night, it's easy. If there isn't enough lubricant on the outside of the cable, sparks fly from the sheaves as the rope passes over them. During the day, the operator recognizes a need for lubrication by the way the rope acts or sounds.

Operators can become quite sensitive to the sights and sounds produced when a rope cries out for lubrication. However, when sparks occur as a result of friction arising from sliding contacts external to the rope and/or the rope is being frictionally stressed so that sounds are emitted, <u>it is definitely too late</u> to avoid life-curtailing wear damage! The rope should not ask for lubrication. Rather, the rope should be provided preventive maintenance that will assure long life. Moreover, during field visits, few operators were found who appeared to be waiting to see sparks fly before directing the oiler to relubricate.

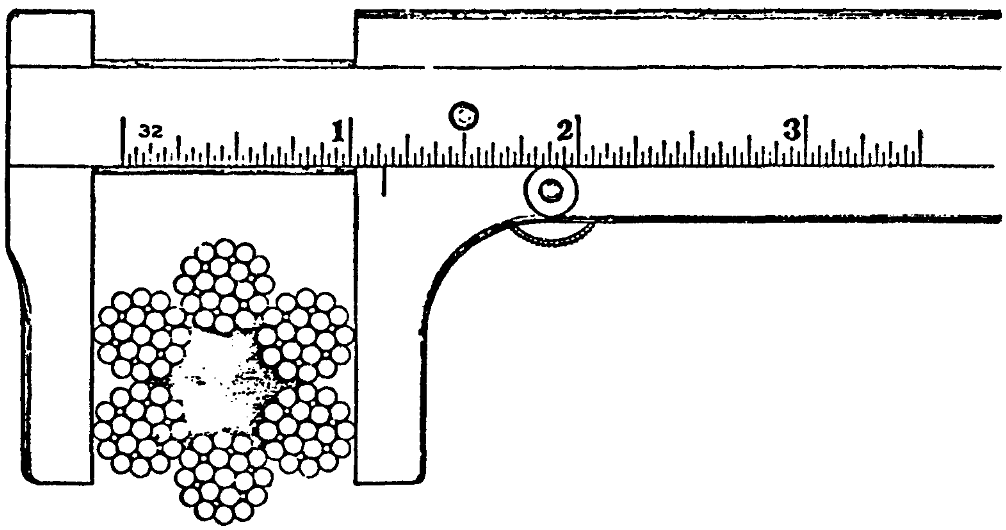
An anonymous article[29] in <u>Lubrication</u> points out that wire rope should be relubricated as often as necessary, and especially <u>before</u> corrosion and/or severe wear has an opportunity to damage or destroy the rope. The usual precautions against wear are presented. In addition, the article indicates that the combined effects of internal and external wear in a rope can be measured (as suggested earlier by Layton[25]), and the rate of wear can be assessed and recorded. The article then diagrams the caliper method of measuring rope diameter, shown in Figure 4.

The latest research in rope wear is described in a summary of the 1975 International Congress of Transportation by Rope.[34] Professor Rossetti and Engineer Dragone of Turin University presented a paper describing wear experiments resulting in a correlation between wear, reduction of rope diameter in the case of broken wires, and the remaining strength of the rope. If such a correlation has been made, then the use of a reliable technique for assessing the wear condition of a rope in service would permit the determination of remaining rope life.

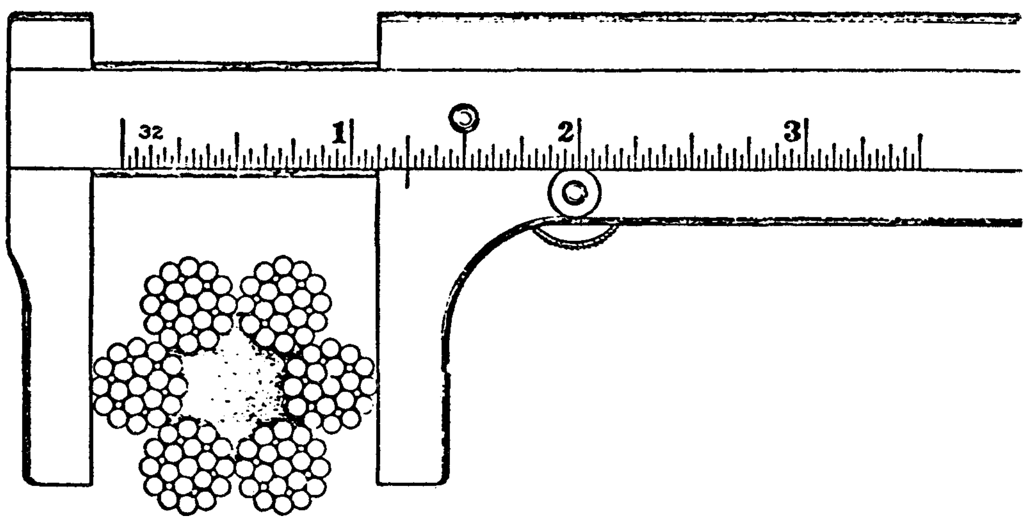
Rope wear problems possible when sheave- or drum-groove diameters are not optimum are described by Russell.[35] If the groove is too large, the rope will tend to flatten and the groove will wear rapidly. On the other hand, if the groove is too small, the rope will be pinched and distorted. As the rope bends around the curve in the sheave, the strands and wires slide upon themselves while adjusting to the curvature. When the rope is pinched in a too-tight groove, the strands and wires bend against each other, increasing abrasion, internal and external friction, and hindering rope readjustment to its load. This an undersized sheave groove causes the wires and strands to carry more than their share of the load.

In checking sheaves, drums, rollers, and idlers, it is important to assure that they are free running; however, some type of trade off is necessary so as to avoid excessive sheave overspin. If these machine elements stick due to improper adjustment, excessive wear, or poor lubrication, the rope will slide across the surface, producing a cutting or filing action on the element that is not turning as it should. This will eventually severely damage the rope.

Mine visits during this investigation revealed that about half the fairlead rollers on the draglines observed were stuck and had been immobile long enough to produce rope cuts due to wear that were definitely a source for rope damage. In one case, the rollers had been cut through completely!



INCORRECT WAY



CORRECT WAY

FIGURE 4. Measuring Rope Diameter

Wear problems resulting from improper sheave grooving, excessive radial bearing pressures on sheaves and drums, and bad fleet angles are discussed quantitatively by Anderson.[36]

Brown[37] describes a wire-development program aimed toward developing a more wear-resistant material for wire-ropes used in haulage. A high-temperature patenting process was used, producing high-tensile strength wires without the need for the usual series of cold work inputs that multiple drawing steps produce. Brown concludes that the kinds of wire properties required for long service lives in specific applications can be produced if the manufacturing technology is carefully controlled and pointed at the properties known to resist failure.

BENDING FATIGUE

In large shovels and draglines, the most often-replaced wire ropes are those which travel repetitively over sheaves and drums. These ropes are removed from service largely because repeated bending eventually causes wire fatigue, culminating in rope degradation and possible breakage. Of course, many other factors already discussed (e.g., corrosion and abrasion) cause rope degradation, these influence work, in combination with bending fatigue, to produce an eventual rope breakage or damage necessitating rope replacement. The relative severity of one condition versus another determines which damaging factor predominates and is responsible for the need to replace the rope.

With some large-diameter wire ropes used in surface-mining machinery, bending fatigue is normally the primary cause for rope removal. The hoist ropes on draglines and shovels and the crowd and retract ropes on shovels are good examples of ropes that are commonly replaced because of bending-fatigue damage.

Volume II of the Navy Wire Rope Handbook[38] recently completed by Battelle-Columbus describes the relative influence on the bending-fatigue life of wire rope of such system design factors as rope load, sheave-to-rope diameter ratio, sheave-groove contour and reverse bends. These topics are reviewed briefly in the next section of this report.

In addition to these system design factors, the rope construction and wire strength are determining factors in a rope's resistance to bending fatigue.

The effects on the bending-fatigue life of wire rope with variations in wire strength are shown in Figures 5 and 6 for one rope and sheave size. These figures show trends of data presented in a German wire rope publication.[39]

According to Figure 5, the bending-fatigue life for small-diameter rope will not change appreciably, even when the strength of wires in a rope is varied over a substantial range (at least from 200,000 to 300,000 psi), with all other factors such as load and D/d (sheave-to-rope diameter ratio) held

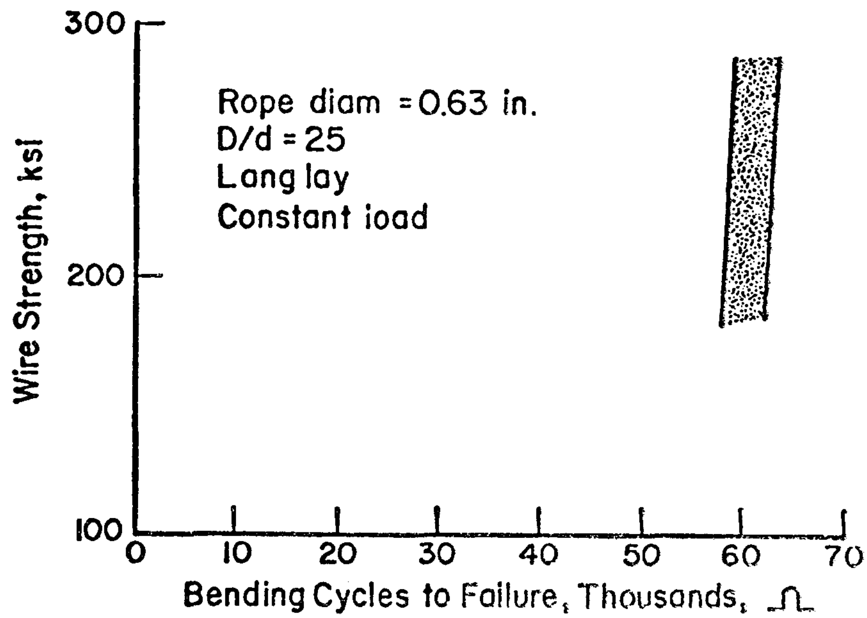


FIGURE 5. Effect of Wire Strength on Bending Cycles to Failure--All Tests Completed at Equal Rope Tensions

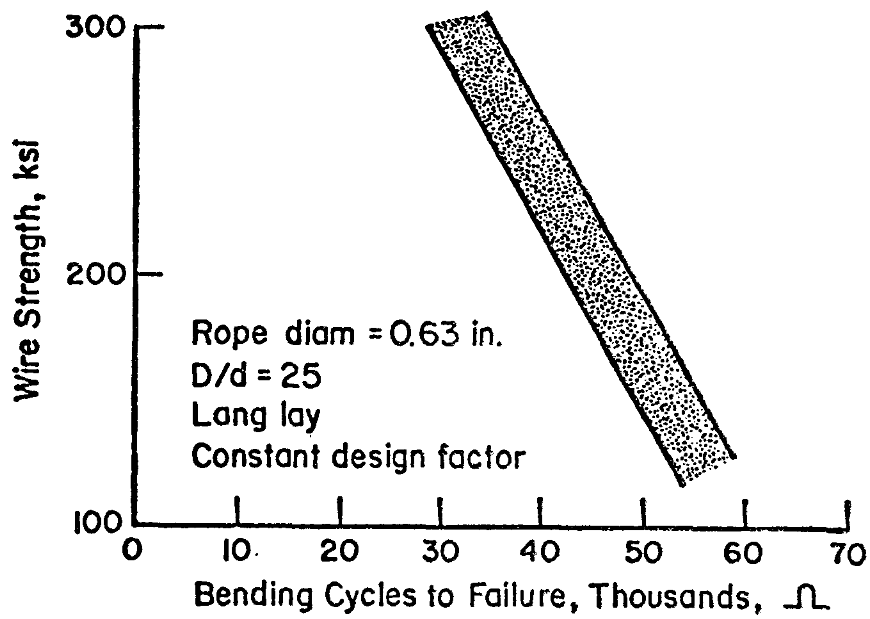


FIGURE 6. Effect of Wire Strength on Bending Cycles to Failure--All Tests Completed at Equal Design Factors

constant. On the other hand, if the design factor is held constant, as in Figure 6, the bending-fatigue life will decrease as the wire strength and rope load are increased. Work by other investigators on rope manufactured in the U.S. has confirmed this trend, at least qualitatively, for higher-strength wires.[40,41]

Rope construction is as important as is wire strength in determining bending-fatigue resistance. Normally, a number of factors must be considered when evaluating a rope's construction--the lay direction, number of strands, arrangement and size of wires, core type, and whether or not the strands are preformed. When speaking of large-diameter surface-mining ropes, however, most of these factors have been relatively standardized as a result of extensive field experience. [Encouragingly, most of the "standard" construction options on large-diameter ropes have been supported through laboratory evaluations as being the most reasonable designs for good bending-fatigue performance.] For example, some German experiments[39] have shown that lang-lay rope (the type used almost exclusively in most large draglines and shovels) is superior to regluar-lay rope in resistance to bending fatigue, as long as sheave-groove conditions are good. It has also been shown by several investigators[42,43] that strand preforming provides superior fatigue resistance and that an IWRC should be used under conditions involving high loads,[41] (although the opposite appears to be true under moderate loading conditions).

The evidence for the superiority of 6-strand ropes is less conclusive; in fact, some experiments with 6-, 8-, and 9-strand ropes have shown that several 6-strand rope constructions have poorer resistance to bending fatigue than have similar 8- and 9-strand ropes.[39] These results were developed using fiber-core ropes, however, it is likely that 8- and 9-strand fiber-core ropes would be totally unsatisfactory under the severe loading and impact conditions imposed on surface-mining machinery.

SYSTEM DESIGN AND OPERATION CONSIDERATIONS

A number of design- and operation-related factors, if properly considered, will tend to considerably improve the life of wire ropes used in surface-mining machinery. This section discusses the various viewpoints presented in the literature concerning the effects of wire rope break-in, sheave design and rope loads, and lubrication practice.

BREAK-IN

A number of published papers recommend a break-in procedure to assure long service life of newly installed wire ropes. Dull[6] has indicated that loads of 200,000 lb are applied to the 5-in. diameter wire ropes used in the largest machines for surface-mining excavation, whereas the 3/4-in. hoist rope of the smaller machines operate under only 7,500 lb. These data indicate that break-in for the 5-in. rope becomes comparatively very important; certainly, the full load should not be applied on any rope without allowing the rope to accommodate.

Russell and Flowers[19] state that at break-in (or any time later), shock loads that overstress the rope should be avoided. If a load is suddenly applied to slack rope, the shock may exceed the rope's elastic limit. This may not result in an immediate failure, but the rope may fail later when a different operator is on the machine. Shock loads can be avoided by making sure there is no slack and no jerking of the rope. Power should be applied smoothly and steadily. Each machine operator should be instructed in the importance of avoiding shock loads and the dangers of mistreating rope.

All wires and strands in a wire rope must work together smoothly. A short break-in period under a light load and at a slow speed permits the rope components to adjust and adapt themselves to the job. The rope becomes accustomed to the various bends and, in effect, fits itself to the reeving. End attachments can be tested and adjusted if necessary before maximum loads are imposed. The actual time consumed in breaking-in a rope is short, compared to the time required to install a rope. And it can pay off in longer rope life.

Of a number of sources reviewed,[11,21,22,25,44] all mention the importance of break-in at low loads and slow speeds.

Proper break-in procedures should also consider care in the initial spooling on the drums, mentioned in detail by several authors.[35,44,45] Poor spooling can crush the rope; regardless of the effectiveness of lubrication, the rope might be damaged during spooling. The operator must know how to spool and break-in the rope without crushing, distorting, or kinking--and he must know how great a load he can put on the rope. Putting a new rope through a break-in working cycle is said to actually strengthen it by setting the strands more firmly in place. In the normal action that takes place when a wire rope is bent in tension over a drum (or sheaves), the outside circumference of the rope tends to stretch while the inner surface is compressed. This action results in a continuous shifting of the wires and strands as the rope moves over the drum (or sheaves), producing internal friction. Rope preforming reduces these effects; however, even with preformed rope, considerable care should be taken in break-in.

Kasten[45] mentions the last break-in phenomenon that might occur. If the rope is "rust bound" when installed, it can experience a precipitous release when "jerked" or not properly broken in. Consequently, the brunt of the shock is confined to one spot instead of throughout its length, resulting in one or more strands breaking. Mine visits and observations noted ropes stored outside for long periods were obviously corroded on their exterior surfaces. Should they have been corroded on the interior as well, they may very well have been rust bound.

SHEAVE DESIGN AND ROPE LOADS

Bending fatigue is a crucial factor in ultimate rope performance. The design and arrangement of the sheaves within a surface-mining machine directly influence bending fatigue. In fact, the size and condition of sheaves within a system probably influence rope life more than the rope and core construction itself, so long as some other factor such as corrosion does not predominate to cause rope degradation.

If the sheaves are too small relative to the rope diameter, less-than-optimum fatigue lives can be expected, even under relatively low loads. This trend is evident in the experimental results for small-diameter rope displayed in Figure 7.[40,41] These data illustrate that, depending on the load level, a sheave-to-rope diameter ratio (D/d) as low as 10 can result in fatigue lives from 15 to 100 times shorter than those obtained for systems in which the D/d ratio is 30. Even a D/d ratio of 20 may cause a reduction by a factor of 2 or 3 in fatigue life for most load levels. Although impractical from a design standpoint in some systems, an increase in sheave diameter resulting in a D/d ratio of greater than 30 would likely result in even longer bending-fatigue lives. However, increased sheave overspin from the increased inertia of the sheave would cause more external wear and would reduce overall rope life.

Obviously, the service life of a wire rope under normal operating conditions also depends strongly on the load that it carries. Figure 8 provides a dramatic illustration[46] of this for small-diameter rope. The curve shapes are said to be typical of those expected for all types of wire rope. Figure 8

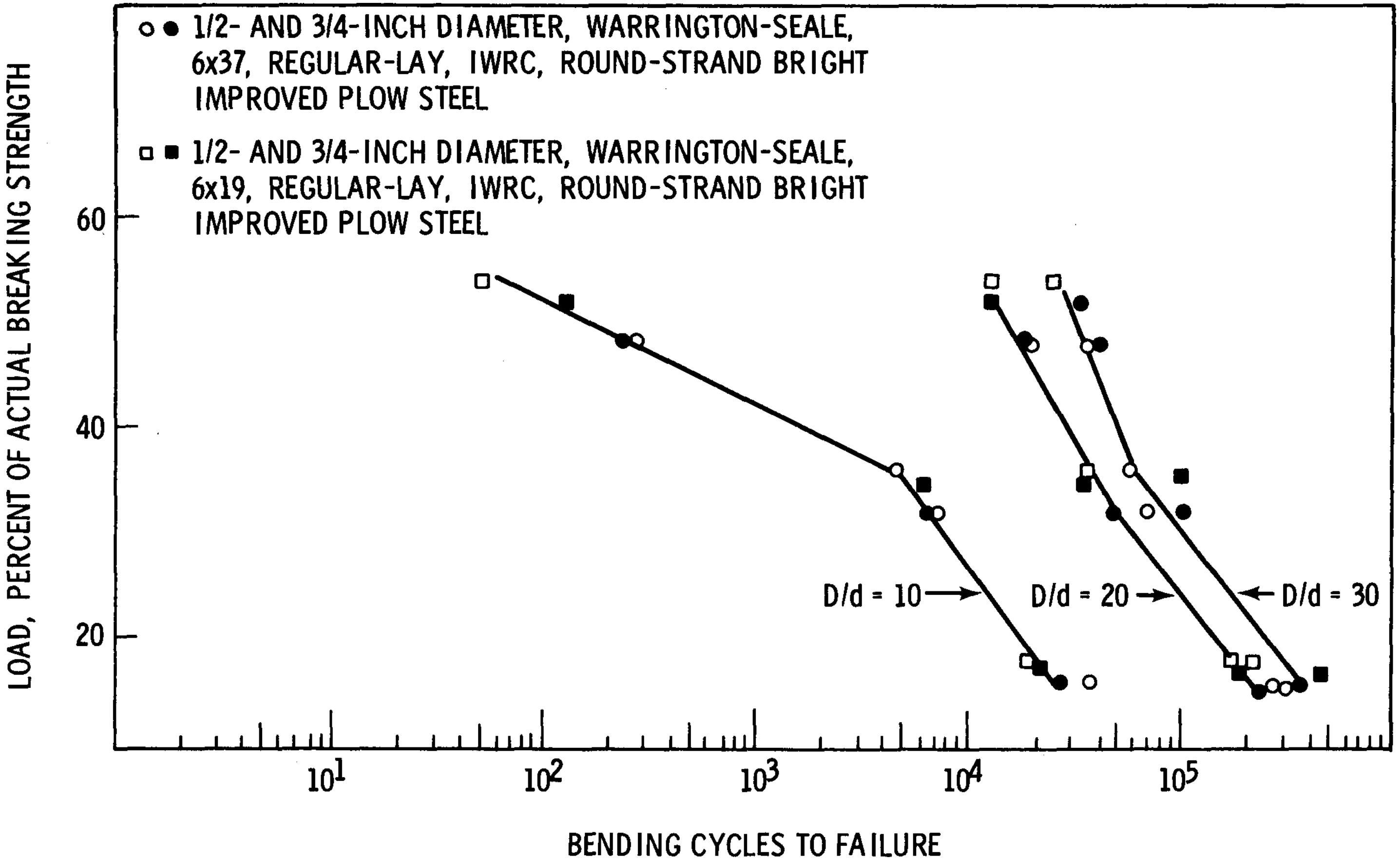


FIGURE 7. Fatigue Data for Small-Diameter IWRC Rope

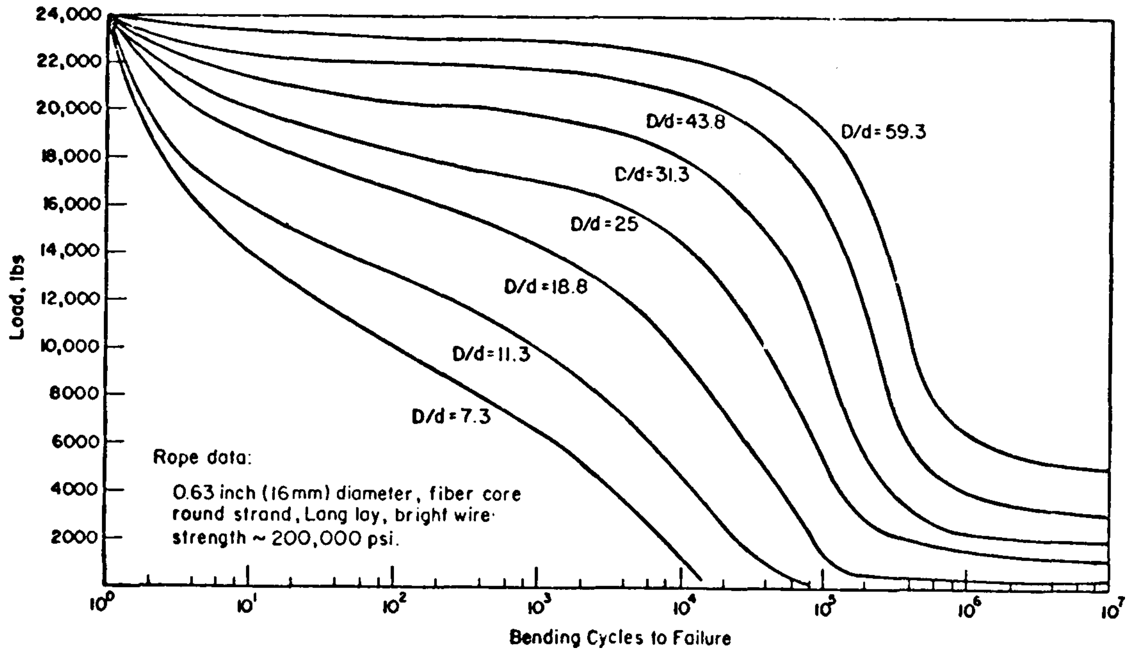


FIGURE 8. Bending-Fatigue Data for a Wide Range of Loads and D/d Ratios

illustrates another important point. While rope life is nearly always inversely proportional to load, the amount of variation depends greatly on the load range and D/d ratio. At load levels near the breaking strength of the rope, the fatigue life decreases rapidly with only small increases in load. The load level at which the drastic reduction in life occurs depends greatly on the D/d ratio, with the lowest D/d-ratio conditions showing the most sensitivity to load increases.

