THIRD EDITION

HANDBOOK OF OCEANOGRAPHIC WINCH, WIRE AND CABLE TECHNOLOGY 2001

John F. Bash,
Editor

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Editor
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Handbook of Oceanographic Winch, Wire and Cable Technology

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PREFACE:

This Winch and Wire Handbook has been prepared for the ship operator, engineer, scientist and technician who is involved in the use, and reliance upon, the various combinations of winches, wires and cables found within the oceanographic and commercial communities. This third edition is an update of the previous edition and is an outgrowth of a Winch and Wire Symposium held at the Tulane/Xavier Center for Bioenvironmental Research in New Orleans Louisiana on 30 November and 1 December 1999. Several of the authors of the original handbook were speakers at this symposium. All fourteen of the original chapters have been reviewed and updated for this third edition. Two chapters have been removed and partially incorporated into other chapters. Several chapters remain with no changes. The index has been reduced and streamlined. The chapters remaining in this third edition were originally written and subsequently reviewed and updated by recognized authorities in their respective fields. The editing has been limited to providing continuity of material without altering the individual author's content or style.

The first edition of this handbook had its beginnings in late 1981 as a result of conversations with ship users. As a result, it became increasingly clear that some consolidated approach to the handling and understanding of both deck machinery and wires used at sea by the oceanographic community was required. The success of the first edition combined with the appearance of new cable types, which were applicable to deep ocean use, prompted the writing of a second revised edition, published in 1991. Alan Driscoll, of the University of Rhode Island, edited the original publication and the first edition. This third edition became necessary because of the continued changing of technology and the need to reprint the manual due to an exhausted supply of copies. Chapters 3, 4, 6, 10 and 11 underwent major rewrites. Chapters 1 and 9 remain as published in the second edition. Chapters 12 and 13 have been removed since much of the subject matter has been covered in other chapters. The remaining chapters have received minor updated changes.

The completion of this third edition could not have been accomplished without the hard work and dedication of many. First credit must be given to Ms. Dolly Dieter of the National Science Foundation for providing guidance and funding for the project. Credit also goes to the Steering Committee established to provide the planning of the update process, the initial review of the manual and direction to the editor for the updating. This committee included:

Jon Alberts	*WHOI*
Tom Althouse	*Scripps*
Sherman Bloomer	*OSU*
Richard Findley	*HBOI/U. of Miami*
William Hahn	*URI*
Kenneth Johnson	*MLML*
Craig Lee	*U. of Washington*
Robert Pickart	*WHOI*
Kenneth Smith	*Scripps*
Michael Webb	*NOAA*
Albert Williams	*WHOI*
Mark Holmes	*U. of Washington*

Setting the stage for the update was accomplished at a Winch and Wire Symposium held 30 November and 1 December 1999 at the Tulane/Xavier Center for Bioenvironmental Research in New Orleans, LA. The following members of a panel provided summaries for future needs of the scientific community. These were:

Tom Althouse	*Scripps*
Sherman Bloomer	*OSU*
Kenneth Johnson	*MLML*
Craig Lee	*U. of Washington*
Kenneth Smith	*Scripps*
Albert Williams	*WHOI*

Speakers at the symposium provided industries response to the scientific needs. These were:

Jon Alberts	*WHOI*
Tom Coughlin	*Vector*
Alec Crawford	*Deep Tek Ltd.*
Phil Gibson	*TMT Laboratories*

Etienne Grignard	*Grignard Co.*
William Hurley	*Glosten Associates*
Michael Markey	*Markey Machine*
Reed Okawa	*North American Crane & Equipment Co.*
James Stasny	*Dynacon Inc.*
Sim Whitehill	*Whitehill Manufacturing*

Considerable credit must be given to the hard work of reviewing, rewriting and updating the manual. These reviewers were:

Chapter 1	*Larry Means*	*Wire Rope Corps. Of America*
Chapter 2	*Len Onderdonk*	*Rochester*
Chapter 3	*Sim Whitehill Jr.*	*Whitehill Manufacturing*
Chapter 4	*George Wilkins*	*Pan Pacific Inst.*
Chapter 5	*Tom Coughlin*	*Vector*
Chapter 6	*Etienne Grignard*	*Grignard Co.*
Chapter 7	*Rich Findley*	*HBOI/U. of Miami*
Chapter 8	*Phil Gibson*	*TMT Laboratories*
Chapter 9	*J. F. Bash*	*URI*
Chapter 10	*Michael Markey*	*Markey Machine*
Chapter 11	*James Stasny*	*Dynacon Inc.*
Chapter 12	*Walter Paul*	*WHOI*

Finally considerable credit must be given to the secretarial support of Mrs. Diane McGannon of URI for her tireless effort of scanning the 550 plus pages of this manual as well as assisting with the layout and printing preparations.

John F. Bash
Editor

INTRODUCTION

This third edition to the Handbook of Oceanographic Winch, Wire and Cable Technology has been prepared to provide a reference and guide for the operation, handling and care of winches, wires, cables and oceanographic rope for the operators and scientists of the oceanographic community. Further references on specific subjects can be found in the various chapters.

Since the original writing of this handbook the operators of oceanographic ships have significantly improved the reliability and safety of deck equipment and deck operations. Some credit must be given to this handbook and the increased interest of both the funding agencies and the ship operators in striving for a higher level of operation and maintenance. Periodic review of this manual and symposia similar to the one held in New Orleans in 1999 provide a greater focus and heighten the interest of the oceanographic community to continuous improvements.

In the original Handbook the following word of caution was made. It remains germane.

It is appropriate to insert a word of caution for the reader who is dealing with the evaluation of a winch and wire system. That caution is that in dealing with a calculated ideal it may not always be possible to achieve that ideal in practice due to the physical restraints that are imposed by research vessel size and configuration. When it becomes necessary to depart from the calculated ideal for one reason or the other, the assessment should be scrutinized carefully in order to define the true operational limits for the equipment at hand. Additionally, problems that manifest themselves in an existing winch and wire system are rarely the result of a single element and are usually the result of several component problems. When problems do arise, the entire system should be evaluated and appropriate corrective or de-rating measures taken.

This handbook contains information suitable to initiate a careful review of existing winch systems as well as being useful for the specification of upgrades or new systems. It is the hope of the authors that this handbook will serve as a ready reference for both the oceanographic and commercial communities, now and in the future.

TABLE OF CONTENTS

CHAPTER 1

3X19 OCEANOGRAPHIC WIRE ROPE

W.A. LUCHT (Reviewed 2000 by Larry Means)

1.0 CONSTRUCTION AND MATERIALS

3 x 19 torque balanced oceanographic wire rope is a relatively new invention in the wire rope industry. It was first put to use as a winch line at Woods Hole with Scripps following a very short time later.

This was during the period a tapered 6-strand wire rope was being tried for use in deep water. The tapered 6-strand rope, as well as all other 6-strand, non-torque balanced rope, unwind when suspending a free hanging load. Any momentary sudden release of tension permits the wire rope to start winding back to its original design because of its spring-like properties. ln so doing, hockles and kinks are formed, which are familiar to all oceanographers.

To permit exploration of the ocean bottom with continuous rope lengths up to 46,000 feet, required the wire rope be made from wire of approximately 300,000 psi. The wire is stranded and the rope is closed on conventional rope making equipment. But here the differences start. Figure 1-1 is a sketch of this construction. It is difficult to tell by the untrained eye whether the rope is torque balanced or not.

Figure 1-2 explains the principle behind torque balancing. Under tension, each wire in the rope exhibits a torque directly related to the angle it makes with the rope axis. The sum of these torques are equally balanced by the opposing rope torque. When these forces are equal, the rope is virtually non-rotating under a free hanging load. This scientific principle eliminates hockles and kinks caused by unequal torque balance.

To achieve torque balance in rope making, the strand lay is shortened and the rope lay lengthened. The wires are fixed into position by heating the rope to a minimum temperature of 675°F. This process relieves internal stress so the wires stay in the position they are in when heated. Other benefits of stress relieving or thermal stabilization will be discussed.

2.0 TESTING

Oceanographic work is very demanding on a wire rope. It must be as strong as stated, it must not yield prematurely, it must not rotate excessively. Modulus of elasticity is important to length stability.

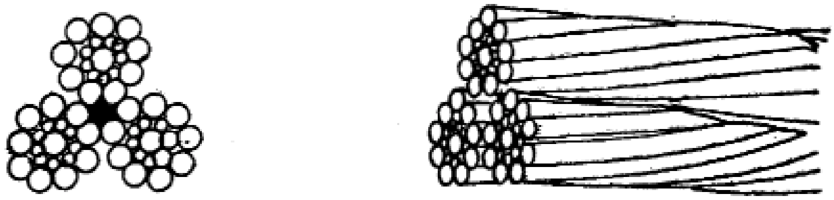


FIGURE 1-1

3 X 19 WIRE ROPE CONSTRUCTION

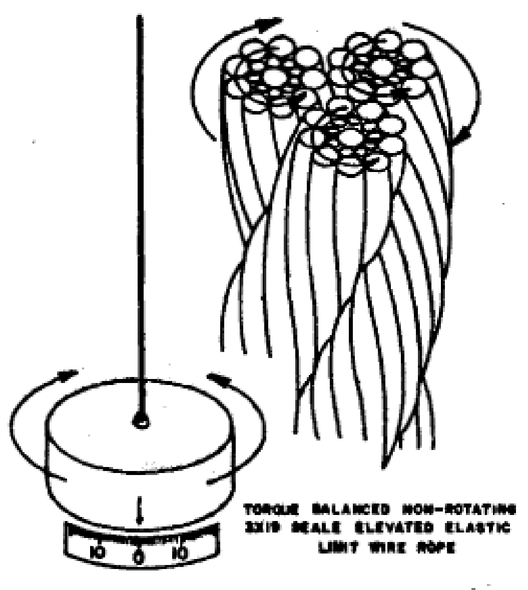


FIGURE 1-2

PRINCIPLE OF TORQUE BALANCED WIRE ROPE

Certified test reports from the manufacturer should be ordered when the wire rope is procured. The wire rope should be tested using the following procedure.

2.1 Breaking Load

A wire rope requires that a fitting be attached to each end of the sample specimen so it may be attached to the tensile test machine. Zinc filled wire rope sockets, resin filled wire rope sockets, or swaged sockets are ordinarily used. Swaging must be done properly in order to develop full strength. With 3-strand wire rope soft annealed wire is inserted in the valleys of the wire rope prior to swaging to improve efficiency. It has been found swaging the full length of the fitting rather than in bites, provides a higher breaking strength.

2.2 0.2% Offset Yield

Because of the torque generated in a 6-strand wire rope, it is virtually impossible to determine yield strength by methods used for testing solid bars. Three-strand torque balanced wire rope, however, can readily be tested as if it were a solid bar because of its non-rotating characteristics. The preferred method of test is known as the 0.2% offset yield, the procedure for which is described in ASTM E8. A high yield strength is important in not overloading ropes subjected to repeated shock loads.

2.3 Rotation

The specification for 3-strand torque balanced ropes is that it will not rotate more than one degree per foot of rope length with one end fixed and the other end free to rotate. It is important that this specification and the method of test is understood. There have been instances where the rope was tested with both ends fixed or an inoperative swivel placed in the system because the purchase order was not specific.

The correct method of test is to fix one end in the tensile test machine and attach a ball bearing swivel to the other end before attaching it to the test machine. This end must be free to rotate as the load is applied.

2.4 Modulus of elasticity

Both modulus of elasticity and rotation characteristics must be known to predict length stability for moorings. Stress relieved or thermally stabilized torque balanced 3-strand rope, follows Hooke's Law because of the heat treatment. There is a straight line relationship between stress and strain. Measurement can readily be made from a stress strain curve.

3.0 PROTECTIVE COATINGS

3.1 Galvanize

The wire composing three-strand torque balanced rope has a drawn galvanized coating (amgal). The process wire is first coated with molten zinc in a hot galvanizing process after which it is cold drawn through a series of dies to the required diameter. Although this process reduced the thickness of the zinc, it restores properties to the wire so it meets the same strength and ductility properties as bright wire. Hot galvanizing provides a metallurgical bond at the zinc-iron interface which is tough and wear resistant. The fatigue life of amgal wire rope in salt water environment is substantially greater than bright rope.

3.2 Plastic Jacketed Wire Rope

Wire rope and strands with extruded plastic coating such as polyethylene provide excellent protection from salt water, salt atmospheres, and chemically corrosive atmospheres. The plastic jacket provides a barrier between the rope and the environment, preventing contact and subsequent corrosion. As added corrosion protection, it is recommended that jacketed ropes be galvanized. Test and field experience have shown that small holidays in the plastic do not destroy the corrosion protection of the jacket. Socket-to-rope interfaces can be fitted with boots which will protect this critical area.

Plastic jackets also are beneficial when the major element of corrosion control is being provided by cathodic protection. The cathodic system only needs to protect areas where the plastic jacket is interrupted, thus reducing greatly the area requiring protection. The current level and number of anodes required are less with a coating, and the total length protected by one anode is much greater. Tests have shown that plastic-jacketed rope resists axial tensile fatique better than non-jacketed rope.

It is believed the vibration dampening properties of the jacket are responsible for this phenomena. Boots further add to the fatigue life of the product.

3.3 Lubrication

Wire rope is a working machine. Hertzian stresses are extremely high because of point contact between wire and sheave and wire and wire. Lubrication is necessary to minimize wear and the effects of wear. It also acts as a barrier between the steel and sea water to minimize corrosion. Good lubrication is known to double rope life.

4.0 STRESS RELIEVING WIRE ROPE AND IMPROVEMENT OF TECHNICAL PROPERTIES

It is a well known fact that a stress relieved treatment on cold worked steel can remove residual stresses and provide better metallurgical properties of the finished steel product. A problem that needed to be solved was how to stress relieve the long length ropes and strands which needed this treatment. A batch process was judged to be impractical. However, heating of wire rope or strand as it passes through an induction coil has proven to be a satisfactory manufacturing process. In this process, the rope or strand is quickly brought to a stress-relieving temperature by the induction heating process and then equally as quickly, cooled off by either an air or water quench. This results in very favorable metallurgical properties. It was also noted that wire ropes and strands subjected to this stress relieved treatment, showed preformed characteristics, whether they had been previously mechanically preformed or not. Once this was realized, it became a logical step to stress relieve the finished 3-strand torque balanced wire rope. Such a product was found to be fully preformed with no tendency towards twistiness, also no tendency to change the lay and destroy the torque balanced characteristics.

Stress relieving removes residual tensile stresses left in the outer fibers of wires due to the wire drawing process. The combination of these outer fiber tensile stresses, the bending stresses imposed in the wires taking their natural shape in the rope and bending over sheaves or drums, plus the tensile stresses involved in carrying the load result in a very high level of applied stress. This high level of applied stress can lead to micro-cracks and ultimate fatique. The removal of these residual tensile stresses enables the wire to better withstand the imposed loads and thus leads to increased fatigue life. In addition, the wires have increased ductility thus leading to improved rope breaking strength efficiency. The improvement of wire properties can be seen graphically and explicitly in Figures 1-3 and 1-4. In addition to improved wire properties, improved rope properties were also noted. Increases in elastic limit, yield strength and modulus of elasticity can be clearly seen in Figure 1-5. These increases taken together increase the shock absorbing capacity of the wire rope as measured by the area under the stress strain curve.

5.0 SPOOLING AND STORAGE OF 3-STRAND TORQUE BALANCED WIRE ROPE

5.1 Lebus Lagging

The use of Lebus lagging in combination with a diamond cut traverse is preferred when spooling 3-strand rope. It can be used with traction winches without developing hockles between the storage drums and haul-off point.

When it is necessary to use a smooth-faced drum, the following procedure has been found to be helpful:

5.2 Spooling Three-Strand Wire Rope on Smooth Faced Drum

Multi-layer spooling of 3-strand wire rope on smooth-faced drums can be accomplished. This requires uniform distribution of wraps on the bottom layer, without nesting.

Nesting permits non-uniform distribution as shown in Figure 1-6. To achieve uniform distribution of wraps with 3-strand wire rope, a filler can be inserted between wraps adjacent to the drum.

The size filler to use with 3-strand Torque Balanced wire rope is given in Table 1. With this information, the following procedure should be used:

1. Measure width between flanges at drum.

2. Determine how many full wraps can be accommodated with spacing from center to center of wrap being from d_1 to d_2 inches. The closer to d_1 the better the spooling.

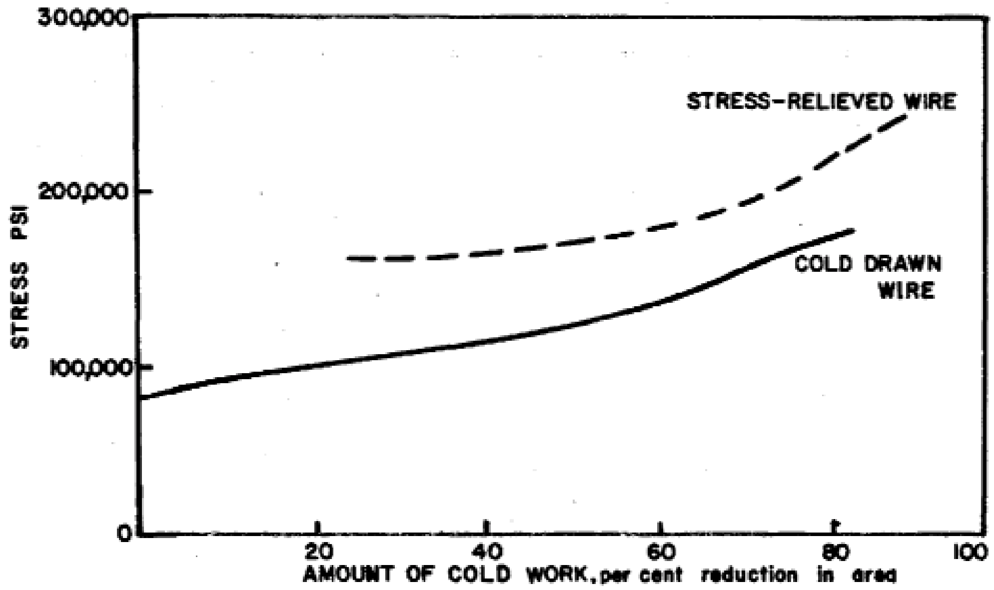


FIGURE 1-3
INCREASE INELASTIC LIMIT (0.01% OFFSET)

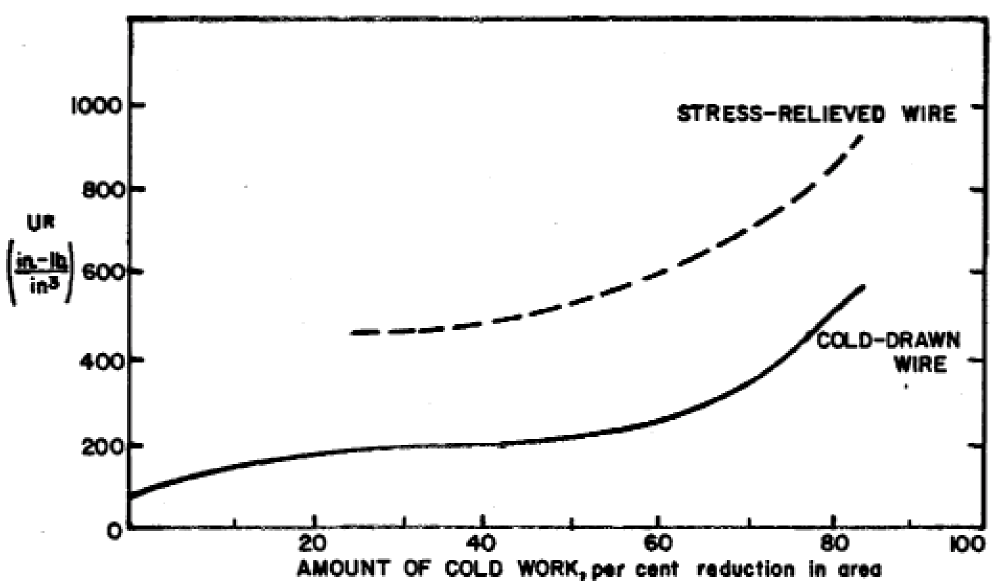
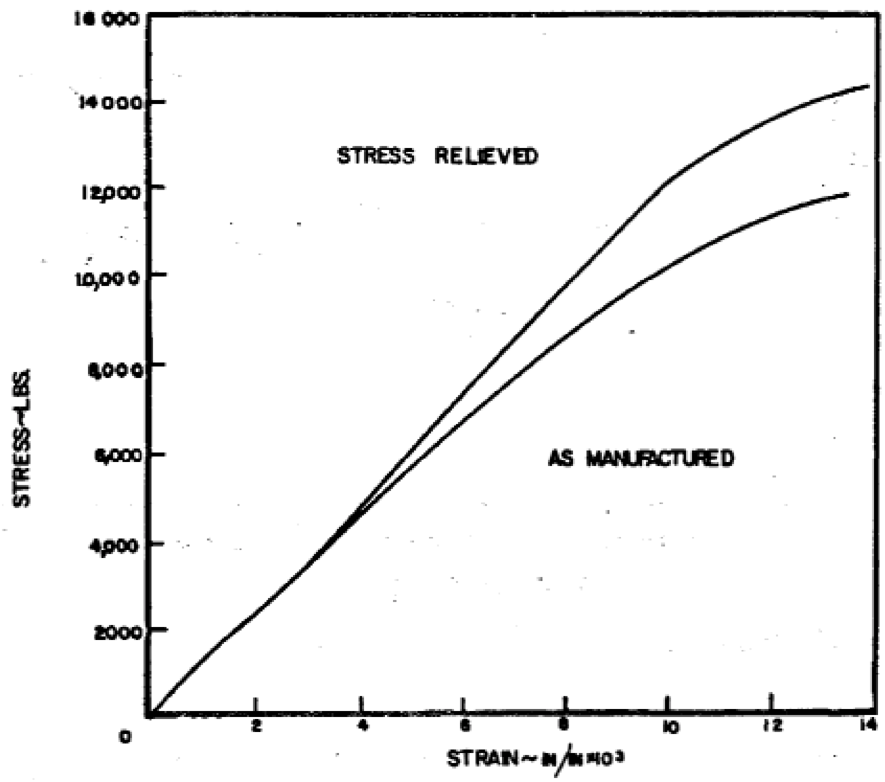


FIGURE 1-4

INCREASE IN MODULUS OF RESILIENCE (Ur)

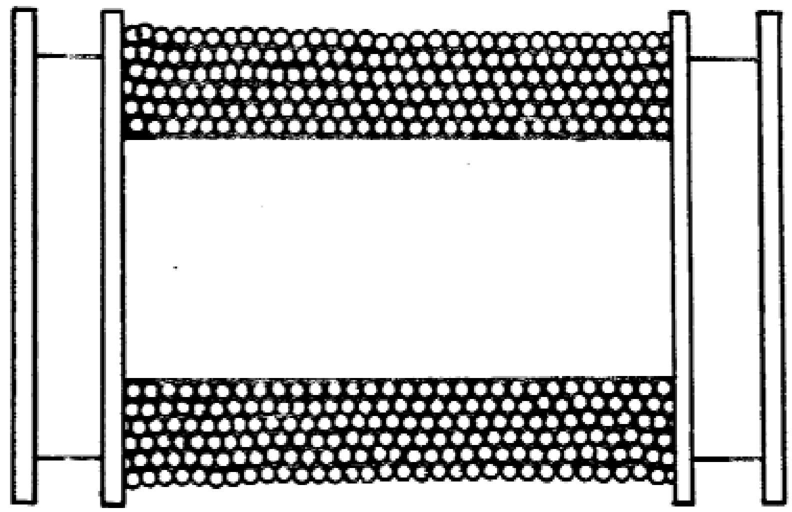


STRAIN IN./IN. x 10^{-3}

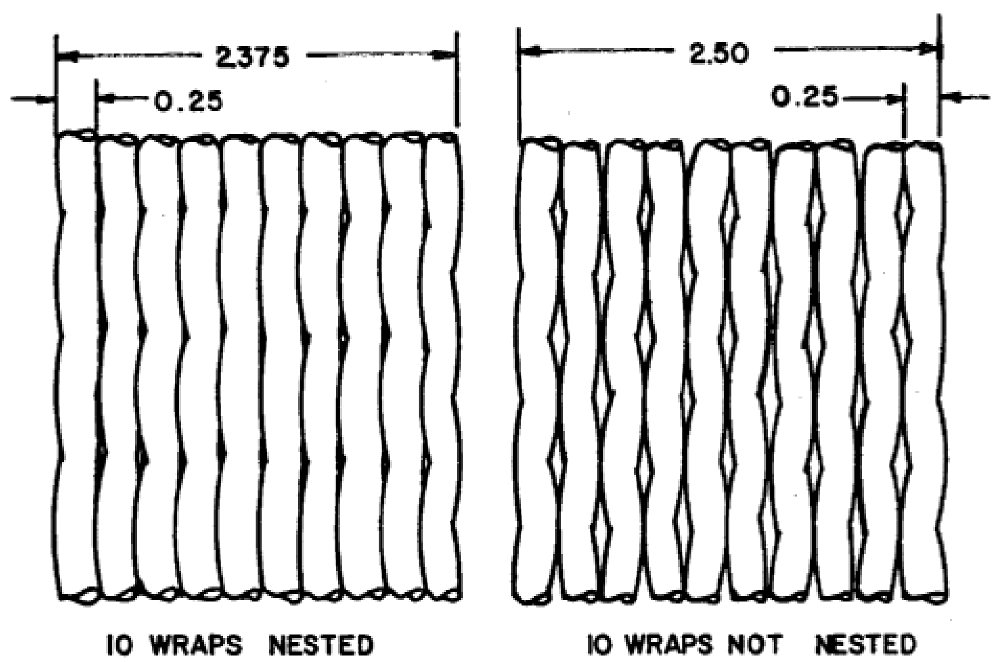
	AS MANUF.	STR. RELIEVED
MODULUS of ELASTICITY (PSI)	19,000,000	20,300,000
JOHNSON'S ELASTIC LIMIT (LBS)	9,600	12,400
2% OFFSET Y. S. (LBS)	11,200	14,000
ELONGATION in 24 IN. (%)	3.0	5.1
BREAKING LOAD (LBS)	14,960	15,260

FIGURE 1-5

IMPROVEMENT IN MECHANICAL PROPERTIES OF 3/8" 3 X 19 SEAL TORQUE BALANCED AMGAL MONITOR AA

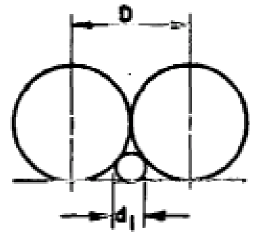


PROPER WIRE NESTING
FIGURE 1-6A

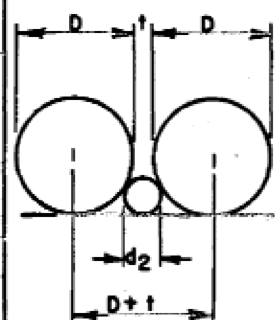


EFFECT OF ROPE NESTING ON SMOOTH FACED DRUM
FIGURE 1-6B

Nom Rope Dia., (in.)	Con-struc-tion	Calcu-lated Rope Dia., (in.) D	Spac-ing Toler-ance t	d_1 (in.)	d_2 (in.)
5/32	3 x 7	.173	1/32	.043	.059
11/64	"	.190	"	.0457	.063
3/16	"	.205	"	.051	.067
7/32	"	.233	"	.058	.074
1/4	"	.268	"	.067	.083
5/16	"	.328	"	.082	.097
3/8	"	.395	"	.099	.111
7/16	"	.456	"	.114	.130
1/2	"	.523	"	.131	.146
9/16	"	.589	"	.147	.163
11/64	3 x 19	.190	"	.0475	.063
3/16	Seale	.205	"	.051	.067
7/32	"	.236	"	.059	.075
1/4	"	.265	"	.066	.082
5/16	"	.328	"	.082	.098
3/8	"	.392	"	.098	.114
7/16	"	.460	"	.115	.131
1/2	"	.522	"	.1305	.146
9/16	"	.586	"	.1465	.162
5/8	"	.647	"	.162	.177
3/4	3 x 19	.781	1/32	.195	.211
7/8	Seale	.917	3/64	.229	.253
1	"	1.041	"	.260	.284
1-1/8	"	1.166	"	.2915	.315
7/8	3 x 25 F.W.	.915	"	.229	.252
1	"	1.045	"	.261	.285
7/16	3 x 36 Seale F.W.	.464	1/32	.116	.132
1/2	3 x 46	.528	"	.132	.148
9/16	Seale	.587	"	.147	.162
5/8	F.W.	.645	"	.161	.177
3/4	"	.766	"	.194	.210
7/8	"	.918	3/64	.2295	.253
1	"	1.045	"	.261	.285
1-1/8	"	1.172	"	.293	.316
1-1/4	"	1.314	1/16	.3285	.360
1-3/8	"	1.437	"	.359	.390
1-1/2	"	1.564	"	.391	.422
1-5/8	"	1.718	3/32	.4295	.476
1-3/4	"	1.843	"	.461	.508



$d_1 = \frac{D}{4}$



$d_2 = \frac{D}{4} + \frac{t}{2}$

TABLE 1 FILLER SIZES FOR SPOOLING 3-STRAND TORQUE BALANCED ROPE ON SMOOTH FACED DRUM

3. Insert filler when spooling bottom layer. This is done simultaneously.

4. Practically anything can be used as a filler, but steel strands or IWAC ropes are preferred.

5. The bottom layer must be tight.

6. If a whole number of wraps cannot be accommodated on the bottom layer, a filler or spacer should be added to the flange.

7. After placing the bottom layer, spooling can proceed in normal fashion for parallel grooved drum.

6.0 RECOMMENDED INSTALLATION PROCEDURE FOR TORQUE BALANCED ROPE

These recommendations should be effective for 3-strand Torque Balanced ropes where the torsional balance is obtained by introducing relatively short strand lays with very long rope lays.

These factors are such that although the rope as manufactured has no twist, or possibly up to one turn of tightening twist, in operating over sheaves the rope develops tendencies to rotate in a direction to unwind more the tightly wound wires in the strands than the long-lay strands in the rope. This results in tendencies of the rope to twist in a tightening direction.

The smaller the sheaves the greater the twist potential; therefore, it is important to provide as favorable sheave-to-rope ratios as is possible.

It has been found that in all of our successfully operating Torque Balanced ropes which initially showed tendencies to rotate in tightening direction, the following procedure was followed:

- o The dead end of the rope was disengaged from its termination point and approximately five turns of tightening twist were induced into the rope and the rope reattached to its termination.

- o The hook block was raised and lowered about four times with no pay-loading in order to distribute the induced twist into the rope reeved in the block.

- o A pay-load was applied to the hook block, then raised and lowered two or three times. In most situations the hook block operated with no indication of rope twist tendencies. Where twist indications were still evident, an additional three or four twists were induced into the rope at the dead end, with the desired results.

It must be remembered that the smaller the sheaves over which these ropes operate, the greater are the radial pressures of the ropes on the sheaves, the more pronounced is the roller-pin effect on the rope, resulting in greater tendencies of the rope to twist up in a tightening direction.

It should be noted that the direction of twist is in the opposite direction of that encountered in standard hoist ropes operating under the same relative conditions.

Steel reels are much preferred over wood for shipment from the manufacturer, storage, and paying off. The rope should be kept covered and dry. It should not be in contact with cinders or dirt, as these often contain injurious chemicals. Last but not least, the rope should be lubricated so it is ready for use when used again.

7.0 STAINLESS STEEL TORQUE BALANCED STAINLESS STEEL ROPE (Type 302 and 304)

Wire rope made from types 302 and 304 wire perform well in fresh water. However, in marine environment at ambient temperature, there have been failures which have been the subject of considerable discussion and disagreement. Failure, when caused by corrosion, often is ascribed to stress corrosion, other times to pitting, crevice corrosion or tunnel corrosion.

In some cases, stainless steels are preferred and even necessary when sample cleanliness or hydrogen embrittlement is a factor.

Type 302 and 304 need access to oxygen to maintain a protective oxide film. Moving water over 5 m.p.h. generally will provide such access.

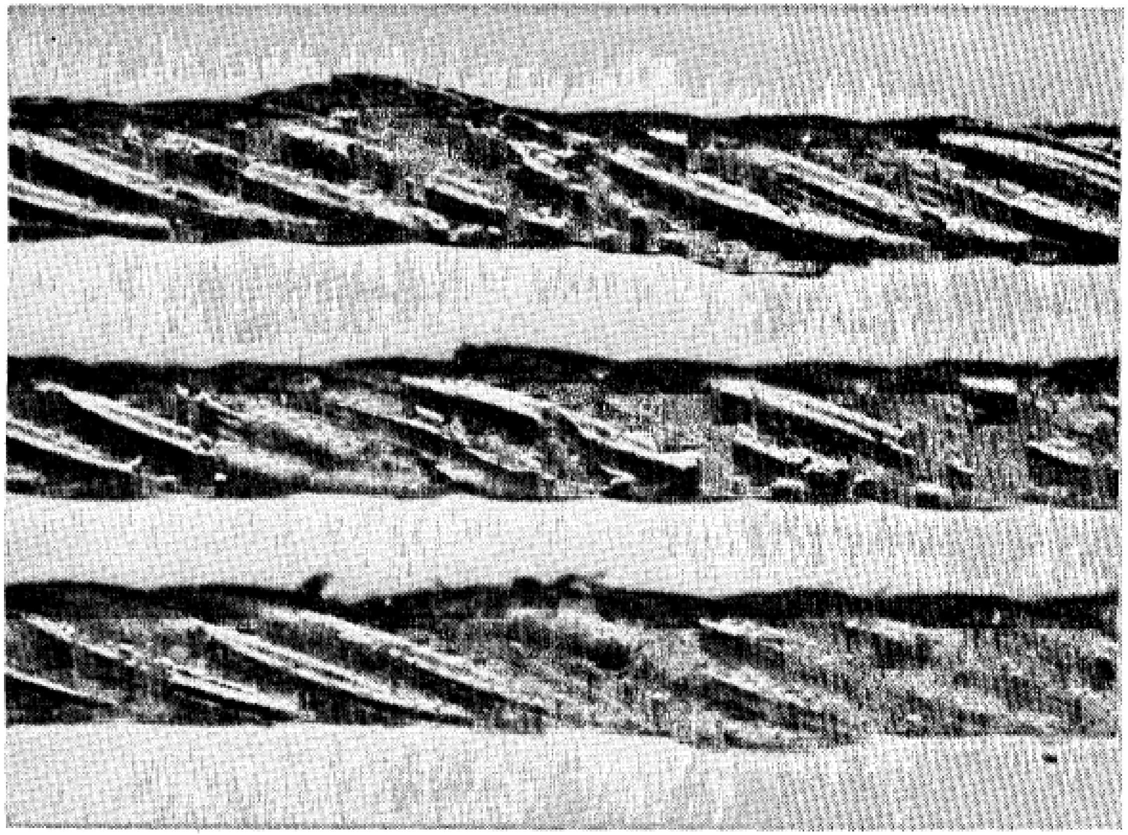
In Marine environments, salt and marine growth tend to deposit heavily in the valleys between strands thus limiting the oxygen supply in these areas. This leads to a rapid corrosion of the wire in selected areas and thus the term crevice corrosion. lf the deposits in the valleys can be eliminated, rope life should be extended.

Another curious phenomenon is known as tunnel corrosion. This is believed to originate from pits in the steel thus exposing a region having a different electric potential than the steel surface. The pit continues to elongate under the wire surface into hollow tunnels.

Photograph 1 shows an example of tunnel corrosion.

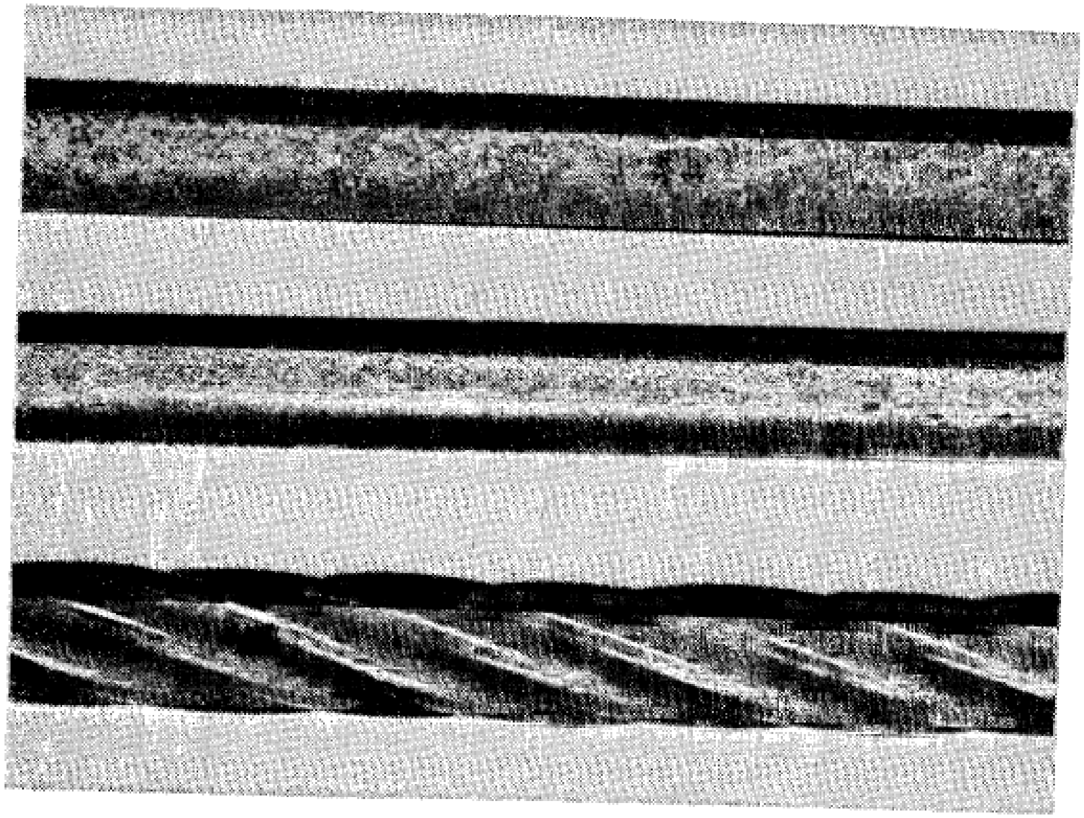
Experiments and experience have shown that stress relieving significantly improved the corrosion resistance of single wires and strands in both marine atmosphere and sea water.

Photograph 2 shows the favorable result of an exposure experiment at the 80 foot International Nickel test lot at Kure Beach, North Carolina.



PHOTOGRAPH 1 (X 15)

PITTING AND TUNNEL CORROSION OF 1 X 7 STRAND, CONSTRUCTED FROM DRAWN AISI 302 STAINLESS STEEL WIRES, CAUSED BY EXPOSURE IN SEA WATER



0.050-INCH-DIAM HARD-DRAWN AISI 304 WIRE

0.50-INCH-DIAM HARD-DRAWN AND STRESS-RELIEVED AISI 304 WIRE

0.018-INCH-DIAM AISI 302 HARD-DRAWN WIRE STRESS-RELIEVED AS STRANDED

PHOTOGRAPHY 2 (X 15)

APPEARANCE OF AISI 302 AND 304 DRAWN AND SUBSEQUENTLY STRESS-RELIEVED WIRES AND STRAND AFTER THREE YEARS OF EXPOSURE IN A SEVERE MARINE ATMOSPHERE

8.0 ENDLESS NASH-TUCK SPLICE FOR THREE-STRAND ROPE

The following dimensions apply to the detailed steps required to make the splice and are denoted in Figures 7 through 13.

Rope Size, Inches	A Feet	B Feet	C Feet	D Inches	E Inches	F Inches
3/16	92	46	45	12	3	5-3/4
7/32	112	56	55	12	3-1/2	6-3/4
1/4	127	61	60	12	4	7-3/4
5/16	163	81-1/2	80	18	5	9-3/4
3/8	193	96-1/2	95	18	6	11-3/4
7/16	224	112	110	24	6-3/4	13-1/2
1/2	254	127	125	24	7-3/4	15-1/2
9/16	254	127	125	24	8-3/4	17-1/2
5/8	254	127	125	24	9-3/4	19-3/8
3/4	256	128	125	36	11-1/2	23-1/4
7/8	256	128	125	36	13-1/2	27-1/4
1	256	128	125	36	15-1/2	31
1-1/8	256	128	125	35	18	34-3/4

- Use “A” length for splicing.
- Unlay both rope ends “B” distance (see Figure 1-7).
- Form marriage at this point and finger lock the three strands of one with the three strands of the other (see Figure 1-8).
- Run back one strand from one end “C” distance, replacing it with the proper strand from the second end (see Figure 1-9).
- Run back another strand from the second end the same distance, replacing it with the proper strand from the first end.
- We now have three pairs of strands protruding from the rope, separated by a distance of “C” (see Figure 1-10).

- Cut off the longer strand ends so that all strands protrude about "D" inches from the rope (see Figure 1-10).
- Unlay each protruding strand "E" distance so that the two strands in each pair protrude "F" apart (see Figure 1-11).
- Split each protruding strand in half back to point of protrusion so that one-half has a center wire, four inner wires, and the four outer wires covering these inner wires, and the other half has the remaining wires (see Figures 1-12).
- When 3 x 7 is used, one-half has the center wire and three adjacent outer wires, and the other half has the four remaining wires.
- Make ten complete wraps connecting half of one strand with half of the other in each pair (see Figure 1-13).
- Using a propane torch, anneal all the protruding split strands thoroughly. Be careful and do not heat the live strands.

The desired temperature at which annealing should occur is approximately 1300°F (704°C). The use of other than a propane torch using an ambient oxygen supply should be avoided due to the much greater heat produced by other devices. For example:

Torch Type	Tip Temperature
Ambient 02 Propane	1700°F (927°C)
Ambient 02/Propane (Swivel tip)	2700°F (1 482°C)
Propane/Oxygen	4579°F (2526°C)
Acc/Oxygen	6000°F (331 6°C)

*All temperatures based on medium flame

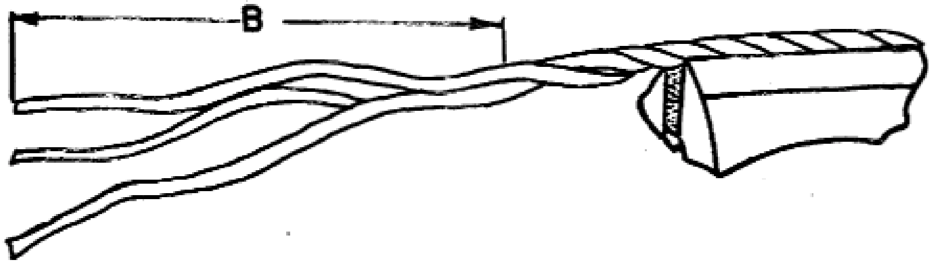


FIGURE 1-7

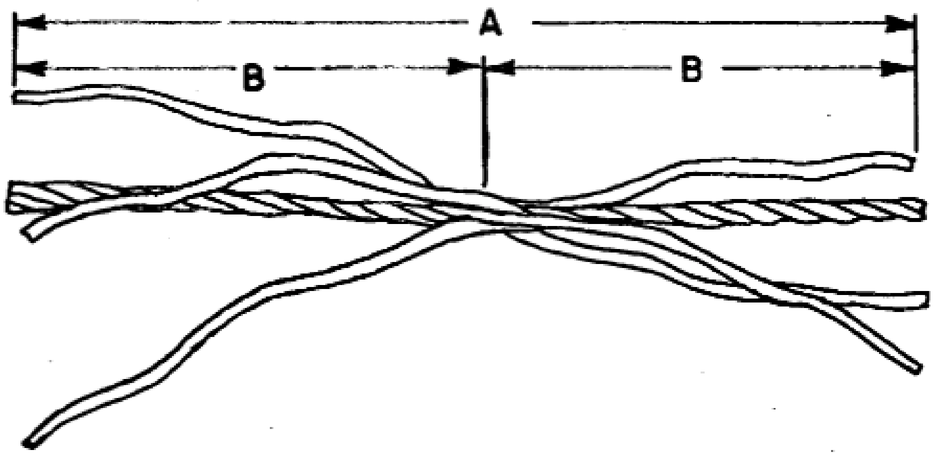


FIGURE 1-8

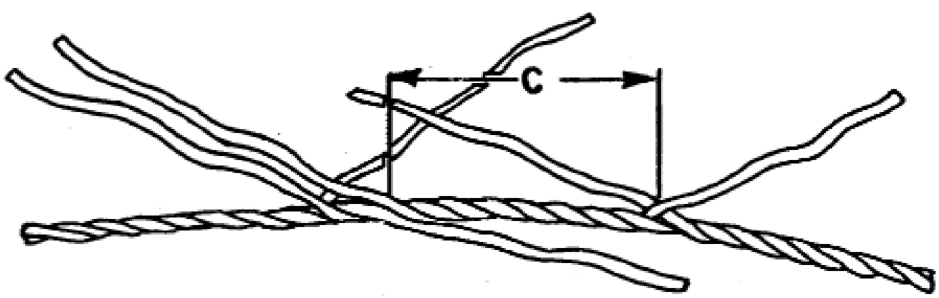


FIGURE 1-9

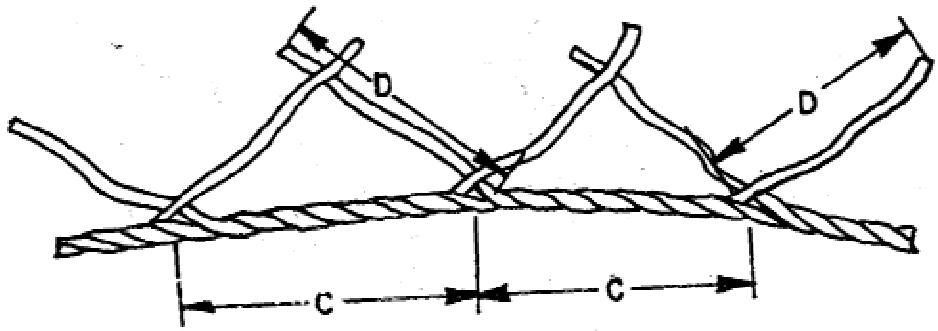


FIGURE 1-10

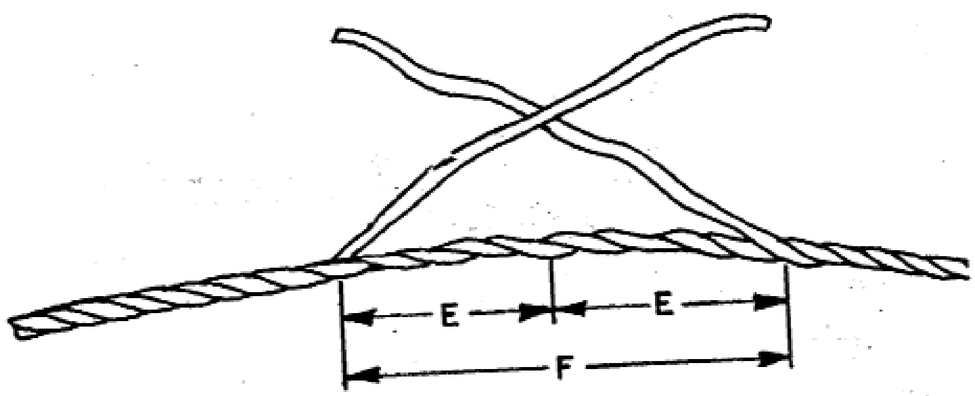


FIGURE 1-11

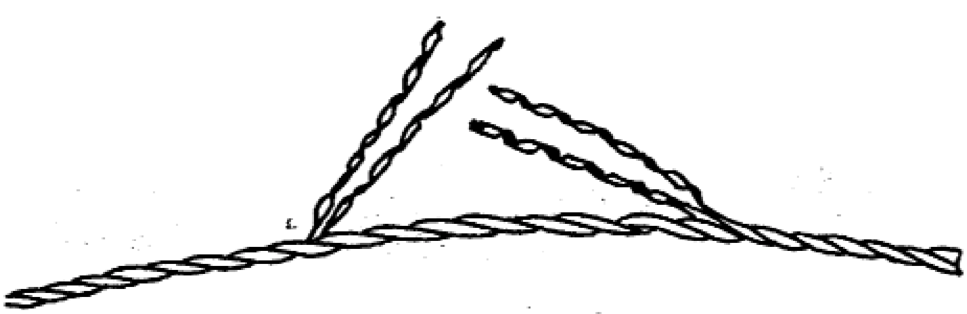


FIGURE 1-12

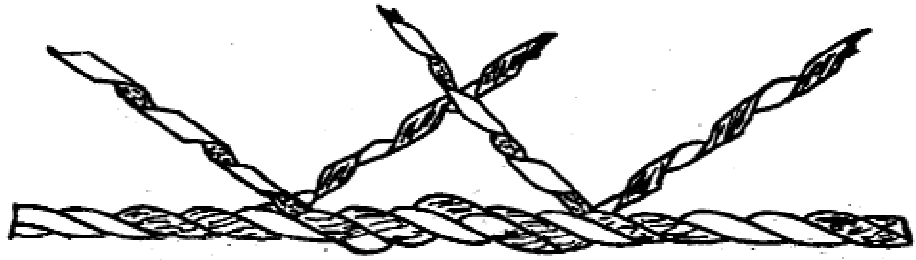


FIGURE 1-13

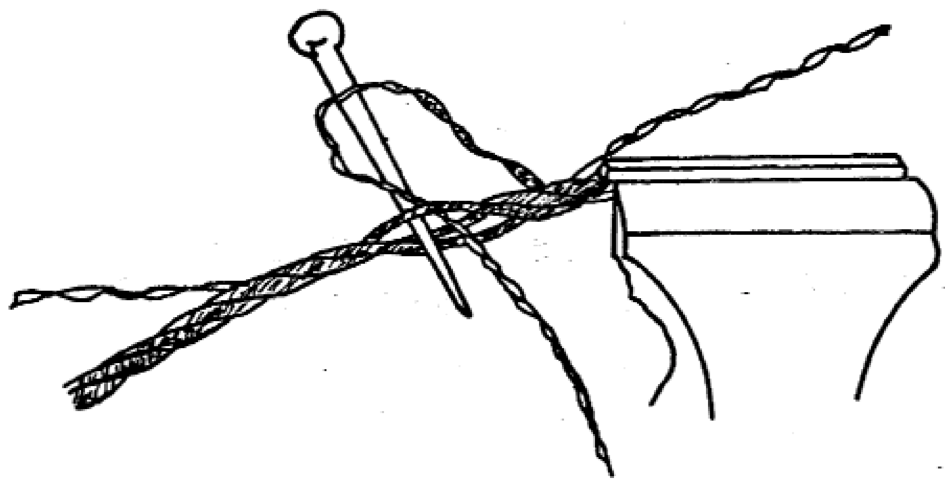


FIGURE 1-14

FIGURE 1-15

- Nash-Tuck each half strand individually with four full tucks by inserting each half strand under the first adjacent strand, over the second, under the third, over the fourth, under the fifth, over the sixth, and under the seventh, following the direction of the crown wires. These half strands must be untwisted at each tuck so that the wires are parallel as they go under and over the rope strands (see Figure 1-14).

- Pound or press the spliced section into a round tight configuration and cut the protruding split strands off as close to the rope body as possible (see Figure 1-15).

Caution, spliced 3 x 19 wire rope should be rated at 80% of its original, new rope breaking strength. This percentage is based on a properly installed Nash-Tuck splice as described above. Shorter splices than recommended, improper technique or accidental annealing of the rope itself will further reduce the efficiency of the splice.

9.0 WIRE ROPE RETIREMENT CRITERIA

A wire rope should be considered for replacement if it has three or more broken wires per strand in one strand lay or six or more broken wires in one rope lay.

It is important the wire surface be sufficiently clean so that the broken wires are visible. A good inspection method is to encircle rope with a rag or cotton waste and run the hoist slowly. If broken ends protrude and catch the waste or ends, bits of it will show location of the broken wires. It is best to face in the direction the rope is moving when holding the rag so it is pulled away from the holder if it snags on broken wires. A bare or gloved hand, rather than a rag or cotton waste, can be dangerous. A rope speed of 50 fpm or less is usually suggested.

Diameter and lay length measurements are most easily made at the same time and at the same location along the rope. A significant reduction in diameter can be the result of loss of metallic area from corrosion or from stretching due to broken wires.

Strand rope should be measured with a 3-point micrometer to obtain a meaningful measurement. Look for corrosion at and under attachments and at the end terminations. Corrosion is a reason for replacement as it is difficult to estimate remaining strength.

Structural damage is fairly easy to see. Types that call for immediate rope removal, If they cannot be removed by cuffing off the ends of the rope include kinks and doglegs.

Some additional, but more subtle forms of wire damage which will effect wire performance and ultimate strength are abrasions of the outer wires and general corrosion; i.e., flaking rust. The flattening of external wires due to abrasion occurring at the outboard sheave, level wind rollers or by the wire coming into contact with a fixed object reduces the wire load carrying capacity and should be watched for during visual inspections of the wire.

A wire condition exhibiting flaking rust indicates that the wires protective zinc coating has been lost and that the individual wire in the rope is being reduced through an oxidization process. This results in a reduction of the metallic area of each strand with a re-suiting lowering of the ultimate strength of the whole wire. Since this condition is difficult to evaluate for the entire rope length a rope exhibiting this condition over a major portion of its length should be retired from service.

10.0 MINIMUM TREAD DIAMETERS OF SHEAVES AND DRUMS IN INCHES FOR 3-STRAND WIRE.

Rope Dia.	3 x 7	3 x 19	3 x 25	3 x 46
1/4	14.5	11.5		
5/16	19	14.4		
3/8	23	17		
7/16	27	20		
1/2	29	23		13
9/16	35	26		15
5/8	38	29		16
3/4	46	35	28	
7/8	54	40	33	23
1	58	46	37	26
1-1/8		52	42	30
1-1/4		58	47	33
1-3/8		64	52	36
1-1/2		70	57	40
1-5/8				44
1-3/4				47
1-7/8				50
2				53

These minimum tread diameters are based on factors of approximately 400 times the diameter of outer wires.

11.0 GROOVES

Grooves in sheaves and drums should be slightly larger than the rope, in order to avoid pinching and binding of the strands, and to permit the rope to adjust itself to the radius of curvature. The greater the angle of approach to the groove, the larger the tolerance required to prevent excessive flange wear (Figure 1-16).

The diameter of an unused rope may exceed the nominal diameter by the amounts specified in the following table:

Diameter Tolerance for Wire Rope

Nominal Diameter of Rope in inches	Undersize %	Oversize %
0 to 1/8	0	8
Over 1/8 to 3/16	0	7
Over 3/16 to 1/4	0	6
Over 1/4	0	5

Grooves which have been worn to the minimum diameter shown in the table should be re-machined to the minimum diameter shown for New or Re-machined Grooves. Grooves of too large diameter do not properly support the rope, and permit it to become elliptical.

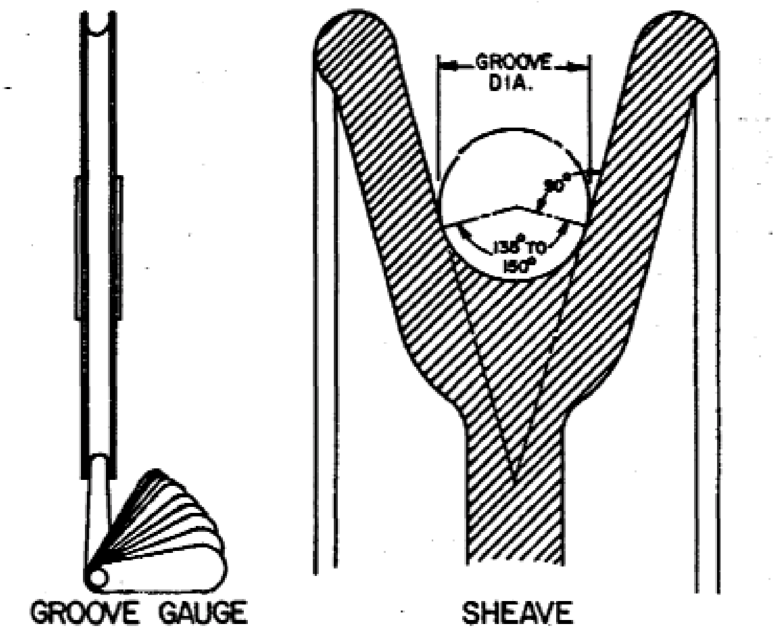


FIGURE 1-16

.SHEAVE GROOVE DIMENSIONS

Tolerance Groove Diameter Should Exceed Nominal Rope Diameter		
Nominal Diameter of Rope in Inches	Minimum (%)	New or Remachined Grooves (%)
0 to 1/8	4	8
Over 1/8 to 3/16	3.5	7
Over 3/16 to 1/4	3	6
Over 1/4	2.5	5

2.0 FLEET ANGLE

On installations where the ‘wire rope passes over a lead sheave then onto a drum, it is important that the lead sheave be located at a sufficient distance from the drum to maintain a small fleet angle at all times. The fleet angle is the side angle at which the rope approaches the sheave from the drum. It is the angle between the center line of the wire rope (Figure 1-17).

FIGURE 1-17
EXAMPLE OF FLEET ANGLES

‘Experience has proven that the best wire rope services is obtained when the maximum fleet angle is not more than 1-1/2 degrees for smooth drums and 2 degrees for grooved drums. The maximum fleet angle is an angle between the center line of the sheave and the rope when it is at the end of its traverse travel on drum Fleet angles of 1-1/2 and 2 degrees are the equivalents Of approximately 38 and 29 feet, respectively, of lead for each foot of rope traverse travel either side of the center of travel in line with the lead sheave should be located not less than 57 feet from the lead sheave. If the drum were grooved, the minimum distance should be approximately 43.5 feet.

13.0 HOW TO ORDER WIRE ROPE SPECIFICATIONS

To ensure that your order for 3-strand wire rope is accurately filled, the following information should be included for each item.

- Length: The length of each piece and number of pieces should be specified.
- Diameter: Diameter should be specified.
- Construction should be specified.
- Finish: Amgal should be specified for carbon steel. Stainless steel would ordinarily be furnished bright.
- Grade: The grade should be stated, such as Monitor AA (Extra Improved Plow), Monitor AAA, etc.
- Preforming: When excellay preforming is desired, the order should so state.
- Lay: Torque Balanced should be specified. Right lay will be furnished unless specified otherwise
- Rotation resistance: Maximum of one degree per foot of length when loaded to 70% of strength with one end fixed and the other free to rotate.
- Special Processing: Stress relieving or elevated elastic limit should be specified.
- Plastic Jacketing: Specified when required.
- Fittings: Details as to type attachment should be specified when required.
- End Termination.
- Modulus of Elasticity: Should be specified when length stability is important. 3 x 19 torque balanced elevated elastic limit 20,000,000 psi minimum.

14.0 NON-DESTRUCTIVE TESTING

Electromagnetic non-destructive testing of wire rope has been in use for over 25 years. During this period, more and more extensive use of this method has been relied on to evaluate the material condition of wire ropes used in situations where personnel or equipment safety are concerned, i.e., mining and the ski industry. Normally such inspections are carried out by agencies whose trained personnel are expert in the interpretation of signals resulting from broken or corroded wires.

Basically, the testing units are either AC or DC in nature. The AC unit measures the differences in cross sectional areas and is best for corrosion detection. The DC units are best used to detect broken wires if sufficient separation of the broken ends exists. However, cracked wires cannot normally be detected with this unit. The principals of operation for both individual AC/DC test units and the unitized AC/DC units is discussed in the following text.

14.1 <u>Individual AC/DC Units</u>

<u>DC Unit</u>: In the DC unit, strong permanent magnets are placed around a section of wire rope so that the rope becomes saturated with magnetic lines of flux (Figure 1-18). Lines of flux can be observed by iron filings sprinkled on top of a piece of paper having magnets underneath (Figure 1-19). The flux appears to "flow" from the north to south pole; however, the lines are stationary. The lines of flux are also distinct because of the existence of both attractive and repulsive forces. Saturation means that if stronger magnets were used, the number of lines of flux for a given cross section (flux density) would remain essentially unchanged.

If a broken wire were present in a saturated section of rope, then a north and south pole would be formed and the lines of flux would "jump" the gap (Figure 1-20). It is these lines of flux, called flux leakage, that can be detected to indicate a broken wire. Pitting from corrosion and localized wear will also interrupt the saturated lines of flux and cause flux leakage.

A classic physics experiment is to demonstrate that a magnetic field can produce an induced voltage in a conductor that is passed through the magnetic field. The conductor, passing at right angles through the lines of

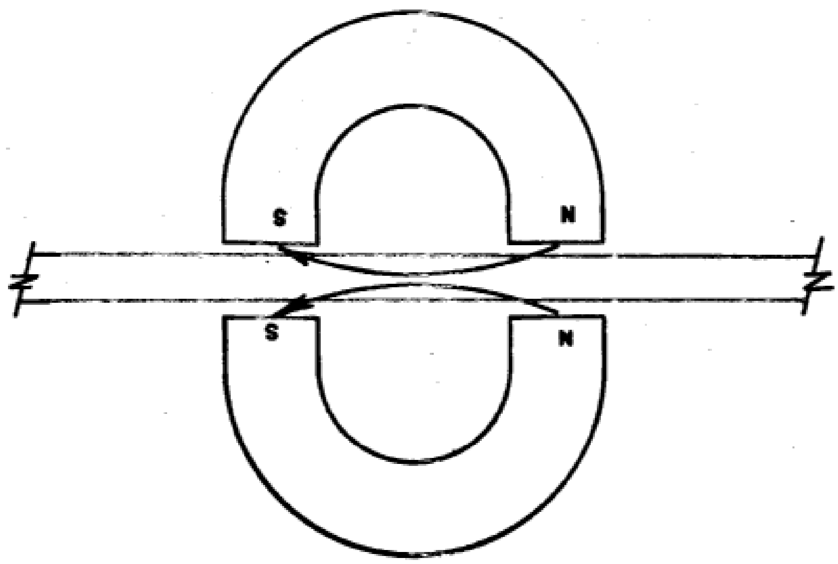


FIGURE 1-18
SATURATION OF WIRE ROPE WITH LINES OF FLUX

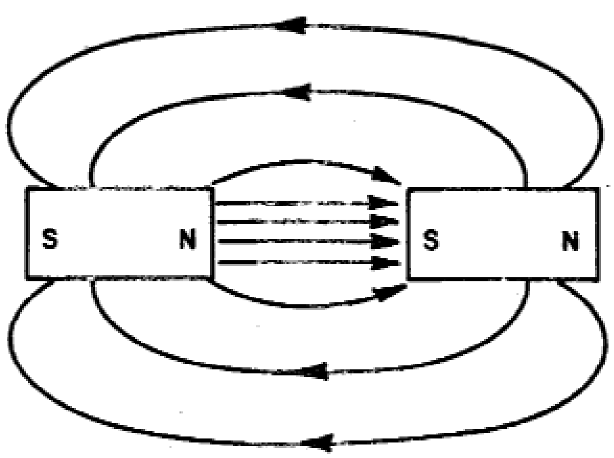


FIGURE 1-19 LINES OF FLUX

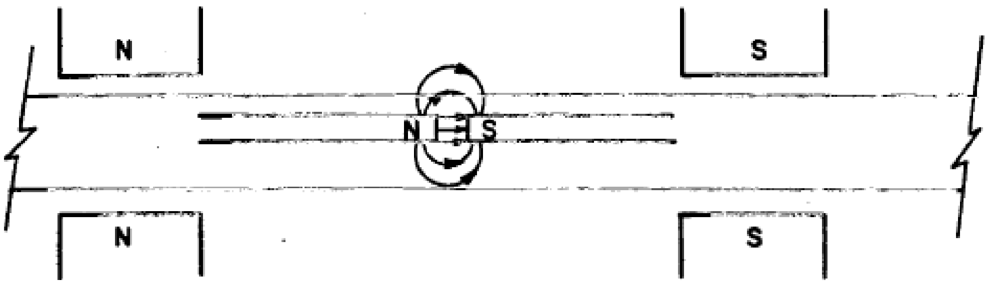


FIGURE 1-20

FLUX LEAKAGE LOCATION OF BROKEN WIRE

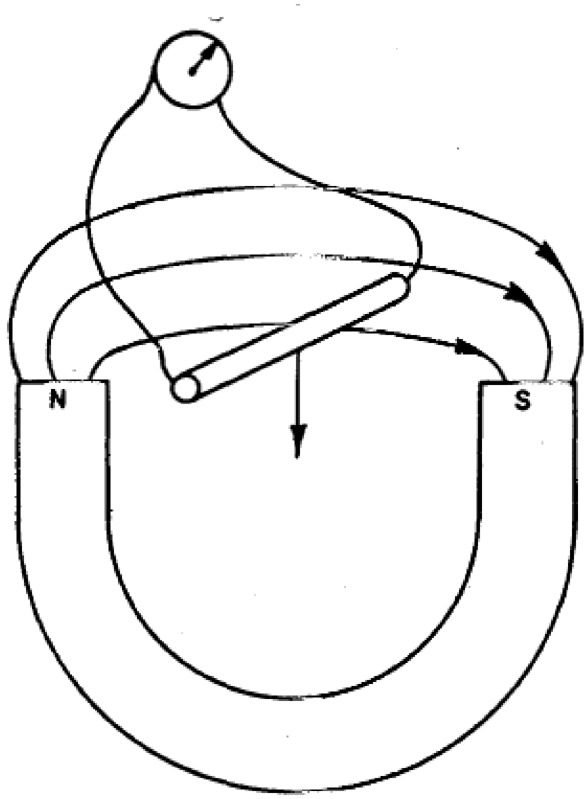


FIGURE 1-21

A CURRENT IS CREATED IN A CONDUCTOR
MOVING THROUGH LINES OF FLUX

flux, must have a minimum travel speed through the flux field in order for the voltage to be large enough to measure (Figure 1-21).

Flux leakage in the wire rope is detected by using this phenomenon. However, in this case, the conductor is a search coil that is held stationary while the magnetic field is moving. In the inspection equipment, search coils are placed around the saturated wire rope between the poles of the permanent magnet. The rope travels at some minimum speed; thus, any flux leakage will also be moving and will pass through the search coils (Figure 1-22). When this occurs, an induced voltage is generated in the search coils, and, by proper amplification and conditioning of the signals, the broken wire is detected.

For the DC unit, there must be relative motion between the sensor coil and the wire rope. This means that the rope must travel through the sensor head or, for a stationary rope, the sensor head must travel along the wire rope. A minimum velocity of about 50 fmp is required.

Below this speed the induced voltage in the sensor coil is too small to detect broken wires. The velocity must also remain constant for signal strength to be consistent; however, to account for changes in velocity, the DC unit is built with a tachometer coupled to an amplifier so that signal strength can be amplified for changes in velocity.

Two search coils are usually built into the sensor head, as shown in Figure 1-22, to allow the head to clamp around the rope. Data output can take several forms because signals from two search coils are available. Usually two output traces are shown so signals from coil A and B can be displayed as a combination of A, B, A + B, (A + B)2 or AB. Typically the data is displayed as A + B on one trace and (A + B)2 on the second trace.

<u>AC Unit</u>: A relatively weak alternating magnetic field is produced by electromagnets in an AC unit sensor head. These magnets function as the primary coil of a transformer (Figure 1-23)[1]. The wire rope serves the purpose of the ferromagnetic core of a transformer. A secondary coil in the middle of the sensor head produces an output voltage that is proportional to the magnetic flux “flowing” through the wire rope. Variations in the cross-sectional area of the wire rope influences the strength of the magnetic flux field and, thus, the strength of the output

[1]Material for Figures 1-18 to 1-23 supplied by ROTESCO, Inc Lakewood, CO.

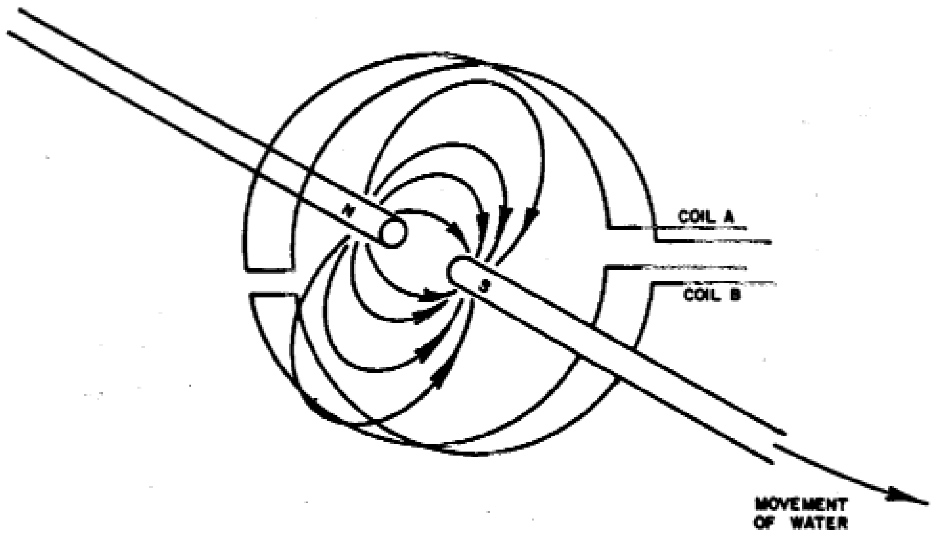


FIGURE 1-22

SEARCH COILS PICK UP RADIAL COMPONENT OF FLUX LEAKAGE

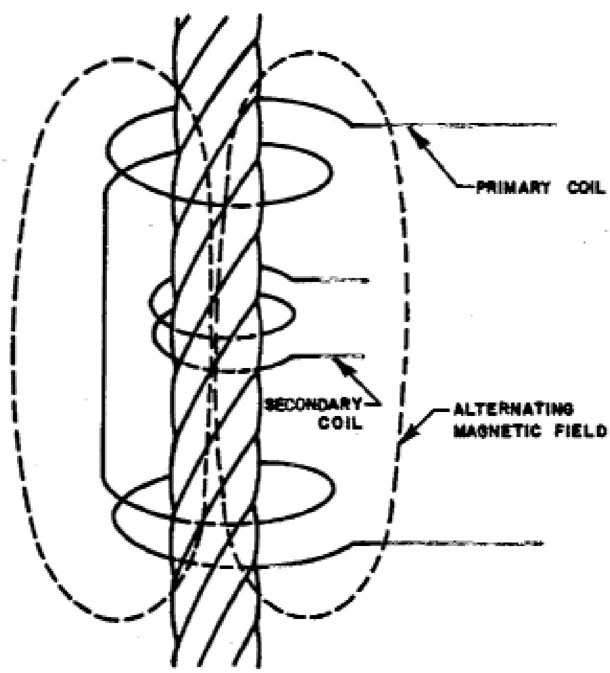


FIGURE 1-23

THE BASIC PRINCIPLE OF THE AC METHOD

voltage. Hence, loss of metallic area can be measured by the output voltage.

The sensor coil measures the metallic volume over a 2 to 3-inch length of wire rope. Wear and corrosion can produce a significant volume change within the finite length, but a single broken wire with a small gap between the ends reduces the volume insignificantly. If many breaks occur within the finite length or a wire is missing, then a defect signal may be recorded.

In the AC unit, the magnetic flux field always varies with time because of the alternating field. Hence, a voltage is produced in the sensor coil whether or not the rope moves.

Because of the alternating magnetic field, small electric currents are induced that circulate around the rope axis within and between the wires. These eddy currents also alternate and produce their own magnetic fields which tend to oppose that from the primary field. This opposition produces a phase shift between the peaking of the magnetizing current and that of the sensor coil voltage. Built-in circuits in the instrumentation utilize the phase shift to produce a second data trace. The first data trace, called X, is essentially proportional to the axial component of the flux field in the rope, and therefore, measures loss of metallic area. The second trace, called A, is proportional to the magnitude of the eddy currents and reflects conditions within the rope that cause changes in the eddy currents. Corrosion products or lay tightening or loosening will affect the passage of eddy currents. Thus, by comparing the X and R traces, wear and corrosion can usually be differentiated.

14.2 Unitized AC/DC Unit

The unitized AC/DC unit uses a sensor head having strong permanent magnets to saturate the wire rope with magnetic flux. This is similar to the individual DC unit; however, the means of sensing the faults in the rope is different. Hall effect sensors are used to detect faults.

Hall effect sensors are solid state devices which can detect and accurately measure magnetic fields. Figure 1-24 shows a sketch of a sensor. Electrical wires are bonded to all four sides of a semiconductor chip. A constant current is passed between two opposing edges. The other two edges develop a potential difference when the semiconductor chip is placed in a magnetic field.