Abnormal sheave-groove geometries and certain sheave arrangements can also reduce fatigue life. Some reductions in fatigue life may result if the sheaves are arranged within a system so that the rope must pass through a reverse bend, i.e., where the rope must bend in one direction and then bend again in the opposite direction during one pass of the rope through the rigging. This effect is shown in Figure 9 for a 6 x 19 lang-lay rope[46] of 0.63-inch diameter. Obviously, for the conditions tested, reverse bends were very detrimental to rope life, especially for low loads (or high design

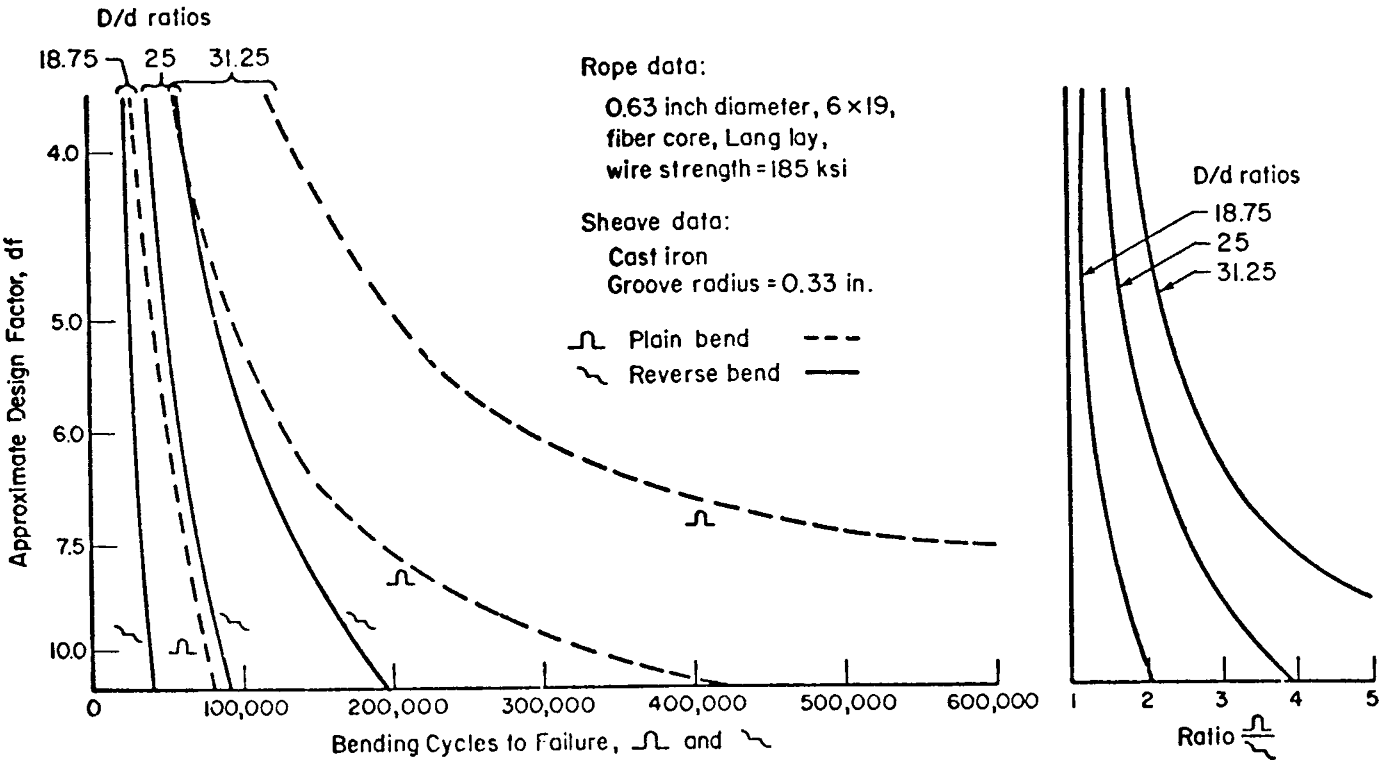


FIGURE 9. Effect of Plain and Reverse Bending on the Fatigue Life of a Lang-Lay Wire Rope

factors). Although no data are available for verification, it has been said that long distances between reverse bends alleviate their adverse effect on rope life because of the natural tendency of a wire rope in a field system to rotate slightly as it moves through the system. Some dragline manufacturers use reversed bending to reduce sheave overspin. Current practice is to allow at least two lay lengths between bends. It is felt that this distance eliminates the reversed bending effects. Specific data are not available on the effects of this system on rope life.

Another factor that is potentially degrading to rope life and is often overlooked, is the use of rollers and deflection sheaves. Because the wrap angle of the rope around these devices is often relatively small, they are assumed to be insignificant in terms of their contribution to bending fatigue of the rope. Some experimental results indicate that this is not true; in fact, sheaves involving rope deflections as small as 10 to 15 degrees may cause as much or more damage to the rope than those sheaves that bend the rope through 90 to 180 degrees.[41,46]

During mine visits, it was reported that the sister-sheave rollers used to support the drag rope as it comes and goes to the drum were a source of rope damage and one drag rope actually broke in that area.

Evidently, some controversy also exists regarding the effects of sheave hardness on the bending-fatigue performance of a rope. Several investigators have found almost no effect of hardness on rope life[47,48] as long as the sheave groove was sufficiently hard to prevent quick deterioration and rope pinching. Some other investigations have revealed, however, that nylon sheave-groove liners may serve to increase rope life substantially.[46]

LUBRICATION PRACTICES AND EQUIPMENT

Lubricants and lubrication practices are covered in this section. The relationship of lubrication with modes of rope failure and with rope life will also be discussed, insofar as information is available. Lubricants and lubrication are known to definitely influence the performance and life of wire rope; however, few real data substantiate the claims of the beneficial effects of lubrication for wire-rope applications in general, much less for mining applications. Some data demonstrate the relationship between lubrication and rope life for ropes used in underground mines,[a] but, thus far, experimental data are nonexistent for wire ropes used in surface-mining machinery. Nevertheless, the results-oriented and intuitive claims of the literature will be summarized and some of the investigators' personal observations during field visits will be presented.

In establishing any guidelines or procedures for removal of wire rope from service, the degradation processes such as corrosion, fatigue (and the results of their interaction), wear and abrasion, and the other failure modes (e.g., wire and strand breakage) are all considerably influenced by lubrication. Removal based on the calculation of remaining-strength criteria

(a) Just beginning is a program sponsored by the U.S. Bureau of Mines that will determine the state-of-the-art in lubricants and lubrication practices for mine-hoist ropes.

appears to be the best way to handle the interactive influences mentioned above. Grimwood[32] has presented the basis for such a retirement approach. He reports some data for the effects of lubrication and the absolute requirement for the lubricant to prevent corrosion so that remaining life can be estimated. First, Grimwood indicates that fatigue, abrasion, and corrosion are the prime causes of rope failure. Fatigue, which for running ropes is caused mainly by bending, cannot be eliminated, but it can be minimized by proper lubrication and by design considerations. Abrasion, too, can be controlled by effective lubrication, provided the sheaves and drum are in good condition. And, if the lubricant is appropriately formulated, corrosion can be prevented. Therefore, it must be concluded that lubrication is of prime importance in the maintenance of wire rope; further, the corrosion-preventive capability of a lubricant is the key function in allowing the prediction of remaining strength by calculation.

Grimwood's data are for a 9/16-inch diameter 6 x 19 rope (probably an excavating crane rope with an IWRC), part of which was laid up lubricated and part of which remained unlubricated. Samples of each of these sections were cycled (load unspecified) over sheaves of two different diameters until the same number of broken wires occurred for each specimen. Table 1 compares the number of bends required to produce an equivalent number of broken wires.

TABLE 1. Bends Required to Produce An Equivalent Number of Broken Wires

	Bends over a 10-inch sheave, x 10^3	Bends over a 24-inch sheave, x 10^3
Unlubricated rope	16	74
Lubricated rope	39	386

These data confirm that the sheave-size effect (about which a great deal is already known) is enormous. In addition, it is obvious that lubrication has an effect dramatically significant in its own right. Therefore, lubrication deserves considerably more attention than it has received in the past. After sheave-design factors have been considered, rope lubrication and maintenance is probably the next most important factor in rope life.

Unfortunately, in the almost 20 years since Grimwood's paper was printed (1968), little more actual data indicating the beneficial effects of lubrication have been published. The VDI Guidelines,(39) published in 1968, for lifting and hauling ropes contain the curves shown in Figure 10. It is obvious from these curves that lubrication enhances the life of the wire rope under the conditions studied. However, the rope is relatively small and it is suspected that it is a fiber-core rope. Therefore, there still exists no data whatsoever (published or unpublished) for large-diameter IWRC ropes typical of those used in surface mining. In 1955, Lex(49) had published a study correlating the retention of lubricant in fiber-core ropes with life in terms of a relative lack of broken wires. Lex's study was somewhat confusing; however, he did show conclusively that fiber-core ropes of relatively small diameter (5/8-inch) under a 5,000 pound load and drip lubrication during cycling (over a 24-inch diameter sheave) yielded longer fatigue lives than did ropes operated containing only the lubricant originally included in the core and strand during manufacture. Such a study should be made for large diameter ropes containing IWRC (rather than fiber cores), to provide maintenance information to the surface-mining industry.

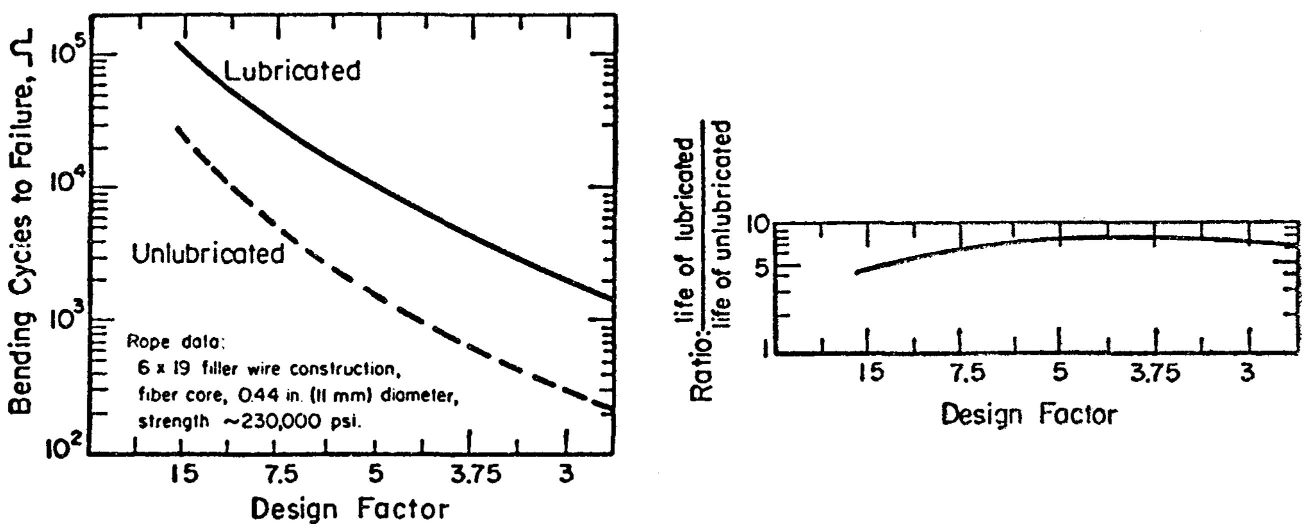


FIGURE 10. Effect of Lubrication on the Bending-Fatigue Life of a Rope for Various Design Factors

The authors described above are the only ones who have presented actual data for the effects of lubrication on rope life for hoisting and haulage applications. However, many publications cite the need for good lubrication practices in service. The arguments in favor of frequent lubrication stem mostly from a hard-to-analyze, but very important, function of the lubricant--that of corrosion prevention. One of the best papers describing the various corrosion processes in wire rope was authored by LaQue.[50] He presents some measures to protect the rope from corrosion. However, they include lubrication in only a general sense and are not very instructive.

Others have likened a wire rope to a machine that has relatively few moving parts operating under loads producing high contact stresses between metallic materials of similar metallurgy--a combination of conditions for producing high levels of friction, plastic deformation resulting in galling and seizure, and wear resulting from material transfer due to adhesion and from material loss due to abrasion. The addition of an effective lubricant to such sliding contacts in relative motion should: (1) increase load support by spreading the effective area of contact when squeeze-film lubrication effects prevail, (2) lower friction and adhesive wear by providing boundary lubrication at asperity contacts, and (3) lessen abrasive wear by chasing dirt and metallic debris out of the areas of high contact stress.

Russell and Flowers[19] laud "proper and systematic" lubrication as one of the important opportunities for increasing the life of wire ropes used in surface mining.

> Proper lubrication is nothing more than applying a light oil--a little at a time--and frequently. There is often a tendency to put too much lubricant on the rope in the belief that the interval between applications can be extended. Too much oil may be a detriment.
>
> It sometimes appears unprofitable to lubricate a drag rope, but it can be done. The key to successful drag-rope lubrication is application of a small quantity of light oil applied continuously or at frequent intervals, and well distributed, the purpose being to get penetration before it gets wiped off.

The Battelle investigators concur with Russell and Flowers, but offer several additional comments:

1. Proper lubrication <u>is</u> more than applying a light oil a little at a time and frequently. The light oil must be formulated to produce effective boundary lubrication and corrosion inhibition; therefore, not just any light oil will suffice.
2. "A little at a time--and frequently" is better than "a lot at a time and infrequently." However, "a lot" is too much only when it causes other problems, e.g., too high a cost, volumes that creep or run by mass transport to an area or a material that should remain unlubricated or is chemically incompatible with the lubricant, or an excess that attracts dirt or becomes a housekeeping problem.
3. It can be profitable to lubricate drag ropes except where the sheave/drum design or rope quality is so far from optimal that these larger factors negate the beneficial influence of effective lubrication.

Although more than 25 years have passed since Russell and Flowers encouraged drag rope lubrication, a recent field survey revealed that only about half of the drag ropes are being lubricated at all. And even some of the hoist ropes on draglines are lubricated only infrequently.

Meger describes lubrication as a "money saver and safety insurance policy."(22) Using the wire rope "machine" analogy, he laments (in 1960) that, despite all instructions and warnings, many ropes are installed, used and discarded without applying any additional lubricant (other than that applied during manufacture). He goes on to say that one of the most successful lubrication methods incorporates automatic or semiautomatic mist applications of small amounts of a "light lubricating oil," just enough to keep the rope damp. Proper lubrication adds a safety bonus because a light oil usually keeps the rope clean enough to facilitate inspection.

The best paper in terms of lubricant technology coverage is Myers'.(23) It covers all facets of lubrication for wire rope, including the properties of lubricants and cleaning and lubrication techniques. This reference not only has a good basis in lubricant technology: it also relates to wire rope used in mining applications, although probably underground mining rather than surface mining. Without bias Myers presents the types of lubricants available and their good and bad points relative to performance. He introduces the controversy concerning which is preferable--a light oil or the heavy asphaltic residua used as wire-rope lubricants. In addition, Myers discusses lubricants used during wire-rope fabrication as well as those applied during rope use. He is one of the few authors of information on wire-rope lubrication citing desired lubricant properties. Such properties are those that can be built into a lubricant by knowledgeable selection of the base stock and skillful formulation with additives. They include:

- rust and corrosion inhibition
- good wetting ability
- adhesiveness and cohesiveness
- extreme-pressure (boundary lubrication) properties.

It is well within the state-of-the-art to produce lubricants having these desired properties; however, few commercial lubricants exhibit all of the properties in an optimal balance.

The controversy concerning lubricant consistency was the major subject of a paper by Barden and Klaus(28) addressed specifically to surface-mining applications for wire rope. The authors indicate that rope is normally lubricated by the manufacturer, but the effect of the initial lubricants lasts for only a short time. Therefore, as soon as the rope is installed and machine operation begins, an immediate lubrication maintenance program by the mine operator is recommended.

Barden and Klaus recommend the use of a light oil--but only a "high-grade" oil, compounded especially for wire rope--because of its ability to penetrate the rope core. The use of a low-viscosity oil requires no change of grade from summer to winter. Light lubricants tend to have a washing effect and clean out dust and abrasive particles rather than attracting them. They point out that any lubrication is better than no lubrication at all, because it "cuts down the abrasiveness of the sand or dust."

Siding with the "light-oil advocates," Barden and Klaus explain that light-oil lubricants are especially appropriate for ropes used on large stripping machines. On the smaller ropes (<2-inch diameter) for loading machines, etc., users often prefer the heavy asphaltic lubricants. Lubrication is a more critical factor on large draglines than on loading shovels having only one moving rope--the hoist rope. On loading shovels, the hoist rope is comparatively small, and replacement cost is nominal. But on large draglines, there can be four hoist ropes and four drag ropes that require early replacement. The authors state that "proper lubrication of these ropes can double or even triple their lives, as proven by the records of several operators, and the savings can be substantial."

During this project's field survey, the authors were not able to secure operator data allowing comparison between the rope-life benefits accrued when good lubrication practice is used and that obtained when no lubrication is used. As mentioned earlier, very few of the operators visited lubricated the drag ropes on a routine basis; most of the hoist ropes were lubricated (at the point sheave and sometimes at the deflection sheave) only once per shift. The authors concur with Barden and Klaus that lubrication can be very beneficial. However, their statement that ".... the average dragline uses about 55 gallons of wire rope lubricant per month" was found to be a gross overestimate of the amounts actually used. In addition, they state that "as a general rule, the entire length of the rope should be lubricated every 8 hours on a machine operating continuously." Again, in practice, only the rope sections passing over the sheaves are lubricated; in many machines the rope sections that take the greatest internal stress (e.g., that section going onto the drum when the haul is begun) are not lubricated at all! These are the sections of rope where some machine operators and/or oilers reported that they make judgments about the progress of rope failure and need for change-out by observing lay length increase and diameter reductions. Lubrication to mitigate destructive internal stresses in wire ropes is used much too sparingly and locally.

Dobie[51] implied that, although development work is being carried out to improve the wire-rope lubricants used during rope manufacture, little was being done in 1962 relative to lubrication effectiveness for ropes during use.

He disparaged the practice of using thick lubricants to coat the outer surfaces of wire rope and indicated that many examples can be found of ropes appearing to be a bar of grease, but containing completely corroded and, in some cases, broken internal wires. Dobie terminates his paper with the following recommendation:

> Low-viscosity lubricants are preferable to thick oils and greases for application in service and these should be applied under high pressure (>800 lb/inch square), wherever possible. Drip feeds should be positioned so that the rope is bent immediately aferwards in order to distribute the lubricant internally.

Special consideration for the lubricant incorporated into wire ropes designed for specific uses was referred to in an article appearing in Engineering.[52] The article stated

> During the time in which the wire department has been carrying out their investigations and drawing up specifications for the production of rope samples (for special or new applications), the rope research department will have been pursuing preparatory work on the all-important lubrication to be incorporated into the final ropes.--It covers independent and collaborative research on such factors as chemical stability, lubricity, corrosion protection, adhesion, weathering and aging, effects of thermal changes, etc.

The above commitment to lubrication sounds impressive but, as Dobie indicated, the maintenance aspects of lubrication are probably the most influential factors affecting rope life.

Sherwood[53] essentially reports on the work of Lex[49] in an article elaborating on the lubrication aspects in Sherwood's other paper[21] describing maintenance procedures designed to produce longer life in wire rope. It appears that in the mid-1950s, the industry was just beginning to recognize the benefits of external lubrication of rope during use. Lex[49] was the most important harbinger of the news that lubricants can penetrate wire rope to restore lubricant that is included during manufacture but is soon lost in use by exiting the rope ".... one way--out." Lex proved that lubricants added during rope use penetrate to the fiber cores of ropes which were intentionally left unlubricated during their manufacture. He also proved that the addition of external lubricant extended rope life significantly.

Figure 11 demonstrates this conclusion for fiber-core ropes lubricated conventionally during manufacture. One set was drip lubricated during bend cycling and was run against a set receiving no external lubrication. As a result of this work and much practical experience since that time, it is obvious that the addition of external lubricants during rope use does, in fact, benefit rope life (and probably sheave and drum life as well). Unfortunately, the magnitude of such benefits for large ropes remains to be quantified.

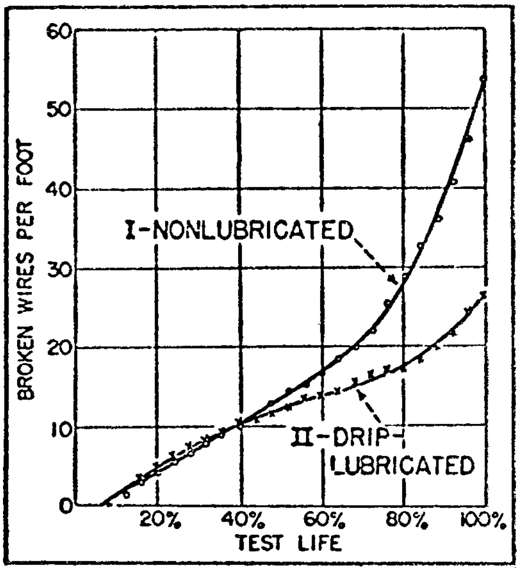


FIGURE 11. Rope Life Lengthened by Lubrication. Eight ropes used in each test, one group run without lubrication and the other with. Nonlubricated ropes removed when inspection showed 50 or more broken wires per foot. Corresponding lubricated ropes removed at same number of test revolutions. After 40% of test life, nonlubricated ropes showed far more broken wires than lubricated.

Meals[26] brings up a point important to users of fiber-core rope. Apparently, when he published his paper in 1929, most of the wire rope used in lifting and hauling contained a fiber core, rather than an IWRC. Therefore, Meals centered on the benefits of lubrication that accrue to the core itself. He says the life of the core depends largely upon its being lubricated throughout its life in service. Core wear is minimized by keeping it impregnated with lubricant at all times. A dry core will not only absorb moisture, but will deteriorate and crush quicker than will a well-lubricated one--all of which affect the wearing qualities of the rope. Because the core supports the rope, Meals concentrated on preservation and maintenance of the core--about which "it has been said that the life of a wire line is largely dependent upon the core."

Meals also deals quite specifically with the selection and application of wire-rope lubricants. He readily admits to the need to add lubricant during rope use; however, he recommends heavy lubricants: "Wherever possible, it is best to utilize a thicker, semi-plastic lubricant and to apply it hot in a thinned condition, rather than to apply cold a more fluid lubricant."

One article written in 1929 discusses the viscosity/temperature problem connected with wire-rope lubricants in cold climates. The assumption is that the high-viscosity lubricants might not adhere or function under winter conditions; therefore, lower-viscosity materials are recommended. Also discussed is the need to apply lubricants in the heated condition--which assumes that one needs to lower the viscosity to effect penetration into the rope. Finally, the split-box method of application is presented.

Written more recently, Reference 29 repeats the same kinds of statements that appeared in the 1929 article. However, it does admit that lubricants for field application generally have different physical properties than do those incorporated into the rope during manufacture, primarily because hot application is impractical. In some cases, the same or a similar lubricant is dissolved in a suitable solvent for ease of application.

Kasten[31,45] produced two papers on wire-rope lubrication, one in 1963 and one in 1964. In the earlier paper, he considers the problem of lubricant melting at high ambient temperatures (130°F), and brittleness during cold weather. Otherwise, he mentions desirable properties of lubricants,[31] and cites the high incidence of rope failures and the possibility of accidents where poor lubrication is responsible. The only new information, however, is Kasten's statement that spooling on a drum can be enhanced by the use of extreme-pressure (E.P.) lubricant additives in the lubricant formulation. It is not surprising that any high contact-stress sliding junction in a wire-rope system or in the rope itself, can benefit from an appropriate boundary-lubrication improver such as an E.P. additive.

Kasten's later paper[45] is occupied almost entirely with some of the requirements for a good wire-rope lubricant. In this article, Kasten provides a 12-point checklist of desirable properties for wire-rope lubricants:

- resistance to melting at high ambient temperatures
- resistance to becoming brittle at low ambient temperatures
- adhesiveness
- physical and chemical stability
- effective lubricating qualities for wire rope
- water repellency and corrosion-preventive properties
- melting point suitable for application
- resistance to acids and salt water
- resistance to caking, balling up, and gumming
- rapid solidification to smooth coating on wire
- prevention of bacterial deterioration of fiber core
- conformation to government standards.

The two papers were remarkably similar with regard to subject matter and wording.

The Soviet scientists Velikanov et al[54] point out that two kinds of lubricants are necessary to lubricate wire rope--one for impregnating fiber cores and the other for lubricating the rope itself. Currently used lubricants, both domestic and imported, have shown inadequate corrosion-preventive qualities and other less-than-optimal characteristics. Therefore, the authors engaged in a lubricant-development program resulting in the formulation of "Compound PSK-185" for fiber-core impregnation and "Cable Lubricant VKS-244-U" for ropes. PSK-185 is described as being water stable, even in water containing corrosive agents. It gives the rope a "high degree of adhesiveness" and its performance is satisfactory over a wide range of temperatures. VKS-244-U surpasses competing lubricants with respect to corrosion protection, physical stability, resistance to the effects of low and high temperatures, adhesion and "other areas." The authors generated an economic model proving that, although more costly than their predecessors, the newly developed lubricants would be economically beneficial because they may increase rope life by a minimum of 10 percent. Unfortunately, the chemical compositions for the new lubricants were not divulged. However, they wouldn't have to be very sophisticated to perform better than the baseline materials: wood resin for the fiber-core impregnant and petrolatum (probably additive-free) for the rope lubricant.

"Lubricating a wire rope is as necessary as putting motor oil in a car."[44] The quote is reported to have been made by wire-rope industry engineers who also recommend that operators move ropes at regular intervals in order to shift wear, much like rotating tires on an automobile.

The Fourth International Congress of Transportation by Rope, held in Vienna, was reported in Reference 34. In a number of papers presented on lubrication, both hauling and carrying applications for ropes were discussed, each calling for quite different lubrication methods. The importance of lubrication in the production of ropes was presented by Gedecke and Naumann of Dresden. Lubrication was claimed to be the fourth most important factor in rope manufacture--after rope construction, rope materials and manufacturing technology.

Correct lubrication by rope users was also mentioned; "unsuitable ones, e.g., thin oils, give no protection against corrosion and, first and foremost, may not be compatible with the lubricant used by the rope manufacturer." Although the latter half of this statement has merit, the Battelle researchers believe the first half is misleading. Certainly thin oils can be made corrosion-preventive by the use of additives.

Another paper presented described the work of Zbinden of Bern. Zbinden expressed the fact that the life and operating safety of wire ropes are considerably affected by cleaning and lubrication.

The title of a third paper on the subject is even more pointed than Zbinden's statement. "More Safety Through Modern Lubricants," by Neutschil, reports on tests covering many years, aimed at finding the best lubricant for ropes used in low-temperature environments. The summary terminates with the statement "Synthetic lubricants are, of course, the best to meet demands for higher safety."

Two articles by Layton(25,27) referenced earlier mention lubrication in passing. The earlier paper(25) advocates frequent lubrication--whether the rope is in use or not. He warns that ropes must be lubricated internally, as well as externally. "Smearing grease on the outside of a wire rope is not lubricating it." He concludes that "generally, wires of smaller diameter are better lubricated with lubricants of lighter viscosity, but plentiful penetration to the inner wires must always be the prime consideration."

Layton's later article(27) concerns itself with a number of good maintenance tips. Lubrication is mentioned as one important step and the "split-box," "pour-on" and "bath" techniques for applying lubricants are pictured.

Comments concerning wire-rope lubrication were made in a paper by Russell.(35) However, he discussed nothing really new in his paper.

REFERENCES

1. Learmont, Tom, "Productivity Improvement in Large Stripping Machines," Transactions of the Society of Mining Engineers, Vol. 258, pp. 239-246 (September, 1975).

2. Reemsnyder, H. S., "The Mechanical Behavior and Fatigue Resistance of Steel Wire, Strand and Rope," Presentation made to the Ad Hoc Committee on Mechanical Rope and Cable, National Materials Advisory Board, (June, 1972).

3. Godfrey, H. J., "The Strengthening of Steel by Wire Drawing," Wire and Wire Products, Vol. 38, pp. 51-59, 129, 130 (January, 1963).

4. Vitelli, V. J., "The Effect of Heat Treatment and Cold Work on the Mechanical Properties of High Carbon Steel Wire," Wire, pp. 1763-1766 (December, 1962).

5. American Iron and Steel Institute (AISI), Specification on the Grades and Strength of Carbon Steel Wire, Spec. XYZ14.

6. Dull, R. G., "Rope Wire for Today's Applications," Wire and Wire Products, pp. 1642, 1643, 1713, and 1714 (October, 1966).

7. Wire Rope Handbook and Catalog E, Armco Steel Corporation, Union Wire Rope, Kansas City, Missouri (1972).

8. USS Tiger Brand Wire Rope, United States Steel, Catalog ADUSS 55-2460-05, (1974).

9. Bethlehem Wire Rope for General Purposes, Catalog 1877, Bethlehem Steel Company, Bethlehem, Pennsylvania (undated).

10. Wire Rope Catalogue, WRI Publication 681, Wire Rope Industries of Canada Limited, Lachine, Quebec, Canada (undated).

11. Wire Rope Handbook, Broderick & Bascom Rope Co., St. Louis, Missouri (1966).

12. Wireco Wire Rope, Handbook W67, Wire Rope Corporation of America, Inc., St. Joseph, Missouri (1971).

13. MacWhyte Wire Rope, Catalog G-17, MacWhyte Wire Rope Company, Kenosha, Wisconsin (1974).

14. Wire Rope and Slings, Catalog 107/3M/M, Greening Donald Ltd., Hamilton, Ontario (April, 1974).

15. Finnis, P. J., "Consideration of Some Factors Affecting the Economic Use of Wire Rope on Open-Pit Mining Equipment," The Canadian Mining and Metallurgy Bulletin, pp. 103-108, (August, 1973).

16. Anon., "Armco Zinc-Epoxy Socket Now Available with Union Wire Rope," Product Data Sheet LK-3075, Armco Steel Corporation (June, 1975).

17. Personal Communications with Bethlehem Wire Rope Company Engineers, Williamsport, Pennsylvania (November, 1975).

18. Anon., "Installation, Maintenance, and Inspection of Wire Rope," Volume III of the Wire-Rope Handbook, prepared for the Navy by Battelle Columbus Laboratories under Contract N00024-74-C-5295 (May, 1976).

19. Russell, W. C. and Flowers, A. E., "How to Get More Life from Wire Rope," Coal Age, pp. 76-81 (April, 1955).

20. Meger, A. F., "Making and Handling the World's Largest Wire Rope," Coal Age, pp. 90-92, (July, 1968).

21. Sherwood, P. W., "Longer Lives for Wire Ropes," The British Steelmaker, Vol. 31, No. 1, pp. 24-25 (January, 1965).

22. Meger, A. F., "Wire Rope Maintenance," Mining Congress Journal, pp. 34-37 (July, 1960).

23. Myers, S. J., "Maintenance, Lubrication, and Preservation of Wire Ropes," Mining Society of Nova Scotia Transactions, Vol. LVIII, pp. 167-174 (1955).

24. Reynolds, E. S., "Proper Lubrication of Wire Rope Pays," Pit and Quarry, pp. 112-115 (September, 1963).

25. Layton, C. H., "Longer Life for Your Wire Rope," Engineering and Mining Journal, Vol. 157, No. 10 (October, 1956).

26. Meals, C. D., "The Selection and Application of Wire Rope Lubricant," Wire, pp. 230-248 (July, 1929).

27. Layton, C. H., "Proper Maintenance Will Make Your Wire Rope Last Longer," The Oil and Gas Journal, Vol. 66, No. 31, pp. 76-80 (July, 1968).

28. Barden, K. L. and Klaus, F. J., "Making Wire Rope Last Longer," Coal Age, pp. 79-82 (March, 1972).

29. Anon., "Wire Rope," Lubrication, Vol. 53, No. 6, p. 73 (1967).

30. Anon., "Wire Rope Lubrication in Winter," Wire, (reprinted from the magazine "Lubrication") pp. 49-50 (February, 1929).

31. Kasten, R. O., "Wire Rope Lubrication--Its Relation to Rope Life and Safety," Wire, pp. 1271-1273 (September, 1963).

32. Grimwood, J. E., "When Do You Change Wire Rope Cables?" Iron and Steel Engineer, pp. 94-97 (June, 1958).

33. Murphy, W. A., "Wire Rope--The Working Arm of Construction Equipment," Society of Automatic Engineers Paper No. 40, from the National Farm, Construction, and Industrial Machinery Meeting, Milwaukee, Wisconsin (September, 1964).

34. Anon., "Technical Investigations of Wire Ropes," Wire Industry, pp. 680-683 (September, 1975).

35. Russell, W. C., "How to Select and Care for Wire Rope and Operating Conditions Which Shorten Its Life," National Engineer, pp. 43-46 (November, 1956).

36. Anderson, W., "Factors Affecting the Efficiency and Safety of Steel Wire Ropes," Lift, pp. 229-230 (November/December, 1972).

37. Brown, R. S., "Wear Resistance of Wire for Wire Rope," Wire and Wire Products, pp. 1037-1047 (November, 1948).

38. Anon., "Wire-Rope Analysis and Design Data," Wire Rope Handbook, prepared for the Navy by Battelle Columbus Laboratories under Contract N00024-74-C-5295, Vol. II (May, 1976).

39. "VDI Guidelines for Lifting and Hauling," VDI 2358 [Published in Germany, English translation is NAVSHIPS Translation 1417] (October, 1968).

40. Gibson, P. T., White, F. G., Schalit, L. A., Thomas, R. E., Cole, R. W., and Cress, H. A., "A Study of Parameters that Influence Wire Rope Life," Final Report prepared by Battelle's Columbus Laboratories for the Naval Ship Systems Command under Contract No. N00024-72-C-5427 (October, 1974).

41. Gibson, P. T., Larsen, C. H., and Cress, H. A., "Determination of the Effect of Various Parameters on Wear and Fatigue of Wire Rope Used in Navy Rigging Systems," Final Report prepared by Battelle's Columbus Laboratories for the Naval Ship Research and Development Center under Contract No. N00600-70-C-1045, AD-776 993 (March, 1972).

42. Cordiano, H. V. and Wolfe, R. J., "Report of Investigation of Uncoated Wire Rope, Preformed and Nonpreformed," Lab Project 4750-18, Material Laboratory, New York Naval Shipyard (May, 1951).

43. Cordiano, H. V. and Wolfe, R. J., "The Resistance to Fatigue of Uncoated Hot Dip Galvanized and Nongalvanized Wire Rope, Preformed and Nonpreformed," Lab Project 4750, Material Laboratory, New York Naval Shipyard (June, 1953).

44. Anon., "The Wonderful World of Wire Rope," Wire, pp. 616 and 661.

45. Kasten, R. O., "Wire Rope Lubrication Means Longer, Safer Rope Life," Lubrication Engineering, pp. 180-183 (May, 1964).

46. Müller, Hugo, "The Properties of Wire Rope Under Alternating Stresses," Wire World International, p. 249 (October, 1961).

47. Gibson, P. T., Karleen, R. E., and Cress, H. A., "The Continuation of Analytical and Experimental Investigation of Aircraft Arresting Gear Purchase Cable," Final Report prepared by Battelle's Columbus Laboratories for the Naval Air Engineering Center, AD-905 835 (June, 1971).

48. Anderson, W., "Factors Affecting the Efficiency and Safety of Steel Wire Rope," Lift, pp. 109-113 (May/June, 1972).

49. Lex, W. I., "Lubricants Do Penetrate Wire Rope," Engineering and Mining Journal, Vol. 155, No. 6, pp. 96-98 (June, 1954).

50. La Que, F. L., "Corrosion and Protection of Mine Hoist Ropes," Corrosion-National Association of Corrosion Engineers, Vol. 6, pp. 8-13 (January, 1950).

51. Dobie, J. M., "Recent Developments in Wire Rope Manufacture and Application," Mine and Quarry Engineering, pp. 356-362 (August, 1962).

52. Anon., "Progress Towards Improved Steel Ropes," Engineering, pp. 798-799 (June, 1962).

53. Sherwood, P. W., "Proper Lubrication and Its Relation to Service Life of Wire Rope," Wire and Wire Products, pp. 1696 and 1717-1715.

54. Velikanov, K. M., Polyakov, V. S., Semenov, R. P., and Frolov, R. I., "The Economic Effectiveness of the Use of Cable Lubricant VKS-244-U and PKS-185, Developed at the M. I. Kalinin Polytechnic Institute," Trudy, Leningrad Polyteckhnicheskiy Institut, No. 282, pp. 166-174 (1967).

APPENDIX B

SOME PRELIMINARY FINDINGS ON WIRE ROPE LIFE IN SURFACE COAL MINING DRAGLINES[a]

(a) This appendix was written by W. E. Anderson of Battelle-Northwest and T. M. Brady of the U.S. Bureau of Mines, Spokane, WA.

SOME PRELIMINARY FINDINGS ON WIRE ROPE LIFE IN SURFACE COAL MINING DRAGLINES

As part of a larger study to identify factors influencing the practical operating life of wire rope used on large draglines in surface coal mining, field trips to operating surface coal mines were made during late 1975. Wire rope performance information was obtained from personnel at thirteen mine sites west of the Appalachians. Seventeen pieces of equipment, utilizing buckets ranging from 30 cubic yards to 220 cubic yards capacity, were documented. This report outlines preliminary findings which may prove useful to many operators in extending wire rope life, thereby increasing surface coal mining productivity.

Although study findings did not reveal anything about dragline rope life that wasn't known or practiced by one operator or another, most operators could expect improved rope life by periodic review of thier practices and by adopting or adapting one or more life-improvement measures employed at other sites. We did not find anyone using presocketed dump ropes or dump rope sheaves that opened like a snatch block, so we are suggesting these as candidate improvements.

We estimate that an improvement of only 10 percent in the average rope life in about 120 large draglines throughout the United States would permit removal of an additional 4 million cubic yards of overburden each year, without significant capital investments by the operators.

DATA ACQUISTION

During the field visits, mine operators provided rope life information about their machines. Some of the rope life data consisted of gross averages; other data was in the form of detailed records, including rope construction and manufacturer, cutoff and resocketing periods, end-for-end switching, and downtime for rope change-out.

Trends of average hoist rope life and average drag rope life are shown in Figure 1 as a function of bucket size. The three lines from the origin have slopes of 1, 1/2 and 1/4. Comparing the data against these, it may be seen that hoist ropes moved about twice as many yards as drag ropes prior to retirement, although more than two-to-one scatter within each category is evident. As expected, the larger machines with the larger buckets moved more total yardage.

The four pairs of solid symbols represent those data for which detailed rope life histories are presented in subsequent figures.

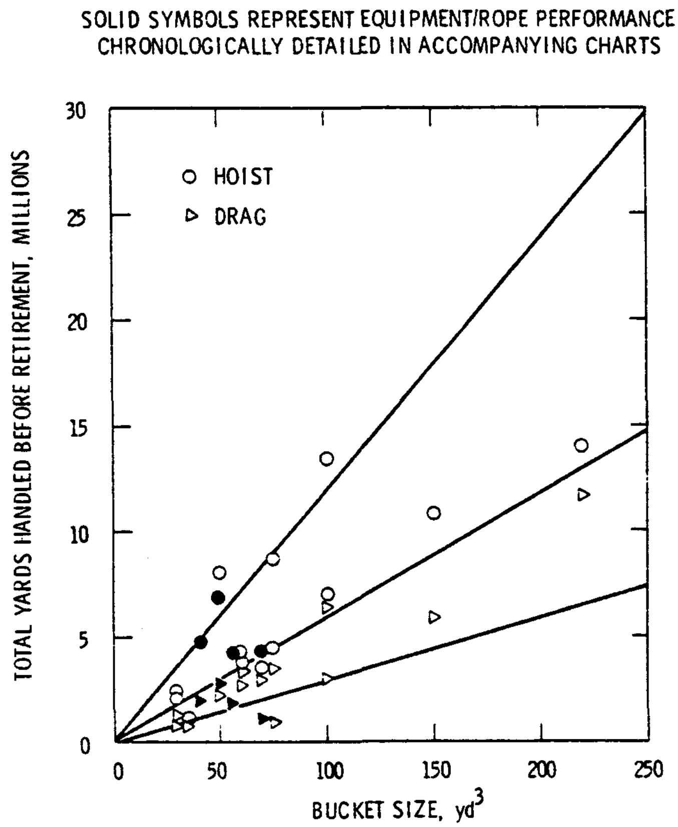


FIGURE 1. Equipment/Rope Performance in Total Yards as a function of Bucket Size

DATA TREATMENT

In this preliminary investigation, it was recognized that only gross features or trends, if any, could be uncovered. Consequently, it was assumed that operators would employ the most appropriate wire rope they could acquire for their equipment. This study did not attempt to ascertain if differences in rope life might arise from differences in rope construction; comparisons between ropes of 6 x 49 versus 6 x 57, for example, were not made. Neither were allowances made for the type of overburden moved, although this factor clearly affects rope life.

Some form of data normalizing, however, was considered desirable. One apparently useful factor was the amount of cross-sectional wire performing the hoist or the drag function. A simple expedient was to sum the squares of the nominal rope diameters used in each function and divide that number into the total yards handled until retirement. The performance of draglines with different bucket sizes could then be crudely compared, since the normalizing factor is approximately proportional to the actual cross-sectional area of the wires in the rope (and hence the weight per unit length of rope, or, roughly, the cost per unit length).