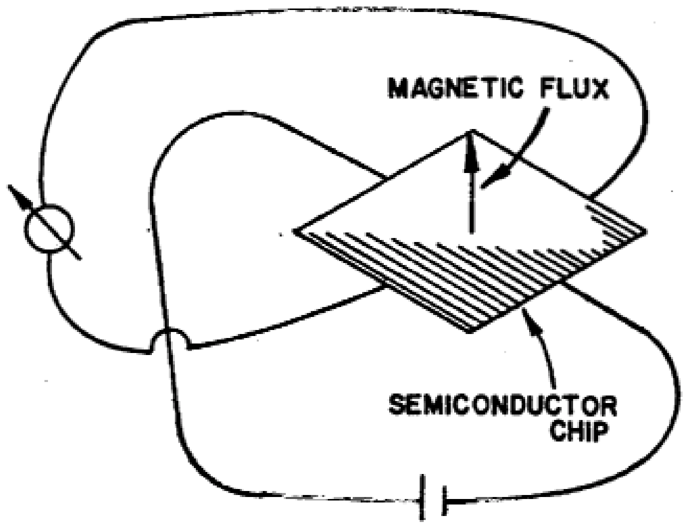


FIGURE 1-24

SKETCH OF A HALL EFFECT SENSOR

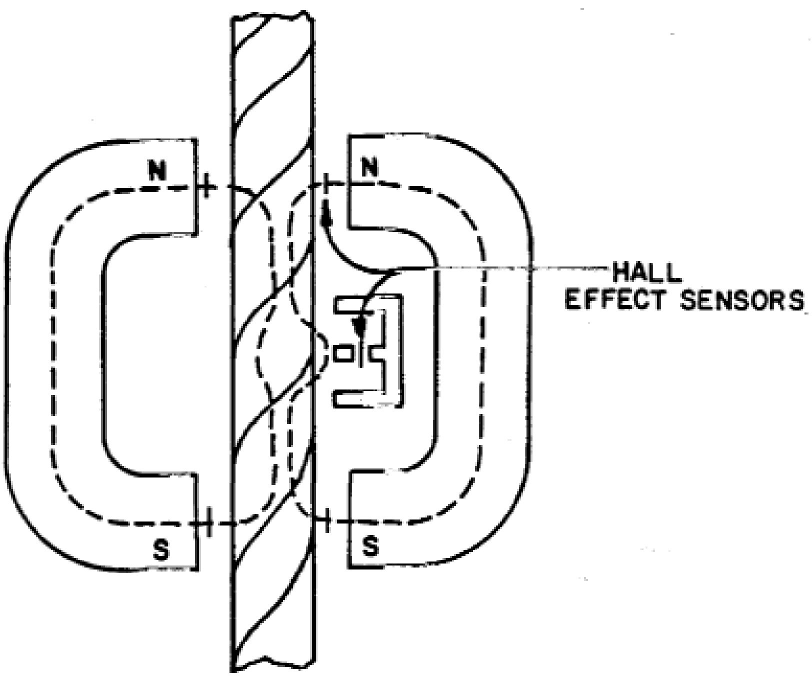


FIGURE 1-25

HALL EFFECT SENSOR PLACEMENT FOR THE MAGNOGRAPH

The potential difference developed by the sensor is directly related to the strength of the flux field. Static magnetic fields can be measured; this is a feature not available in the individual AC/DC units. This means that ropes traveling at extremely slow speeds can be inspected, which is desirable when trying to pinpoint a fault location.

Figure 1-25 shows the placement of Hall effect sensors in the Magnograph sensor head which clamps around a wire rope. The Hall effect sensors located between the poles of the magnets pick up flux leakage, which indicates broken wires or other local faults (or LF). The Hall effect sensors at the poles of the magnets measure the quantity of flux “flowing” into the wire rope. When the cross-sectional area of steel changes, so does the flux “flowing” into the rope; thus, loss of metallic area (or LMA) is measured.

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CHAPTER 2

OCEANOGRAPHIC ELECTRO-MECHANICAL CABLES

Albert G Berian (Reviewed and edited 2000 by Len Onderdonk)

1.0 CONSTRUCTION CHARACTERISTICS

Electro-mechanical (E-M) cables constitute a class of tension members which incorporate insulated electrical conductors. The spatial relationship of these two functional components may be:

1.1 Coincident (Figure 2-1), as in an insulated, copper-clad steel conductor conventionally used in sonobuoy and trailing cables of wire-guided missiles.

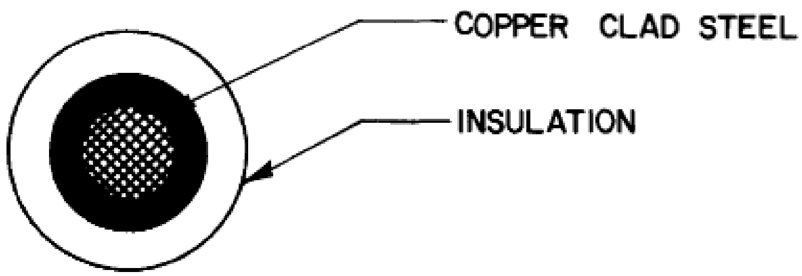
1.2 Center Strength Member (Figure 2-2), such as for elevator traveling control cables. In this, as in most constructions wherein the strength member and electrical component are separate elements, the strength member may be one of several metals or non-metallic materials. Also, the construction of the strength member may be a solid but more generally, it is a structure of metal or yarn filaments. The electrical components of the cable are arranged around the strength member and an outer covering jacket is usually used.

1.3 Braided Outer Strength Members (Figure 2-3), involve a center arrangement of electrical conductors (one, coax, twisted, pair, triad, etc.) with the braided metal or non-metal strength member external to the electrical conductors. Because of the mechanical frailty of the relatively fine filaments a protective covering or jacket is usually required.

1.4 Electro-Mechanical Wire Rope, (Figure 2-4), uses standard wire rope constructions; a three-strand is illustrated. The insulated electrical conductors can be located in two parts of the cross section, in the strand core and in the outer valleys or interstices. When conductors are placed in the outer interstices, a protective covering, or jacket is needed.

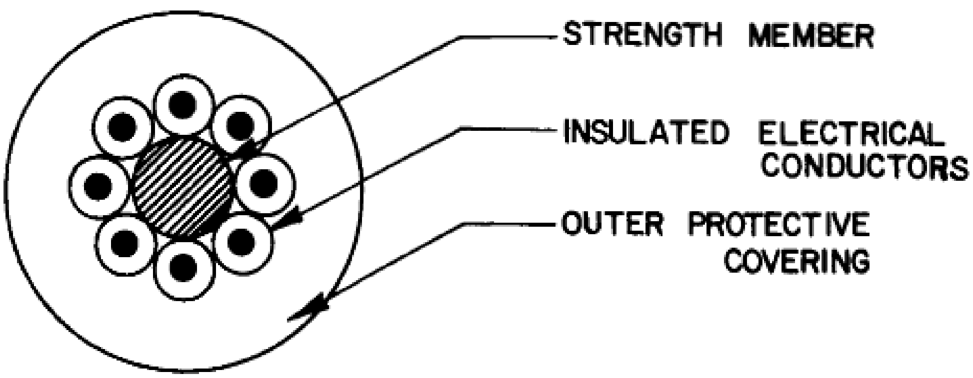
1.5 Outer Single Served Strength Member (Figure 2-5), utilizes metal or non-metal fibers which are helically wrapped around the electrical core which contains the insulated electrical conductors. The metal or non-metal fibers are helically wrapped around the electrical core so that they completely cover the surface. Because this construction has a high rotation vs tension characteristic, it is impractical as a tension member; the wrapping being used to increase resistance to mechanical damage.

1.6 Outer Double Served Member (Figure 2-6), has two helical serves of metal or non-metal fibers which are rapped around the electric cord. The two



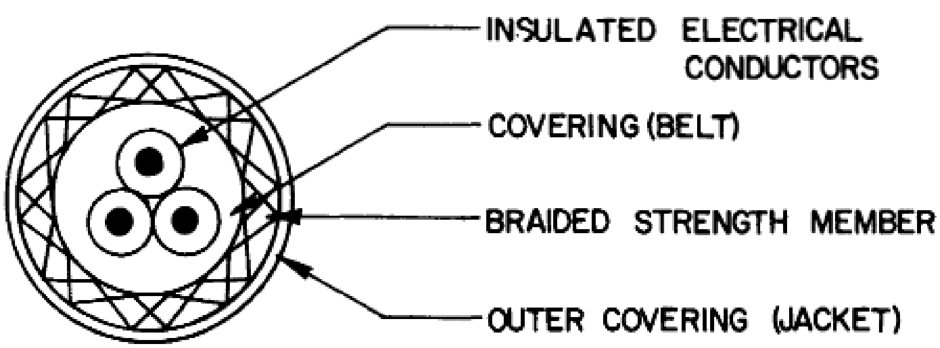
COINCIDENT STRENGTH-MEMBER AND ELECRICAL CONNECTION

FIGURE 2-1



CENTER STRENGTH-MEMBER E-M CABLE

FIGURE 2-2



BRAIDED OUTER STRENGTH MEMBER

FIGURE 2-3

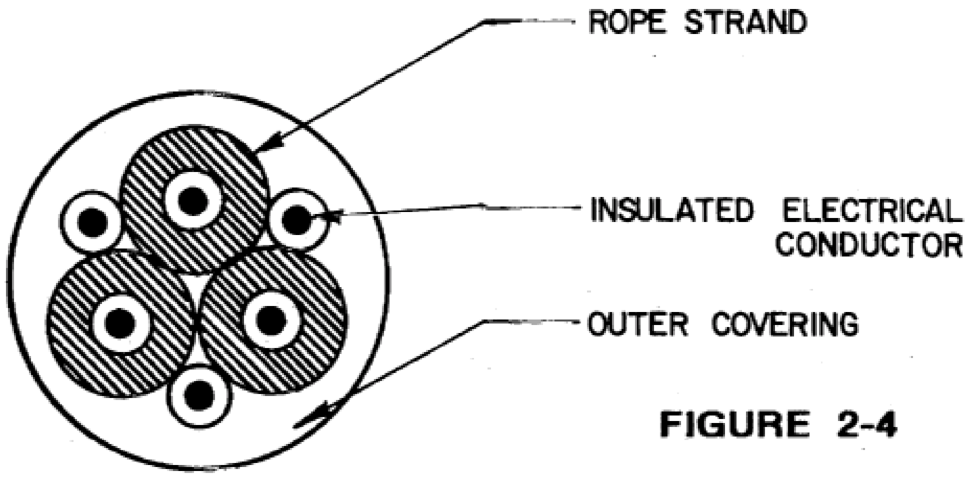


FIGURE 2-4

ELECTRO-MECHANICAL WIRE ROPE

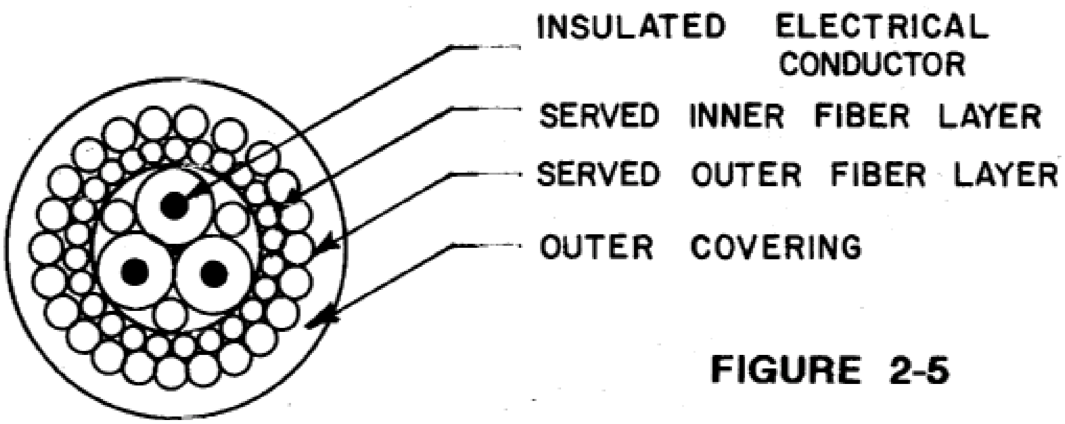


FIGURE 2-5

SINGLE SERVED STRENGTH-MEMBER E-M CABLE

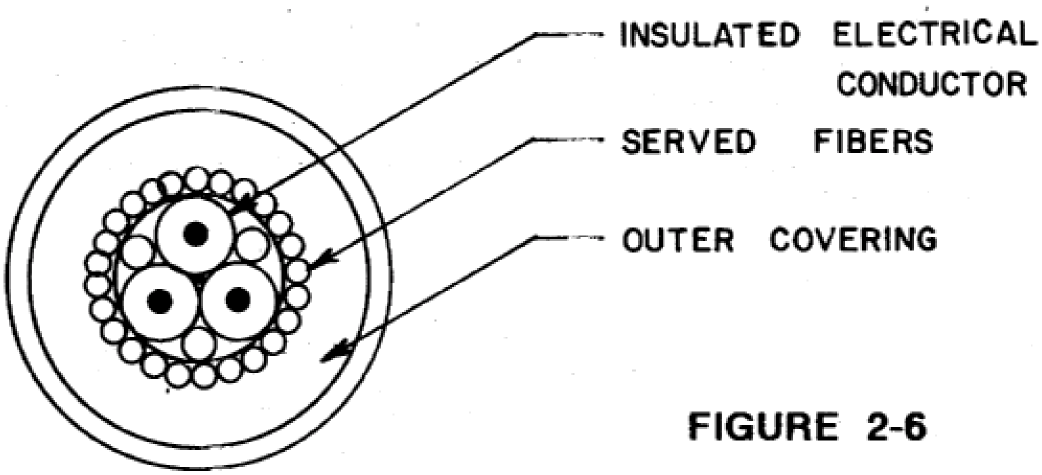
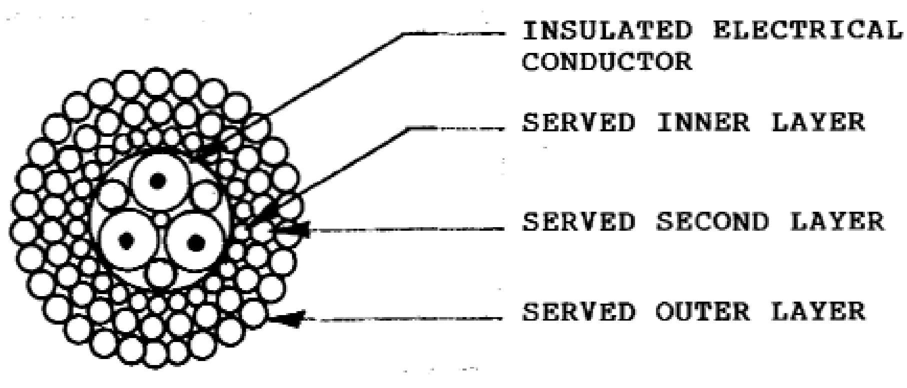


FIGURE 2-6

DOUBLE SERVED STRENGTH-MEMBER E-M CABLE

helical wraps are usually served in opposite directions to obtain a low torque or low rotation vs. tension performance characteristic. An outer covering may be used; its purpose being primarily corrosion protection.

1.7 3, 4, 5 Layer Served Strength Member (Figure 2-7), utilize more layers of the served strength member to increase the ultimate tensile strength, or breaking strength of the E-M cable. The direction of helical serve for a three-layer serve is, from inner to outer serve, right-right-left (or Left-left-right). For a four-layer serve the directions are left-right-right-left, or a combination that permits proper load sharing and package stability.



TRIPLE SERVED STRENGTH-MEMBER
E-M CABLE
FIGURE 2-7

2.0 WORKING ENVIRONMENT

In the above discussion of construction of E-M cables, no mention was made of the working environment, which for this discussion is oceanographic.

The hazards of this environment, which are important to E-M cables, include:

2.1 Flexing

In most applications operating from ships there is constant motion in service with resulting bending of the E-M cable at points of changing direction, such as on sheaves, fairleads, winch drums, capstans, level winds, motion compensators, etc.

2.2 Abrasion

This motion results in the development of two forms of abrasion; between cable internal components and external between the cable and the handling equipment. This abrasion degradation can progress to a point where either a failure occurs or it is observed to be unfit for continued use and is retired from service. The latter is, of course, the more desirable approach.

The rate of abrasive wear varies with several operational factors including line speed, tension, cable to sheave alignment and bend diameter as a ratio of cable diameter. Also, maintenance factors such as allowing abrasive materials (sand, corrosion, etc.) to remain in the cable and maintaining the proper lubrication of rubbing metal parts have a significant affect on the deleterious effects of flexing.

2.3 Tension Cycling

When deployed from a moving platform, the tension in the EM cable will vary constantly. The magnitude of the tension variations can be reduced by use of such devices as motion compensators. Because the E-M cable is an elastic member, it has a tension/elongation characteristic defined by its elastic modulus (see Appendix 1). As the magnitude of stretch varies, the components change their geometrical relationship and create internal friction very much similar to that in flexing. The same damage alleviating and enhancing factors apply as for flexing conditions.

2.4 Corrosion

Applying to metal, primarily steel, this is a major concern in the marine environment. Galvanized steel is, because of its low life cycle cost, the most common metal used for the very common double layer armored cables. The galvanized coating, usually about 0.5 oz./ft^2, is usually electrolytically dissolved very quickly leaving basic steel to be attacked by the sea water. Figure 2-8 shows the equivalent thickness to be about 0.0005 inch. Using an average surface reduction by corrosion for steel of 0.001 inch per year, this thickness would be completely eliminated in six months.

Figure 2-8

Thickness of Zn Coating
on GIPS Armour Wires

Usual specification $= 0.5 \frac{0z}{ft^2} =$

t = thickness (inch)

$\rho =$ density $\frac{lb}{ft^3}$

$$\frac{t}{12} = \frac{\frac{0.5\,oz}{ft^2}}{\frac{16\,oz}{1b}} = 0.3125 \frac{lb}{ft^2}$$

$$t = \frac{12 \times 0.03125}{\rho} \text{ for } Zn_1 = 12 \times 62.4 \frac{lb}{ft^3}$$

$$t = \frac{12 \times 0.03125}{12 \times 62.4} = 0.0005 \text{ in}$$

2.5 Fishbite

This hazard applies to cables having an outer surface which is soft relative to steel. This class of cables include those with extruded outer coverings, or jackets, and those having a covering of braided yarns such as polyester and aramid.

2.6 Abrasion Rate Factor

The rate of this degradation in internal surfaces such as interarmor surfaces can be reduced by maintaining a clean, lubricated condition. On outer cable surfaces accelerated wear is usually the result of improperly selected or installed handling equipment.

2.7 Kinking/Hockling

A kink results when the coil of a cable is pulled to an increasingly smaller coil diameter to the point where permanent deformation of the cable occurs. E-M cables armored with multi-layers of round metal wires are most susceptible to this condition because they usually have a tendency to rotate about the cable axis as tension increases. At high tensions, therefore, a large amount of torsional energy is stored in the cable. At low rates of tension changes this torsional energy will dissipate by counter-rotating the cable about its axis.

At high rates of tension reversals the internal friction of the cable prevents the torsional energy from being dissipated by axial rotation and coils are formed, one coil for each 360° of cable rotation.

2.8 Crushing

The crushing of an E-M cable usually occurs in situations where high compressive forces exist. Crushing can occur on a winch drum when the cable is allowed to random wind and the tension coil crosses over another single coil or when the bed layers are in capable of supporting the cable due to improper spool tension. The high concentration of compression force can cause permanent deformation of metal strength members and other components.

3.0 PARTS OF CONTRA-HELICALLY ARMORED E-M CABLES

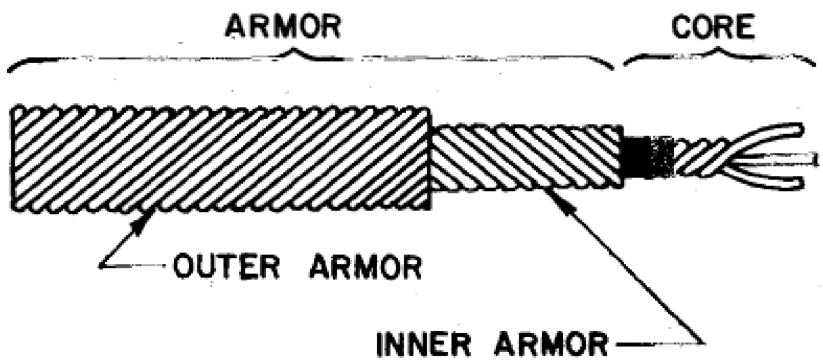
Because over 90% of all E-M cables used in dynamic oceanographic systems use a contra-helical armor strength member, they will be discussed most completely in this chapter.

As shown in Figure 2-9, this type of E-M cable consists of two parts, the core and armor. The core consists of all components under the inner layer or armor. The armor consists usually of two layers of helically wrapped round metal wires, although 3, 4 and 5 layer armors are used. The term contra-helical indicates that the layers have opposing helices.

3.1 Direction of Lay

The convention for determining right-hand and left-hand lay is the direction of the helices as they progress away from the end of the cable as viewed from either end.

The cable shown in Figure 2-9 has a right-hand lay inner armor and left-hand lay outer armor. This arrangement has become an industry standard having its roots in the logging cables used in the oil industry. Because the full splicing of a cable is common practice in the oil industry, standards for armors became necessary. These standards informally developed from usage patterns of the major oil field cable users.



PARTS OF AN ARMORED E-M CABLE

FIGURE 2-9

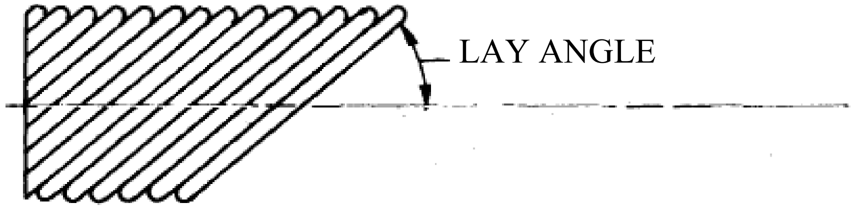
There is no evidence that a right-hand lay outer armor, with a left-hand lay inner armor would not provide the same performance characteristics. Right-hand lay outer armors have been designed and used pending the application and desired performance characteristic.

3.2 Lay Angle

This is the angle the armor helix forms with the axis of the cable as illustrated in Figure 2-10. The magnitude of the lay angle is conventionally between 18° and 24°. Different lay angles may be used for the inner and outer armors, depending on the design characteristic and interrelationship with other cable components.

3.3 Preform

This is a process preformed during armor application to shape the wire in a helical form. Before the armor wires are assembled over the underlying components (core for the inner armor and inner armor for the outer armor) they are formed into a spring-like helix. Preforming wire reduces strain on core components, improves cable flex properties, allows for easier handling and termination, and reduces the stored energy (torque) within the wire.



ARMOR LAY ANGLE
FIGURE 2-10

3.4 Height of Helix

As shown in Figure 2-11, the height of the helix of the coils is determined by the internal diameter of the coil.

3.5 Percent Preform - The ratio for the diameter of under-lying surface to the height of preform is termed the percent preform.

Example: Core dia. = .320

Height of preform = .240

$$\% \text{ Preform} = \frac{.240}{.320} \times 100 = 75$$

A 70% to 80% preform is used in current practice. Note that zero armor compression onto underlying components at 100% preform; a highly undesirable condition.

3.6 Length of Lay

The length of the helix to encompass a 360° traverse is termed the length of lay. This crest-to-crest dimension is shown in Figure 2-11.

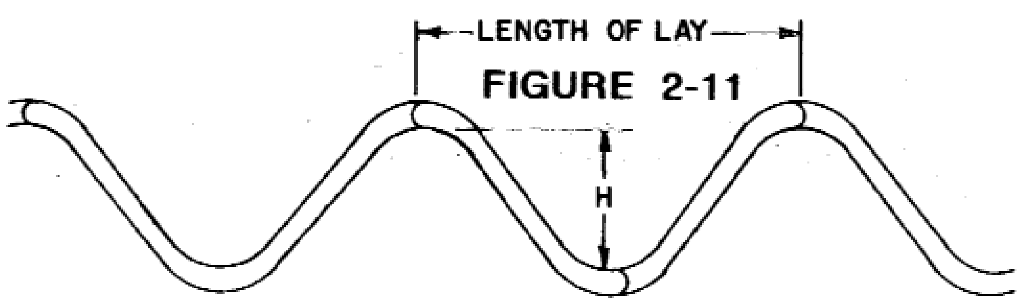


FIGURE 2-11

HEIGHT OF ARMOR HELIX AND LENGTH OF LAY

3.7 Pitch Diameter

This dimension is the diametrical distance between the center lines of the coiled wires. This dimension is illustrated in Figure 2-12 for the inner and outer armor wires.

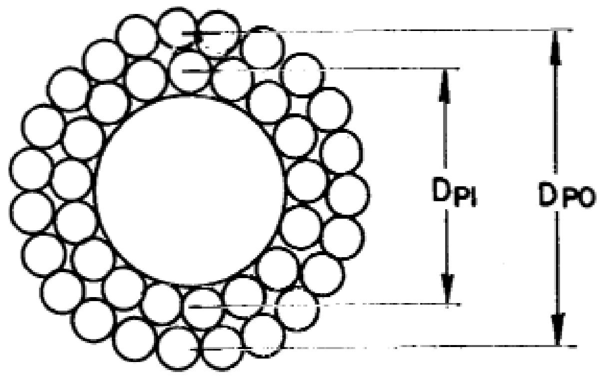
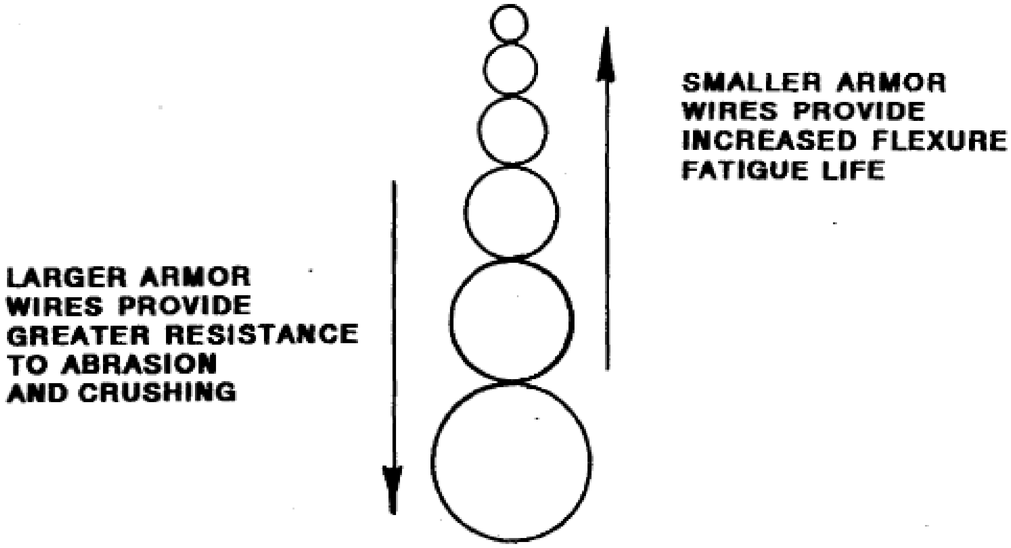


FIGURE 2-12 PITCH DIAMETER

3.8 Number of Armor Wires

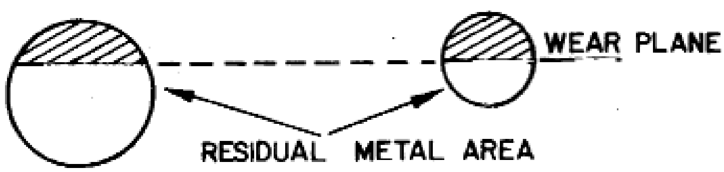
The number and diameter of armor wires are selected to cover 96%-99% of the surface or as determined by the application. There is a balance between the number and size of wires to obtain this coverage. As illustrated in Figure 2-13, for the same pitch diameter and metal type the larger diameter armor wires provide greater mechanical stability; this stability relates both to resistance to distortion and to abrasion. The residual metal remaining after the same diametrical reduction by abrasion on large and small armor wires is illustrated in Figure2-14. The percent residual metal and therefore, strength of the larger armor wires is greater.

But, for the same pitch diameter and metal type, the smaller armor wires offer a greater flexure fatigue life. As illustrated in Figure 2-15, the smaller diameter armor wires will have the smaller outer fiber stress; they will, therefore, have a greater flexure fatigue life.



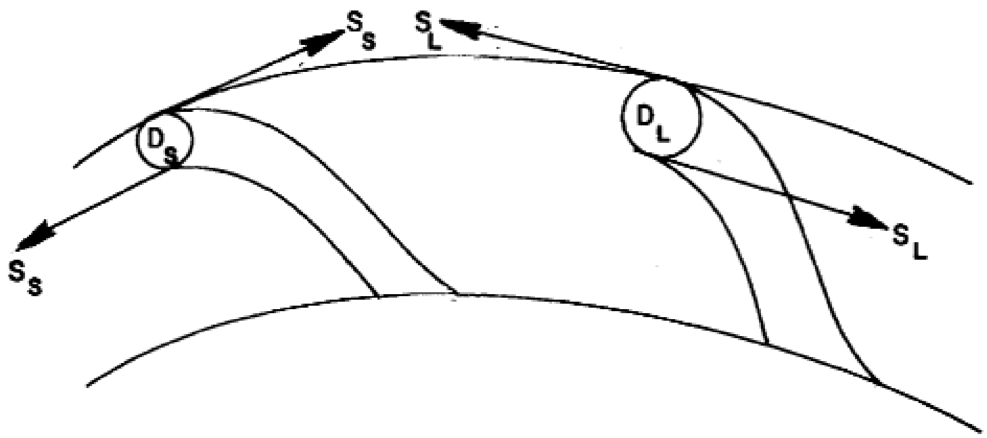
PERFORMANCE RELATIONSHIPS
OF ARMOR WIRE DIAMETERS

FIGURE 2-13



EFFECT OF WEAR ON ROD METAL AREA
OF LARGE AND SMALL DIAMETER ARMOR WIRES

FIGURE 2-14



SMALLER OUTER FIBER STRESS

IN SMALL DIAMETER ARMOR WIRES

FIGURE 2-15

3.9 Armor Coverage

The circumference of the cable is not completely covered by the armor wires; instead, a space is allowed. This space permits greater relative movement of the individual armor wires as the cable is flexed. Also, this space permits settling of the armor layers to a smaller diameter, a natural transition for E-M cables, without overcrowding the armor wires. In a greatly overcrowded condition there will be insufficient space for all armor wires and one or more will be forced out to a large pitch diameter. In this position the wire will be higher than the others and, therefore, much more subject to snagging and increased wear; it is termed a high wire. A normal coverage is about 96% to 99%.

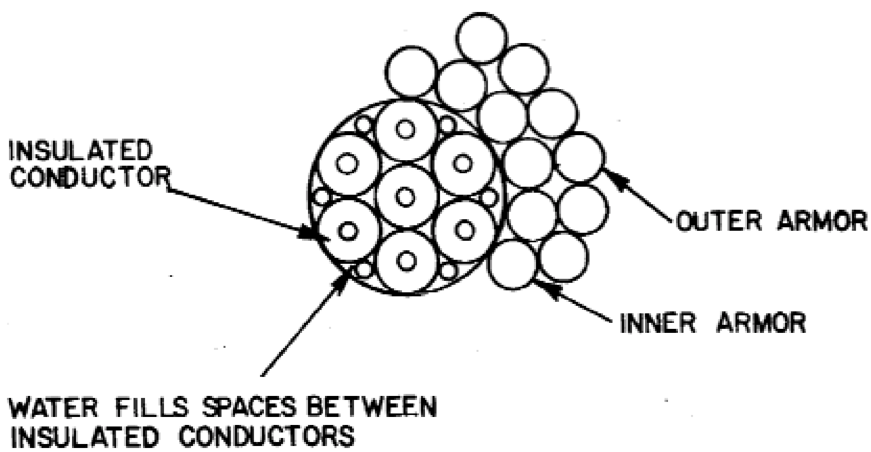
3.10 Cable Seating

The tendency for the high compressive forces caused by the low, circa 70% - 80%, preform to settle the inner armor into the core is termed seating. It results from the plastic deformation of the jacket or insulating surface. While much of this cable seating occurs during manufacturing and post-conditioning, it progresses during the early part of the usage period and is highly dependent on operational loads. The diametrical decrease resulting from cable seating varies depending on end-use and operational scenarios.

3.11 Core

The core may be of two general types, free-flooding or jacketed.

a. The free-flooding type of core is commonly used for oil well logging where the environment media is a mixture of oil and water at pressures which can exceed 20,000 psi. As shown in Figure 2-16, water is free to migrate through the internal parts of the core, filling the internal voids or interstices. A free-flooding cable is considered very reliable because each component is designed to be pressure-proof. Failure of one component, therefore, does not affect the function of others.



FREE FLOODING E-M CABLE

FIGURE 2-16

b. In a jacketed core a pressure-restricted covering is applied on the outside surface as shown in Figure 2-17. The function of the jacket is to form a pressure-restricted barrier against the intrusion of water or other media into the internal parts of the core and to act as an additional support layer for subsequent layers.

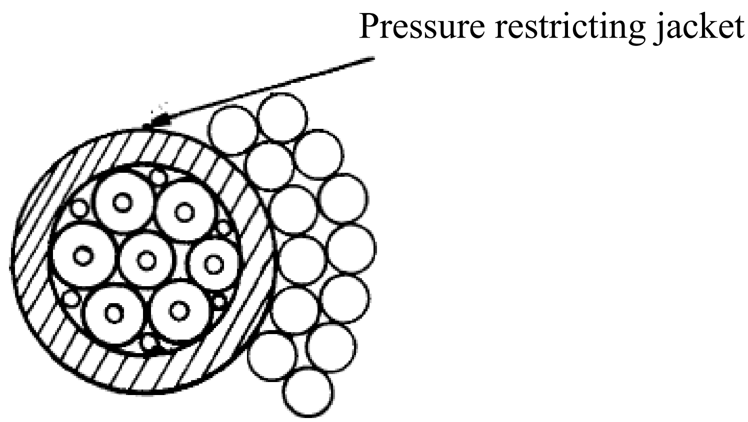


FIGURE 2-17 JACKETED CORE

3.12 Void Filled

This term designates the type of core within which the interstitial spaces are filled with a soft material that could be depolymerized rubber, silicone rubber, and/or cured urethane (today there are many materials available for this purpose, each selected based on the final application). The purpose of this filling can be one of several, the primary one being the restriction of water migration axially within the core in the event of a rupture in the jacket. This filling of the interstitial voids has another benefit; it increases the compression modulus of the core as well as decreases permanent deformation of the structure.

Other parts of the core may also be void-filled. The braided or served outer conductor of a coaxial core may be so treated as may the conductor stranding. The latter measure is infrequently used; the rationale being that cable damage severe enough to penetrate the conductor insulation has rendered it inoperable.

4.0 PERFORMANCE CHARACTERISTICS OF C-H-A, E-M CABLES

4.1 Torque Balance.

This term relates to the ratio of the torque in the outer armor to that of the inner armor. Each armor unrestrained will tend to unlay; i.e., uncoil as tension is increased. The first order equation which provides a figure of merit called torque ratio (Rt) is:

$$Rt = \frac{N_0 d_0^{\,2} D_0 \sin q_0}{N_I d_I^{\,2} D_I \sin q_I}$$

where

N = number of wires per armor layer

D = armor wire diameter

D = pitch diameter of armor layer

θ = lay angle

subscripts

0 = outer armor

I = inner armor

The derivation of this equation is shown in Appendix 2. The torque ratio of most oceanographic cables is between 1.5 and 2.0. With a trade-off for other performance factors, the torque ratio can be reduced to one. Today, with the availability of proven software packages the design engineer can evaluate cable rotation, torque, elongation and a variety of other characteristics to assure the functionality of the product meeting the desired requirements.

But caution must be used because:

- the above torque ratio calculation applies at one tension only; as tension is increased the magnitude of both the pitch diameter, D, and the lay angle, θ, will decrease.

- to decrease the torque ratio, Rt, a larger number of smaller diameter outer armor wires relative to those of the inner armor is necessary. This results in the trade-offs discussed under "Number of Armor Wires," Section 3.8.

The effect of the number of armor wires on the armor ratio equation is illustrated in Figure 2-18. The data in the chart was taken from a selection Of cables currently used in oceanographic applications. The expected trend toward a unity value of armor ratio as the armor wire factor increases occurs because the:

$$\frac{d_0^{\,2}}{d_I^{\,2}}$$

ratio becomes unity, or in extreme torque balanced cables may become less than unity.

The $\frac{D_0}{D_I}$

ratio becomes very small as the diameter of armor wires (d) decreases relative to the pitch diameter (D).

The $\frac{\sin\theta_0}{\sin\theta_I}$

ratio usually varies only between 0.72 and 0.83, a 15% range. So the number of armor wires is the predominant factor in determining the armor torque characteristic.

Cable O.D. (in.)	Inner Armour Wires (no./Dia-In.)	Outer Armor Wires (No./Dia-In.)	Armor Ratio
.125	12/.017	18/.017	2.2
.292	18/.028	18/.0385	2.6
.349	18/.037	24/.037	1.6
.680	22/.065	36/.050	1.04

FIGURE 2-18 EFFECT OF NUMBER OF ARMOR WIRES ON ARMOR RATIO

A development of an equation expressing the torque of each armor layer and the net unbalanced torque is shown in Appendix 16.0.

4.2 Twist Balance

As compared with torque balance which is a potential energy function twist balance is a kinetic energy function. The two are related in that a cable having a lower net torque can be expected to have a lower rotation vs tension characteristic. In general this is the case, but not in a direct ratio.

An important axiom to emphasize is that the three and four armor layers must counterrotate relative to each other for cable rotation to occur. Some factors which will decrease rotation relative to torque include:

- high armor interlayer friction
- extruded outer jacket material entering the armor interstices (cusps)
- foreign matter entering the inter-armor interstices
- a well-conditioned armor wherein the pitch diameters of both layers have reached a stable value and there is intimate contact between the inner armor and core and between the two armor layers.

4.3 Crush Resistance

This external force varies in the manner of application; it may be:

a. across one diameter as would occur by a heavy object hitting the cable when it rests on an unyielding surface,
b. uniform radial pressure such as occurs on the underlying layers of cable spooled under tension,
c. random hydrostatic stress such as would occur on a bottom layed cable on a shifting rocky bottom,
d. self-deformation caused by the load end of the cable crossing over a stray loop on the drum,

e. point or line contact such as would occur when a cable displaces from a sheave groove and bore on the lip of the groove while under high tension.

The crush resistance of a cable increases with the use of larger diameter armor wires as depicted in Figure 2-13.

4.4 Corrosion Resistance

This form of armor degradation in sea water is usually associated with steel but it also occurs with various types of stainless steels. E-M cable design techniques to minimize or eliminate corrosion problems include:

a. isolation from the media by use of a covering jacket over the armor

b. use of a corrosion-resisting metal for the armor wires,

c. Avoiding stainless steels whenever possible. The common types of ferritic (400 series) and austenitic (300 series) stainless steels have been found to be very ineffective for armoring materials. In addition to providing a lower ultimate tensile strength (UTS), they suffer severe pitting, referred to as crevice corrosion. This condition is aggravated by a low oxygen level in the water and is most severe in areas where there is stagnant water. Stainless steels depend on the maintenance of a self-repairing oxide coating for protection against corrosion and failure to maintain this protective coating causes severe localized metal removal by corrosion.

e. (higher alloy metals) Because of the relative low cost of galvanized improved plow steel (GIPS), the most commonly used armor metal, higher alloy stainless steels have been found cost effective in very few oceanographic cable systems. The properties of some metals which have been shown to have good corrosion-resisting properties in sea water are presented in Figure 2-18a.

A vital factor in the evaluation of cost-effectiveness of these higher cost alloys is the relative importance of corrosion among other cable life limiting factors such as:

- flexure fatigue
- handling damage
- abrasion

f. Factors affecting GIPS Corrosion. Because GIPS is the most commonly used armoring metal it is appropriate to examine factors which can affect the corrosion rate in sea water.

Metal	Cost	Corrosion
GIPS	10	1
Nitronic5O(1)	4	3
AL-6X(2)	3	4
MP-35N(3) multiphase	1	10
Inconel 625(4)	2	6

Figure 2-18a: COMPARATIVE COST/CORROSION RESISTING METALS (1 = greatest; 10 = least)

Trademarks:
(1) Armco
(2) Allegheny Ludlum
(3) SPS Co.
(4) INCO Alloys International

The corrosion rate of GIPS in sea water could be increased by:

- stray electric fields causing electrolysis,
- connection to system parts containing materials which are higher in the electromotive series thus rendering the steel sacrificial.

g. Decreasing GIPS Corrosion. The sea water corrosion rate could be decreased by:

- using a fresh water rinse and a relubricating procedure after retrieval from salt water,
- ensuring that the steel armor is at ground potential by the proper use of grounds within the system,
- use of sacrificial zinc anodes at the terminations.

4.5 Abrasion Resistance

This is a metal removal degradation which can be greatly minimized by the use of proper handling equipment. Common causes of excessive abrasion include:

- improper fitting sheave grooves
- rough sheave groove surface
- cable allowed to rub against stationary surface
- unnecessary dragging of cable on the sea bottom

A technique for markedly decreasing sheave groove induced abrasion is the coating of the groove surfaces with a material such as polyurethane or Nylon 12.

4.6 Elongation

The percent elongation at 50% of UTS for sizes of cables which are typical to oceanographic use is listed in Appendix 17. This characteristic applies after length stabilization as described under "Prestressing" in the Manufacturing Process Section and for the same diameter of cable, will vary with:

a) core softness

b) armor tightness

c) armor construction

4.7 Sea Water Buoyancy

This buoyancy becomes more important as the immersed volume (length X cross-sectional area), increases. Calculation for weight in water, specific gravity, and strength to weight ratio are shown in Appendix 18.

4.8 Breaking Strength

Assuming the full conversion of armor wire strength to cable strength the cable strength becomes the sum of the strengths of the armor wires or:

$$Pc = \Sigma P_0 + \Sigma P_I \quad 1.$$

P = breaking strength of armor wire
Subscrips
0= outer armor
I = inner armor

The component of armor wire tension which is parallel to the cable axis is

$$P = P\omega \cos \theta \quad 2.$$

Where: θ = lay angle
$P\omega$ = wire strength

The wire strength is

$$P\omega = \frac{\pi}{4} d\omega^2 S\omega \quad 3.$$

where: $d\omega$ = the armor wire diameter
$S\omega$ = wire tensile strength

substitute 3. into 2.

$$P = \frac{\pi}{4} d\omega^2 S\omega \cos \theta \quad 4.$$

Substitute Eq. 4. into Eq. 1. for the inner and outer armor wires:

$$Pc = \frac{\pi}{4} (N_0 d_0{}^2 S_0 \cos \theta_0 + N_I{}^2 dS_I \cos \theta_I) \quad 5.$$

This ignores the effects of contact stresses.

4.81. The armor wire diameter is determined by equating the circumferential length at the pitch diameter to the sum of armor wire diameters, or:

$$L = \Sigma W_c \quad 6.$$

where L= circumferential length at the pitch diameter
ΣW_c = space occuppied by wire

The circumferential space occupied by each armor wire is:

$$W_c = \frac{d}{\cos\theta} \qquad 7.$$

and the sum of all wires is:

$$\sum W_c = \frac{Nd}{\cos\theta} \qquad 8.$$

The circumferential length at the pitch diameter:

$$L_c = \pi D \qquad 9.$$

which is decreased to allow for coverage, C(%) and

$$L = \frac{\pi C D}{100} \qquad 10.$$

Substitute 8. and 10. into 6.

$$\frac{\pi C D}{100} = \frac{Nd}{\cos\theta} \qquad 11.$$

Solve for d:

$$d = \frac{\pi C D \cos\theta}{100 N} \qquad 12.$$

The use of Eq. 12. above to determine the diameters of the inner and outer armor wires and subsequent use in Eq. 4. will yield a relationship between cable breaking strength and cable O.D. This relationship is shown in Appendix 20.

5.0 MANUFACTURING PROCESSES FOR E-M CABLES

The processes used to manufacture E-M cables differ from those used for general industrial cables in that much greater care in quality control is mandatory. This greater attention to ensure the design integrity of components, subassemblies and the final product is necessary because of the high mechanical stresses which are imposed by the armor and by the system use of the cable.

5.1 Conductor Stranding

To decrease fiber bending stresses, the electrical conductors of E-M cables are stranded; i.e. they contain several individual wires, common strandings being 7-19, and 37. The lay-up is usually "bunched" which means that all wires are twisted in the same direction. Properties of copper conductors commonly used in oceanographic cables are shown in Appendix 3.

5.2 Insulation

The majority of electrical insulating materials are thermoplastics with the most commonly used being ethylene propylene copolymer (polypropylene), polyethylene, and fluoropolymers. These thermoplastic materials are supplied in granular pellet form. These pellets are put into an extruder which melts them and feeds the melt to the extruder head where the semi-liquid thermoplastic is formed around the conductor wire as it traverses through the extruder die. The coated wire is then cooled in a long trough filled with flowing water.

Tests which are usually conducted at this stage are insulation diameter and electrical integrity by means of a spark test.

a.) Diameter measurements are electro-optically made in two orthogonal planes by electro-optical instruments, laser based instruments being popularly adopted.

b.) The spark test consists of electrically stressing the insulation by a voltage generally in the region of 6,000 to 14,000 volts. The purpose is to induce an insulation breakdown where a weakness may occur. These weak insulating points may be caused by voids (bubbles), inclusions (foreign material) or extreme non-uniformity of the wall
voids (bubbles), inclusions (foreign material) or extreme non-uniformity of the wall
thickness (non-concentricity).

5.3 Wet Test

Insulated conductors are typically subjected to another electrical test while submerged in water. The reel containing the completed conductor is fully immersed, except for the ends, in fresh water to which a chemical (wetting agent) may be added to lower surface tension and thereby improve wetting of all the insulation surface. After soaking for a specified period (4-24 hours), electrical tests are made of:

- dielectric strength (hipot)
- insulation resistance (IR)

The insulation resistance values range above tens of thousands of megohms.

5.4 Cabling

In this process several conductors are twisted together either to form a group which may, in turn, be further cabled with other groups to form the final electrical part of the E-M cable core.

5.4 Braiding

When the electrical core is a coaxial conductor, the outer conductor shield may be braided which is the same construction used on coaxial cables specified in MIL-C-17.

5.6 Serving

Because of the self-cutting tendency of braided copper outer conductors at the wire crossover points served shields have become popularly used. This construction consists of helically wrapping several wires around the insulation in the same manner as armor wires are applied,followed by a metal or metal-coated polyester tape.

5.7 Jacketing

The same extrusion process as used for insulating is also used to apply the jacket over the core and/or the armor. To test the pressure-resistant in-water integrity of jacketed constructions, tank testing is sometimes used. Because of the limited availability of pressure test tanks of sufficient volume and because of the high cost, these tests are usually omitted.

a. One technique sometimes used to increase the reliability of a jacket involves the use of a double layer extrusion. This procedure greatly reduces the chance of a pin-hole, bubble or other flaw in one layer from being coincident with a similar minor defect in the outer layer.

b. Reinforced Jacket. When a two-layer jacket is used there is an opportunity to greatly increase its tensile strength by using an open braid of a high strength fiber over the first extrusion. Candidate materials include polyester or aramid yarns. This jacket construction is called a reinforced jacket.

c. Common thermoplastics used for jackets include:

- polyethylene (high density, low molecular weight)
- polyurethane (polyether)
- Hytrel[4]
- nylon

d. When specifying or selecting a jacket material foresight should be given to the termination procedure. If a potted termination is to be used, the bonding procedure should be established.

5.8 Armoring

When the electrical core is completed it is installed into the armoring machine and the spools of armor wires are loaded into cradles of the armoring machine. Two general types of armoring machines are in common use; (1) the tubular type and (2) the planetary type. Both types are in successful use for producing high quality armors.

a. The tubular armoring machines are favored due to their production efficiency. They operate at up to 1,000 rpm as compared with a usual maximum of 300 rpm for planetary machines.

b. A major consideration during the armoring process is to have sufficient lengths of wire in each spool to make the entire cable. This avoids planned welds in the armor wire. User specifications frequently limit the number of welds in the outer armor layer and specify the minimum distance between welds along the cable.

c. An example specification control of welds is that:

- minimum distance between welds shall be (x) lay lengths.
- no more than one weld in any one armor wire.
- no more than three welds in each armor layer.

d. Armor Wire Welds. Broken armor wires are usually butt fusion welded. The heat of welding anneals the metal in the vicinity of the weld and vaporizes the galvanize coating. The primary function of the weld is to provide a smooth mechanical transition across the broken section. Therefore, the loss of approximately 50% of the unwelded wire strength has a minor effect on the performance capability of the cable.

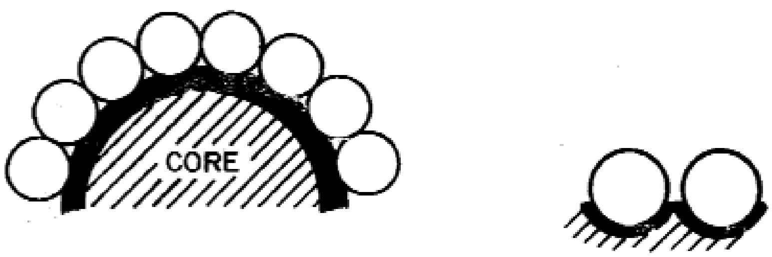
5.9 Prestressing

Prestressing is a term applied to the stabilizing of the construction of an E-M cable; it is also termed length stabilization. When first manufactured, the inner armor wires seat into the underlying thermoplastic insulation or jacket as shown in Figure 2-19. This is an unstable condition because of the very high surface stress which at working loads can exceed the yield strength of thermoplastic.

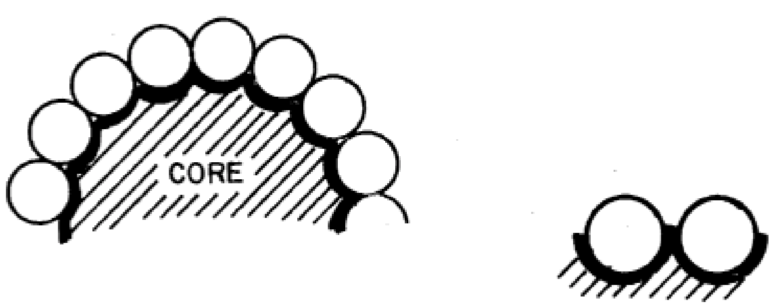
Manufacturers, therefore, may prestretch the cable by passing it over several sheaves at a tension of about 40% of the breaking strength. The equipment used for this operation, called prestressing, varies but functionally includes equipment shown in Figure 2-20.

FIGURE 2-19 GEOMETRY CHANGES DURING CABLE RECONDITIONING

2-19A AS MANUFACTURED THERE IS A SMALL INDENTATION OF INNER ARMOR WIRES INTO THE CORE MATERIAL



2-19B WHEN LENGTH STABILIZED, OR PRESTRESSED, THE INNER ARMOR WIRE CONTACT WITH THE CORE INCREASES



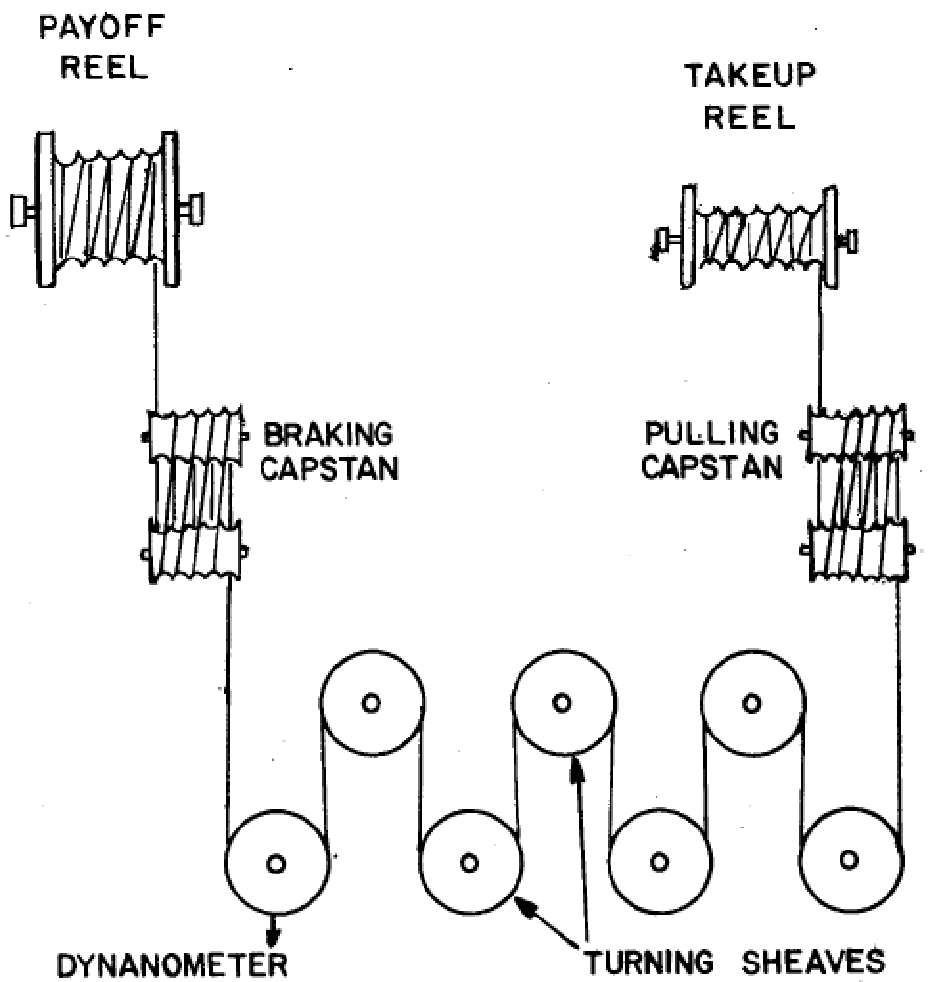
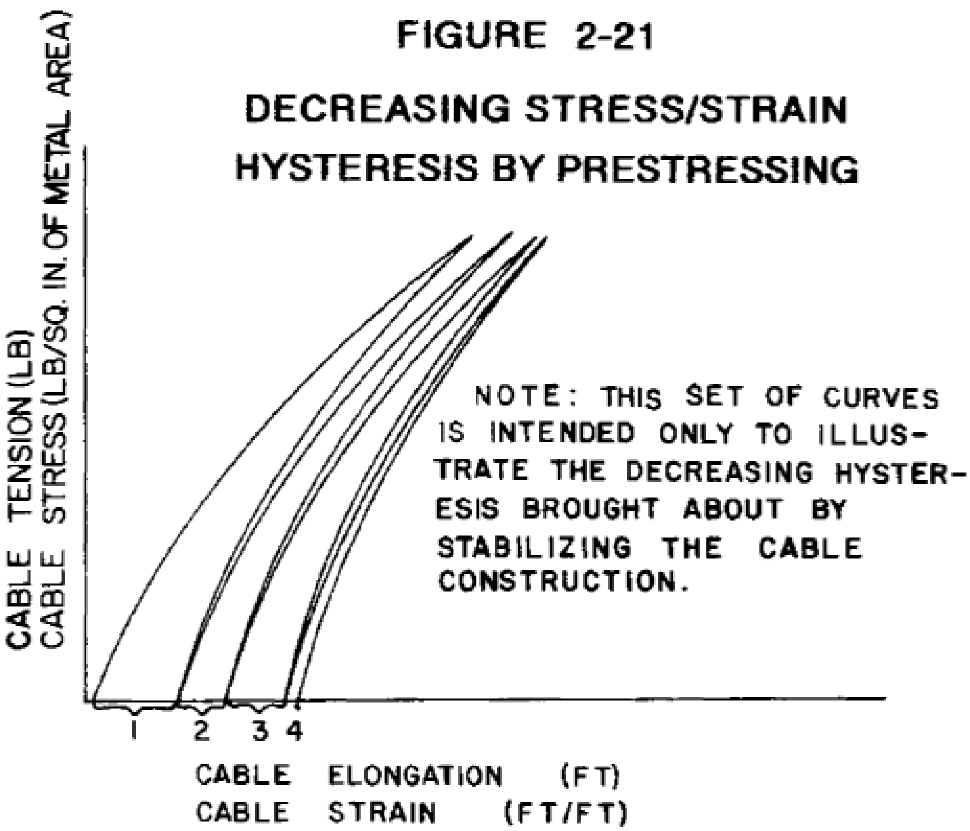


FIGURE 2-20 PRESTRESSING EQUIPMENT

The objective of this operation is to operationally stabilize the cable; i.e., reach a condition wherein the same dimensional parameters and consequent cable stretch will occur on subsequent tensioning operations. This hysteresis phenomena, graphed in Figure 2-21, is decreased as the inner armor contact with the core is increased. This indentation will continue until a contact surface is formed which results in a stable contact stress value.

FIGURE 2-21

DECREASING STRESS/STRAIN HYSTERESIS BY PRESTRESSING



6.0 HANDLING E-M CABLES

The discussion of handling E-M Cables starts with the assumptions that the cable had been properly specified and procured.

6.1 Storage Before Use

E-M cables are usually supplied on heavy duty steel or wood shipping reels. The cable will be uniformly thread-layed on the reel; i.e. it will be tightly coiled with no gaps or crossovers. This practice is to prevent in transit damage to the cable which can occur due to self-crushing at these crossovers.

The reel should always be stored upright; i.e., resting on the two flanges. Storing the reel on the flat of one flange can cause coils to cross over into a random tangle. Subsequent righting of the reel and proper re-reeling of the cable can be very difficult. When stored in an unsheltered area, the reels should be covered with the bottom left open for ventilation. If the storage period is to be more than a few months, the spraying or wiping of an extra amount of lubricant onto the surface layer of cable will provide added protection.

The reel should be lifted by using a bar through the center holes. In no case should fork lift blades bear onto the coiled cable.

6.2 Spooling Effect on E-M Cables

The spooling of a cable onto a storage drum can be performed with only sufficient tension to tightly pack the cable in a thread-lay. A tension of 3% of the cable breaking strength is reason able.

When using a single drum winching system the winching power and storage function are provided by a single unit and the installation of the cable becomes critical. Before discussing spooling procedures, the effects on the cable should be noted. A contra-helically armored E-M cable is most resistant to damage by compressive forces when all components, i.e., two armor layers and core, are intimately in contact so that there is little relative movement from the external force. When these cable parts act as a unit, the resistance to damage by compression is maximized. Also, when the compressive force is distributed around the cable circumference rather than across one diameter, less distortion will result in a much lower damage possibility.

For succeeding layers the tension in the third layer is maintained for about one-half the total cable length.

For the remaining half of the total cable length the tension is reduced in equal increments every 1,000 ft to the first layer value at the outer layer of cable.

6.3 Smooth Drum Spooling

Smooth drum spooling uses a plain cylindrical winch drum and is most commonly used on small oceanographic winches containing 1,000 to 2,000 meters of cable. Because of the low deployment forces involved, the spooling onto these winches is less critical. However, good practice dictates that a uniform thread-lay be used.

6.4 Tension Spooling Objectives

When longer cables are to be handled by a tension winch formalized spooling procedures become mandatory to prevent cable damage. The procedure has three objectives:

a. Tightly thread-lay the cable under tension to ensure that the cable cross-section has resistance to crushing.

b. Provide sufficient rigidity of cable in lower layers to prevent nestling or keyseating of the tension coil.

c. Provide sufficient spooled tension to balance some of the deployment tensions to reduce coil slippage caused by tightening of the tension coil.

6.5 Tensions for Spooling

Spooling tensions vary in succeeding layers according to schedules which vary according to the experience of many able technicians. The schedules shown in Table 1 apply for a selection of small diameter cables. The tensions shown in Table 1 are typical for EM cables similar to those used in oil well work. They will vary for different types of armor.

The schedule shown in Figure 2-22 displaying spooling tensions expressed as a percentage of the cable UTS, or breaking strength, is applicable to a wide variety of cable types.

TABLE 1
Recommended Typical Spooling Tension Schedules

			Spooling Tensions (lbf)		
Cable in.	Dia. mm.	Approximate UTS lbf.	first layer	second layer	third layer
.185	4.7	3,900	650	900	1,050
.203	5.2	4,500	750	1,025	1,200
.223	5.7	5,500	925	1,250	1,500
.250	6.4	6,800	1,150	1,550	1,825
.316	8.0	11,200	1,900	2,575	3,025
.375	9.5	14,600	2,500	3,350	3,950
.426	10.8	18,300	3,000	4,200	4,925
.462	11.7	18,300	3,000	4,200	4,925
.472	12.0	22,200	3,800	5,100	6,000

The schedule shown in Figure 2-22 displaying spooling tensions expressed as a percentage of the cable UTS, or breaking strength, is applicable to a wide variety of cable types.

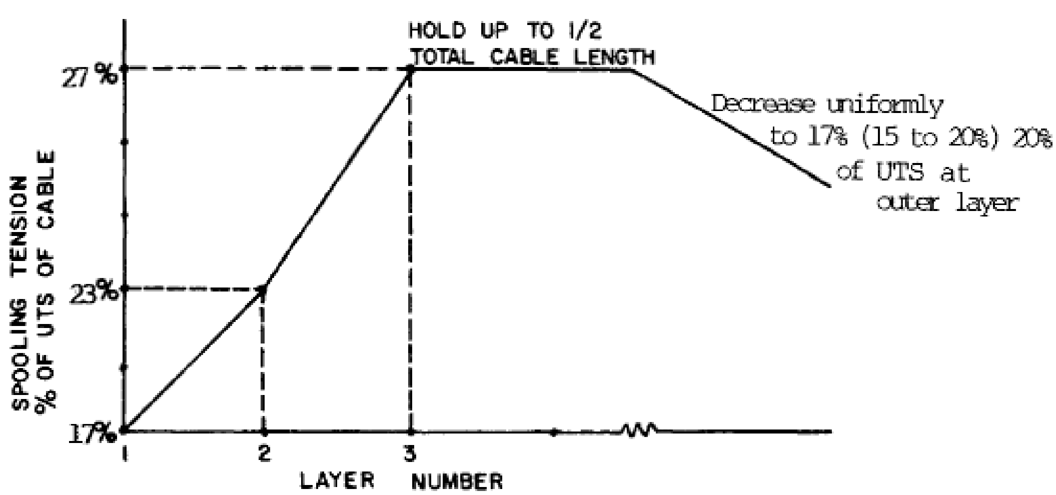


FIGURE 2-22 SPOOLING TENSION SCHEDULE

6.6 Lower Spooling Tensions

It is very possible to obtain satisfactory performance using lower spooling tensions with the provisions:

a. Care is taken that the bed (first) layer is properly started using a sufficiently high spooling tension with coils in tight contact and uniformly distributed across the winch drum.

b. The remaining spooling operation is performed with good workmanship at tensions as close to the recommended values as possible.

c. MOST IMPORTANT! Initial deployments are made at low tensions and slow winching speeds. The tensions can be gradually increased in subsequent deployments.

d. Remember: single winch drum deployments require bed layers to support the operational load. Each time a cable is deployed, the operational load will be profiled back on the winck

6.7 Grooved Drum Sleeves

These grooved drum sleeves, made by Lebus, Inc., are described in a later chapter. Their use is encouraged because they:

a. determine the spooling thread-lay at the bed layer and, therefore, more positively ensure that the remaining procedure will be correct.

b. are similar to correct sheave grooves, these grooved sleeves provide support for the cable to increase its crush resistance.

6.8 Sheaves

The correct sheave groove design as illustrated in Figure 2-23 provides for circumferential cable contact of about 1400 of arc.

a. Groove surface finish: The surface of the groove should be smooth with a surface roughness not exceeding 32 microinches. As grooves become worn, the surfaces will become corrugated and cause accelerated abrasive wear of the cable; they should then be refinished.

b. Hardness of groove surface: The hardness of the groove surface should be less than that of the armor wire which is about Rockwell (C-scale) 55. It is less costly to resurface a sheave groove than to replace an E-M cable. Good results have been obtained with the use of polyurethane coated sheave grooves; other thermoplastics have also been successfully used.

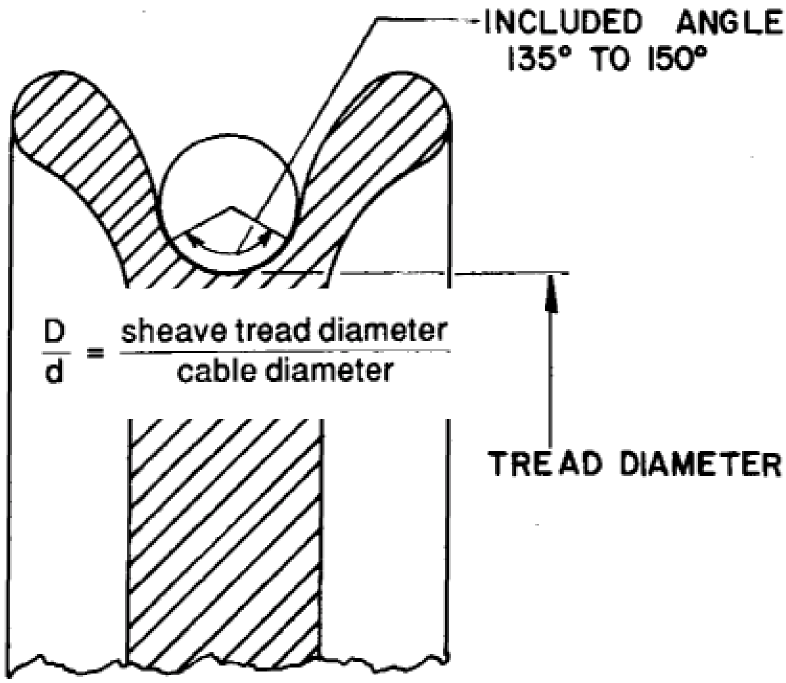


FIGURE 2-23 CORRECT SHEAVE GROOVE DESIGN

c. Tread diameter: The tread diameter of the sheave should be as large as possible, the minimum diameter for a cable following the general rule:

$$D = 400\ dw$$

where

D = sheave tread dia (see Figure 2-23)

dw = largest diameter of armor wires in the cable

d. Bending stresses: The importance of designing an E-M cable handling system to impose a minimum number of directional changes in the cable can be understood by an examination of bending phenomena. To allow a cable to bend, the inner and outer armor layers must move relative to each other. This causes abrasion between the armor layer; i.e., the importance of lubrication. Also, the shortening of the armor lay angle at the sheave entrance point of tangency and normalizing at the exit point of tangency causes a rubbing action between the cable and sheave.

The wear rate resulting from this abrasion will increase:

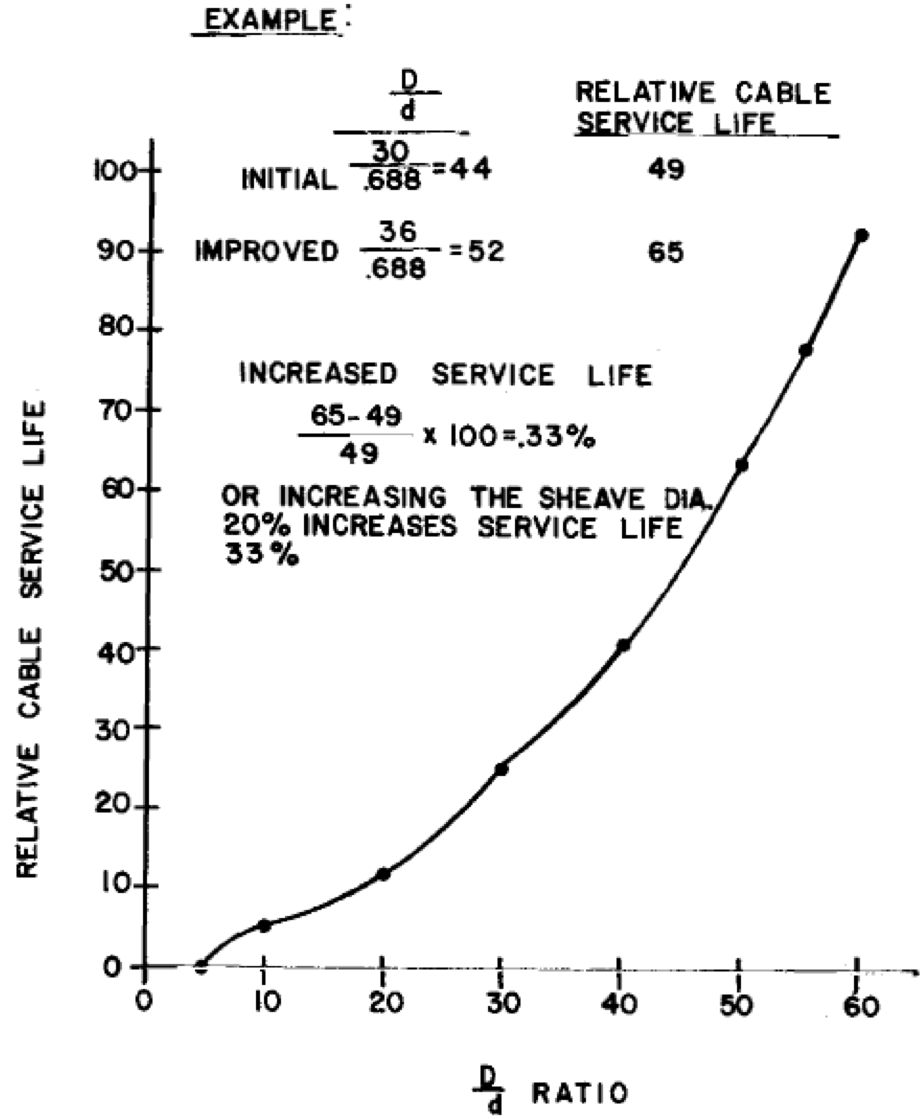
- cable tension
- winching speed,

and a decreased ratio of

$$\frac{D}{d} = \frac{\text{sheave tread diameter}}{\text{cable diameter}}$$

e. Sheave tread diameter effect on flexure fatigue life:

Illustrations of the importance of sheave tread diameter to the flexure fatigue life are presented in Figures 2-24 and 2-24a. These data were generated from tests on wire rope but they generally also apply to CHA cable.



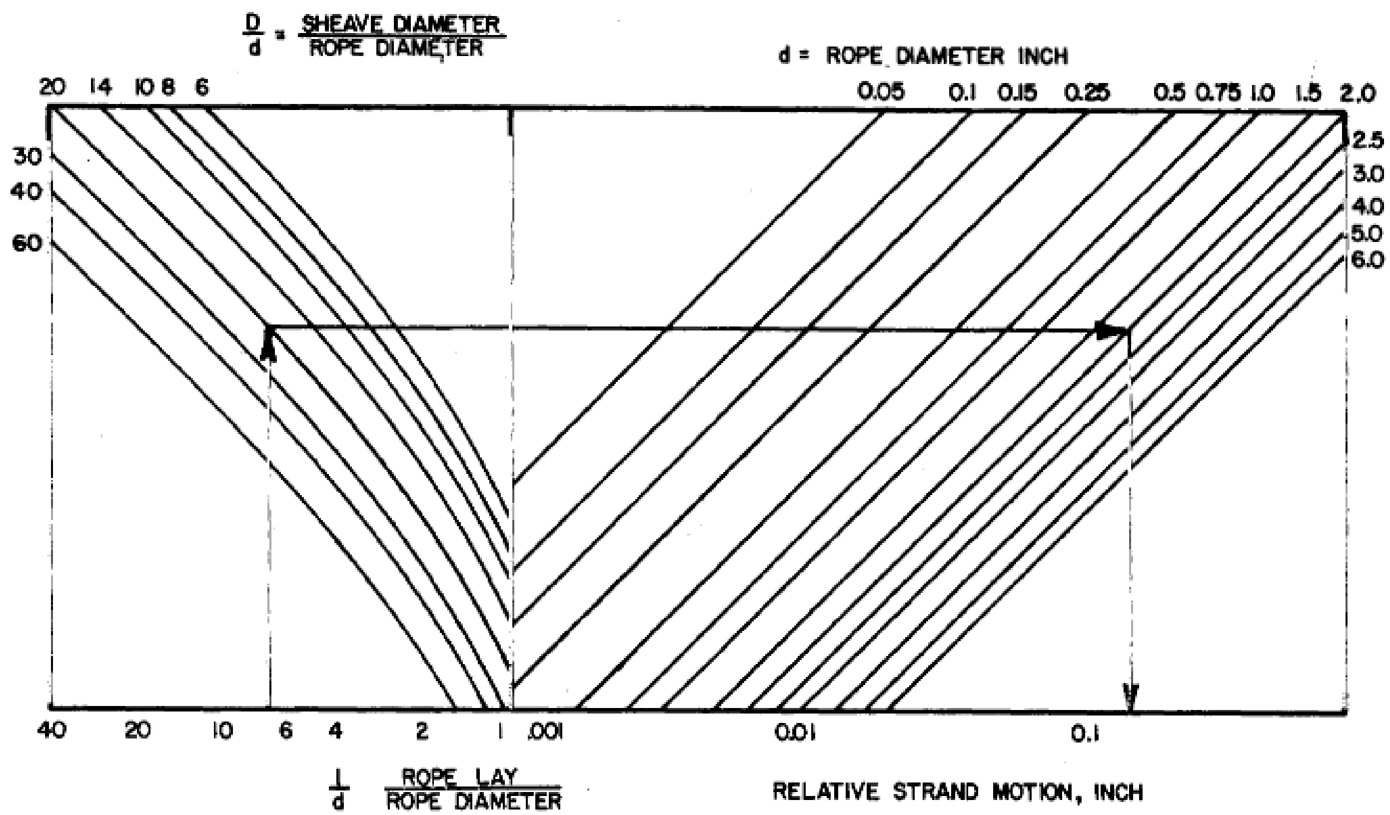
REF: WIRE ROPE USERS MANUAL, 1981
AMERICAN IRON AND STEEL INSTITUTE

RELATIVE SERVICE LIFE FOR VARIOUS RATIOS

$\frac{D}{d}$ = RATIOS

FIGURE 2-24

FIGURE 2-24a INFLUENCE OF ROPE AND SHEAVE GEOMETRY ON RELATIVE STRAND MOTION FOR A SIX STRAND CONSTRUCTION



7.0 FIELD INSPECTION AND TESTING

7.1 General

When an E-M cable has been properly specified (see Section 10.0 of this chapter) the inspections advisable upon receiving it essentially concern verification that the receiving records conform to the purchase specifications.

7.2 Required Inspections

After a cable has been put into service, there is need to use inspection procedures for:

a. location and identification of performance defects which may occur because of cable handling or service oriented accident.

b. monitoring of changes in cable geometry and performance characteristics caused by usage wear. Information from this data can prove to be very valuable in deciding to retire the cable from service.

7.3 Cable Record Book

As is common practice on oil well electrical wireline (oil well logging) trucks which use E-M cables similar to those used for oceanographic instrumentation, a cable record book is used. This is highly recommended for the oceanographic community. It is much more necessary for cables used on oceanographic survey ships because of the rotation of personnel; operating engineers on oil well electrical wireline trucks may use the same E-M cable for its entire life. They, therefore, will know of any special performance characteristics and of the usage history.

Data appropriate for the Cable Record Book and other data arrangements may be found more applicable to certain systems. The important consideration is that a record of cable usage is maintained. The cost effectiveness of E-M cables and the relative performance lives of similar cables in similar systems otherwise cannot be accurately evaluated.

7.4 Cable Log

This parallel record to the record book is a history of the status of cable characteristics and of maintenance and repair procedures.

7.5 Inspection

An E-M Cable should be inspected and undergo maintenance after each sailing or at regular intervals of time. The inspections should include:

a. visual to observe, evaluate, and document any damage or severe wear and log its location in the Cable Log;

b. armor tightness which can serve as a warning that the cable was overstressed or that the cable should have Service Shop maintenance. This latter activity is described in a later section of this chapter;

c. conductor electrical resistance, which can be another indicator of overstressing. Also, it can indicate conductor damage from other causes. In any event, this is valuable data for determining the suitability of the cable for continued use;

d. outside diameter, like outer armor lay length is an indication of length stability. But, when visual examination shows abrasion of the outer surface of outer armor wire, it is an indication of residual metal cross-section and, therefore, residual UTS;

e. need for lubrication, before storage a cable should always be lubricated. Materials and procedures are discussed in another chapter.

7.6 Visual Inspection Practice

This is performed while re-spooling to or from the winch or from a storage reel at a slow, less than 50 ft/min speed. Observe for any changes in the armor which can include:

a. permanent offset, which is an indication that there may have been the onset of a kink or that the cable was overstressed by bending over a very small diameter while under a tension in excess of 10% of UTS;

b. scraped or nicked wires, as evidenced by their being at a larger diameter than the others. This is indicative also of these wires experiencing shear forces caused by the cable being drawn over stationary surfaces. It may also indicate an overtension experience.

7.7 Armor Tightness Inspection

Three methods are possible for this evaluation; they are:

a. pick test which is the easiest but the most qualitative of the three is illustrated in Figure 2-25. If an outer armor wire can be raised above the outside diameter of other outer armor wires, a loose armor condition is indicated. The same test should be performed at several locations to determine if any loose armor indication may be localized.

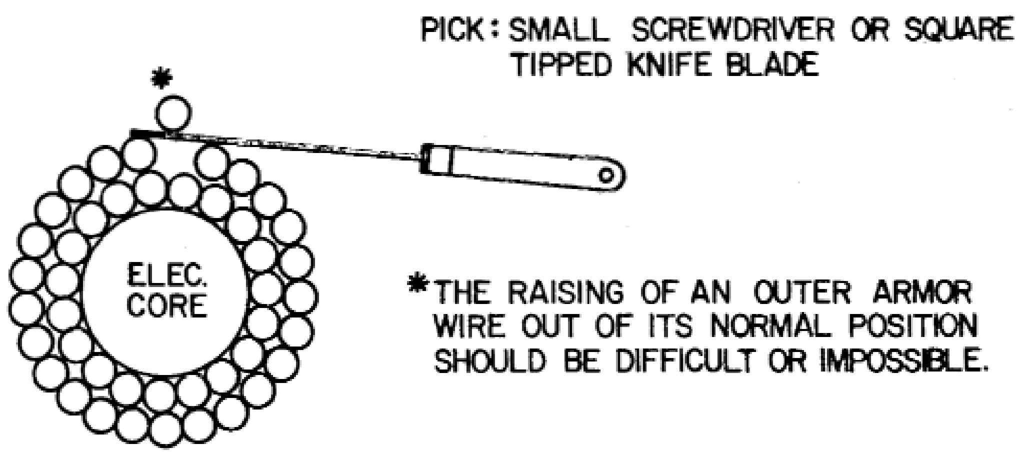
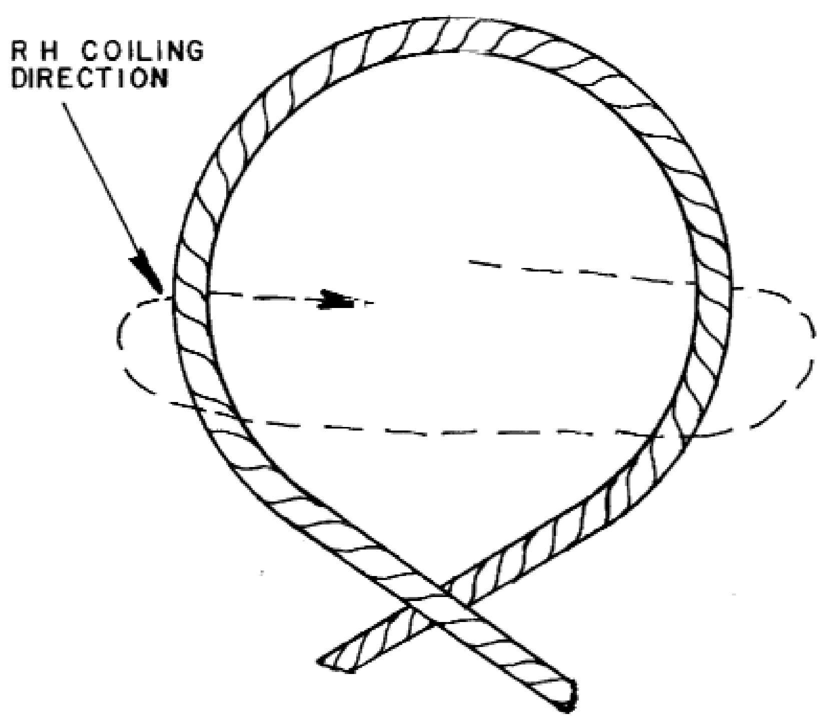


FIGURE 2-25 PICK TEST FOR LOOSE ARMOR DETERMINATION

b. coil test which is performed on a run of cable by forming an in-line coil having a coiling direction which tends to tighten (axially rotate the cable in a direction opposite the outer armor lay direction) the outer armor. For the standardized L-H-L outer armor the coil direction becomes R-H. The coil test procedure is described in Figure 2-26.

FIGURE 2-26 LOOP TEST FOR LOOSE ARMOR DETERMINATION (FOR L-H-L- OUTER ARMOR)

1. OPERATOR STANDS BESIDE A RUN OF SLACK CABLE WHICH IS BY THE RIGHT HAND.
2. WITH THE PALM OF THE RIGHT HAND FACING PWAY FROM THE BODY,THUMS TO THE REAR, THE OPERATOR GRASPS THE CABLE.
3. THE HAND IS TURNED CLOCK-WISE I8Oo OR UNTIL THE THUMB IS POINTING FOR WARD, THUS FORMING A COIL AS DIAGRAMMED BELOW.
4 IF THERE IS A LOOSE OUTER ARMOR THE CABLE WILL REMAIN COILED OR TEND TO CONTINUE FORMING ADDITIONAL COILS.



c. catenary test, which is performed on a run of cable by forming a catenary on a 30 ft. to 40 ft. run. When the catenary is aligned with a vertical reference such as a plumb bob, a deviation of the catenary in the direction tending to tighten (see above) the outer armor indicates a loose outer armor condition. For a L-H-L outer armor a loose armor condition will be indicated by a deviation to the right of vertical; the larger deviations indicating a higher severity of looseness. This inspection procedure is diagrammed in Figure 2-27.

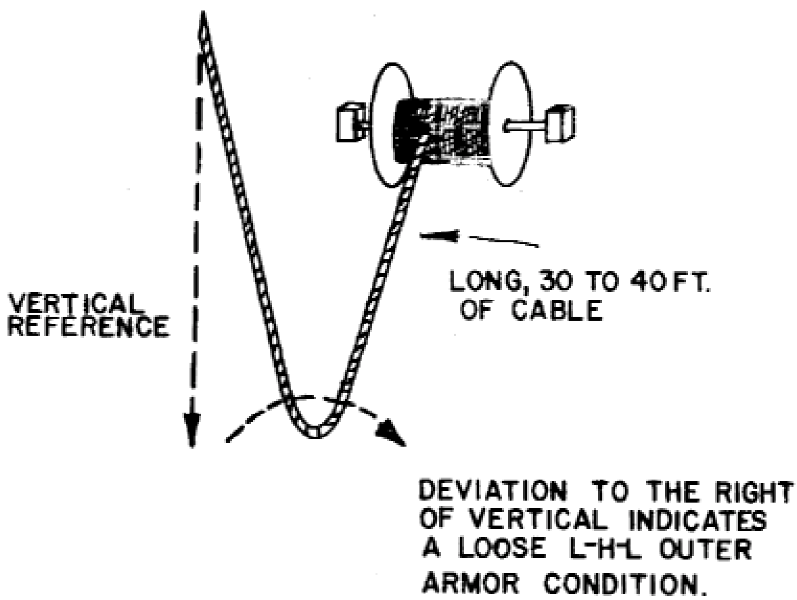


FIGURE 2-27 CATENARY TEST TO EVALUATE LOOSENESS IN L-H-L OUTER ARMOR

7.8 Lay Length of the Outer Armor

a. The general lay length of a single armor wire helix is shown in Figure 2-1 1. This lay length will increase as the cable becomes length stabilized, a natural occurrence in service. The progression of this lay length increase can be used as an indicator of our armor wire loosening or of cable core deterioration. It is therefore, one of the cable's vital signs.

b. Procedure: The procedure for field determination of lay length is shown in Figure 2-28. To obtain accuracy, the rubbing from the cable must be in one plane and the markings of uniform length; the paper must not be allowed to rotate while the rubbing is being made. Also, remember that when a cable is held in a curved position, the lay length on the outside surface of the curve has a longer lay length. Therefore, it is important to perform this inspection procedure on a straight run of cable.

The rubbing is measured by laying a high accuracy scale on the rubbing and reading the scale with optical magnification. Remember, the lay length will vary with tension; therefore, the rubbing should always be taken on the cable while under the same light tension, about 5% of UTS. The accuracy is improved by measuring over several lays and dividing by the number of lays.

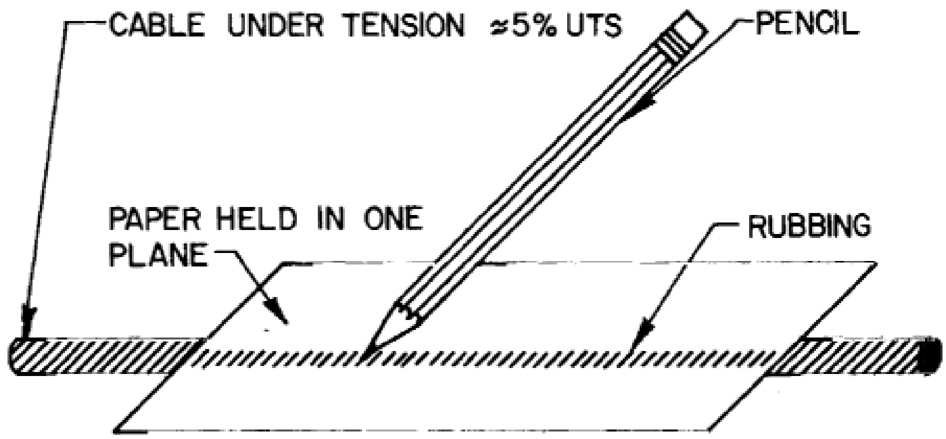
7.9 Conductor Electrical Resistance

a. Procedure: Changes in the conductor electrical resistance may indicate that damage has occurred and an evaluation must be made of the continued usefulness of the cable.

The desirable instrument for this test is a resistance bridge having an accuracy of 0.1%. The temperature of the conductor has a significant bearing on resistance so that the cable must be kept ma shaded, near constant temperature area for a minimum of twelve hours prior to taking this measurement to ensure that the entire length is at a uniform temperature.

The standard temperature for expressing electrical resistance is 20°C (68°F); to convert the electrical resistance taken at any other temperature to the value at 20°C, use the correction factors shown in Appendix 4.

b. Cabled conductor resistance: Remember that the measured electrical resistance is for the conductor in the cable and will apply directly only if the conductor is coincident or parallel, with the cable axis. Otherwise, the readings must be corrected for cabling. Appendix 5 shows the correction which must be made to express electrical resistance as cabled to straight wire electric resistance. When the wire is axially coincident with the cable $\theta - 0$ and $\tan \theta - 1$.

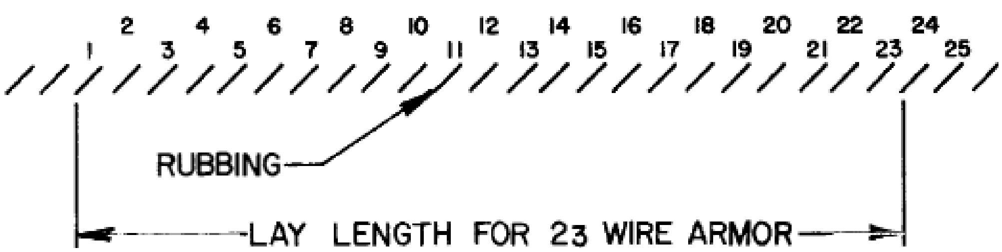


1. LAY PAPER OVER A STRAIGHT RUN OF CABLE.

2. RUB THE PENCIL ON THE PAPER ABOVE THE CABLE WHILE THE PAPER IS HELD IN ONE PLANE.

3. THE RUBBING SHOULD SHOW A MARK FOR EVERY COIL OF THE WIRES. THE MARKS SHOULD BE APPROXIMATELY THE SAME LENGTH.

AN ALTERNATE PROCEDURE INVOLVES INKING THECABLE AND PRESSING IT BETWEEN TWO BOARDS ON WHICH THE PAPER IS PLACED BETWEEN CABLE AND BOARD.



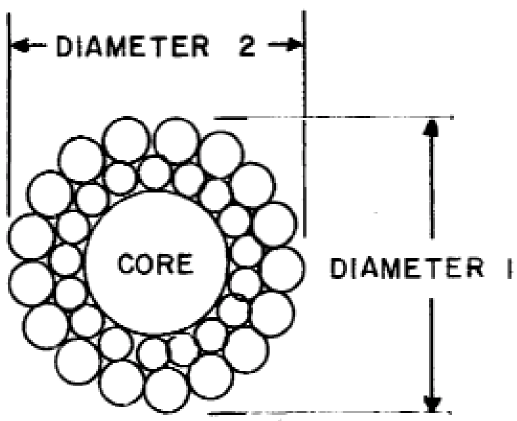
MEASURING PROCEDURE TO DETERMINE ARMOR LAY LENGTH

FIGURE 2-28

7.10 Outside Diameter

As for measuring the lay length of the outside armor wires, the cable must be maintained under a known, repeatable, constant tension when measuring the diameter. A reasonable value of this tension is about 5% of UTS.

a. Caliper method: The micrometer or vernier caliper method of diameter measurement is shown in Figure 2-29. Note that the two orthogonal measurements are a minimum. When a large variation (greater than 1%) occurs between the two measurements, others should be taken to locate the largest diameter. The measurements are averaged to obtain an average diameter.



MEASURE DIAMETERS ACCROSS WIRES ON TWO ORTHOGONAL AXES AND AVERAGE.

FIGURE 2-29 MEASURING DIAMETER USING A MICROMETER OR VERNIER CALIPERS

b. Measuring tapes: Measuring tapes give the measurement of circumference divided by it. These tapes are available from manufacturers of measuring tapes and the ones used for diameters less than one inch have 0.002 inch graduations. These tapes are useful because they provide an average diameter directly (see Figure 2-30).

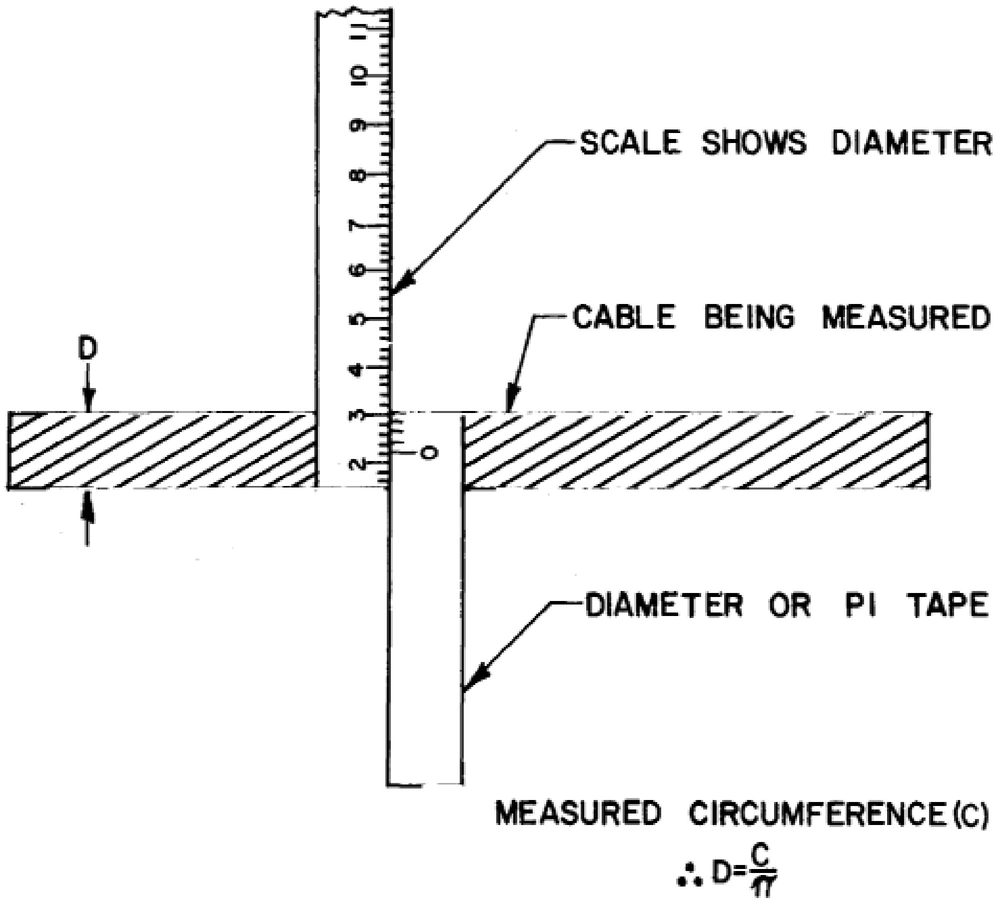


FIGURE 2-30 DIAMETER TAPE METHOD OF MEASURING DIAMETER

7.11 Need for Lubrication

An inspection to determine the need for lubrication is based more on judgment than on any measurable property of the cable.

a. Opening the outer armor: The most direct method would involve using clamps and axially twisting the cable in a direction to open the outer armor. For a L-H-L out armor this involves imposing a R-H rotation to the A-H clamp as shown in Figure 2-31. Clamp surfaces are to be smooth to prevent damaging the wire.

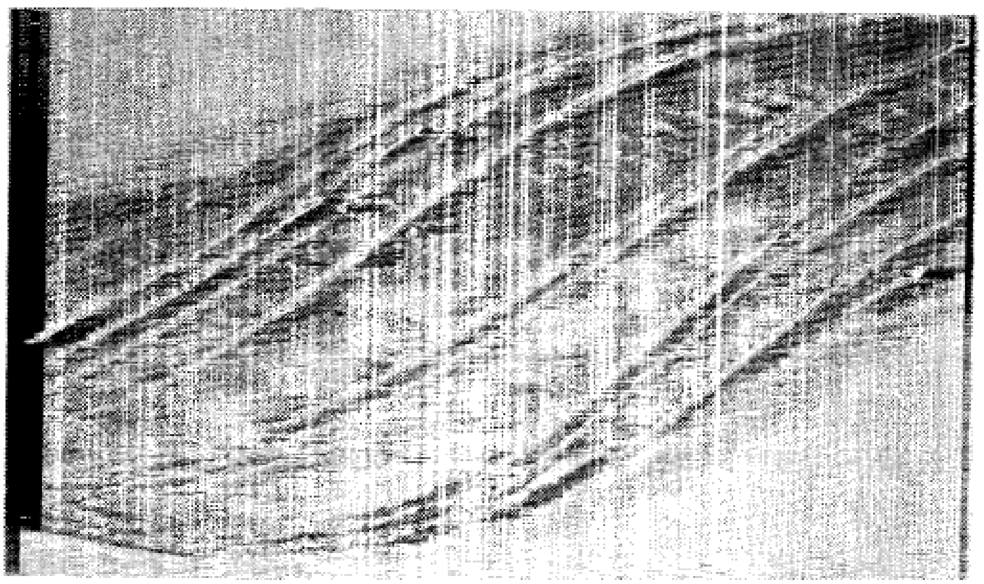


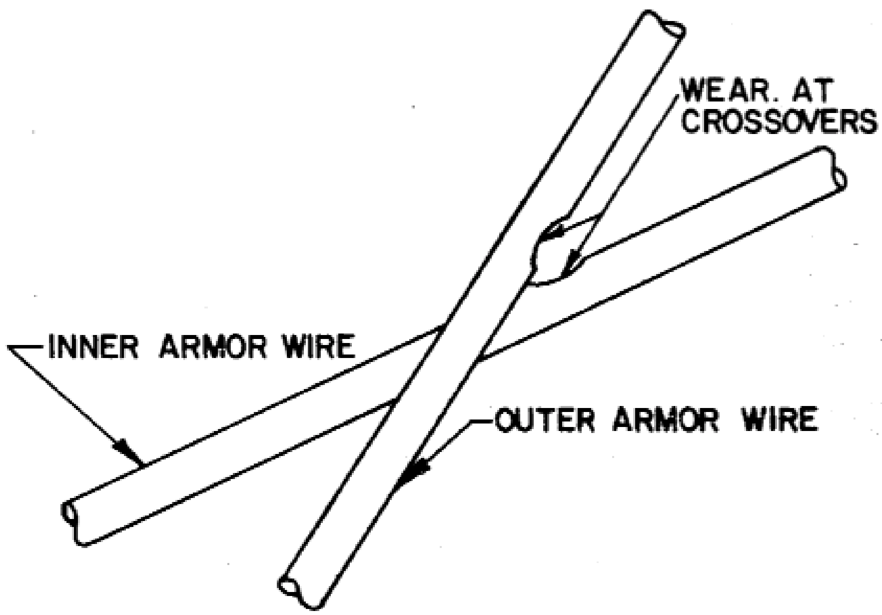
FIGURE 2-31

This procedure should initially be tried on a piece of scrap cable to learn how much the outer armor can be displaced without causing any permanent deformation.

When the outer armor is opened, observations should be made of the presence of lubricant and corrosion.

b. Inner-armor layer wear: While the outer armor is opened, also observe the extent of inner-armor wear; i.e., wear at the armor wire crossovers (see Figure 2-32). The rate of this wear is greatly affected by the maintenance of lubrication. Other factors affecting this rate include:

1. bearing pressure over sheaves (see Appendix 6)
2. winching speed
3. presence of abrasive materials



ARMOR WIRE WEAR AT CROSSOVERS

FIGURE 2-32

c. Bearing pressure determination: The bearing pressure parameter is described in Appendix 6 where the maximum allowable value for wire rope use with cast carbon steel sheaves is 1,800 lbf/sq. in. The calculated values of bearing pressure for typical oceanographic instrumentation E-M cables as shown in Appendix 7 are less than 800 lbf/sq. in. The reasons for such low values is the much lower strength-to-diameter ratio of E-M cables compared with wire rope and the common use of a 5:1 safety factor for E-M cables in oceanographic systems.

7.12 Location of an open in a conductor

A. For a single conductor cable:

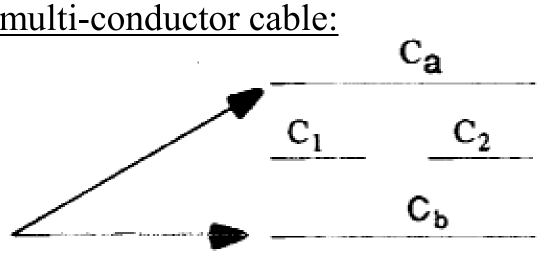
1. Using a capacitance bridge measure the capacitance from both ends of the cable: C1 and C2.

2. The length to the open is determined by:

$$L_1 = \frac{C_1}{C} \qquad L_2 = \frac{C_2}{C}$$

C = capacitance in pf/ft from manufacturers' data

B. For a multi-conductor cable:



1. Measure the capacitance of conducts adjacent to the faulty conductor, record C_a and C_b.

2. Average C_a and C_b

$$C_{avg} = \frac{C_a + C_b}{2}$$

3. Measure C_1 and C_2 and locate the open by:

$$L_1 = \frac{C_1}{C_{avg}} L \qquad L_1 = \frac{C_2}{C_{avg}} L$$

C. A time domain reflectometer (TDR) may be used to locate an open in a conductor of a single or multi-conductor cable. The proper use of this instrument requires calibration on a length of the same cable to determine the dielectric constant. When used with multi-conductor cables, corrections for the lay angle must be incorporated.

7.13 Fault Location, Conductor Short

The short may be conductor-to-conductor or, most commonly, conductor-to-armor. The detection of a high resistance short is extremely difficult to locate and, in most cases, the residual insulation resistance must be reduced to a direct short by applying a high voltage for burning through the remaining insulation. These procedures require specialized equipment and skills and should be performed only by adequately experienced personnel. Service Centers as maintained by manufacturers of oil field electrical wire-lines offer this and other services as further discussed in Section 11. The modified Murray Loop Test which is applicable, is described in Appendix 21.

7.14 Re-Reeling

The setup for performing many of the inspections in this section is diagrammed in Appendix 13. Motor power can be provided by any means which permits the operator to start and stop easily. Although well equipped cable service centers use variable speed, reversible hydraulic drives field inspection/repair stations have successfully used electric motors and:

a. friction drive against the edge of the reel flange,
b. V-belt

7.15 Cable Length Determination

The original length of an E-M cable will reduce as service continues. This reduction can be caused by the normal wear factors or by handling damage. The measurement of the length of a long cable can be determined by:

- a footage marker tape; this is a continuously marked tape which is installed in the cable during manufacturing. See Section 10 for specification coverage.

- re-reeling
- conductor resistance,
- weight

a. The footage marker tape offers the most convenient method of determining the approximate length of an E-M cable. This tape is installed in the cable by the manufacturer and it is marked with sequential footage figures. To determine the length of a cable it is necessary to read the footage numbers at each end and subtract. The specification of the footage marker tape for having it included in a cable is covered in section 10.

b. Re-reeling is convenient for length determination when it is being performed for inspecting the cable. Otherwise it is a very time consuming and difficult method.

c. Conductor resistance offers an accurate and convenient means for length determination; it requires a high accuracy resistance bridge. The procedure for single and multi-conductor cables is shown in Appendix 14. Note that a temperature correction is required.

d. Weight of the cable offers an approximation of cable length, but is the least accurate of all methods. The procedure is described in Appendix 15.

8.0 RETIREMENT CRITERIA

8.1 Considerations

Optimum, known reliability in use is the objective of the several activities bearing on the cable during conception to retirement (cradle to grave). These activities ideally encompass:

a. systems analysis to establish a full set of requirements (Chap. 9)

b. using these requirements to draft a cable procurement specification (Chap. 2)

c. verification of conformance of the cable by review of manufacturers test reports and conducting Receiving Inspections (Chap. 2)

d. proper design of the handling system (Chapters. 8, 9, 10, 11, 12)

e. proper installation in the cable system (Chapter. 2)

f. proper operation of the cable system (Chapter. 2)

g. proper cable maintenance (Chapters. 2, 6, 7).

h. evaluation against established criteria to determine fitness of the cable for continued use.

8.2 Broken Wire Criteria

The wire rope retirement criteria established by the American National Standards Institute and discussed in Chapter 1 are not applicable to E-M cables. The major percentage of total cable strength of E-M cables is contributed by the outer armor which is composed of single wire, not multi-wire, strands of armor wires. The breaking of any one of these armor wires is of major concern because it is subject to unlimited unstranding or unlaying. This unstranding relates to the ability of the broken wire to become continuously unwound from the cable.

Therefore, when an outer armor wire of an E-M cable becomes broken, the first action is to determine the cause. These causes together with follow-on activities include:

a. external abrasion: local or general. The usual cause is rubbing against a stationary surface or roughened sheave grooves. Should other wires appear serviceable, the broken wire may be repaired and service resumed.

b. broken factory weld: in this case an inspection of other factory welds which may be in the same cable is indicated. If the remainder of the cable appears satisfactory, the broken weld may be repaired and the cable returned to service.

c. nicking by a sharp object: as for the broken weld above, the remainder of the cable should be carefully inspected to determine if other nicks exist. If the damage is not extensive, a decision could be made to repair the broken wire and other nicked wires and return the cable to service.

d. broken wire in a rushed cable section: very careful electrical measurements are necessary to determine circuit integrity. If the cable is electrically satisfactory and the other wires in the damaged area are satisfactory, the decision could be made to return the cable to service.

e. wear between armor layers: a broken wire attributed to this cause indicates a general deterioration of the cable. The remainder of the cable should be carefully examined to determine the extent of this wear by opening the outer armor at measured intervals along the cable, not more than 1,000 ft. or 20% of total length, whichever is greater.

f. corrosion: this cause is difficult to distinguish from "e," above, because corrosion usually causes an acceleration of wear between armor layers as shown in Figure 2-32. The same inspection procedure as in "e" should be performed and if a decision is made to repair the broken wire and resume service, the cable should be carefully cleaned and lubricated.

g. kink: this massive, localized deformation is cause for immediate removal from service, or splicing of usable lengths (Appendix 19).

h. birdcaging: this is a localized evidence of a general improper operational condition. A birdcage is caused by a sudden release of tension whereby the potential energy of cable stretch induces an axial compressive strain causing permanent deformation of the wires. As for a kink, this condition is cause for immediate removal from service. The electrical core will usually have been damaged.

8.3 Life Cycle Criteria

a. Repetitive Cable Usage Systems: In systems which make repetitive use of the same cable, the reliability requirements may be so high that periodic replacement whether on service, mission life or cycle life may be imposed. All of these criteria demand the maintenance of accurate records. When these criteria are used, much information usable for modifying the retirement criteria are obtainable from the used cables.

b. Cable log information usage: The log of these used cables, together with final inspection reports, form a valuable information bank not only for use in modifying retirement criteria, but for use in identifying the life limiting factors. These factors can lead to investigations for improvements in the areas of:

1. E-M cable design concept
2. E-M cable engineering design
3. handling system design
4. handling procedures
5. maintenance procedures

c. Resulting increase in service life: the collected data would be used to increase the service life and cost effectiveness of the EM cable.

8.4 Non Destructive Testing

a. History: In the period of 1975 to 1982, there has been a steady development of techniques for detecting anomalies in steel wire constructions. They are generally based on ultrasonic, electromagnetic and Hall effect phenomena.

b. The ultrasonic method of anomaly detection and identification was the standard technique used in Project THEMIS which was conducted in the period of 1972 to 1979, at the Catholic University of America, Washington, D.C. An ultrasonic transducer was connected to one end of a wire rope specimen being tested for UTS and pickup mounted on the other end. The change in transmission through the wire rope specimen provided warning that changes in the structure were occurring. By comparing the recording of the nature of changes in ultrasonic transmission to the observed structure changes, a set of standards were developed which permitted accurate failure prediction later in the program.

c. Hall effect: Instruments using the Hall effect principle are becoming popularly used at this writing and within the Navy (NSRDC Annapolis and NCEL) work is in progress to develop a system for ultimate Navy operational use. A Hall effect instrument is commercially available from a company in the Netherlands. That instrument has been approved by Det Norske Veritas and Lloyds of London for certification of ropes used in aerial tramway systems across the Alps. The largest mining company in Canada, Noranda Ltd., has developed a Hall-effect instrument for inspecting and certifying wire ropes used in their mines. Two companies presently offer a service for inspecting steel wire structures.

d. Development of retirement criteria: That these instruments and services for using them are becoming available is encouraging for establishing retirement criteria based on the change of metal area and construction characteristics throughout the cross-section and the entire length of cables.

9.0 CABLE MATERIALS

9.1 Conductors

The most frequently used conductor material is copper because of its high conductivity and reasonable price. This is the only material to be considered in this section.

a. Circular mils: The area of round conductors is expressed either in circular mils (CM) or square millimeters (mm2). A circular mil is the diameter of a circle (expressed in mils) squared.

From Appendix 3 the seven wire strand of a No 20 AWG conductor is seven wires, each .0126" dia. The diameter of each wire in mils is 12.6 and the circular mils of each wire is 158.76. For the seven wires, the total circular mil area is 7 x 158.76 = 1111 as shown in Appendix 3.

b. Gaging system: The U.S. standard is American Wire Gage (AWG). This system is organized on the basis of defining two wire sizes and basing all others on those sizes. The two defined sizes are:

4/0 diameter = 0.4600 in.

36 diameter = 0.0050 in.

There are 38 sizes (39 increments) between these base points. Therefore, the ratio of diameters of adjoining sizes is:

$$\sqrt[39]{\frac{0.4600}{0.0050}} = \sqrt[39]{92} = 1.12$$

An approximation for estimating the relative properties of wires of various AWG gage numbers is:

- an increase of three gage numbers doubles the area and weight and halves the electric resistance.

c. Coating: The major purpose for coating copper is to prevent oxidation or to improve solderability for terminations. The use of tin, silver or nickel depends on expected temperatures, the limits being:

tin	135°C (275° F)
silver	200° C (392° F)
nickel	300° C (572° F)

Because of the near exclusive use of non-corroding thermoplastics for conductor insulations in oceanographic E-M cables, bare copper is generally used for the conductor material.

d. Physical properties: The physical properties of annealed copper are:

specific gravity	8.89
tensile strength, psi	35,000
elongation at 10% of UTS, %	20

e. Stranding: All conductors used in E-M cables are of a standard construction, i.e., they contain several individual wires which are twisted into a composite. As shown in Appendix 3, the number of wires can be varied, generally 7 and 19-wire for conductors up to #6 AWG. A larger number of wires can be used where even greater flexibility and tolerance to bending fatigue is required. The method used for twisting wires of a conductor is called stranding; the common construction being “bunched” or twisted together so that all have the same lay. The lay length of strands is generally eight times the strand diameter.

A common specification for copper conductors is ASTMSTD-B286, Specification for Copper Conductors for Use in Hookup Wire for Electronic Equipment.

9.2 Electrical Insulations

With very few exceptions, the insulations used in oceanographic E-M cables are thermoplastic. As the name implies, this class of plastics have a repeatable relationship of physical properties with temperature. The properties of the most commonly used materials are shown in Appendix 8.

The most commonly used insulating material, polyethylene, has a low specific gravity and very good electrical and mechanical properties.

The government specification covering insulated conductors is MIL-W-16878, and cables are covered by MIL-C-17.

9.3 Shielding

This part of a cable is usually the outer conductor of a coax, but can also be an electromagnetic interference shield of a single or multiconductor component.

The material for shields may either be tapes or a construction of round wires in either a braid or a serve. Because of the tendency of tapes to break in small E-M cables, their use is usually limited to large multiconductor cables.

a. Shielding tapes commonly used are a polyester base with a film of copper on one side. To provide electrical continuity of wraps and a means for termination, a drain wire is used. This drain wire is cabled with the conductor bundle and lies in one of the outer interstices. The size and location of the drain wire must provide for this electrical contact with the conducting surface of the shielding tape. Shielding tapes have the advantage of low cost and 100% shielding, but the disadvantage of poor mechanical properties, particularly those required in E-M cables for flexing service.

b. Braids utilize small diameter copper wires in sizes generally within #30 AWG to #38 AWG. The coverage is between 85% and 95% and it is the highest cost shielding method.

Its use in flexing E-M cables should be adopted with caution because of widespread experience with extreme degradation. The nature of this degradation is self-cutting of the wires at the crossover point. The very high compressive forces of the covering contrahelical armor imposes extremely high stresses at these point contacts.

c. Serves use small diameter copper wires as in braids, but the difference lies in the construction. A served shield is like the serving of the armor in E-M cables; it may be single layer or double layer. The percent coverage for a single serve is lower than that for braids being in the range of 80% to 90%. A contrahelical served shield may provide a coverage approximating that of a braid. The advantage of a served shield is a longer flexure fatigue life than either the taped or braided shields. The higher flexure fatigue life compared with braids results from elimination of the high stress.

Although not substantiated, the practice of filling the voids in served shields may additionally increase the flexure fatigue life. This filling may be silicone rubber, Vistonex, or other suitable material.

9.4 <u>Jackets</u>

Two types of jackets must be considered in an E-M cable, that which is under the armor and that which covers the armor. The requirements for the physical characteristics of the materials do not differ extremely; they may be summarized as:

- low water permeability
- low cold flow characteristic
- high abrasion resistance
- high cut-through resistance
- resistance of petroleum compounds

Three of the compounds included in Appendix 8 (HDPE, polyurethane, Nylon 6) generally satisfy these requirements. TPR is greatly affected by petroleum compounds and has poor abrasion and cut-through resistance; its primary attribute being the low, 0.88, specific gravity.

The thickness of jackets for the usual diameter range of E-M cables approximately follows the 10% rule; i.e., the normal jacket thickness is 10% of its inner diameter.

9.5 Armor

The most common metal used for armor is steel because of its relative low material cost, excellent mechanical properties, and ease of fabrication and assembly.

a. Steel grades, the steel wires for E-M cables, are covered by the same AISI (American Iron and Steel Institute) specification which applies to wire rope. The grades covered in this specification are:

- mild plow steel (level I)
- plow steel (level II)
- improved plow steel (level III)
- extra-improved plow steel (level IV)
- extra-extra-improved plow steel (level V)

The tensile strength increases to the highest in extra-improved plow steel.

The two grades commonly used in E-M cables are improved and extra-improved plow steel. The breaking and tensile strengths for a selection of wire sizes normally used in oceanographic cables is shown in Appendix 9.

When quoting breaking strengths, manufacturers usually state a minimum value. The E-M cable manufacturer, having a wire mill, has the opportunity to tailor the wire and, therefore, more idealize the strength-to-diameter characteristics. A standard AISI test for ductility of these wires is to wrap the wire in a close helix for six complete turns around a mandrel having a diameter twice that of the wire being tested. There should be no tendency to develop cracks or to break.

The Government specification for these steel wires is: RR-W410, "Wire Rope and Strand."

b. Stainless Steels: The austenitic (300 Series) Stainless steels, particularly Type 316, have been used in oceanographic cables with no success in obtaining a longer life by eliminating corrosion as the life limiting operational factor. Corrosion experienced by steel was found much less hazardous than the insidious crevice corrosion to which this class of stainless steels are susceptible.

b1. Crevice Corrosion - Stainless steels depend on the maintenance of a protective oxide film to isolate the base metal from seawater oxygen starvation can expose the highly reactive basis metal. In double (contra-helical armor construction there is little water flow into the inter-armor area and oxygen depletion occurs. The consequent breakdown of the oxide film allows localized corrosion which is termed crevice corrosion.

b2. High Alloy Steels - Two alloys, Inconel 625[5] and MP-35N[6] have been very successfully used in highly corrosive environments. Their properties are shown in Appendix 10. Although their use has been very successful with no history of difficulties, the very high cost has discouraged any more than highly specialized use.

b3. Nitronic 50[7] and AL-6X[8] - Both of these proprietary alloys are being used in current government systems. Nitronic 50 armored cables are being used in cables for Navy Tow systems and at this writing fleet evaluation is still in progress. AL-6X was

[5] lnconel 625: Trademark of Alloys International

[6] MP3SN: Trademark of SPS Co.

[7] Nitronic 50: Trademark of Armco Steel

[8] AL-6X: Trademark of Allegheny Ludlum Steel Co.

extensively tested before being selected as the armoring metal for the OTEC power cables. Properties are shown in Appendix 10.

With the tradeoffs in lower tensile strength and higher cost, both alloys appear to offer a longer service period before corrosion becomes the basis for retirement. The economic study for each system must be based on the effect of corrosion as being the major life-limiting factor. Also, in many cases, the higher cost of these corrosion resisting alloys must be balanced with the cost effectiveness of an improved cable maintenance program.

10.0 CONTRA-HELICALLY ARMORED, E-M CABLE SPECIFICATIONS

The development of a meaningful cable specification requires a thorough analysis of system equipment and phenomena which affect the operation of the cable. Because of the uniqueness of each system generalized specifications do not help either the user or the manufacturer. The approach in this section will be the discussion of information which should be considered in the development of a TAILORED procurement specification.

10.1 Performance vs Construction Specification

A construction specification is the most simple tool for communication of requirements to the manufacturer. This approach is based on the presumption that the same, exact cable construction provided satisfactory performance in the same or in a similar system.

10.2 Construction Specification

When using a construction specification, all known restraints should be explicitly stated. For instance, to fit an established Lebus grooving or conform to some other handling system restraint, the cable feature or system description should be stated.

a. A convenient technique is to use a manufacturer’s part number for the description. It must be remembered that changes may have been made in the design of the cable and the statement should either refer to a particular procurement or date in addition to the part number.

b. Important data to include in a Construction Specification are shown in Appendix II. Note especially the request for test data to be furnished to the Buyer. These data are valuable to permit any later troubleshooting the cable should it become damaged in service.

10.3 Performance Specification

This is the most useful type for the system designer because it requires less knowledge of cable design, but more knowledge of performance requirements. Also, this approach does not restrict the manufacturer in the effort to provide the best design for the application.

As shown in Appendix 12, the major specification elements are:

a.) scope

b.) referenced documents

c.) requirements

d.) test, or quality control

e.) marking and shipping,

and each section will be discussed further.

a. Scope: The type of system and operational information will, together with the life objective, provide the cable designer with valuable information regarding tolerance to tensile or flexure fatigue and mechanical trauma.

b. Referenced documents: Care must be taken in referencing only those documents which are used in describing the Requirements, Section 3. The statement used in Government Specifications is "The following documents are a part to this specification to the extent specified herein." The last statement, "to the extent specified herein," requires that there is specific reference to a particular part of the document and the only purpose for listing the document is for the convenience of the reader.

c. Requirements: This section contains the details of requirements which should extend to the requirements impacting the electrical, mechanical and termination performances.

d. Test: Care is needed to include all testing which is necessary to ensure that critical performance characteristics are covered in the qualification test program and verified in production tests.

e. Marking and shipping: length. The cable may be marked by:

- a footage marker tape
- magnetic marking

The footage marker tape is a thin polyester tape which is layed longitudinally along the cable core. It is marked in feet and permits a very convenient determination of residual length and positively discriminates one end from the other. Also, by recording the footage at each end upon receiving the cable, the amount of cable cut from either or both ends can be determined as the cable continues in service. The cost of these footage marker tapes is extremely low and they are easily installed during manufacture.

Magnetic marking is a technique of permanently magnetizing points on steel armor. lt is a standard technique for oil well logging cables wherein these marks are used for measuring the length of payed-out cable. The readout instruments are commercially available and all manufacturers of oil well logging cables have the magnetic marking equipment. Unlike paint or tape markers, these magnetic marks are not removed by abrasion or reaction with sea water.

Consideration should also be given to handling of the shipping reel and any restrictions to the diameter or width dimensions should be stated.

11.0 AVAILABLE CABLE SERVICES

11.1 General

Just as the origin of E-M cables has occurred in oil industry service for logging, perforating, etc., service centers have also become available from the same industry. E-M cables in the oil field services receive much more continuity of use than in oceanographic services. Therefore they experience wear phenomena which is partially reversible by proper services. These services are tailored to the needs of a standardized range of oil field cables that have an upper diameter of about ½ inch. Many oceanographic cables are included in this range and some services may be extended to encompass other diameters.

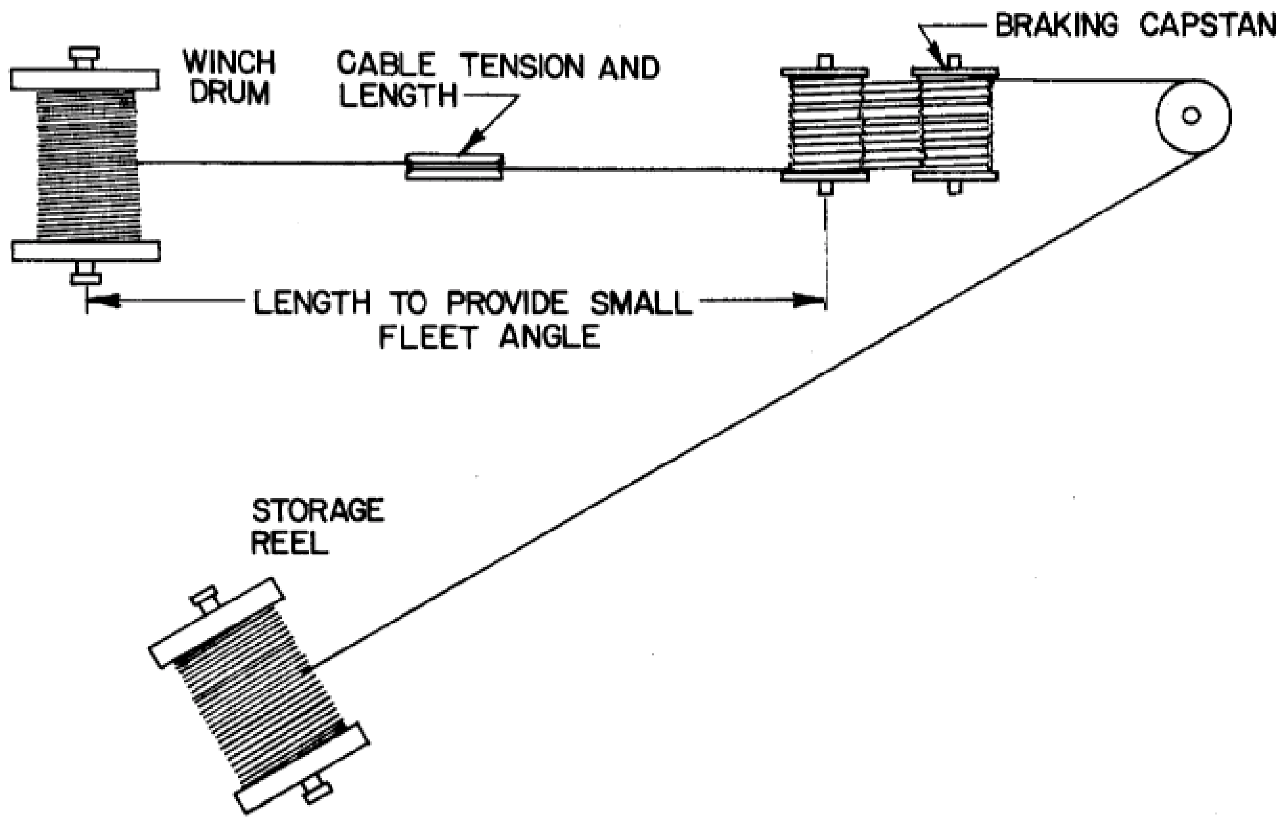


FIGURE 2-33 SPOOLING SETUP

It is useful to know of the availability of these services and in general the procedures used to obtain maximum, satisfactory service life.

The simplified splicing steps presented in Appendix 19 are intended only to show the principles which govern this process.

11.2 Spooling

To obtain an optimized spooling setup specialized equipment and skills are required. Although some oceanographic fitting-out facilities have adequate equipment, many do not.

The elements of a spooling setup used in Service Shops are shown in Figure 2-33. The Braking Capstan provides a regulated back-tension on the cable to obtain the spooling schedule discussed in Section 6.0 and in Chapter 10.

11.3 Splicing

A full cable splice including all core components and the armor is possible and is routinely performed on oil field cables. This is one of the E-M cable services which depends on apprenticeship learning. Very little is published and, outside the oil field cable community, it is relatively unknown. Principles which apply to an E-M cable splicing procedure are presented in Appendix 19.

11.4 Fault Location

All equipment and skills required for fault location are available at service shops. In general, all inspection and testing discussed in Section 7.0 can be performed at Service Centers.

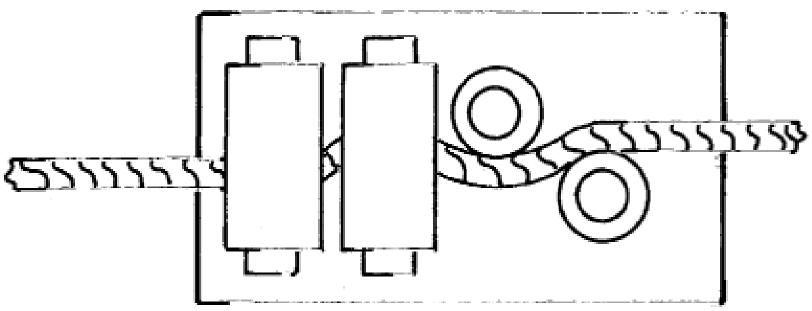
11.5 Reconditioning

Reconditioning is a series of operations performed on a used cable to effect:

(a) cleaning
(b) retightening the outer armor
(c) relubrication

a. Cleaning is accomplished by rotary wire brushing the external surfaces. When foreign materials are lodged in the interarmor area dislodging is encouraged by passing the cable through offset rolls as shown in Figure 2-34.

FIGURE 2-34 OFFSET ROLLS



DUAL SETS ARE PLACED IN ORTHORGONAL AXES

b. Outer armor tightening is performed by mounting the reel of cable turner illustrated in Figure 2-35. The amount of tightening is evaluated by one of the methods described in Section 7.7 or, in some cases, observing the tendency of the cable to rotate about its axis as a point translates from the lower turning sheave to the tensioning device (capstan or hoist).

Rotation in a LHL outer armor tightening direction (near end rotates CW) indicates looseness, and a need for additional rotations of the reel turner per 100 feet of cable.

c. Lubrication is performed in a pressurized tube which is fitted with end glands to seal around the cable. The general schematic of a pressure lubricator is illustrated in Figure 2-36.

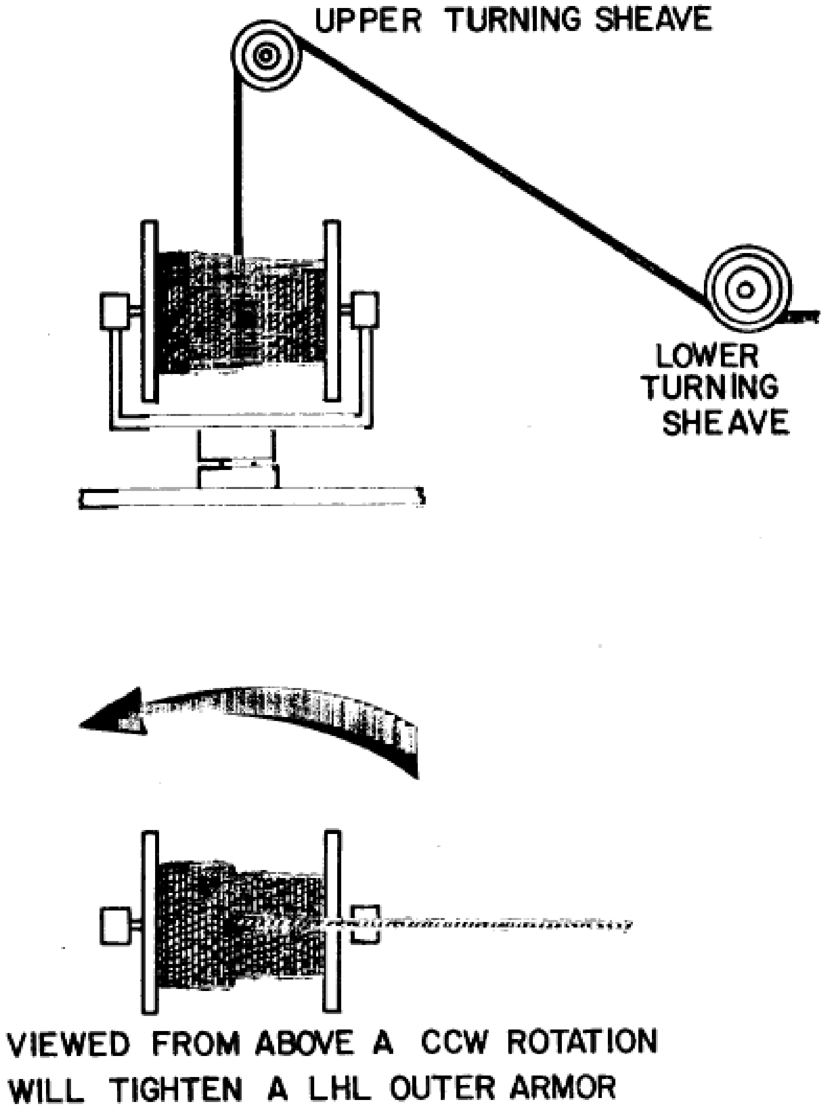


FIGURE 2-35 REEL TURNER

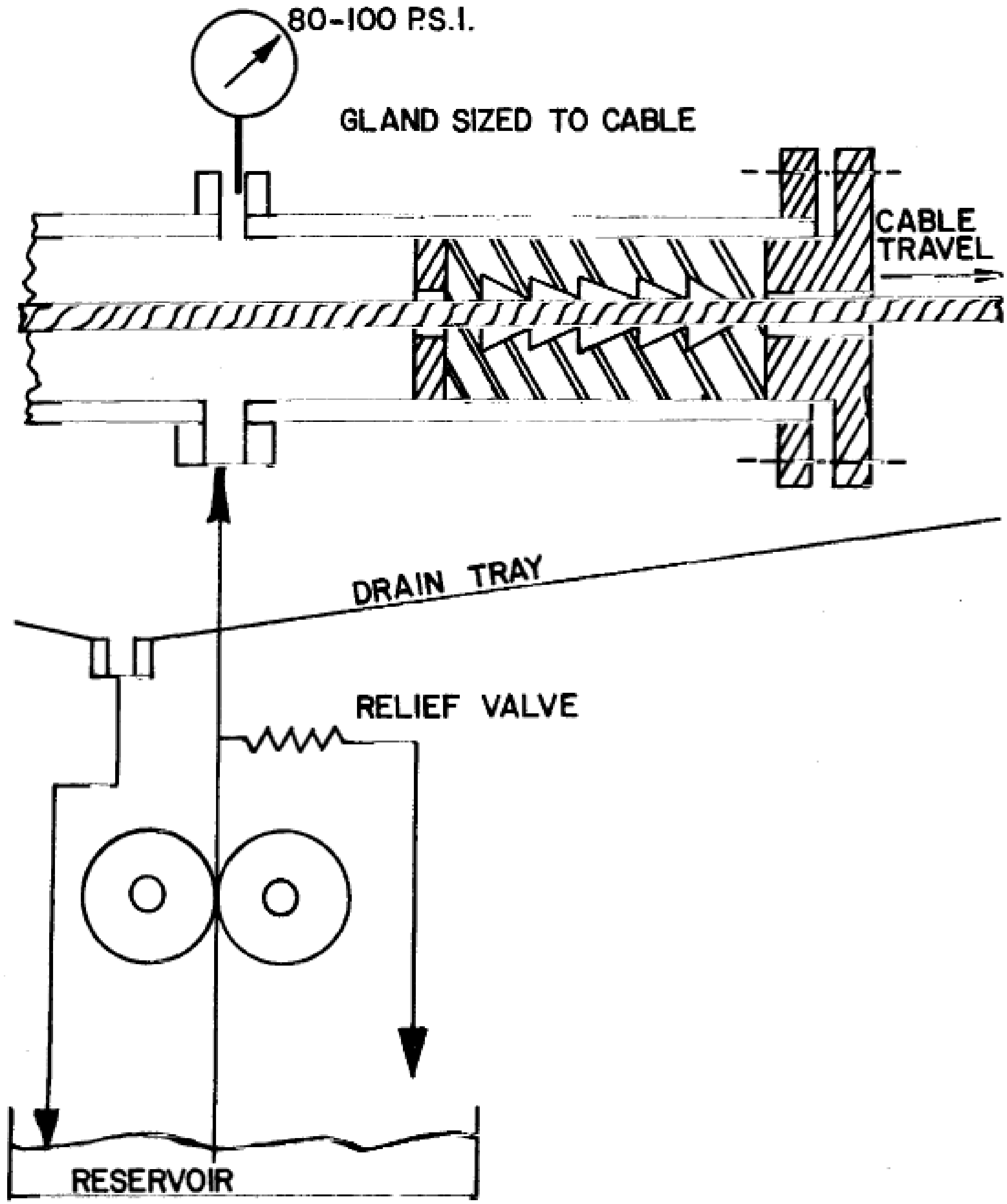


FIGURE 2-36 CABLE LUBRICATOR

11.6 Magnetic Marking

A means for reliable, accurate cable pay-out determination, is performed by applying a high level magnetic flux to a localized part of the armor. The measurement between magnetic marks has been standardized at 100 feet and these length increments are determined automatically by referencing to the previous mark.

This marking means can be applied to any material having a high magnetic permeability and the detection life is known to be over a year.

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APPENDICIES

18.0 Calculations for Physical Properties of E-M Cables

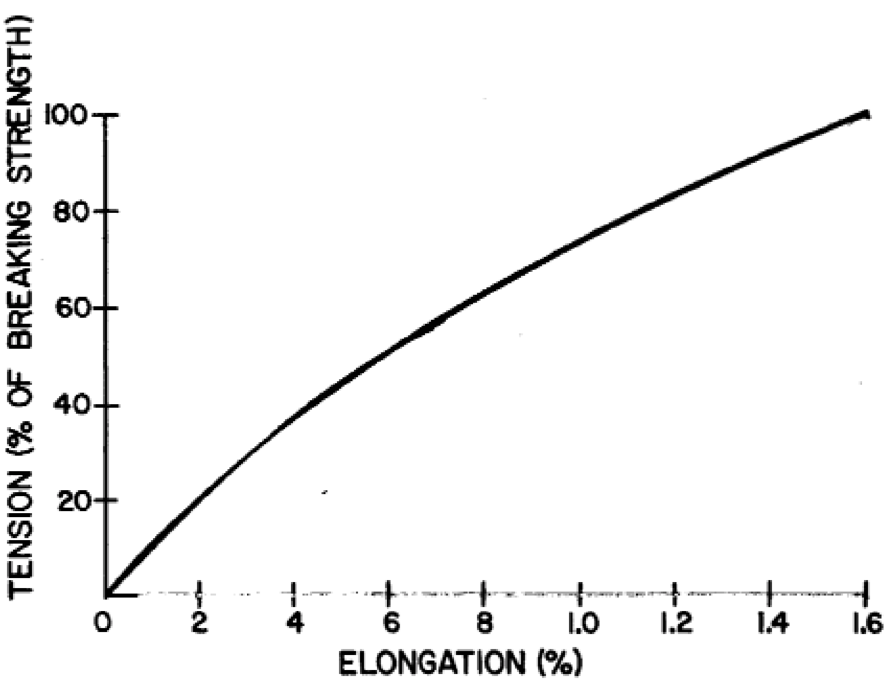
19.0 Principals of E-M Cable Splicing.

20.0 Armored Cable Diameter vs Breaking Strength.

21.0 Location of Short to Armor in Multiconductor Cables.

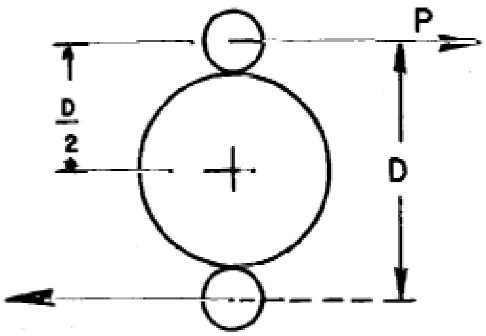
APPENDIX 1.0

TYPICAL TENSION/ELONGATION CHARACTERISTIC OF A DOUBLE ARMORED CABLE



APPENDIX 2.0

TORQUE RATIO EQUATION



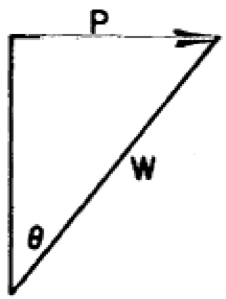
$$\text{Torque} = T = (PD)1/2 \qquad (I)$$

(From Fig. 2-1)

$$\sin\theta = \frac{P}{W}$$

(From Fig. 2-2)

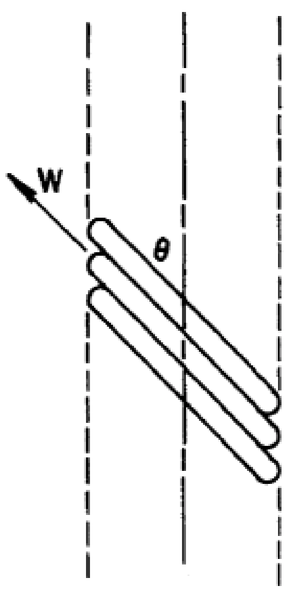
$$w = \frac{P}{\sin\theta} \qquad (2)$$



For a wire, Youngs Modulus is:

$$E = \frac{\frac{W}{A}}{\varepsilon} \qquad W = EA\varepsilon \qquad (3)$$

using (2)



$$\frac{P}{\sin\theta} = EA\varepsilon \qquad (4)$$

$$P = EA\varepsilon \sin\theta \qquad (5)$$

substitute (5) in (1)

$$T = \frac{EA\varepsilon D \ \sin\theta}{2} \qquad (6)$$

summing the torque for all wires in an armor layer

$$\Sigma T = \frac{NEA\varepsilon D \ \sin\theta}{2} \qquad (7)$$

Express the ratio of the torques of outer and inner armor layers

$$R_T = \frac{\sum T_0}{\sum T_I} = \frac{N_0 E_0 A_0 \varepsilon_0 D_0 \sin\theta_0}{N_I E_I A_I \varepsilon_I D_I \sin \varepsilon_I} \qquad (8)$$

E and $\in$ can be assumed the same for the inner and outer armors although the magnitude of $\in$ will differ.

$$R_T = \frac{N_0 A_0 D_0 \sin\theta_0}{N_I A_I D_I \sin\theta_I} \qquad (9)$$

$$A = \frac{\pi}{4} d^2 \qquad (10)$$

substitute (10) in (9) and cancel the constant, $\frac{\pi}{4}$:

$$R_T = \frac{N_0 d_0^{\,2} D_0 \sin\theta_0}{N_I d_I^{\,0} D_I \sin\theta_I} \qquad (11)$$

Note that this analysis is based on "P", the tangential force of armor wires, not on cable tension. Figure 2 cable tension =

$$(L) = \sum(L_0 + L_I)$$

$$= \sum\left(\frac{Pa}{\operatorname{Tan}\theta_0} + \frac{Pt}{\operatorname{Tan}\theta_I}\right)$$

A =	cross-sectional area of an armor wire	(in^2)
d =	armor wire diameter	(in)
D =	pitch diameter of an armor layer	(in)
E =	Young's Modulus	($\frac{lb}{in^2}$)

N = number of all armor wires per layer	
P = tangential force of armor wires in (1) Layer	(lb)
R_T = torque ratio	
T = torque	(lb-in)
W = tension in a wire	(lbf)
θ = armor lay angle	(degrees)
ε = armor wire strain	$\frac{(in)}{in}$

APPENDIX 3.0

PROPERTIES OF STRANDED COPPER CONDUCTORS

Conductor Size	Stranding		Overall Diameter		Copper Area		Weight		Strand Break Strength		DC Resistance at at 20°C (68°F)	
											ohms/mft	ohms/Km
Awg	Inch	mm	Inch	mm	Cir.mils	Sq.mm	1bs/mft	Kg/Km	1bs	N	ohms/mft	ohms/Km
24	7/.008	7/.203	.024	.610	448	.227	1.38	2.06	12.7	56.3	24.0	78.6
	19/.005	10/.127	.024	.617	475	.241	1.47	2.18	13.4	59.7	22.6	74.1
23	7/.009	7/.229	.027	.686	567	.287	1.75	2.60	16.0	71.3	18.9	62.1
22	7/.010	7/.254	.030	.762	700	.355	2.16	3.21	19.8	88.0	15.3	50.3
	19/.0063	19/.160	.031	.778	754	.382	2.33	3.46	21.3	94.8	14.2	46.7
20	7/.0126	7/.320	.038	.960	1111	.563	3.43	5.10	31.4	139.8	9.7	31.7
	19/.0071	19/.180	.035	.876	958	.485	2.96	4.40	27.1	120.5	11.2	36.8
	19/.008	19/.203	.039	.988	1216	.616	3.75	5.58	34.4	152.9	8.8	29.0
19	7/.0142	7/.361	.043	1.082	1411	.715	4.36	6.48	39.9	177.5	7.6	24.9
	19/.0089	19/.226	.043	1.099	1505	.763	4.65	6.91	42.6	189.3	7.1	23.4
18	7/.0152	7/.386	.046	1.158	1617	.819	4.99	7.93	45.7	203.4	6.6	21.8
	19/.010	19/.254	.049	1.234	1900	.963	5.86	8.73	53.7	239.0	5.6	18.5
	41/.0063	41/.160	.051	1.295	1627	.825	5.02	7.47	46.0	204.7	6.6	21.6
16	7/.020	7/.508	.060	1.524	2800	1.42	8.64	12.86	79.2	352.1	3.8	12.6
	19/.0112	17/.284	.054	1.383	2382	1.21	7.36	10.95	67.4	299.7	4.5	14.8
14	7/.0242	7/.615	.073	1.844	4099	2.08	12.65	18.83	116	516	2.6	8.6
	19/.0142	19/.361	.069	1.753	3831	1.94	11.82	17.59	108	482	2.8	9.2

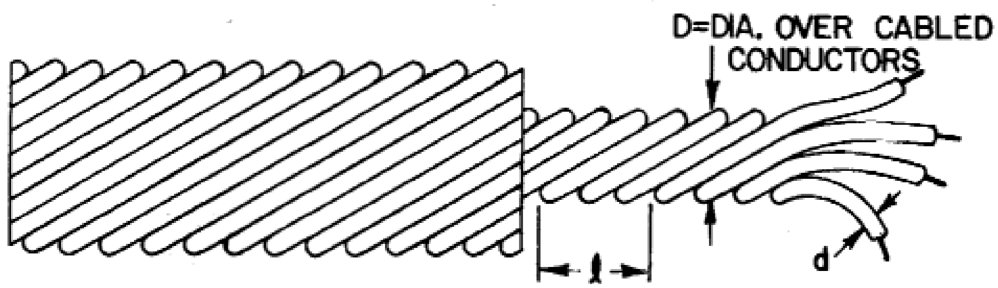
12	7/.0305	7/.775	.092	2.324	5612	3.08	20.10	29.91	184	819	1.6	5.4
	19/.0179	19/.455	.087	2.210	6088	3.08	18.79	28.0	172	766	1.8	5.8
	19/.0185	19/.470	.090	2.284	6503	3.29	20.07	29.9	184	818	1.65	5.42
10	7/.0385	7/.978	.116	2.934	10376	5.26	32.02	47.65	293	1305	1.03	3.39
	19/.0234	19/.594	.114	2.889	10403	5.27	32.11	47.78	294	1308	1.03	3.39
9	7/.0432	7/1.097	.130	3.292	13064	6.62	40.32	60.00	369	1642	.82	2.70
	19/.0242	19/.615	.118	2.987	11127	5.64	34.34	51.10	315	1399	.96	3.76
8	7/.0486	7/1.234	.146	3.703	16534	8.38	51.03	75.93	463	2079	.65	2.13
	19/.0295	19/.749	.143	3.642	16535	8.38	51.03	75.94	468	2079	.65	2.13
6	7/.0612	7/1.554	.184	4.663	26218	13.29	80.92	120.4	741	3297	.41	1.34
	19/.0372	19/.945	.181	4.592	26292	13.32	81.15	120.8	743	3307	.41	1.34
4	7/.0772	7/1.961	.232	5.883	41719	21.14	128.8	191.6	1180.	5247	.26	.84
	61/.0253	61/.643	.228	5.788	39045	19.78	120.5	179.3	1104.	4911	.27	.90
3	61/.0285	61/.724	.257	5.784	49547	25.11	152.9	227.6	1401.	6231	.22	.71

NOTE: All values nominal. 19 strand copper diameters are bunched construction. Break strength is based on approximately 40000 PSI for soft annealed copper. DC resistance of conductors in cable may vary due to helix or slight elongation.

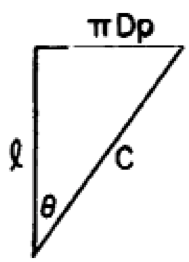
APPENDIX 4.0

COPPER ELECTRICAL RESISTANCE TEMPERATURE CORRECTION MULTIPLIER

Measurement of Cooper °C	Temperature °F	Multiplier
0	32	1.084
5	41	1.061
10	50	1.040
15	59	1.020
20	68	1.000
25	77	0.981
30	86	0.963
35	95	0.945
40	104	0.928
45	113	0.912
50	122	0.896
55	131	0.881
60	140	0.866
65	149	0.852
70	158	0.838
75	167	0.825
80	176	0.812



1. MEASURE THE CABLED CORE DIA. (D), CONDUCTOR DIAMETER (d) AND LAY LENGTH (ℓ)



LAY DIAGRAM I

2. USING THE MEASURED "Dp" AND "ℓ" CALCULATE THE LAY ANGLE (θ) USING THE LAY DIAGRAM

$$\theta = \arctan \frac{\pi D p}{\ell}$$

WHERE

$$D_p = D - d$$

3. REFERRING TO THE LAY DIAGRAM DETERMINE THE RELATIONSHIP BETWEEN CABLE LENGTH " " AND CABLED CONDUCTOR "C"

L = CABLE LENGTH

C = CONDUCTOR LENGTH

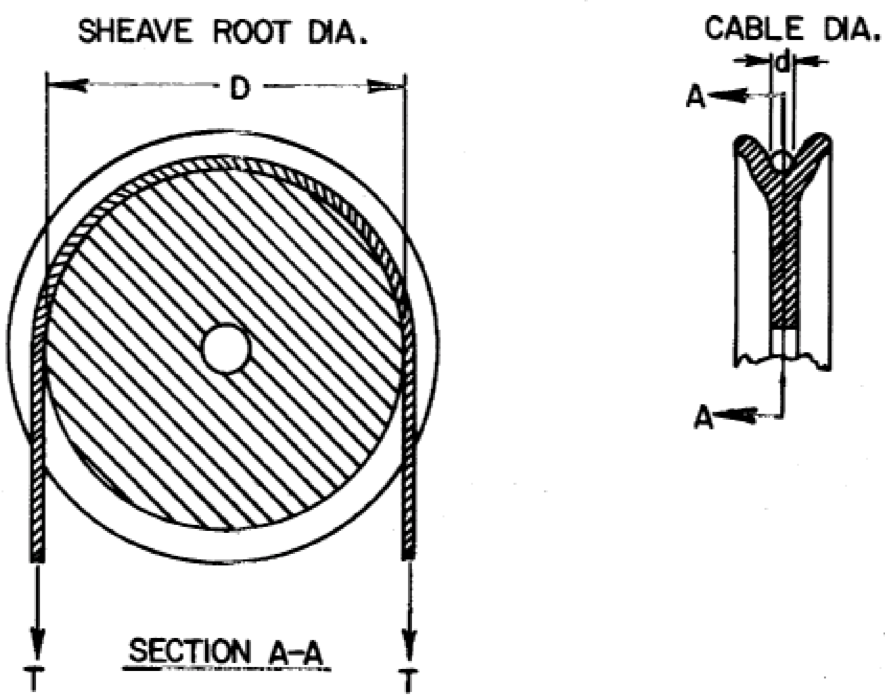
$$L = C \cos\theta = C \cos\left[\arctan\left(\frac{\pi D p}{\ell}\right)\right]$$

OR

$$C = \frac{L}{\cos\left[\arctan\left(\frac{\pi D_p}{L}\right)\right]}$$

APPENDIX 5.0 DETERMINING THE LENGTH OF A CABLED CONDUCTOR

APPENDIX 6.0 SHEAVE TO CABLE BEARING PRESSURE



PRESSURE $(p) = 2T \times \frac{1}{A} = \frac{2T}{A}$

A = Projected Area of Cable Contact Surface = dD

$\therefore\ p = \frac{2T}{dD}\ \left(\frac{lbf}{in^2}\right)$

FOR WIRE ROPE USE WITH CAST CARBON STEEL SHEAVES THIS BEARING PRESSURE CAN ACCEPTABLY BE AS HIGH AS 1800 LBF/sq.in.