Equipment/rope performance data of Figure 1 have been normalized in this manner and are presented in Figure 2, showing normalized rope performance in terms of cubic yard bucket size. Average and median performances are indicated. The normalized performance may be viewed as an efficiency rating of rope use; further, there is about a five-fold difference in utilization efficiency for hoist ropes and for drag ropes among the pieces of equipment. This difference is certainly due in part to variation in type of overburden handled and the nature of the digging operation itself. However, some of the difference might well arise from operating practices. From this vantage, we believe some modest improvements in average rope life could be developed.

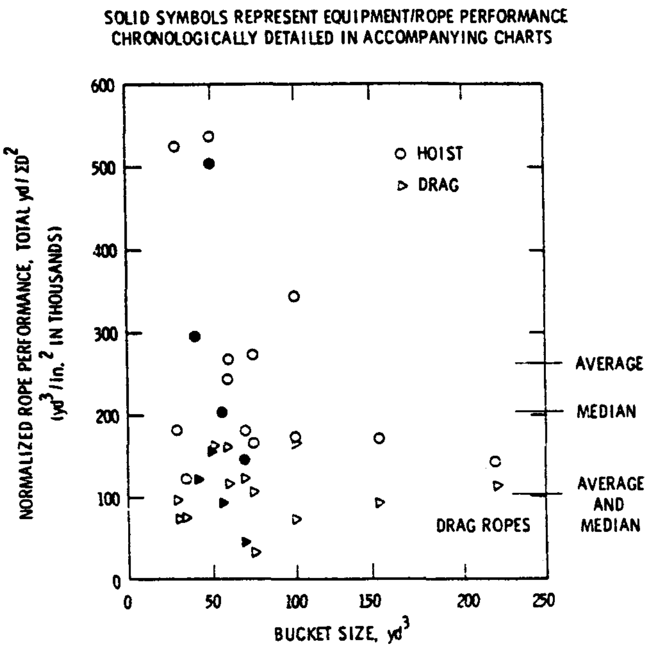


FIGURE 2. Normalized Equipment/Rope Performance as a Function of Bucket Size

In Figure 3, the normalized rope performance for both hoist and drag ropes is randked; median and average performances are noted. The light lines connecting solid symbols in Figure 3 show respective ranking positions for ropes from the same equipment.

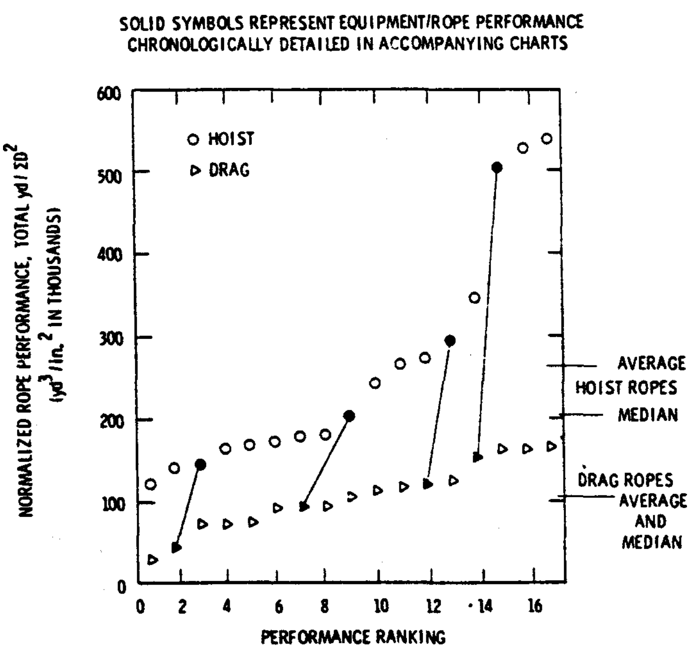


FIGURE 3. Ranking by Normalized Roper Performance

HISTORICAL PERFORMANCE

Detailed rope performance is shown in Figures 4 through 7, containing rope life historical information previously indicated by solid symbols. Mine/equipment code numbers were employed to ensure anonymity of information sources. Similarly, only the approximate bucket sizes are given, and references to rope manufacturer are omitted.

CODE 6-2

Normalized hoist and the drag performance for mine 6, equipment identification -2 is plotted in Figure 4. This was a 40-yard class dragline. Points indicate the calendar date on which each rope was retired, versus the normalized performance of each pair of ropes (hoist and drag were two ropes, 2-7/8 inches in diameter).

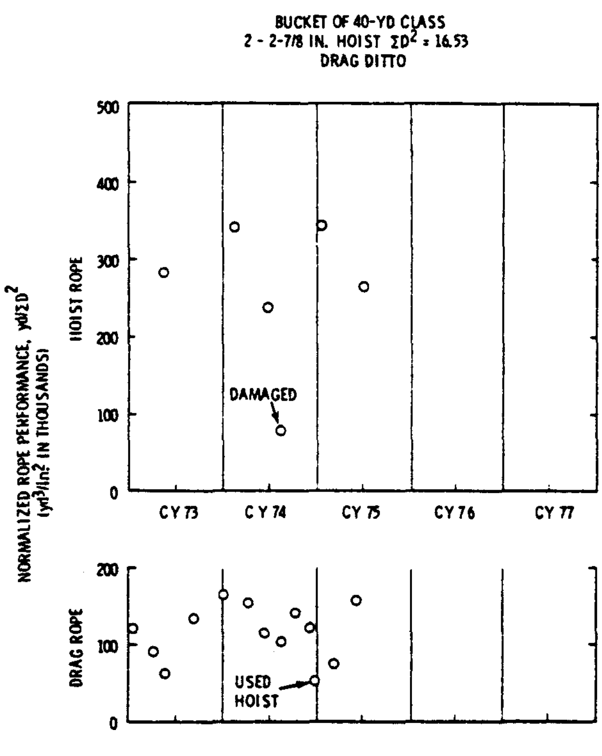


FIGURE 4. Normalized Hoist and Drag Rope Performance Mine/Equipment Code 6-2

Mine records indicated drag rope damage in the summer of 1974; that information is noted beside the symbol. In a similar manner, a used hoist rope was put on as a drag rope late in 1974; that item of information is also noted.

Average normalized performance of the hoist ropes may be visually estimated as about 300,000 units, while the drag rope performance was about 125,000 units. There is some variation, depending on whether the damaged rope of the used rope data are considered in calculating an average.

CODE 14-2

Figure 5 describes rope performance on equipment at mine code 14, which carried a bucket of the 50-yard class; the two hoist ropes were reported as 2-5/8 inches in diameter and the two drag ropes as 3 inches in diameter. Numbers with arrows above them indicate data points which are offscale.

Rope performance on code 14-2 equipment is characterized by about 500,000 units for hoist ropes about 150,000 units for drag ropes.

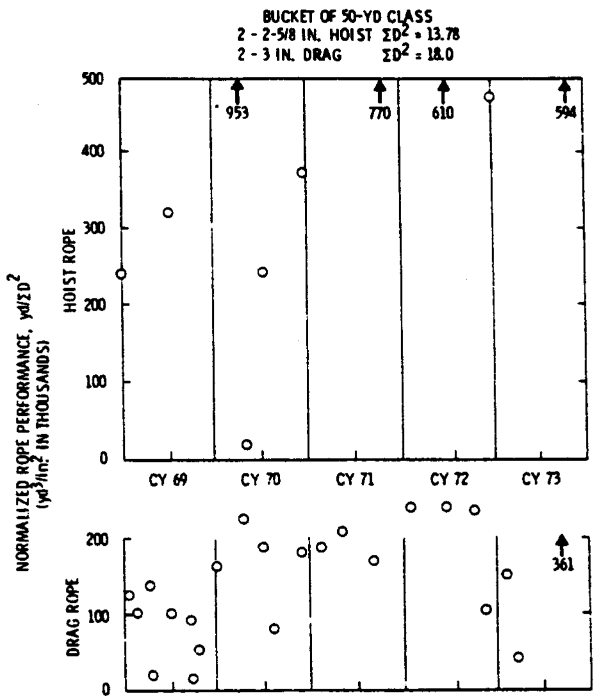


FIGURE 5. Normalized Hoist and Drag Rope Performance Mine/Equipment Code 14-2

CODE 4-1

Mine/equipment code 4-1 data are shown in Figure 6. Average hoist rope life was approximately 200,000 units, normalized; average normalized drag rope life was almost 100,000 units. The bucket was of the mid-50's class, and all rope was 3-1/4 inches in diameter.

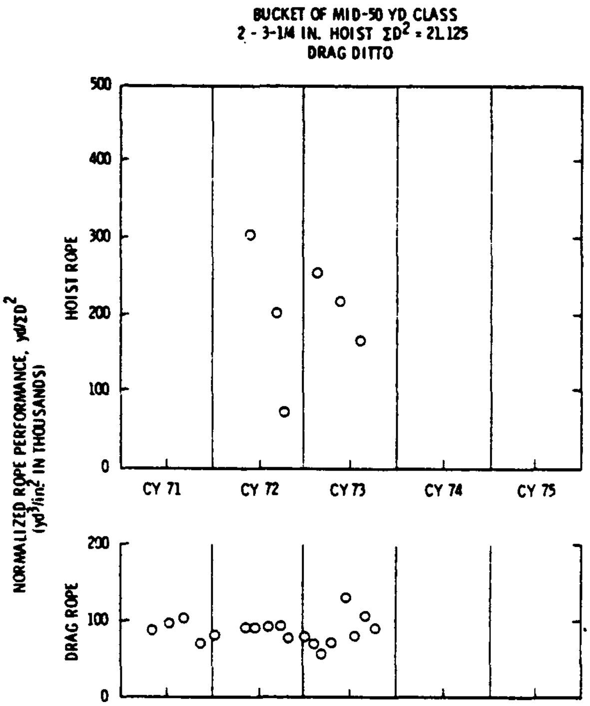


FIGURE 6. Normalized Hoist and Drag Rope Performance Mine/Equipment Code 4-1

CODE 2-1

Mine/eqiupment code 2-1 is for a dragline of 70-yard class; all ropes were 3-1/2 inches in diameter. Hoist rope averaged about 150,000 normalized units, while drag rope performance was approximately 50,000 units. These data are shown in Figure 7.

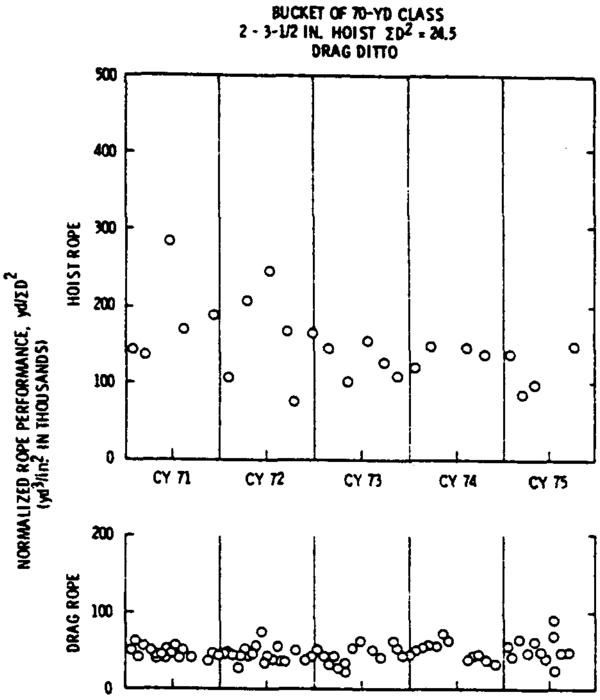


FIGURE 7. Normalized Hoist and Drag Rope Performance Mine/Equipment Code 2-1

DISCUSSION

Information describing dump ropes was insufficient to permit data treatment like that carried out for the hoist and drag ropes. Dump ropes were a very expensive nuisance problem at some of the mines, while they were said to constitute no appreciable difficulties at other mine sites. This aspect will be discussed in more detail below.

Although the importance of bucket performance is well recognized and must be connected with rope life in several ways, this report does not directly treat such aspects. Discussions of bucket performance were outside the purview of this study and thus necessarily avoided, although bucket/overburden factors will certainly affect rope life from site to site. In a somewhat related manner, other operational, maintenance and equipment factors are known to influence rope life, but they are not treated in substantial detail for this preliminary report.

DOWNTIME DUE TO ROPE CHANGES

The downtime and consequent lost production due to stoppages for rope change-out or resocketing varied considerably from site to site, but the overall picture tended to confirm previous estimates.[a] In one case we timed the drag rope change-out on a 60-yard class machine with 3-3/8 inch ropes; it took one hour and fourteen minutes from bucket set-down to bucket pick-up. We judged less than twenty man-hours were required for the complete operation.

At another site, we were told that the change-out of drag or hoist ropes (on a 100-yard class bucket) would take the better part of one shift and would involve at least five men. We estimated that, on the average, change-outs, resocketing and swapping ends was taking about 105.6 hours/year; total effort was about 500 man-hours for hoist, drag, and dump combined.

(a) E. P. Pfleider (ed.), Surface Mining, AIMMPE, N.Y. (reprinted) 1972, p. 458.

Reported difficulties or problems with dump ropes were even more varied than for drag or hoist ropes. One operator noted that more total time was spent in chaning dump ropes than for drag and hoist combined. Several operators reported using sections of retired rope from other equipment which would then last for a satisfactorily long period. On the other hand, some operators remarked that even new dump ropes would wear out in a distressingly short time.

We understand it is common knowledge that dump ropes need to be set to rather tight limits for proper operation and long life. Yet, from this preliminary study, we could not identify if the operators having dump rope problems were appropriately prudent in this regard, or if there was some complication traceable to the equipment itself. It did appear that at least some operations could be improved if the dump ropes were presocketed to a given length and then simply cast over the dump sheave by employing a special sheave assembly which could be quickly opened and resecured. Even presocketing one end of the dump rope for the conventional sheave might prove cost effective in some operations.

Our estimate of total production loss, due to all aspects of downtime from rope changes or resocketings, was based on an "average" 60-yard machine. This loss amounted to a total of 40,550,400 yards not moved, for an estimated 120 machines of this average capacity. If the downtime losses could be improved only 10 percent by adjusting practices which increase rope life, the annual increase in overburden removal would total more than 4 million yards across the nation.

PERFORMANCE OVERVIEW

Figures 1, 2, and 3 show the rope performance for the mine sites and equipment. As stated earlier, these are averaged results for each item of equipment and particular operations of overburden removal.

The crude performance data fell in two obvious bands, visible in Figure 1. Rope-to-sheave ratios, type of overburden or digging conditions, and other refinements relating to lubrication, weather environment or rope construction details, were not considered. Nonetheless, our findings are consistent with previous information,[a] thereby enchancing the suitability of this limited study for generalized interpretations.

After normalizing the rope performance (Figure 2), it became apparent that the larger machines were not as efficient in rope use as were the medium-size machines. Although rope efficiency is only one of several improtant criteria in judging overall machine efficiency, it appears worthwhile to explore this aspect more thoroughly when additional information on larger machines becomes available.

Outstanding performance of hoist ropes from three equipment items is obvious in Figures 2 and 3. These biased the average hoist life for all hoist rope data. The only clear difference between these machines and other, two-rope, hoist arrangements is that the rope sizes are all smaller than the trend line of rope diameter versus bucket size. This implies a larger-than-average sheave-to-rope ratio and hence less severe bend-over-sheave fatigue stresses in the ropes. On the other hand, performance may by so high simply because the overburden is lighter than normal. Further studies could clarify such points.

DETAILED RECORDS

Mine/equipment code 6-2 is identified with the data of Figure 4. It is a releatively new operation, but the performance is above the median on a normalized basis.

Figure 5 presents data for mine/equipment code 14-2, with records back to 1968. There appear to be several points of interest about this operation. We understand that a change in drag rope maintenance practice was made in late 1969; this action is apparently reflected in the performance data. By permission of the operator, we can report that one change was the replacement of wooden slide blockes in the cab with rollers. Anogher

(a) Pfleider, op. cit., pp. 457-459.

change was a shutdown at nearly every lunch period to walk the drag rope, burning off any broken whiskers which were observed.

A change in hoist rope lubrication was reportedly effected in 1970; this consisted of applying a penetrating fluid in a systematic manner. Again, results bear out the value of the changed practice. As mentioned above, the hoist rope is somewhat smaller than "normal" trend, so some of the generally improved performance may be traceable to this aspect. However, we do not have available information on the type of overburden or other site details at this time, and these factors may account for some portion of the unusually good performance.

Recent data for mine/equipment code 4-1 is not presently available; earlier information is shown in Figure 6. We are reluctant to comment further about operations at this site, for there may be several important factors affecting rope performance which we cannot incorporate into a meaningful comparative analysis for this introductory study.

Figure 7 reflects performance of the drag ropes for mine/equipment code 2-1, which might well be attributable to site conditions. We understand this operation is conducted in sandstone and is of such configuration that the drag ropes are ordinarily pulled through the abrasive overburden. In this situation, it may prove least expensive, in the total cost picture, to pay for the additional drag ropes rather than change procedures in mining. The poor hoist rope performance may be similarly traceable to an unusually dense overburden. Information is probably not available which whould permit calculation of any cost effectiveness resulting from using a smaller bucket, although additional studies might develop such data.

SUMMARY AND CONCLUSIONS

A limited investigation of dragline rope performance was carried out by preliminary field observations and discussions with selected surface mine operators. The obtained data appeared consistent with previous information on releative performance of hoist and drag ropes, providing some optimism that the findings are roughly representative of U.S. surface mining practices.

Attempts were made to compare rope performance among equipment carrying different bucket sizes by normalizing the rope yardage at retirement. The total yardage moved was divided by the sum of the squares of the rope diameter for the hoist and for the drag. On this basis, a five-to-one difference was found in both hoist and drag rope performance between the extremes of the seventeen draglines examined. Median hoist life was approximately twice the median drag life.

Downtime resulting from rope change-out varied considerably from one operation to another. Even more striking was the difference in experiences with dump rope lifetimes; some operators recited significant difficulties and frequent changes, while others reported little additional downtime loss.

We suggest that nearly every operator could profit from periodic review of his operational proactices in one or more of the following areas:

- Development of applicable techniques and training methods for expediting normal rope changes.
- Improvement of relevant knowledge about dump rope performance on the equipment employed and for the particular site digging; consider dump sheave modification and presocketed ropes.
- Maintenance/housekeeping procedures generally, from attention to rust and damage prevention in storage to whisker-trimming on the drag ropes and to improved lubricants/lubrication of both drag and hoist; consider abrasion-reduction practices in moving ropes through cab to the drums.

We conclude that periodic publication of rope performance data will serve to encourage operators to effect useful improvements on their equipment and useful practices at their sites. We recommend industry development of such survey and publication activity.

APPENDIX C

DESIGN AND CONSTRUCTION OF LARGE-DIAMETER WIRE ROPE BEND-OVER-SHEAVE FATIGUE MACHINE[a]

(a) This appendix was written by Mike Morgenstern of Battelle, Pacific Northwest Laboratories, Richland, Washington.

INTRODUCTION

The cost of maintaining and replacing wire ropes on surface mining equipment is one of the major expenses in strip mining operations. Battelle, Pacific Northwest Laboratories proposed a program to the Bureau of Mines in January 1975 to "Investigate the Feasibility of Improving the Efficiency of Surface Mining Equipment by Extending Rope Life."(1) A major part of this program was to design, build and operate a machine which would fatigue test 3-in. diameter wire ropes by bending each rope over a 90-in. sheave while cycling the rope back and forth, simultaneously applying a tensile load.

The basic design was completed in December 1975. Testing began in February 1976 and the first rope was failed in March 1976. This rope was tested at 30% of ultimate tensile strength (UTS) and failed after 30,805 cycles. During Phases II and III of the program, 23 ropes have been tested and 17 ropes have been tested to failure. The program was modified in early 1976 to include the design and construction of a tensile test machine adjacent to the fatigue machine. The tensile tester was designed to test 3-inch diameter IWRC Wire Rope up to 100% of manufacturers' predicted breaking strength. The tensile tester has been used to test ropes after initial testing on the fatigue machine. In all but one case the rope tested was the non-failed companion rope from the fatigue test. In order to obtain additional fatigue data, tensile tests were suspended during Phase III.

DESIGN-FATIGUE MACHINE

The fatigue machine is designed to test two ropes ∿60 ft long simultaneously. The ropes are cycled back and forth over sheaves while simultaneously loaded in tension. Cycle time during Phase 2 and most of Phase 3 was held at approximately 33 seconds and was increased to 40 seconds during the latter half of Phase 3. The fatigue machine consists of:

- pad and support structure
- sheaves, bearings and shafts
- diesel engine and hydraulic system
- safety systems
- instrumentation and recording system

The fatigue machine basically consists of two sheaves with their supports and bearings, and two hydraulic cylinders. The larger cylinder provides the load by extending one sheave assembly; the smaller double-ended double-acting cylinder cycles the ropes back and forth. The fatigue machine is shown in Figure C-1.

The large hydraulic cylinder has a 24-in. diameter bore and a 10-in. diameter shaft. It is double-acting single-ended and is capable of developing over a million pounds of force.