APPENDIX 7.0

SHEAVE-TO-CABLE BEARING PRESSURES OF TYPICAL OCEANOGRAPHIC CABLES

(d) O.D. (In)	(D) Sheave root (In)	(T) 20% of UTS (lbf)	(P) Bearing Pressure (lbf/sq.in)
.183	10	580	634
.305	13	1,480	747
.224	12	880	655
.254	14	1,100	619
.282	16	1,440	638
.351	19	2,080	624
.375	20	2,360	629
.421	23	3,200	661
.670	28	6,600	704
.726	30	8,000	735

Calculated from the formula $P = \dfrac{2T}{2d}$

from Appendix 6.0.

APPENDIX 8.0

NOMINAL PROPERTIES OF INSULATING AND JACKETING MATERIALS

	Polypropylene (PP)	Nylon 610 (N)	Polyethylene High Density (HDPE)	Polyethylene Low Density (LDPE)	Teflon (TF)	Polyurethane (PU)	Hypalon (HY)	Thermoplastic Rubber (TPR)
Specific gravity	.902	1.08	.947	.920	2.16	1.25	1.15	0.88
Ultimate Tensile Strength, psi	5,000	8,000	3,400	2,200	3,000	6,000	2,500	2,000
Ultimate elongation, %	200	200	250	625	250	600	325	300
Dielectric Constant, 1 kHz	2.22	4.5	2.32	2.25	2.1	7.5	8	2.2
Rated mix Temp, °C	-10	-40	-65	-65	-65	-55	-40	-70
Relative Cost	0.4	1.2	0.4	0.4	12.0	1.8		

Appendix 9.0 has been removed. New charts are available in AISI Steel Products Manual Level III and Level IV.

APPENDIX 10.0

PROPERTIES OF CORROSION RESISTANT ARMORING MATERIALS

	GIPS	NITRONIC 50[1]	MP35N[2]	ALLEGHENY 6X[3]	INCONEL 625[4]
density	7.8 gm/cc	7.88 gm/cc	8.57 gm/cc	8.0 gm/cc	8.44 gm/cc
	281 1b/in^3	.285 1b/in	.309 1b/in	.293 1b/in	0.305 1b/in
Electrical resistivity	19 microhm-cm	82 microhm-cm	101 microhm-cm	82 microhm-cm	129 microhm-cm
UTS	270 Ksi	246 Ksi	280 Ksi	205 Ksi	269 Ksi
60% cold reduction	1862 Mpa	1696 Mpa	1793 Mpa	1412 Mpa	1853 Mpa
.2% offset yield	216 Ksi	234 Ksi	240 Ksi	184 Ksi	253 Ksi
	1489 Mpa	1613 Mpa	1655 Mpa	1270 Mpa	1743 Mpa
Modulus of elasticity	28 x 10^6 psi	28 x 10^6psi	33.6 x 10^6psi	29 x 10^6psi	30 x 10^6 psi
	193,054 Mpa	193,100 Mpa	200,000 Mpa	206,844 Mpa	206,884 Mpa
Coefficient of	.0063 ft/M'/F°	.0092 ft/ M'/F°	.0076 ft/M'/F°	.0089 ft/M'/F°	.0073 ft/m'F°
thermal expansion	.0113 m/km/C°	.0162 m/Km/C°	.0137 m/Kn/C°	.0160m/Km/C°	.0131 m/Km/C°
magnetic properties	magnetic	non-magnetic	non-magnetic	non-magnetic	non-magnetic
Analysis (Typical)					
Ni		12.5%	35%	24.50%	57.0 Ni
Co		---	35%	---	1.0 Co
Cr		---	20%	20.25%	21.0 Cr
Mo		2.2	10%	6.25%	9.0 Mo
Mn		5.0	—	.50%	0.5 Mn
Material Cost Ratio	1	8	65	19	41
Cable Cost Ratio	1	3	16	6.5	11

The above data are presented for comparison only and are not intended as accurate design or quotation data

Albert G. Berian
10 July 1980

Trademarks:

1. Nitronic 50 -- Armco
2. MP35N -- SPS Co.
3. Allegheny 6X -- Allegheny Ludlum
4. Inconel 625 -- Alloys International

APPENDIX 11.0

ELEMENTS OF A CONSTRUCTION SPECIFICATION

<u>Electrical</u>

Conductor size and stranding; insulation and thickness

Belt and jacket thickness, if used

Insulation resistance

Capacitance

Dielectric strength

<u>Mechanical</u>

Cable length and tolerance

Number of wires and diameter

Breaking strength, minimum

Permissible number of welds in the finished armor wire

Weight of zinc coating on wires

Overall diameter and tolerance

<u>Test Data to be furnished</u>

Conductor resistance

Capacitance

Dielectric strength

Insulation resistance

Breaking strength of armor wires

Cable UTS and yield strength

APPENDIX 12.0
ELEMENTS OF A PERFORMANCE

SPECIFICATION

1.0 SCOPE

1.1 System types include:

- tow
- umbilical
- vertical array
- floating
- bottom deployed.

1.2 Mission profiles cover:

- speed of deployment - environmental
- frequency and duration of deployment
- geographical location
- steady-state and varying tensions
- known hazards such as fishbite, corrosive conditions and mechanical impact on the cable such as crushing, abrasion, etc.

1.3 Type of handling system

- winch characteristics (drum diameter., traverse, fleet angle, drum grooving, which type (storage or tension)
- level wind)
- sheave diameter and groove design (geometry, smooth ness, hardness, coating used
- coiling into bales, cannisters, tanks, etc.

2.0 REFERENCED DOCUMENTS

These could include:

U.S. Government

MIL-C-915 Shipboard cable frequently referenced for the non-hosing test requirement and conductor color coding.

MIL-W-16878 - Insulated Wire Covers

MIL-C-I7; RF Cables
a basic document for RG cables

MIL-I-45208

covers basic quality assurance
requirements

MIL-Q-9858

covers comprehensive quality assurance
requirements.

MIL-C-242l7

Underwater Electrical Connectors

Independent Power Cable Engineers Association (IPCEA)

various sections cover insulation construction and other requirements
for power cables to 35 KV.

3.0 REQUIREMENTS

3.1 Electrical

- Power conductors: phases, frequency, watts or current, max. voltage drop Or resistance, corona-initiation and suppression voltages

- Control: voltage, current

- Signal or communication: characteristic impedance, resistance, voltage, Crosstalk, attenuation capacitance.

3.2 Mechanical

- Tension: maximum steady-state working, maximum varying tension, factor of safety.

- Diameter and tolerance

- Weight objective

  positive buoyancy
  neutral buoyancy
  maximum negative buoyancy

- Torque characteristics

  rotation per 1000 feet
  torque ratio

- Bending fatique

  number of cycles under specified conditions of
  sheave/cable D/d ratio, excursion speed

3.3 Cable Accessories (supplied by cable manufacturer or purchaser)

- Electrical connector: configuration, cable connector, type of seal to cable, bulkhead feed through

- Mechanical connector: configuration, type of mechanical bending strain relief.

- Fairings: length to be faired, description of fairing.

4.0 TEST

4.1 Qualification

Mechanical testing may include:

- ultimate breaking strength
- flexure cycling
- torque balance
- vibration
- non-hosing hydrostatic
- hydrostatic pressure cycling
- elongation and diameter of deformation

4.2 Acceptance

Electrical: IR, dielectric strength

Mechanical:
- OD
- Surface quality

5.0 MARKING AND SHIPPING

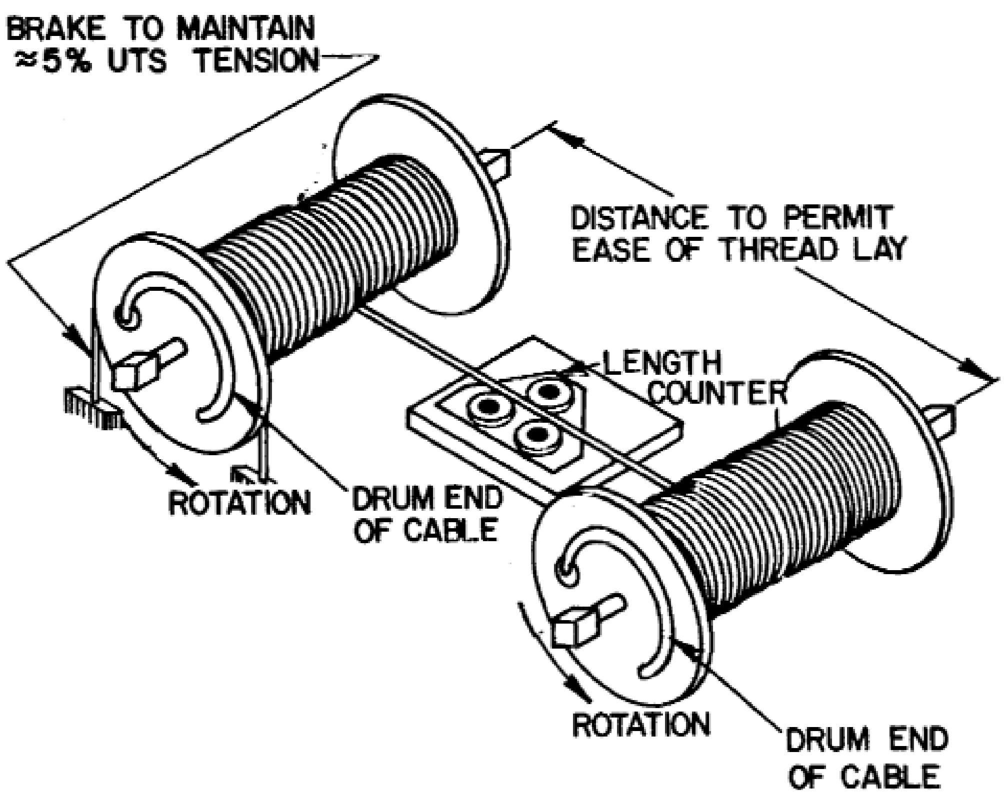
- method of marking cable

- reel requirements
  special lagging requirements
  marking of reels

- length of inner end of cable to be free

APPENDIX 13.0

REREELING SETUP



APPENDIX 14.0

CABLE LENGTH DETERMINATION BY CONDUCTOR RESISTANCE MEASUREMENT

A. CABLE HAVING A CENTER CONDUCTOR

1. Refer to the as-received conductor resistance and length.
2. Using the most accurate ohmmeter or resistance bridge available measure the center conductor resistance.
3. The cable length is:

where: L = present length of cable x 10^{-3}

$$L = \frac{R_m}{R}$$

R = conductor resistance, ohms/1000 ft.
R_m = optional measured resistance corrected to 20°C (see Appendix 4.0)

B. CABLE HAVING NO CENTER CONDUCTOR

1. Determine the relationship between the cabled conductor length and cable length using the procedure of Appendix 5.
2. As for a center conductor cable, measure the conductor resistance; this measurement will be of the cable conductor, C (correct to 20°C per Appendix 4.0).
3. Calculate the present length:

$$L = C \cos\theta \left(\frac{R_m}{R}\right)$$

θ = conductor lay angle (usually 5°-9°)

APPENDIX 15.0

CABLE LENGTH DETERMINATION BY WEIGHT

Although less accurate than the re-reeling or resistance measurement methods it may be used when only an approximate present length measurement is needed.

1. Determine the tare weight ,W_r, of the reel.

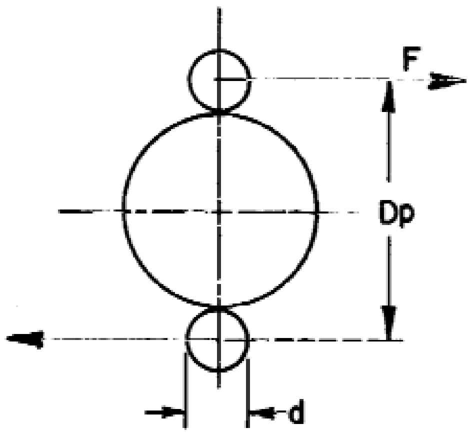
   Determine the weight per 1000 ft of cable, W, from the manufacturer's data.

   Weight the reel of cable, W_2, and calculate the length by:

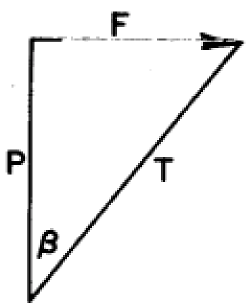
$$L = \frac{W_2 - W_r}{W} \times 1000 \text{ ft}$$

APPENDIX 16.0

E-M Cable Torque



$$\text{Torque} = M = \frac{F\,Dp}{2} \qquad (1)$$



$$F = P \tan\beta$$

$$T = \frac{P}{P\cos\beta} \qquad (2)$$

Combine equations (1) and (2)

$$\text{Torque} = M = \frac{P\,D_p \tan\beta}{2} \qquad (3)$$

Hooke's law for wires

$$E = \frac{\frac{T}{A}}{\varepsilon} \qquad E\varepsilon = \frac{T}{A} \qquad (4)$$

by assuming $E\varepsilon$ equal for the inner and outer armor layers, the ratio $(\frac{T}{A})$ for both the inner and outer armor will be equal, or:

$$\frac{T_0}{A_0} = \frac{T_I}{A_I} \tag{5}$$

express T in terms of P from (2)

$$\frac{P_0}{A_0 \cos\beta_I} = \frac{P_I}{A_I \cos\beta_I} \tag{6}$$

$$\frac{P_0}{P_I} = \frac{A_0 \cos\beta_0}{A_I \cos\beta_I}$$

sum for all armor wires

$$\frac{\sum P_0}{\sum P_I} = \frac{N_0 A_0 \cos\beta_0}{N_I A_I \cos\beta_I} \tag{7}$$

$$\sum P_0 = \sum P_I \frac{N_0 A_0 \cos\beta_0}{N_I A_I \cos\beta_I} \tag{8}$$

$$\sum P = \sum P_0 + \sum P_I \tag{9}$$

substitute (8) in (9)

$$\sum P = \sum P_I + \sum P_I \frac{N_0 A_0 \cos\beta_0}{N_I A_I \cos\beta_I} \tag{10}$$

$A = \frac{\pi}{4} d^2$ and the constant $\frac{\pi}{4}$ cancels

$$\sum P = \sum P_I \left(1 + \frac{N_0 d_0^{\,2} \cos \beta_0}{N_I d_I^{\,2} \cos \beta_I}\right) \tag{11}$$

$$\sum P = \sum P_I \left(\frac{N_I d_I^{\,2} \cos \beta_I + N_0 d_0^{\,2} \cos \beta_I}{N_I d_I^{\,2} \cos \beta_I}\right) \tag{12}$$

$$\frac{\sum P_I}{\sum P} = \frac{N_I d_I^{\,2} \cos \beta_I}{N_I d_I^{\,2} \cos \beta_I + N_0 d_0^{\,2} \cos \beta_0} \tag{13}$$

$$\sum P_I = \sum P \frac{N_I d_I^{\,2} \cos \beta_I}{N_I d_I^{\,2} \cos \beta_I + N_0 d_0^{\,2} \cos \beta_0} \tag{14}$$

$$\sum P_0 = \sum P \frac{N_0 d_0^{\,2} \cos \beta_0}{N_0 d_0^{\,2} \cos \beta_0 + N_1 d_1^{\,2} \cos \beta_0} \tag{15}$$

To obtain an expression for inner and outer armor torques, substitute (15) and (16) respectively into (3).

$$\sum M_I = \sum P \frac{N_0 d_0^{\,2} \cos \beta_0}{N_I d_I^{\,2} \cos \beta_I + N_0 d_0^{\,2} \cos \beta_0} \frac{\left(D_{pI} \tan \beta_I\right)}{2} \tag{16}$$

$$\sum M_0 = \sum P \frac{N_0 d_0^{\,2} \cos \beta_0}{N_0 d_0^{\,2} \cos \beta_0 + N_I d_I^{\,2} \cos \beta_I} \frac{\left(D_{p0} \tan \beta_0\right)}{2} \tag{17}$$

$$\sum M = \sum M_0 - \sum M_I \tag{18}$$

Substitute (16) and (17) into (18).

$$\sum M = \sum P \frac{N_0 d_0{}^2 D_0 \cos\beta_0 \tan\beta_0 - N_I d_I{}^2 D_I \cos\beta_I \tan\beta_I}{2(N_0 d_0{}^2 \cos\beta_I + N_I d_I{}^2 \cos\beta_I} \tag{19}$$

$$\sum M_I = \sum P \frac{N_0 D_0 \cos\beta_0 \tan\beta_0 - N_I D_I \cos\beta_I \tan\beta_I}{2\left(N_0 \cos\beta_0 + N_I \cos\beta_I\right)} \tag{20}$$

$\left(d_0 = d_I\right)$

$$\sum M_I = \sum P \frac{D_0 d_0{}^2 \cos\beta_0 \tan\beta_0 - D_I d_I{}^2 \cos\beta_I \tan\beta_I}{2\left(d_0{}^2 \cos\beta_0 + d_I{}^2 \cos\beta_I\right)} \tag{21}$$

$\left(N_0 = N_I\right)$

F = component of armor ire tension tangential to cable axis (lb)

Dp = pitch diameter of wire (in)

T = armor wire tension (lb)

P = component of armor wire tension parallel to cable axis (lb)

β = armor lay angle (degrees)

M = torque (in-lb)

β = armor lay angle (degrees)

subscripts:

0 = outer armor
I = inner armor

APPENDIX 17.0

REPRESENTATIVE LOAD vs ELONGATION VALUES
(at 50% of Breaking Strength)

Dia. (inch)	Cable No. of Conductors	Wires (No./Dia.)	Amor Elongation at 5O% of BS (%)
.185	1	12/.0355 12/.022	0.50%
.206	1	15/.033 9/.033	0.80%
.221	1	15/.0355 11/.031	0.48%
.319	1	18/.044 12/.044	0.66%
.428	1	181/.059 181/.042	1.10%
.376	7	23/.042 17/.042	0.60%
.427	7	18/.059 18/.042	0.70%
.464	7	24/.049 24/.039	0.64%
.520	7	20/.064 19/.052	0.61%

APPENDIX 18.0

Calculations for Physical Properties of E. M. Cables

Wa	=	weight in air (lb/M^1)
B	=	buoyancy (lb/M^1)
Pw	=	density of sea water (lb/fl^3) ~64 lb/ft^3
Ac	=	cable cross-sectional area (in^2)
d	=	cable dia. (in)
M^1	=	1,000ft
Scw	=	specific gravity of cable in sea water
Pca	=	specific gravity
T_B	=	breaking strength (lb)
Ww	=	weight in water (lb/m’)

1.0 Weight in sea water (Ww) of jacketed cables (armored or unarmored)

$$Ww = Wa - B$$

$$B = 1{,}000 \text{ ft } \times Ac \times Pw$$

$$B = 1{,}000 \times \frac{\frac{\pi}{4d^2}}{144} \times 64 \frac{lb}{ft^3}$$

$$B = 349d^2$$

$$Ww\Big|_{\text{Jacketed}} = Wa - 349d^2$$

2.0 Weight in sea water of non-jacketed, armored cable approximately 10% void in armor interstices (i.e., 0.9 x Ac)

$$Ww\Big|_{\text{nonjacketed}} = Wa - d^2\,(349 \times 0.9) = 314d^2$$

3.0 Specific gravity in sea water (jacketed)

$$Scw = \frac{Pc\,a}{Pw}$$

$$Pca = \frac{144\ Wa}{\frac{\pi}{4} d^2 \times 1{,}000}$$

$$Scw = \frac{144\ Wa}{\frac{\pi}{4} d^2 \times 1{,}000 \times 64} = \frac{0.2865\ Wa}{100\,d^2}$$

4.0 Specific gravity in sea water (non-jacketed)
(approximately 10% voids in armor interstices)

$$Scw = \frac{144\ Wa}{0.9 \times \frac{\pi}{4} d^2 \times 1{,}000 \times 64} = \frac{.3185\ Wa}{100\,d^2}$$

5.0 Strength-to-weight ratio in water

$$= \frac{T_B}{Ww}$$

APPENDIX 19.0

PRINCIPLES OF E-M CABLE SPLICING

The following presentation is intended to only present the principles which apply to E-M cable splicing, not for a working procedure.

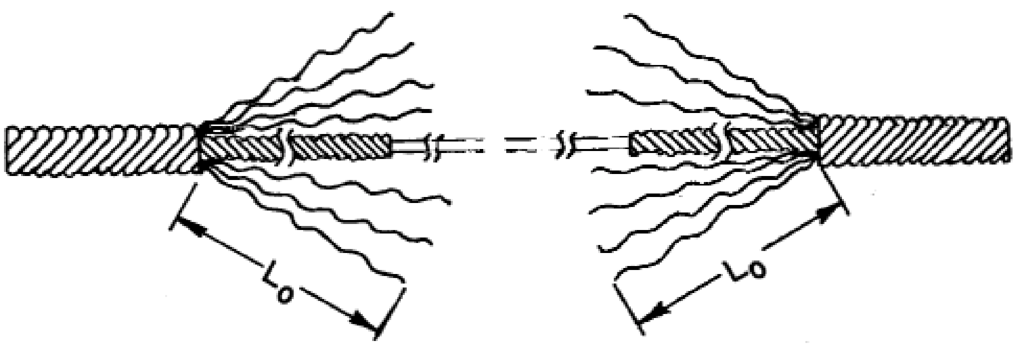
1. From the ends of the cable to be spliced unstrand the outer armor a distance from each end equal to:

$$L_0 = 6N_0I_0$$

where L_0 = unstranded length of outer armor wires

I_0 = lay length of outer armor

N_0 = number of outer armor wires



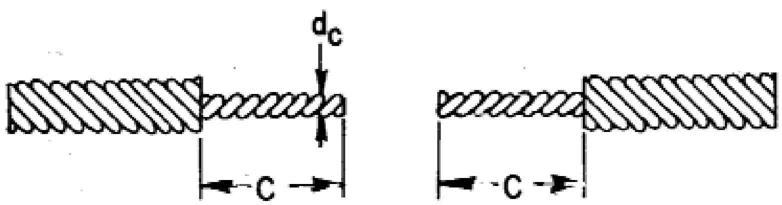
For convenience in handling the unstranded wires may be taped in groups of three.

2. Using the above length determination, unstrand the inner armor wires and tape them. The shorter lay length of the inner armor will result in $L_I < L_0$ where L_I = unstranded length of inner armor wires.

3. Note the rifling, or helical grooving, which the inner armor has compressed on the core. It is desirable to replace the inner armor wires into this grooving.

4. Square cut the core so that there is a continuity of the helical grooving.

5. Prepare the insulated conductors for splicing by cutting the insulation at a distance of approximately 20 times the conductor diameter shown in Appendix 3.0.



$c = 20\, d_C$

where C = length of stripped conductor

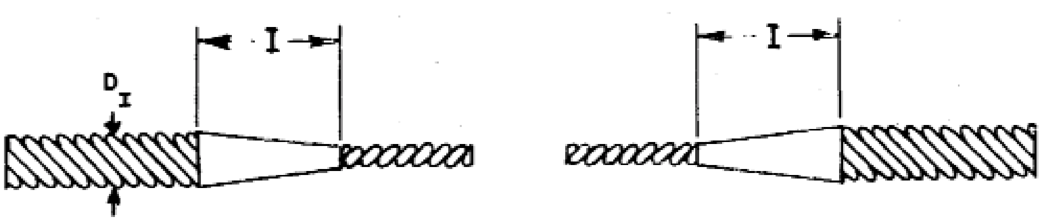
d_C = diameter of conductor

6. Prepare a conical surface on the ends of the insulation with the height of the truncated cone being:

$I = T\, d_I$

where I = Height of truncated cone

d_I = Insulation diameter



7. Separate the conductor wires in preparation for splicing. Two splicing methods are used:

   (a) combing and tying

   (b) soldering

The soldering method consists of combing, or enmeshing the wires and, using a minimum amount of solder, bond the wires without allowing the solder to wick along the stranded conductor. A long soldered joint creates a stiff section having end discontinuities which form points of stress concentration.

The combing and tying procedure for conductor splicing has two versions which are in popular use:

(a) comb complete strand to obtain a splice cross-section having twice the number of wires as the basic strand.

(b) cut wires from each of the ends to be spliced so that the number of wires in the splice cross-section is the same as that of the basic strand.

When the wires of each strand are combed, or intermeshed, the strands not just overlapping, tie them using the lowest denier, unwaxed dental floss available. This tying is intended to bind the wires to obtain a few pounds pull-out strength but allow a length adjustment of the spliced conductor when the completed cable splice is tensioned.

The insulation splice void is filled with wraps of commercially-available, self-adhesive Teflon tape which is 0.0005 in. thick and 0.25 wide. Avoiding wrinkles or other discontinuities, the wrapping uses a 50% overlap of the tape. Continue wrapping to maintain an increasing uniform diameter. As the diameter increases the taped length will also increase until the splice diameter reaches the original insulation diameter.

8. The inner armor wires are restranded into the original grooving in the core. The wire splices should alternate between ends; i.e., a short wire of one end should adjoin a long one of the same end. The wire splices should be separated by a distance of about five times the lay length.

When the restranding operation is completed the wire splice locations can be reviewed prior to cutting the wires and laying them in place! It is not necessary to bond the inner armor wires.

9. The outer armor wires are restranded using the same procedure as for the inner armor. There are four procedures in use for treatment of the wire ends of the outer armor; they are:

    (a) butted only as is the inner armor.

    (b) soldered, whereby the butted wires are silver soldered to the adjoining wires.

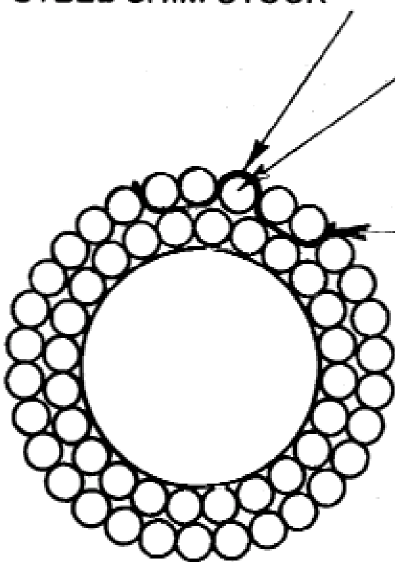
    (c) shim stock spliced as illustrated on page 2-21.

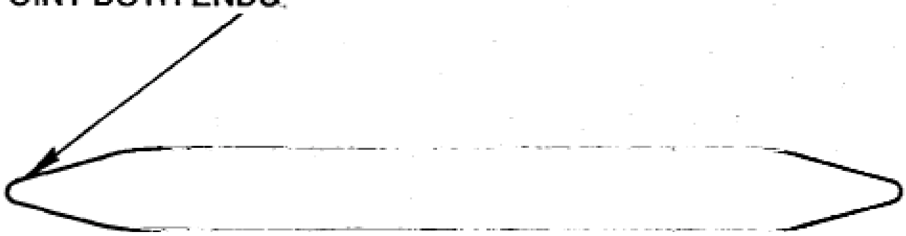
    (d) butt welded. This procedure is used to join broken wires during the armoring process as discussed in Par. 5.8 but is rarely used as a part of cable repair procedure.

10. The diameter of the spliced section will usually be found to be larger than an unspliced section. It is therefore desirable to condition the splice before releasing the cable for service. This can be accomplished by running the cable over a sheave several times; somewhat duplicating the manufacturers prestressing procedure as described in Par. 5.9. The cable diameter as well as electrical continuity should ‘be monitored.

    Service shops may use offset rolls for this prestressing procedure; they are depicted in Fig. 2-34.

SHIM STOCK SPLICE

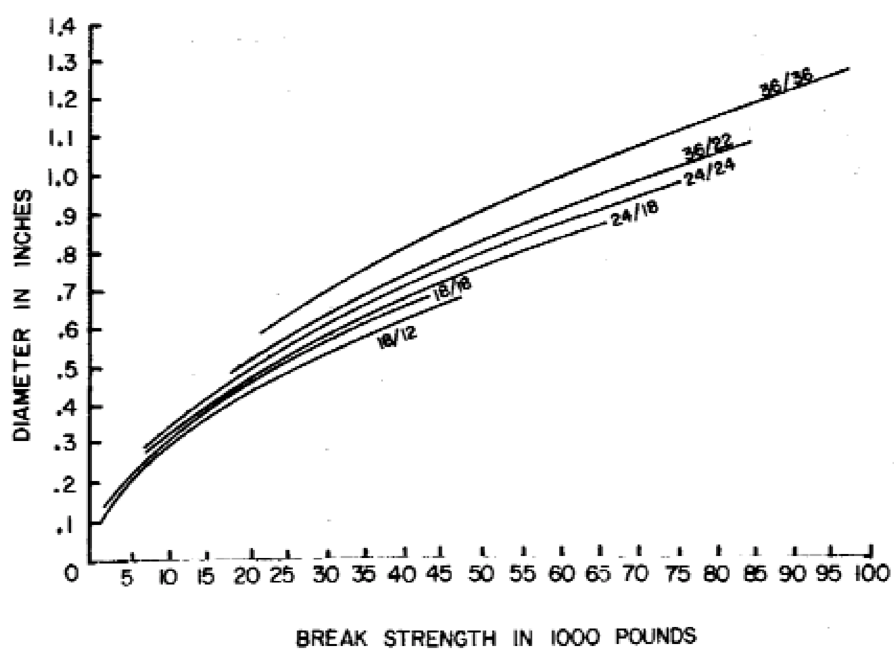




APPENDIX 20.0

ARMORED CABLE DIAMETER vs BREAKING STRENGTH

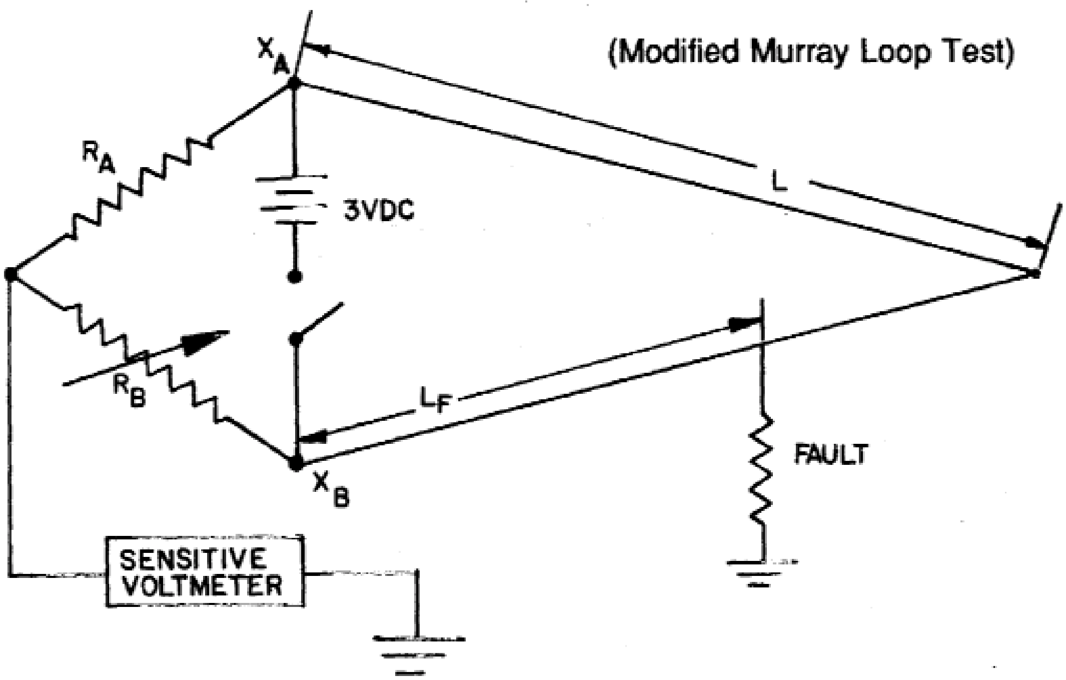
XX/XX= NO. WIRES OUTER ARMOR LAYER/NO. WIRES INNER ARMOR LAYER



The above curves indicate Outer Diameter verses Break Strength for various double armor constructions. The data is based on using standard galvanized improved plow steel tensiles. For example, a 3/4" diameter cable, with 24 outer wires and I8 inner wires, has approximately 46,000 lbs ultimate break strength.

APPENDIX 21.0

LOCATION OF SHORT TO ARMOR IN MULTICONDUCTOR CABLES



where R_A & R_B = Resistances of bridge arms at balance

L	= length of each conductor
L_F	= distance to fault from X_B
R	= resistance of an unfaulted conductor of length L
R_F	= resistance of conductor length L_F from X_B

Ratio the two sides of the bridge

$$\text{or} \quad \frac{R_F}{2R} = \frac{R_B}{R_A + R_B}$$

Express R_F and R in terms of length equivalent

$$\frac{R_F}{R} = \frac{L_F}{L_B} \quad .$$

CHAPTER 3

HIGH STRENGTH SYNTHETIC FIBER ROPES

A. Simeon Whitehill, Jr.

1.0 INTRODUCTION

Chapter 3 describes ropes made from high strength synthetic materials for oceanographic towing, mooring and lifting. The chapter is broken in four main sections.

High strength synthetic rope compared to wire

Materials available for rope construction

Constructional changes that alter rope performance

Summary and Reference Material

2.0 HIGH STRENGTH SYNTHETIC FIBER ROPES VS STEEL

A primary advantage of Synthetic Fiber Rope is their lightweight. Lightweight lines are easier to handle and reduce topside weight. High Strength Synthetic Fiber Rope also can be used in greater depths than wire.

2.A Weight Comparison

Kevlar density is less than 1/5 that of steel and Spectra's density is less than 1/8 that of steel. A 1" Kevlar, Spectra, and wire each have approximately 125,000 pound break strength. However, the weight per 100/ft is very different:

	Approximate Weight/100'	
	in air	in water
Steel	185 Lbs.	161 Lbs.
Kevlar	36 Lbs.	10 Lbs.
Spectra	26 Lbs.	0 Lbs.

2.B Payload Comparison in Water

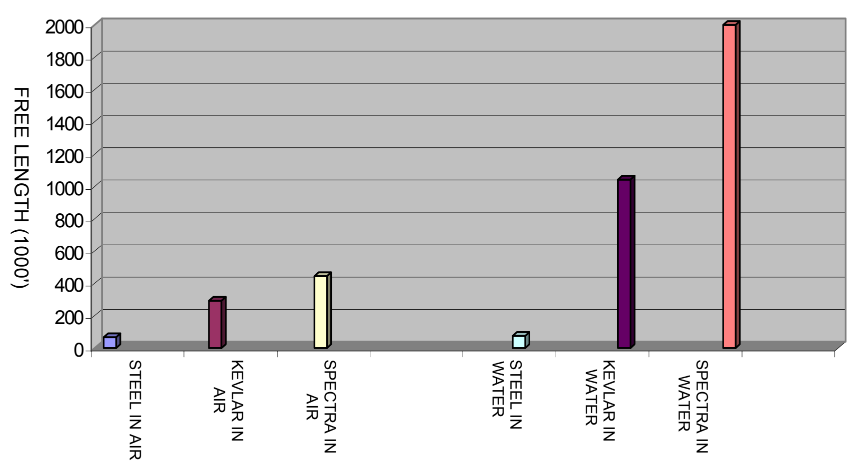
There are significant weight savings when synthetic rope is used in water. As stated above, Kevlar rope is 1/5 the weight of steel in air. In water, Kevlar weighs 28% of its weight in air. This means significantly higher payloads in oceanographic lifting. An extreme example is a 20,000 foot long, 1/2" wire rope (8,570 pound working load - 25,700 pound break strength). The rope itself weighs 6,820 pounds. With 1,000 pounds needed for over-pull, the payload of this system is 750 pounds.

A 1/2" diameter Kevlar rope could be used in this system, still getting the required 26,000-pound break strength, but 20,000 feet of rope would weigh 500 pounds in water. Assume the same 1,000 pound over-pull, the payload can be increased fivefold to 3,750 pounds. Using Kevlar, the total system would have a factor of safety approximately 5 to 1, not the 3 to 1 of steel.

2.C Free Length

Free length is that length at which a rope breaks under its own weight. It can be found by dividing a rope's break strength by its weight per foot. Taking the same 1/2" diameter rope discussed above, Figure 3-1 shows graphically, the difference between steel, Kevlar, and Spectra ropes in air and sea water.

FIG 3-1



2.1 FATIGUE RESISTANCE

The fatigue properties of synthetic rope are outstanding and the rope's construction can be adjusted to achieve additional cyclic life.

2.1.A Bend Over Sheaves

Kevlar ropes have been demonstrated to withstand 50,000 bending cycles over sheaves forty times the rope's outside diameter at 35% of the rated break strength, without failure. Residual strength is 95% of rope's original rated break strength. Therefore, Kevlar rope's performance, under these conditions, is comparable to 6 strand steel wire rope that is not torque balanced.

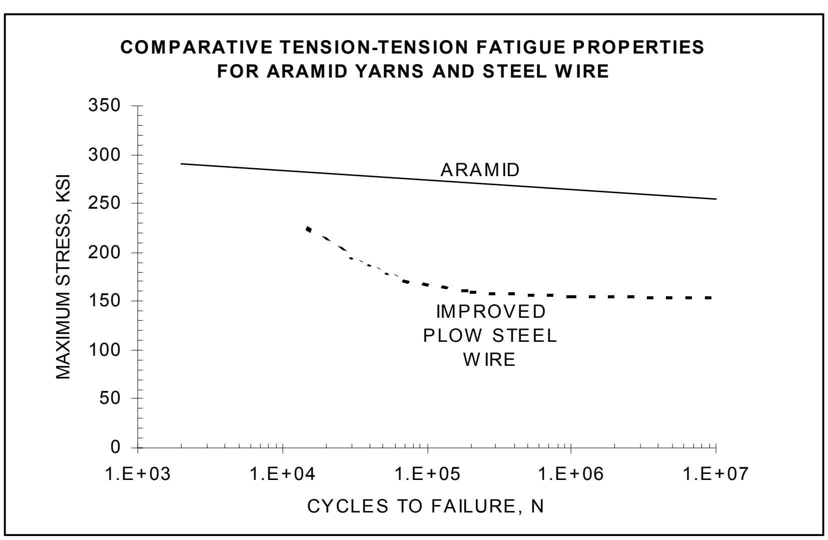
Kevlar has been demonstrated to have outstanding performance at a sheave diameter/rope diameter (D/d) ratio of 40:1 and a relatively high factor of safety. It is more common, however, to use Kevlar ropes at a 25 to 1 or 30 to 1 D/d ratio and a 5 to 1 or 6 to 1 factor of safety.

Space limitations can limit the sheave size. D/d's as small as 20 to 1 have been used but reduced fatigue life. Endless cycling of the same section of a rope, as on a motion compensator, can wear a rope locally, due to the fast accumulation of bending cycles. In either case special rope designs can be produced that will extend fatigue life.

2.1.B Tension-Tension Fatigue

Since all ropes are subject to fluctuating loads, tension-tension fatigue performance is an extremely important characteristic. Figure 3-2 shows the superior fatigue performance of Kevlar 29 yarn to improved plow steel wire. Residual strength of Kevlar 29 and steel wire at 10,000,000 cycles was above 95% for both materials.

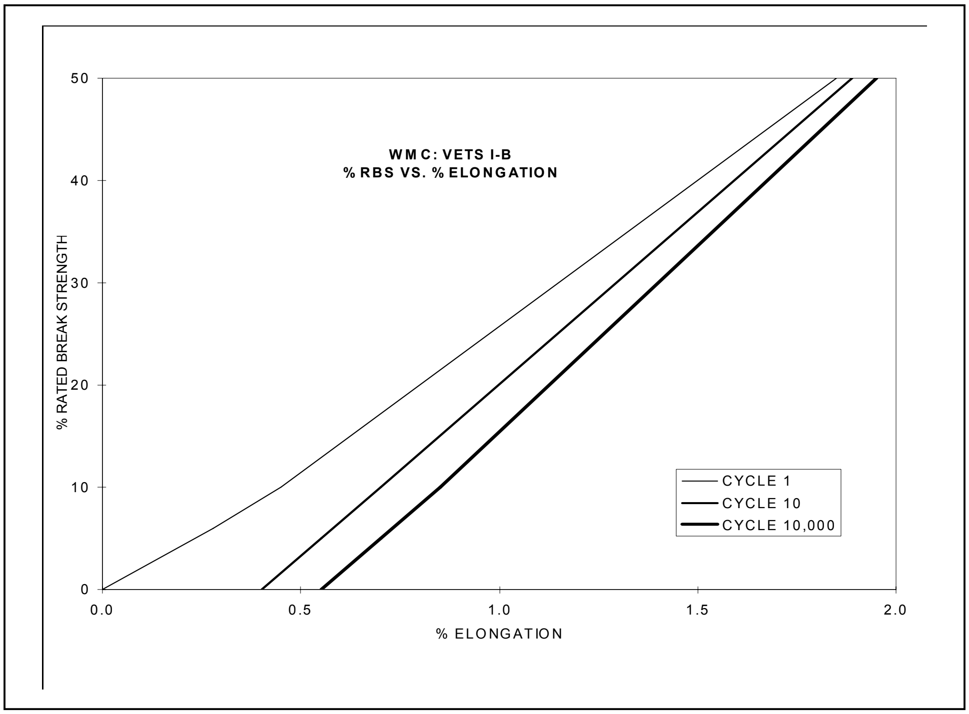
FIG 3-2



2.2 LOW STRETCH

Due to the high modulus of Kevlar there is minimal stretch and stored energy over the working load range. Figure 3-3 is a load-elongation curve for Kevlar. Ropes made from other synthetic fibers like nylon have about 10 times the stretch of Kevlar. Low stretch gives the user better control over the load's position and faster reactions to the load touching the bottom.

FIG 3-3



Less stretch also means less stored energy in Kevlar ropes. That can be invaluable. For example, taking a core sample, there is one tenth the recoil with Kevlar than nylon rope. Less recoil reduces the damage to the core sample.

2.3 DISADVANTAGES

The main disadvantage of these materials is that they are easy to damage. Being less tough than steel wire rope, synthetic rope shows the damage it receives. Wire rope damage often tends to look less severe than it actually is and is frequently used even when it should be retired.

3.0 MATERIALS

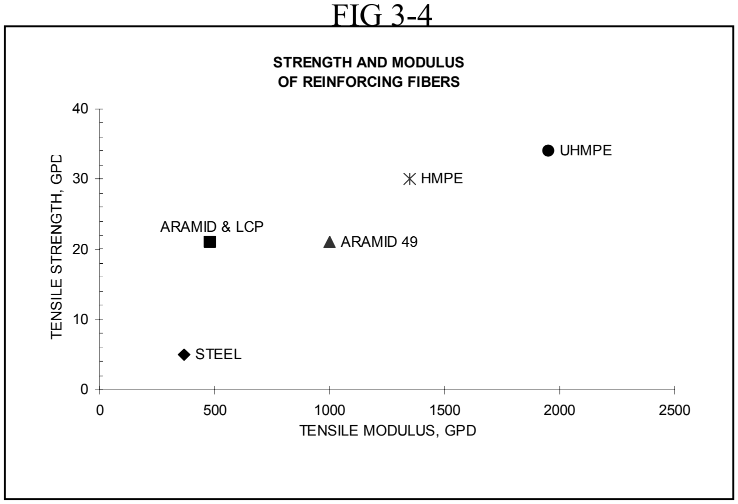
Several fibers are currently available for rope in oceanographic applications. Several of the material's physical properties are compared numerically in the following table. Figure 3-4 graphs the materials specific strength vs specific tensile modulus.

Physical Property Comparison of
Kevlar, Sepectra, and Steel

	Kevlar 29	Kevlar 49	Spectra 900	Spectra 1000	Steel
Tenacity (gpd)	23	23	30	35	2.9
(psi)	400,000	400,000	375,000	435,000	285,000
Modulus (gpd)	525	850	1400	2000	200
(1,000,000 psi)	10	18	17	25	30
Elongation (%)	3.7	2.5	3.5	2.7	2.0
Density (g/cc)	1.44	1.44	.97	.97	7.86
Meltig Point (°C)	500°**	500°**	147°	147°	>1199°
Denier	1500	1500	1200	650	
No. of Filaments	1000	1000	118	120	

*Galvanized Improved Plow Steel
**Does not melt - It chars

FIG 3-4



3.A Kevlar

DuPont introduced two para-bonded aromatic polyamide fibers, Kevlar 29 and Kevlar 49, in 1971. Kevlar exhibits high strength, high flexibility, high modulus, low elongation, low density, non-conductivity, and corrosion resistance. Kevlar also has exceptional thermal stability over a wide range of temperatures, from -46°C to 160°C with minimal change in tensile strength. Kevlar 29 and 49 have similar tensile and thermal properties but Kevlar 49 has higher modulus, lower elongation, and a higher price. Most of the information contained herein applies to Kevlar 29 because of the experience with Kevlar 29 over the years.

3.B Other Aramid Fiber

Two types of Aramid are manufactured outside of the United States. Teijin Limted produces Twaron in Holland and Technora in Japan. These fibers demonstrated properties similar to Kevlar except Technora, which shows slightly better chemical resistance to high concentrations of acids and bases. These fibers are available in the United States on a limited basis.

3.C Spectra

Spectra 900 and Spectra 1000 are ultra high modulus polyethylene fibers developed by Honeywell/GE. Spectra fibers combine a high degree of molecular orientation with a density lower than water. Therefore, Spectra can be made into buoyant ropes. Spectra demonstrates high specific modulus, high specific strength, excellent chemical resistance, and high abrasion resistance. The disadvantages of Spectra are its high price, tendency to creep, and limited temperature range. Principal advantages are lighter weight (buoyant) and longer cyclic bend over sheave flex life.

4.0 JACKETS

For use on oceanographic winch applications Kevlar ropes generally require protective jackets. Such jackets are intended to provide protection from ultraviolet light and external abrasion.

4.A Braided Jackets

Several fiber materials have been evaluated for braided jackets and polyester has proven best for general purpose use. Polyester stays tight on the rope and facilities load transfer from the force of a traction winch to the rope.

Braided polyester is relatively soft and conforms to the outside surface of the Kevlar assuring direct load transfer while cushioning and protecting the load bearing fiber.

4.B Extruded Plastic Jackets

Polyurethane, Polyethelene, Zytel, Hytrel and other materials have been used for protective jackets. Such materials provide somewhat better protection from ultraviolet light and other environmental considerations but generally lack the toughness to protect the rope from abrasion. For long term standing rigging, such as tower guy, extruded jackets perform well but they are not recommended for working ropes.

When a rope is in tension the diameter may be reduced, thus causing the jacket to become a loose sleeve. When pulled around a traction the jacket will bunch up and tear.

4.C Combination Jackets

For extreme cases, combination jackets have been successful. For combination jackets the rope first has an open braid with about 50% coverage applied to the Kevlar rope. The desired plastic is then pressure extruded into the rope. The braid provides a form of fiber reinforcement to the plastic as they become interlocked.

Such jackets provide a degree of better protection but increase diameter (more drag), bending stiffness, weight, and cost.

5.0 TERMINATIONS

The final break strength of a rope is often determined by the efficiency of the rope's termination. Some terminating techniques can develop a rope's full break strength, while others severely limit the break strength. Ropes can be supplied already terminated or the user can terminate them himself as needed.

5.A Improved "Hood" Splice

The best terminations of the multi-strand ropes with a single jacket is the Improved "Hood" Splice. It is a modification of the Braidback Splice and develops 100% of the break strength of a rope. Directions for this splice are beyond the scope of this paper but may be obtained from the writer.

5.B Eye Splice

An efficient method to terminate jacketed ropes is the eye splice, similar to the splices used for wire and natural fiber ropes. These splices are constructed by forming an eye near the end of the rope, then tucking the tails of the rope back into the rope's body. The point where the last tuck enters the rope is where it usually fails. The actual type of splice required depends on the rope's construction. Detailed splicing directions for many types of ropes are available from the manufacturer.

5.C Mechanical Terminations

Several forms of mechanical terminations have been tested with Kevlar ropes including: potted epoxy plugs, swage fitting, nicopress sleaves, and wire rope clips. These terminations develop 50% - 75% of the rated break strength of a rope due to problems with load concentration at the end of termination. These terminations may be useful in applications where rope has excess strength and quick easy terminations are important.

Testing to-date, of mechanical terminations, are basically break strength. Cyclic tension-tension fatigue tests, at reasonable working levels, should be conducted before selecting a mechanical termination for a dynamic application.

6.0 OTHER CONSTRUCTIONS AND APPLICATIONS

This chapter discusses only two constructions of synthetic rope that perform well with oceanographic winch systems. Kevlar ropes are made in most wire rope constructions for wide variety of applications. Some additional uses of Kevlar and Spectra ropes include:

Oceanographic Mooring
Balloon Tethers
Mine Sweep Cables
Riser Tensioners
Moorings on Oil Rigs
Winch Lines for Utility Trucks
Helicopter Slings
Oil Containment Booms
Lift Lines of Cranes

7.0 SUMMARY

Since there are so many uses for synthetic rope, it is impossible to have one material or construction to suit each application, making a variety of ropes necessary. However, for most oceanographic lifting applications the WMC:VETS 1-B type, stranded Kevlar ropes, listed in the "User Information" section, provide a good combination of characteristics. These characteristics include: small diameter, light weight, low stretch, torque balance with good bend over sheave, and outstanding tension-tension fatigue life. Combining these types of rope with a reliable termination provided 100% strength translation.

7.A Reference Tables

KEVLAR

Outside Diameter (inches)	Break Strength (pounds)	Weight In Air (lb./1000')
3/16	3,500	16
1/4	7,500	28
5/16	12,000	40
3/8	17,800	62
7/16	23,500	74
1/2	28,750	89
9/16	37,500	102
5/8	46,000	128
3/4	63,000	180
7/8	89,000	255
1	125,000	360
1-1/4	160,000	520
1-1/2	200,000	600

SPECTRA

Outside Diameter (inches)	Break Strength (pounds)	Weight In Air (lb./1000')
3/16	3,200	12
1/4	6,800	21
5/16	11,000	29
3/8	16,500	45
7/16	22,500	54
1/2	28,500	64
9/16	35,000	78
5/8	43,500	100
3/4	59,500	137
7/8	85,000	195
1	120,000	274
1-1/4	165,000	368
1-1/2	200,000	460

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CHAPTER 4

FIBER OPTIC TELEMETRY IN OCEAN CABLE SYSTEMS

George A.Wilkins

1.0 INTRODUCTION

The 1980s have seen major improvements in the strength, ruggedness, and attenuation of deepsea armored coaxial cables. Some of these gains have been due to more efficient geometries, and to the development and application of precise cable analysis techniques. Another contributor has been the availability of armor steels with higher strengths at little or no sacrifice of flexibility.

During these same years, even greater advances have occurred in the performance and reliability of fiber optic cables. These advances have been so striking that fiber optics should soon begin to replace electrical telemetry in many areas of undersea telemetry. This replacement should be especially rapid in systems where performance is constrained by some mixture of:

(a) A need for more bandwidth than conventional telemetry cables can supply without compromising systems constraints on the cable's diameter, volume, weight, and handling system.[1]

(b) A need, beyond the capability of conventional telemetry, for the cable and handling System to be very much smaller, lighter, and more transportable.

(c) An anticipated growth of system data requirements that will require a major expansion of telemetry bandwidth, but with no increase in cable diameter. Cable diameter must become essentially independent of bandwidth.

(d) A requirement for long cable runs (many 10's of km) without a repeater. For example, if the cable cannot be used to transmit electrical power, then transmitters and receivers must be located (only) at the ends of the telemetry system.

(e) A constraint that telemetry system cost (including handling and deployment) must be as low as possible. For example, the data link may be expendable.

[1]The classic reason usually given for (often reluctantly) choosing fiber optics is the need for telemetry bandwidth. Other constraints, of the types described above can convert this initial timidity to enthusiasm.

This chapter will discuss recent developments which have brought armored-coax telemetry to a high level of maturity. It will go on to describe contributions that fiber optics can make in exceeding these levels, as well as special guidelines and hazards involved in the use of this new technology. Cable design examples will be presented and explained. Finally, in a topic where knowledge and guidelines are still deficient, this chapter will discuss the influences that fiber optic telemetry can and should have on the cable handling system.

2.0 THE STARTING POINT--THE DEEPSEA ARMORED COAX

It has been the author's experience that the development of deepsea optical cables has been an evolutionary sequence rather than revolutionary. We have known the capabilities and the limits of those conventional cable technologies from which the development began. The development objectives have generally been known. Finally, we have been able to maintain a relatively clear view of the path which best connected these two end points. This is a reasonable definition of continuous evolution. In a revolution, progress tends to be sudden and often discontinuous and many of the participants are likely to lose their heads.

2.1 The Navy/Scripps Armored Coax

The armored coaxial cable sketched in Figure 4-1 should be familiar to most readers, since it has been in common use as a deepsea oceanographic tether since about 1970. It is often known as the Navy or Scripps deep tow cable, and has its most-probable origin in the 1960s as an armored lift cable for elevators and mine shafts. This cable, which is still in general use, has a number of serious design flaws.[2]

(1) The soft copper wires of the center conductor are too large and stiff, and the conductor becomes inelastic at a very low value of tensile strain. When the cable is subjected to severe tensile or flexure cycling, these wires are likely to "ratchet." That is, each stress cycle can cause the center conductor to suffer another increment of permanent strain. Finally, the conductor becomes so much longer than the surrounding

[2]Ref. (1) describes the response of this cable to accelerated flexure over 35.5-cm-diameter sheaves at 6,350-kg tension.

cable structure that it Z-kinks into the dielectric spacer. At this point, the conductor may either break or its component wires may force their way through the insulation to short against the shield conductor.

(2) The shield conductor is fabricated as a soft copper braid and tends to break up rapidly during loaded flexure.

(3) The elastomeric materials that form the dielectric spacer and the coax jacket are soft and can easily be punctured by the metal shards resulting from defects (1) and (2).

(4) The contrahelical steel armor package is initially stress balanced but, because of the larger outside wires, has a 2/1 torque mismatch. As a result, the cable suffers very high rotation under load when one end is unconstrained. This unbalances stress in the two armor helices and results in a 30% loss of strength. (The “one end free” mode is common in deep oceanographic systems where the instrumentation package is normally fitted with a swivel.)

While this strength loss is serious, the lack of torque balance is of even greater concern. If the package strikes the seafloor and unloads the cable, then the considerable rotational energy stored in the lowest cable section is suddenly released. The resulting reaction can lift the now-slack cable high enough to form a loop. When the cable is again loaded, this loop can tighten into a hockle--or worse, with catastrophic results.

2.2 <u>The UNOLS Armored Coaxial Cable</u>

During the summer of 1983, the Woods Hole Oceanographic Institution (WHOI) was asked by the National Science Foundation to negotiate a large purchase of deepsea armored cables for the U.S. National Oceanographic Laboratories (UNOLS). In response, WHOI hosted a meeting of UNOLS representatives to determine what the nature and design of these cables should be. The author was working at WHOI that summer and was asked to attend.

The UNOLS group quickly agreed on the critical issues. <u>First</u> a single cable design should be adopted so that deepsea tethered systems could be readily transferred among the UNOLS laboratories. <u>Second</u>, the diameter of this standard tether should be 17.3 mm (0.680); a “best” fit to the oceanographic winches used by most laboratories. Third the new design should be a major improvement over the current deepsea tether cable.

A fourth point was accepted in principle. The new tether cable should be a "last generation" version of the conventional coax. That is, the next generation deepsea cable should include fiber optic telemetry. As a goal, the new armored coax should also be compatible in size, strength, power transfer and handling characteristics with this new electro-optical (E-O) tether.

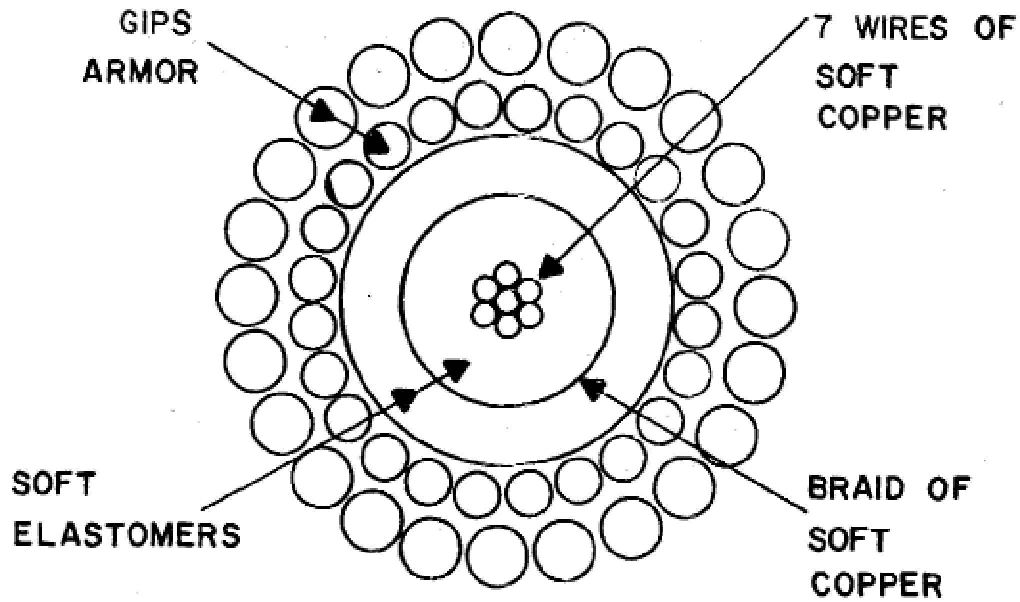
The new armored coax, Figure 4-2, *was* built by The Rochester Corporation (TRC) in late 1983. It is similar to one formally proposed to the Navy by the author in 1973 and built by TRC for the Naval Oceanographic Office in 1982. It differs from these primarily in the use of work-hardened copper[3] and ultra-high-strength steels. Key features of the 1983 design include:

(1) The center conductor was changed to a 12-around-6 structure of half-hard copper wires, formed as a unidirectional helix around a monofilament. This increased the tensile compliance of the conductor during load or flexure cycling.

(2) To increase its penetration resistance, the coax dielectric was fabricated from a high density polyethylene.

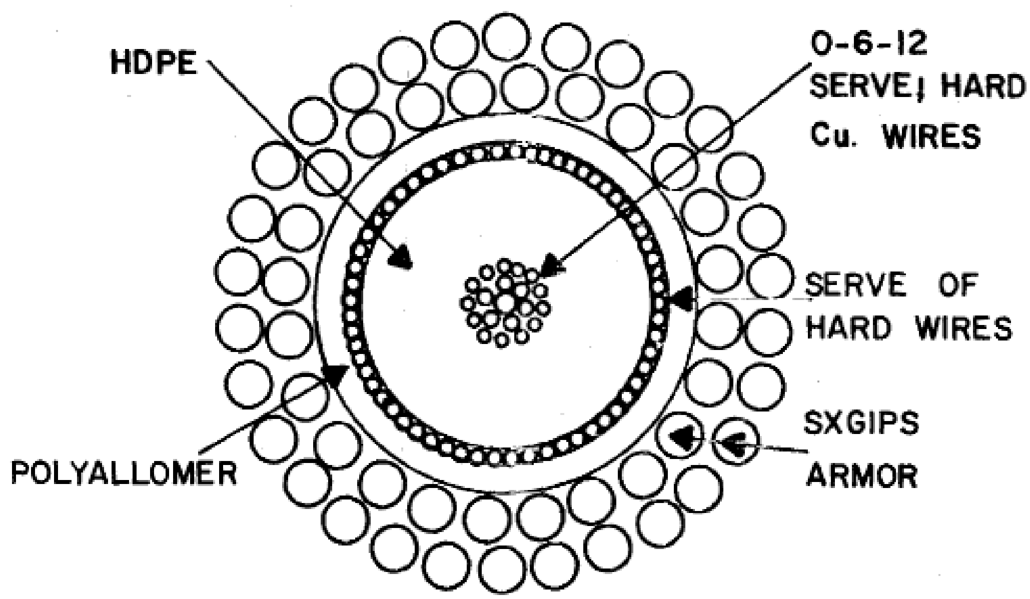
(3) To eliminate crossovers and self abrasion, the braided-shield conductor was replaced with a served shield of work-hardened copper wires. A conductor of this type is likely to exhibit radiation leakage and crosstalk, since the interwire openings behave somewhat like the Taylor's slits in an optical diffraction experiment. To correct for this, a copper-backed-polyester tape was wrapped around the copper wires to establish short circuits across the slits. This technique was used successfully in 1974 during the development of the KEVLAR-

[3]Copper wires are usually work hardened during the drawing process. But the cable manufacturer, often responding to a customers specification, is likely to purchase the wires in a soft annealed temper. To satisfy this requirement, the hardened (i.e., elastic) wires must be heated and annealed to return them to a dead soft condition. Instead of paying a cost penalty for tempered wires, the customer may actually pay a premium to eliminate this desirable feature.



NAVY/SCRIPPS DEEPSEA COAX---CIRCA 1972

FIGURE 4-1



UNOLS DEEPSEA COAX---CIRCA 1983

FIGURE 4-2

armored tether cable for the Navy's Remote Unmanned Work System (Ref. 2 & 3).

(4) The armor wires were a special galvanized steel, which combined excellent flexure performance with an ultimate tensile strength of 22,000 kg/sq-cm (315,000 psi). A common wire diameter was used to simplify shipboard armor repairs.

2.3 Comparison of the New and Old Coax Designs

Complete torque balancing of the new cable design would have required unbalancing the tensile stresses in the two armor layers and, therefore, would have resulted in a severe reduction of cable strength. As a compromise, a slight residual torque was allowed. The resulting rotation caused barely 1% loss in cable strength. More important, the energy stored in cable rotation was so small that, for reasonable payload in-water weights, the sudden loss of tension during any bottom impact could never cause the cable to lift high enough to form a loop or hockle. This had been the first design goal.

Table (1) compares the performance of the new armored coax to that of its Navy/Scripps predecessor. Note that, for the both-ends-fixed operating mode, there was very little increase in cable ultimate strength. This was because much of the additional strength that could have been gained by selection of SXGIPS steel armor was reinvested to reduce the RF attenuation of the cable's (now larger) coax core. (Attenuation in the UNOLS cable was approximately 67 dB at 5 MHz through a 6,000-meter length).

The second major design goal had been to reduce the loss of tensile strength that normally results when one cable end is free to rotate. This is a much more general operational situation. As Table (1) shows, the new armored coax loses little strength for this condition. In fact, it is about 45% stronger than the Navy/Scripps design.

The payoff of this low tension/rotation sensitivity is made more clear if we examine its impact on the tethers ability to sup-port deepsea weights. Because of its weight and low strength, the Navy/Scripps cable had an inherent (i.e., self-weight) safety factor less than 2.5 during deployment (one end free) to 6,000 meters. Adding the weight of an instrumentation package at this depth forced the cable static safety factor to fall even farther short of this reasonable constraint.

TABLE 1

PERFORMANCES OF TWO DEEPSEA ARMORED COAXES

Cable Parameter	1972 Navy/SIO	1983 UNOLS
Diameter (mm)	17.3	17.3
Strength (kg)		
Ends Fixed	17,000	17,800
One End Free	11,900	17,300
Weight (kg/km)		
In Air	1,070	1,020
In Water	820	795
Free Length (m)		
Ends Fixed	20,800	22,300
One End Free	14,500	21,800
Payload (kg)*		
Ends Fixed	1,880	2,350
One End Free	None	2,150

*For operations to 6000 meters, with the lower cable end free to rotate. The system's static strength/weight safety factor is 2.5

The Navy/Scripps cable can carry a 1,500-kg (in-water weight) package to a depth of 6,000 meters only if its static safety factor is reduced to 1.85. The low torque and rotation of the UNOLS design give it a static safety factor of 2.76 for this same payload and depth.

Eight copies of the UNOLS cable have been purchased from TRC since 1983. These are now being operated by both Canadian and U.S. oceanographic laboratories, including Scripps and WHOI. The cable is reported to be well behaved, with little performance degradation during tensile or flexure cycling. Cable torque and rotation are reportedly so low during deep-tow operations that no differences are observed when the payload swivel is switched in and out of the system (Ref. 4).

The UNOLS armored coax cable was used successfully by WHOI's Deep Submergence Laboratory during the search for RMS TITANIC in 1985. In its role as the primary instrumentation and support tether for the deepsea search system ARGO, this "last generation coax" telemetered the first video photographs of the wreckage of that ship.

2.4 New Non-LinearAnalvsis Technique

In these newer designs, both performance and reliability have been aided immeasurably by the introduction of computerized analysis techniques (Ref. 5, 6,& 7). These recognize non-linearities in the cable geometry and allow cable performance to be precisely balanced in both stress and torque. The results are precise, with accuracy and reliability that fit well with normal manufacturing tolerances.

New versions of these analysis programs can analyze the effects of bending on cable strength and even support a limited insight into lifetime and failure modes if the cable is subjected to loaded flexure. Two of these analysis programs are available in a condensed version, which can be run on a (IBM compatible) personal computer (Ref. 8).

2.5 An Insight Into Fiber Optic Telemetry

To reduce the UNOLS cable's RF attenuation, it was necessary to increase the cross section of its coaxial core. If it had not been for this constraint, the use of ultra-high-strength steels in the new armor package would have increased cable strength to more than 23,000 kg (50,000 1b).

But bandwidth in the armored coax is inescapably entwined with the core's cross section and, through it, with the cable's diameter and strength. The low conductor resistances and thick dielectrics demanded by telemetry constraints leave little room for optimization of the cable's power or strength functions.

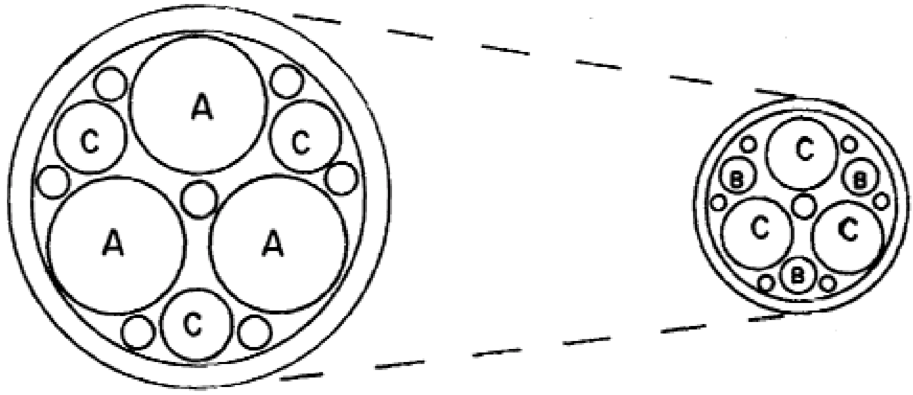
The tyranny of these interactions served as an important stimulus for UNOLS desire to convert from coaxial to fiber-optic telemetry in the next generation deepsea tether. That decision will be rewarded by several advantages offered by fiber optics.

Advantage #1. It is almost axiomatic that one optical fiber can provide an enormous increase in telemetry bandwidth. If single-mode fiber optic technology is used, data rates of at least 1,000 megabits/second can be achieved over an 8 to 10 km cable run. When system telemetry constraints require it, a single optical fiber can simultaneously send data in one direction and command signals (at a different wavelength) in the opposite direction. Even if we discount such high data rates by 20-times in order to convert from digital data to an equivalent analog, a single tiny (0.125-mm diameter) optical fiber can support uplink telemetry at data rates which correspond to at least 10 channels of high resolution television.

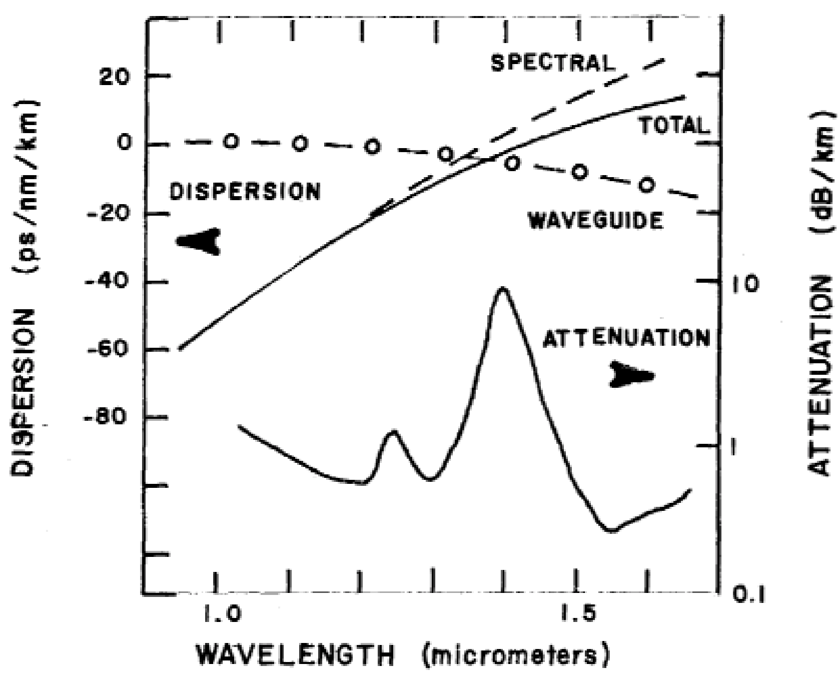
Advantage #2. This bandwidth can be almost independent of cable diameter. In many designs, the tiny optical fiber can be placed into inconsequential crannies of the cable cross section or can even be used in roles normally relegated to void fillers.

This characteristic is illustrated in Figure 4-3, which shows the effects of replacing three coaxial subcable units (A) with three optical fibers (B) in a hypothetical cable core. The three power conductors (C) are left unchanged. Before the switch, these power conductors were relatively inconspicuous, in fact, were used as void fillers to firm up and round out the cross section of the core. After the conversion, they became the dominant elements in the core--and the void filling role in the now much smaller cable core has been assumed by the optical fibers. Any further diameter reduction must be obtained through a rethinking of the cable's power and strength functions.

Advantage #3. In most electro-optical (E-O) cables, the power and telemetry functions will be nearly independent, so much so that they can be separately optimized. This is a nearrevolutionary departure from the coax design approach, where optimization of one function (generally telemetry because it is so critical) is likely to ride roughshod over subordinate system requirements.



EFFECT OF CONVERTING COAXES (A)
INTO OPTICAL FIBERS (B)
FIGURE 4-3



ATTENUATION AND DISPERSION SPECTRA
FOR FUZED SILICA OPTICAL FIBERS

FIGURE 4-4

For example, the conductors and dielectric insulation in an "acceptable" telemetry coax can usually transfer far more power than the system needs. In response to this freedom, the power subsystem is often allowed to grow, i.e., to consume the excess power and, ultimately, to demand it.

3.0 THE NATURE OF FIBER OPTIC COMMUNICATIONS

An optical fiber can be considered as a simple quartz wave-guide, which traps light rays and constrains them to propagate within a very small and extremely transparent rod. Fiber attenuation (Figure 4-4) can be less than 0.1 dB/km, and telemetry over 250-km continuous fiber lengths has been demonstrated in the laboratory. Bandwidths, i.e., bandwidth-length products, of 200,000 Mb-km/sec have been also achieved (Ref. 9). As the figure shows, bandwidth is normally greatest at 1.3 micrometers, where total dispersion passes through zero.

On the bandwidth/attenuation scale noted above, fiber optic communication through the cable length needed to support deepsea operations seems almost trivial. For example, a single 10-km optical fiber, operated at a standard T4 rate of 270 MB/sec, can easily support four or five digital TV channels. Yet such a system's bandwidth-length product is only 1 % of demonstrated capability.

3.1 Physical Properties of Optical Fibers

Physically, an optical fiber is a solid fuzed silica rod, with a normal diameter of 0.125 mm (0.005"). For protection and isolation from external stresses, this rod is coated with one or more layers of plastic. Typically, silicone rubber is used as the primary coating, but UV-cured acrylates are becoming popular in this role. The secondary coating is usually an extruded elastomer (e.g., HYTREL or NYLON).

In a short gauge, the fiber's ultimate tensile strength can be almost 50,000 kg/sq-cm (700,000 psi). This is equivalent to an ultimate strain of about 7%, far greater than the breaking strains of cabling steels or the lightweight KEVLAR or SPECTRA fibers.

But in a practical telemetry system, the optical fiber must be used in continuous lengths of several kilometers. And in such lengths, an optical fiber acts very much like a chain, a chain that, somewhere, must have a weakest link. It is the physical strength of this weakest link that will determine the practical tensile strength (or ultimate strain) of the optical fiber.

Typical optical fibers for deepsea tether cables can be specified to survive a "proof" stress of about 10,500 kg/sq-cm (150,000 psi). This is equivalent to a tensile strain of 1.5%. In this test, the fiber is passed between two sheaves so that it experiences a tensile stress with a duration of about 1 second. Any fiber section weaker than the proof stress will fail.

At a relatively small cost premium, the proof strain can be increased to about 2.0%. But even the lower of these strain limits is greater than the elastic limit of cable armoring steels--although it is still much less than the 2.4% failure strain of KEVLAR-49.

3.2 Light Propagation in Optical Fibers

Figure 5-a sketches the cross section of an optical fiber. Light propagation along the fiber will take place in a central core (the shaded zone). The sketch shows a ray of light moving, at incidence angle θo from a medium of refractive index no into a core of index n_1. If the external medium is air, then $n_0 = 1$. Refraction at the core/air interface reduces the external angle of incidence to a value $\theta 1$ The two angles are related by the Law of Sines.

$$n_0 \mathrm{Sin}\theta_0 = n_1 \mathrm{Sin}\theta_1 \qquad (1)$$

To trap the light ray inside the fiber, the optical core is surrounded by a transparent cladding[4], which has a lower index of refraction ($n_2 < n_1$). The largest value of θ_0 that can propagate within

[4]The optical fibers core and cladding are normally ultrapure silica and one of these components is doped to achieve the desired values of n_1 and n_2 The fiber may contain a second silica cladding, which provides physical protection. Finally, the fiber will be protected by one or more plastic jackets or "buffers" to a diameter of 0.25-1.0 mm.

the core belongs to a ray that suffers total internal reflection at the core/cladding interface. This occurs when θ_0 is equal to the "Brewsters angle," defined by the relationship:

$$\mathrm{Sin}\,\theta \;<\; (n_2 / n_1)\,\mathrm{Sin}\,(90^{\circ}) \;=\; (n_2 / n_1) \tag{2}$$

$$= \;\mathrm{Cos}\;\theta_1 \tag{3}$$

$$= \sqrt{1 - \mathrm{Sin}^2\,\theta_1} \tag{4}$$

Using Equations (2) and (4):

$$\mathrm{Sin}\,\theta_1 \;=\; \sqrt{1 - \left(n_2^2 / n_1^2\right)} \tag{5}$$

Combine Equations (1) and (5) with $n_0 = 1$ to solve for the external incidence angle.

$$\mathrm{Sin}\,\theta_0 = \; n_1 \sqrt{1 - \left(n_2^2 / n_1^2\right)} \tag{6}$$

$$= n_1 \sqrt{(n_1^2 - n_2^2 / n_1^2} \tag{7}$$

The fiber's core and cladding indices of refraction seldom differ by more than one percent, so the quantity $(n_1 + n_2)$ is usually set equal to $2n_1$. lso, it is customary to define an index difference term $A = (n_1 - n_2)/n_1$. Equation (7) becomes:

$$\mathrm{Sin}\,\theta_0 \;=\; n_1 \sqrt{\frac{(n_1 + n_1)}{n_1}\frac{(n_1 - n_2)}{n_1}} \tag{8}$$

$$= n_1 \sqrt{2\Delta} \tag{9}$$

$$= \;\mathrm{NA} \tag{10}$$

This last term is the "numerical aperture," the maximum external incidence angle at which light can be "launched" into an optical fiber, that is, trapped within the fiber and propagated by it. The square of the NA is proportional to the solid angle defined by this launch cone, so that fiber NA is equivalent to the "speed" of a camera lens.

In Figure 4-5a, fiber refractive index was discontinuous at the core/cladding interface. This type of behavior defines a "step index" fiber. Many other index profiles are possible. In almost all of them, the fiber core's index of refraction varies with radius according to the general constraint that;

$$n = n_1 [1 - \Delta(r/a)^a] \qquad (11)$$

where,

n Is the index of refraction on and along the fiber axis.

R Is the radial distance from that axis.

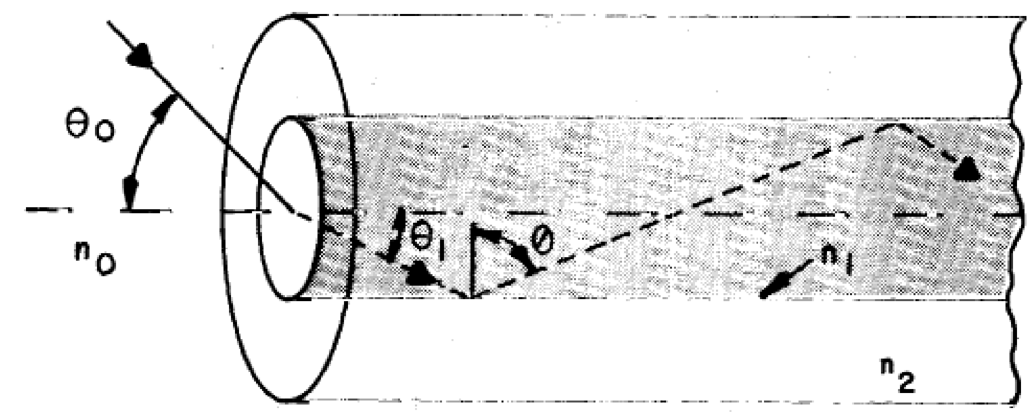
a Is the radius at which (by definition) the optical core ends and the cladding begins.

α Is a parameter which defines the shape of the transition between core and cladding.

For a step index fiber, $\alpha = \infty$, although few manufacturers can manufacture a fiber with a value greater than about 50 for this parameter. The optical behavior of a step index fiber is shown in Figure 4-5a.

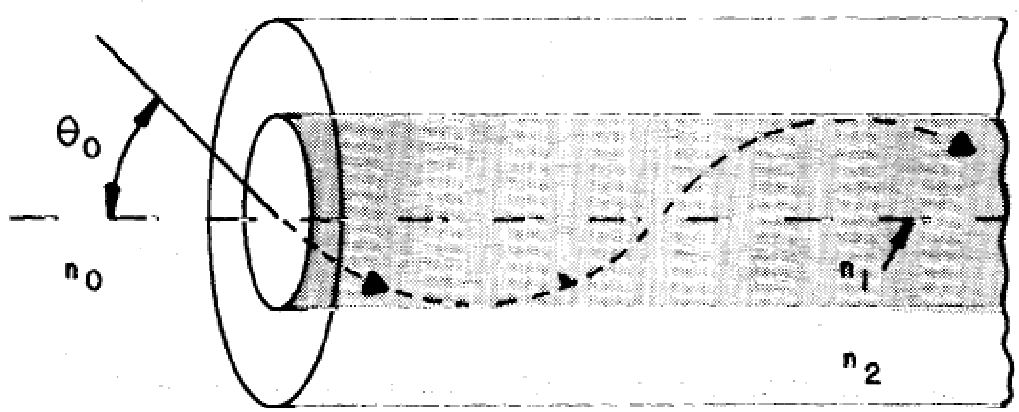
When $\alpha = 2$, the index of refraction of the fiber core changes from n_1 to n_2 as a parabolic function of fiber radius between $r = 0$ and $r = a$. Figure 4-5b shows how a light ray moves along the core of this "graded index" or "GI" fiber.

If all other parameters are identical, the graded index optical fiber can have a much greater bandwidth-length product than a fiber with a step index. The reason becomes apparent when the light paths in Figures 4-5a and 4-5b are compared.



STEP INDEX FIBER

FIGURE 4-(5-a)



GRADED INDEX FIBER

FIGURE 4-(5-b)

STEP The optical power in a digital data "bit" contains light rays at all angles less than Sin^{-1} (NA). Those rays that enter the fiber at large angles of incidence must travel a greater total distance and, therefore, move more slowly along the fiber axis than the lower-angle rays. Due to this speed difference, the bit's initial square shape begins to broaden and round off. Ultimately, the bit interferes with adjacent bits. Bit contrast is reduced, and a bandwidth-length limit has been reached.

GI Here, the core's refractive index decreases from n_1 on the fiber axis to n_2 at the core/cladding "interface." As it enters a region of smaller index, the tight ray must move faster. This helps high-angle light rays keep up with their paraxial counterparts. In fact, if α is precisely equal to 2, then all light rays of the same wavelength propagate along the core at the same speed. This increases the bandwidth-length product as much as 50-times, compared to the step index fiber.

3.3 Fiber Optic Propagation Modes

It can be shown by wave theory that light propagating in an optical fiber is quantified. That is, the fiber can accept only certain discrete angles of incidence (i.e., angular modes). Also, the number of angular modes "N" that can be accepted by a fiber is finite.

$$N = \frac{\alpha \, . \, \Delta}{2 + \alpha} \left[\frac{2\pi \, n_1 \, a}{\lambda} \right]^2 \tag{12}$$

where the parameters a, Δ, n_1, and α have already been defined, and λ is the wavelength. Note that, because its high α value, the step index optical fiber contains twice as many propagation modes as the graded index fiber.

3.4 Single Mode Fibers

If Δ or a are decreased, then the number of modes that can be propagated by an optical fiber must also decrease. In the limit, the fiber propagates only a single mode. This limit is reached (Ref. 12) when the fiber satisfies the inequality:

$$V = (2\pi n_1 a / \lambda)\sqrt{2\Delta} < 2.405 \qquad (13)$$

Table (2) shows typical fiber parameters for step index, graded index, and single mode fibers. Note that the single-mode fiber can have quite different values of index difference and core radius, so long as they satisfy Equation (13).

TABLE 2

TYPICAL OPTICAL FIBER PARAMETERS

Parameter	Step	GI	S-M #1	S-M #2
$\lambda(\mu m)$	1.3	1.3	1.3	1.3
$a(\mu m)$	25	25	2.0	4.0
n_1	1.5	1.5	1.5	1.5
α	40	2	40	40
A	1%	1%	1.38%	0.34%
N	328	164	1	1

Each fiber type has advantages and disadvantages. The large core diameters of the step index and graded index fibers allow them to accept greater launch powers from injection laser diodes (LED's)and light-emitting diodes (LED's). But the single-mode has a bandwidth-length product far greater than any obtainable with the large core fibers. These properties are shown in Table 3.

TABLE 3

POWER/BANDWIDTH COMPARISON OF
THREE OPTICAL FIBER TYPES

Fiber Type	Bandwidth-Length (Mbit-Km/second)	Launch Power (relative)
Step Index	20	80
Graded Index	1,000	40
Single-Mode	15,000	1
Single-Mode[5]	>200,000	1

3.5 Fiber Bending And Microbending

In designing for the use of optical fibers in cables, it is extremely important, in fact, essential, to recognize that the optical fiber cannot be treated like a copper conductor. For example, the fiber will be absolutely unforgiving of small-radius bends. The effect of such a bend will be to convert the highest order propagation modes into modes which leave the fiber.

This loss is immediate and can be quite large. It occurs even if the fiber completes only a very small arc of such a bend. After the initial loss, the curved fiber will continue to exhibit a steady-state radiation loss which is caused by:

(1) A repopulation of the highest-order modes because the bend is also perturbing the fibers more-tightly-bound modes.

(2) The immediate stripping of the new “highest order residents out of the fiber.

That fraction of total propagated energy which will be immediately lost is approximately equal to (ref. 11 & 13);

[5]When the signal is launched by an ultra-monochromatic (e.g., distributed-feedback) single-mode laser.

1

$$\text{Loss} = 1 - \frac{\alpha + 2}{\alpha}\left[\frac{1}{2\Delta}\right]\left[\frac{0.002\text{a}}{\text{R}} + \frac{0.003\lambda}{4\pi\, \text{n}_1 (1-\Delta)\text{R}}\right]^{2/3} \qquad (14)$$

Here, “R” is the radius of curvature of the fiber axis in millimeters. Figure 6 shows how typical optical fibers respond to uniform bending at various radii of curvature. The four fibers plotted in the Figure are identical to the ones described in Table 2.

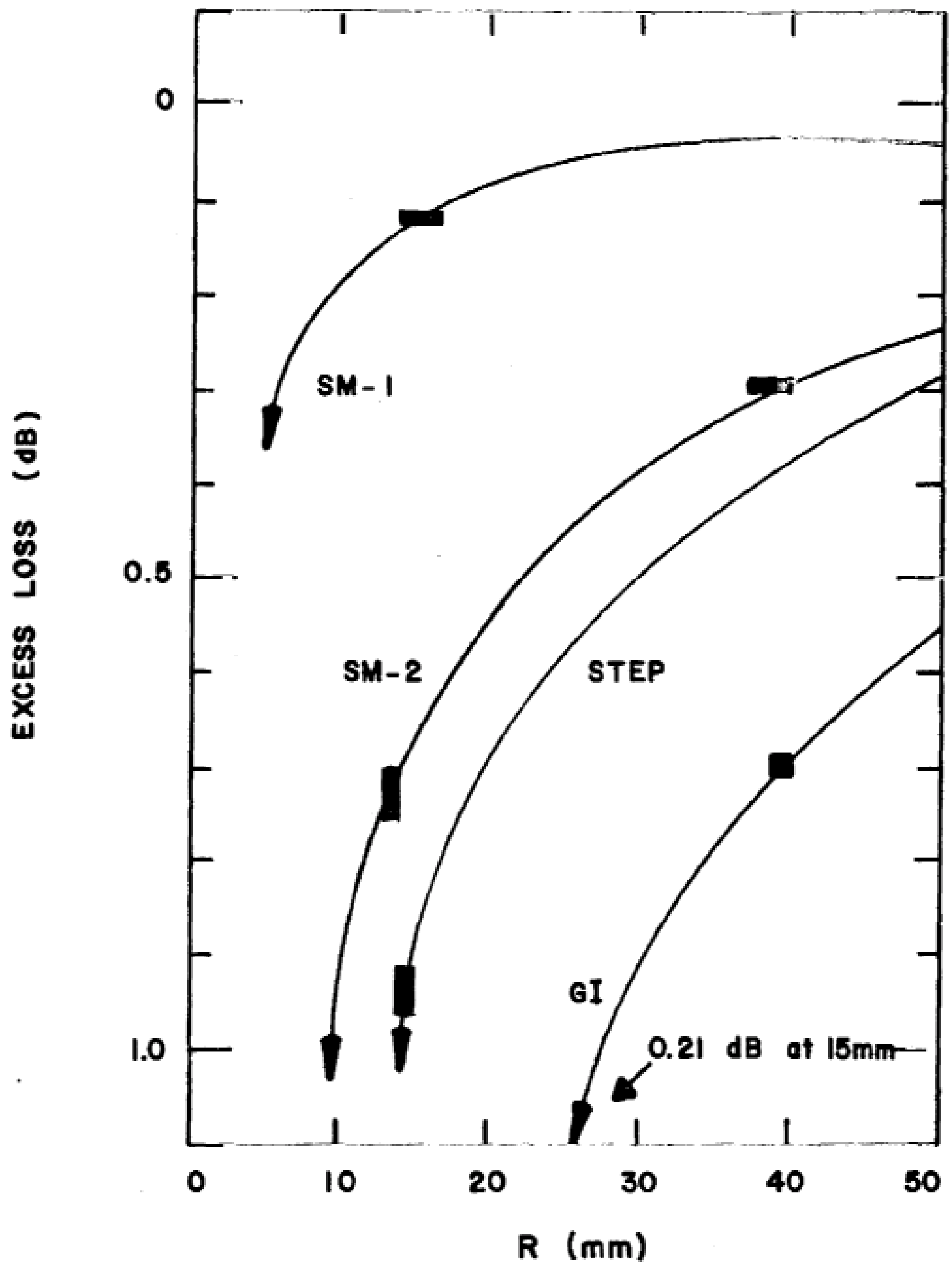
Because of its high bandwidth and large core, the Gl fiber has become quite popular in undersea tether and tow cables. As Figure 4-6 shows, it is also extremely sensitive to even moderate radii of curvature. Part of this sensitivity, a fault shared with the step index fiber, is a concentration of optical power at higher (i.e., more loosely bound) modes. Another reason (this time not shared) is the high order modes in the Gl Fiber have a closer angular spacing than those in the step index fiber. As a result more modes (and more optical energy) can be perturbed by a given fiber bend.

Notice the low bending responses of the two single-mode fibers. Note also how that response is sharply reduced if core radius a is decreased while increasing index difference A. The freedom to make such changes in a single-mode fiber is limited by the need to satisfy the constraint imposed by Equation (13).

Figure 4-6 also shows the effect of “microbends” a much more dangerous form of fiber curvature. In a typical microbend, the linearity of the fiber axis is perturbed by some imperfection in the local environment. As a result, the fiber axis suffers a highly localized bending, which combines a small amplitude with a small spatial period. Typical microbending conditions will have a spatial period comparable to the fiber’s buffered diameter.

This is, unfortunately, just the condition that can excite resonant scattering (i.e., leakage) from the fiber’s more loosely bound propagation modes. Energy in these modes will be converted into unguided radiation, modes.

Under these conditions, a single microbend can cause an immediate attenuation loss of approximately (Ref. 14):



EFFECTS OF BENDING AND MICROBENDS ON FIBER LOSS

FIGURE 4-6

$$\text{Loss (dB)} = \left[\frac{H^2}{\Delta^3}\right]\left[\frac{a^4}{b^6}\right]\left[\frac{K\,E'}{E}\right]^{3/2} \qquad (15)^6$$

where:

H Is the displacement induced by the microbend (μm).

B Is the radius of the fiber optical cladding (μm), with a value generally less than the physical fiber radius.

E'/E Is the ratio of the fiber buffer's tensile modulus to the Modulus of the fiber.

K Is a constant with a value near unity.

Figure 4-6 includes a semi-quantitative model to illustrate how a relatively gentle microbend can cause a very large local attenuation increase. The model assumes that an already-curved fiber is subjected to a perturbation which reduces its radius of curvature by 1 mm. At a bend radius of 40 mm, additional optical attenuation is almost negligible. lf the microbend occurs at 15 mm, then the resulting loss is quite painful. This is especially true if one considers that, in a poorly designed or badly built cable, such microbends can occur with a frequency of 100's/km.

The author remembers having to report the performance of one such cable, which had an excess attenuation of 1,400 dB/km, with the comment that he expected the sponsor to be "underwhelmed."

This bending/microbending scenario is not academic. It is, in fact, a reasonable description of what actually happens in a cable when a microbend perturbs a fiber that has been assembled in a relatively small-radius helix. Causes of microbending range from roughened surfaces of adjacent components, to imperfections in the protective fiber buffer, to localized Z-kinking caused by yielding of metal conductors. Several methods can be employed to reduce the impact of fiber bending and microbending.

[6] Murata (Ref. 15) describes an equation with a similar form, except that b^6 is replaced by b^4d^2, where d is the physical radius of the optical fiber. See also Kao (Ref. 16).

(1) Use single-mode optical fibers. In addition to microbend protection, these fibers will increase system bandwidth by one or two orders of magnitude (compared to a Gl fiber).

(2) Increase the radius of fiber curvature in the cable. As a rule of thumb, this radius should never be less than 50 mm.

(3) Float the fiber within a hydrostatic cable environment, so that it is not forced to make high-pressure contacts with adjacent surfaces.

(4) Make these adjacent surfaces as smooth as possible. For example, a conductor neighbor should contain small wires, and the specification for assembly of its jacket should require a pressure extrusion.

The combination of large wires and a tubed extrusion can allow the memory of these wires to carry through to the outer surface of the insulation. If the fiber is forced to conform to this surface, its axis may be repeatedly bent so that it acts like a grating. Extremely high excess losses can result.

(5) Increase the thickness of the fiber buffer to isolate the fiber from external anisotropic (i.e., bending) forces.

(6) Encase the optical fiber in a hard protective metal tube. This is an extreme version of the technique normally used to protect the fiber - an initial coating with a soft plastic buffer to form a quasi-hydrostatic environment, followed by a secondary annulus off a harder plastic.

For certain applications, this sixth approach is attractive, since multiple roles can be assigned to a protective metal tube. In addition to its primary anti-microbending assignment, the metal tube can seal the fiber inside a hydrostatic environment. It can serve as a dedicated conductor, and as a major strength element in the cable. For selected applications, the fiber, metal tube and (optional) jacket can serve as the entire cable.

Previous papers (Ref. 17 & 18) have reported success with techniques to form multi-kilometer lengths of metal tube as hermetic containers around one or more optical fibers. The application of this approach to undersea cables has also been discussed (Ref. 18, 19 & 20).

4.0 THE USE OF OPTICAL FIBERS IN OCEAN CABLES

The shotgun marriage of power and telemetry functions in an electromechanical coax cable allows a limited set of options in choosing the geometry of that cable. Conductor resistances and dielectric thicknesses can be varied, but the basic shape of the cable remains quite static. Figures 4-1 and 4-2 are probably a fair statement of the limits to the design freedom one has in the conventional E/M coax.

This is not so in the electro-optical (E-O) cable. Here, the ability to separate power and telemetry functions allows a much greater freedom to choose from among distinctly different cable geometies. Three of these configurations, representing designs with 1-, 2- and 3-electrical conductors, are sketched in Figure 4-7. The operating characteristics of each of these types will be discussed in some detail.

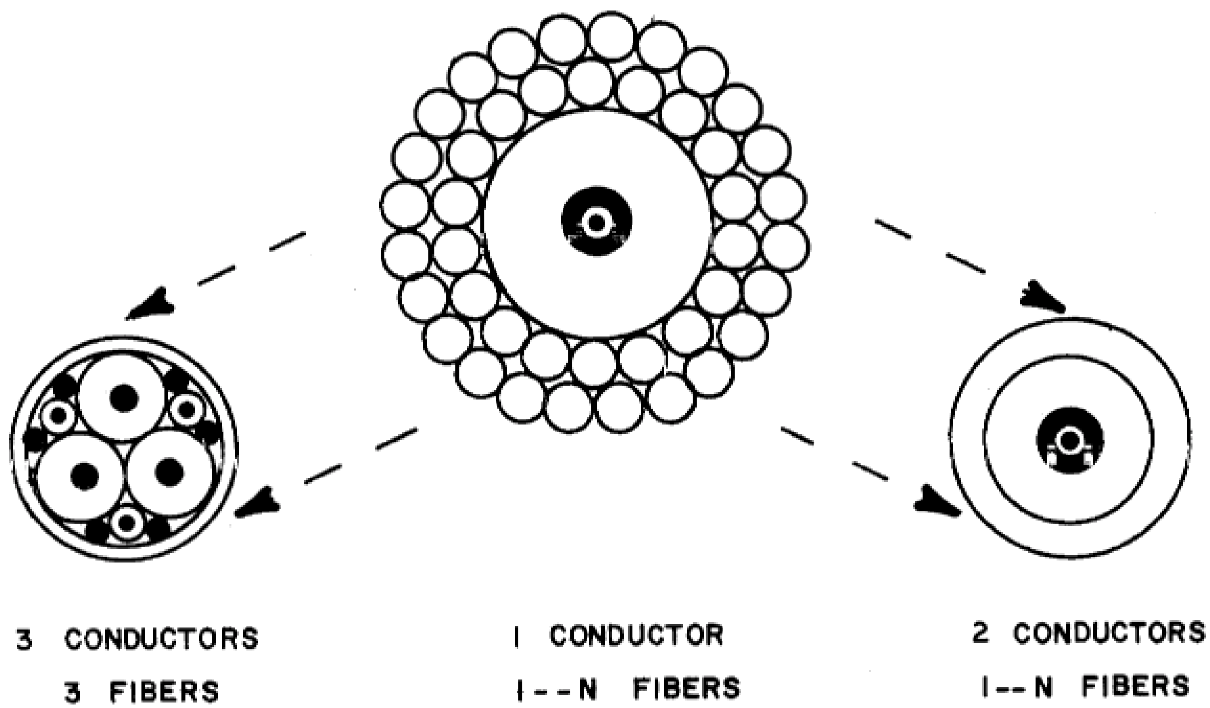
One generic type of E-O cable design is not treated here. That is the “hybrid” or “straddle” approach, a design which attempts to keep one foot in the new optical technology and the other foot solidly planted in electromechanical technology. A typical member of this cable design family will contain several types and sizes of power conductors. It will contain one or more coax subcables, and perhaps a twisted-pair data cable or two. And somewhere within this jungle, it will try to accommodate “N” (usually a large number to ensure “safety”) optical fibers.

This chapter intends to present a rational set of recommendations for the proper use of fiber optics in undersea cables. The hybrid cable design violates almost every one of these recommendations.

4.1 The N-Conductor E-O Cable

3-Conductors. While this cable design (Figure 4-7) is intended for use with 3-phase power transfer, it allows considerable operating freedom for the power system. The design can operate with a cable-return power circuit (unbalanced line). lt is even possible to parallel the conductors and operate the system with a seawater return.

The optical fibers in the 3-conductor cable design are overjacketed to a size that rounds out the core’s cylindrical cross section. They then ride gently in the channels defined by the several conductors. It will be shown



ALTERNATIVE CORE DESIGNS FOR ELECTRO-OPTICAL CABLES

FIGURE 4-7

that this design is relatively inefficient in trading off power transfer versus strength. At the same time, it gives excellent protection and strain relief to the electrical and optical conductors.

Some 3-conductor designs incorporate a strength member (e.g., KEVLAR-49) within the fiber overjacket. The author avoids this approach like the plague. The main value of the KEVLAR is to make the cable company's task easier, so that more back tension can be applied to the fiber unit as it is served into the conductor structure. Once the cable has been built, the higher tensile modulus of the strengthened fiber unit - driven by the fact that cable strain is determined by cable load acting on the cable armor, causes the fiber unit to experience relatively high tension in the E-O core. This forces the unit to press against local conductors while attempting to migrate to the cable center. Under such high tensile and bearing stresses, the KEVLAR yarns can also act as local microbending centers which increase fiber optical attenuation.[7]

2-Conductors. The right-hand sketch in Figure 4-7 shows a coax-like structure, with two electrical conductors and one or more optical fibers. The associated ability to provide in-cable circuit return for a power system can be a critical advantage. For example, this design might be used for an E-O cable that must be dipped into the sea from a helicopter.

But the cable-return capability is achieved at the cost of lower power-versus-strength efficiency, since the requirement for two layers of insulation reduces the cross section that can be dedicated to loadbearing structure. The coaxial E-O cable design is not recommended if diameter, strength, and weight are critical system elements, unless the use of a cable circuit return is a categorical imperative.

1-Conductor. In both the 1- and 2-conductor designs, the optical fiber(s) are contained within and protected by a closed-and welded metal tube. In the single-conductor design approach to be discussed here, this tube! also serves as the cable's only electrical conductor, so that the power circuit must be completed by a seawater return.

[7]lf strength is required in the buffered fiber, it should be obtained with materials that offer stiffness in axial compression; e.g., fine steel wires or glass filaments.

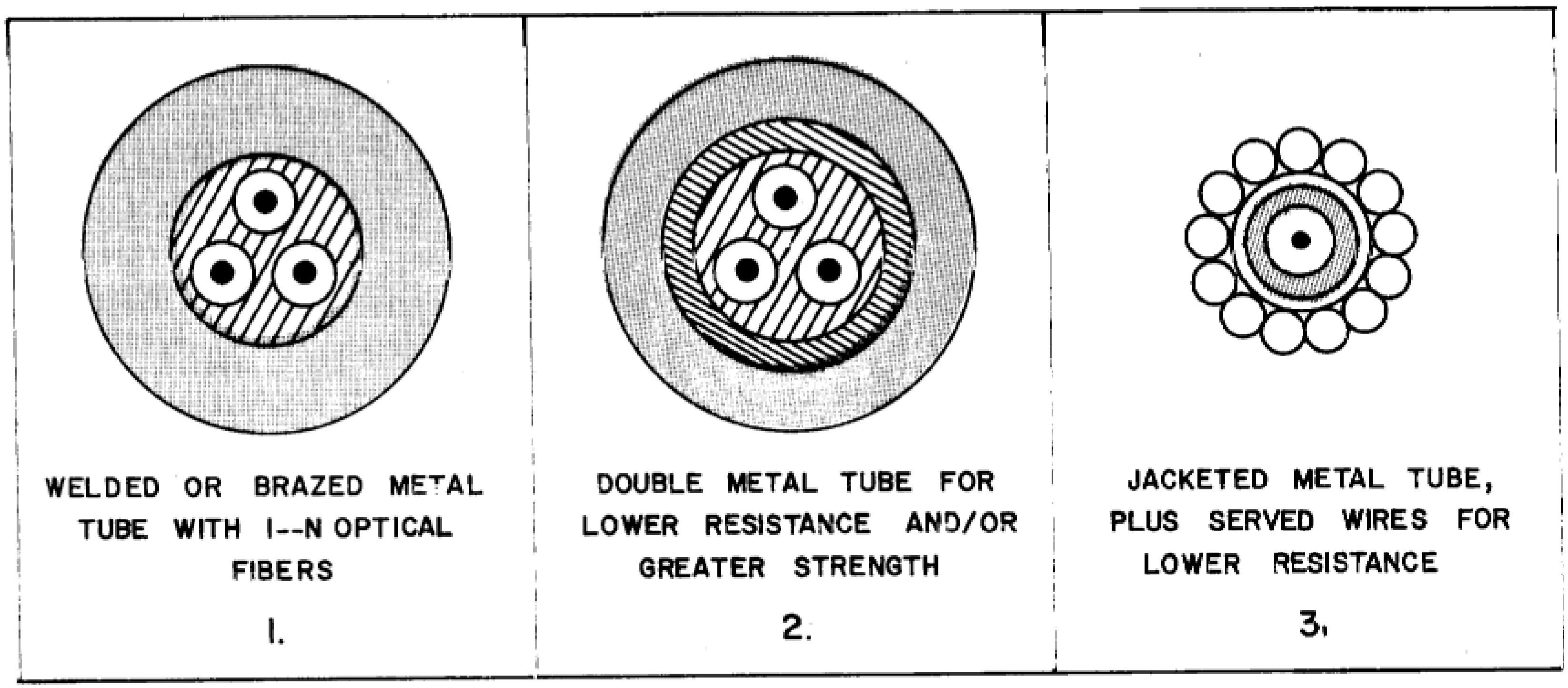
Figure 4-8 shows the centers of three design options for this conductor. The shaded region in the centers of designs (1) and (2) is a void-filling gel that provides the optical fibers with a hydrostatic environment. In design (3), the metal tube is given a thin elastomer jacket. This layer serves as a bedding layer for the (work hardened) copper conductors added to reduce and tune cable resistance.

In the limit, any of these designs might contain only one optical fiber, and that fiber might operate in a full-duplex mode. For example, telemetered data can move in one direction at one wavelength while command signals flow in the other direction at a second wavelength. (This capability has been demonstrated in the laboratory through fiber lengths greater than 100 km.)

The attenuation of the optical fiber in the formed-and-welded tube will generally be less than the level claimed by the fiber manufacturer. (This means only that those fiber stresses which can increase fiber attenuation are less inside the tube than they are on a “zero-stress” measurement spool.)

The single-conductor cable design operates with a seawater circuit return, an approach which is sometimes challenged as a critical weakness of the single-conductor concept. The author has even heard this from scientists who then defend grounding the shield conductor to the armor of their deepsea coax cables with the argument: “But the circuit return is through the armor.” At best, less than 1 % of system current will flow through the armor. The remainder will pass through the sea, through the painted hull of the ship and, by the most devious means, back to the system ground. How much better it would be to operate the system with a dedicated, efficient (and known) seawater return!

A more valid question might be: “What are the physical limits to a seawater return circuit?” Europe offers the most compelling operational experience with this type of circuit. For example, the Skagerrak Sea power cable delivered 250 megawatts between Norway and Denmark over a 130-km single-conductor circuit with a seawater return. Until a second cable was installed in the late 1970’s, to take advantage of the fact that the cable’s resistance was slightly lower than that of the ocean, this power system operated uneventfully with a seawater circuit return of 1,000 amperes (Ref. 21).



CENTER-CONDUCTOR OPTIONS
FOR METAL-TUBED OPTICAL FIBERS

FIGURE 4-8

In the 1- and 2-conductor designs in Figure 4-7, the metal tube lies parallel to the cable axis. This means that helical geometry cannot be used to provide the E-O conductor with strain relief. The conductor must furnish whatever tensile compliance is needed to avoid permanent strain at very high cable tension.

Figure 4-9[8] demonstrates typical stress/strain behavior for one alloy commonly used to build ultra-miniature versions of the metal-tubed optical cable. The tube is almost completely elastic for strains as great as 0.85%. For an alloy with 50% conductivity (e.g., OLIN 195), the elastic strain limit is 0.7% (Ref. 17). Both of these limiting-strain levels are higher than the elastic limits of conventional "compliant' stranded copper conductors.

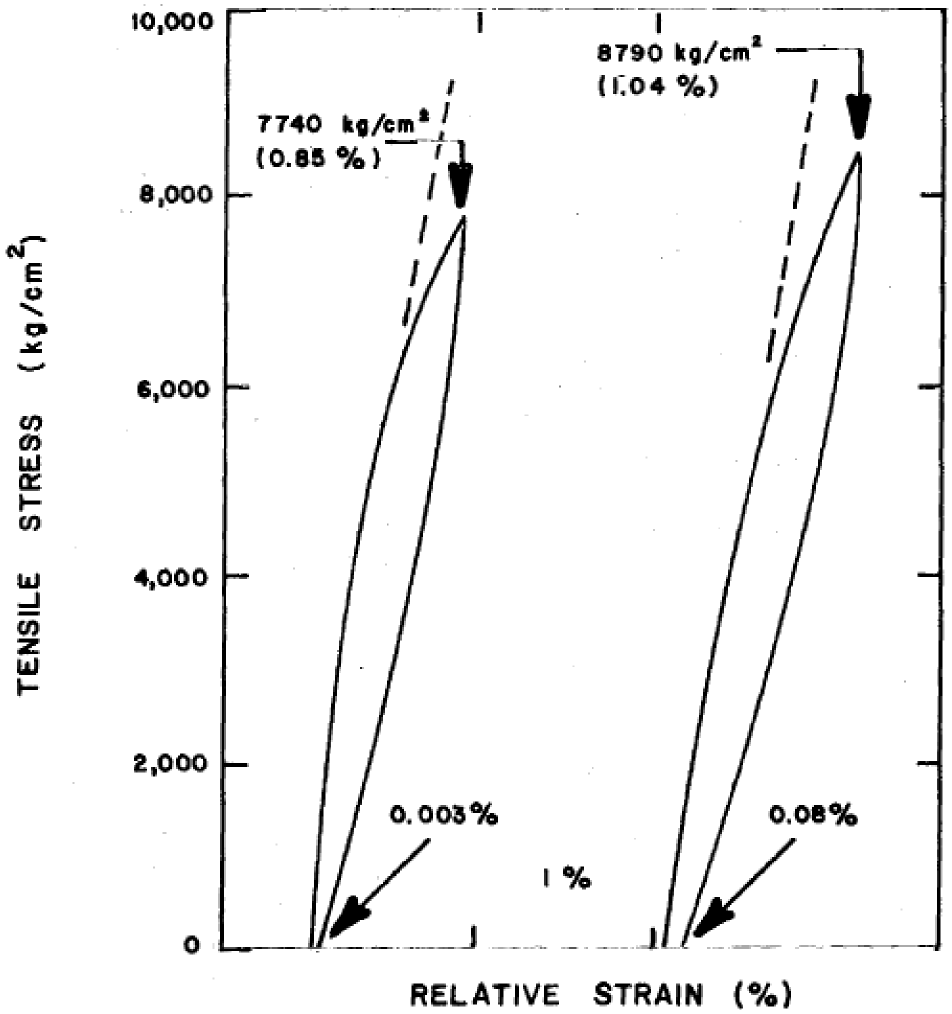
4.2 Designs For Electro-Optical Undersea Cable

No single E-O cable design can be "optimum" for all system constraints and for all operating conditions. Design choices must still depend on the often painful resolution of conflicting requirements for cable strength, power transfer, and diameter. This is not unusual---in fact, it is the norm in the design of tether cables. What makes the E-O cable unique is the fact that system bandwidth no longer plays a critical role in that design conflict. As an example, let's follow the path of a typical E-O tether design. (See also Ref. 22 & 23.)

(1) The design analysis begins with a cable diameter constraint, to ensure that the cable can operate from existing handling systems. (This analysis evaluates cable performance at 7.94-, 9.53-, and 17.3-mm diameters (5-, 6-, and 11/16-in.).

(2) A second constraint sets a minimum value for the power that must be delivered to system instrumentation. This limit is best expressed as a power-length product (P-L), i.e., as a power P delivered to a payload through cable length L.

[8]Two U.S. companies are able to fabricate metal-tubed optical fibers in (lengths of 10+ km. They are OLIN Corporation (New Haven, CT) and KT Armor Tech San Diego, CA).



OLIN TYPE 638 BRONZE ALLOY
SPRING TEMPER
10% CONDUCTIVITY

FIGURE 4-9

(3) The P-L specification may be accompanied by a specified value of supply voltage or cable current. This constraint is often imposed for no obvious reason and can have a very serious impact on cable strength/diameter tradeoffs. This interaction will be discussed later.

(4) Other parameters that can affect the choice of geometry for the E-O core include the number of fibers, requirements for strain relief, or even special constraints placed on the electrical power system. For example:

   (a) The need for optical and electronic simplicity at the ends of the cable might veto the use of full duplex telemetry, forcing the cable to contain two or more optical fibers.

   (b) The need for additional tensile strain relief might force the selection of a 3-conductor cable core because of its helical structure.

   (c) A propulsion motor in the instrumentation package might demand 3-phase power.

(5) With these parameters and constraints in hand, a (somewhat) arbitrary initial value can be chosen for system supply voltage. This choice allows the cable electrical resistance to be calculated, leading to design values for conductor dimensions, for insulation thickness and, finally, for the cross section and overall diameter of the E-O core. As a special constraint to minimize electrical pinholes, the dielectric might be specified to have a thickness greater than some minimum value.

(6) The cable- and core-diameters define the annulus available for loadbearing armor so that, assuming reasonable values were selected for the armor coverage and helix angles, we can calculate cable strength, weight, payload capability, and "free length."

(7) If the conflict between strength and diameter is critical, then the supply voltage should be "optimized." That is, supply voltage will be adjusted until a value is found for which cable strength (or strength/weight) is a maximum.

(8) Finally, the approximate (i.e., linear) cable design can be refined. Such tuning involves the use of nonlinear analysis techniques to adjust the conductor and armor helix angles for simultaneous stress and torque balance (Ref. 5, 6, 7 and 8).

4.3 Power-Diameter-Strength Optimization of E-O Cables

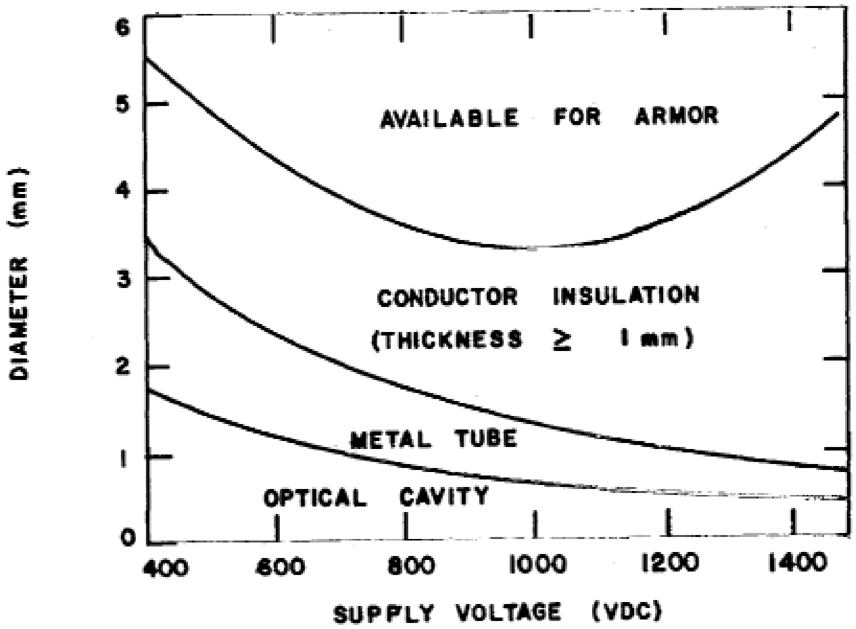
The sensitivity of component diameters to supply voltage is shown in Figure 4-10 for the 1-conductor cable design. Note that the metal tube's thickness is limited to 27% of tube diameter. This constraint recognizes that the tube wall may buckle in the forming die if the relative thickness exceeds this limit.

Also, a 1.0-mm minimum thickness constraint has been imposed on the cable insulation to avoid pinhole effects. This constraint affects only the low-voltage side of the curve, where a thinner insulation will satisfy the constraint that dielectric stress be equal to 1,970 V/mm (50 V/mil).

The diameter of the E-O core must also grow as the system's P-L product is increased. This effect is shown in Figure 4-11 for all three cable geometries. Each curve represents a family of "optimum" design solutions, i.e., the smallest jacket or dielectric diameter possible for a given P-L product. A 50% voltage drop in the cable is assumed for all analyses except the one plotted in Figure 4-15.

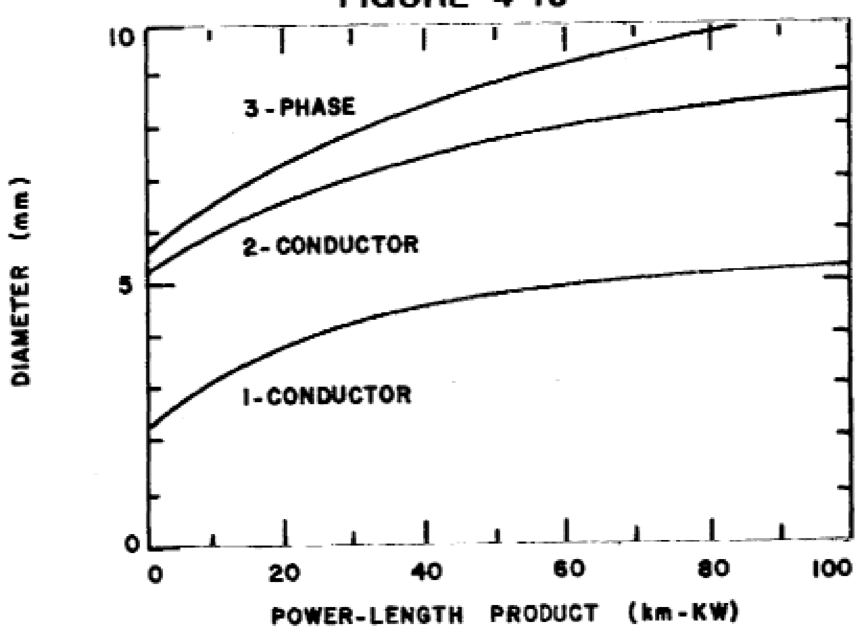
Note that the diameters of the 2- and 3-conductor cores are much more sensitive to the value of P-L. This is because the multi-conductor cores must use more insulation to achieve the required voltage protection and physical isolation. In addition, the 3-phase cable core is inefficiently filled.

Figure 4-12 shows the effects of P-L growth on cable strength for all three conductor designs and for a cable diameter of 7.94 mm (5/16"). Figures 4-13 and 4-14 present similar data for cable diameters of 9.53 mm (3/8") and 17.3 mm (0.680").



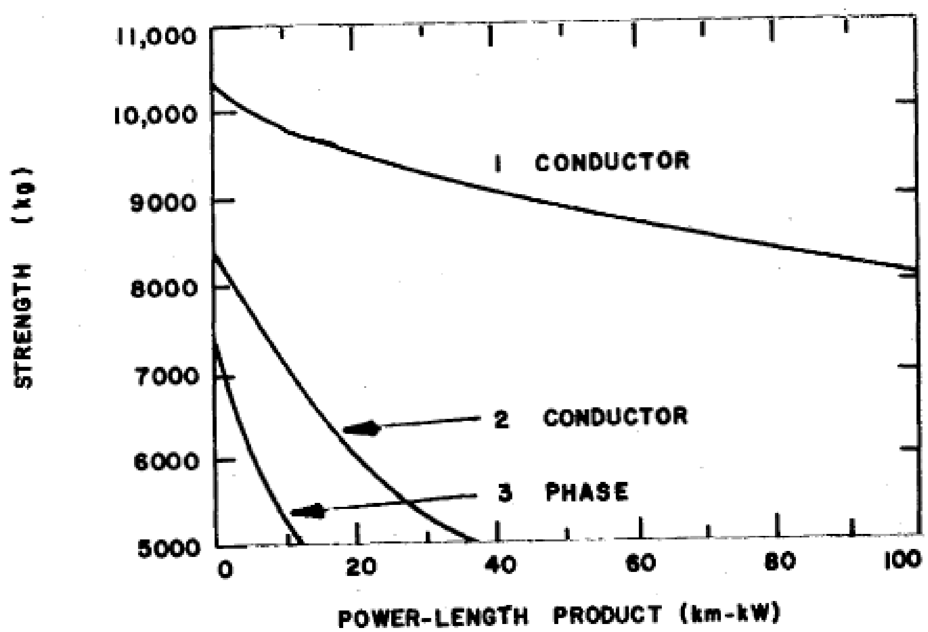
COMPONENT DIAMETERS vs SUPPLY VOLTAGE
FOR: P-1000 Watts, L- 8 km, T-Tube Thickness/ Diameter Ratio - 0.27

FIGURE 4-10



GROWTH OF E-O CORE DIAMETER WITH POWER-LENGTH PRODUCT

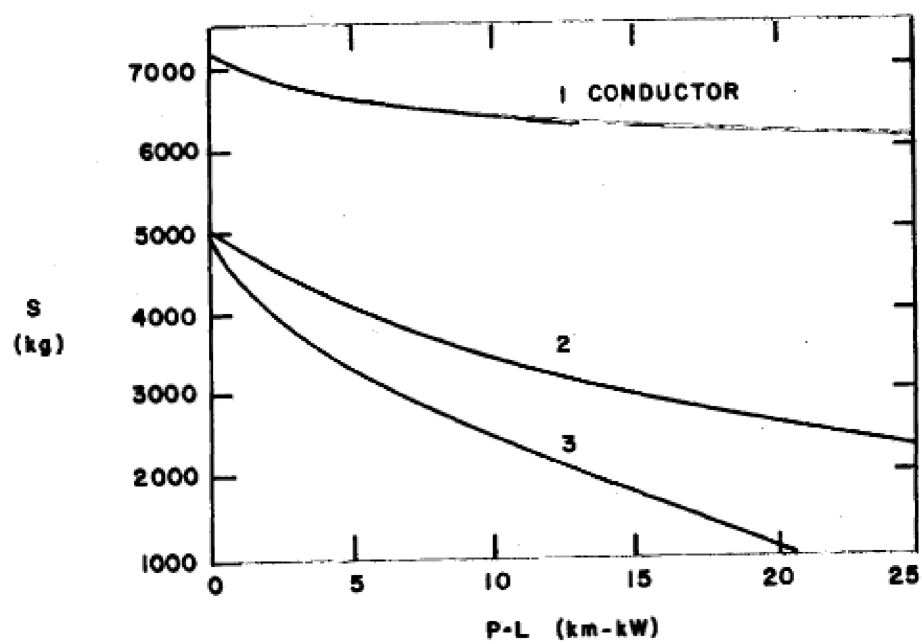
FIGURE 4-11



POWER-LENGTH PRODUCT vs STRENGTH

CABLE O.D. = 7.94 mm

FIGURE 4-12

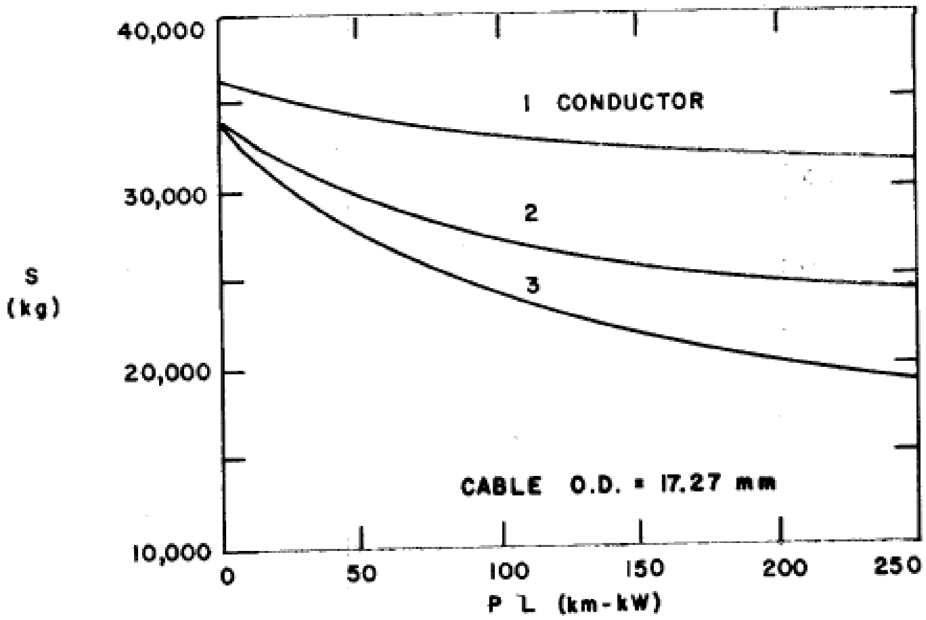


VARIATION OF STRENGTH

WITH POWER-LENGTH PRODUCT

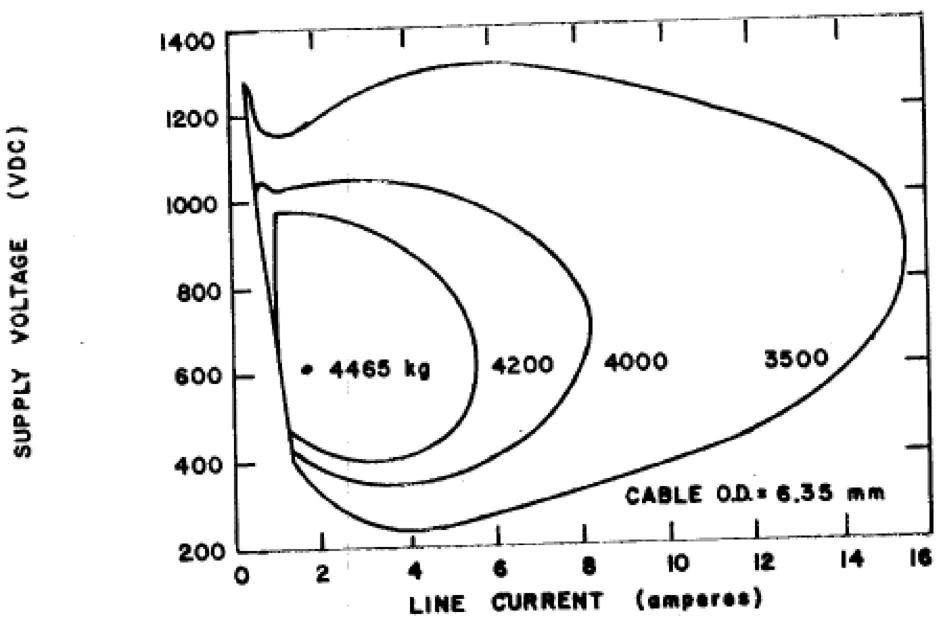
CABLE O.D. = 9.53 mm

FIGURE 4-13



POWER-LENGTH PRODUCT vs STRENGTH

FIGURE 4-14



CONTOURS OF CONSTANT CABLE STRENGTH
IN VOLTAGE/CURRENT PLANE FOR:

DELIVERED POWER - 500 Watts	TUBE CONDUCTIVITY - 50%
CABLE LENGTH - 7 km	TUBE T/O.D. RATIO - 0.27

FIGURE 4-15

In the smaller cables, it is clear that the multi-conductor tie-signs have severely limited abilities to supply both power and strength. A somewhat smaller penalty is exacted at the largest diameter (17.3 mm), since that cable has sufficient cross section to absorb moderate P-L values without too great a theft of space from the loadbearing armor

4 4 System Tradeoffs And "Off-Optimum" Cable Design

Figure 4-15 gives another example of the tradeoffs that become possible once telemetry bandwidth has been removed from the design conflict. Here, contours of constant cable strength are plotted in voltage-current space. The maximum achievable strength for this 7.94-mm-diameter, single-conductor cable is also shown. The cable design satisfies all constraints listed in Figure (4-10).

The performance contours in Figure 4-15 could as easily have been drawn in terms of cable payload capability, or cable weight, or cable strength/weight ratio. Whatever parameter is most critical to the system can be plotted.

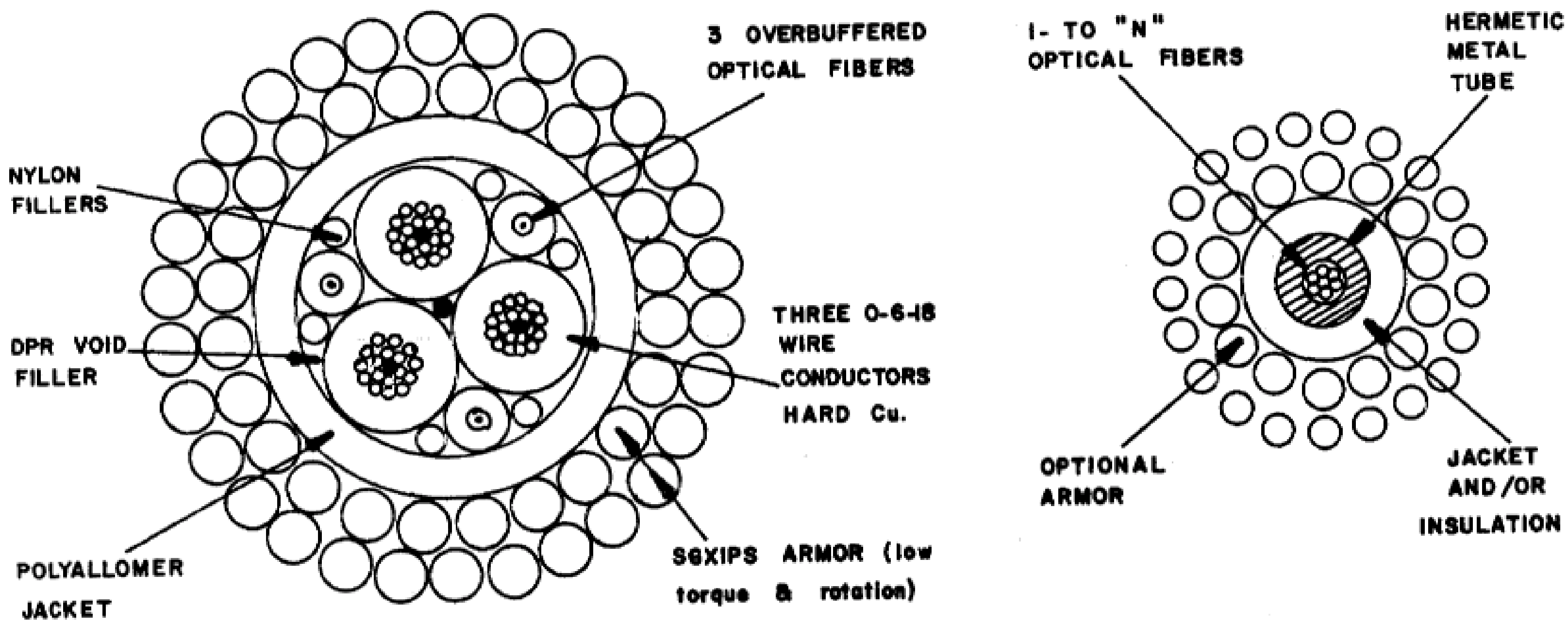
The insights offered by contour plots of this type can be of considerable value in helping to resolve design conflicts among such system parameters as cable diameter and strength, operating voltage, and line current. At the very minimum, this insight can force system conflicts and compromises to take place on more rational grounds, i.e., with more light and less heat.

4.5 The Next Generation

With some degree of reluctance, the author has recommended the 3-phase, 3-fiber cable geometry for the "next generation" deepsea E-O tether cable. A typical cross section for this cable; is shown in Figure 4-16. The primary reasons for selecting this geometry were that:

(1) Its larger diameter (17.3 mm) gives some measure of relief from the normal inefficiency of this design in trading off strength versus power-length product.

(2) The use of three optical fibers simplifies the telemetry system. One fiber can be dedicated to uplink data, one to downlink commands, and one kept in reserve for expansion or for use as a spare.



OCEANOGRAPHIC CABLE
(17.3-mm Diameter)

HYDROGRAPHIC CABLE
(7.94-mm Diameter)

OPTIONS FOR DEEPSEA ELECTRO-OPTICAL CABLES
FIGURE 4-16

(3) The helical core geometry gives additional strain relief for both fibers and electrical conductors.

(4) The cable geometry is a relatively close match with the well tested and successful design of an earlier E-O tow cable which was also built for operation under very high stress conditions (Ref. 18).

At the same time, it is hard to ignore the many advantages of the single-conductor E-O cable. It may well be that the “next” generation should see two basic types of deepsea tethers.

OCEANOGRAPHIC	This heavy duty cable will be similar to the unit sketched in Figure 4-16, and will operate from the standard oceanographic winch. It can support massive instrumentation packages, supplying them with at least 10 KW through a 10-km length.
HYDROGRAPHIC	This smaller cable (probably 7.94 mm) will operate from a standard hydrographic winch. It can deploy moderate-sized packages to abyssal depths and can supply them with a few kilowatts of power. It will probably contain one optical fiber-operated full duplex, but can incorporate at least three fibers within a slightly larger conductor tube.

Table 4 shows how these two designs might perform in deep ocean operations. The two examples given for each design differ only in the amount of power they export (through a common 8,000-meter length).

5.0 HANDLING SYSTEMS FQR E-O CABLES

The handling characteristics of the armored deepsea E-O tether should be much like those of conventional armored coaxes or 3-phase cables. In either case, the cable’s internal geometry will closely resemble that of its electromechanical counterpart. The armor packages are (deliberately) as similar as possible. Differences in cable handling should occur primarily as responses to new operational capabilities offered by the fiber, or to new requirements that the fiber brings with it.

5.1 Optical Slip Rings

At the shipboard end of an E-O cable, uplink telemetry will be received as very low level optical signals. As in conventional E/M systems, these signals must be continuously fed from the rotating cable storage drum through some kind of slip ring to a stationary receiving point. The problem is more difficult (but not impossible) if the cable contains more than one optical fiber. Choices for handling the data at the winch include:

(1) Pass the optical signals directly through the slip ring. But the signals will be quite weak, as much as 30 dB (1,000-times) weaker than the power levels launched into the fiber at the other end of the cable. Any oil, dirt, or particulate matter within the slip ring can have disastrous effects on data transmission, especially, on bit error rate which, with fiber optics can readily be 10^{9}.

TABLE 4

FOUR OPTIONS FOR DEEPSEA ELECTRO-OPTICAL TETHER CABLE

Performance Parameter	17.3-mm Diameter 3-Conductor Cable		7.9-mm Diameter 1-Conductor Cable	
	10,000 W	20,000 W	1000 W	5000 W
Optimum Voltage	864 V	1,026 V	1,148 V	1,397 V
Strength (kg)	23,470	20,240	6,600	5,790
Weight (kg/km)				
In Air	1,135	1,032	276	260
In Water	927	818	233	216
Free Length (km)	25.3	24.8	28.3	26.8
Payload (kg)*	3,830	3,190	1,240	1,020

For operations to 6000 meters, with strength/weight safety factor of 2.5

(2) Receive (and transmit) the optical signals at a repeater which is in a splash-proof housing within the core of the storage reel. The power level required by each optical channel is no more than 1-3 watts, so that the repeater can easily be powered for missions of days with conventional batteries. Signal transfer options from this point include:

(a) Multiplex the data at a high bit rate, then transmit a composite signal through the slip joint.

(b) Accomplish the same purpose with spectral multiplexing. that is, transmit each signal at a discrete wavelength. This approach involves combining a spectral multiplexer (two signals sent in one direction) with a spectral duplexer (two signals sent in opposite directions). Optical signals can be sent through both ends of the cable reel axle.

(c) As a third alternative, use angular multiplexing. Let each half of the rotary joint be a blown up version of the step index fiber in Figure 5-a[9]. Light injected in one end of the joint at a given incidence angle will exit the other end at the same angle. Different data channels pass through the joint at different angles.

(d) As a near term fallback, optical digital data can be converted to an electrical data stream, and telemetry can be fed through a conventional coaxial slip joint. If this is done, it is vital that the telemetry system remain digital, i.e., that data precision not be lost through premature conversion to an analog format.

5.2 Fiber Flexure Performance

Although its absolute strength is low, the optical fiber is not the most tender element in the cable. lt is much like a strand in a spider web, weak, but oh so tough.

[9]We might use a section from the preform from which a fiber is drawn, since it is a scaled up (about 200-times) model of the optical fiber.

As long as the optical fiber is neither stressed to a level more than (about) 50% of its proof strain, nor assaulted by other components, its performance under loaded flexure should be at least as good as that of any other cable element.

5.3 First Pressure/Temperature Response

Optical fiber attenuation is almost completely insensitive to (at least) the -50°C to +75°C temperature range. Any attenuation response to deepsea pressures, to topside temperatures or to handling system bearing pressure will actually be due to other causes. Likely culprits include imperfections in the fiber's plastic buffer, and microbending induced in the fiber by adjacent cable components. These should have been eliminated (or at least observed) during cable manufacture and initial testing.

5.4 E-O Cable Operating Stresses and Strains

It is important to remember that the optical fiber is not at all like its copper wire ancestor. For example, the copper wire can be, in fact, normally is, fully yielded and deformed during the production and cabling of an electrical conductor.

A braided copper conductor is a good example. In a good braid, each wire repeatedly makes small-radius, fully-yielded crossovers of its companion wires. This does not increase wire resistance, since electrons can easily blunder around such sharp turns. Just one of these crossovers would either break an optical fiber or would induce sufficient attenuation to consume most or all of the system's optical power margin.

As a general rule, the optical fiber should not be subjected to high tensile loadings for long periods of time.

(1) During a deployment of hours to days, fiber tensile strain should be limited to no more than 30% of the proof strain to which it was originally subjected. Typical values might be 1.5% proof strain and 0.45% deployment strain. Any strains due to fiber curvature must be counted. Some would argue that the percent-of-proof-strain limit level should be 20%.

(2) During storage periods of weeks to years, the fiber's strain level should be no greater than (about) 0.10%. (See below for a related precaution.)

5.5 Storage Conditions for E-O Cables

In conventional cables, high operating loads can permanently strain the copper wires. This is not necessarily bad as long as the cable is never allowed to become completely slack. If all cable tension is released, drawback of the elastic armor may cause the copper conductors to Z-kink[10].

SOLUTION #1 Using monofilament core wires and work-hardened wires, make the copper conductors as elastic and compliant as possible.

SOLUTION #2 If the cable has been loaded to such a point that Z-kinking is likely, then it must never again be allowed to see low or zero tension.

The first solution also applies for the E-O cable, since it is no more than a common sense design technique for all highly-stressed cables that contain materials subject to low-strain yielding.

The second solution is easy to apply to the E/M cable, since long term tensile loads are unlikely to damage a cable component. In fact, it is common practice to operate such cables directly off a winch, so that the inner wraps see full deployment stresses and strains throughout their time in storage.

This method of operation would probably be fatal to an E-O cable. Without re-reeling, the fiber spends its operating life suffering the highest stresses it has ever sustained during deployment. The most likely result of such storage is that the optical fibers will creep to failure, and failure will occur on the storage reel, since that is where the cable spends most of its life.

5.6 Winches For E-O Cables

In deep ocean operations, the E-O cable should never be operated directly off a storage winch. (This statement is made to give extra emphasis to the discussion just finished.)

[10] The Japanese name for this phenomenon is “zaku-tsu”, which sounds even more frightening.

The conventional traction winch should work well with an optical or electro-optical cable. It also solves the storage tension problem, since selection of on-reel storage tension can be independent of operating conditions.

For smaller E-O cables (like the hydrographic unit described in Section 4.5) serious consideration should be given to use of the linear or “caterpillar winch. The traction that this winch can deliver is proportional to the diameter of the cable. But the pulling force required is proportional to the cable’s cross section, i.e., to the square of its diameter. Therefore, the winch becomes more efficient as the cable becomes smaller.

Another advantage of such a winch is that the cable need not be bent during the transition from high to low tension. In very small cables, this leads to a handling system concept in which the linear winch is rotated so that the pulling axis is parallel to the axis of the loaded cable. The approach can essentially eliminate cable degradation due to scuffing and loaded flexure.

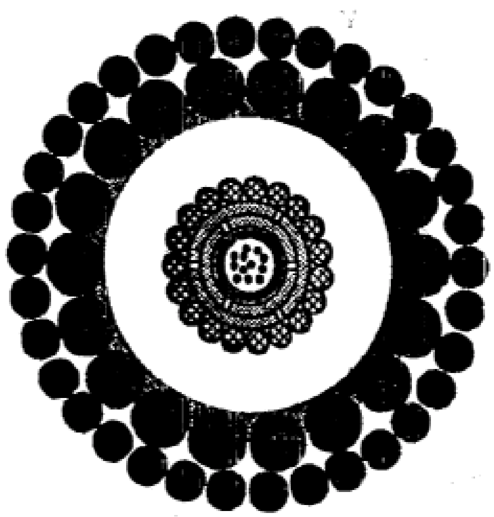
A GLIMPSE OF A NEW E-0 GENERATION

Just before the author retired from the University of Hawaii, he had the opportunity to design a new deepsea E-0 cable which incorporated most of the principles presented earlier in this chapter. He also assisted in its deployment into 4700 meters of water about 28 km west of Keahole Point on the Big Island of Hawaii.

The cable was built in four 9-km sections. (These lengths were limited by availability of continuous optical fibers.) Four sections were joined with inline E-0 connectors to serve as a power and communications link from shore to a deepsea optical array. The connectors were designed to pass through a portable Figure 8 traction winch with 6-foot-diameter drums. The cable cross section is sketched in Figure 4-17.

The program was known as DUMAND (Deep Underwater Muon and Neutrino Detector). Its purpose (Ref. 24 and 25) was to map our portion of the Milky Way galaxy by the "light" of ultra-high energy (12 TEV) nuetrinos which rech the earth after passing with little absorption through thick interstellar dust and gas clouds. The array was designed to look through the earth and then out into space—i.e., it used the mass of the earth as a noise filter to block out cosmic ray radiation.

We believed that DUMAND might also serve to give early warning for supernovae, since neutrinos from such events would leave the exploding stellar mass well ahead of optical photons, and should reach the earthy sensibly in advance of such photons.



1. Stainless steel tube with 12 single-mode fibers in a void filling jell (OD = 2.34 mm).
2. Nylon jacket to 2.64-mm O.D.
3. Conductor: 2 layers of copper tapes overserved with Cu wires to OD of 6.1 mm.
4. Insulation to OD of 10.4 mm and bedding layer to OD of 11.0 mm.
5. Contrahelix of 2.01 and 1.32-mm GIPS steel wires to 17.2-mm OD.

FIGURE 4-17
FINAL DESIGN OF DUMAND E-0 SHORE CABLE

The DUMAND shore cable contained several new and/or unique features:

1. For the first time we were able to incorporate more than 4 optical fibers into a single hermetically welded, metal tube. Each tube/fiber combination was jell filled; with better than 98% efficiency

   Laser welding was continuous over each 9-km tube length; with no interruptions or flaws in hermeticity. The optical fibers were provided by Bell Telephone, and were proof strained to 2%. Insertion of the 12 fibers and jell, and forming/welding of the stainless steel tube, were performed by Laser Armor Tech in San Diego.

2. Midway through assembly of the E-0 core, the need for more array power forced a design change in the conductor. This was accomplished by Vector Cable, by simply adding a helix of copper wires around the (already assembled) layers of copper tapes.

3. The armor was designed to have low torque and rotation under load. As a result, its strength was essentially the same (about 16,000 kg) with both ends fixed as it was when the lower cable end was free to rotate.

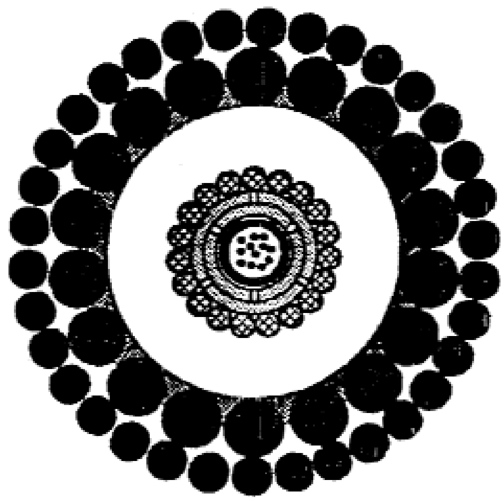
During deployment, the sea end of the cable was attached to one riser element (of 9) in the DUMAND array. This element was then lowered to the sea floor at a depth of approximately 4700 meters. When contact and position were assured, the ship began to move slowly along a 28-km surveyed path toward Keahole Point while paying out cable. Water depth, cable tension, ship speed and payout speed were monitored and computer-controlled so that the cable draped along the seafloor with essentially zero tension.

Near shore, the ship dropped anchor and a measured length of cable was cut from the figure-8 storage bin, buoyed along its length, and floated to a point just offshore. (This required almost every gallon milk jug on the Island of Hawaii.) There, it was pulled through a cased shaft which had been horizontally drilled under the sea cliffs to provide a safe and storm-proof shore-crossing route for the cable.

The cable has been in place for several years, and continues to show optical and electrical integrity. Eleven of the 12 optical fibers survived deployment. All survivors now show less attenuation than was measured before deployment.

DUMAND Proposed Redesign
Full Armor

A serve of 0.032" copper wire over the formed copper strip has been added to the design to reduce the conductor resistance by 50%. The insulation wall thickness has been increased and armor package is now low torque.



1 - Stainless Steel Tube			
With 12 fiber optics	OD = 0.092"	(2.34 mm	
Nylon Buffer, wall = 0.006"	OD = 0.104"	(2.64 mm)	
Conductor			
3 - 0.017 x 0.113" formed Cu strips	OD = 0.138"	(3.51 mm)	
3 - 0.017 x 0.145" Formed Cu strips	OD = 0.172"	(4.37 mm)	
18/0.034" served Cu wire	OD = 0.240"	(6.10 mm)	
All void spaces filled.			
Insulation			
EPC, wall = 0.085"	OD = 0.410"	(10.4 mm)	
Compliant Armor Bedding	ØD = 0.434"	(11.0 mm)	
Estimated compressed OD = 0.415"			
Low Torque Armor			
1st Layer: 18/0.079" GIPS	OD = 0.573"	(14.6 mm)	
2nd Layer: 34/0.052" GIPS	OD = 0.677"	(17.2 mm)	
Rust inhibitor applied to armor.			

NOMINAL PARAMETERS

Electrical:

DC resistance		=	0.32	Ω/kFt	(1.1 Ω/km
Insulation resistance		=	2000	MΩ/kFt	(610 MΩ/km
Voltage Rating		=	2000	V. rms	

Optical:

Attenuation	1300 nm	=	0.4	dB/km	
	1550 nm	=	0.3	dB/km	

Mechanical:

Calculated weight	air	=	774	Lbs/kFt	(1150 kg/km
	sea water	=	632	Lbs/kFt	(940 kg/km
Calculated break strength	ends fixed	=	36,000	Lbf	(160 kN
	ends free	=	35,000	Lbf	(155 kN
Recommended minimum bend diameter	storage	=	14	inch	(34 cm
	25% of BS	=	27	inch	(69 cm

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CHAPTER 5

ROPE AND CABLE TERMINATIONS

Author: C. Robert Shaw   Reviewer: Tom Coughlin

1.0 INTRODUCTION

Selection of the proper type of wire or cable termination used in oceanographic applications is a key factor in the safe and effective utilization of the winch and wire system in the deep sea. The purpose of this section will describe the seven basic types of wire rope terminations, as well as those used with electromechanical cable. In particular, the characteristics, advantages, limitations, and assembly procedures for each type will be fully illustrated.

The selection of a wire or cable termination for a particular application should be considered carefully, bearing in mind that there are advantages and disadvantages to each termination type discussed in this section. To select the correct type of terminal for a particular application, the following factors should be carefully evaluated:

- Type and size of cable involved
- Corrosion potential
- Loading and cable fatigue
- Efficiency required
- Assembly requirements and cost

Primarily, seven basic types of wire rope terminations will be discussed as follows:

a. Wire rope clips - A U-bolt and saddle combination or a “fist-grip” nut and bolt arrangement used to fasten a loop of wire rope that is formed around a thimble.

b. <u>Open-wedge terminations</u> -- Also called “wedge sockets”; the cable is looped around a wedge, which is inserted into a socket or “basket” and held secure by tension on the line.

c. <u>Poured-socket termination</u> - - Also known as Spelter sockets; molten zinc or an epoxy compound is poured into the socket to bond the wire rope inside the fitting.

d. <u>Compressed sleeves</u> -- Also known as Nicopress terminations; sleeves are crimped or compressed around the cable, usually by use of hand tools.

e. Swaged terminations - - Attached by cold forming under high pressure so that the metal of the fitting flows around and between strands and wires of the rope.

f. Mechanical terminations - - Also known as Electroline fittings; these devices utilize wedge or plugs of various sizes and configurations to hold the cable inside a threaded lock sleeve.

g. Helical terminations - - Also known as Preformed Dyna-grip terminations. This device utilizes helical gripping wires, which wrap around the cable and are finished in a thimble or epoxy filled fitting.

2.0 SELECTION CRITERION

As mentioned in the introduction, a series of five criteria should be considered in the selection of a wire or cable termination. The following discussion of these criteria will provide insight into the problems, which can be encountered in the selection process.

2.1 Type and Size .of Wire or Cable

The terminations selected must be compatible with the type of cable being used and must result in the maximum effective holding strength. For example, swaged terminations, compressed sleeves and wire rope clips are not efficient terminals for hemp-core wire rope, armored electrical cables or synthetic cables. Application of such terminations requires squeezing them onto the cable, and “soft-core” cable material will give way under the pressure, thereby weakening the effectiveness of the termination. Mechanical, poured-socket and open-wedge terminations can be use effectively with these types of cables since they achieve their efficiency from bonding or compressing only the steel of the wire or cable.