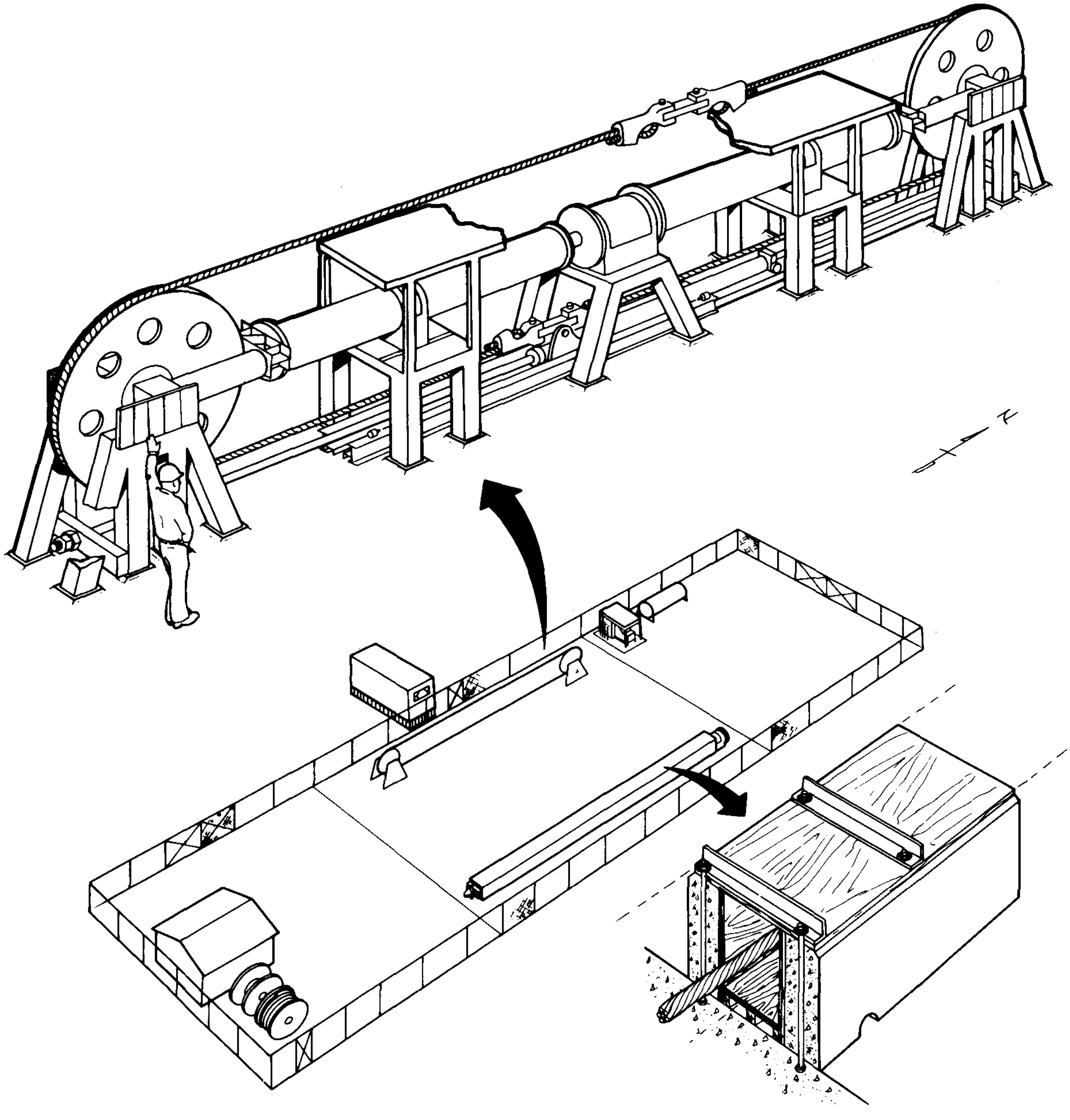
The smaller cylinder has a 7-in. diameter bore and a 3-in. diameter shaft. Double-acting and double-ended, it has a 20-ft stroke. The small cylinder is used to cycle the ropes back and forth. The cylinder itself cycles with the rope while the shafts are kept stationary and attached to the foundation through bolting to the machine structure. As a result the shafts are loaded in tension to react against the buckling movement caused by the cycling of the cylinder.

SHEAVES

The sheave assembly consists of a 90-in. diameter sheave made of A-36 mild steel plate. The shafts are 14 in. in diameter made from 4142 normalized forged steel and pressed in place. Two pillow blocks 20 x 20 x 13 in. contain bronze bearings.

BEARINGS

The bearings are cast from S.A.E. 660 bronze. There was originally a shrink fit between the bearing and the pillow block (bearing housing). Lubrication was provided through grease grooves in the bearing, fed through a Zert fitting at the top of the pillow block.



(Three-inch diameter capacity; machine owned by the U.S. Department of Energy and installed on Battelle-Northwest property)

FIGURE C-1. Wire Rope Fatigue Test Machine and Tensile Test Facility

INSTRUMENTATION

Instrumentation was designed to measure load on the rope (by measuring pressure in the tensioning cylinder) and elongation of the rope while cycling under load.

Pressure in the tensioning cylinder is measured by a pressure transducer and recorded continuously in the instrumentation trailer. Rope elongation was originally measured by monitoring the extension of the tensioning cylinder. This system, however, did not provide accurate data on rope elongation because we were also measuring the seating-in of the rope within the blocks. A new system was designed using extension transducers of the "pot and string" variety. The pot is bolted to a clamp made from schedule 40 pipe which is in turn clamped to the rope. A 1/4-in. channel was attached to one edge of the sheave. One-sixteenth-in. coated aircraft cable attached to the "pot" runs through the channel and is clamped to the other end of the rope. The stretch of each rope is recorded continuously on a strip recorder during the fatigue test.

SAFETY SYSTEM

The safety system consists of a series circuit containing a number of devices that will shut the machine down if a condition develops that could result in damage to equipment or injury to personnel.

The circuit consists of:

- thermocouple in each bearing to sense an overtemp condition,
- over travel limit switches at both ends of the machine,
- over-extension limit switch on the south pillow block to shut the machine down when a rope fails and prevent the block from sliding completely off its support stand,
- a high pressure sensor on the hydraulic system,
- a low hydraulic oil sensor,
- ground fault detectors on all external power pumps,
- low oil pressure cut-off switch on the diesel.

OPERATIONAL EXPERIENCE

The fatigue machine operation was initially satisfactory and the first rope failed after 30,805 cycles. The only problem encountered was that bearing temperatures tended to exceed the high temperature shutdown temperature of 150°F. An auxiliary water cooling system was added; this kept the bearing temperature below the 150°F at which the high temperature trip was set.

The most significant problem encountered during testing was preventing bearing failure. The service to which the bearings were subjected was severe. The bearing loads were high, rotation speed was slow (a cycle = 30 to 40 seconds) and the direction of rotation was continuously reversed without completing a revolution. Discussions with bearing manufacturers confirmed that the type of bearing used was probably the best since experience with other types of bearings such as roller bearings in similar applications had proven unsatisfactory. Shaft-bearing alignment was improved by contour plotting bearing-shaft contacts using soft solder.

During the second set of rope tests two bearings failed. The apparent failure mechanism was loss of lubrication due to rotation of the bearings within the pillow blocks. The failed bearings were replaced by thicker bearings of the same material. The new bearings were pinned in place and H-Pattern grease grooves were centered at the maximum load point. Since the new bearings were installed, bearing temperatures have been kept under control by using a combination of automatic lubrications and water cooling of the bearings. The high temperature shut-off system is currently set to shut down the fatigue machine when any of the thermocouples sense more than 185 to 190°F. This rather conservative number was chosen because the thermo-couples are not located at the point of maximum load and to minimize the possibility of bearing failure. In addition, used grease samples are taken periodically and examined for metal particles.

APPENDIX D

PROCEDURES FOR WEDGE-SOCKETING THREE-INCH DIAMETER WIRE ROPE

PROCEDURES FOR WEDGE-SOCKETING THREE INCH DIAMETER WIRE ROPE

Minimum Manpower and Equipment Requirements

Two Men	Experienced in handling heavy equipment and loads
One Vehicle	Capable of dragging 60 ft of rope with sockets attached (about 3500 lb) through the available work area.
Assorted Prys, Chains, Hammers, etc.	For moving and positioning sockets, wedges and rope.
One Deadman	Solid anchor of some sort, set such that the vehicle used cannot move it.

Procedure

1. Determine the pin to pin length for the socketed rope.
2. Due to the loop in the rope at each socket, the new rope should be cut six feet, five inches longer than the desired pin to pin length (for Esco socket 66908-1 and wedge W-25 with three inch rope).
3. Chain the clevis end of the socket to the deadman and remove the wedge. Socket on its side, pocket facing the maneuvering area.
4. The rope should be grasped about six feet from the end by wrapping a chain with a sliphook around the rope three times and placing the sliphook around the standing length of the chain. (This grasp hereafter called a "choker".) Fasten the standing end of the chain to the vehicle and drag the rope toward the socket, maneuvering the end of the rope to pass through and extend beyond the socket opening until the choker comes up to the socket. Remove the chain and rechoke on the other side, pulling until about 30 ft of rope is available for making an eye.

Now change direction, rechoke if necessary, and pull the rope (with five to six ft of free end) back along its length toward the socket, maneuvering the end into the wedge pocket. Continue until 18 in. of rope extends beyond the nose of the socket. (The length of this "tail" is critical and is the only control on the finished length of the rope.)

At this point, there should be an eight to 10 ft diameter loop extending from the wedge pocket. Place the wedge inside the loop with the tip resting on the inside edge of the wedge pocket.

Transfer the choker from the tail to the running end of the rope and slowly pull the loop out. Be sure to maintain the 18 in. tail and continually sweep the rope to prevent gravel and debris from being dragged into the wedge pocket.

As the loop approaches the wedge, it will pick it up and carry it into the pocket. When satisfied that the 18 in. tail has been maintained, continue to pull to the limits of the vehicle's traction to seat the wedge and rope into the socket.

The process is repeated at the other end of the rope.

Critical Points

A socket should never be installed unless the rope has been laid out for its entire length. Loops, bends and kinks in the running length of the rope should be avoided at all times.

For fatigue testing, both ropes must be installed with loops facing the same direction. That is, both wedges should be on the same side with no torque in the rope.

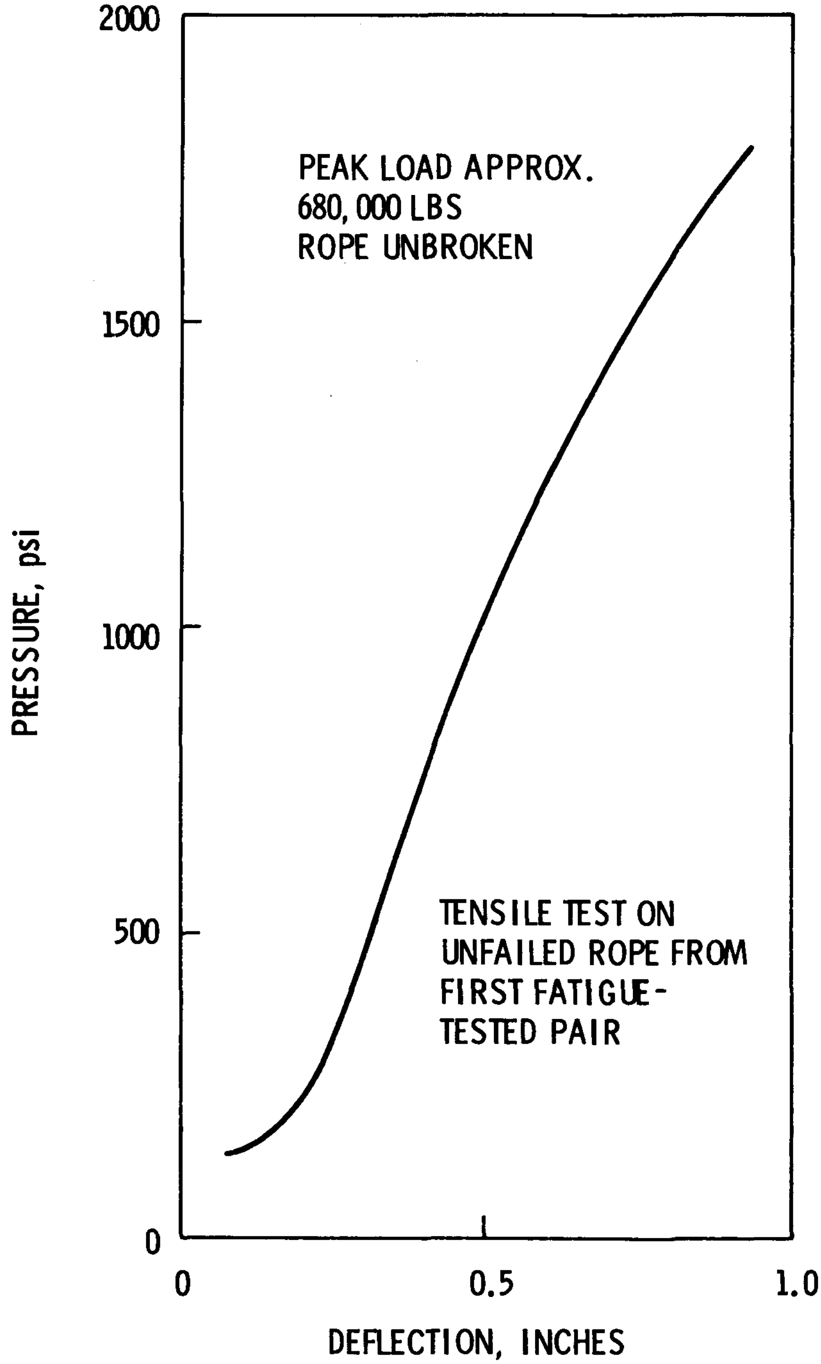
Following this procedure will produce a socketed rope within two in. of the desired pin to pin length.

Caution

These components are heavy! Standing in the wrong place, having hands or feet in the wrong place, or other careless actions can cause serious injury. Both operators should always be aware of the other's position and what he is doing. All intended movements should be carefully thought out and clearly communicated to the other operator. Anyone not familiar with this kind of heavy work or the equipment required should not be allowed to participate. Spectators should never be allowed within striking distance of the vehicle or the components.

APPENDIX E

LOAD VERSUS DEFLECTION CURVES FROM PHASE II TENSILE TESTS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PRESSURE, psi
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DEFLECTION, INCHES
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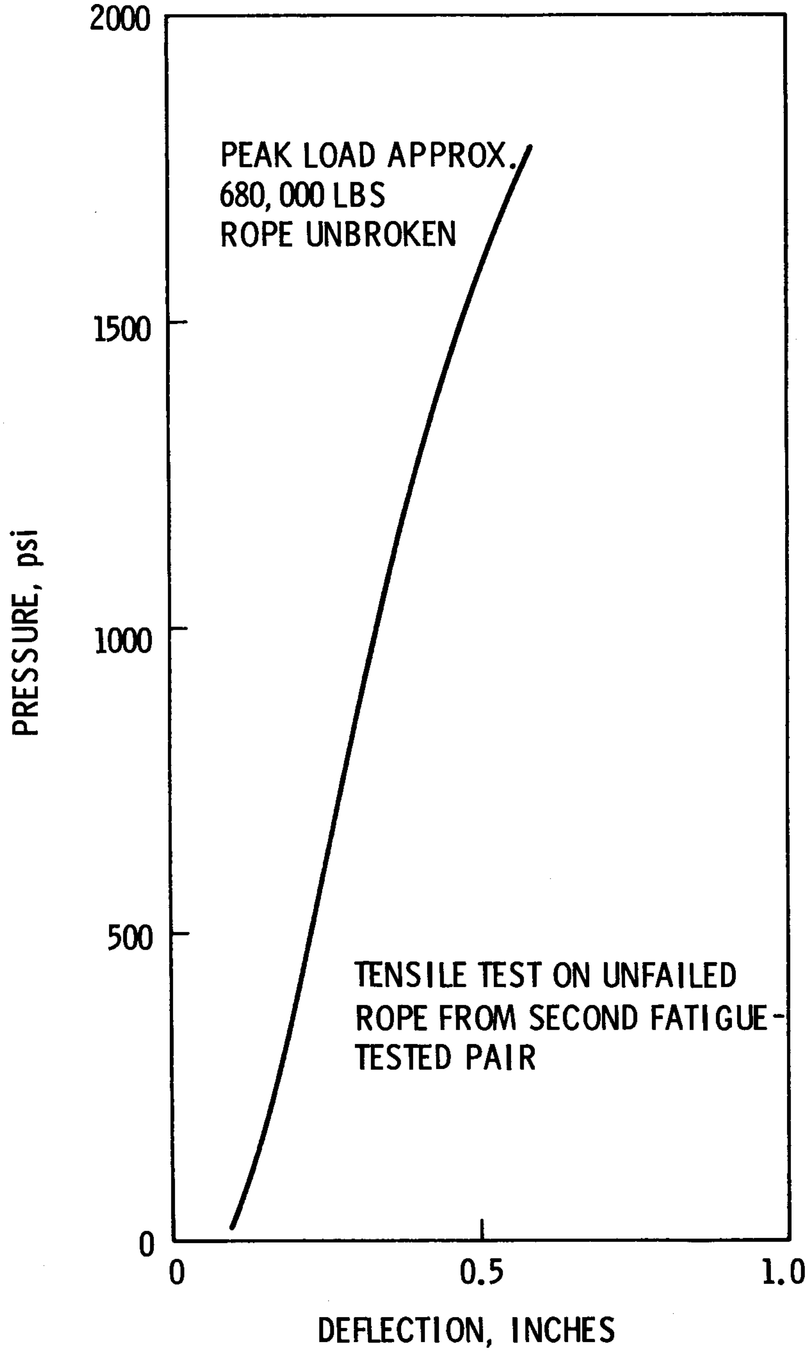
ROPE UNBROKEN
TENSILE TEST ON
UNFAILED ROPE FROM
FIRST FATIGUE-
TESTED PAIR