Cable size is a major consideration because of the standard capacities of termination devices that are generally available. Compressed sleeves can be used on cables up to 5/8-inch diameter. Wire rope clips, open wedges and mechanical fittings are standard for cables up to 1-1/2 inches in diameter. Swaged terminals can be obtained as large as 2-1/2 inches and poured sockets up to 4 inches.

2.2 Corrosion Potential

The corrosion problems experienced in an oceanographic application, when wires and terminations are subjected to alternate immersion and drying cycles, is well known. Given this as an existing condition, it is important to consider the standard materials in which the various termination devices are available. In the main, poured sockets, helical terminations, wire rope clips, and open-wedge sockets are stocked only in stell, although zinc plating on these terminations is available. Compressed sleeves (Nicopress), swaged terminations, and mechanical fittings (Electroline), are available in a wide variety of materials ranging from steel to stainless steel. Compressed sleeves and the mechanical fittings are also produces, in certain sizes, in both bronze and aluminum for special applications. The variety of materials available in termination constructoin, make the matching of a specific fitting to a particular application a fairly simple process.

2.3 Loading and Cable Fatigue

All seven basic types of terminations are suitable for static and moderately cyclic loads such as those imposed by cranes, hoists, guy wires, tie downs, slings, etc. Only the mechanical and helical terminations, however, are designed to accommodate the shock and vibration loads imposed by winches, buoys and towed bodies in the marine environment.

The mechanical fittings have a “transition zone” in the nose where the cable enters the termination. In this “semi- loose” transition zone, the tension, compression and bending stresses in the rope strands are dissipated. Because of the ways in which other types of terminations are affixed to the cable, they have a hard transition from the cable to the terminal which can contribute to shorter cable fatigue life The helical type termination provides a long cable life by dissipating the shock and vibration loads in the spring action of the helical gripping wires.

2.4 Terminal Efficiency

The more efficient the terminal, the smaller and lower cost the cable may be. This also can affect the cost of winches and other handling equipment.

The swaged and helical terminals are rated for 100% of the cable's rated breaking strength. Poured sockets, compressed sleeves and mechanical fittings are rated at 95% to 100%, while wire rope clips and open-wedge terminations are rated at 75% to 85%. Wire rope clips tend to lose their holding strength with use and must be retightened from time-to-time. At the other end of the scale are the mechanical and open-wedge terminals in which the wedging action actually increases efficiency with loadings (Table I).

2.5 Assembly Requirements and Cost

Wire rope clips are both the least expensive and easiest termination that can be applied in the field. They require only attention to clip spacing, placement, and tightening torque to perform efficiently. The open-wedge termination is only slightly more expensive, but is just as simple to install, requiring only hand tools for the application. Although the simplest, they are also the least efficient of the six terminations discussed. The compressed sleeve (Nicopress) fitting represents another low cost, but highly efficient means of terminating a wire rope. Special tooling, which is available from the manufacturers, is available to ensure proper compression of the sleeve and the installer needs only to match the required number of compression to the sleeve size selected for use. Used in the proper situation, the compressed sleeve can be rapidly and efficiently reapplied in the field with no special training.

The swaged and poured socket (Spelter) terminations represent a moderately priced fitting with a high efficiency rating. Swaged terminations require large hydraulic presses for proper installation and do not readily lend themselves to reapplication in the field under most circumstances. The poured socket represents a highly efficient fitting, which can be reapplied in the field using either molten zinc or an epoxy resin to achieve wire bonding. This approach, however, requires careful attention to detail and the use of an aid to clean the wire ends prior to fitting installation.

The mechanical termination represents a more expensive fitting type discussed in this section due to the number of components involved in each assembly. Although it appears to be a complicated termination, it can be easily installed without special equipment and with only the training received from the manufacturers' literature. Attention to assembly detail and adequate proof loading are all that is required to produce a highly efficient termination using this fitting.

The helical terminal is the most expensive. Assembly can be accomplished easily in the field with no special training or equipment. However there is a 24-hour cure for the epoxy filling.

Inspection is another important assembly consideration. The swaged and compressed sleeve terminals can be inspected for effective assembly by measuring the final diameters. Wire rope clips can be inspected with a torque wrench. The poured socket cannot be inspected to determine if the assembly is proper. The mechanical terminal has an inspection hole built in to facilitate visual checking. The helical termination can be inspected to assure that no wires are crossing themselves and that the body is filled with epoxy.

Perhaps the single most important thing to remember in the selection of a termination for either wire rope or cable is that the fitting should be chosen at the same time as the cable is specified. A second major consideration is physical size of the termination relative to the instrument or device it will be attached to. This is especially important when the fitting is required to pass through an instrument as in the case of a piston-coring device or over a sheave. In these cases physical size and configuration of a fining will influence the type selected. Since the wire or cable termination is vital to the safe and efficient use of the wire or cable, it should be viewed as an integral part of the system and gives careful consideration then purchased.

In order to select the proper terminations for an application, the factors discussed here should be carefully considered as well as those presented in Table I.

3.0 INSTALLATION PROCEDURES

In order to ensure the highest possible reliability in applying the particular termination selected for use, the following procedures have been detailed, along with information, which affects long-term performance. For any type of termination there are “tricks of the trade” which assure the integrity of the fitting once applied, and which should not be deviated from if full reliability is to be achieved. The following procedures, if carefully followed, will assure the reliability required in an oceanographic application.

TABLE 1

CHARACTERISTICS OF SEVEN TYPES OF TERMINALS FOR WIRE ROPE AND CABLE

TYPE OF TERMINAL	TYPE OF ROPE/CABLE	ROPE SIZE	STANDARD MATERIALS	DESIGN LOADING	EFFICIENCY % OF RBS	CABLE LIFE	ASSEMBLY	COST
Wire Rope Clips	IWRC Strand	3/16 - 1 1/2	Steel	Static Cyclic (needs re-adjustment)	75% - 85%	Short	No training needed. No special tools. Fast field assy and disassembly, Reuseable. Torque inspection.	Low
Open Wedge	IWRC Hemp Center Strand Synthetic	3/8 - 1 5/8	Steel	Static Cyclic (effectiveness increases with load)	75% - 85%	Short	No training needed. No special tools. Fast field assy and disassembly, Reuseable. Visual Inspection.	Medium
Poured Socket	IWRC Hemp Center Strand	3/16 - 4	Steel	Static Cyclic	95% - 100%	Long	Training required. Special tools and equipment req'd. Difficult field Assy. Not reuseable. No visual inspection of acceptable assy.	Medium
Compressed Sleeve (Nicopress)	IWRC Strand	1/16 - 5/8	Steel Stainless Aluminum Bronze	Static Cyclic	95% - 100%	Short	No training req'd. Special tools req'd. Field assy on only smaller sizes. Not reuseable. Dimensional Inspection.	Low
Swaged Socket	IWRC Strand	1/8 - 2 1/2	Steel Stainless	Static Cyclic	100%	Medium	Training needed. Tools and Equipment req'd. No field assy. Not reuseable. Dimensional Inspection	Medium
Mechanical (Electroline Fiege)	IWRC Strand Hemp Center Double-armored Synthetic	1/16 - 1 1/2	Steel Stainless Aluminum Bronze	Static Cyclic Shock Vibration (effectiveness increases w/load)	95% - 100%	Long	No training needed. No special tools. Fast field assy, reuseable. Visual inspection.	High
Mechanical (Preformed Dyna-Grip)	IWRC Hemp Center Strand Double-armored	1/8 - 3/4	Steel	Static Cyclic Shock Vibration	100%	Long	No training needed. No special tools. 24 hour cure for the epoxy. Not reuseable No method of inspection.	High

3.1 Wire Rope Clips

Wire rope clips (Figure 5-1) are sized and marked on the body of the clip with the wire diameter that they are to be used. It is important that the clip be matched to the diameter wire that is in use, as mismatches will result in a drastic reduction in termination holding power and efficiency. Placement of the wire rope clip is of prime importance to achieve maximum holding power. It should be noted that most available wire rope clip data sheets specify only a minimum number of clips needed for ordinary loads. Where heavy loading is anticipated, it is strongly recommended that two additional clips be added to each installation.

The recommended procedure for applying wire rope clips to achieve the maximum termination holding power is as follows:

-- Turn back the amount of wire required based on Table 2. This distance is measured from the thimble to the bitter end of the wire.

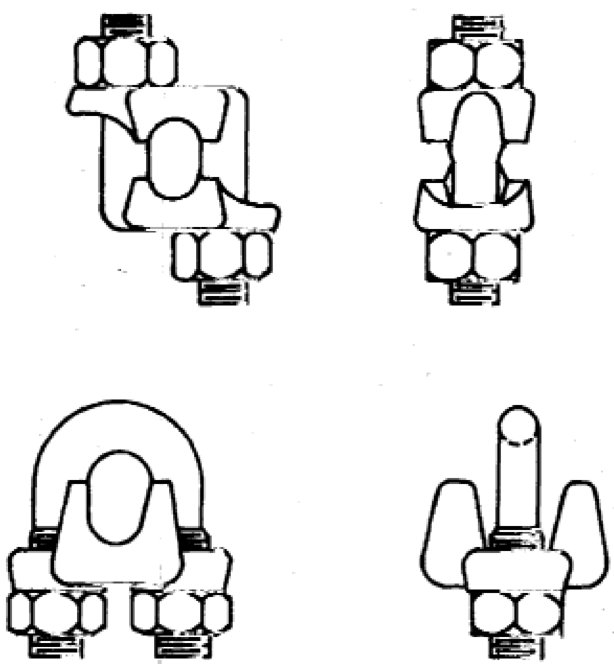


FIGURE 5-1

WIRE ROPE CLIPS

-- The U-Bolt portion of the clip must be placed over the bitter end of the wire while the saddle of the clip is placed on the standing part of the wire. Any reversal of this procedure or a staggering of the clips will result in reduced efficiency of the termination.

TABLE 2

Wire Rope Clip Assembly Data **

ROPE DIA	NO. CLIPS REQ NORM	H.D.	CENTER TO CENTER CLIP SPACING	LENGTH OF ROPE TURNBACK-INCH NORM	H.D.	TIGHTENING TORQUE FT. LBS.
1/8'	2	2	1 3/8"	3 1/4"	6"	5
3/16'	2	4	1 7/16"	3 3/4"	6 5/8"	9
1/4'	2	4	1 7/8"	4 3/4"	8 1/2"	18
5/16'	2	4	1 7/8"	5 1/4"	9	30
3/8"	2	4	2 5/8"	6 1/2"	11 3/4"	42
7/16'	2	4	2 7/16"	7"	11 7/8"	70
1/2'	3	5	3 15/32"	11 1/2"	14 31/3"	75
9/16'	3	5	3 1/2"	12"	19"	100
5/8'	3	5	3 1/2"	12"	19"	100
3/4'	4	6	4 1/4"	18"	27"	150
7/8'	4	6	4 1/4"	19"	27 1/2"	240
1'	5	7	4 3/4"	26"	35 1/2"	250
1 1/8'	6	8	5 3/8"	34"	37 5/8"	310
1 1/4'	6	8	5 3/4"	37"	48 1/2"	460
1 3/8'	7	9	5 15/16"	44"	55 7/8"	520
1 1/2'	7	9	6 1/2"	48"	60 7/8"	590
1 5/8'	7	9	6 29/32"	51"	64 27/3	730
1 3/4'	7	9	7"	53"	66 1/2"	980
2'	8	10	8 15/32"	71"	87 31/32"	1340
2 1/4"	8	10	8 5/8"	73"	90 13/32"	1570
2 1/2'	9	11	8 27/32"	84"	101 11/16"	1790
2 3/4'	10	12	9 9/16"	100"	118 15/16"	2200
3'	10	12	10"	106"	125 1/4"	3200

** Table based on Crosby Group Data

-- The first clip should be installed within one saddle width of the end of the turned back wire and the nuts evenly tightened. The second clip should be installed as near the thimble or loop as is possible with nuts firmly installed, but not tightened.

-- Space additional required clips evenly between the two clips already on the wire. Light tension should be applied between the terminal loop and the standing part of the cable before tightening all clips to their recommended torque. This process will eliminate stack occurring in the bitter end of the cable and produce a more uniform application.

-- An initial load should be applied to the termination and all nuts retightened to their recommended torque prior to use of the termination. Once applied in accordance with the above procedures, the resulting termination will have an efficiency rating of 75-85% of the breaking strength of new wire.

-- When a wire rope clip type of termination is used, it is advisable to periodically tighten the nuts to their recommended torque since wire vibration can result in a loosening of the U-bolt nuts.

3.2 Wire Rope Thimbles

The use of a thimble is highly recommended with this type of termination and some discussion of thimble styles is necessary at this point. Essentially, thimbles fall into three broad categories: I) Standard Wire Rope Thimbles, 2) Extra Heavy Thimbles, and 3) Solid Thimbles. Specific data relating to each style discussed will be found in the Useful Information section at the end of this handbook.

a. Standard Wire Rope Thimbles -- This class of thimble is designed primarily for use in light duty situations where loading is minimal. Their use in heavy-duty situations will result in a complete deformation of the thimble and the placing of excessive stress on the wire at the head of the loop. Under this situation, a failure of the wire can be expected, which will occur below the rated strength of both the termination and the wire.

b. Extra Heavy Wire Rope Thimble -- This style of thimble has been designed for heavy-duty service where high loading conditions are expected to Occur. They are far more resistant to deformation due to loading and work to maintain an even wire loading condition in the loop of the termination. As in all thimbles under load, the size of the pin used to attach the toad to the cable is critical. Its diameter should be closely matched to the internal diameter of the thimble in order to reduce point loading.

c. Solid Thimble - This thimble is designed as the ultimate in crush-proof thimbles due to its solid steel construction. Primarily a unit for very heavy load conditions, it has a single disadvantage in that the hole-sizes available are more limited than those found in the extra-heavy thimble. Where heavy loading situations are anticipated on a regular basis, it is recommended that a solid thimble be considered.

3.3 Open-Wedge Termination

Although this type of termination is typically found on crane cables, it has occasionally been used for limited trawl wire operations. Because of its diverse usage in the field, it is felt prudent to provide proper assembly instructions for this type of fitting.

The open wedge termination (Figure 5-2), although a simple style of fitting to install, should be approached with a certain amount of caution during installation of the wire. Improper placement of the wire can result in excessive wire stress at the termination resulting in a reduction of wire loading potential.

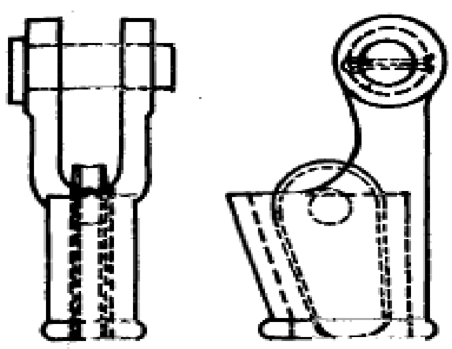


FIGURE 5-2

OPEN WEDGE TERMINATION

a. Installation Procedure - The simplicity and rapid installation potential of the open-wedge can be further enhanced by attention to a few details calculated to produce the maximum efficiency from this style of termination. The proper installation procedure is as follows:

-- An inspection of both the socket and wedge should be made to identify any rough or burred surfaces on the wire path, which could damage the wire under load. If irregularities are discovered, they should be removed, if possible, or the socket or wedge replaced.

-- The bitter end of the wire should be clean cut and served in order to prevent unlaying. It is important that the bitter end be clean cut rather than fused due to cutting with a torch in order to allow the individual wire strands to adjust around the sharp bend of the wedge. If the wire end is fused on installation, the movement of individual wire strands will be translated to the standing part of the wire causing irregularities in shape and unequal loading.

-- To install the wire in the socket, the socket must be in an upright position (ears downward). The wire is then brought into the socket to form a large, easily handled loop. Care should be taken to ensure that the standing part of the wire is in line with the sockets ear (Figure 5-2).

-- The bitter end of the wire should extend above the socket for a distance equal to nine (9) times the diameter of the wire used. At this point the wedge is placed in the socket and a wire rope clip placed around the bitter end of the wire by clamping it around a short length of wire, which has been attached to the bitter end to provide the mass required for the wire clips seating. The U-bolt of the wire clip should bear on the bitter end and the saddle on the added short piece of wire.

-- By securing the fitting to a convenient pad eye or bit, a load should be placed on the standing part of the wire. This load is steadily applied until the wedge and wire are pulled into position with enough strain to hold them in place when the load is released. During the seating of the wedge, sudden surge or shock loading should be avoided.

3.4 Poured or Spelter Sockets

The spelter socket (Figure 5-3) represents a highly reliable termination with 100% efficiency, when properly applied. The key factor in achieving the 100% efficiency of this termination is careful cleaning of the wire ends with a solvent solution and the positioning of the socket on the wire prior to the pouring of either the zinc or resin used to hold it in place. The cleaning of the wire ends allows for the maximum bonding action of the filler chosen while exact positioning of the socket on the wire ensures an even loading of the wire strands in the field.

A certain amount of caution should be used when dealing with this style of termination. The rigidity, which is caused by the bonding action of the zinc or resin on the end of the wire, causes a sudden dampening of wire vibration at the point where the wire enters the fitting. This dampening of vibration can lead to fatiguing of the wires at this point and frequent, careful inspection of the area for broken wire should be made. The detection of any broken wires should dictate the immediate replacement of the fitting.

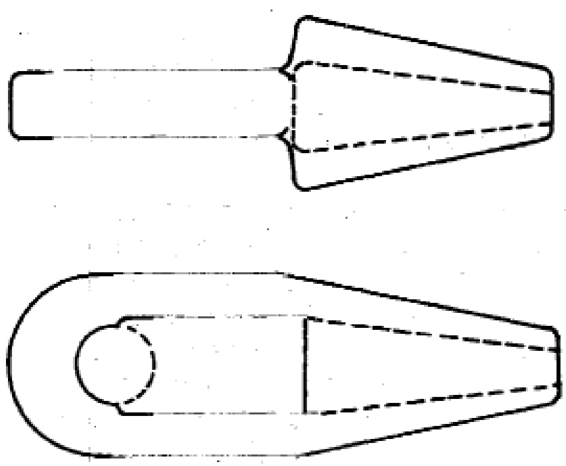


FIGURE 5-3

POURED SOCKET

a. Zinc Socketing Procedures (Figure 5-4) - The procedures for applying zinc-pouted sockets is as follows:

-- Measure the rope ends for socketing and apply serving at the base of the socket. As indicated in Figure 5-4a, the length of the rope end should be such that the ends of the wires when un-laid from the strands will be at the top of the socket basket. Apply a tight wire-serve band for a length of two, rope diameters begin-fling at the base of the socket and extending away from it.

-- Broom out strands and wires in the strands (Figures 5-4b and 5.4c). Un-lay and straighten the individual strands of the rope and spread them evenly so they form an included angle of approximately 60°. If the rope has a fiber core, cut out and remove the core as close to the serving band as possible. Un-lay the wires from each individual strand for the full length of the rope end, being careful not to disturb or change the lay of the wires and strands under the serving band. If the rope has an independent wire rope core (IWRC), un-lay the wires of the IWRC in the same manner.

-- Clean the broomed-out ends (Figure 5-4d). A suggested solvent for cleaning is SC-5 Methyl Chloroform. This solvent is also known under the names of Chlorothane VG or 1,1,1-trichloroethane.

<u>CAUTION</u>: Breathing the vapor of chlorinated solvents is harmful; use only with adequate ventilation. Follow the solvent manufacturer's instructions; observe the label instructions.

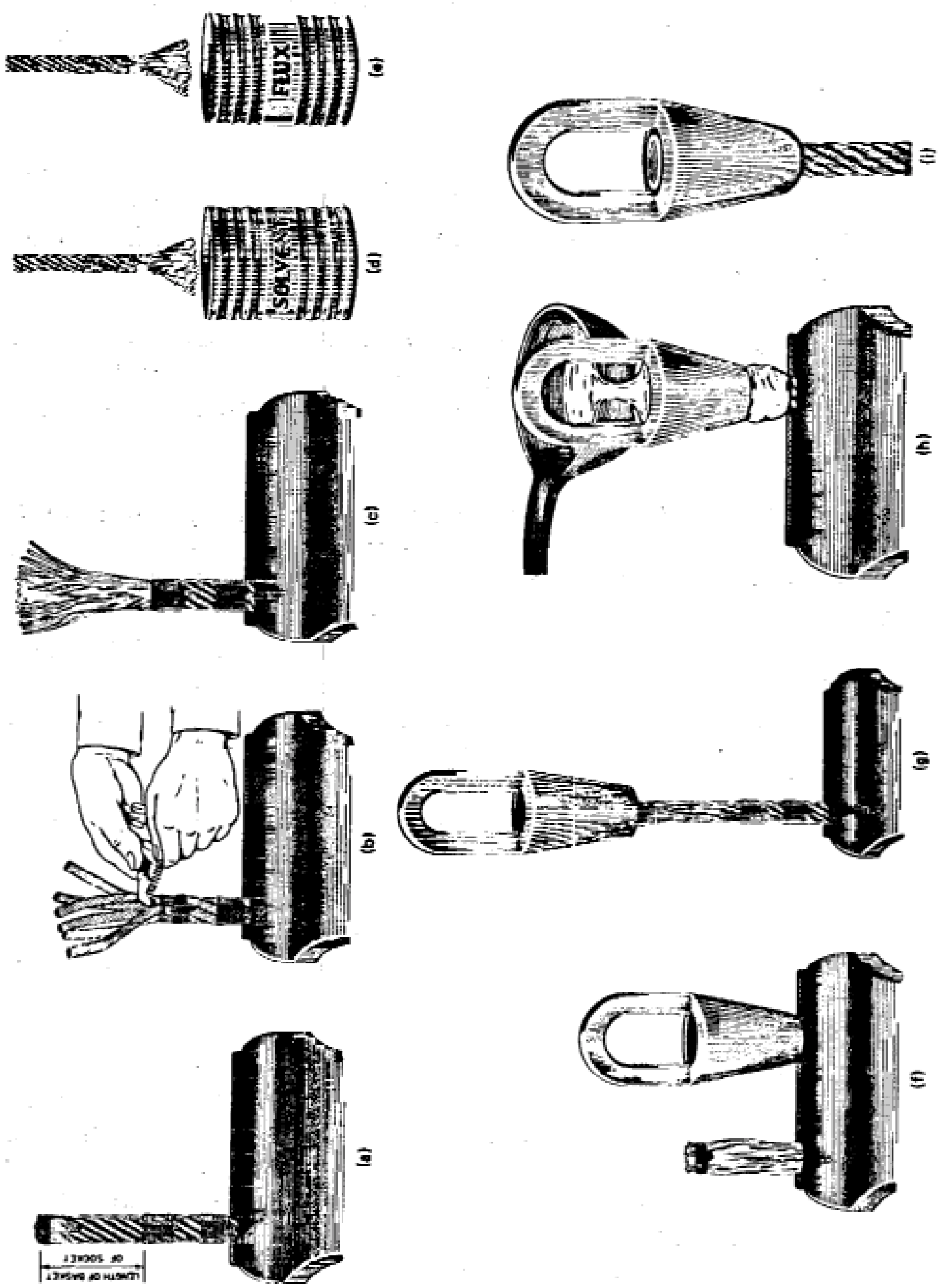
When using a solvent, swish the broomed-out rope end in the solvent and vigorously brush away all grease and dirt making sure to clean all the wires of the broomed-out portion to a point close to the Serving band. A solution of hydrochloric (muriatic acid) may be used for additional cleaning. However, if acid is used, the broomed-out ends of the rope should be subsequently rinsed in a solution of bicarbonate of soda to neutralize any acid that may remain on the rope. Care should also be exercised that acid does not enter the core, particularly if the rope has a fiber core. Ultrasonic cleaning is a preferred method for cleaning rope ends for socketing.

After cleaning, put the broomed-out ends upright in a vise until it is certain that all the solvent has evaporated and the wires are dry.

- <u>Dip the broomed-out rope ends in flux</u> (Figures 5-4e). Make a hot solution of zinc-ammonium chloride flux such as Zalcon K. Use a concentration of one pound of zinc-ammonium chloride in one gallon of water and maintain the solution at a temperature of 180° to 200°F. Swish the broomed-out end in the flux solution, put the open end upright in the vise, and permit all wires to dry thoroughly.

FIGURE 5-4

ZINC SOCKETING



-- Close rope ends and install the socket (Figures 5-4f and 5-4g). Use clean wire to compress the broomed-out rope end into a tight bundle so that the socket can be slipped over the wires. A socket should always be cleaned and heated before placing it in the rope. The heating is necessary to dispel any moisture and to prevent premature cooling of the zinc.

CAUTION: Never heat a socket after it has been placed on the rope because of the hazard of heat damage to the wire rope.

When the socket has been put on the rope end, the wires should be evenly distributed in the socket basket so that zinc can surround every wire. Use utmost care to align the socket with the centerline of the rope and to ensure that there is a vertical, straight length of rope exiting the socket that is equal to a minimum of 30 rope diameters. Seal the base of the socket with fire clay or putty, but be sure this material is not inserted into the base of the socket; if this were done, it would prevent the zinc from penetrating the full length of the socket basket and would, create a void, which would collect moisture when the socket is placed into service.

-- Pour the zinc (Figure 5-4h). Use zinc that meets the requirements in ANSI/ASTM B6-70, Specification for Zinc Metal (Slab Zinc), for "high grade" or Federal Specification QQ-Z351-a Amendment I, and Interim Amendment 2. Pour the zinc at a temperature of approximately 95O~F to 975°F making allowances for cooling if the zinc pot is more than 25 feet from the socket.

CAUTION: Do not heat zinc above 1100°F or its bonding properties will be lost.

The temperature of the zinc may be measured with a portable pyrometer or a Templistik. Remove all dross before pouring. Pour the zinc in one continuous pour to the top of the socket basket so that all the wire ends are covered; there should be no "capping" of the socket.

-- Remove the serving band (Figure 5-4i). Remove the serving band from the base' of the socket and check to see that zinc has penetrated to the base of the socket.

-- Lubricate the rope. Apply a wire rope lubricant to the rope at the base of the socket and on any section of the rope from which the original lubricant has been removed.

b. Procedure for Thermoset Resin of Wire Rope

-- General -- Before proceeding with thermoset resin socketing, the manufacturers instructions for using this product should be carefully read. Particular attention should be given to sockets that have been designed specifically for resin socketing.

-- Seizing and Cutting the Rope - - The rope manufacturer's directions for a particular size or construction of rope should be followed with regard to the number, position length of seizing, and the seizing wire size to be used. The seizing which will be located at the base of the installed fitting, must be positioned so that the ends of the wires to be embedded will be slightly below the level of the top of the fitting's basket. Cutting the rope can best be accomplished by using an abrasive wheel.

-- Opening and Brooming the Rope End - - Prior to opening the rope end, place a short temporary seizing directly above the seizing that represents the base of the broom. The temporary seizing is used to prevent brooming the wires the full length of the basket and also to prevent the loss of lay in the strands and rope outside the socket. Then move all seizings between the end of the rope and the temporary seizing. Un-lay each of the strands that make-up the construction of the rope. Open each strand of the rope and broom or un-lay the individual wires.

When the brooming is completed, the wire should be distributed evenly within a cone so that they form an included angle of approximately 60°. Some types of sockets require a different brooming procedure and the manufacturers instructions should be followed.

-- Cleaning the Wires and Fittings - Different types of resin with different characteristics require varying degrees of cleanliness. For some, the use of soluble oil for cleaning wires has been found to be effective. For one type of polyester resin on which over 700 tensile tests on ropes in sizes 1/4 to 3-1/2 inches in diameter were made without experiencing any failure in the resin socket attachment, the cleaning procedure is as follows:

Thorough cleaning of the wires is required to obtain resin adhesion. Ultrasonic cleaning in recommended solvents such as trichloroethylene or I-I-I trichloroethane or other nonflammable grease-cutting solvents is the preferred method of cleaning the wires in accordance with OSHA Standards. Where ultrasonic cleaning is not available, brush or dip cleaning in trichloroethane may be used; but fresh solvent should be used for each rope and fitting and discarded after use. After cleaning, the broom should be dried with clean compressed air or in some other suitable fashion before proceeding to the next step. The use of acid to etch the wires before resin socketing is unnecessary and not recommended. Since there is a variation in the properties of different resins, the manufacturer's instructions should be carefully followed.

-- Placement of the Fitting - Place the rope in a vertical position with the broom up. Close and compact the broom to permit insertion of the broomed rope end into the base of the fitting. Slip on the fitting, remove any temporary banding or seizing as required. Make sure the broomed wires are uniformly spaced in the basket with the wire ends slightly below the top edge of the basket; make sure that the axis of the rope and the fittings are aligned. Seal the annular space between the base of the fitting and the existing rope to prevent leakage of the resin from the basket. A non-hardening butyl rubber-base sealant gives satisfactory performance. Make sure that the sealant does not enter the base of the socket so that the resin may fill the complete depth of the socket basket.

-- Pouring the Resin - Controlled heat curing (but without open flame) at a temperature range of 250° to 300°F is recommended -- and is essential if ambient temperatures are less than 60°F. When controlled heat curing is not available and ambient temperatures are not less than 60°F, the attachment should not be disturbed and tension should not be applied to the socketed assembly for at least 24 hours.

-- Lubrication of Wire Rope after Socket Attachment - After the resin has cured, re-lubricate the wire rope at the base of the socket to replace the lubricant that was removed during the cleaning operation.

c. Description of the Resin

-- General - Resins vary considerably according to the manufacturer; it is important to refer to the manufacturer's instructions before using resins as no general rules about them can be established.

Properly formulated thermoset resins are acceptable for socketing. These resin formulations, when mixed, form a pourable material that hardens at ambient temperatures or upon the application of moderate heat. No open-flame or molten-metal hazards exist with resin socketing since heat-curing, when necessary, can only be carried out at a relatively low temperature (250° to 300°F) that can be supplied by electric-resistance heating.

Tests have shown satisfactory wire rope socketing performance by resins having the properties of a liquid thermoset material that hardens after mixing with the correct proportion of catalyst or curing agents.

-- Properties of Liquid (Uncured) Material - Resin and catalyst are normally supplied in two separate containers, the complete contents of which, after thorough mixing, can be poured into the socket basket. Liquid resins and catalyst should have the following properties:

1) Viscosity of Resin-Catalyst Mixture - - The viscosity of the resin-catalyst mixture should be 30,000 to 40,000 CPS at 75°F immediately after mixing. Viscosity will increase at lower ambient temperatures and resin may need warming prior to mixing in the catalyst if ambient temperatures drop below 40°F.

2) Flash Point - Both resin and catalyst should have a minimum flash point of 100°F.

3) Shelf Life – Resin and catalyst should have a minimum of one-year shelf life at 70° F.

4) Pot Life and Cure Time - After mixing, the resin-catalyst blend should be pourable for a minimum of eight minutes at 60°F and should harden in 15 minutes. Heating of the resin in the socket to a maximum temperature of 250°F is permissible to obtain full cure.

-- Properties of Cured Resin

1) Socket Performance - Resin should exhibit sufficient bonding to solvent-washed wire in typical wire rope end fittings to develop the nominal strength of all types and grades of rope. No slippage of wire is permissible when testing resin-filled rope socket assemblies in tension; however, after testing, some “seating” of the resin cone may be apparent and is acceptable. Resin adhesion to wires shall also be capable of withstanding tensile shock loading.

2) Compressive Strength - The minimum compressive strength for fully cured resin should be 12,000 lb/in^2.

3) Shrinkage - - Fully cured resin may shrink a maximum of 2%. The use of an inert-filler in the resin is permissible to control shrinkage, if the viscosity provisions specified for the liquid resin are met.

4) Hardness - A desired hardness of the resin is in the range of Barcol 40-55.

-- Resin Socketing Compositions - Manufacturers directions should be followed in handling, mixing, and pouring the resin composition.

-- Performance of Cured Resin Sockets - - Poured resin sockets may be moved when the resin is hardened. After the ambient or elevated temperature cure recommended by the manufacturer, resin sockets should develop the nominal strength of the rope, and should also withstand shock loading sufficient to break the rope without cracking or breakage. Resin socketing materials that have not been tested to these criteria by the manufacturer should not be used.

3.5 Compressed Sleeves (Nicopress)

The compressed or Nicopress sleeve (Figure 5-5) represents a style of wire and cable termination which has been available for the past thirty years. Its high efficiency, 95% - 100% of the breaking strength of the wire, and its simplicity of installation have made it an ideal type of termination for general field use. The success of the compressed sleeve is dependent upon the selection of the proper sleeve to match the wire to be terminated and matching the compression requirements of the sleeve. Specialized tooling is produced which will ensure proper compression to achieve the maximum holding power of the fitting.

When terminating a rope or cable, both the sleeve and tool should match the requirements of the cable size. Table 3 provides a guide to this selection as well as the recommended compressions needed for maximum efficiency.

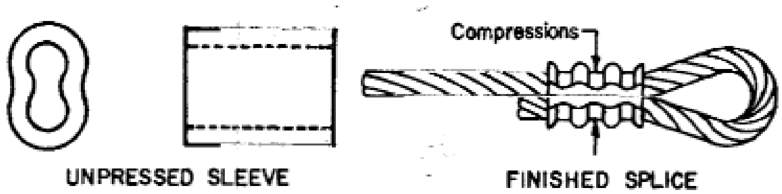


FIGURE 5-5

COMPRESSED SLEEVE (NICOPRESS)

It should be noted that in Table 3 the number of sleeves, required per installation, to achieve the maximum holding power is the same for all wire sizes. The manufacturer recommends a single sleeve per termination and the addition of more sleeves in no way increases the ultimate holding power of this type of termination. One factor, which can affect the efficiency of the compressed sleeve, is excessive compression. The recommended compressions shown in the table and in Figure 5-5, allow for ultimate holding while providing an adequate stress relief at both ends of the sleeve. This factor can become extremely important when Stainless steel sleeves are used in the termination of a wire.

As with other types of terminations, which require the wire to be looped at the termination point, it is necessary to use a properly sized thimble to protect the wire. The discussion found in Chapter 6, Section 3.2, also applies to the use of compressed sleeves.

3.6 Swaged Terminations

The swaged style of wire termination, possesses a 100% efficiency rating when compared to the breaking strength of the wire being

TABLE 3

CABLE SIZE	SLEEVE STOCK NO. PLAIN	SLEEVE STOCK NO. PLATED	TOOL STOCK NO.	LENGTH BEFORE COMPRESSION	LENGTH AFTER COMPRESSION
1/16"	18-1-C	28-1-C	51-C-887	3/8"	7/16"
1/8"	18-3-M	28-3-M	51-M-850	9/16"	3/4"
3/16"	18-6-X	28-6-X	51-X-850	15/16"	1-3/16"
1/4"	18-10-F6	28-10-F6	3-F6-950	1-1/8"	1-1/2"

CABLE SIZE	SLEEVE STOCK NO. PLAIN	SLEEVE STOCK NO. PLATED	DIE STOCK NO.	WIDTH OF PRESS	NUMBER OF PRESSES	GAUGE
5/16"	18-13-G9	28-13-G9	OVAL-G9	1-5/16"	1	.730
3/8"	18-23-H5	28-23-H5	OVAL-H5	1-5/16"	1	.795
7/16"	18-24-J8	28-24-J8	OVAL-J8	7/8"	2	.915
1/2"	18-25-K8	28-25-K8	OVAL-K8	13/16"	2	1.000
9/16"	18-27-M1	28-27-M1	OVAL-M1	9/16"	3	1.125
5/8"	18-28-N5	28-28-N5	OVAL-N5	5/8"	3	1.285

terminated. Its high efficiency is achieved through the use of large hydraulic presses, which exert a uniform compressive force on the fitting (Figure 5-6).

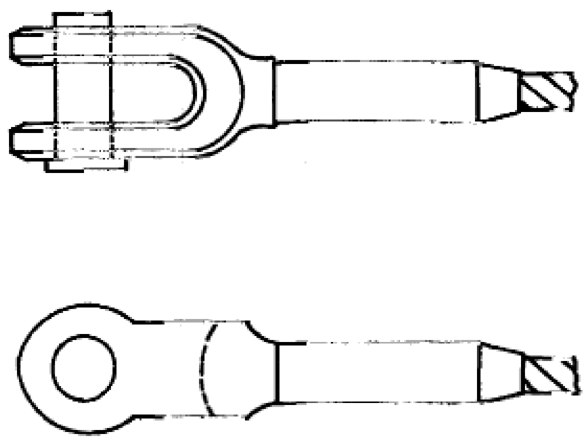


FIGURE 5-6

SWAGED TERMINATION

When swaged sockets are used with 3 x 19 wire rope, it is necessary to insert filler pieces into the spaces between the strands, inside the socket, prior to compression of the fitting. The soft wire fillers serve to increase the effective surface area of the 3 x 19 rope and allow a more uniform compression and holding power to be achieved (Figure 5-7).

The specialized nature of the swaged termination does not lend itself easily to field applications due to operate the equipment. The frequent re-termination of both trawl and hydrographic wires that is required at sea also precludes the use of this particular type of fitting for working cable applications. In addition, the corrosion potential within the swaged socket occurs in an area, which is impossible to inspect. Corrosion of the filler wire can occur over time and eventually result in a failure of the termination without prior warning or evidence of weakening.

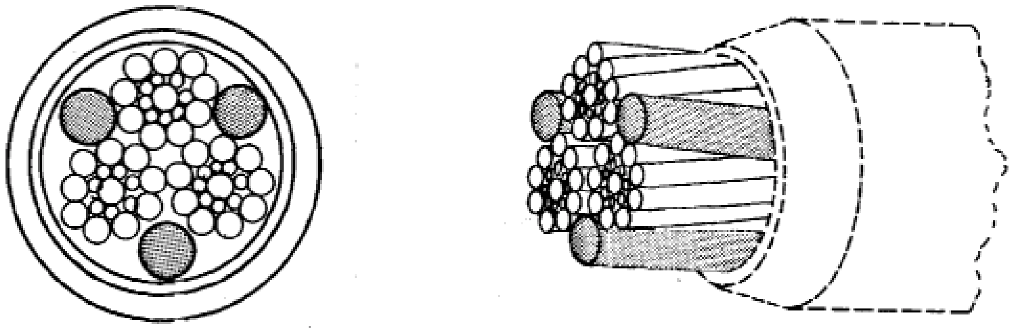


FIGURE 5-7

FILLER WIRES FOR SWAGED TERMINATION

3.7 Mechanical Terminations (Electroline or Fiege)

Perhaps the most common fitting used to terminate the deep-sea trawl wire is the Electroline eye socket assembly or “Fiege fitting” as it is frequently called. The Electroline termination (Figure 5-8), is a three component device consisting of a threaded sleeve, socket assembly and a plug which, when properly assembled, will result in a termination strength equal to 95%-100% of the ropes’ breaking strength. The high level of fitting efficiency is achieved through the use of the plug as a wedge and by carefully following the assembly instructions.

Although this type of fitting can be used with all styles of wire rope construction, only three-strand, torque balanced wire rope will be discussed here. Changes in wire construction will require changing the style and possibly the material used to construct the plug (Photograph I). The particular plug we are interested in for the purpose of this discussion is a triangular plug specifically designed for three-strand wire rope (Figure 5-9).

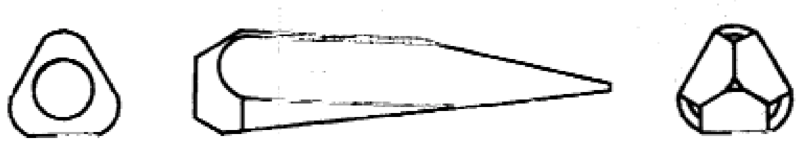


FIGURE 5-9

3 x 19 WIRE ROPE TRIANGULAR PLUG

The following assembly procedures have been developed to detail the application of the Electroline mechanical termination for 3 x 19 torque-balanced wire rope. It should be remembered that this application requires that an oversized fitting and special plug be used.

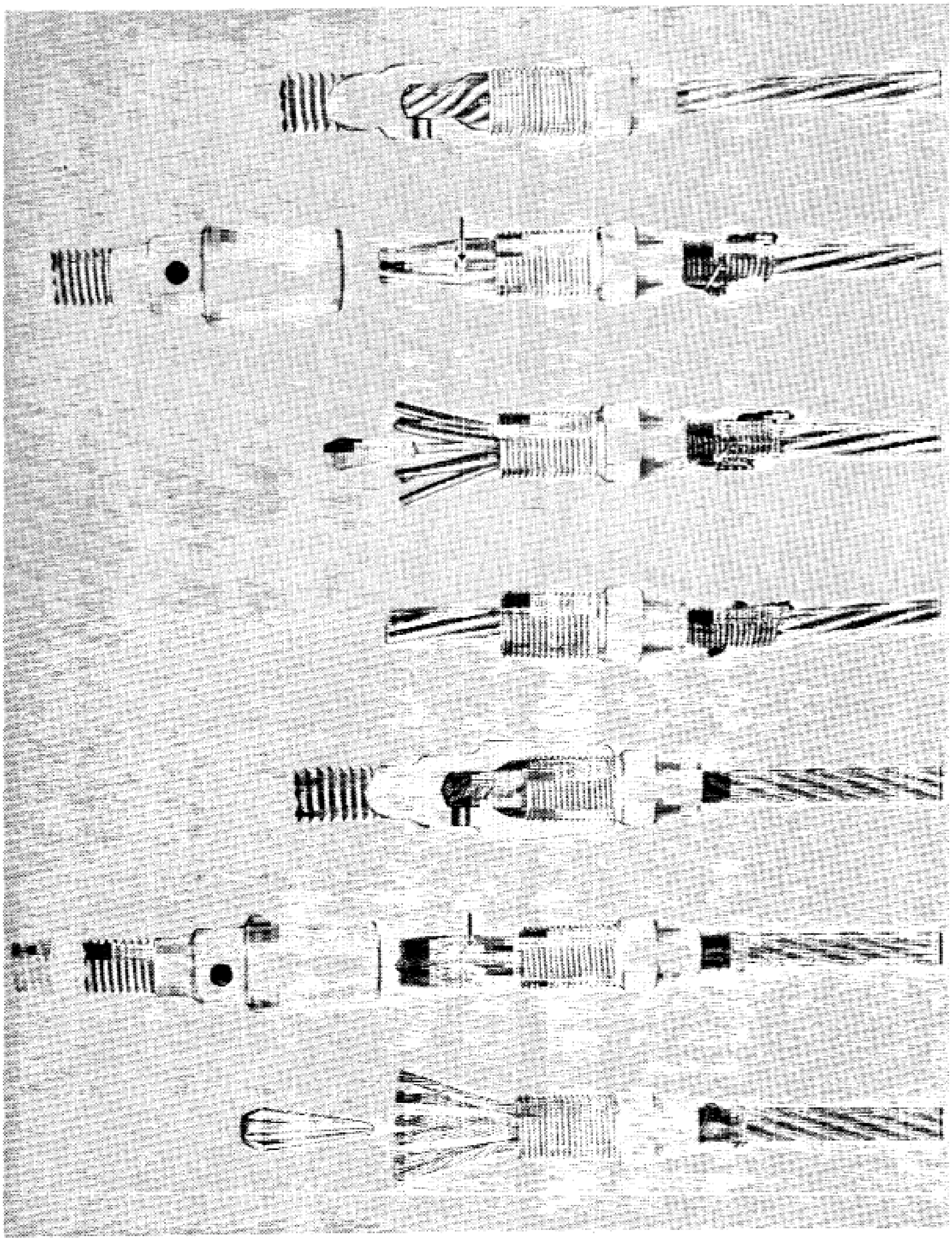
1. When assembling Electroline fittings on wire rope it is recommended, that assembly blocks be used to prevent the rope from being nicked by the jaws of the vise, to protect the lay of the rope, and to hold all wires in the strands firmly so the plug can be driven to a solid seat.

Assembly kits are available in the following three sizes:

Part No.	For Rope Sizes
SP-307A	1/8” thru 9/16”
SP-307B	5/8” thru 1”
SP-307C	1-1/8” thru 1-1/2”

2. The bitter end of the wire to be terminated should be carefully cut to insure a 90° surface. Prior to cutting, the wire

PHOTOGRAPH 1



should be seized with light wire to prevent unraveling of the rope when cut.

3. Place the Assembly blocks on the rope and place blocks in vise. If the assembly blocks are missing or unavailable, two short pieces of wood strapping can be used between the vise jaws. The wire length should be carefully measured (Table 4) before securing the wire between the wooden blocks in the vise.

4. Position end of rope to dimension "A" as shown on the chart for the size rope being assembled. Tighten vise firmly. This position of the assembly procedure should be carefully checked with a good ruler to ensure that the exact measurements specified in the chart are met. In addition, care should be taken to ensure that the wire is perpendicular to the assembly block, before tightening vise.

5. Remove seizing on end of rope.

6. Twist threaded end of sleeve over end of rope. Twist in direction of rope lay. Check dimension "B" as shown on Table 4.

7. Un-lay one of the three strands. If rope has a right lay, un-lay each of the other two strands in counter-clockwise order. If rope has a left lay, un-lay each of the other two strands in clockwise order. When done correctly, the three outer strands form a symmetrical basket. Do not attempt to straighten the spiral lay of the three strands.

8. Place the plug in the center of the three strands. Drive the plug downward with a hammer while making certain that each of the strands is positioned properly along the sides of the plug.

9. Once the plug is in position it should be driven to a solid seat using a hammer and draft pin of at least V2" diameter. When seated, the top of the plug should be well below the tops of the wires.

10. Remove assembly from vise, remove assembly blocks, and clamp the hex of the sleeve in vise. At this time wrap a layer of tape around the wire where it exits the sleeve. The tape should be as close to the base of the sleeve as possible.

11. With a piece of tubing (1.0. of tubing should be 1/32” to 1/16” larger than 0.0. of each strand of rope), bend each of the three strands in toward the center of the plug. Tubing is furnished in each of the three assembly kits.

    As an alternative to using the tubing, a small 2” hose clamp can be used to squeeze the ends of the wires toward the center of the plug, allowing the socket to slide over the wire ends and mate with the threaded sleeve.

12. Place socket over ends of strands, twist on in the direction of the lay of the rope. Engage threads of sleeve and tighten socket securely on sleeve.

    During the tightening of the fitting it is recommended that a pair of large crescent wrenches (20”) be used instead of a bar through the eye of the socket. The reasoning behind this is that the bar can and will deform the eye of the socket making insertion of a bushing or screw pin difficult.

13. If assembled correctly, the end of the rope will be visible in the inspection hole. Several threads will be visible on the sleeve below the eye socket after tightening. The best method for checking the visibility of the wires through the inspection hole is with the use of a flashlight. If the ends of the wires are not visible the fitting should be removed, the wire cut, and the termination installed again.

    At this time it would pay to check the tape on the wire below the fitting sleeve to determine the amount of slippage that has occurred. The average amount allowable is approximately 1/2”.

14. After a proof load is applied to the assembly, the plug will seat further in the sleeve and the rope will not be visible in the inspection hole. This final seating of the plug insures an assembly of maximum strength. After the proof loading of the fitting has been accomplished the fitting should tightened again to recover any exposed threads. The recommended proof load should be 8,000 lbs. for 1/2-9/16’, 5/8 wire ropes.

TABLE 4

Rope Size	Fitting Rope Size	Sleeve Rope Size	Plug No. -MZ	Dimensions A+1/8 -0	B+1/8 -0
1/16	1/8	1/8	MZ 1606	1-9/16	5/8
1/8	3/16	3/16	"1612	1-31/32	3/4
3/16	1/4	1/4	"1618	2-1/4	13/16
1/4	5/16	5/16	"1625	2-3/4	1
5/16	3/8	3/8	"1631	3-1/8	1-1/8
3/8	7/16	7/16	"1437	3-5/8	1-5/16
7/16	1/2	1/2	"1443	4-1/8	1-1/2
1/2	9/16	9/16	"2250	4-3/4	1-5/8
9/16	5/8	5/8	"2250	4-3/4	1-5/8
5/8	3/4	3/4	" 1462	5-1/2	1-7/8
3/4	7/8	7/8	"2275	6-1/2	2-1/4
7/8	1	1	"2287	7-7/8	2-3/4
1	1-1/8	1-1/8	"2299	9-3/8	3-1/8

ELECTROLINE TERMINATION ASSEMBLY DIMENSIONS

MISCELLANEOUS

The following items are not recorded in any of the literature available to date, but constitute a series of points that can help in detecting or preventing failure, etc.

A. Eye Socket Hole

It has proven to be a good idea to accurately measure the hole diameter of each new fitting prior to its first use and again at the end of each cruise. In this way elongation of the hole due to applied stresses can be detected and the fitting discarded prior to a failure.

Some form of indexing and logging of this data should be established. It is also recommended that when the hole is measured, at least three separate readings be taken to determine an average.

B. Eye Socket Bushing

It is recommended that a bushing be used inside the eye socket hole when the pin passing through the eye is smaller than the opening. This approach will lessen the point load exerted on the fitting during periods of high stress.

C. Removal of Fitting and Plug

When a termination is to be removed it is suggested that the wire be cut off as close to the sleeve as possible. The fining is then disassembled, the sleeve with plug clamped in a vise with the sawn off section of wire uppermost, and the plug and wire driven out with a 1/2" drift pin. It will be noted that a considerable amount of force will be required to accomplish this task.

D. Re-use of Triangular Plug

It is in no way recommended that the triangular plug be reused after it has been removed from the sleeve. The potential for damage to the plug is very high and re-use can only jeopardize the equipment deployed on the next lowering. Also a careful inspection of the sleeve and eye socket should be made prior to their re-use or storage. Items to look for are broken threads, elongated holes, and cracks in either the sleeve or eye socket.

4.0 ELECTRO-MECHANICAL TERMINATIONS

Since it would he impractical to discuss fully all of the available styles of electro-mechanical terminations this chapter will, instead, concentrate on three specific types of terminations. Specifically these will include: 1) straight mechanical fittings; 2) combination mechanical and epoxy terminations; and 3) helical terminations. It is felt that other termination styles, which are available lie somewhere within the range of those, discussed in this chapter.

One factor, which cannot be stressed often enough, is the need to carefully follow the recommended installation procedures for the fitting selected. If the termination is to perform as it is advertised, it is necessary for the individual installing of the fitting to understand that recommended procedures should not be compromised by a lack of attention to detail. Required wire lengths should not be estimated and the necessary curing times for epoxies must not be disregarded in an attempt to re-deploy an instrument. The result of such careless practices is usually the loss of an expensive piece of equipment.

4.1 Electroline E-M Cable Terminations

This style of termination, like its wire rope counterpart, relies on the use of a series of plugs to achieve its holding power. Because this is a purely mechanical termination it can be used immediately after assembly and testing. Other terminations, which combine an expoxy filler in the mechanical fittings are restricted by the curing time of the epoxy and therefore have a relatively long time period between assembly and use.

The purely mechanical termination is quite simple to install in the field requiring only basic hand tools and no special technical training on the part of the installer. The only critical aspect of assembling this style of fitting is close attention to the installation instructions. By carefully following the Instructions detailed below the maximum efficiency of the fitting can be assured.

4.2 Installation Procedures

The following procedures have been developed for the termination of double armor electro-mechanical cable. The success of this termination rests on the proper installation of two hollow plugs (Figure 5-10) and the careful measurement of the wire lengths required in Table 5.

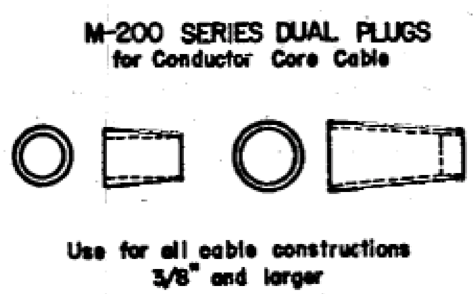


FIGURE 5-10

TABLE 5

E.M. FITTING ASSEMBLEY DIMENSIONS

FITTING SIZE	CABLE SIZE	NO. WIRES PER. LAYER	CONDUCTOR HOLE SIZE	PLUG PART NO.	DIMENSIONS A	DIMENSIONS B	DIMENSIONS C
1/8"	1/8"	12 x 18	3/32"	ME-212	1-3/8"	CUT TO REQUIRED LENGTH PLUS 1" TO 1-1/2" ADDITIONAL SPACE TO ALLOW FOR SEATING OF PLUG	7/16"
3/16"	3/16"	18 x 18	1/8"	ME-218	1-23/32"		1/2"
1/4"	1/4"	15 x 15 18 x 18	3/16"	ME-225	1-15/16"		9/16"
5/16"	5/16"	15 x 15 18 x 18 18 x 24 24 x 24	3/16"	ME-231	2-5/16"		5/8"
3/8"	3/8"	18 x 18	1/4"	ME-237	2-5/8"		3/4"
7/16"	7/16"	18 x 18	5/16"	ME-243	3-5/16"		3/4"
1/2"	1/2"	18 x 18 24 x 24	3/8"	ME-250	3-1/2"		15/16"
9/16"	9/16"	24 x 24	7/16"	ME-256	4-1/4"		1-1/8"

Installation of the fitting can be accomplished using the following procedure:

1. The cable to be terminated should have the better end seized and the cable cut off square. Once this is accomplished place the cable in the assembly block, position the end of the strand to dimension A and B for the size cable being terminated and clamp the assembly block in a vise (Figure 5-11). When the cable is clamped in the vise the seizing can be removed.

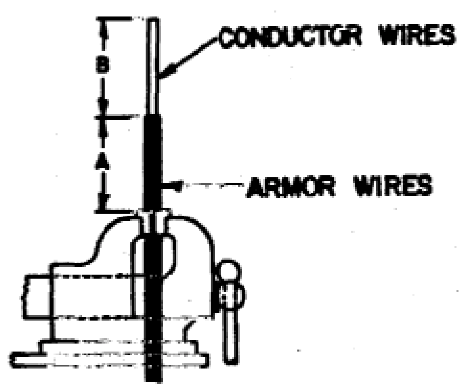


FIGURE 5-11

2. Twist the threaded sleeve over the end of the cable and check the length of the exposed armor strand against dimension C in Table 5. If this length is compatible with the table proceed to un-lay the outer armor wires. However, if this dimension is not met reposition the cable in the blocks before continuing. Place the large hollow plug over the center wires of the cable and carefully drive it to a solid seat (Figure 5-12). Once seated the inner armor wires should be un-laid and the small hollow plug slipped over the conductor wires and driven to a solid seat.

3. Remove the assembly blocks from the vise and the cable and clamp the threaded sleeve in the vise as shown in Figure 5-13. The broomed-out armor wires can now be bent inward around the conductor wires. In order to protect the conductor wires during the process, a piece of tubing should be slipped over the conductors prior to bending the armor wires inward.

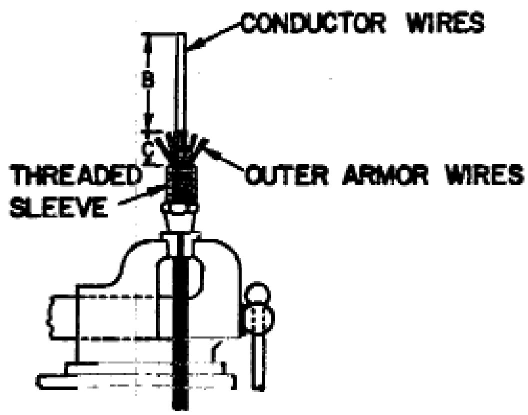


FIGURE 5-12

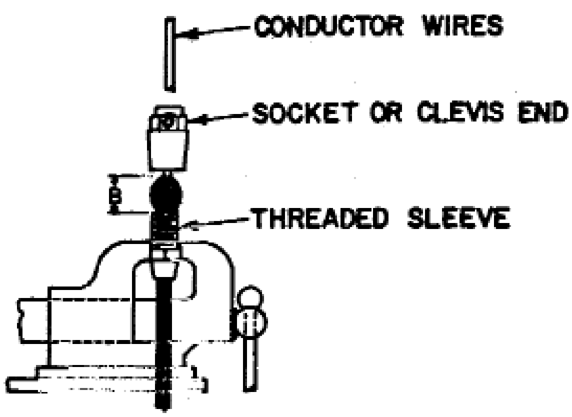


FIGURE 5-13

Once this step is completed twist the socket or clevis end portion of the fitting over the ends of the wires and feed the conductor wires through the hole provided. Engage the threads of the sleeve and tighten securely. When tightening the socket or clevis it is advisable to remember the procedure discussed in section 3.7(12).

4. Once the socket or clevis has been securely tightened on the sleeve the ends of the armor wires or tubing will be visible in the inspection hole. If the installation has been properly accomplished (Figure 5-14), several threads will be exposed on the sleeve. Should the armor wires or tubing not be visible in the inspection hole the termination should not be used, but should instead be removed and the cable re-terminated.

5. It is advisable after the termination has been successfully installed to perform a pull test at a load equal to at least two times the anticipated instrument load to assure the safety of the equipment.

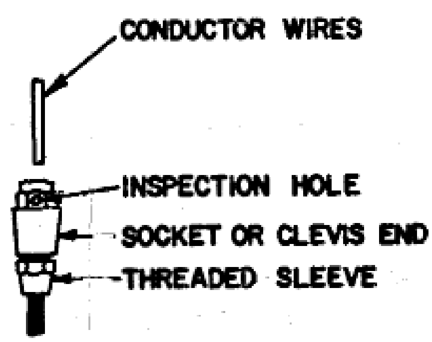


FIGURE 5-14

4.3 Combination Mechanical/Epoxy Termination

The particular fitting, which will be discussed here, is the DYNA-GRIP termination for electro-mechanical cables. The basic termination design consists of an oval-shaped hollow insert which slips over the cable, a set of helically-formed rods which wrap (by hand) over the cable and insert, a housing with internal contour to match the rods and insert and a threaded retainer (Figure 5-15). Holding strength of the termination is developed by the preformed principle of the helically formed rods and gripping by the matching insert and housing (Figure 5-16). There is no reliance upon special tools or user proficiency. Every installation is uniform and repeatable in holding ability and appearance. There is no crushing or deformation of the cable elements, and yet, the termination will hold the full-rated strength of the cable for which it is designed.

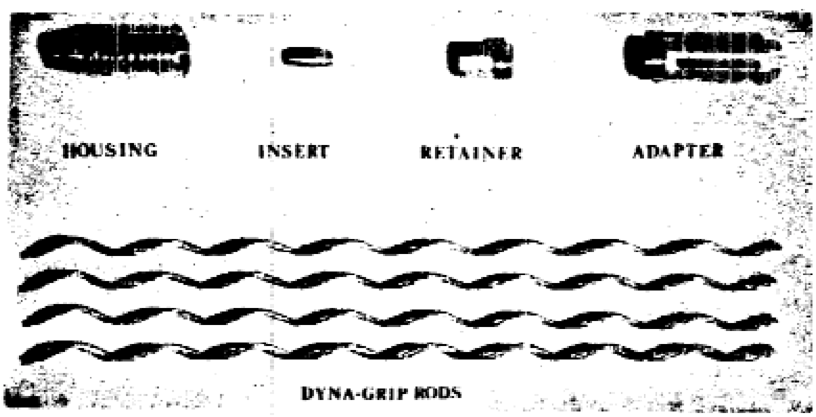


FIGURE 5-15

1. Cable Preparation: If the armor is to be terminated at the fitting, care should be taken to allow for a sufficient length of armor to extend beyond that of the Rods. The outer armor wires should be cut about 1/4” shorter than the inner wires and then taped. After application of the Dyna-Rods, the inner armor should be bent away from the cable to prevent chaffing of the insulation.

2. For proper positioning of the helical rods (Figure 5-17), match the center mark on the insert to the color mark on the rods. For proper and easy installation of the rods, the following are important:

   a. Begin application with a two-rod subset and end application with a two-rod subset.

   b. Wrap the rods one subset at a time about the cable and ‘over the insert starting at the trimmed end of the cable.

   c. Align the ends of the rods closely with each other.

   d. Do not allow gaps between subsets or accumulation of gaps between subsets or accumulation of gaps could interfere with application of subsequent subsets.

   e. Do not allow any of the helical rods to cross each other.

3. After complete set of rods is wrapped on (Figure 5-18), slide housing over rod and insert assembly until it seats over insert.

4. Insert retainer and screw tightly into housing. Use the clevis, with hex keys, as a spanner wrench (Figure 5-19). When the retainer is properly in place, there will be no looseness, in the entire assembly.

5. Place the housing in a nose down position. The epoxy filler is then prepared.

Preparation of epoxy:

Thoroughly mix the contents of each can before combining them. Each part tends to separate into layers. Each component must be thoroughly mixed until it is homogeneous. This is critical to proper cure of the material. Combine the contents of the two cans. Mix the combination thoroughly for five to ten minutes. The epoxy is then ready to pour.

6. Dam the leading edge (nose) of the housing. A mastic-modeling clay or similar material can be used. Pour the epoxy material into the housing. It is important that the epoxy penetrates into the interstices between individual armoring rods. Breaking away the dam and visually checking to see if the adhesive is seeping through the fitting may check adhesive penetration.

7. Screw in mounting adaptor and lock thread with groove-pin (or set screws) before the epoxy filler hardens (Figure 5-20).

8. Completed assembly (Figure 5-21).

9. The following techniques should be observed in order to terminate:

   a. Do not terminate the armor wires of electro-mechanical cable inside the housing or apply the epoxy filler in a manner that would cause a bond of the cable armor to the electrical core.

   b. Allow the epoxy filler material to cure for 24 hours (normal temperatures) before using.

   c. Reuse of the preformed helical rods is not generally recommended after load has been applied.

   d. Caution should be used when overboard sheave has a groove diameter less than 40 times the diameter over the helical portion of the assembly.

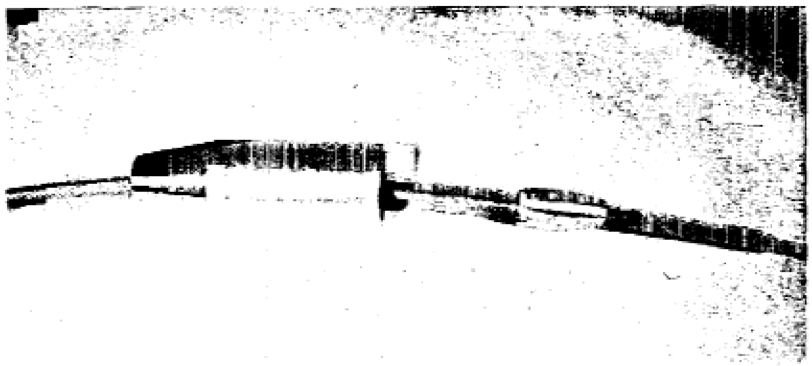


FIGURE 5-16

FIGURE 5-17

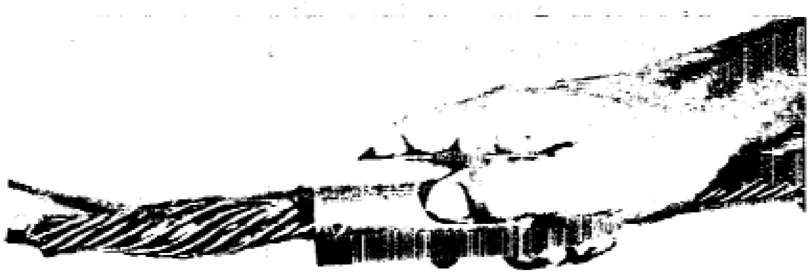


FIGURE 5-18

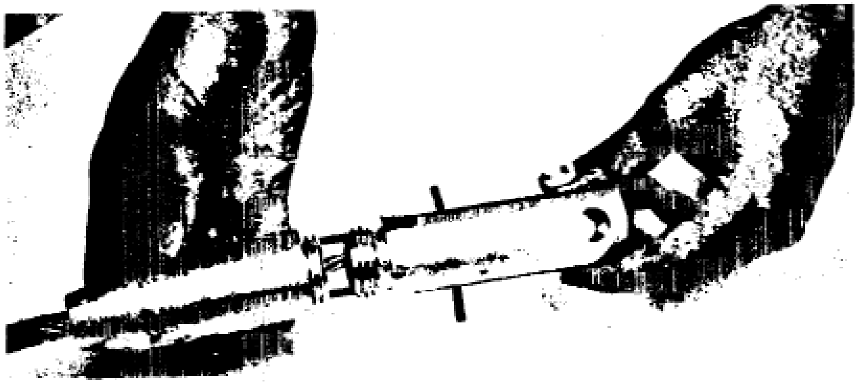


FIGURE 5-19

FIGURE 5-20

FIGURE 5-21

4.4 Helically Wound Terminations

This particular style of termination is designed for electro-mechanical cables larger than I inch diameter (Figure 5-22). Certain limitations are present in the use of this termination and are expressed below.

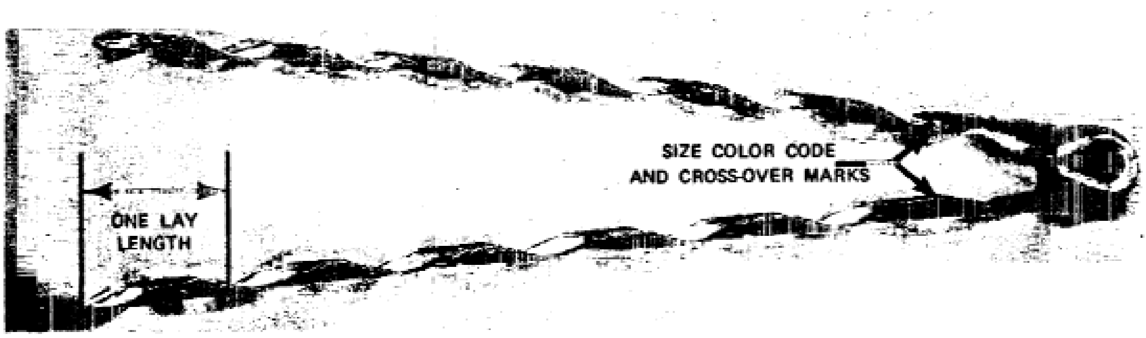


FIGURE 5-22

1. The preformed Cable Stopper must have lay direction as the cable. CAUTION: preformed Cable Stopper with an opposite lay cable.

2. In the application of preformed Cable Stoppers, marlinspikes or screwdrivers should be used only as an aid in splitting the legs and snapping the ends in place.

3. Wire rope thimbles or equivalent fittings of the same size as the Cable Stopper should be used.

   Successful application of this termination can be achieved by following the procedure detailed below.

1. Start application by wrapping on two lay lengths of first leg, starting at crossover marks (Figure 5-23).

2. At this point of the application, install a heavy-duty wire rope thimble (if required) (Figure 5-24). If the eye is intended to be attached by a shackle, it should then be placed on a pin or shackled to a pad eye to keep it from turning. Match the crossover marks and apply the second leg the same lay length as the first. Application is made easier if the leg of section being applied is pulled out and around the cable in one continuous motion.

FIGURE 5-23

3. Use a marlinspike or screwdriver and split the legs into sections for the one pitch length as shown in Figure 5-25.

4. Split both legs back to the applied portion of the Cable Stopper as shown in Figure 5-26. The section closest to the cable (#1, above) should be applied and followed consecutively by the sections closest.

5. At this point the first section should be applied to completion, followed consecutively by each of the other legs (Figure 5-27).

6. As shown in Figure 5-28, a marlinspike or screwdriver can be used to snap the ends into place.

7. Completed application of the preformed Cable Stopper (Figure 5-29). Make sure all rods are in contact with the cable. Should there be one rod under another, remove to that point and re-apply.

8. The following should also be considered when this type of termination is selected:

    a. The preformed Cable-stopper dead-end was developed as a Strain-relief fitting for data cables. These can be applied to either bare armor or jacketed cables and are normally designed to hold 50% of the rated breaking strength.

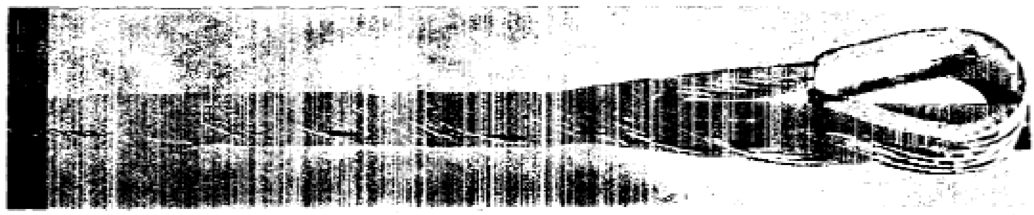


FIGURE 5-24

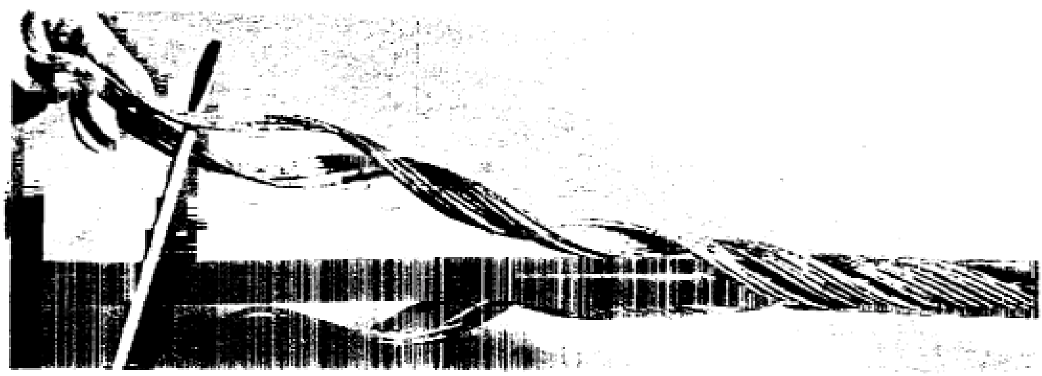


FIGURE 5-25

FIGURE 5-26

FIGURE 5-27

FIGURE 5-28

FIGURE 5-29

b. The eye of the grip is made with sufficient length to allow the cable to proceed on through and beyond the termination point. This provides complete continuity of the cable to an over-boarding application with a strain-relief member of proven performance.

c. The preformed helical concept is the only method of termination or holding of a cable that will not crush the cable or create a high-stress potential failure point.

5.0 KEVLAR ROPE CONSTRUCTIONS

The construction of KEVLAR® ropes may be similar to that of wire rope or to synthetic braided ropes. The best construction depends on the particular application. This decision should be left to the experts such as the rope manufacturers. To date there are no MLL-R or MIL-W specifications for KEVLAR® ropes as there are for wire ropes. The state-of-the-art is expanding so rapidly that whatever is written for this text may be obsolete by the time it is printed. There are certain generic types of KEVLAR® ropes. Without going into great detail on the internal constructions, we will attempt to explain their basic designs.

5.1 Working Ropes and Cables

Aramid working ropes are a high strength, low stretch flexible rope for the same applications as nylons and polyesters synthetic braided ropes. Aramid has the advantages of a higher strength to weight ratio in addition to greater personnel safety. Due to low stretch, the snap-back safety hazard at failure is virtually non-existent. Working ropes are used for:

Boat davits & gripes	pendant lines
canopy stays	pusher cables
constant-tension mooring lines	tow lines
safety ropes	trawling lines
lashing ropes	

Working ropes are supplied with a braided polyester jacket although other jacket materials are available.

5.2 Static Ropes

Aramid static ropes are a high strength to weight replacement for wire ropes in static applications. They are designed for cyclic-loading but has a low fatigue life when used on winches or over sheaves. For long-term applications, an extruded jacket is recommended (as flexibility is not required) to exclude dirt, water and ultra-violet light. Aramid static ropes are effective where low weight and non-conductivity are required. Typical applications for static ropes are:

naval lifelines
antenna guys
canopy stays
barrier ropes

The selection of the best jacket material is dependent on the requirements for:

abrasion resistance
flexibility
insulation resistance
aeolian vibration dampening
protection from light
resistance to chemicals
protection from heat
ice shedding
cold flow
water permeability
cut thru resistance

See Table 6.

5.3 Winch Ropes

Aramid winch ropes are a replacement for wire rope where long lengths are required. Oceanographic winch wire ropes have corrosion problems and their length is limited by their own weight. An aramid rope of the same size as wire rope has a greater operating load capacity due to the low weight of the rope. The internal construction or lay of the aramid yarn is a function of torque, strength, internal abrasion, flexibility, operating load, winch cycle life, etc. As with wire rope, each different construction has its own unique advantages and disadvantages.

The life of a winch rope may be improved by (1) adding an extruded jacket, (2) separating the extruded jacket from the aramid core with a braided jacket or Mylar tape, and by (3) pre-stressing the aramid core prior to extruding the jacket. The two primary reasons for the extruded jacket are (1) abrasion resistance for use with a sheave or winch, and (2) the jacket will maintain the original shape of the aramid core when the rope passes over a sheave. If the rope is allowed to flatten, internal abrasion of the aramid fibers will shorten its life.

The purpose of the intermediate braid or tape is to reduce the abrasion at the aramid/jacket interface in bending. Pre-stressing or tensioning the rope prior to extruding the jacket is used to remove the constructional stretch of the aramid rope. If this is not accomplished, the rope will shrink under load and pull away from the jacket. This, in turn, will allow the rope to deform when passing over a winch or sheave and the subsequent internal abrasion will shorten the operating life of the system. Pre-stressing is an additional expense and the user must decide if the increase in rope life is justified.

5.4 Dynamic Cycling Ropes

These ropes are designed to survive constant cyclic loading and working over sheaves. An example would be Riser Tensioner ropes for the risers of oil drilling rigs. Ropes of this type may range from 7/8" diameter to as large as 6" diameter. Normally they were constructed of multiple small "working" ropes in a typical wire rope lay, such as 1 x 6 x l2, etc. Each of the individual small ropes or strands will be jacketed with a braid or braid and extruded jacket or only a thin extruded jacket. An outer jacket is normal over the finished construction to maintain the shape of the rope. The outer jacket may be braided or extruded as determined by the need for abrasion protection and flexibility.

5.5 Electromechanical Cables

EM cables are a composite of electrical conductors, jackets, separators and a KEVLAR® strength member. These cables are of special construction to suit the user. A typical construction would be a core of multiple electrical conductors or fiber optics, around which is an extruded jacket or bedding for the KEVLAR® strength members. The KEVLAR® may be braided or applied in contra-helical layers. Finally, there is an extruded jacket for protection. When the cable must survive bending under load, the layers of KEVLAR® may be separated by a braid or tape to reduce internal fretting of the KEVLAR®. As KEVLAR® has more stretch than the electrical conductors, it should be pre-stressed or the outer jacket extruded with the KEVLAR® under some tension to reduce the constructional stretch.

6.0 JACKET MATERIALS

The selection of the best jacket material for the application is as important to the life of the rope as is the internal construction.

An explanation of the relative merits and characteristics of jacket materials may be seen in Table 6. There are various formulations for each of these materials that alter their relative attributes. Consult the rope manufacturer for their cost, availability and specific attributes.

6.1 Braided

The braided jacket will, provide the most flexible of all jacket materials but will allow the rope to be degraded by dirt and ultraviolet light. Also, the abrasion resistance is less than that of the extruded jackets. The braid is easily removed for terminations such as eye splices and potted terminals.

6.2 Extruded Polyethylene

Polyethylene is the easiest to extrude and the least expensive of the jacket materials. It has excellent dielectric and ice shedding properties. It is more flexible than some other extruded materials.

6.3 Extruded Nylon

These are perhaps the most all around jacket materials, but make for a very stiff rope.

6.4 Extruded Polyurethane

Polyurethane provides a flexible rope with good dielectric isolation properties, abrasion resistance and, is the easiest to chemically bond electrical connectors. It is ideal for underwater ropes and electromechanical cables.

6.5 Extruded Hytrel®

Hytrel® can be formulated to have dielectric, abrasion, flexibility and ice shedding properties equal to or exceeding any other material. It is recommended for antenna guys where ice and high winds cause aeolian vibrations, which can fatigue polyethylene antenna guys. Hytrel® is, however, the most expensive of the extruded jackets.

TABLE 6 JACKETING MATERIALS

JACKET MATERIAL	ABRASION RESISTANCE	FLEXIBILITY	ELECTRICAL INSULATION RESISTANCE	AEOLEAN VIBRATION DAMPING	PROTECTION FROM UV	PROTECTION FROM HEAT	ICE SHEDDING	COLD FLOW	LOW WATER PERMEABILITY	RESISTANCE TO CUT THRU
BRAIDED	1	5	NONE	NONE	1	1	NONE	N/A	NONE	1
EXTRUDED NYLON (POLYAMIOS)	4	2	4	3	5		3	5	4	5
EXTRUDED HYTREL (POLYESTER)	5	3	5	5	5		5	5	5	4
EXTRUDED POLYURETHANE	3	3	5	5	5		3	5	5	4
EXTRUDED POLYETHYLENE	3	3	5	2	5		5	2	5	2
EXTRUDED TEFLON	3	3	5	2	5	5	5	2	6	2

6.6 Extruded Teflon®

Extruded Teflon® is used where the highest resistance to external heat is required. Neither Teflon®, nor polyethylene is recommended for use with terminations, which have an external compressive action, due to their poor cold flow properties.

7.0 TERMINATIONS FOR KEVLAR® ROPES

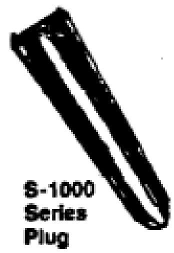
There are a multitude of considerations in the selection of the terminations for KEVLAR® ropes. The construction of the rope, in many cases, will be primary in the selection of the end fitting. These fittings are much more rope and jacket construction dependent than those of wire ropes. Wire rope terminations are usually not satisfactory for KEVLAR® ropes. KEVLAR® will withstand high tension and compression. However, the KEVLAR®, being very brittle in shear, will not tolerate the high shear forces at the tension/compression interface. Wire rope fittings depend on high compressive forces, which produce these high shear loads.

The most favorable termination for KEVLAR®, produces a low compressive load, over a much longer length, to reduce these shear forces. The relative attributes of these terminations can be seen in Table Il. The descriptions and assembly methods of these terminations are as follows.

7.1 Internal Plug or Wedge Terminals

The internal wedging terminal is easy to install in the field (Figures 5-30 and 5-31). The holding ability is somewhat dependent on the technique of the installer. It has not proven satisfactory for dynamic applications or when loads cycle thru zero. There is some damage to the KEVLAR® fibers during assembly. Installation on Naval Lifelines has shown that the KEVLAR® will creep out of the fitting after a period of time. These fittings may be acceptable for short-term applications. Lengthening the plug, as seen in the SEFAC and Linear Composite fittings, may increase the holding strength of these fittings.

Assembly Instructions



S-1000 Series Plug

ROPE SIZE	PLUG CAT. NO.	DIA. 'A'	ROPE SIZE	PLUG CAT. NO.	DIA. 'A'
1/8	SD-1012	1-3/16	5/8	SD-1062	3-3/16
3/16	SD-1018	1-1/2	3/4	SD-1075	4-1/2
1/4	SD-1025	1-9/16	7/8	SD-1087	5-1/8
5/16	SD-1031	1-7/8	1	SD-1099	5-1/2
3/8	SD-1037	1-15/16	1-1/8	SD-10112	7-5/8
7/16	SD-1043	2-3/16	1-1/4	SD-10125	9
1/2	SD-1050	2-1/2	1-3/8	SD-10137	10-1/2
9/16	SD-1056	2-13/16	1-1/2	SD-10150	11-1/2

Double Braided Construction

1. Insert the rope into the non-threaded end of the sleeve.

2. Place the sleeve into a vise.

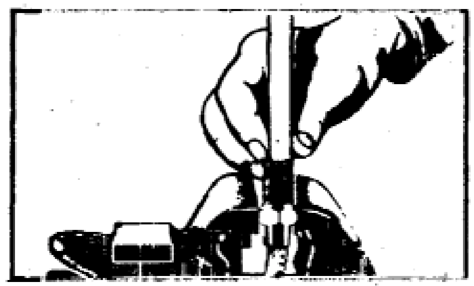
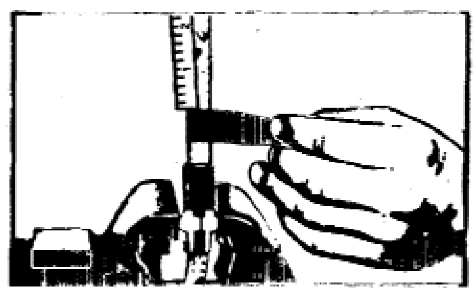
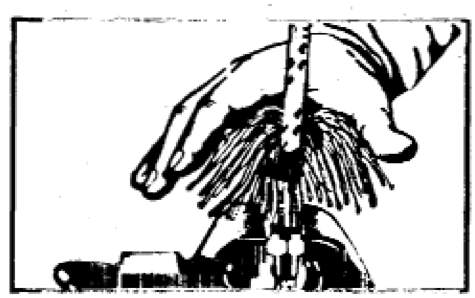


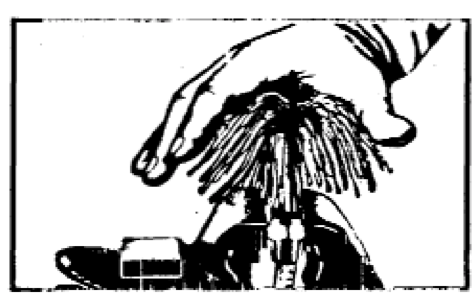
FIGURE 5-30

3. Use tape to seize the rope so that a section of rope that is equal to the 'A' dimension protrudes. (See chart for 'A' dimension of each plug.)





4. Unlay both the inner braids.



5. . . . and the outer braids.

NOTE: Before inserting plug into center of fibers pull a section of unlayed rope, equal to twice the diameter of the rope, into the sleeve.

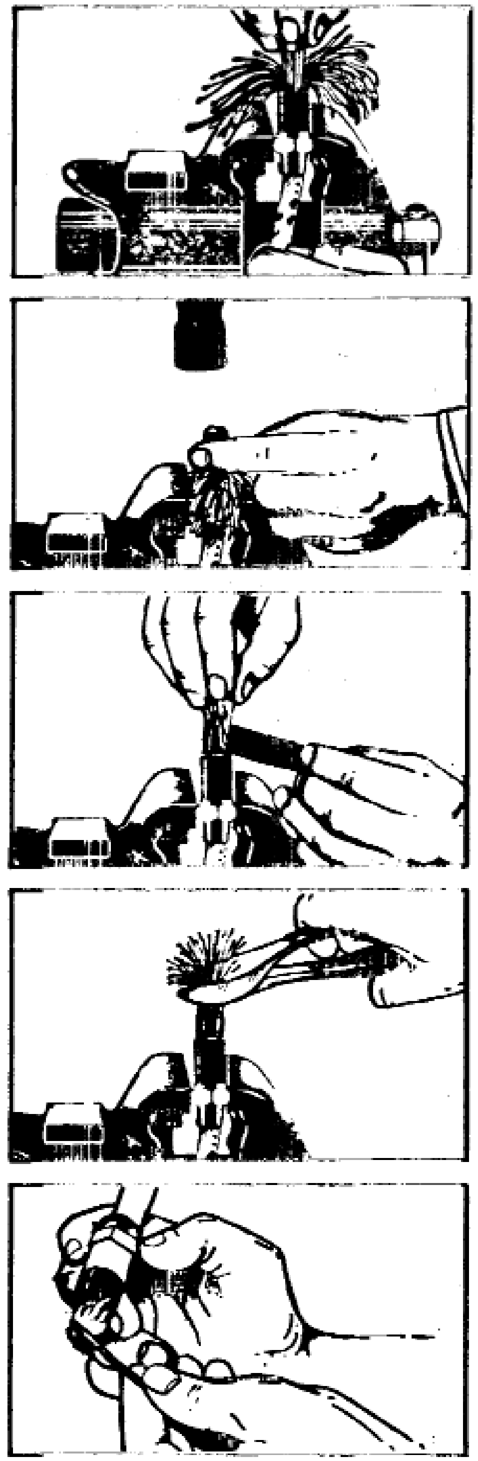


FIGURE 5-31

6. Insert the plug into the center* of the fibers, then pull the rope, with the plug inside, into the sleeve. *Plug must be exactly centered in order to achieve the best possible termination.

7. Make certain that all the fibers are uniformly distributed around the plug. Then use a hammer to tap the plug to a solid seat.

8. Make certain that all the fibers are uniformly distributed around the plug. Use tape to tightly seize all the fibers against the body of the protruding plug.

9. Trim excess fibers to within one-half inch of the end of the plug.

10. Use a torch, match or heat gun to fuse the protruding fibers together. **(IMPORTANT: Do not melt the fibers below top of plug.)** For proper assembly, diameter of fused rope must be less than the inside diameter of the fitting.

7.2 External Plug or Wedge Terminal

Electroline Termination for Jacketed KEVLAR® Rope

The Electroline mechanical termination has been developed for Aramid ropes with extruded jackets. The efficiency is 100% when applied to ropes of the following construction: thick jacket of medium hardness and materials not susceptible to cold flow under pressure. Less than 100% efficiency is obtained on extruded jackets of polyethylene and Teflon® and on ropes with a braided jacket between the aramid and the extruded jacket. This termination is also efficient on jacketed, aramid electro-mechanical cables. This termination is easily installed in the field and required no special tools or training. They are manufactured in aluminum, bronze and stainless and with various types of clevis, stud or eye adaptors. As shown in Figure 5-32 jacket removal is not required.

7.3 Chemical Potting

Chemical potting requires removal of the jacket. It is a high efficiency terminal but is not readily field installable. There is a possibility of fatigue failure, after long-term tension cycling, at the nose of the terminal. This is due to possible damage of the Kevlar on jacket removal and the weak transition from jacketed to unjacketed rope. For many rope constructions and applications, this is the only suitable method of terminating.

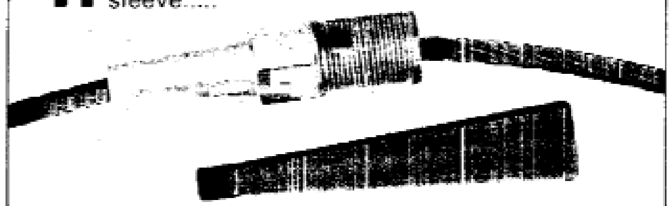
7.4 Eye Splicing

Eye splicing is making a termination from the rope itself, by looping the end of the rope and braiding the tail back onto the outside of the rope. This is a more efficient method of termination but requires some training. In many cases, it is done by the rope manufacturer or by the sling house. If done properly it will provide a full strength termination that will withstand dynamic loading.

Variations of the eye splice include: 1) the “breakout” splice where the end of the rope is looped and unbraided, then braided back into the main member of the rope; and (2) the “hollow braid” where the end of the rope is looped into an eye, run back and forth through the main member of the rope, then continued down the center of the rope.

FIGURE 5-32

1. Feed the rope through the sleeve.....



2. Expand the plug segment by hand to approximately 15°.....

3. Feed the rope through the plug. (be sure rope extends beyond the end of the plug a minimum of 2-times the rope diameter).....

4. Fit the plug into the sleeve and drive to a solid seat.....

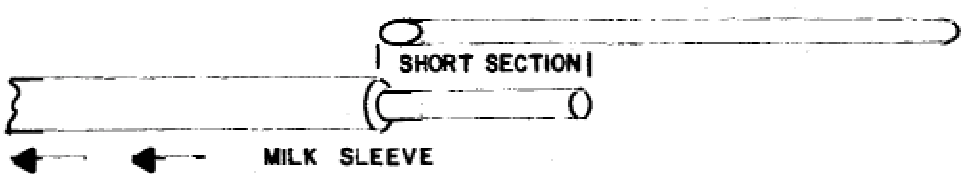
5. Thread sleeve into the socket of the clevis, eye or stud, tighten securely and the installation is complete!

There is a different technique for each construction of rope. Some typical methods of eye splicing are as follows:

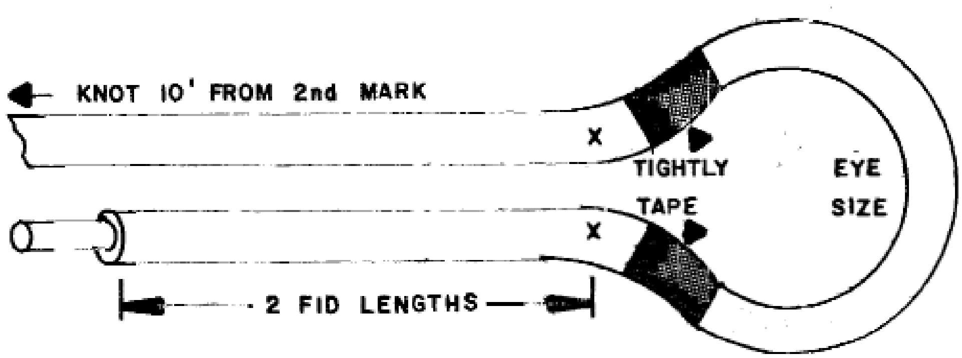
The following eye splice instructions are intended for experienced riggers familiar with conventional braid-on-braid splicing techniques. This splice method will allow the KEVLAR® core (primary strength member) of AraCom rope to pass around the eye inside the sleeve and down into the neck of the splice employing the Chinese finger grip principle. AraCom is difficult to splice. Yet the quality of the splice determines the usable strength of the rope.

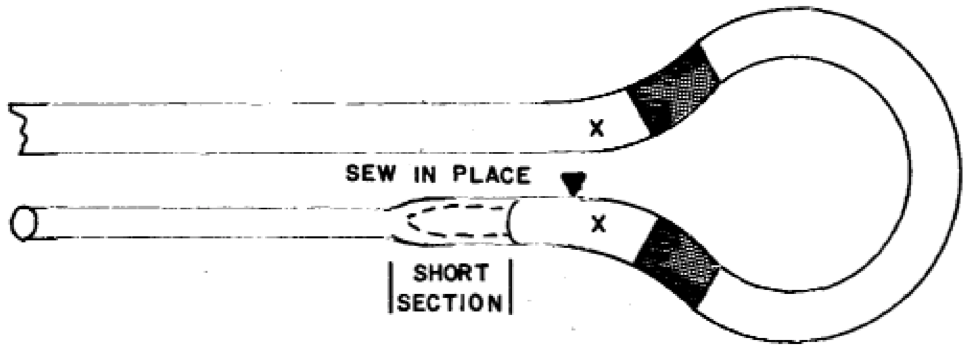
IMPORTANT: KEVLAR® is an extremely low elongation fiber demanding that close attention be paid to avoid twisting the braid or misaligning the fibers while splicing.

I. Pull core out from end of rope a distance of (1) short section. Hold end of sleeve down on core and milk excess sleeve back about five (5) feet.

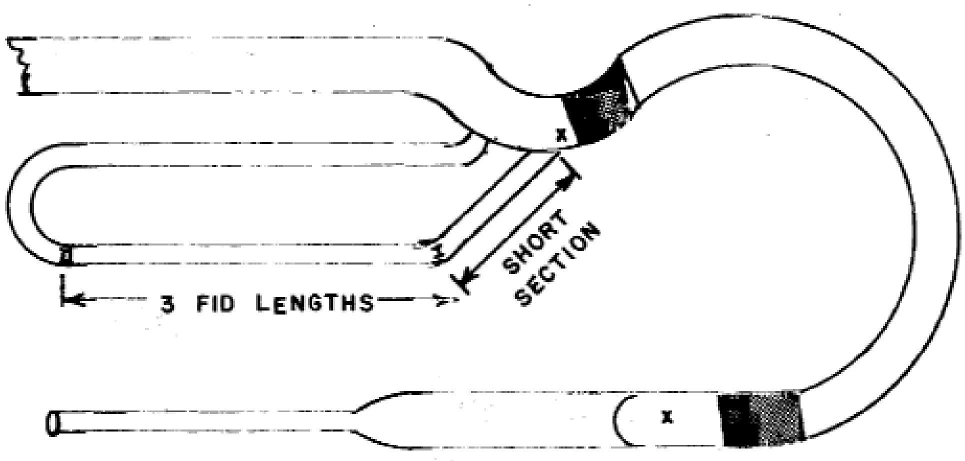


2. Measure two (2) fid lengths (approximately 40x diameter) from end of sleeve and mark sleeve. Form desired eye size and place Second Mark on sleeve opposite the First Mark. Lightly, tape sleeve on the eye side of these marks (the core and sleeve must not move in relation to each other between these marks). Tie a slipknot approximately ten (10) feet from Second Sleeve Mark.

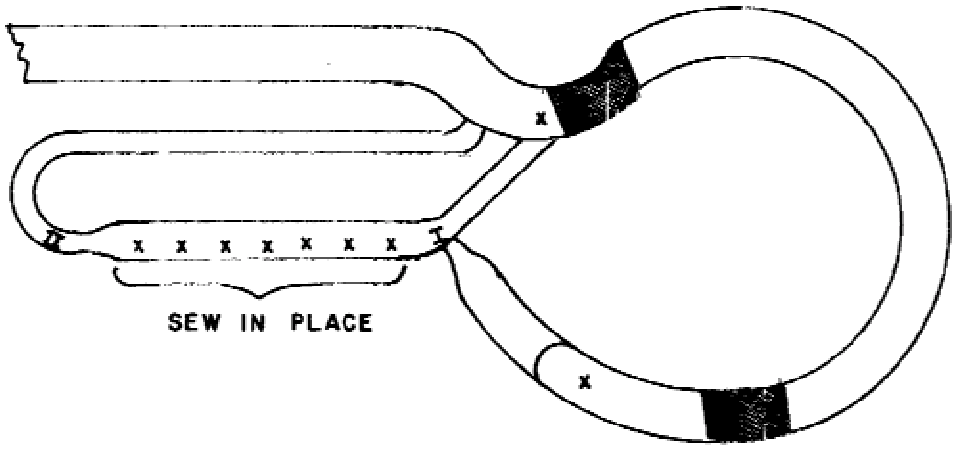




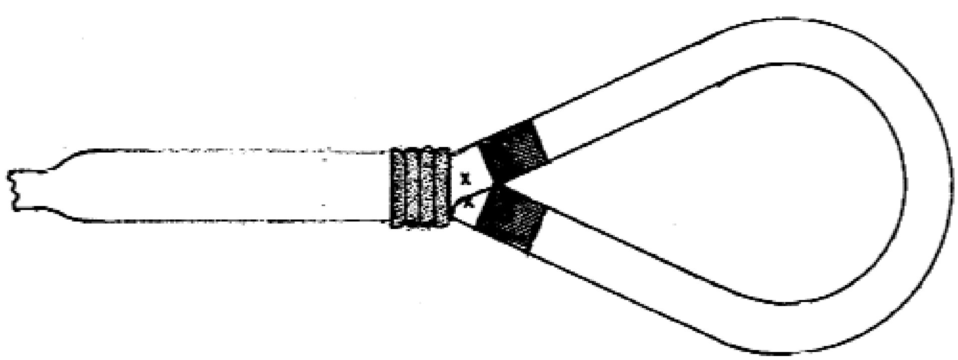
3. Extract core from sleeve at First Mark. Insert sleeve into core for a distance of one (1) short fid section from First Mark toward end of core. Sleeve end should be tapered and sewn in place.



4. Extract core from sleeve at Second Mark and pull out from direction of knot bunching sleeve toward knot. Place core Mark 1 a distance of one (1) short section from point of extraction and core Mark 2 a distance of three (3) lid lengths from core Mark 1.



5. Insert core tail into core at Mark 1 and bring out at Mark 2. Taper core tail. Align sleeve Marks I and 2. Smooth core from Mark 1 to Mark 2, allowing core tail to disappear at Mark 2. Sew from core Mark I to end of tapered core tail. DO NOT pull thread hard, distorting braid.



6. Smooth sleeve from knot toward eye allowing the core to feed back into the sleeve. DO NOT allow the core to roll or twist as it goes back into the sleeve. Repeat this process until sleeve Marks 1 and 2 are aligned and the eye is correct size.

7. Whip the throat.

7.5 Swaged Ferrules

The swaged ferrule is a quick method of terminating but will not produce an efficient terminal. The high compressive loads damage the KEVLAR®. This method is satisfactory for low static loads.

SIZE OF ROPE	NICOPRESS OVAL SLEEVES STOCK NO.	NICOPRESS HAND TOOLS STOCK NO.	- OVAL SLEEVES ~ LENGTH BEFORE COMPRESSION
1/16"	1700-C	51-C-887	3/8"
1/8"	1700-M	51-M-850	11/16"
3/16"	1582-P	51-P-850	1"
1/4"	1700-X	51-X-850	1-1/4"
NO. 635 HYDRAULIC TOOL DIES			
5/16"	1700-G3	1700-G3	1-1/8"
3/8"	1700-H5	1700-H5	1-9/16"
1/2"	1700-J8	1700-J8	2"

7.6 Line Pulling Grips

These grips are primarily for short-term applications such as line stringing. They are easy to field install and are reusable. They are not a full strength termination, but do no damage to the rope. A variety of grips exist and the majority are constructed similar to the Kellems® grip shown in Figures 5-33 and 5-34.

NON-CONDUCTIVE GRIP

FIGURE 5-33

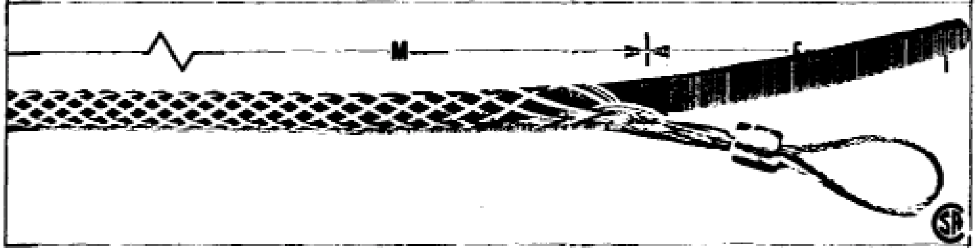
KELLEMS GRIP

Catalog Number	Description	Color Code	Cable Diameter Range -Inches-	Approx. Breaking Strength -Lbs.-	E (INCHES)	M (INCHES)	A (INCHES)
036-28-001	NCP 050	Green	.50- .62	4,000	5.5	24.0	.44
036-28-002	NCP 063	Yellow	.63- .74	5,000	5.5	26.0	.44
036-28-003	NCP 075	Red	.75- .99	6,000	6.0	31.0	.63
036-28-004	NCP 100	Blue	1.00-1.24	6,000	6.5	36.0	.63
036-28-005	NCP 125	White	1.25-1.49	6,000	6.7	41.5	.63

E – Eye length

M – Mesh length at nominal diameter

SLACK GRIP — CLOSED MESH



DOUBLE WEAVE

	Catalog Number	Description	Cable Diameter Range -Inches-	Approx. Breaking Strength -Lbs.-	E (Inches)	M (Inches)
STANDARD	033-08-003	UD075-A	.75- .99	2,600	7	12
	033-08-004	UD100-A	1.00-1.24	4,000	8	15
	033-08-005	UD125-A	1.25-1.49	5,400	8	16
	033-08-006	UD150-A	1.50-1.74	6,600	9	17
	033-08-007	UD175-A	1.75-1.99	10,000	10	18
	033-08-008	UD200-A	2.00-2.49	11,000	10	19
	033-08-009	UD250-A	2.50-2.99	11,000	10	20
	033-08-010	UD300-A	3.00-3.49	14,500	12	21
	033-08-011	UD350-A	3.50-3.99	14,500	12	22
LONG	033-08-012	UD075-M	.75- .99	2,600	7	20
	033-08-013	UD100-M	1.00-1.24	5,400	8	20
	033-08-014	UD125-M	1.25-1.49	5,400	8	23
	033-08-015	UD150-M	1.50-1.99	6,600	9	25
	033-08-016	UD200-M	2.00-2.49	11,000	10	26
	033-08-017	UD250-M	2.50-2.99	11,000	10	29
	033-08-018	UD300-M	3.00-3.49	16,000	12	32
	033-08-019	UD350-M	3.50-3.99	16,000	12	35

E – Eye length

M – Mesh length at nominal diameter

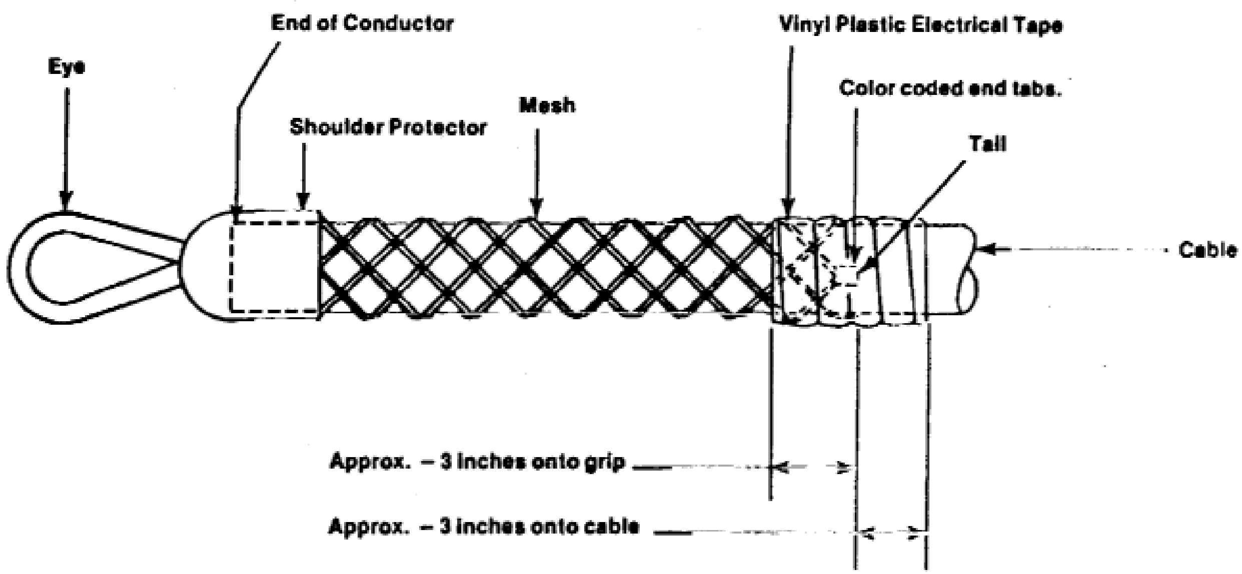
FIGURE 5-34 KELLEMS® GRIP

The following steps should be taken to assure proper assembly of Non-conductive Pulling Grips.

1. Assemble the grip so the cable is inserted into the molded shoulder protector.
2. All slack should be smoothed out of the mesh starting at the molded shoulder protector and working back toward the tail.
3. Apply vinyl plastic electrical tape starting three inches from the tail of the grip winding onto three inches of the cable. Continue back and forth, wrapping tape tightly, until two to three layers of tape have been applied. Taping is required to insure maximum reliability and guard against accidental release.

   Note: When higher loads are required, use Kellems® Multiple Strength or DUA-pull Grips.

   Note: DO NOT USE FOR PULLING ROPE. FOR PULLING ROPE, USE KELLEMS® DUA-pull GRIPS.



GRIP PART NUMBER	COLOR CODE	WORK LOAD (LBS)*	SIZE RANGE O.D. (INCHES)
036-28-001	Green	550	.50 — .62
036-28-002	Yellow	700	.63 — .74
036-28-003	Red	850	.75 — .99
036-28-004	Blue	850	1.00 — 1.24
036-28-005	White	850	1.25 — 1.49

*Rated working load is determined by using a safety factor of 5 based on approximate breaking strength.

ACKNOWLEDGMENTS

The author wishes to acknowledge the following companies who have provided material and information used in this section.

Cortland Cable Company
E. I. du Pont de Nemours and Company
Esmet Corporation (Electroline)
Hubbell Incorporated
Philadelphia Resins Corporation

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Kellums® is a trademark of Hubbell Corporation

Some useful links:

http://www.wireropenews.com

http://www.awrf.org

http://www.craneinstitute.com

http://www.MTSociety.org

http://www.ropecord.com

CHAPTER 6

WIRE ROPE AND E.M. CABLE LUBRICATION

Etienne Grignard

1.0 INTRODUCTION

One of the major causes of wire or cable deterioration and the reduction of its serviceable life is a lack of proper field lubrication techniques. The lack of proper re-lubrication of an operational wire is roughly equivalent to the purchase of a new automobile and then ignoring the need to add more oil as the vehicle is used; ultimately the engine lubricant is consumed and the moving metal parts begin to abrade each other until the engine fails. However, with proper re-lubrication, that engine, like a rope or cable, will last for an indefinite period of time. It is important to realize that any rope or cable is a complex mechanical system that operates in a hostile environment. Not only is it subject to both internal and external abrasion during use, but it is also prone to the corrosive effects of the sea water. Without proper re-lubrication techniques in the field, both the abrasion of the strands and corrosive effect of the sea water combine to reduce both the wire's load-carrying capacity and serviceable life.

Corrective Maintenance Vs Preventative Maintenance.
Questions to determine:

Corrective Maintenance:

- Cost to replace is relatively low
- Product is easily and readily available
- Product is replaced utilizing minimal man hours without significant shut downs
- Cost to replace is less expensive that cost to maintain
- Man hours to replace is less than man hours to maintain
- Safety is not jeopardized by lack of maintenance
- A replacement schedule is enforced
- Company can not commit to a follow a preventative maintenance program
- Maintenance materials distort research findings

Preventive Maintenance:

- Cost to replace is (relatively) high
- Product is not readily available
- Product replacement utilizes excessive man hours
- Product replacement stops production / research
- Cost to maintain is less expensive that cost to replace / install
- Man hours to maintain is less than man hours to install
- Safety may be jeopardized by lack of inspection
- Maintenance materials do not distort research findings
- Company commitment is to make preventative maintenance programs work

2.O WIRE LUBRICATION THEORY

The lubricants applied to working ropes and cables provide a dual form of protection in that individual wires are protected from one another and the whole wire is preserved against the corrosive action of sea water. In order to understand the importance of wire and cable lubrication, it is necessary to realize that a wire, when in use, is a dynamically complex mechanical tool which is composed of numerous moving parts. As the wire passes over the sheave train, it is subjected to corrosion, bending, tension, and compressional stresses as it attempts to equalize the effects of the load it is carrying. The lubricant added to the wire during manufacturing permits this equalization to occur with a minimum of abrasion to the individual wires within each strand.

2.1 Wire Re-lubrication

The reapplication of a lubricant in the field cannot be stressed too strongly since it ensures that the friction between individual wires is reduced to minimum; remember, each wire within a rope or cable is in constant contact with other wire along its entire length. If the user neglects to follow a program of field lubrication, the manufacturer supplied lubricant is soon dissipated and direct metal to metal contact established between the individual wires of the rope. As the “dry” rope is used, the abrading of the individual wires reduces their metallic area and subsequently, the total load carrying capability of the wire itself. The effects of operating a dry rope versus a lubricated rope are best illustrated in the following chart taken from the Roebling Wire Rope Handbook which shows the results of cyclic testing of non-lubricated versus lubricated wire rope.

	10" Tread Dia Sheave Sheave/Rope Dia Ratio = 18	24" Tread Dia Sheave Sheave/Rope Dia Ratio = 43
Dry Rope	16,000 Bends	74,000 Bends
Lubricated	38,700 Bends	386,000 Bends

During this test series a 9/16" dia 6 x19 wire rope was used. The results indicated that based on sheave size and lubrication, a lubricated wire will operate 2.4 times as long as a dry rope in the case of the 10" tread dia and 5.2 times as long when the tread dia is increased to 24". The results of such tests clearly indicate the benefits and the need for continued field lubrication of working wires and cables, as well as the importance of utilizing properly sized sheaves. The success of the 24" diameter sheave is due to the reduction in wire stresses due to the larger contact area the wire is exposed to and the reduction in individual wire movement within the rope.

Definition of Corrosion

"Corrosion is the deterioration of a substance (usually a metal) or its properties because of a reaction with its environment."

Basic Forms of Corrosion

There are many forms of corrosion. Some are frequently encountered in everyday life and on the job. Others require specific combinations of materials and environments that are rarely encountered. Gaining an understanding of the mechanisms at work in these forms of corrosion and how the mechanism results in the specific forms of corrosion is an important first step in controlling corrosion. What we understand, we can more easily control.

In this course, we will find that all forms of corrosion, with the exception of some forms of high-temperature corrosion, occur through th action of the electrochemical cell. We will find that this electrochemical cell can act in many ways, but that it's general principles, once understood, can be applied to the understanding of most forms of corrosion. We will find that many of the methods that are used to control corrosion involve intentional interruption of the action of the electrochemical cells responsible for corrosion

2.2 Corrosion Protection

Corrosion Occurs Through Electrochemical Reactions

- Except for high temperature corrosion, all corrosion reactions are electromechanical reactions occurring in electrolyte. –NACE

- Electrochemical reactions:

Occur in electrolytes, which are liquids that can carry and electrical current

Occur through the exchange of electrons

-The exchange of electrons in electrochemical reactions occur at separate sites

-The electrons flow through the metal from one of these separate sites to another

- Electrolyte:

  An electrolyte is a liquid that contains ions. An electrolyte can conduct electricity through the flow of ions. Anions flow towards the anode, cations flow towards the cathode.

Protecting a wire from the corrosive effects of the salt water environment may well be the most important aspect of increased wire life. As discussed, the wire is usually delivered with some type of lubricant already in place on the wire, which acts as a corrosion prevention device as well as a lubricant. If renewed in the field, a rust preventative/lubricant can extend the useful life of the rope or cable by as much as five times the working life currently being experienced.

The full effect of corrosion damage to an unlubricated wire is virtually impossible to assess fully due to the complexity of the problem. Simultaneously a corroding wire is affected by a loss of metallic area in the individual wires due to chemical and electrochemical action on the bare steel, the bare metallic contact areas between wires are pitted and toughened causing an uneven surface which forms stress points in individual wires, and finally the corroded contact surface inhibits the normal, smooth movement of the wires relative to one another generating high stress concentrations, speeding corrosion fatigue and crack propagation. All of these factors operate internally within the wire and may not be visible during a casual inspection other than the presence of surface or leaking rust.

2.3 Splash Zone Corrosion

The marine environment represents what is perhaps the most hostile climate a wire or cable may be required to operate in during its working life. At sea the wire is alternately subjected to the corrosive effects of the marine atmosphere and short immersions in seawater during lowerings. Combine this with frequent exposure to salt spray blowing aboard and a set of circumstances equivalent to splash zone conditions is produced.

When the wire in use is allowed to lose its factory-applied lubrication through a lack of re-lubrication at sea, it quickly loses its protective coating and becomes subject to the full corrosive effects of its environment. Figure 6-1 represents the typical corrosion rates which can be expected of bare steel in the marine environment. Based on the data presented in this figure, it is obvious that metals used in marine atmospheric conditions are subject to the highest rate of corrosion and therefore, require proper field lubrication techniques to be practiced if maximum wire life is to be achieved.

2.4 Marine Atmospheric Corrosion

Of secondary importance to the possible mechanisms for at-sea wire degradation are the affects of the marine atmosphere on unprotected steel. Based on the data presented in Figure 6-1, atmospheric corrosion has a lesser affect upon a wire than one in the splash zone. Nonetheless, its continuing effects will be realized over time. Given the circumstances of a reel of wire stored on a vessel without adequate protection or a covering, it can be expected that at least the upper layers of the wire will be rendered useless in a relatively short period of time.

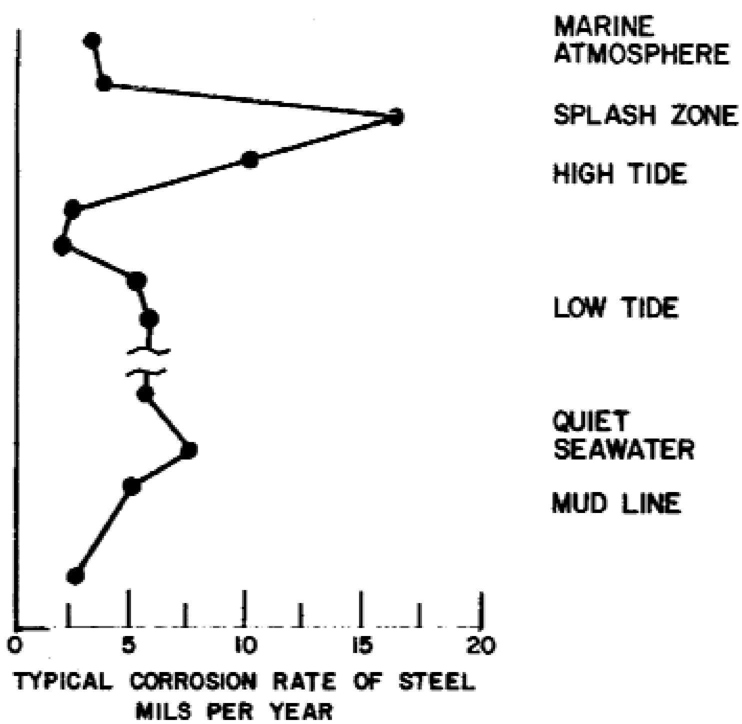
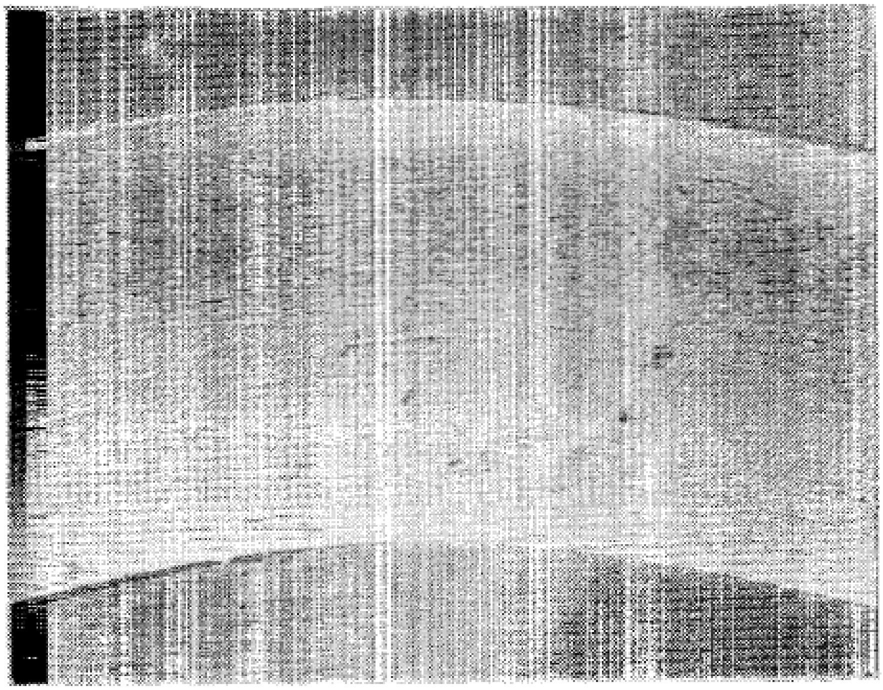
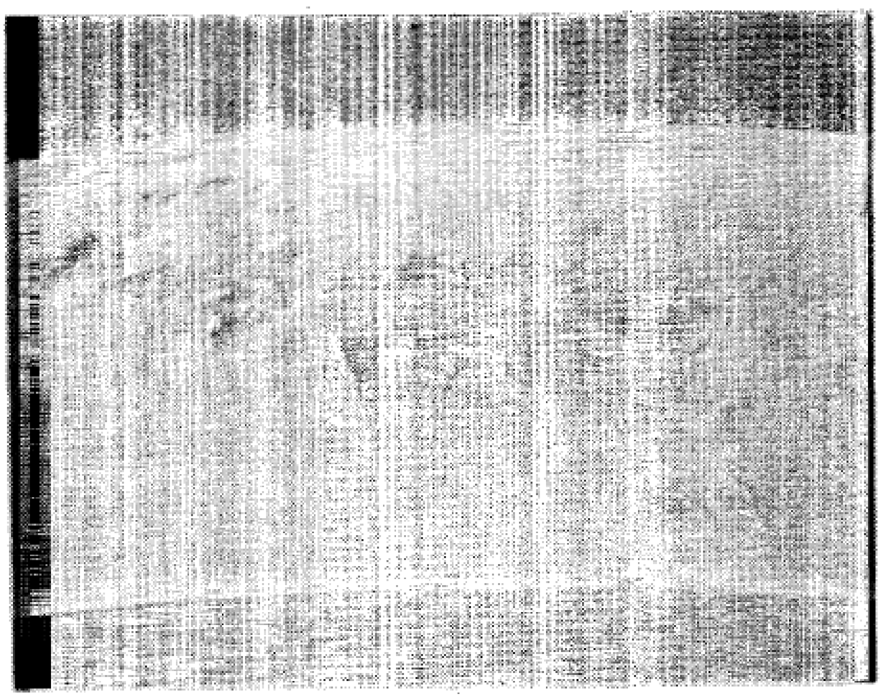


FIGURE 6-1

One prime example of the affects of atmospheric corrosion resulted from an experiment conducted by U S Steel of Trenton, NJ in 1969. In this experiment samples of unprotected steel were exposed to the salt air environment at distances up to 800 feet from the water edge. Although corrosion was present on all samples, it was determined that samples 80 feet from the water corroded at a rate of 10% to 15% faster than the 800 foot sample. Translate this to a research vessel whose wires are never more than 20 feet from the water and this potential for rapid atmospheric corrosion is obvious.



WIRE PROTECTED WITH LUBRICANT/RUST PREVENTATIVE
PHOTOGRAPH 1 (100x)



UNPROTECTED WIRE
PHOTOGRAPH 2 (100x)

2.5 Corrosion During Wire Immersion

The corrosive effects of sea water on wires that are immersed for long periods of time, such as buoy moorings, etc., has been an area of interest for some time. It has been observed that the rates of corrosion vary from location to location in the ocean when the wire in use is unprotected by either a lubricant or a rust preventive. Accelerated rates of corrosion have been observed, which were ultimately traced to a combined effect of chemical levels in the water and the temperature of the seawater.

This problem was fist addressed by the Grignard Chemical Company in 1969 when advanced states of corrosion were first noticed on bright steel samples from the Antigua area. With the assistance of the Woods Hole Oceanographic Institution, Oregon State University, the Halan Company and U.S. Steel, a series of experiments were conducted to determine the causes of the accelerated corrosion rate observed in the Antigua sample. Test specimens of bright steel wire, galvanized wire and electromechanical cable, supplied by U.S. Steel, were submerged in sea water for half their length at three widely spaced locations, i.e., Massachusetts, Oregon, and Antigua, W.I. for a period of three months. At the conclusion of the test, all specimens were removed from the sea water and the corrosion level present in each wire was evaluated.

The analysis performed jointly by the Woods Hole Oceanographic Institution and Grignard Chemical Company indicated that water temperature was the prime factors involved in the accelerated corrosion rate that had been observed. Water analysis from the three test sites were as follows.

Location	Solids (Sodium Chloride)	Temperature
Oregon	3.71% Total Solids	44^0 F
Massachusetts	3.61% Total Solids	40^0 F
Antigua	3.98% Total Solids	84^0 F

3.0 WIRE LUBRICANTS AND CORROSION PREVENTION

Essentially the function of any wire dressing is two-fold in that it must act as a lubricant between individual wires to prevent premature wear and it must also prevent corrosion of the wire in the long term. Of these two functions, corrosion prevention is probably the most important as more wires fail or are discarded due to the effects of corrosion than service wear. It is a sad but true statement that oceanographic wires tend to rust out before they wear out.

Corrosion is basically a chloride reaction process where rate is increased by temperature, i.e., sodium chloride, and pollutants in the air and water. Corrosion of an unmaintained wire or cable is not restricted to only its outer surface, but instead, attacks all of the wires individually. The result is a steady reduction of the metallic area of each wire and the susceptibility of the wire to corrosion fatigue during bending over a sheave.

A corroded surface, Photograph 2, is made up of a myriad of microscopic pits and craters that under loading establish failure planes in the wire strands. The bending stresses involved in working wires are a factor in fatigue failure of a wire, but when this condition is combined with active corrosion, the failure point of the wire becomes totally unpredictable. The longevity of any wire or cable can be substantially lengthened through a program of re-lubrication in the field and in the specification of wire dressings at the time of manufacture.

3.1 Incorporation of Rust Preventative During Manufacture

Present day manufacturing practices of applying either an asphaltic or petrolatum compound to the wire is not wholly satisfactory. Given current concerns over the longevity of ropes and cables in oceanographic service, perhaps the time is right to address new techniques for these ropes and cables at the time of their production and to develop testing procedures for the ropes. Even though most ropes and cables used in oceanographic applications are purchased on a competitive price basis, it would seem, that the additional cost of including an effective rust preventative would be outweighed by the greater length of service that would be obtained.

Desired Characteristics:

- Water Displacement
- Corrosion Protection
- Penetrate All Voids of the Cable
- Lubrication
- Regulatory Compliance(Environmentally Safe/Non Toxic)
- Easy to Apply ***
- Can Not Camouflage the Cable (McDermott)

3.2 Testing

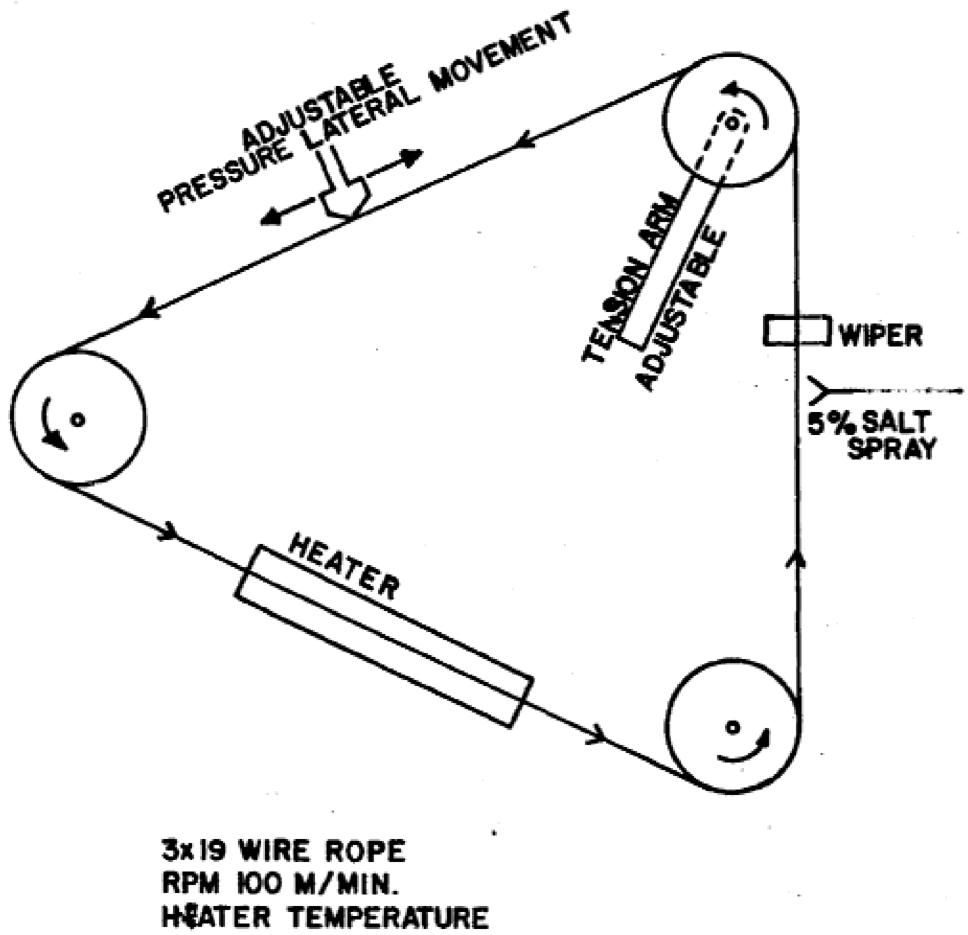
In addition, it would appear that a standard corrosion test for oceanographic ropes and cables should be developed. The current ASTM Salt Spray Panel Test is of little value where ropes and cables are concerned. In effect, the ASTM test achieves its results from monitoring the corrosion of a single steel panel in a static condition. The results of these tests confirm only that a static, high viscosity compound such as cosmoline or a shellac based product will provide good protection form corrosion. Since the hard surfaced rust preventatives are subject to cracking and flaking when the specimen is flexed, it is impossible to compare them or the ASTM test to a wire rope that is bent, stressed, and cavities during use.

Figure 6.2 illustrates a potential test bed for oceanographic ropes and cables that could be developed to assess not only the lubricating value of compounds, but also rust preventatives. In this diagram the wire would be subjected to equivalent corrosive conditions and stresses that are present in the field. This or a similar test method would certainly provide more relevant data than the simplified panel test.

3.3 Rust Preventatives

The largest problem in the effective applications of rust preventatives is being able to successfully coat the inner wire of a rope or cable. In the manufacture of electro-mechanical cables a cotton or synthetic braid is often used between the insulated conductor and the armor which provides an ideal reservoir for a medium viscosity rust preventative. Although the practice of incorporating a preventative is not, at present, commonly practiced by cable manufacturers, it is a concept worthy of evaluation.

Inspections of new electro-mechanical cable that had been coated with a rust preventative after the armor was installed revealed no penetration of the preventative to the cotton braid. In use, the type of cable construction is subject to water impregnation of the braid and progressive deterioration. In a section of cable, only one and a half years old, inspection revealed that the inner armor wires were badly corroded and the braid disintegrated.



PROPOSED LUBRICATION AND CORROSION TEST FOR WIRE ROPE

FIGURE 6-2

The use of an unlubricated or untreated cotton or synthetic braid can result in abrasion between the braid and the inner armor wires. In a report by the Roebling Wire Rope Company it was stated that if during manufacture the amount of lubricant is reduced too low, or is non-existent, an abrasive action between the dry fibers and the armor wires occurs as the cable is worked. This results in a form of fretting corrosion which wears extremely fine particles of steel from the armor wires which oxidize rapidly and can usually be noticed by wire discoloration. This process also results in the reduction of the ultimate strength of the cable over time.

It should be realized that a requirement, by the user, for the addition of a lubricated cotton or synthetic braid would raise the final cost of the cable by some percentage. However, this author feels that if an adequate lubricant/rust preventative is incorporated in the braid, it would be possible to extend the working life of the cable by as much as a factor of five (5) over present experience. Given this type of cable longevity, the additional cost of a lubricated braid is truly insignificant.

CABLE LIFE LONGEVITY CAN INCREASE WITH A PROPER COATING

EXAMPLE: EM Cable Lube #2 manufactured by Grignard Company is a transparent material with excellent penetrating abilities (does not setup, dry or form a tacky film). EM Cable Lube #2 provides twice the lubricity of petroleum based materials, compatible with synthetic and natural rubber, does not readily pick up dust and grime.

ADVANTAGES:

- Environmentally Safe
- 100% Synthetic Base
- Displaces Moisture/Remains Stable
- Effective in Sub-Zero Temperatures

TECHNICAL DATA:

Property	Test Method	Typical Value
Appearance	Visual	Clear
API Gravity	ASTM D 287-39	25.9
Density	ASTM D 1217	7.486
Flash Point	ASTM D-92-52	350^{o} F
Fire Point	ASTM D-92-52	420^{0} F
Viscosity 100^{0}F	ASTM D-4127	390-410 SUS
Four Ball Ware	ASTM D-4127	.31 mm
Water Displacement & Stability	FTM 3007	Pass

REMEMBER: The use of solvent cutbacks should be avoided due to their low flash points and high evaporation rates which result in only partial penetration of the wire leaving the base strands open to corrosion.

4.0 COMPATIBILITY OF LUBRICANT AND RUST PREVENTATIVES

With the identified need to re-lubricate ropes and cables, it is appropriate to mention a few items that are of importance in achieving this process. Principally, the compatibility of the selected lubricant/rust preventative with the wire dressing provided by the manufacturer. Problems that can result from the use of incompatible materials include partial penetration of the wire, a leaching out of the components of the original compound or flaking area of the reapplied dressing due to exposure to ultraviolet light. Since these effects are not readily apparent, careful selection and evaluation of the product to be used is advised.

4.1 Selection of a Field Dressing

The solvents that are commonly used in field dressing include petroleum distillates, chlorinated hydrocarbons, diesters, glycols, and alcohols. The major problem with inexpensive petroleum solvents is their low flash point which presents a serious fire hazard when used in confined spaces or in areas where motor sparking, etc., is likely to occur. Prior to the selection of field dressing, it is advisable to contact the manufacturer and explain the intended use of the product. The fact that it is an oceanographic cable and not an elevator hoist, machinery, or automotive application may be of prime importance in obtaining an effective field dressing.

Oceanographic cable manufactured or dressed with a lubricant should not form a sheen or be iridescent on the water.

The manufacturers of available field dressings are numerous and it would be impractical to list them fully in this chapter. *The wire rope compound used in manufacture and dressing must be inhibited to meet the Water Displacement and* Stability Test FTM 3007. A product developed by Grignard Company, EM CableLube #2 properties can be used as a guide line when selecting a cable lubricant.

4.2 Galvanizing of Ropes and Cables

It has been a standard practice in oceanographic ropes and cables to galvanize the wires prior to laying them up in final form. Zinc, used in the galvanizing process, has good resistance to seawater and usually corrodes at a rate of about 1/1000 of an inch per year, as long as the zinc cladding is intact. However, in a working oceanographic wire internal abrasion and sheave wear can rapidly reduce or pierce the zinc on the wire.

Once the galvanized surface is broken and “white rust” is seen to form, it is indicative that degradation process has begun. In this case, the two dissimilar metals, zinc and steel, are acting against each other in the common electrolyte formed by the seawater causing an electrical flow between the two metals. One metal, the zinc coating, will become the anode while the steel act as the cathode resulting in a steady deterioration of the galvanized coating.

It is important to remember that when the “white rust,” zinc oxide, is seen, there is an abrading action taking place in the wire which, if left alone will result in the piercing of the galvanized coating.

The use of a lubricant/rust preventative on galvanized wire will provide the lubrication necessary to eliminate any oxide formation resulting in longer service life of the wire.

5.0 FIELD APPLICATION OF LUBRICANTS/RUST PREVENTATIVES

Considering the hostile environment in which oceanographic ropes and cables are expected to perform, it is obvious that a program should be developed to perform regularly scheduled re-lubrication of wires in service. Given the current and escalating costs of ropes and cables, the small investment in time and materials required to re-lubricate a wire are certainly outweighed by the greater wire life which can be obtained. The majority of major wire producers recommend a re-lubrication procedure and fully realize the benefits to the users in the reliability of their product.

Due to the variety of conditions and use rates that exist between oceanographic organizations, it is impossible to set down a firm schedule for re-lubrication of working cables. The frequency of this procedure is best determined by the individual user, based on his own experience and on the wire documentation he maintains, At a minimum, the wire should be dressed before being placed in storage ashore or every six months where actual use is on a limited basis.

For organizations with high use rates, this redressing should probably occur at two month intervals and should be applied beginning at the greatest deployment length occurring during this period. Again, the wire documentation will prove highly useful in determining where the redressing should begin. One additional consideration would be a redressing of the entire wire length prior to long periods of disuse aboard the vessel and the covering of the winch reel, where exposed to the elements, with a tarp or similar covering.

What should be remembered is that the wire or cable is an investment upon which many people depend to perform their work. Without proper care and maintenance the wire rapidly becomes unreliable and failures are subject to occur with little or no warning.

6.0 FIELD APPLICATION TECHNIQUES

In the use of field dressings it is important to remember that the majority of products on the market have a specific gravity of less than one. What this translates to in reality is that the dressing will replace moisture in and on the wire, but it will have no effect on entrained water. This brings us to the first concern in the field application of a wire dressing; the removal of excess seawater.

6.1 Cable Drying

The primary concern in preparing a rope or cable to receive a field dressing compound is that it be free from entrained water. This excess water can be removed by using a series of spaced flexible wipers constructed of either rubber, Teflon or leather. It is inadvisable to use rags or other porous materials as a wiper since they quickly saturate and become ineffective. Their purpose is to scrape off the entrained water that is carried along by the moving wire. Due to the construction of 3 x 19 wire rope, a larger number of wipes are required to remove the excess water than are needed with a more concentric electro-mechanical cable.

The placement of any wiping device is relatively crucial to the success of the re-lubrication process. Conceptually, the wiper should be located at a point midway between the outboard sheave and the point at which the field dressing is applied. In this way some of the entrained water will naturally be shed by the wire due to vibration before reaching the wiper and the wire will have a brief opportunity to air dry before being dressed.

Under ideal circumstances it would be preferable to incorporate a compressed air dryer immediately after the wiper to further reduce the moisture level of the wire. It is realized, however, that some vessels have limitations on the availability of service air supplies which can preclude this approach. This lack of an air supply can be partially overcome by reducing the speed at which the wire is recovered during a re-lubrication operation.

6.2 Field Dressing Application

As stated in the first edition of the Handbook of Oceanography Winch, Wire and Cable Technology, extensive work has been done on the investigation of preparing the rope prior to dressing it.

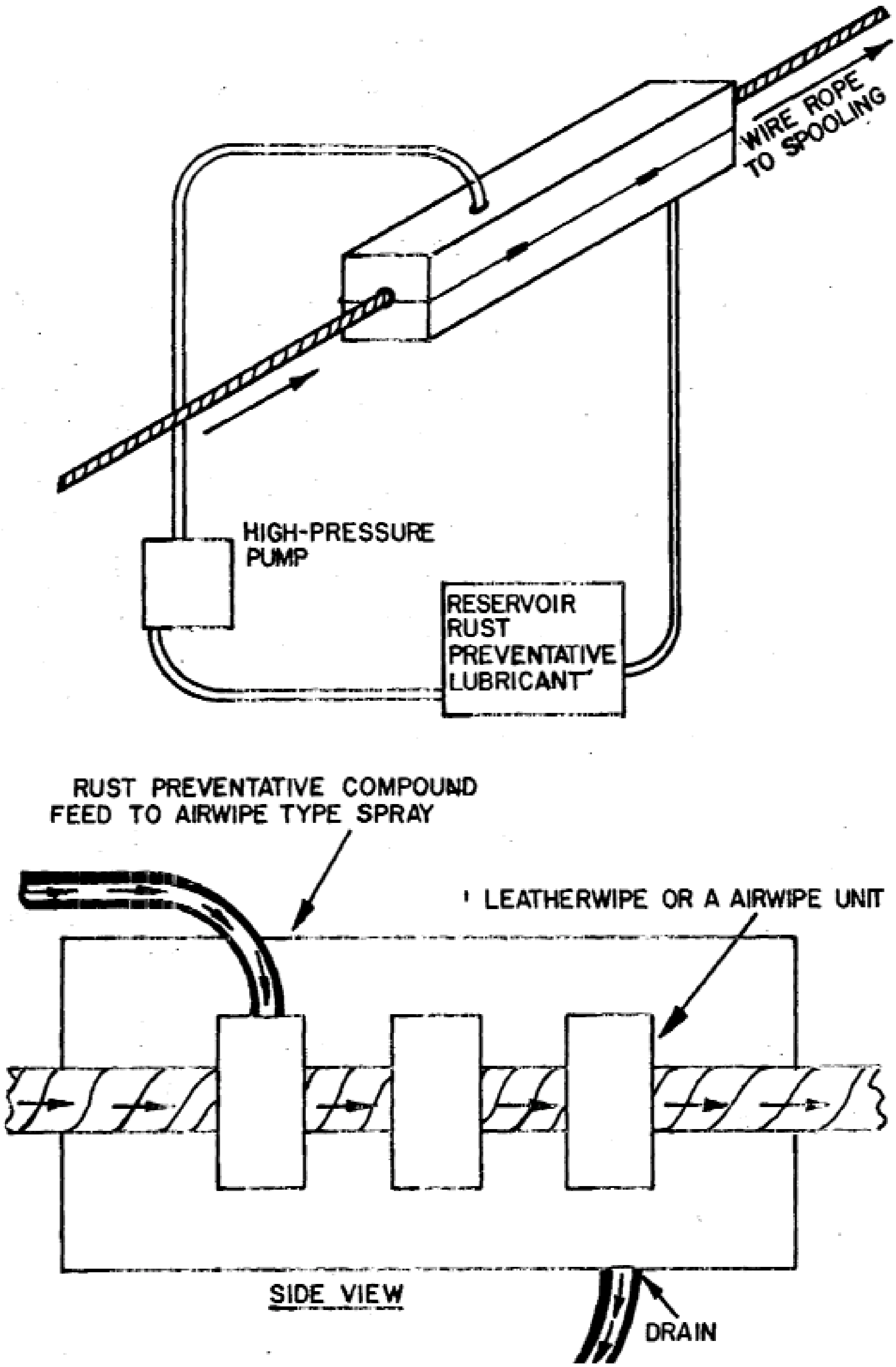
Four objectives must be obtained or the protective coating is worthless.

(1) Cleaning the rope of scale, dirt, salts, oxides and etc.
(2) High pressure water wash.
(3) Air Dry the rope of any water prior to dressing it in the lubricator.
(4) A light weight lubricator that will permit the lubricant to penetrate the wires in the strand and allow little leakage.

The first objective as shown in Figure 6-3 is the most important because if the valleys between the wires are not cleaned of solid debris and old wire rope compound, the dressing material even under pressure will not penetrate and if it does it can carry the abrasive debris between the wires an shorten their life by wear. Many lubricators are being marketed but no mention is being made for preparing used dirty rope prior to its lubrication or dressing. Investigation unveiled a wire rope cleaning unit for ski lift lines than can be adapted to small oceanographic 3 x 19 or 6 x 12 mechanical cable.

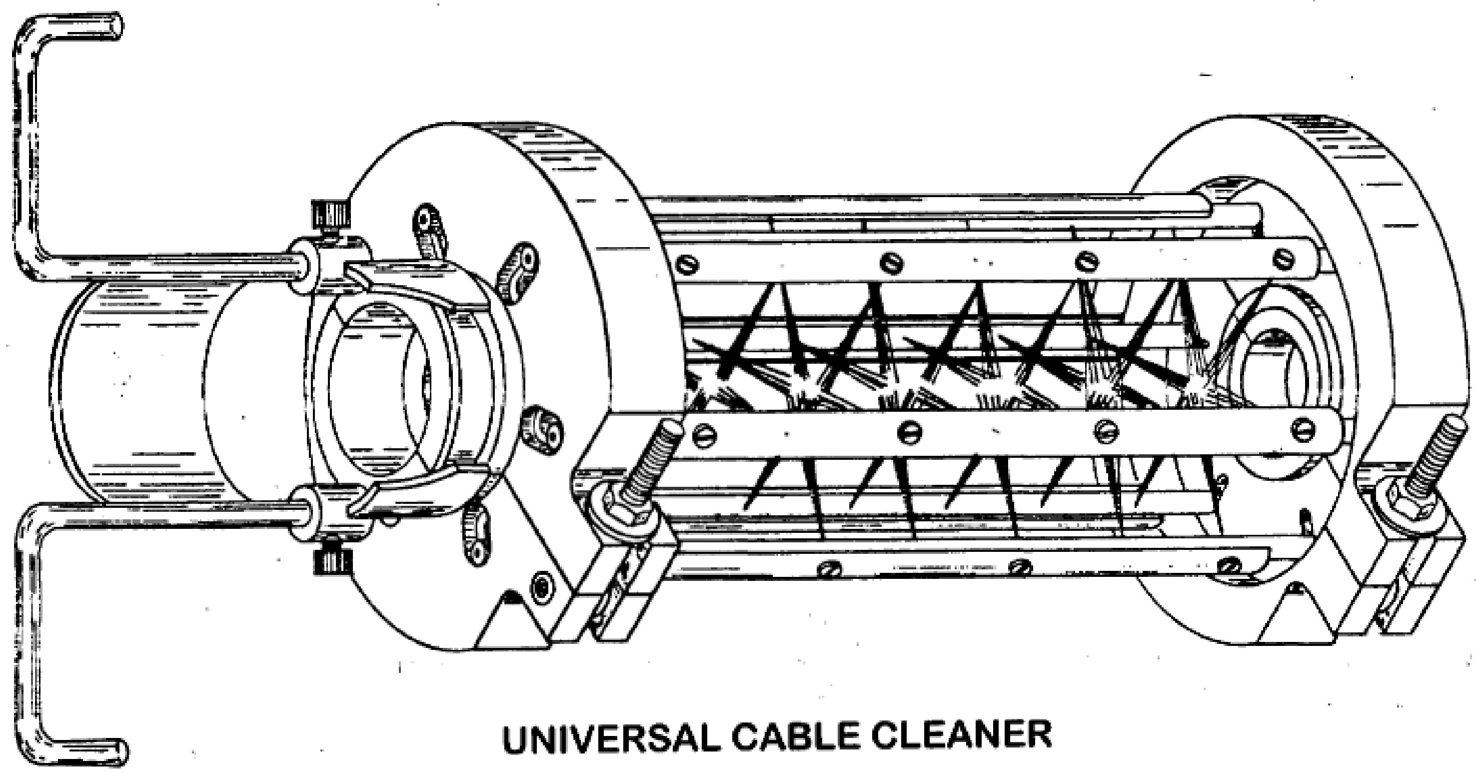
As shown in Figure 6-4, this unit is made of wire brushes that revolve with the lay of the rope as it passes through it. The worn steel wire brushes can be easily replaced in minutes and the same unit can be used for different size ropes.

The spiral wire brush unit designed by Brooke Ocean Technology shown in Figure 6-5 is excellent for cleaning the surfaces of electromechanical or strand cable.



CONCEPTUAL AUTOMATIC SPRAY UNIT

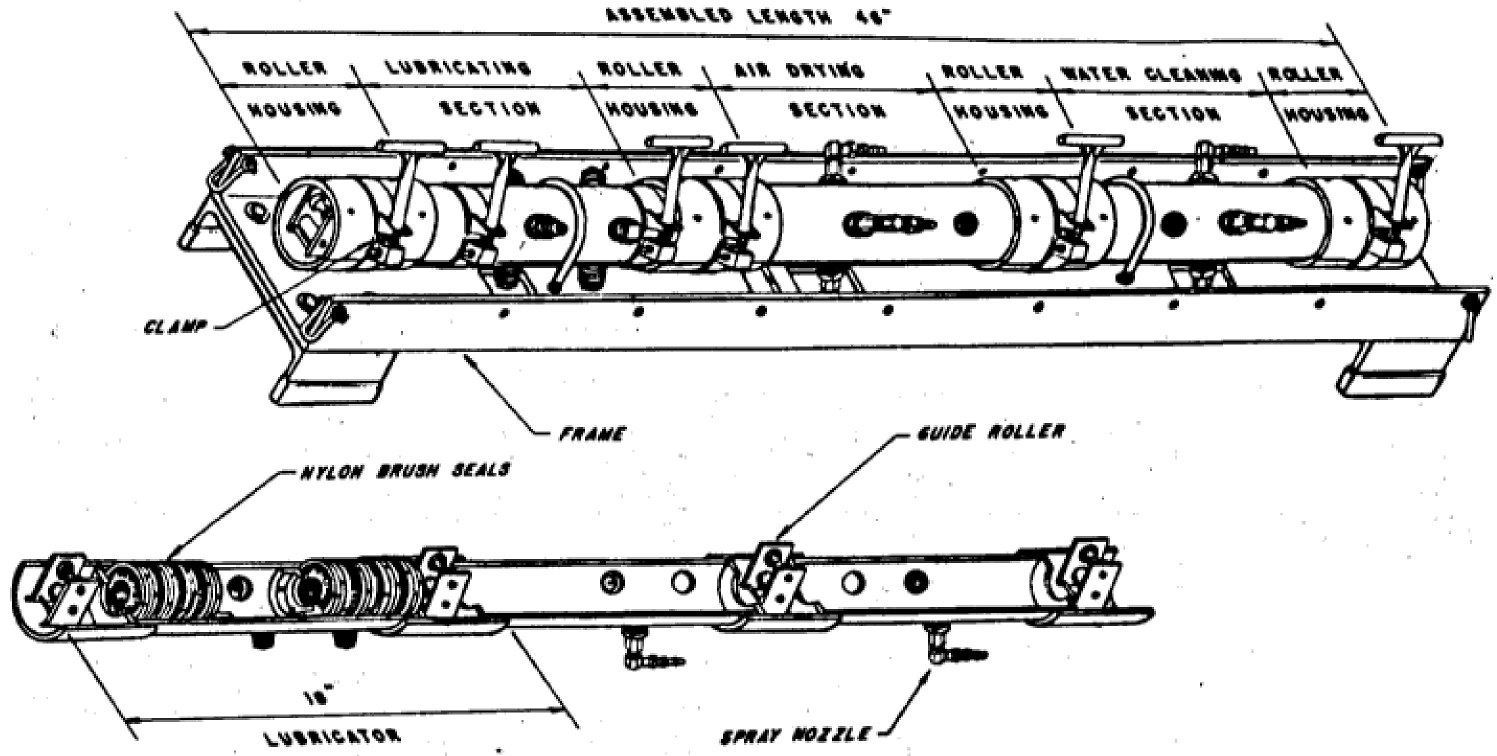
FIGURE 6-3



UNIVERSAL CABLE CLEANER

FIGURE 6-4

Courtesy --- Grignard Company



WIRE ROPE CLEANER LUBRICATOR

COURTESY BROOKE OCEAN TECHNOLOGY, LTD.

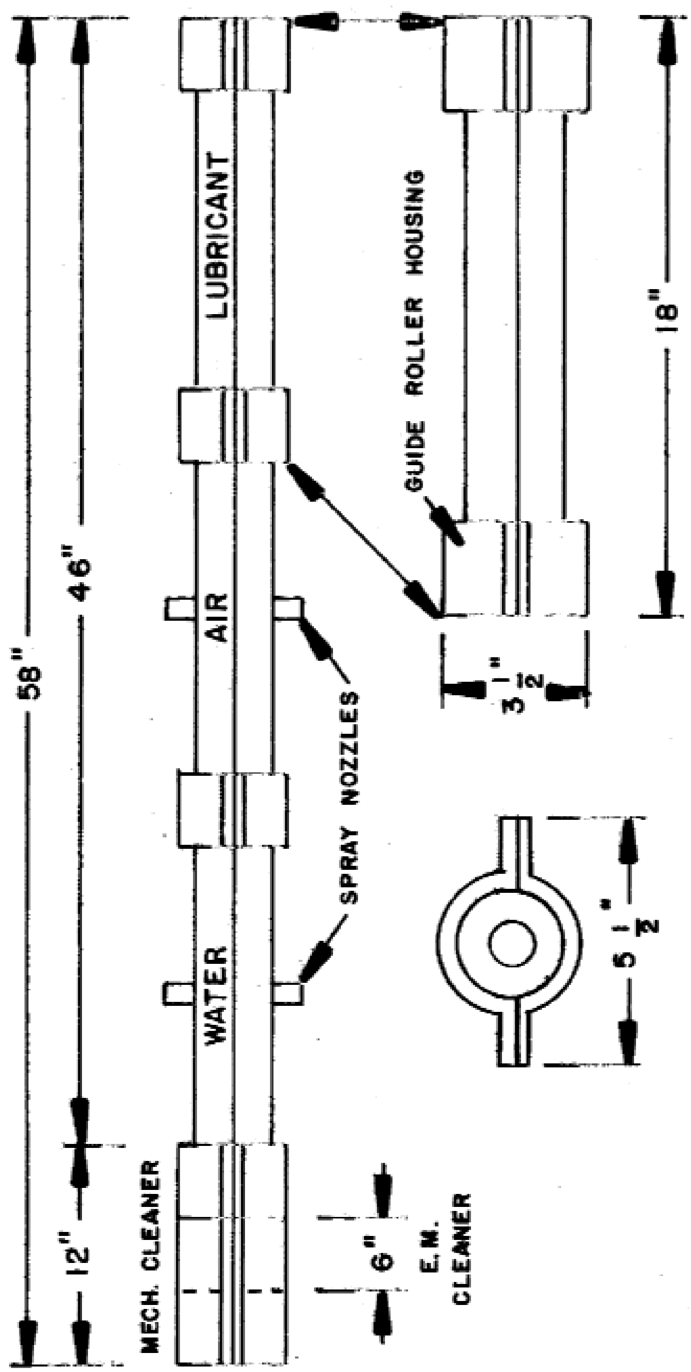
FIGURE 6-5

After examining different lubricators available, we highly recommend the combination of Brooke Ocean Technology's cleaner and lubricator with the rotating wire brush cleaner manufactured by Grignard Company (Figure 6-4). This combination as shown in Figure 6-6 of cleaner, water wash, air dry and lubricator is sectional and therefore only the required sections need be used depending on the condition of the rope. The combination of all the units are made of aluminum and weighs approximately 55 pounds (lubricator-8lbs.).