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PRESSURE, psi
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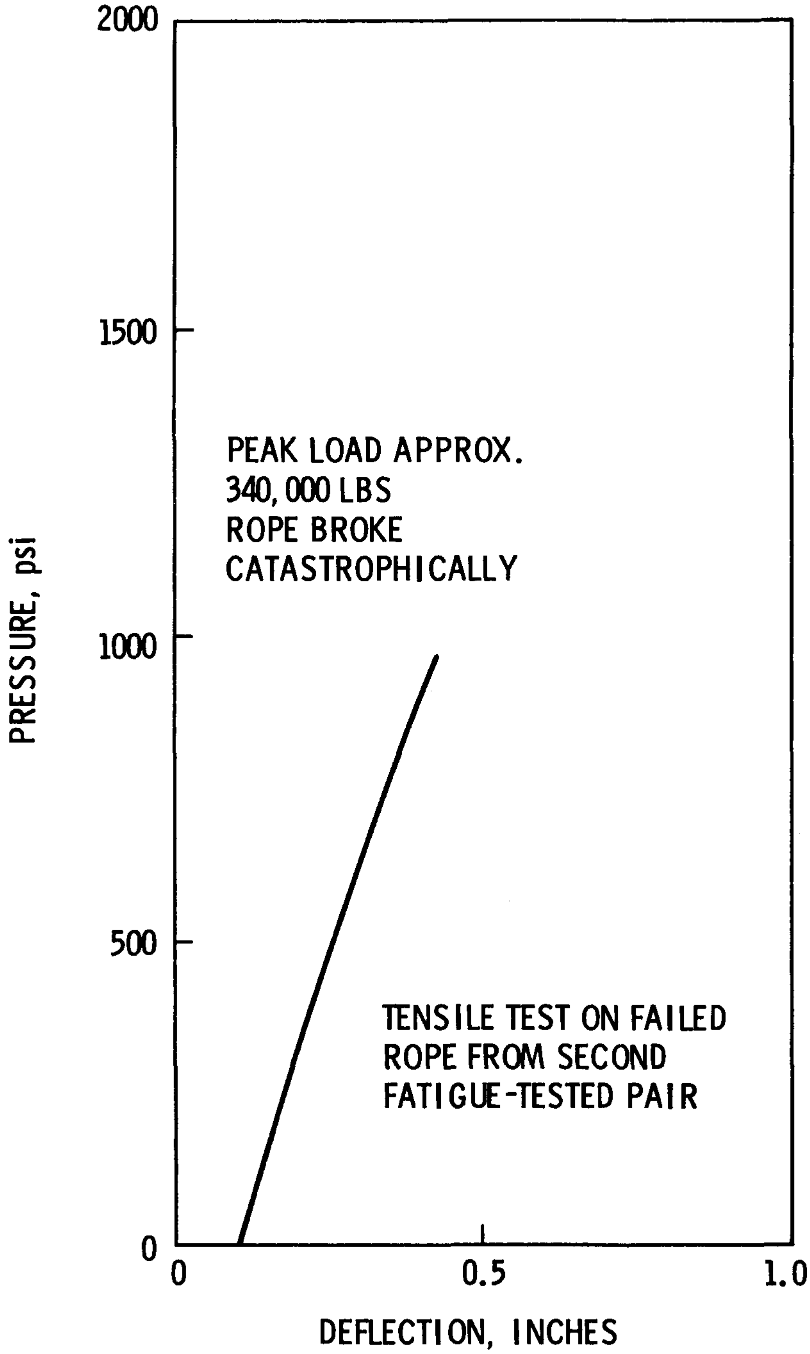
DEFLECTION, INCHES
PEAK LOAD APPROX.
680,000 LBS
ROPE UNBROKEN
TENSILE TEST ON UNFAILED
ROPE FROM SECOND FATIGUE-
TESTED PAIR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PRESSURE, psi
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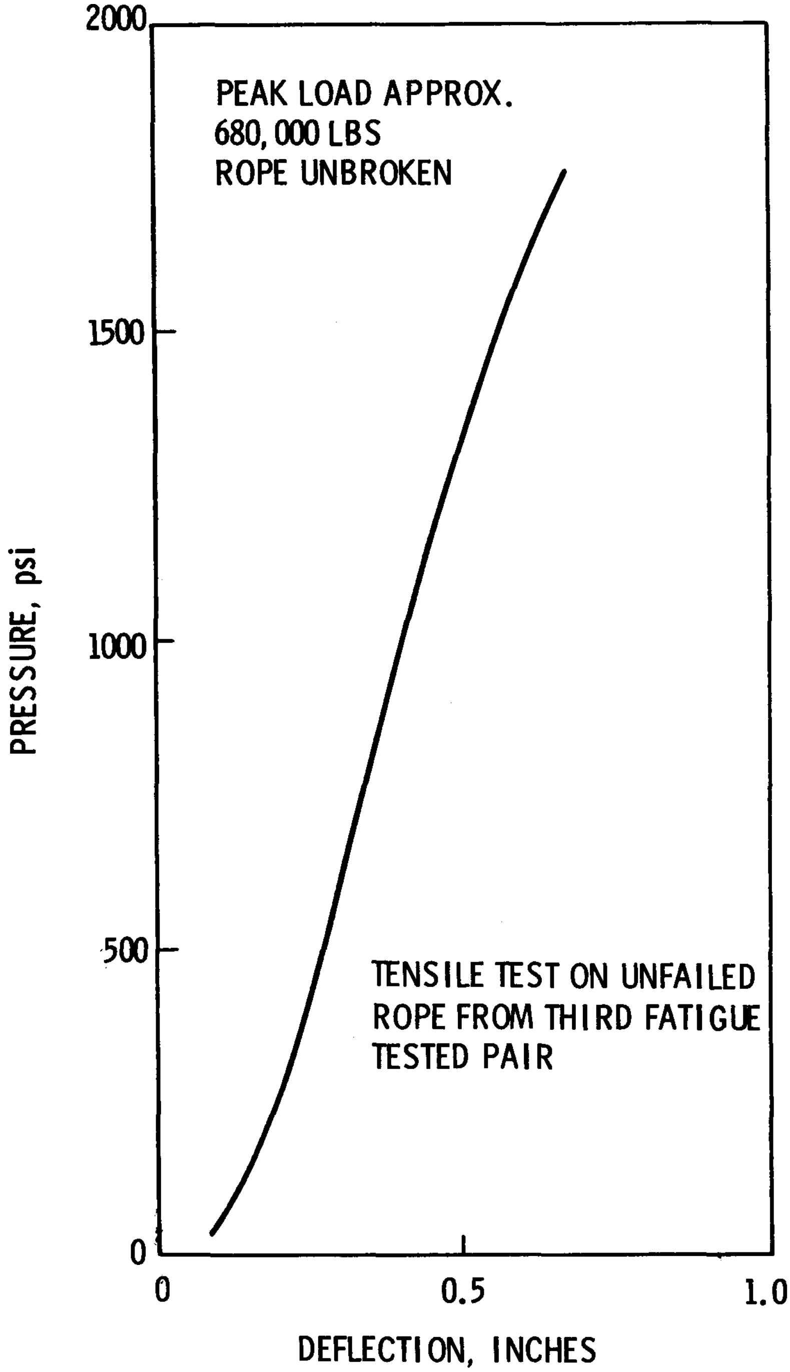
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ROPE UNBROKEN
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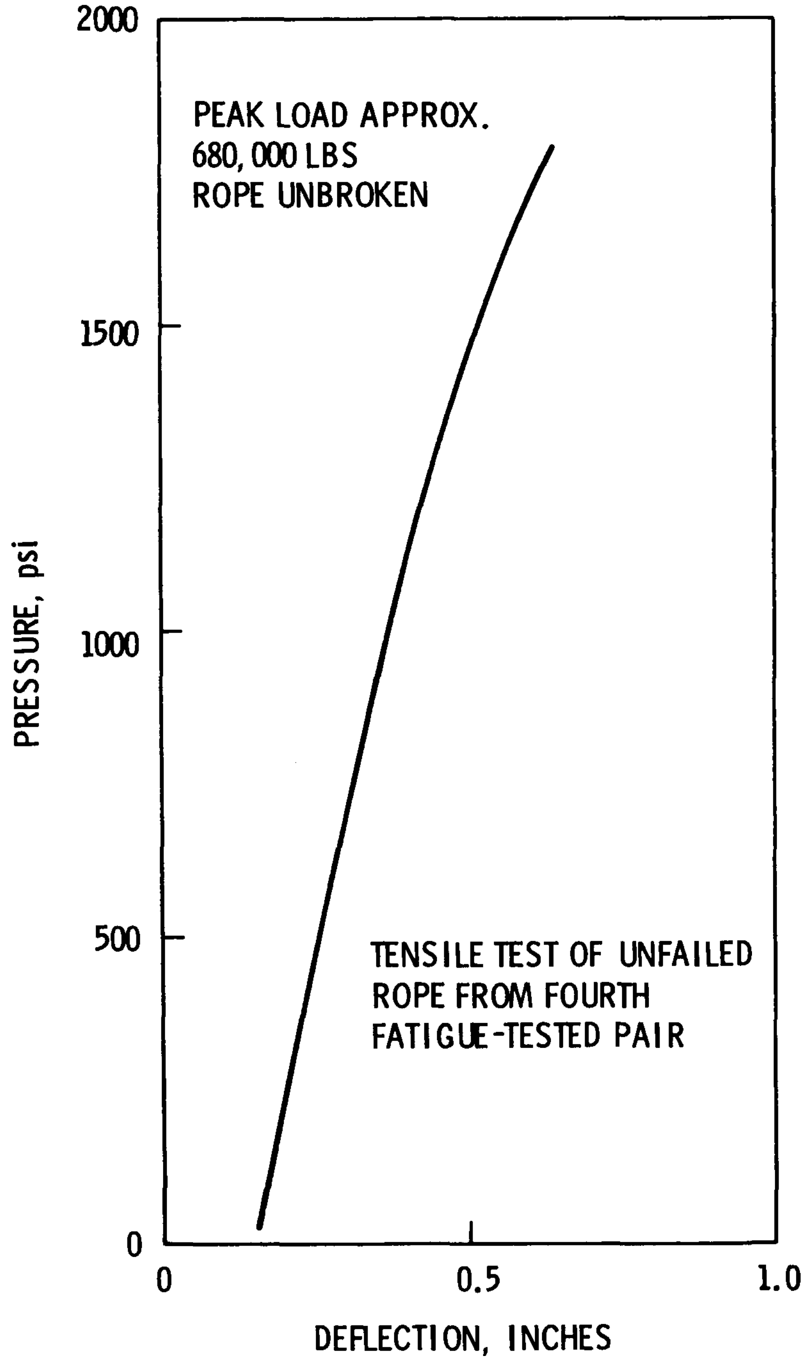
DEFLECTION, INCHES


PEAK LOAD APPROX.
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ROPE UNBROKEN
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ROPE FROM FOURTH
FATIGUE-TESTED PAIR
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PRESSURE, psi
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1.0
DEFLECTION, INCHES

APPENDIX F

CRACK DETECTION TECHNIQUES

CRACK DETECTION TECHNIQUES

During our search for a crack detecting technique applicable to individual wires of large wire rope, several methods were investigated. Of these, dye penetrant and magnetic particle seemed the only cost effective methods for the wire rope work. After initial testing, it was apparent that the magnetic particle technique offered advantages very desirable for our application.

For an effective dye penetrant test the wires must be very clean. Also, the indication caused by the developed dye penetrant is very fragile and relatively short lived. In contrast, the magnetic particle method is effective on wires that have been simply degreased, and, due to the high magnetic retentivity of the material, the defect indications on the specimen are very durable and permit repeated inspections over extended periods.

The magnetic particle method of nondestructive testing is a method for locating surface and subsurface discontinuities in ferromagnetic material. It depends for its operation on the fact that when the material or part under test is magnetized, discontinuities which lie in a direction generally transverse to the direction of the magnetic field will cause a leakage field to be formed at and above the surface of the part. The presence of this leakage field, and therefore the presence of the discontinuity, is detected by the use of finely divided ferromagnetic particles applied over the surface, some of these particles being gathered and held by the leakage field. This magnetically held collection of particles forms an outline of the discontinuity and indicates its location, size, shape and extent.

Applying this basic technique to our problem, the wires are passed through a coil and given a permanent magnetic field. Then they are sprayed with a light oil containing fluorescent magnetic particles in suspension. If a crack is present, it causes a local distortion in the magnetic field. This distortion will cause the magnetic particles to outline the crack. The crack can then be detected when the part is examined under black light. Cracks in the micron range can be detected by this method.

APPENDIX G

TEST PROCEDURES

LARGE DIAMETER WIRE ROPE FATIGUE AND TENSILE TEST

TEST PROCEDURE

1.0 TEST OBJECTIVES

1.1 Fatigue Test Data

1.1.1 Number of cycles to failure

1.1.2 Extension of rope as a function of cycles

1.1.3 Load history of rope

1.2 Tensile Test Data

1.2.1 Original length of rope

1.2.2 Elongation as a function of load

1.2.3 Maximum load obtained

1.2.4 Final length of rope

1.3 Rope Examination Data

1.3.1 Location of sample in rope

1.3.2 General appearance of rope

1.3.3 Number of broken wires

1.3.4 Number of cracks

1.3.5 Characteristics of cracks

1.3.6 General appearance of sample

2.0 TEST PROCEDURE

2.1 Socketing and Revving Ropes

2.1.1 Stretch rope and measure.

2.1.2 Wrap rope with wire both sides of cut location.

2.1.3 Cut ropes to length with cut in torch.

2.1.4 Locate socket near "deadman".

2.1.5 Tie chain around rope about 6 feet from end to be socketed.

2.1.6 Using small "cat", pull rope through socket, up to chain.

2.1.7 Retie chain onto end of rope that was pushed through the socket.

2.1.8 Pass chain around the wedge and back through the socket, until about 18 inches stick through the socket.

2.1.9 Rehook chain on the long side of the rope and pull until wedge is seated.

2.1.10 Repeat the steps 4-9 on the other end of the rope.

2.1.11 Check pin-to-pin length before final seating of the wedges. Length should be 57 1/2 ft.

2.1.12 Socket second rope using above procedure.

2.1.13 Move socketed ropes into fatigue test enclosure.

2.1.14 Using a fork lift, place one socketed end of each on top of wooden platform and hook it to load cell.

2.1.15 Place ramp and dunnage under the fatigue machine such that the cycling cylinder shafts are protected.

2.1.16 Using the cycling cylinder, drag the socket rope under the fatigue machine far enough to permit hooking it to the carriage.

2.1.17 Using "com-longs" suspend rope end from the compression member of the fatigue machine.

2.1.18 Return the cycling cylinder to the other end and repeat steps 15-17.

2.1.19 While ends are hanging under the machine, hook ropes to the carriage on the cycling cylinder.

2.1.20 Ropes are now on the machine and fatigue testing can start.

2.1.21 The tensioning cylinder is then activated and load is slowly applied not to exceed the desired test load. This loading is required to give a final seating to the socket wedge.

2.2 Constant Load Bend Over Sheave Fatigue Test

2.2.1 The desired load is established from the test requirements. Tests have been run at 30-40% of the manufacturers' rated breaking strength and others are planned for 50%. The maximum planned load in field use is about 30% of rated UTS.

2.2.2 The stroke of the cycling cylinder is ∿20 feet. This is long enough to subject part of the rope to a double bend each half cycle, onto and off the sheave.

2.2.3 The hydraulic system is started. Load is applied to the tensioning cylinder, then the cycling cylinder is started and the test proceeds. The cycle time is governed by flow rate of the hydraulic pump. The present cycle time is about 33 seconds.

2.2.4 The ropes under test are inspected at least once every two days, and generally are inspected every day. The inspection consists of observing the ropes while they are cycling, and looking for broken wires. The location of the broken wires is noted in the log book, and is sprayed with paint. When the broken wire extends far enough to cause damage to other wires, it is cut off.

2.2.5 The test will be shut down automatically for over extension of the tensioning cylinder. This is an adjustable limit switch that is usually placed about one inch from the running position. Tension cylinder overpressure will also shut the machine down. This is an adjustable trip which is set near the desired operating pressure.

2.2.6 All other trips are for protection of the hydraulic system or the fatigue machine such as cycling cylinder overtravel, fatigue machine bearing temperature, hydraulic oil level, and cycling cylinder overpressure.

2.2.7 Several measurements are made during the test. (1) The output from the load cell is recorded continuously. (2) There is a cycle counter. (3) Periodically the extension of the tension cylinder is measured and the diameter of the rope. (4) Photographs are taken for record.

2.2.8 The test continues on a 24 hour-a-day schedule until the equivalent of damage that would cause retirement in the field occurs.

2.3 Tensile Test

2.3.1 A testing machine is available for tensile testing ropes or other long slender objects. The machine can presently handle objects of pin-to-pin lengths of 63 feet to 57 feet.

2.3.2 The load is applied by a hydraulic cylinder similar to the tension cylinder on the fatigue machine. The cylinder is capable of applying a load of 935,000 lb and has a stroke of 30 inches.

2.3.3 The rope sample to be tensile tested is chosen. This can be the damaged rope if the damage is not too severe, otherwise the second rope of the pair is tested. To date when one rope failed, the second rope remained in apparent good condition.

2.3.4 The rope to be tested is moved from the fatigue machine using a fork lift. Care is taken to not damage the rope. The rope is then installed in the tensile machine using the same sockets that were used during the fatigue tests.

2.3.5 When the rope is in the tensile tester, the protective cover emergency energy absorbers are put in place.

2.3.6 The hydraulic system is activated. Control and monitoring of the test is done from within the instrument trailer. No personnel are allowed in the enclosure when the tensile load is greater than 100,000 lb.

2.3.7 A pressure transducer at the load cylinder is used to monitor the load on the rope. An extensometer is clamped to the rope. An x-y plot of load versus extension is generated for each rope tested.

2.3.8 The load is increased until the rope breaks, or when a load of 680,000 lb is reached.

2.3.9 After the tensile test is completed the rope and sockets are removed from the machine. The rope is cut off with a cutting torch and desocketed.

2.4 Rope Examination

2.4.1 Engineers examine the ropes and select locations where samples will be taken.

2.4.2 The samples are cut from the ropes with a cutting torch. Each sample is at least 1 lay length long. The samples are given a preliminary degreasing.

2.4.3 The samples are then cut to final length on an abrasive cutoff machine and disassembled in strands.

2.4.4 The samples are given a thorough degreasing, examined and the strands that are to be given a more thorough examination are chosen. These strands are now disassembled into wires.

2.4.5 Wires are examined using magnetic particle testing. Cracked wires are broken and the cracks characterized as to sizes, initiation, and cause.

2.4.6 After the broken and cracked wire examination is completed and the data are recorded, the test is complete.

APPENDIX H

WIRE TESTING

WIRE TESTING

One of the major variables on the Drucker-Tachau factor is the ultimate tensile strength of the wire. The $\frac{2T}{Dd}$ term in the Drucker-Tachau ratio represents a nominal bearing stress between the rope and the sheave. The ultimate tensile strength of the wire is the value they decided on to use to relate ropes of varying strength wires. They indicate that a value for the endurance limit might give better results but that data are not readily available.

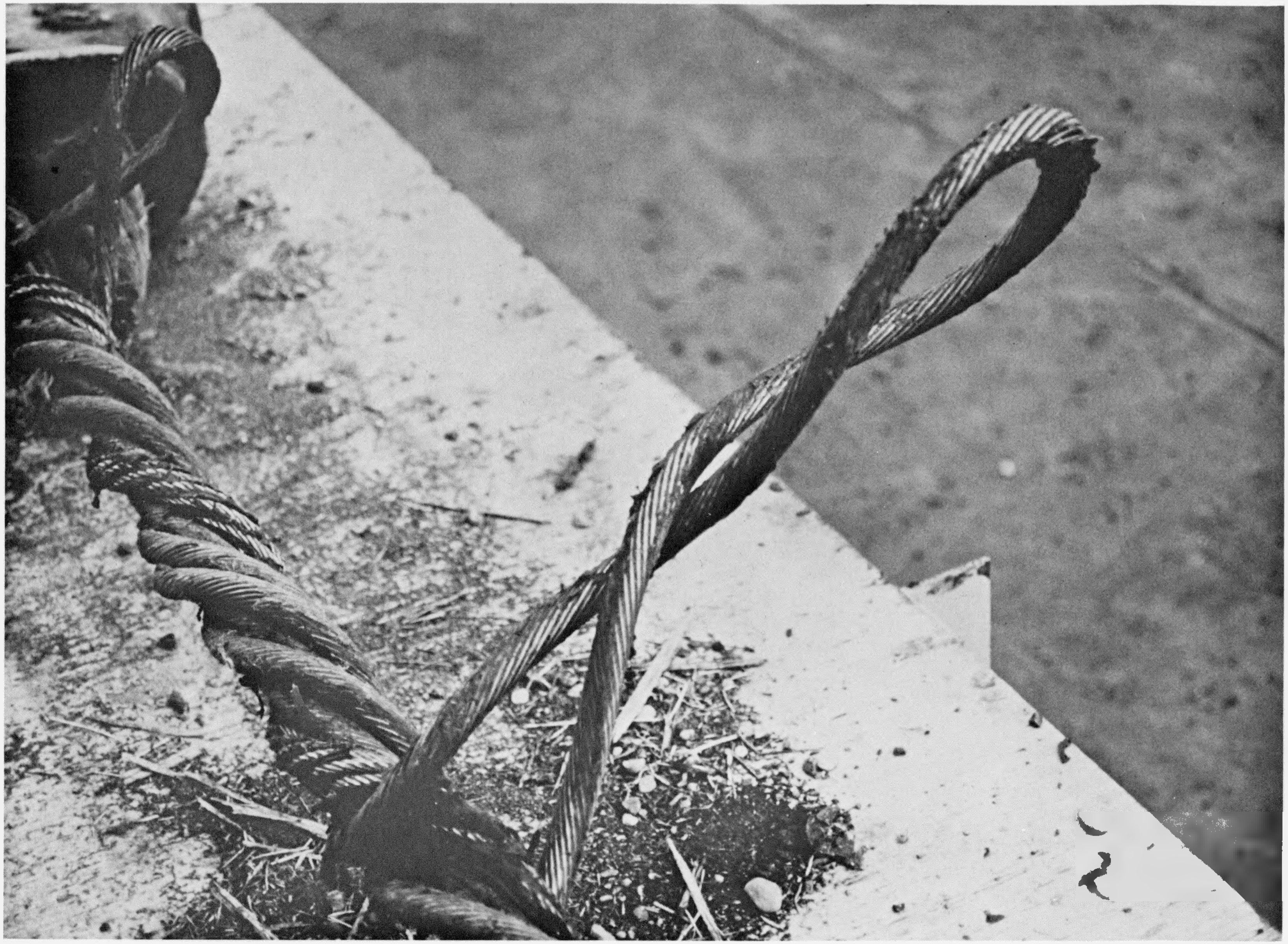
Three wires from each manufacturer were tested to failure in tension to determine their UTS. Results of this testing are shown below.

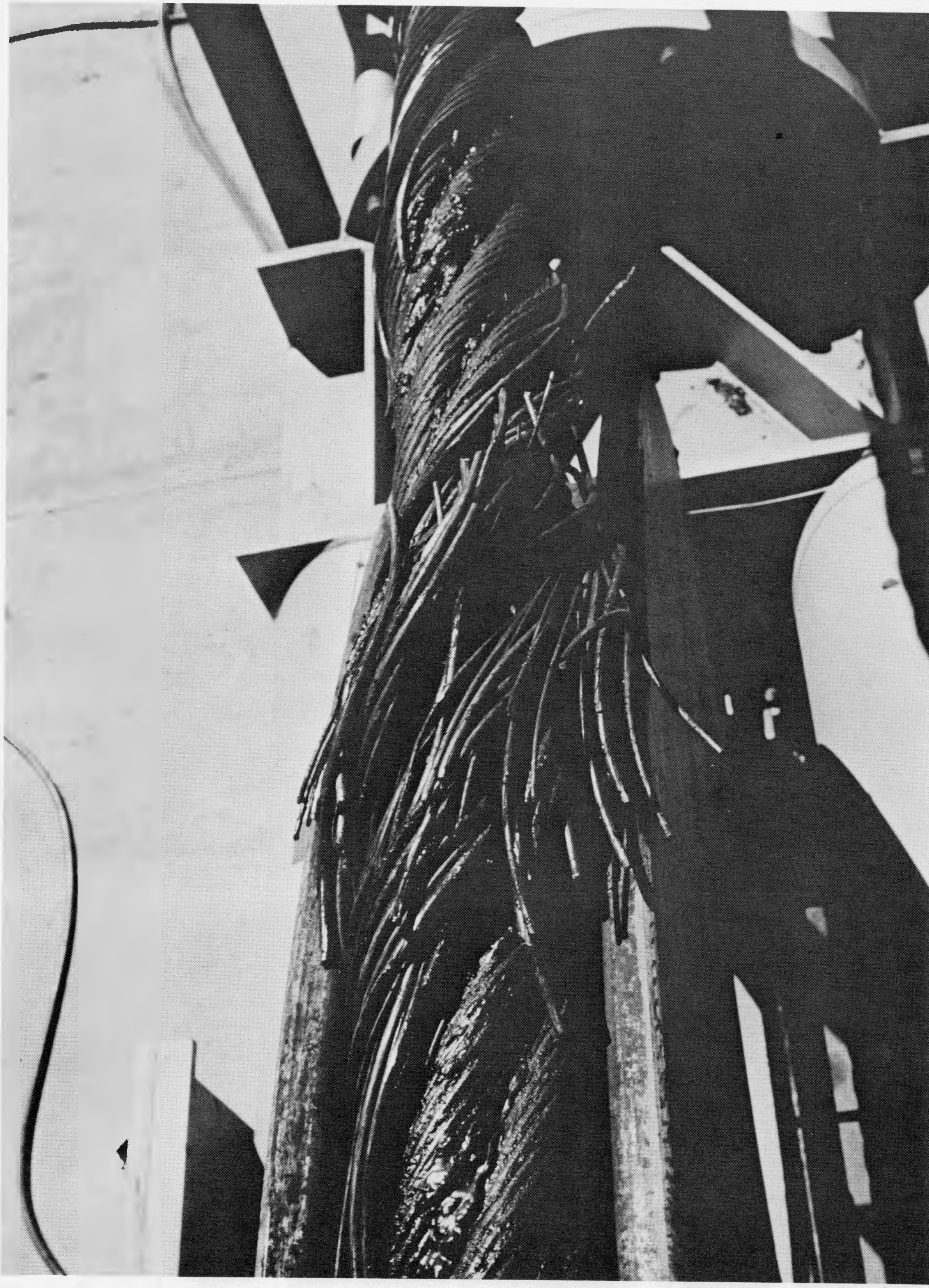
Manufacturer	UTS (Ksi)	Average
B	260.5	
B	258.9	259.6
B	259.4	
D	228.8	
D	229.1	228.9
D	228.8	
B'	241.9	
B'	239.2	239.7
B'	238.0	

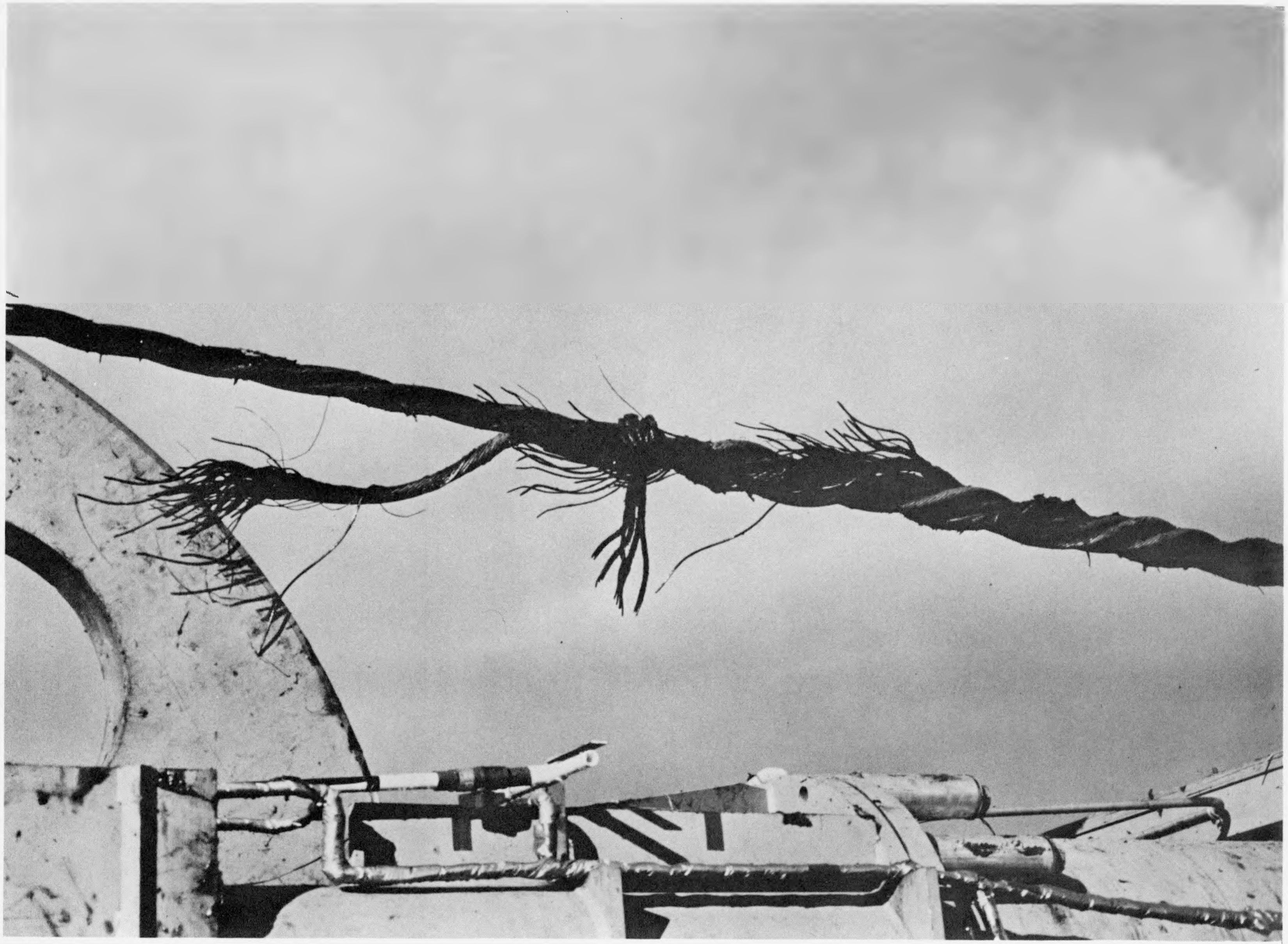
All wires necked significantly before breaking. All breaks were out of the grips so that bias from this did not occur.

APPENDIX I

PHOTOGRAPHS OF TEST ROPE FAILURES







APPENDIX J

LITERATURE REFERENCE RELATED TO THE SERVICE LIFE OF WIRE ROPE (WITH PARTICULAR EMPHASIS ON LARGE WIRE ROPES USED IN SURFACE MINING)

LITERATURE REFERENCE RELATED TO THE SERVICE LIFE OF WIRE ROPE (WITH PARTICULAR EMPHASIS ON LARGE WIRE ROPES USED IN SURFACE MINING)

A vast number of references deal with various aspects of wire ropes and cable. Some describe specific applications or uses of wire rope; most deal with one or several aspects of the overall design, construction, usage or maintenance of wire rope. To make this literature survey as useful as possible, groups of references for some of the primary areas or interests related to wire rope that have some bearing on the specific application of large wire ropes for surface mining have been identified. The list presented is by no means exhaustive but it does include most of the known useful references.

With only a few exceptions, all references are more recent than 1950; many were published after 1965. Some references present information on several aspects of wire rope--those reports are included under the category which identifies the primary consideration within the document or article. In a number of cases references were purposely omitted if they appeared to present information redundant to that presented in other cited references.

References are listed for these conditions or parameters:

- Rope Materials and Constructions
  - Wire Materials
  - Coatings
  - Constructions
  - Core Material
  - Lubrication
  - Terminations
- System Conditions
  - Abrasion and Wear
  - Bending Fatigue
  - Corrosion

Load and D/d Ratio

Internal Friction and Flexibility

Sheave Design

Stress Analysis

Nondestructive Inspection

Wire Rope Design Criteria

Maintenance

Wire Rope Handbooks (dealing with large-diameter rope)

Vibration and Dynamics

Dragline and Haulage Systems

Running, Rigging, Requirements, and Testing

Underground Mining (for rope uses of relevance to surface mining)

WIRE MATERIALS

Anderson, W., "Factors Affecting the Efficiency and Safety of Steel Wire Ropes", Lift, pp 109-113, (May 1972).

Armstrong, B. M., "Effect of Ausforming on the Properties of Carbon Steel Wire", Journal of the Iron and Steel Institute, pp 655-659 (August 1964).

Dull, R. G., "Rope Wire for Today's Applications", Wire and Wire Products, Vol 41, pp 1642-1643 (October 1966).

Friedman, H. H., "Clad Metal Wire Outperforms Solid Wire", Material in Design Engineering, Vol 65, No 2, pp 66-67 (February 1967).

Godfrey, H. J., "The Fatigue and Bending Properties of Cold Drawn Steel Wire", ASME Trans, Vol 29, (March 1941).

Godfrey, H. J., "The Strengthening of Steel by Wire Drawing", Wire and Wire Products, Vol 38, No 1, pp 51-60, 129-130 (January 1963).

Kenyon, J. N., "Fatigue Properties of Steel (Rope Wire)", Materials Research and Standards, Vol 2, No 7, pp 553-556 (July 1962).

Lowe, R., and Hanford, A. G., "Processing and Applications of Stainless Steel Wire", Wire and Wire Products, Vol 42, pp 1473-1478, 1554 (September 1967).

Nakamura, Hiroshi, "High Tension Steel Wire of Low Carbon Content", Bulletin of JSME, Vol 3, No 11, pp 305-311 (1950).

Tingle, M. I., "Recent Developments and Current Practices in the Manufacture and Production of High-Carbon Steel Wire in South Africa", The South African Mechanical Engineer, Vol 16, No 1, pp 219-237 (June 1967).

Vitelli, V. J., "The Effect of Heat Treatment and Cold Work on the Mechanical Properties of High Carbon Steel Wire", Wire, pp 1762-1766 (December 1962).

----, "Fatigue Evaluation of Aluminum Coated Improved Plow Steel Wire Rope", Lab Project 9300-44, Technical Memorandum No. 5, SF-020-01-02, W. U. 3406, Naval Applied Science Lab (February 28, 1967)

----, "Stainless Steel Wire: Where It Is Today", Wire and Wire Products, Vol 42, pp 1483-1487 (September 1967).

COATINGS

Cordiano, H. V., Wolfe, R. J., "Investigation of the Resistance to Fatigue of Uncoated Hot Dip Galvanized and Drawn Galvanized Wire Rope Preformed and Nonpreformed Manufactured by the American Chain and Cable Co.", Report No. 414017 to Naval Applired Science Lab, (June 1953).

Dingley, W., "Laboratory Development of Corrosion Inhibiting Coatings for Mine Hoise Rope", Paper presented Eastern Division, Canadian Region, National Association of Corrosion Engineers, Montreal, November 14-16, 1960.

Macco, J. G., "Investigation of the Resistance to Fatigue and Wear of Aluminum Coated and Galvanized Improved Plow Steel Wire Rope", Report No. NASL-930-44-PR-2, Naval Applied Science Lab, (January 1963).

Rigo, Jane H., "Laboratory and Service Evaluation of Aluminized Steel Wire", Corrosion-National Association of Corrosion Engineers, Vol 17, pp 245-250 (May 1961).

"Research Program to Determine Fatigue Properties of Wire Rope Having Individually Coated Wires", Final Report, Contract N156-47208, Bureau of Engineering Research, University of Alabama, 30 October 1966.

----, "Report of Investigation of the Resistance to Fatigue of Uncoated, Hot Dippled Galvanized and and Drawn Galvanized Wire Rope, Preformed and Non-preformed", Lab Project 4750-17, Final Report NS 653-016, Naval Applied Science Lab (June 1953).

----, "Report of Investigation of the Resistance to Fatigue of Wire Rope Zinc Coated and Black, Preformed and Non-Preformed, Lab Project 4750-16, Final Report NS653-016, Naval Applied Science Lab (January 1959).

CONSTRUCTIONS

Hitchen, H., Characteristics and Applications of Round, Flattened, and Locked Strand Rope", The Mining Society of Nova Scotia--Transactions, Vol LX, pp 167-175 (1957)

Kastner, J., "The Importance of Shape of Wire in Wire Rope Design", DRAHT, Vol 13, No 9, pp 517-524 (September 1962) and Vol 13, No 10, pp 582-590 (October 1962).

----, "AL-U-FLEX Wire Rope", Bulletin No. 2, American Chain and Cable Co., Inc. (June 1964)

----, "Comparative Evaluation of Replenishment-at-Sea Span Wire Ropes of Foreign and Domestic Manufacture", Lab Project 6738-1, Final Report, Naval Applied Science Lab (September 1962)

----, "Compacted Strand Cable Ups Strength and Flexibility", Wire and Wire Products, Vol 44, p 69 (january 1969).

----, "Investigation of the Effects of Core and S-rand Construction on the Endurance Life of Wire Rope", Lab Project 6402-3, Technical Memorandum, Final Report SR 007-01-01, Task 3406, Naval Applied Science Lab (September 24, 1964).

----, "Investigation of the Resistance to Fatigue of Various Types and Constructions of Wire Rope - Bethlehem Steel Company, Inc., NS653-016, Lab Project 4750-16, Naval Applied Science Lab (July 1948)

----, "Newly Designed Steel Cable Has Increased Strength", Wire Journal, pp 48, 49 (February 1969).

----, "A Report on the Comparison of Bending Fatigue Properties and Physical Characteristics of Wire Rope with Polypropylene vs. Sisal Cores," Project No 2 Report, American Chain and Cable Co. (January, 1967)

----, "Evaluation of a Wire Rope with a Plastic Core", Lab Project 4750-23, Final Report, NSM 653-016, Naval Applied Science Lab (July 16, 1957).

----, "Evaluation of Wire Rope with a Spring Core", Lab Project 4750-24, Final Report, NSM 653-016, Naval Applied Science Lab (August 9, 1957)

----, "Fatigue Life of IWRC Wire Rope", Lab Project 9300-54, Technical Memorandum No 4, SR 101-03-20, W. U. 8366, Naval Applied Science Lab (March, 1967)

----, "Research Report on the Effects of Core and Strand Construction on the Fatigue and Wear Resistance of Wire Rope", Lab Project 4750-12, 4750-14, Final Report NS 653-016 Naval Applied Science Lab (May, 1957)

----, "Specification for Fiber Cores for Wire Ropes", British Standard 525:1962, British Standards Institution.

----, "The Effect of the Use of a Metallic Core in Place of a Fiber Core on the Fatigue and Wear Resistance of Wire Ropes of Various Constructions", Progress Report No 3, NS 653-016, Naval Applied Science Lab (January 6, 1954).

LUBRICATION

Kasten, R. P., "Proper Lubrication Means Longer Rope Life", Lubrication Engineering, 20, pp 180-183, (May 1964).

Kasten, R. O., "Wire Rope Lubrication--Its Relation to Rope Life and Safety", Wire, pp 1271-1273 (September 1963).

Lex, W. I., "Tests on this Wire Rope Fatigue Machine Proved that Lubricants Do Penetrate Wire Rope", Eng. M. N. J., 155, pp 96-98 (June, 1954)

Reynolds, E. S., "Proper Lubrication of Wire Rope Pays", Pit and Quarry, Vol 56, pp 112-115 (September 1963)

Rhodes, R., and Turner, F. H., "Proper Lubrication of Wire Rope Pays", Pit and Quarry, Vol 56, No 9, pp 112-115 (September 1963).

White, E. L., Miller, P. D., and Peoples, R. S. "Antigalling Coatings and Lubricants for Titanium", Titanium Metallurgical Laboratory, Battelle Memorial Institute, Columbus, Ohio TML Report No. 34 (February 24, 1956).

----, "Lubrication - Wire Ropes", Mining Journal, Vol 268, p 299 (April 1967)

----, "Wire Rope", Lubrication, 53 (6), pp 67-74 (June 1967).

TERMINATIONS

Cordiano, H. V., and Wolfe, R. J., "Report on Investigation of 'Fiege' Wire Rope Connectors", Lab Project 5366-2, Naval Applied Science Lab (February 1953).

Delmonte, J., "Epocast Terminated Wire Cables (and Roper)", Wire and Wire Products, Vol 34, No 9, pp 1092-1093 (September 1959).

Smolhinger, C. W., and Cornish, J. B., "Investigation of Variables in the Zinc Socketing of Wire Rope and Strand", Wire, pp 1677 - 1679, 1725-1727 (December 1961).

Swaging Instructions, ESCO Corporation, Portland, Oregon, (March 1965)

----, "Correct Socketing", Pit and Quarry, Vol 59, pp 104-105 (June 1967).

----, "Correct Socketing Prevents Premature Rope Failure", Engineering and Mining Journal, Vol 168, pp 126 (November 1967)

----, "How to Splice Wire Rope Eyes", Coal Age, Vol 72, pp 172 (ctober 1967)

----, "Method of Socketing - Round Strand Rope", British Ropes Limited, London, Publication 112 (no date).

----, "Recent Developments in Ferrule Connections for Steel Wire Ropes", Wire, Vol 74, pp 247-248 (December 1964).

----, "Report of Investigation of Various Factors Affecting the Efficiency of Wire Rope Connections Made with Wire Rope Clips", Lab Project 4750-11, Naval Applied Science Lab (August 1949)

----, "Socketing", British Ropes Limited, London (no date).

----, "Sound Splicing is Key to Long Wire Rope Life", Modern Materials Handling, Vol 19, pp 64 (November 1964)

----, "Wire Rope and Fittings", coal Age, Vol 68, No 4, pp 101-116 (April 1963)

ABRASION AND WEAR

Franke, E. A., "Inspection of Wire Ropes Used in Lifting and Haulage Equipment", Industrie-Anzeiger, Vol 82, pp 23-25 (October 1960).

Kaplan, S. M., "Wire and Cable Abrasion Testing Machines", Report No. FA-TN-1014, Frankfort Arsenal, Philadelphia, PA (January 1953).

Nishioka, T., "Surface Conditions and Fatigue of Wire Rope", Wire World International, Vol 8, No 3, pp 67-73 (May 1966).

Schneider, A., and Sztankoczy, E., "Wear and Quality Charactertics of Steel Wires in a Systematically Reversed Pulley Rope with a Low Safety Factor", Wire World International, Vol 14, pp 245-250 (November 1972).

----, "Report of Investigation of the Resistance to Fatigue and Wear of Wire Ropes with Various Strand Constructions", Lab Project 4750-4 Prog. Report 1, NS653-016, Naval Applied Science Lab, (October 2, 1953)

(See also "Determination of the Effect of Various Parameters on Wear and Fatigue of Wire Rope Used in Navy Rigging Systems")

Colderale, P. M., "Investigation of the (Fatigue) Strength of Wire Rope", DRAHT, Vol 11, No 8, pp 407-415 (August 1960).

Gibson, P. T., et al, "Analytical and Experimental Investigation of Aircraft Arresting Gear Purchase Cable", Final Report from Battelle's Columbus Laboratories to Naval Air Engineering Center under Contract No. N156-47939 (July, 1967) Distribution limited.

Kahan, D. S., and Jubb, W., "New Fatigue Machine Developed to Improve Testing of High Carbon Spring Wires," Wire and Wire Products, Vol 40, No 1, January 1965, pp 96, 97, 142 and 143.

Kenyon, J. N., "Fatigue Properties of Steel Rope Wire", Mater, Res. and Stand. 2, 7 (July, 1962), pp 553-556.

Kenyon, "Fatigue Properties of Wire Rope", Materials Research and Standards (July 1962).

Kenyon, J. N., "The Rotating Wire Arc Fatigue Machine for Testing Small Diameter Wire," Proceedings of the American Society for Testing Materials, Vol 35, Part 2, 1935, pp 156-164.

Miller, Hugo, "The Properties of Wire Rope Under Alternating Stresses", Wire World International, Vol 3, No 5 , (October 1961)

Rossette, U., "Fatigue Strength of Wire Ropes", (In Italian) Ingeghere Vol 27, No 7, pp 769-771, (July 1953)

Shelton, S. M., and Swanger, W. H., "Tests of Galvanized Wire Under Pulsating Tensile Stress", ASTM Proceedings, Vol 33, Part II (1933).

Sherwood, P. W., "How to Stretch the Life of Your Wire Rope", Rock Products, Vol 69, p 125 (January 1966).

Van DeMoortel, D. G., "A New Method for the Fatigue Tests on Steel Wires and Wire Ropes and the New Viewpoint Opened by the Results Obtained", Wire and Wire Products, Vol 34, No 10, pp 1317-1321, 1324-1325 (October 1959)

"Accelerated Fatigue Tests of Wire Rope", Final Report, Contract N-156-47208, Bureau of Engineering Research, University of Alabama, 9 March 1968.

Bulletin 10, International Organization for the Study of the Endurance of Wire Rope (O.I.P.E.E.C.), 20 Rue D'Athenes, Paris 9E, France (July 30, 1967).

Bulletin 19, same as above (1971)

----, "Investigation of the Resistance to Fatigue of Various Types and Constructions of Wire Rope", Lab Project 4750-12, Progress Report 1, Naval Applied Science Lab (May 25, 1953).

CORROSION

Hoover, H. M., "Wire Rope", Geomarine Technoloyg, Vol 1, No 3, pp 32-33 (February 1965)

Rigo, J. H., "Seawater Tests Determine Corrosion Resistance of Stranded Steel Wire", Material Protection, Vol 1, No 7, pp 28-37 (July 1962)

Sproles, E. S., and Hoke, J. K., "Zinc or Aluminum Coating - Which Gives Wire Rope the Best Corrosion Fatigue Resistance", Wire and Wire Products, Vol 43, No 9, pp 150-152, 218 (October 1968)

REMAINING STRENGTH

Davidson, W., "Investigation and Calculation of the Remaining Tensile Strength in Wire Ropes with Broken Wires", ACTA POLYTECHNICA- Mechanical Engineering Series, Vol 3, No 6, (1955)

(See also <u>Roebling Wire Rope Handbook</u>)

(See also "Determination of the Effect of Various Parameters on Wear and Fatigue of Wire Rope Used in Navy Rigging Systems")

LOAD AND D/d RATIO

Gibson, P. T. and Cress, H. A., "Analytical and Experimental Investigation of Aircraft Arresting Gear Purchase Cable", Final Report from Battelle's Columbus Laboratories to the Naval Air Engineering Center, (November 1968)

Gibson, P. T., et al, "A Study of Parameters that Influence Wire Rope Life", Final Report prepared by Battelle Columbus Laboratories for the Naval Ship Systems Command under Contract No N00024-72-C-5427 (October, 1974)

Gibson, P. T., et al, "Determination of the Effect of Various Parameters on Wear and Fatigue of Wire Rope Used in Navy Rigging Systems", Final Report prepared by Battelle's Columbus Laboratories for the Naval Ship Research and Development Center under Contract No. N00600-70-C-1045 (March 1972).

Muller, Hugo, "The Properties of Wire Rope Under Alternating Stresses", Wire World International, p 249 (October 1961)

----, "Comparative Wire Rope Performance and Effect of Shear Diameter on Fatigue Life", USNAEL Code 80020, NAEL 7279 (April 13, 1965)

INTERNAL FRICTION AND FLEXIBILITY

Hind, E. C., "Wire Rope Transmission Losses", Proc. Inst. of Mechanical Engineering, Vol 178, Pt 1, No 30 (1963-1964)

Macco, J. G., "Report of Investigation of the Pliability of Wire Rope as Indicated by the Energy Required to Drive the Rope over a Sheave", Lab Project 6402-4, Final Report, SR-007-01-01, Task No. 3406, Naval Applied Science Lab, (September 1963)

----, "Investigation of the Pliability of Wire Rope as Indicated by the Force to Drive the Rope over a Sheave", Lab Project 4750-8, Final Report, NS 653-016, Naval Applied Science Lab, (July 1957)

SHEAVE DESIGN

Cassell, L. D., "Elastomeric Sheave Liners", Wire, pp 20-21, 279-280, (February 1963)

Czitary, E., "Behavior of a Cable on a Pulley with Electrically Deformable Lining" (In German), Ost. Ing. Arch. 10, 4, 349-359 (1956)

Czyzewski, M., "The Friction Coefficient Between Rope and Sheave in a Lift", (In Polish), Archiwum Budowy Maszyn 13, 1, 31-70 (1966).

Skillman, Edward, "Some Tests of Steel Wire Rope on Sheaves", U. S. Bureau of Standards, Technologic Paper Vol 17, No 229 (March 2, 1923), pp 227-243.

Sprangler, F. L., "Drum and Sheave Design", Product Engineering pp 13-15, (January 1940)

(See also "Factors Affecting the Efficiency of Steel Wire Rope")

(See also VDI Guidelines for Lifting and Hauling)

Bert, Charles W. and Stein, Robert A., "Stress Analysis of a Wire Rope in Tension and Torsion", Journal of Engineering for Industry (May 1962), p 621

Bullen, N. I., "Investigation of Tension in Running Wire Ropes", R.A.E. Tech. Memo. N.A. 17

Carstarphen, F. C., "Effect of Bending on Wire Rope", A.S.C.E., Vol 98, (1933), pp 562-621.

Chapman, R. W., "The Stress of Wire Rope Due to Bending", Engineering Reviews, October 1968.

Diratsu, V. S., "Determination of Stresses in the Strands of Spirally Wound Steel Cables by the Electromagnetic Method", (in Russian), Nauch. Zap. Odessk. Politekn, in-ta 1, (1953), 63 - 82, Ref. Zh. Mekh. No 12 (1956), Rev. 8689

Gibson, P. T. and Cress, H. A., "Analytical Study of Aircraft Arresting Gear Cable Design", Report from Battelle's Columbus Laboratories to U. S. Navy Bureau of Weapons, (May 1965)

Gibson, P. T. et al, "The Continuation of Analytical and Experimental Investigation of Aircraft Arresting Gear Purchase Cable", Final Phase Report from Battelle's Columbus Laboratories to Naval Air Engineering Center under Contract No N156-69-C1501, (April 1970)

Gibson, P. T., et al, "Torsional Properties of Wire Rope", American Society of Mechanical Engineers, Paper 69-DE-34 (May 1969)

Glushko, M. F., "mechanical Testing of Wire Rope", ZAVODSKAYA LABORATORIYA, Vol 28, No 8, pp 981-983 (August 1962)

Griffith, J. H. and Bragg, J. D., "Strength and Other Properties of Wire Rope", Proc. of ASTM, vol 32, part 2 (1932), pp 398-412

Heilig, R., "Statics of Heavy Ropes", (in German), Stahlbau 23, 11, E 12 a (October 1955)

Heller, S. R., "The Contact Pressure Between Rope and Sheave", Naval Engineers Journal, pp 49-57 (February 1970)

Howe, James F., "Determination of Stress in Wire Rope as Applied to Modern Engineering Problems", ASME Trans (1918)

Hruska, F. H., "Calculation of Stress in Wire Rope", Wire and Wire Products, Vol 26, No 9, pp 766-767, 799-801 (September 1951)

Hruska, F. H., "Radial Forces in Wire Rope", Wire and Wire Products, Vol 27, No 5, pp 459-463 (May 1952)

Hruska, F. H., "Tangential Forces in Wire Rope", Wire and Wire Products, Vol 28, No 5, pp 455-460 (May 1953)

Ikada, A., and Ueno, I., "Equivalent Bending and Torsional Rigidity of Wire Rope When It Is Bent Along a Sheave", Proc. 1st Japan National Congress on Applied Mechanics, 1951 National Committee for Theoretical and Applied Mechanics, pp 198-208 (May 1952)

Kahan and Jubb, "Fatigue Properties and Stress Analysis (of Wire Rope)", Wire (November 1966)

Leissa, A. W., "Contact Stresses in Wire Ropes", Wire and Wire Products, Vol 34, No 3, pp 307-314, 372-373 (March 1959)

Otto, P., "Rope Creep on Sheaves", (in Dutch) Ingenieur, Vol 65, No 31, pp 138-146 (July 1953)

Starkey, W. L., and Cress, H. A., "An Analysis of Critical Stress and Mode of Failure of Wire Rope", ASME Annual Meeting, Paper No 58-A-63 (November 1958)

Thomas, H. R. and Hoersh, V. A., "Stresses Due to the Pressure of One Elastic Solid on Another", Bulletin 212, Engineering Experimentation Station, (June 1930), University of Illinois.

Vanderweldt, H. et al, "Mechanical Behavior of Stranded Wire Ropes", Report 69-4, Themis Program No 983, Catholic University of America (july 1969)

Wyatt, T. A., "Secondary Stresses in Parallel Wire Suspension Cables", Proc. American Society of Civil Engineers, 86, ST 7 (J. Structural Div.) July 1960, pp 37-59

Zhdanoy, G. P., "Bending Stress in Steel Wire Cable (Spiral and Double Twist)", (in Russian), Nauch. tr. Khar. Kovsh. gorn. in-ta 2, (1955), pp 129-135, Ref. Zh. Mekh 1956, Rev. 5553

NONDESTRUCTIVE INSPECTION

Higginson and Butler, "Non-Destructive Examination of Steel Wire Ropes", National Coal Board, Production Department, Submitted to Safety in Mines Research Advisory Board (March 1969)

----, "Electromagnetic Rope Tests in Ontario, May 1961 to August, 1962", Progress Report to Ontario Mining Association by McPhar Geophysics Limited, 93 pages (1962)

----, "How Wire Rope Tester Promotes Safety - Increases Service Life", Metal Mining and Processing (July 1964)

WIRE ROPE DESIGN CRITERIA

Calderale, P. M., "Programming and Statistical Analysis of Fatigue Tests on Wire Rope with the Aid of Least Squares", Wire, Issue 66, pp 131-138, (August 1965)

Drucker, D. S., and Tachau, H., "A New Design Criterion for Wire Rope", Journal of Applied Mechanics, Trans., ASME 12 (1945)

MAINTENANCE

Anderson, W., "The Servicing of Wire Rope", The B.H.P. Technical Bulletin No 24

Carlson, B. N., "Extending Wire Rope Life", Mining Congress Journal, Vol 43, No 7, pp 60-63 (July 1957)

Chase, L., "Why Wire Rope Breaks and How to Prevent It", Ocean Industry, Vol 102, pp 98-100 (May 1969)

Kasten, R. O., "Inspection of Wire Rope Boom Supports for Operating Safety", Wire and Wire Products, Vol 39, pp 1179-1180 (August 1964).

Layton, C. H., "Proper Maintenance Will Make Our Wire Rope Last Longer", The Oil and Gas Journal, Vol 66, No 21, pp 76-80 (July 29, 1965)

Lex, W. I., "How to Inspect Wire Rope", Modern Concrete, pp 38-39 (July 1965)

Lex, W. I., "Operation, Maintenance and Inspection of Wire Rope", U. S. Coast Guard, Proceedings of the Merchant Marine Council, pp 91-96 (May 1968)

Meger, A. F., "Wire Rope Maintenance", Mining Congress Journal, pp 34-37 (July 1960)

Sherwood, P. W., "Longer Lives for Wire Ropes", The British Steelmaker, Vol 31, No 1, pp 24-25 (January 1965)

Care and Abuse of Wire Rope, Independent Wire Rope Manufacturers Association, Silver Springs, Maryland (Undated)

Care and Use of Wire Rope, First Edition, Vocational Training Courses, Petroleum Industry Series, Issued by the University of Oklahoma Business and Industrial Services and State Board for Vocational Education. Order from University Book Exchange, University of Oklahoma, Norman, Oklahoma (1954)

Here's How to Keep Your Wire Rope Working, Hazard Wire Rope Division, American Chain and Cable, Inc., Wilkes-Barre, Pennsylvania (Undated)

Longer Life From Your Wire Rope, United States Steel, Pittsburgh, Pennsylvania (Undated)

Tuffy Tips on Abuses and Correct Uses of Wire Ropes and Slings, Union Wire Rope, Armco Steel Corporation, Middletown, Ohio (1963)

----, "Use and Care of Wire Rope", The Timberman, Vol 61, p 77 (1960)

WIRE ROPE HANDBOOKS (DEALING WITH LARGE DIAMETER WIRE ROPE)

Bethlehem Wire Rope for General Purposes, Catalog 1877, Bethlehem Steel Company, Bethlehem, Pennsylvania (Undated)

MacWhyte Wire Rope, Catalog G-17, MacWhyte Wire Rope Company, Kenosha, Wisconsin (1974)

Roebling Wire Rope Handbook, The Colorado Fuel and Iron Corporation, Roebling Wire Rope, Trenton, New Jersey (1966) (Out of Print)

Selection of Wire Rope for Engineering Applications, British Ropes Group, Wire Rope Technical Department, Carr Hill, England (1967)

USS Tiger Brand Wire Rope Engineering Handbook, United States Steel, Pittsburgh, Pennsylvania (1968)

Wire Rope Catalogue, WRI Publication 681, Wire Rope Industries of Canada Limited, Lachine, Quebec, Canada (1974)

Wire Rope Handbook, Broderick and Bascom Rope Co., St. Louis, Missouri (1966)

Wire Rope Handbook and Catalog E, Armco Steel Corporation, Union Wire Rope, Kansas City, Missouri (1972)

Wireco Wire Rope, Handbook W67, Wire Rope Corporation of America, Inc., St. Joseph, Missouri (1971)

Ayre, R. S. and Abrams, J. I., "Dynamic Analysis and Response of Aircraft Arresting Systems", Proc. A.S.C.E. Engr. Mech. Div, April, 1958.

Carrier, G. F., "A Note on the Vibrating String", Quart. Appl. Math., Vol 7, (1949), pp. 97-101.

Carrier, G. F., "On the Non-Linear Vibration Problem of an Elastic String", Quart. Appl. Math., Vol 3 (1945), pp. 157-165.

Cole, J. C., Dougherty, C. B., and Huth, J. H., "Constant Strain Waves in Strings", J. Appl. Mech., Vol 20, No 4, (1953), pp. 19-22.

Cristescu, N., "Elastic-Plastic Waves in Strings" (in English), Arch. Mech. Stus. 12, 5/6 (1960), pp. 597-615.

Debriunyi, I. E., "An Investigation of the Damping of the Free Vibrations in Wire Cables", (in Russian), Trans. Sci.-Teck, Conf. on the Damping of Vibrations, Kiev., Akad. Nauk USSR 1960, pp 145-150; Ref. Zh. Mekh. no 11, 1961, Rev. 11 v 175

Engle, E., "The Torsional Tendency and Torsional Stiffness of Cables", (in German), Ost. Ing. Z. 1 (Jan. 1958), pp. 33 - 39.

Engle, E., "Twisting of Wire Ropes and Their Torsional Rigidity", (in German), Ost. Ingenieur Z. 1, 1 (Jan. 1958), pp. 33-39.

Eshelby, J. D., "The Interaction of Kinks and Elastic Waves", Proc. Roy. Soc. Ser. A 266 (1962), pp. 222-246.

Goldsmith, W., "An Elongating String Under the Action of a Transverse Force", J. Appl. Mech. 24, 4, (Dec. 1957) pp. 609 - 616.

Islinskii, A. Yu., "On the Equations of Longitudinal Motions of a Cable (elastic cord) of Variable Length", Dokl. Akad. Nauk. SSSR (N.S.) 95 (1954), pp. 939-941.

Kawashima, Sukeo, "On the Vibration and Impact of Elastic Cables", Mem. Fac. Engng. Kyushu Univ. 12, 2, (September 1950), pp 60-138

Kawashima, Sukeo, "Suppliment to the Report of the Vibration and the Impact of Elastic Cables", Memo. Fac. Engng. Kyusho Univ., 12, 3-4, (1951), 235-243.

Li, Wen-Hsiung, "Elastic Flexible Cable in Plane Motion Under Tension:, J. Appl. Mech. 26 (1959), pp. 587-593.

Marble, F. E., "The Motion of a Finite Elastic Cable", North American Instrument Company, Rept. (1954), Altadena, Calif.

Neidhardt, G. L., Eslinger, N. F., and Sasaki, F., "An Analytical Approach to the Alleviation of Dynamic Tensions in Aircraft Arresting Cables", WADC Tech. Rept. 58-217, Astia No. AD155542.

Pugsley, A. G., "On the Natural Frequencies of Suspension Chains", Quart. J. Mech. Appl. Math. 2, (1949), pp. 412-418.

Ringleb, F. O., "Basic Problems in the Problems of Aircraft Arresting Gear", from A Decade of Basic and Applied Science in the Navy, Office of Naval Research (1957), pp 59-75

Ringleb, F. O., "Cable Dynamics", Naval Air Material Center Rept., Philadelphia, Pa. NAEF-Eng-6169

Ringleb, F. O., "Dynamics of a Moving Cable", Naval Air Material Center, Rept. M-4812 (1948), Philadelphia, Pa.

Ringleb, F. O., "Motion and Stress of an Elastic Cable Due to Impact", A.S.M.E. Summer Conf., Berkley, Calif. (June 1957), pap.57-APM-10, 9pp.

Savin, G. M., "Dynamic Forces in a Hoisting Rope on Raising a Load from an Immovable Foundation", (in Russian), Dopovidi, Akad. Nauk. Ukrain. RSR No. 2 (154), pp. 140-147.

Savin, G. N. and Goroshko, O. A., Dynamics of Threads of Variable Length (Dinamika Niti Peremennoi Dliny), Kiev, Nauk. UKSSR (1962), 331 pp., 1.05r.

Savin, G. M. and Goroski, O. O., "Parameters of a Naturally Twisted Thread", Dopovidi Akad. Nauk. Ukrain. RSR 1959.

Savin, G. N., Postolnik, Yu. S., and Sevelo, V. N., "On the Equation of the Dynamics of an Elastic or Visco-Elastic Thread of Variable Length", Akad. Nauk. Urkrain. RSR. Prikl. Meh. 1 (1955), pp. 36-40. In Ukranian.

Sokolov, Yu. D., "Approximate Solution of the Basic Equation for the Dynamics of a Hoisting Cab le", (in Russian), Dop. Akad. Nauk. USSR No 1, (1955), pp 21-25,; Ref. Zh. Mekh. No. 1, (1957), Rev. 999

Sokolov, Yu. D., "Determination of Dynamic Forces in Mine Shaft Hoisting Cables", (in Russian), Prikl. Mekh. 1, 1 (1955), pp 23-35; Ref. Zh. Mekh. No. 12 (1956), Rev 8556.

Stevens, G. W. H., "Experiments on Waves in Wires with Fixed Ends", R. A. E. Rept. D. I. 57.

Thomlinson, J., "Theoretical Treatment of Aircraft Arresting Gear Problems", R.A.E. Tech. Note N.A. 194, A.R.C. 12,579, May, 1949

Truman, C., "High Velocity Engagement of Arresting Wires", Naval Air Missle Test Center, Point Mugu, Calif. 1954

Walten, T. S. and Polacheck, H., "Calculation of Non-linear Transient Motion of Cables", David W. Taylor Mod Basin Rep. 1279 (July 1959), 50 pp

Willis, J. M. N., Chisman, S. W., and Bullen, N. I., "Measurement and Suppression of Tension Waves in Arresting Gear Rope Systems", Areo. Res. Counc. Lond. Rep. Mem. 2981, (1957), 26 pp.

Yu, A. T., "Vibration Damping of Stranded Cable", Proc. Soc. Exp. Stress A alysis, 9, 2 (1952), pp 141-158.

----, "Haulage and Hoisting from Face to Surface", Coal Age, pp 220-225 (July 1967).

----, "Haulage and Hoisting from Face to Surface", Coal Age, Vol 73, pp 193-198 (July 1969).

VDI Guidelines for Listing and Hauling VDI 2358 (October 1968) (Published in Germany, English Translation is NAVSHIPS Translation 1417).

Berkovitz, H., "Evaluation of Hoist Rope Performance", Iron and Steel Engineer, (September 1960), p 111.

Goodwin, R. L., Jr., "Report on Two-Inch Towing Hawser Wire Rope", U. S. Navy, U.S.S. Munsee (ATF-107) (January 9, 1968).

Greenough, G. K., "The Safe Use of Wire Ropes in British Mines and Quarries", O.I.P.E.E.C., Bulletin No. 11, pp. 16-21 (1968).

Greenough, G. K., et al, "The Strength of Haulage and Suspension Components for Coal Mines", The Mining Engineer, No. 73-84, pp 160-174 (October 1966 - September 1967).

Grove, G. W., et al, "Some Haulage Safety Devices for Use on Grades, Slopes and Inclined Shafts", United States Department of the Interior, Bureau of Mines, Miners' Circular 43, (1942).

Hurt, F., and Walker, R. E., "Design Considerations and Applications of Modern Wire Rope Haulages", Journal of the Association of Mining Electrical and Mechanical Engineers, pp. 216-224 (October 1967).

White, M. C., "A Survey of the Wire Ropes Examined by SMRE after Failure or Serious Deterioration During the Five-Year Period 1961-1965", Safety in Mines Research Establishment, Research Report No. 249, (1967).

RUNNING, RIGGING, REQUIREMENTS AND TESTING

Deforest, A. U. and Hopkins, L. W., "The Testing of Wire Rope and Rope Wire", American Society for Testing and Materials (1932).

Dodson, M., et al. "Investigation of Wire Rope Service Requirements and Design Parameters", Final Report to Naval Ship Systems Command (Contract No. 00014-68-C-0492), Battelle Memorial Institute Columbus Laboratories (August 29, 1968)

Flowers, A. E., "How to Get More Life from Wire Rope", Coal Age, pp 76-81 (Paril 1955)

Franke, E. A., "The testing of Wire Rope", DRAHT, Vol 2, No. 2, pp 56-59 (February 1960).

Franke, E. A., "Testing of Wire Rope for Load Lifting Service", DRAHT, Vol. 13, No. 3, pp 114-116 (March 1962).

Gambrell, S. C., Jr., and Case, R. O., "New Machine for Accelerated Fatigue Tests of Wire Rope", Wire and Wire Products, pp 46-49 (June 1968).

Gaper, J. G., and Gunn, N., "Strength and Fatigue Tests on 20 Cut Steel Wire Ropes", Royal Aircraft Establishment, Farnborough, England.

Glushko, M. F., "Mechanical Testing of Wire Ropes", Industrial Laboratories 28, 8, pp 1038-1041 (February 1963).

Hagarman, P., and Pressley, L., "Evaluate, Test and Manufacture an Improved Wire Rope and Cable", Texas Instruments, Inc. (July 1967).

Laura, P. A., and Cassarella, M. J., "A Survey of Publications on Mechanical Cables and Cable Systems", Report 68-1, Themis Program No. 893, N00014-68-0506-0001, Catholic University of America (December 1968).

Layton, C., "Longer Life for Your Wire Rope", Engineering and Mining Journal, Vol 157, No 10, pp 100-103 (october 1956).

Russell, W. C., "How to Select and Care for Wire Rope, and Operating Conditions which Shorten Its Life", National Engineer, pp 43-46 (November 1956)

----, "Wire Rope - Simplified Practice Recommendation", National Bureau of Standards, Washington, D. C., SPR-190-50 (June 1950).

----, "Wire Rope Testing", Lab Project 930-44, Letter Report 932, Naval Applied Science Lab (March 8, 1968)

Drucker, D. S., and Tachau, H., "A New Design Criteria for Wire Rope", Journal of Applied Mechanics, Trans. ASME 12 (1945).

Meebold, R., Cables and Their Practical Use (Die Drahtseile in der Praxis), 2nd rev. ed., Berlin, Springer-Verlag, (1953)., vi + 108 pp, 121 figs. DM 12

Murphy, W. A., "Wire Rope - The Working Arm of Construction Equipment", paper prepared for presentation at National Fram Construction, and Industrial Machinery Meeting, Milwaukee, Wisconsin, September 14-17, 1964, Society of Automative Engineers, Inc.

Ropeman's Handbook, Second Edition, National Coal Board Production Department in collaboration with the Safety in Mines Research Establishment, Ministry of Power (England). Hazell, Iatron and Viney Ltd., Aylesbury, Backs (1966).

Wire Rope/Its Engineering, Wire Rope/Its Selection (2 booklets) The Rochester Corporation, Wire Rope Division, Culpeper, Virginia (undated).

UNDERGROUND MINING
(For Rope Uses of Relevance to Surface Mining)

Dean, Humphrey, "Hoist Rope Stress Analysis and Preventive Maintenance", Paper No 58, prepared for presentation at the 1970 Annual Meeting of the Canadian Institute of Mining and Metallurgy (1970).

Egawa, T. and Taneda, M., "External Pressure Produced by Multi-Layers of Rope Wound on a Hoisting Drum", Bulletin of JSME, Vol 1, No 2, pp 133-138 (1958).

Fuller, C. N., and Wainwright, E. J., "Recent Developments in Steel Wire Ropes for Hoisting from Great Depths", The South African Mechanical Engineer, Vol 16, No 11, pp 238-245 (June 1967).

Ilsley, L. C., and Mosier, M., "Inspection and Maintenance of Mine Hoisting Ropes", Department of the Interior, Bureau of Mines, Technical Paper 602, (1940).

Jehmlich, Dr., et al, "The Effects of Lubrication Corrosion Protection, Rope Construction and Pulley Material on the Durability of Mine Winding Ropes", Wire, Issue 95, pp 130-134 (June 1968).

Pettit, J., "A Technological Approach to the Mechanical Design, Application and Use of Equipment Used with Steel Wire Ropes in Coal Mines", The Mining Engineer, pp 611-622 (July 1969)

Richards, C. D., "Steel Wire Ropes for Hoisting Machinery and Lifts", Machinery, pp 1302-1309 (December 1962).

----, "Ropes for Deep Shafts", Mining Magazine, Vol 114, No 3, pp 209-211 (March 1966).