7.0 WIRE PRESERVATION DURING LONG TERM STORAGE

Wire rope or cable that will be stored for long periods in marine atmospheric conditions should be treated somewhat differently than the wires maintained at sea. This process involves three steps which, in actuality, occur simultaneously and involve a fresh water spray, air drying, and the applications of a lubricant/rust preventative. The process can be accomplished as the wire is removed form the ship's winch to the storage reel.

The fresh water spray acts to dissolve and remove any salts that may have remained on the wire after its last use. It is preferable that the freshwater be heated prior to spraying so that a maximum dissolution of salts, etc., can be achieved. The use of a spray rather than a washing with a hose allows for a maximum effect with a minimal water volume thereby increasing the efficiency of the air drying step.



COMBINATION UNIT

FIGURE 6-6

Air drying can be accomplished using a shoreside compressor and moderate pressure of approximately 80 psi. This, as in the field technique, serves to reduce the entrained moisture level and allow proper penetration of the lubricant. The lubricant/rust preventative can be applied utilizing the same unit as described in the field technique above.

Once the preservation process is completed, the reel should, ideally, be stored inside a building out of the weather as further protection. If this in not possible, a second alternative of covering the reel with a tarp is suggested. A periodic inspection and rotation of the stored reel is suggested as a means of keeping the lubricant/rust preventative evenly distributed throughout the spooled wire.

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Fink, F.W. and W.K.Boyd. The Corrosion of Metals in the Metals in the Marine Environment, Defense Metals Information Center, DMIC Report 245, 1970

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CHAPTER 7

INSTRUMENT LOWERING SYSTEM DOCUMENTATION

Richard J. Findley

1.0 INTRODUCTION

An instrument lowering system is comprised of several components; the winch, wire and blocks are all an integrated part of the system. Each of these components needs to have usage and maintenance logged to clearly understand failure modes and to establish when a wire has served its useful life and should be retired. In the previous chapters, a number of factors have been identified, which directly affect wire life and also best practices for prolonging wire life and establishing criterion for the retirement of working cables.

The focus of this documentation will be to assist in determining how well these best practices are being followed and gathering the required data to determine if the criteria for retiring a cable have been reached.

In addition, winch performance, maintenance, and repair records are required in order to field and maintain a safe and reliable system. Compliance with ISM code will require maintenance and log keeping documentation consistent with each institution's written operating procedures.

The accurate recording of the "Life History" of individual wires also serves a number of important functions when conscientiously applied across the community. Primarily it acts as a management control tool at the institutional level, a quality control device from the standpoint of the scientific user, a valuable source of background information in the event of a wire failure, and as justification for replacement of a wire or cable when required. Simple attention to the details involved in recording required information can make this documentation process an easy, yet effective means of control. The objective of the documentation process is to provide the winch and wire system user with a demonstrable performance record and previous cable history.

From a carefully practiced procedure of documentation, it is possible for the vessel operator to assess his wire and winch inventory and to make accurate assessments of his future requirements, as well as presenting, upon request, a complete history of any winch, block, or wire in his inventory.

The necessity for establishing a consistent means of documenting winch, wire and block histories cannot be stressed too heavily. In a climate of increased safety awareness and rising product and instrument costs, each loss, due to the failure of a instrument lowering system, through unrecorded incidents, inadequate practices, or operation of a wire past its realistic retirement condition, directly affects the entire community. This latter aspect is of importance where inter-institutional loans of wires occur. In this case, the borrower can receive a wire of known condition via its life history record. Since it is the task of the vessel operator to provide the scientist user with a safe and reliable winch and wire system at sea, it is important that the operator know the condition of the wires in his inventory with a fair degree of reliability. This condition is achievable through the documentation process.

The idea in lowering system documentation would be the use of a simple standard series of forms within the oceanographic community. Such a standardized format would allow easy evaluation of borrowed wires, provide an engineering database previously unavailable, and provide important feedback to wire manufacturers for product improvement. The implementation of a standard documentation format would in no way infringe on an institution's individuality, but would, instead, demonstrate the high level of cooperative effort that has been a part of this community.

2.0 WIRE AND CABLE DOCUMENTATION

Within this section, we will concentrate on the development of a basic documentation scheme designed to encompass all of the aspects of a wire's useful life. In addition, sample formats of suggested spreadsheet forms have been provided as a guide to establishing a comprehensive wire life history record.

This system only requires two forms to maintain a complete history of the cable these forms are referred to as the Activity Log and the Use Log. The Activity Log is used to record and track operations on the wire such as spooling, lubrication, termination, testing etc. The Use Log is used to maintain a record of meters deployed, and tension of each lowering or cast.

2.1 Wire Activity Log (Appendix A)

This document is presented as a spreadsheet, which would be the simplest method of maintaining this log, however it can be printed and maintained by hand. The correctly completed Wire Activity Log provides all the information needed to determine the status and location of a wire in a single place. There are two main sections to the form, the first part is the header data, and the second portion is in a tabular format.

Header Data

The information in this section would be filled out when the wire is originally acquired. It identifies the specific cable and provides the manufacturer information, specifications, and physical characteristics of the cable. Locating all of this information at the top of the log provides a constant reminder of the design limits of the cable. The information would not normally change during the life of the wire.

Wire Identification #: If accurate records are to be maintained; it will be necessary to establish a means of clearly identifying each individual wire or cable within the inventory. Any simple alphanumeric system can be adapted as long as it clearly identifies the individual wire or cable in some unique and non-repeatable form. The following illustrates one method of wire identification that can be employed.

Block #	1	2	3	4	5	6	7
Code	T	625	04	82	R	A	1

The particular alphanumeric combination shown above can be interpreted in the following manner:

Block 1 This letter refers to the particular cable type, i.e., F = fiber optic, T = trawl wire, H = hydrographic, E = Electro-mechanical cable, S = synthetic (i.e., Kevlar, Spectra, etc.). The letter used only identifies wire type as an aid in locating the wire in a storage area or during an inventory situation.

Block 2 The three-digit code shown in the example refers to the decimal diameter of the wire of that particular wire. These digits would obviously change for different diameter wires. Since a wire's diameter can always be reduced to a decimal equivalent, the three digit code would form a simple means of further identification.

Blocks 3 &4 - These two dual digit codes refer, respectively, to the month and year a specific reel of wire was received by an institution and placed in inventory. The date at which a reel of wire is placed in service becomes important when high reliability of the wire is required. By including the date of receipt in the identification number, a ready reference is provided at a glance.

Block 5 - The single letter code in this block can be used to identify the institutional point of origin of a particular reel of wire or cable and can become important as wires are loaned or traded amongst institutions. By being able to identify the institution managing the wire in use, its background and use record are always available.

Block 6 - This block would be reserved for the identification of simultaneous purchases of the same wire by an institution. In this case, either an A or a B suffix would be added to the code to differentiate the individual wires, thereby preserving their individuality.

Block 7 - This variable length code would be reserved for the identification of sub-divisions of the original wire after delivery from the manufacturer. The original uncut wire would be designated 1. If that wire was cut into two pieces it would be designated as 1.1 and 1.2. If the 1.1 wire was further divided into three pieces it would be designated as 1.1.1, 1.1.2, and 1.1.3 thereby preserving their individuality and remain traceable to the original wire as delivered from the manufacturer.

Manufacturer: Self-explanatory.

Date of manufacture: Self-explanatory.

Manufacturers part #: Self-explanatory.

Manufacturers serial #: Self-explanatory.

Wire Construction: This line should contain a description of wire size, construction (i.e., 3 x 19, 3 x 46, 6 x 19, etc.), or for electro-mechanical cables the armoring, number of conductors, etc.

Original Length: Self explanatory

Lay Length: From previous chapters it has been shown that lay length determines optimal wrap angles for each sheave.

Minimum Bending Radius: The minimum bending radius for the cable, usually provided by the manufacturer.

Established by: Indicates who established the minimum bending radius. This entry ties responsibility of the selection of a safety related issue to a particular person or document. It should reference a person by name or refer to a document and who made the decision to apply the document. Typically, the manufacturers recommendation is used, but an institution may choose to adopt a more conservative value to increase the working life of the cable. A minimum bending radius should never be adopted which is less than the manufacturers recommendation!

Breaking Strength (with end free to rotate): The breaking strength with the end free to rotate as provided by the manufacturer. Since almost all use of oceanographic cables are used in this mode this specification should be used as opposed to breaking strength with both ends fixed.

Safety Factor: The value that will be used to determine the safe working load of the cable.

Established by: This entry ties responsibility of the selection of a particular safety factor to a particular person or document. It should reference a person by name or refer to a document and who made the decision to apply the document.

Elastic limit: As defined in previous chapters, the value which when exceeded causes permanent deformation of the wire. Wires that have been stressed past this point will have a reduced breaking strength.

Additional items would be recorded for conducting cable such as wire gauge, number of conductors, and resistance per unit length. Fiber optic cable adds additional parameters such number of fibers, db loss per unit length, fiber diameter etc.. These items should also be included in the header information.

Tabular Data

The data in this section of the form is used to record activities performed on the wire such as spooling, re-termination, lubrication and pull tests, etc..

A new line is created for each activity with multiple lines being used for multiple activities at the same time. All information from the line above that is still valid should be carried down to the next line or left blank if it is not valid.

Seq. #: When a spreadsheet version of the log is used, it is desirable to be able to sort the rows by activity. As an example, the information could be sorted to allow all of the times the cable was lubricated to be grouped. The sequence number allows for resorting back to the original chronological order. Since multiple activities may happen on the same day the date alone is insufficient.

Date: Self explanatory.

Entered by: The person entering the data and bearing the responsibility for assuring the accuracy of the information.

Wire ID #: Because the wire identification number may need to change because of a sub-division and the previous history will still apply it is necessary to maintain a record of the changes in the wire ID #.

Activity: A keyword that describes the activity. Try to minimize the number of different keywords and keep the keyword consistent (i.e. don't spell out the activity one time and abbreviate it the next.) Some recommended keywords are: new, lubricate, pull test, rerig, terminate, subdivide, megger, remove, inspect, store, retire and dispose.

Ship/Location: Identifies the ship the wire is currently on or when ashore the building where it is located.

Winch/Spool ID: Identifies the winch the wire is currently installed on, or when ashore the spool, rack or other method of helping to locate the wire.

Orig. Bottom Now: In the process of transferring wires to and from storage drums the wire is end for ended. When the wire is in use tension and bending fatigue data will be recorded starting from the end of the wire in the water. When the wire is end for ended the subsequent data would apply to different section of cable, for this reason it is necessary to maintain a record of all end for ending activities. To maintain this information a convention needs to be established for identifying the specific end of the cable. In this system, the end of the cable that was at the root of the reel when delivered from the manufacturer is termed "Orig. Bottom." At the end of each wire activity the current location of the "Orig. Bottom" is recorded as being on either the top or bottom of the drum.

When a wire is sub-divided the end that was closest to the “Orig. Bottom” becomes the “Orig. Bottom” for that section.

Current Length: Self Explanatory.

Sheave # (1 through n): Because it has been established in previous chapters that bending radius, wrap angle and a correctly sized groove are critical factors in determining the working life of a cable, it is important to record the information about each of these parameters to aid in determining when a wire has reached the end of its life. Additional columns should be added for each sheave that the wire passes over. In the case of identical multiple sheaves such as would be encountered with a traction head or accumulator, a forth column would be added to the information for that sheave #, identifying the number of sheaves in the group.

Root Radius: Radius of the sheave at the bottom of the groove. The minimum value should be twice the minimum bending radius for the wire.

Wrap Angle: The total number of degrees that the wire is in contact with the sheave.

Groove Radius: The radius of the groove that the wire rides in. While a correctly sized groove diameter is an important consideration for all cables it is especially critical with conducting and fiber-optic cables.

Comments: Short comments can be place directly in the table. Longer discussions should be placed in a separate document and referenced here.

3.0 USE LOG (Appendix B)

This particular aspect of the documentation may well be the most important item in the system as it a record of the actual work done by the all of the components in the system. When properly recorded either by computer or manually, it will reflect length of wire deployed at each station, maximum stress placed on the wire, and documents the accumulated bending stress cycles. The data from this form when combined with the Activity Log provides a complete picture of the wire history.

The following example (Figure 7-2) represents a typical Use Log that has been filled out manually. For this form manually refers to either hand written or manually entered into a spreadsheet verses a computerized data logging system.

If a computerized data logging system were being used some of the data would need to be entered into the data logging system by the operator. In this situation, typical parameters would be winch operator, station number, station type and sea state.

<u>Wire ID #</u>: The number would be the same ID # as on the Wire Activity Log. By incorporating it in the "Use Log," a clear trail is established for that particular wire.

<u>Cruise Number and Date:</u> This information is used to correlate events with the Wire Activity Log and may be used by the science party.

<u>Station Number</u>: Primarily used by the scientific party. Different organizations may use different terminology such as lowering number or cast number.

<u>Station Type</u>: This item simply identifies the instrument being deployed. Since piston cores, dredges, or mid-water trawls all place unique stresses on the wire, a great deal of guesswork is eliminated in later analysis of the station. Type is identified on the form. In addition, this item is also used to identify wire tensioning deployment as well as the wire lubrication process.

<u>Payload Weight (Air)</u>: Since the weight of any instrument attached to a reel of wire or cable should be known or can be determined easily, it is important that it be recorded as part of the permanent record. This will also provide a spot check of the tension measurement system if the same instrumentation is being used repeatedly at the same weight.

<u>Available Wire Length:</u> This number refers to the length of wire which is available for use. It would represent the amount of wire available after subtracting the amount the institution determines is

suitable to remain on the winch from the wire length reported on the last entry of the Wire Activity Log. Some institutions only require several turns others require an entire layer.

Sea State/Weather: Since the motion of the vessel, due to sea conditions and weather, imparts stresses to the wire, the recording of these conditions becomes a necessary part of any later wire analysis and should be included as part of the permanent record.

Maximum Wire Out Maximum Tension: These two numbers are derived from the actual lowering record and are repeated at
the top of the Use Form as a means of ready reference for wire analysis or the determination of a maximum wire length to be paid out during the lubrication process. A place to mark if the safe working load or elastic limit has been exceeded should also be provided. **Any time the elastic limit is exceeded a record should be made in the Wire Activity Log.**

The bulk of the remainder of the form is self explanatory. The frequency of recording the amount of wire over the side is an individual choice, but should be no less than once every 500 meters, or at the time of specific events (i.e., winch speed changes, corer pullout, etc.). Major events, such as winch malfunctions, and cables jumping of sheaves occur pertinent to a specific lowering. Any events that may reduce the strength of the wire should be recorded on this form and on the Wire Activity Log.

3.0 WINCH DOCUMENTATION

Given the variety of winches found on most research vessels and the fact that they are the primary working tool of the oceanographer, it is fairly obvious that they deserve special attention to ensure their operational readiness. This special attention should take the form of programmed maintenance, and repair combined with a procedure of reporting the performance of such maintenance. Compliance with ISM code will require regularly scheduled maintenance and documentation of that maintenance.

7-12

Winch Activity Log (Appendix C)

A Winch Activity Log is similar to the Wire Activity Log and should be employed. It is again suggested that this information be maintained in a spreadsheet but it could also be printed and filled out by hand. An examination of the header and tabular data on the example form is self explanatory and therefore is not listed here.

Each operation is logged on a separate line as was done on the Wire Activity Log.

4.0 BLOCK DOCUMENTATION

Blocks are a critical part of the instrument lowering system. Failure of a block under load can be one of the most dangerous situations on board a vessel an improperly sized bloks can significantly reduce the breaking strength of the wire and reduce wire life. As in the other parts of this system compliance with ISM code will require regularly scheduled maintenance and documentation of this critical component of the system.

Block Activity Log (Appendix D)

An examination of the header and tabular data on the example form is self explanatory and therefore is not listed here with one exception below that needs special attention.

Hook Safe Working Load: Most blocks are rated on the load at the point of attachment. This value is not the same as the wire tension. A block with a 180 degree wrap angle will have a load on it which is twice the wire tension. A block that the wire just “kisses” will have a small fraction of the wire load. If the manufactures Safe working load (SWL) specification does not specify hook load, contact the manufacturer and get in writing how they have defined the safe working load.

5.0 INCIDENT REPORTING

Although no one likes to consider all the things that can go wrong, it is important that when they do occur they be fully and accurately reported. The minimization of a problem or a catastrophic failure by either refusing to face the problem or by simple acceptance of such a failure as a natural occurrence does nothing to eliminate the initial causes of the problem. By minimizing a problem all that is accomplished is the recreation of the same set of conditions that led to the problem or failure in the first place and the probability of another problem or failure in the future is more or less assured.

In order to derive a comprehensive record from a system failure it is necessary to collect and analyze the available information. There are three distinct phases in analyzing a system failure. The onboard data gathering with initial analysis, the post engineering analysis which is done ashore, and the corrective measures taken to prevent reoccurrence. Each phase should be combined in a single incident report.

It is felt that this style of report should be circulated to other users engaged in the same type of at-sea work as a warning and as a preventative against similar failures. It should be remembered that unless adequate and accurate data is made available to users and manufacturers alike, it is not possible to influence product improvement or increase our at-sea reliability.

Phase 1 <u>Shipboard Data Gathering and Preliminary Analysis</u>

This section of the report should be completed at sea immediately following a system failure. The report should list all of the hardware involved including the winch, wire, block, instrument, and any shackles or hardware that were in use. Any visible damage to the hardware should be described and supplemented by photographs. In the event of catastrophic failure, use of all the hardware involved should terminated since the hardware may no longer be capable of meeting its original load specifications. These items should be preserved for later analysis.

A brief description of all items of equipment lost should be listed in the space provided. The equipment may not be a direct cause of the failure, but its description will provide useful data during a later analysis.

It is crucial that the actual point of failure be identified as accurately as possible to aid the engineering analysis. Major differences in failure modes can be determined if the location of the break is known. For instance, a failure at the winch probably does not have the same cause as one occurring at the outboard sheave. A list of all possible witness should be made along with any of his or her observations. Be aware that their observations may conflict. Be sure to take them down as they describe the event. Determine if anyone onboard happened to be taking video or still pictures at the time of the failure, if so make arrangement to ensure that copies are obtained.

Once all pertinent data has been assembled, a preliminary cause of the failure should be determined and recorded. This evaluation while preliminary but is nonetheless valuable in an assessment of the actual problem. Once this on-site evaluation is completed, it should be endorsed by either the vessel's captain or other responsible party. The Winch Activity Log, Wire Use Log, and Block Activity log should be updated with the report name added to the remarks column in each report.

Phase 2 <u>Ashore Post Engineering Analysis</u>

In this phase the in shipboard report is reviewed, an analysis of all of the hardware involved should be conducted which may include non-destructive or destructive testing. The results of the investigation should be included in this section of the report. Any hardware, which is suspected to have been stressed beyond its normal safe working load, should either be load tested or clearly marked as scrap.

Phase 3 <u>Corrective Action</u>

Once the cause of a system has been established, it is obviously necessary to institute some form of corrective action to ensure that a repeat failure will not occur. When a course of action has been selected a summary of that action should be recorded in the report. and added to the Incident Report to complete the package. Corrective action may take the form of either procedural changes in its simplest form or may result in major structural or component changes in the winch system.

6.0 SUMMARY

The documentation system that has been described in the preceding sections constitutes a basic minimum that would be required to effect a safe and reliable instrumentation lowering system.

The success of this or any other documentation scheme rests with the individuals who are responsible for submitting the required information and their cooperation can be more easily achieved if the demands on their time are kept to a minimum. In other words, the proliferation of unique forms and reports should be kept to a minimum if the system is to have any real value.

It is strongly encouraged that the documentation scheme, no matter what it is, be established as a standard within an organizational group such as UNOLS, NOAA, etc. Once this is accomplished, the control over, and reliability of the wires and winches within these Organizations becomes a realistic approach to the problem.

7-16
REFERENCES

Winch and Wire Handbook, Second Edition. University National Oceanographic Laboratory System

Wire Activity Log (Mechanical Cable)

Wire Identification #	T5000298WA1.1		
Manufacturer	Rusty's Wire Inc.	Minimum bend radius (in inches)	12
Date of Manufacture	12/12/98	Established by	I. M. Responsable
Manufactures Part #	1243	Breaking Strength (with end free to rotate)	25000
Manufactures Serial #	45638	Saftey Factor	5
Construction	1/2" 3x19 GIPS	Established by	I. M. Responsable
Original Length	10,000	Safe working Load	5000
Lay Length (in inches)	9.25	Elastic limit	19000

									Sheave #1			Sheave #2			
Seq. #	Date	Entered by	Wire ID #	Activity	Ship/ Location	Winch/ Spool ID	Orig. Bottom Now	Current Length M	Root Radius Inches	Groove Radius Inches	Wrap Angle Deg.	Root Radius Inches	Groove Radius Inches	Wrap Angle Deg.	Coments
1	02/02/98	T. Allen	T5000298WA1	Receive	Shed 32	B	Bot	10000							New
2	02/02/98	T. Allen	T5000298WA1	Lubricate	Shed 32	B	Bot	10000							Lubed at Factory
3	03/15/98	B. Sims	T5000298WA1	Spool	RV Rolly	SN 2345	Top	10000	12	0.510	45	10	0.252	90	Spool_report.doc
4	03/15/98	B. Villa	T5000298WA1	Terminate	RV Rolly	SN 2345	Top	10000	12	0.510	45	10	0.252	90	Electroline
5	03/15/98	A. Bundy	T5000298WA1	Pull Test	RV Rolly	SN 2345	Top	10000	12	0.510	45	10	0.252	90	Tested to 5500 #
6	06/15/98	T. Allen	T5000298WA1	Terminate	RV Rolly	SN 2345	Top	9985	12	0.510	45	10	0.252	90	Electroline
7	06/15/98	T. Allen	T5000298WA1	Pull Test	RV Rolly	SN 2345	Top	9985	12	0.510	45	10	0.252	90	Tested to 5500 #
8	03/01/99	B. Sims	T5000298WA1	Rerig	RV Rolly	SN 2345	Top	9985	12	0.510	45	16	0.260	90	Rplcd. Undersized block
9	03/01/99	B. Villa	T5000298WA1	Terminate	RV Rolly	SN 2345	Top	9985	12	0.510	45	16	0.260	90	Electroline
10	03/01/99	A. Bundy	T5000298WA1	Pull Test	RV Rolly	SN 2345	Top	9985	12	0.510	45	16	0.260	90	Tested to 5500 #
15	03/15/99	T. Allen	T5000298WA1.1	Subdivide	RV Rolly	SN 2345	Top	3000	12	0.510	45	16	0.260	90	Incident_03_05_1999.doc
16	03/15/99	B. Sims	T5000298WA1.1	Lubricate	RV Rolly	SN 2345	Top	3000							Prelube 19
17	03/15/99	B. Villa	T5000298WA1.1	Remove	Shed 32	L	Bot	3000							
18	12/01/00	A. Bundy	T5000298WA1.1	Spool	RV Pitch	SN 3723	Bot	3000	6	0.510	180				Does not levelwind
19	12/01/00	T. Allen	T5000298WA1.1	Terminate	RV Pitch	SN 3723	Top	3000	6	0.510	180				U-Bolt Clamps
20	12/01/00	T. Allen	T5000298WA1.1	Pull Test	RV Pitch	SN 3723	Top	3000	6	0.510	180				Tested to 5500 #
21	12/09/00	B. Sims	T5000298WA1.1	EL Exceed	RV Pitch	SN 3723	Top	2300	6	0.510	180				Incident_2000_12_09.doc
22	12/15/00	B. Villa	T5000298WA1.1	Remove	Shed 32	C	Bot	2300							
23	12/20/00	A. Bundy	T5000298WA1.1	Retired	Shed 32	C	Bot	2300							
24	01/15/01	F. Sanfor	T5000298WA1.1	Disposed											

Appendix A - Sample Wire Activity Log

Use Log

Wire ID #	T5000298WA1.1	Date	9-Dec-00
Cruise #	PI0017	Station Type	Rock Dredge
Station #	1	Sea State	4
Payload Wt. (air)	600	Available Wire Length	2700 M
Breaking Strength	25000	Winch Operator	Bart Sims
Safety Factor	5		
Safe Working Load	5000	Exceeded this cast ?	Y or N
Elastic Limit	19000	Exceeded this cast ?	Y or N
Max. wire paid out	2500		
Max. tension observed	9958		

Time GMT	Wire Out Meters	Max Observed Tension	Rate Out	Rate In	Comments
8:30	0	0	0	0	Tensiometer reads 0 with no load
	-2	595			Lifted dredge off deck, air weigh ok
8:30	3	440	60		Started down
8:39	500	777			
8:48	1020	1114			
8:57	1490	1451			
9:06	2100	1788			
9:10	2500	2125	0		Manual Brake Set, Pawl Set
9:12	2500	3688	0		
9:14	2500	3756	0		
9:16	2500	3897	0		Al Bundy relieved Bart Sims as winch operator
9:20	2500	4615	0		
9:25	2500	5345	0		
9:30	2500	6345	0		
9:35	2500	8721	0		
9:40	2500	9958	0		
9:42		1961	0		Shudder throught ship
9:42		1952	0		
9:50	2500	1946		70	Start bringing in wire
10:14	200			0	Reached end of wire, dredge gone
					See Incident_2000_12_09.doc

Appendix B - Sample Use Log

WINCH ACTIVITY LOG

Manufacturer	X. Pensive Machinery Corp
Model #	BAD-322
Serial #	3723
Date of Manufacture	1/1/98
Maximum Pull (Full Drum)	5000 lb.
Maximum Pull (Bare Drum)	10000 lb
Wire Size Designed For	.322"
Capacity	10,000 m

Seq. #	Date	Entered by	Activity	Ship/ Location	Installed Wire ID #	Comments
1	02/02/98	T. Allen	Receive	Shed 32		
2	02/02/98	T. Allen	Lubricate	Shed 32		Per manufacturers startup procedure
3	03/15/98	B. Sims	Install	RV Rolly		01 Deck lined up with aft eye of side a-frame
4	03/15/98	B. Villa	Spool	RV Rolly	C3220298WA1	
5	04/14/98	A. Bundy	Lubricate	RV Rolly	C3220298WA1	Per manufacturers 30 day lube schedule
6	05/14/98	T. Allen	Lubricate	RV Rolly	C3220298WA1	Per manufacturers 30 day lube schedule
7	06/13/98	T. Allen	Lubricate	RV Rolly	C3220298WA1	Per manufacturers 30 day lube schedule
8	07/13/98	B. Sims	Lubricate	RV Rolly	C3220298WA1	Per manufacturers 30 day lube schedule
9	08/12/98	B. Villa	Lubricate	RV Rolly	C3220298WA1	Per manufacturers 30 day lube schedule
10	09/11/98	A. Bundy	Lubricate	RV Rolly	C3220298WA1	Per manufacturers 30 day lube schedule
15	10/11/98	T. Allen	Lubricate	RV Rolly	C3220298WA1	Per manufacturers 30 day lube schedule
16	11/10/98	B. Sims	unspool	RV Rolly		
17	12/10/98	B. Villa	Remove	Shed 32		
18	01/09/99	A. Bundy	Install	RV Pitch		
19	12/01/00	T. Allen	Spool	RV Pitch	T5000298WA1.1	
20	12/01/00	T. Allen	Lubricate	RV Pitch	T5000298WA1.1	Per manufacturers 30 day lube schedule
21	12/09/00	B. Sims	Incedent	RV Pitch	T5000298WA1.1	See Incident_2000_12_09.doc
22	12/15/00	B. Villa	Unspool	RV Pitch		
23	12/20/00	A. Bundy	Remove	Shed 32		
24	01/15/01	B. Villa	Ship			Sent to manufacturer for inspection

Appendix C - Winch Activity Log

BLOCK ACTIVITY LOG

Manufacturer	Tiny Block Inc.
Model #	6/500
Serial #	1
Date of Manufacture	Unknown
Safe Working Load	20000 lb.
Safety Factor	?
Root Radius	8" 6"
Groove Radius	0.51

Seq #	Date	Entered by	Activity	Ship/ Location	Installed Wire ID #	Comments
1	02/02/98	T. Allen	Receive	Shed 32		Found block on warehouse shelf
2	03/04/98	A. Bundy	Install	RV Pitch		Took specs off plate on side of block
3	12/09/00	C. Bleigh	Incedent	RV Pitch	T5000298WA1.	See Incident_2000_12_09.doc
4	12/15/00	B. Villa	Remove	Shed 32		
5	12/20/00	F. Sanford	Disposed			

Appendix D - Sample Block Activity Log

Appendix E
Incident Report

Date: December 9, 2000
By: Capt. Bleigh

At 9:42 GMT we were underway at 1.5 knots in the process of performing a rock dredge at a position Lat 26 00 N, 80 00 west, and the wire parted. There was 2500 meters of ½" 3x19 wire out. The tension never got over 9958 lbs which is less than half the breaking strength. The winch in use was SN 2345, the wire in use was T5000298WA1.1. The block in use was a Little Block serial number

1. The winch was located on the after deck about 15 ' from the stern. Our in house manufactured rock dredge and 200 meters of wire lost.

Crewmembers Sims, and Allen were both standing on the aft deck taking a cigarette break by the a-frame when the wire parted. They both report feeling the ship shudder. Since the wire parted underwater, they did not see anything else. No one happened to be taking pictures at the time.

The winch operator was ordinary seaman Al Bundy, he reported that he had been writing down the tension every 5 minutes and did not observe the tension exceeding 9958 pounds.

The first mate, Fletcher Christian was watching the radar as another ship was in the area; he also felt a shudder but did not have anything else to report.

The Wire Use Record, Wire Activity Log, Winch Activity Log, and Block Activity Log were annotated with the file name of this report.

A photo of the end of the cable is attached.

Preliminary cause of failure: Defective cable.

Respectively Submitted

Captain Bleigh

Shoreside Analysis
By: Bob Villa

The wire was physically inspected and appears to be recently lubricated. There was minimal corrosion. A review of the wire activity log and incident report dated March 3, 1999 indicates that this section of wire came from a longer wire that was damaged when the wire jumped a sheave. This particular section was from the bottom of the drum and had never been in the water before.

A section of the cable was taken from the end of the cable and at 2000 meters from the "Original Bottom of Cable" both were pulled to destruction. The end section failed at 19,238 pounds. The section from 2000 meters from the failed 21356 pounds.

An additional section was taken from the wire that had not been deployed in this incident. By reviewing the Wire Use Log it has been determined that his section of wire has never been in the water. This section was pulled to destruction and failed at 25873 pounds.

From these results, it appears that the cable has been weakened in use during the cast made at station 1.

A review of the wire use record indicates that the wire tension was being observed every 5 minutes. The manual brake and pawl were sent which would have prevented the winch from slipping. I have also reviewed the over boarding sheave and found that the radius of the sheave is only 6". The minimum radius for this cable is 12". The radius of the sheave groove is .51 inches which is suitable for a 1" cable not the .5" which was in use.

I interviewed the scientist Bill Nye who had attached the dredge to the cable and determined that a weak link was not in use on the rock dredge.

Final determination of cause of failure:
The wire exceeded its breaking strength, the manual observation of the readout was not often enough to note the true tension of the cable.

Additional Findings:
The block that was used was the wrong size for the wire. The manufacturer of the block is no longer in business and the safety factor in determining its SWL is unknown. The block was placed in service without knowing it's previous history. It may have previously abused which can lower it's load capacity.

The safe working load of the cable was exceeded while the crewmembers were loitering around an over the side operation. There was no reason for them to be in this area.

The manual brake and the pawl were set on the winch which would prevent the operator from being able to pay out wire while the ship was slowed down.

There wasn't a weak link in place, which almost guarantees that a catastrophic failure will occur if the dredge hangs up.

The winch has been subject to a load in excess of its design load. It will be returned to the manufacturer for inspection.

The elastic limit of the wire has been exceeded and will be disposed of.

The block will be cut up and disposed of. It has been stressed past its SWL, its history is uncertain and its dimensions are unsuitable for oceanographic work..

All of the shackles and small rigging hardware has been marked scrap and disposed of

Respectfully Submitted

Bob Villa

Corrective Action

Corrective Action Recommended as of a result of incident on November 9, 2000
By: Bob Villa

To prevent the use of undersized blocks and to familiarize personnel with the correct procedures for safely performing over the side operations, all crewmembers and shore personnel have attended a one-day in house seminar on winches and wires. The seminar was based on the Winch and Wire Handbook supplied by UNOLS.

The Wire Activity Log should be modified so that when an improperly sized sheave is entered into the tabular data the entry turns red.

Crewmembers should review all of the existing Incident Reports so they learn from past mistakes.

Signs should be posted on the deck warning all personnel who are not engaged in the current activities to keep clear of the area.

Weak links should always be used in all operation where the ship is moving such as trawling and grappling.

An electronic tension measurement and logging system should be installed with alarms alerting the operator when the SWL of the cable is exceeded.

The manual brake and pawl should not be set.

CHAPTER 8

OPERATIONAL CHARACTERISTICS OF ROPES AND CABLES

Philip T. Gibson

1.0 INTRODUCTION

This chapter of the handbook describes the basic operational characteristics of ropes and cables. The emphasis is on "working" ropes and cables used under dynamic conditions and requiring high strength and the ability to survive cyclic tension loading and cyclic bending. Examples include ropes and cables used to suspend, tether, or tow various payloads from floating or submerged platforms. Ropes and cables used in more passive applications (such as for guy lines or bottom-laid power or sensor cables) are not addressed directly, although many of the following chapter sections can be applied to these applications, as well.

1.1 Nonmenclature

The term "rope" applies to a flexible tension member used to transmit a tensile load to a remote location and which has sufficient flexibility to accommodate repeated bending over sheaves and drums. Included in this category are wire ropes and also nonmetallic ropes made of high-modulus fibers (for example Kevlar fiber from du Pont or Spectra fiber from Allied Chemical). Ropes made of low-modulus fibers (for example polyester, nylon, or polypropylene) are not discussed.

The term "cable" applies to a flexible tension member which, in addition to a strength member, includes power and/or signal conductors within its structure. As in the case of ropes, cables are used to transmit tensile loads to remote locations, and they typically have sufficient flexibility to accommodate repeated bending over sheaves and drums. Again, the strength member may be either metal wires or non-metallic fibers.

1.2 Chapter Organization

The intent of this chapter is to provide information about various types of working ropes and cables having both metallic and non-metallic strength members. Since cables are more complex in both design and materials selection than are ropes, a majority of the following discussion is directed at cables. Additional information specific to ropes is presented separately.

The chapter begins with descriptions of typical cable configurations. Cable reaction to tensile loading is then explored with discussions of constructional and elastic stretch, the significant internal loads and stresses, tension-induced torque and rotation, the potential for hockling and kinking, and cable reaction to rotation.

Next, cable reaction to bending is described, including the associated stresses and motions experienced by the cable components, the effects of bending on breaking strength, and the influence of sheave wrap angles and cable cycling stroke amplitudes on fatigue performance.

Also included are discussions about motion compensation systems, sheave design, cable reeving configurations, winding on drums, and terminations. Finally, cable failure mechanisms and retirement criteria are discussed.

The focus of the chapter then turns to ropes with emphasis on those aspects of rope behavior which differ from the behavior of cables. Simple equations are presented for estimating the torque characteristics of common 6-strand wire ropes. Also included are the results of extensive laboratory tests to evaluate the cyclic bend-over-sheave fatigue life of several selected wire ropes.

The chapter is concluded with a brief discussion of recent developments in the area of mathematical modeling of ropes and cables.

2.0 TYPICAL CABLE CONFIGURATIONS

Cables typically fall into one of three basic design categories. The most common configuration is one in which the power and/or signal conductors are contained within the center of the cable and are surrounded by the strength member and, perhaps, an overall jacket. Another configuration has a center strength member that is surrounded by the power and/or signal conductors and, perhaps, an overall jacket. Finally, in rare cases, a cable may have a strength member which lies along side of and is attached to a separate assembly of power and/or signal conductor elements. A large number of design and material choices exist within these categories.

Operational systems which subject cables to high tension loads combined with bending over sheaves and drums typically use cables which have an external strength member and, perhaps, an overall jacket. This configuration provides the best protection and service life for the core conductors, it is easily handled using conventional winch systems, and it can be designed to provide high strength, good torque balance, and good cyclic tension and cycle bending fatigue performance. Because of its widespread use, this cable configuration will be the main subject of the following discussion. However, many of the concepts presented may also be applied to other cable configurations.

Typical configurations of cables having external strength members of steel wire are shown in Figure 8-1. Cables with two layers of armor wires are most common. However, more than two layers of armor wires can be used if increased cable strength or weight is required.

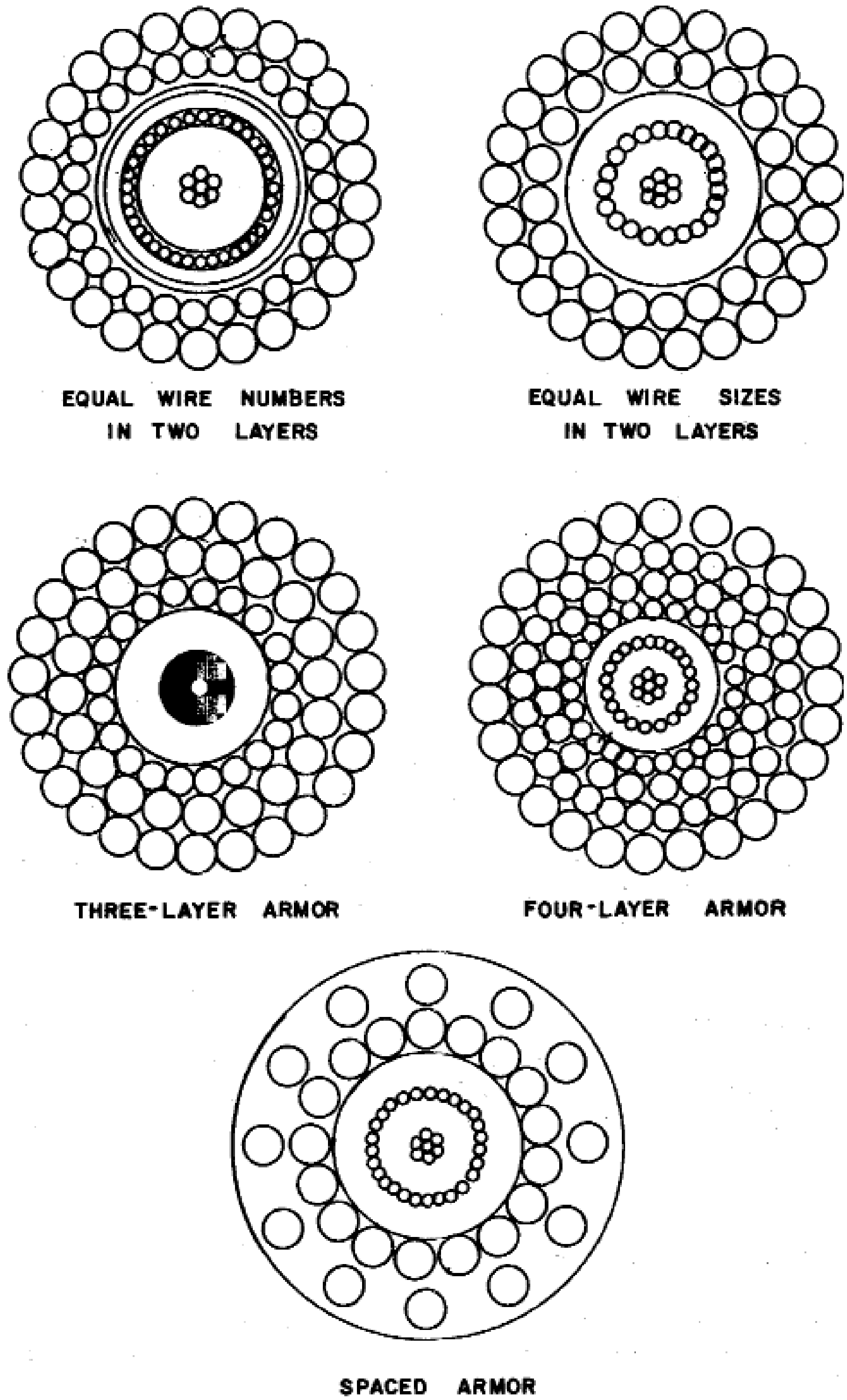
If a cable has a large core assembly with many conductors, then a full double armor design may provide excessive cable weight and more strength than required. The alternative is a spaced armor design where the armor wires within each layer are widely spaced and are held in position by an integral extruded jacket.

Cables which use non-metallic fibers rather than steel wires as the strength member elements typically have the fiber applied either as a braid or in several contrahelically served layers. Higher strength, lower stretch, and better flexure performance can typically be achieved with a multiple layer, contrahelically served strength member with the fiber layers separated by some type of low-friction isolation tape. In either case, the cable typically includes an overall extruded or braided jacket.

With few exceptions, all elements of an operating cable (power conductors, signal conductors, and strength members) are assembled with helical paths within the cable structure to accommodate bending of the cable. While the discussion which follows is directed primarily at cable strength members, it applies to all helically wrapped elements within the cable structure.

3.0 CABLE REACTION TO TENSILE LOADING

This section of the chapter discusses the reaction of a cable to straight tensile loading. A later section explores the effects of combined tension and bending.



TYPICAL CABLE CONFIGURATIONS

FIGURE 8-1

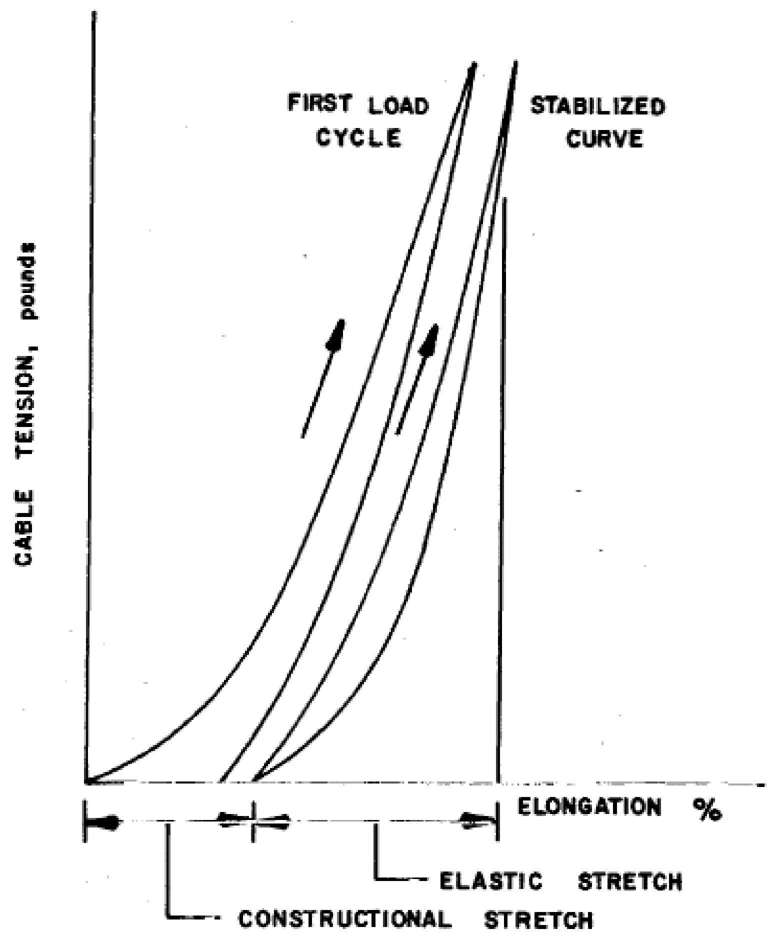
3.1 Constructional and Elastic Stretch

The tension-induced elongation of a new cable consists of two components, the constructional stretch and the elastic stretch, as shown in Figure 8-2. The magnitude of either of these components is likely to be both time and strain rate dependent.

Constructional stretch is most evident in cables having external strength members. As tension is applied to such cables, the strength members exert a radial pressure on the core. This pressure produces deformations of the core elements and filler materials due to both material compressibility and the elimination of voids within the core structure. In addition, for the case of steel wire strength members, the inner wire layer presses against the core jacket and causes the jacket material to move into the interstitial cusp-shaped voids between adjacent wires. There may also be slight contact deformations at the interface between the strength member layers. For the case of non-metallic fiber strength members, no cusp filling occurs, but the individual fiber layers experience some degree of compaction. All of these factors contribute to a reduction in cable diameter and a corresponding increase in cable length.

When tension is removed from a cable, there is some recovery of diameter and a corresponding reduction in cable length (in addition to the length reduction caused by the elasticity of the load bearing elements). However, a portion of the core compression and cusp filling or fiber compaction may be relatively permanent. The result is a residual cable elongation referred to as constructional stretch. Since some cable materials may exhibit time dependent elastic properties, a portion of the constructional stretch may dissipate as the cable remains a low tension for a period of time. However, the lost portion will be regained when tension is again applied to the cable.

In some cases, this constructional stretch may actually be larger than the additional elastic stretch at normal Operating tensions. This quasi-permanent change in cable length must not be overlooked as a potential contributor to the premature failure of certain cable components. For example, the total strain experienced by the core elements will be a function of both the constructional and elastic elongation. A large constructional stretch may produce excessive strain on optical fibers and, in the long term, may contribute to fiber failures at moderate cable tensions or even during storage of the cable at low tension between missions.



TYPICAL CABLE ELONGATION CHARACTERISTIC

FIGURE 8-2

3.2 Stress and Torque Balance

As discussed later, it is usually desirable for a cable to have good torque balance so that it produces little torque when loaded with both ends constrained and little rotation when loaded with one end free to rotate. Certain outdated approaches to cable design (specifically the use of simplified, linear analyses such as the "torque ratio" equation) may produce cable configurations which, at first, appear to offer good torque balance. However, these cables may actually have very large torque and rotation imbalances. The cause of this imbalance is often related to the tension-induced or pressure-induced diameter reductions experienced by the cable component layers.

The change in diameter of a cable having an external strength member (with the exception of a braided strength member) can alter the load sharing among the strength member layers. As the load sharing (stress balance) changes, so does the torque contribution of the various layers.

For example, if a double-armored cable should have the same or a higher helix angle for the inner armor wires as compared to the outer, then the effect of tension-induced cable diameter reduction will be to cause the outer armor layer to carry most of the applied tension. Conversely, if the helix angle of the outer armor layer is significantly larger than that of the inner, the applied tension will be carried primarily by the inner layer. In either case, the cable elongation, breaking strength, and torque and rotation balance are adversely affected.

Cables with non-metallic fiber strength members pose the additional complication that the fiber layers may experience compaction (the elimination of void area) as a consequence of cable tension or external hydrostatic pressure. The compaction of a given fiber layer can affect not only the tension and torque contributions of that layer, but also the contributions of any layers wrapped around it. Whenever one fiber layer experiences compaction and becomes thinner, the overwrapped layers are allowed to seek a reduced pitch diameter. When this happens, the outer layers shed some or all of their tensile bad. The result is a potential reduction in cable strength and elastic modulus and an upset of the cable torque balance.

There is, however, an optimum combination of helix angles for the various strength member layers which will allow the strength member to maintain good stress and torque balance regardless of the magnitude of the tension-induced diameter changes. Cable design and analysis software (such as CABLE SOLVER developed by Tension Member Technology) can be used to establish this optimum strength member geometry.

It is important to note that the cable strength member is not the only potential source of cable torque imbalance. The cable core may carry a significant portion of the applied cable tension and may produce a large torque. The tension and torque contributions of the core can be quite high in cables which have large power conductors or other core elements with a significant cross-sectional area of material with a high elastic modulus.

The analysis of the core contribution to overall cable behavior is complicated by the fact that most core materials have nonlinear stress versus strain characteristics. For example, the copper wire used in most power and signal conductors has such a low elastic limit that it often experiences strains well above its yield point. As a consequence, the tension and torque contributions of the conductors may be nonlinear and may change with repeated load cycling or bending of the cable. Here, again, modern nonlinear cable design and analysis software is required to gain an understanding of the core contribution to cable behavior.

A cable characteristic which is distinct from the torque versus tension behavior is the torsional stiffness. Torsional stiffness is a measure of the ease with which a cable will rotate in response to internally generated or externally applied torque. In general, small diameter cables are torsionally soft, and they will exhibit large amounts of rotation in response to relatively small amounts of torque imbalance. Also, cables with non-metallic fiber strength members are generally much more torsionally soft than are similar cables with metallic strength members.

The torsional stiffness of a cable may be highly directional, especially for cables with two or three layers of armor wires. Cables of these designs will typically rotate much more easily in the direction to loosen the outer layer of wires. In this case, to produce a cable with a minimum amount of rotation, it is desirable for any amount of torque imbalance to be in the direction which causes a tightening of the outer layer of wires. Cables with four layers of armor wires tend to have high torsional stiffness in either direction of rotation.

4.0 CABLE HOCKLING AND KINKING

It is usually desirable for a cable to have good torque and rotation balance to minimize the possibility of cable hockling and kinking in service. A hockle is a loop which forms in a cable and then becomes twisted so that the portions of the cable on either side of the loop become helically wrapped around each other. The hockle, itself, may not seriously damage the cable, but it renders the cable useless where a tension load must be transmitted to a payload. Any application of tension to a hockled cable may cause the hockle to tighten, thereby producing permanent cable deformation and kinking. In a steel wire armored cable, the outer armor wires may become badly displaced or bird caged as a result of this hockling and kinking.

The generation of a hockle in a cable requires only that a slack loop of sufficient size be allowed to form in a cable which contains a sufficient amount of stored torsional energy. If a cable contains no torsional energy, then the formation of a slack loop is not likely to produce a hockle. Similarly, if even a small amount of tension is maintained on the cable so that a slack loop cannot form, then no hockling will occur even if the cable contains a rather large amount of torsional energy.

There are a number of ways in which a cable can inadvertently form a slack loop. For example, when a payload is lowered to the sea floor, it may be difficult to determine exactly when the payload touches bottom, especially in deep water. If excess cable length is deployed after bottom contact, then a slack loop will form at the lower end of the cable. In other situations, it is possible to form a slack loop near the upper end of a tow or tether cable due to motions of the support platform. For certain combinations of payload weight, cable deployed length and elasticity, and platform motions, a resonant condition may produce snap loading of the cable. During snap loading, the cable may repeatedly experience slack loop formation.

There are several sources for the torsional energy required to potentially force a cable slack loop to become a hockle. Obviously, a cable which is not a torque-balanced design will produce torque in response to applied tensile loading. it is usually assumed that when the tension is reduced to zero, the torque will become zero, also. However, this assumption is true only if the tension is zero over the entire cable length.

Consider, for example, a long, heavy, nontorque-balanced cable used to lower or tow a payload which is not allowed to rotate. Because of cable weight and/or hydrodynamic drag, the tension at the surface will be higher than the tension at the payload. The cable torque versus tension behavior as determined in the laboratory may indicate that the cable torque will be highest where the tension is highest. However, in the absence of externally applied twisting moments along the cable, the cable will not support a torque gradient over its length.

To seek uniform torque in the presence of a tension gradient, the cable will experience mid-span rotation (even though no rotation occurs at the cable ends). The rotation near the surface will be in the direction to reduce the tension-induced torque, and the rotation near the payload will be in the opposite direction to produce the opposite effect. The magnitude of the rotation will depend on the cable length, tension gradient, torque imbalance, and torsional stiffness.

This rotation will tend to make the torque uniform throughout the cable length. Thus, if the cable tension should go to zero so as to form a slack loop at any location, then there will be a potential for hockling to occur. In other words, if a nontorque-balanced cable has significant tension anywhere along its length, then hockling is possible even at remote locations if slack should occur.

The use of a swivel will not eliminate the possibility of hockling of a nontorque-balanced cable. The swivel may allow the cable to rotate at the payload so as to maintain zero torque, and depending on the cable length and torque imbalance, the swivel may experience a large number of turns. However, if the cable tension should suddenly drop to zero, the cable rotational inertia and hydrodynamic drag and the swivel friction may prevent the swivel from spinning back fast enough to maintain zero cable torque. The result may be a hockle if a slack loop should form. In addition, any cable rotation allowed by a swivel can be harmful to the cable as discussed below.

Even if a cable has been designed to have good torque balance, it may still develop some torsional energy if it has experienced any twisting. Induced twist can occur during the lowering or raising of a nonsymmetrical payload, by maneuvering of a tethered vehicle so as to accumulate turns in the cable, or by the cable handling techniques. For example, if a cable is deployed manually and is allowed to pull out of a coil which is lying on the deck, it will develop one turn of twist for each wrap in the coil. Similarly, a cable handling system which does not incorporate a drum, but which allows the cable to lie in a cage or basket, will produce one complete twist of the cable for each loop of cable in the basket. Depending on the diameter of the cable and on its inherent torsional stiffness, this twisting may be sufficient to produce a hockle if a slack loop should be allowed to form.

Whether a cable actually forms a hockle depends on the magnitude of the torque imbalance, the size of the slack loop, and the bending and torsional stiffness of the cable. For example, a cable with a high bending stiffness will require a huge slack loop and a high residual torque before the loop will close to form a hockle. Conversely, a very flexible cable may develop a hockle with a relatively small slack loop.

5.0 CABLE ROTATION

Cable rotation (twisting) has a number of adverse effects other than the potential formation of hockles. One of the major consequences of rotation is a reduction in cable breaking strength. This effect is most significant in cables having external contrahelical strength members arranged in either a braid or multiple layers. When a cable is rotated, the strength members which are wrapped in one helical direction are tightened, while those which are wrapped in the opposite helical direction are loosened. The resulting stress imbalance not only reduces the cable breaking strength, but also reduces the cable fatigue performance.

Cables which have high-modulus fiber strength members (such as KEVLAR) may exhibit a dramatic reduction in breaking strength as a result of small amounts of induced rotation. On the other hand, steel armored cables may be able to better accommodate small amounts of rotation because the ductility of the steel allows the wires in both helical directions to reach their yield point prior to rope failure in tension.

Another potential consequence of cable rotation is the rapid failure of conductors within the core. Most cables having a complex core design incorporate several layers of conductors which are typically assembled with alternately right-lay and left-lay helical directions. With this type of core design, no matter which way the cable rotates, some of the conductors will tend to tighten while the others tend to loosen.

Since cables having external strength members tend to become shorter no matter in which direction they are rotated, the conductors which tend to tighten will experience some strain relief due to shortening of the cable. However, those conductors which tend to loosen and develop excess length as a result of cable rotation will experience even more loosening due to shortening of the cable. In the extreme, the conductors may develop z-kinks which can rapidly lead to conductor or insulation failures.

If it is known that a cable will experience induced rotation in service, it is possible to design the cable to be twist tolerant. All conductor layers must have the same helical direction so that they will tighten and loosen together in response to cable rotation. Furthermore, if the direction of the cable rotation is known (such as the rotation induced by a cable handling system), the helical direction of the conductors must be in the direction that causes the conductors to be tightened as the cable rotates. Finally, the lay angle of each conductor layer must be carefully chosen to minimize the additional conductor strain induced by the cable rotation.

Extensive cable rotation tests have revealed that properly designed cables can survive many thousands of cycles of severe twisting without electrical or mechanical failure. Conversely, cables which have not been designed for twist tolerance may survive only a few cycles of moderate twisting. Cable design software (such as CABLE SOLVER developed by Tension Member Technology) can be used to establish the optimum helix angles for the conductor layers.

Of course, whenever possible, cable rotation should be avoided so as to achieve maximum cable breaking strength and fatigue performance. In some systems, it may be necessary to employ a swivel to decouple a torque-balanced cable from a turning payload. Conversely, it may be equally important to eliminate a swivel in a system which uses a nontorque-balanced cable with a stable and nonrotating payload. Regardless of the details of the service conditions for a specific cable, it is usually quite helpful for the cable to be manufactured with an obvious and permanent stripe positioned longitudinally along the cable jacket. This stripe will allow any cable rotation to be identified and quantified so that measures can be taken to minimize the number of accumulated turns.

6.0 CABLE BEHAVIOR IN BENDING

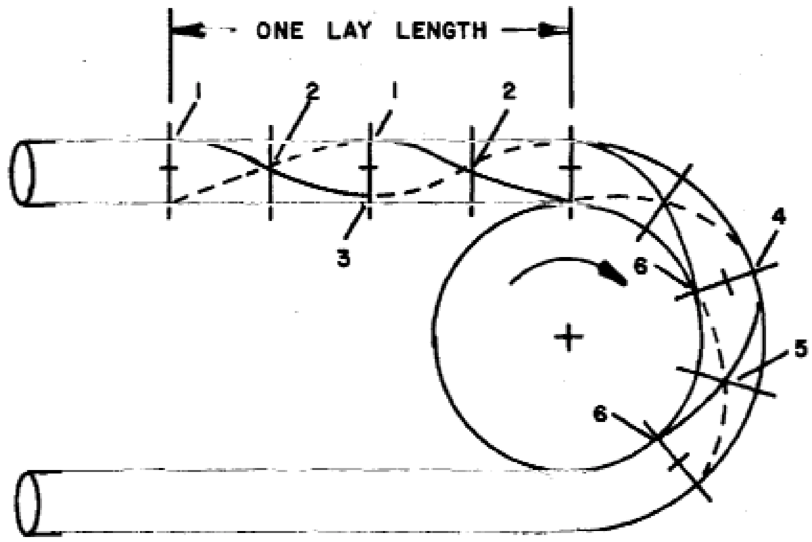
When a cable is subjected to combined tension and bending, the forces and motions imposed upon the individual elements are responsible for the deterioration and final retirement of the cable. It is useful to understand the factors which affect the magnitudes of these forces and motions so that cables may be designed and used property, thereby avoiding premature failure.

6.1 Element Motions During Cable Bending

Consider a cable which is passing over a sheave as shown schematically in Figure 8-3. In the straight portion of the cable, all elements within a given layer (for example, all outer armor wires) have precisely the same length within a given length of cable. Furthermore, if the straight portion of cable is divided into sections of equal length (for example, one-fourth lay length increments as shown in Figure 8-3), then the elements in one cable section have the same length as elements in another cable section. In other words, the length of an element between Positions 1 and 2 is the same as the length of that same element between Positions 2 and 3.

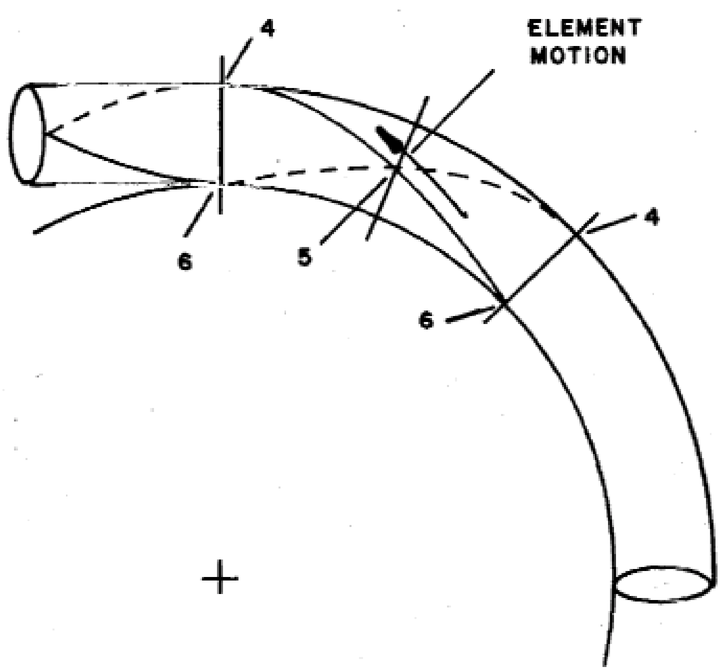
However, after the cable has been bent into a sheave, the length of an element from Position 4 to Position 5 is greater than the length of the same element from Position 5 to Position 6. Thus, in the process of being bent onto the sheave, the cable experiences relative motions among its individual elements to accommodate the distortion of the helical geometry. As the cable passes onto the sheave, the high contact forces at the sheave-to-cable interface prevent element motions in this region (Position 6). To accommodate the length differences described above, each cable element experiences a small amount of motion relative to adjacent layers in the vicinity of Position 5 as shown in Figure 8-4. Little or no motion actually occurs among elements located at Position 4.

To determine the magnitude of these element motions, a mathematical model was developed by Tension Member Technology. This model allows the analysis of any helical structure which is deformed to any desired bending diameter. The details of this analysis are beyond the scope of this chapter, but the results are summarized in Figure 8-5.



ELEMENT GEOMETRY IN A BENT CABLE

FIGURE 8-3

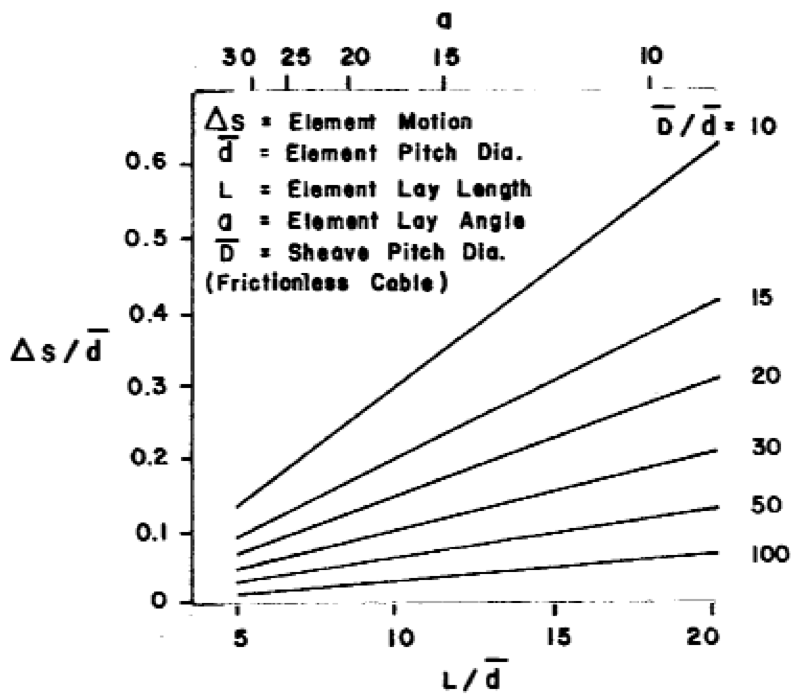


ELEMENT MOTION BY CABLE BENDING

FIGURE 8-4

Consider, for example, a cable having two layers of steel armor wires and an outside diameter of one inch. Assume that the diameter of each outer armor wire is 0.080 inch and, therefore, the pitch diameter of this layer of wires is 0.920 inch. Also assume that the lay length of the outer armor wires is 9.2 inches or 10 times the pitch diameter.

If this cable is bent over a sheave which provides a 27.6-inch bending pitch diameter for the cable (30 times the pitch diameter of the outer armor wires), then, as shown in Figure 8-5, the motion of each outer armor wire relative to the cable axis is approximately 0.10 times the pitch diameter of the outer wires or approximately 0.009 inch. Physical measurements of cable specimens during bending at zero tension (a condition which approaches the frictionless-cable model of the theoretical analysis) have confirmed that the actual element displacements are approximately the same as those predicted in Figure 8-5.



CABLE ELEMENT MOTIONS DUE TO CABLE BENDING OVER A SHEAVE

FIGURE 8-5

Within a given layer, the relative motion between any two adjacent elements becomes smaller with increasing numbers of smaller diameter elements in that layer. However, the motion of an element in one layer relative to an adjacent contrahelically wrapped layer is affected only to a small degree by the number of elements and the element diameters in each layer. All relative motions (both interlayer and intralayer) become smaller with increasing element helix angles.

Similar motions are exhibited to a greater or lesser degree by all helically wrapped elements within a cable, not just the strength members. It is this relative motion within and between element layers which provides a cable with its flexibility. However, these motions also give rise to the shearing forces and abrasive deterioration which leads to cable failure.

If all of the elements were locked together so that no relative motion could occur, then a cable would have a very high bending stiffness, and the elements would experience high tensile strains on the side of the cable away from the sheave throat and compressive strains on the side of the cable adjacent to the sheave throat. However, the mobility of the elements within the cable structure allows the excess element length on the side of the cable toward the sheave throat to make up for the deficiency in length on the side of the cable away from the sheave throat. As a result, the tensile stress remains much more uniform along the length of each element than would be the case if all elements were locked up so that no relative motions could occur.

6.2 Effects of Element Motions on Cable Strength Members

The consequences of element motions near Position 5 in Figure 8-4 can be observed in the laboratory during cyclic-bend-over-sheave fatigue tests of cables having braided Kevlar fiber strength members. During repeated bending over sheaves with normal operating tensions, such cables eventually fail mechanically due to fiber abrasion in the vicinity of Position 5, with little abrasion being apparent in the vicinities of Position 4 and 6.

Improved performance is obtained if a fiber strength member is applied in contrahelically served layers separated by low-friction isolation tapes (such as Mylar). The isolation tapes all but eliminate the layer-to-layer abrasion. Furthermore, since the individual fibers are so tiny and the number of fibers in each layer is enormous, the relative motion and, thus, the abrasive wear between adjacent fibers within a given layer is negligible.

In the absence of significant fiber wear within and between layers, outstanding cable flexure performance may be achieved.

In the case of cables with steel wire strength members, the relative motions within and between the armor layers contribute to some degree of wear which may remove the protective zinc coating on galvanized wires and, thus, promote the corrosive deterioration of the armor. However, wire armored cables usually do not exhibit a significant strength loss due to internal wear. Rather, they develop fatigue failures of individual wires due to variations in the tension, bending, torsion, and localized compressive contact stresses produced by cable tension, bending, and twisting.

Any reduction of element motions typically improves cable bending fatigue performance. An examination of Figure 8-5 reveals that the element motions can be reduced either by increasing the bending diameter of the cable or by decreasing the element lay lengths (increasing the element helix angles). An increase in helix angles is usually accompanied by a slight increase in cable diameter and a slight reduction in cable elastic modulus and achievable breaking strength for a given quantity of load bearing wire or fiber. The reduction in breaking strength is usually of no consequence, since the usable service life of a cable is determined by the residual breaking strength after some period of flexure cycling and not by the original breaking strength. Thus, the use of higher helix angles to improve fatigue performance is usually advantageous if the corresponding slight increase in cable diameter and elasticity can be tolerated.

The tension applied to a cable causes the helically wrapped elements to exert a radial force on the portion of the cable around which they are wrapped. This radial force acts in conjunction with the internal cable friction to retard the element motions described above. As a consequence of the friction forces within a cable, each element experiences a variation in tension along its length as the cable is bent. For example, referring to Figure 8-4, the portion of an element between Positions 4 and 5 experiences an increase in tensile loading as internal friction forces retard the motion of that element. Only in ideal frictionless cable would the tensile loading remain uniform along the length of a given element. Thus, a factor which contributes to cable deterioration during bending, in addition to element wear, is a variation in the effective tensile load experienced by each element. The resulting variation in tensile stress acts to accelerate fatigue crack initiation and propagation in metallic components. Of course, good cable lubrication to reduce these friction effects will improve cable bending performance.

Another consequence of the element motions and friction forces within a cable is a distortion of the geometry of that portion of cable immediately adjacent to a sheave tangent point. Because of the non-uniform tensile load distribution in the strength members around the cable circumference (higher element tensions away from the sheave, and lower element tensions toward the sheave), the cable does not remain a smooth circular cylinder, but develops a helical distortion or corkscrew over a short section of its length near the sheave. Close observation of a cable which is passing over a sheave will reveal a small standing wave at each sheave tangent point as a consequence of this helical deformation.

Cables elements with small helix angles experience greater motions and larger tension variations than do elements with larger helix angles. Therefore, cables containing elements with small helix angles often exhibit more obvious helical deformations during bending. In extreme cases, this helical deformation can lead to the circumferential migration and bunching of the strength members on one side of the cable. When this occurs, the inner layers of cable elements may protrude through the outer layers, and gross deformation of the cable structure may be the result.

The element displacement and friction forces described above also give rise to cable heating during flexure over sheaves. Such heating is of little consequence during normal deployment and retrieval operations where the cable may experience bending over several sheaves, but only infrequently. On the other hand, if a cable is subjected to repeated bending over the sheaves of a motion compensation system while at a high tension load, then the friction-induced temperature build up can be quite significant and, when added to any electrical resistive heating, can lead to accelerated failure of certain insulation materials. Cable heating can be reduced by the application of a lubricant to a cable to reduce the internal friction and to improve the heat transfer away from the cable.

6.3 Effects of Element Motions on Cable Core Components

The electrical conductors or optical fibers within the core of a cable also experience similar element motions and friction forces. However, in this case, the elements may have insufficient tensile strength to accommodate the induced element motions in the presence of high internal cable friction.

As a consequence, conductors may experience strains far in excess of their yield point between Positions 4 and 5 in Figure 8-4, while the same conductors may experience longitudinally compressive loads and z-kinking between Positions 5 and 6. Then, if the same section of cable should pass over another sheave so as to be bent in the opposite direction (a reverse bend), then that portion of the conductor which was previously strained beyond its yield point will be forced into longitudinal compression, while the adjacent section which was previously compressed will be strained beyond its yield point. The consequence will be rapid failure of the conductor and insulating materials. This type of deterioration is frequently observed in cables which incorporate small interstitial conductors at locations well away from the cable centerline.

On the other hand, larger power conductors which have ample tensile strength may be able to accommodate the, induced motions without rapid failure. However, even these conductors may eventually exhibit cup and cone tensile failures (rather than classical fatigue failures) as a result of the large strains induced in the conductors during cable bending.

There are several steps that can be taken to avoid premature failure of conductors which must be located at a significant distance from the cable centerline. One approach is to combine several small conductors together into a twisted pair, triad, or quad so that the assembly has ample extensibility to accommodate the length changes imposed by cable bending without exceeding the yield point of the conductor material. If it is not possible to combine small conductors into complexes, then it may be necessary to use a high conductor helix angle so as to minimize the motions within the cable structure or to fabricate an elastic conductor by wrapping one or more layers of copper filaments around a small diameter nylon rod.

In any case, careful attention must be paid to the design of a cable core so that all elements have sufficient strength, elasticity, and helix angle to accommodate the deformations which occur as the result of cable bending. Also, all core materials should be selected with due consideration given to their friction characteristics, since low element-to-element friction will enhance the bending performance. In this regard, the cable void-filling material should not “glue” the core elements together so as to retard element motions.

6.4 Cable Strength Reduction Due to Bending

Cable breaking strength is an important parameter of most cable design specifications, and the strength is usually determined by a tensile test of a straight cable specimen. For many cable applications, this approach is adequate. However, in some cases, it may be important to know the cable strength under conditions of bending over the sheaves and/or drum of the cable handling system, since the maximum cable tension usually occurs at this location.

Laboratory experiments have shown that if a cable is wrapped around a sheave with both ends attached to a loading plate and is then pulled to failure (without sheave rotation), the cable will break at one of the sheave tangent points (at a point where the cable becomes bent onto the sheave) and with some reduction in breaking strength. This reduction may be significant if the sheave is small. If the same type of cable is pulled to failure while it is <u>moving</u> over the same sheave (such as during retrieval of a payload), the reduction in breaking strength will be even greater.

In the static case, the cable is bent over the sheave at essentially zero tension prior to being pulled to failure. As the cable is initially bent, all internal elements are able to move as necessary to accommodate the bend without their motions being retarded by high internal cable friction. Then, during the break test, the load sharing among all elements is fairly uniform, and the cable breaking strength approaches that of a straight cable (unless the sheave is quite small).

In the dynamic case, the cable experiences continuous bending as the sheave rotates. As the tension increases, so does the internal cable friction which retards the element motions induced by the bending. The result is a non-uniform tension distribution among the cable elements and a reduction in the achievable cable breaking strength. During laboratory experiments, this strength reduction has been as great as 30 percent for some cables. Usually, cables with small helix angles for the strength members exhibit the largest strength reduction during bending. Again, smaller sheaves produce more strength loss.

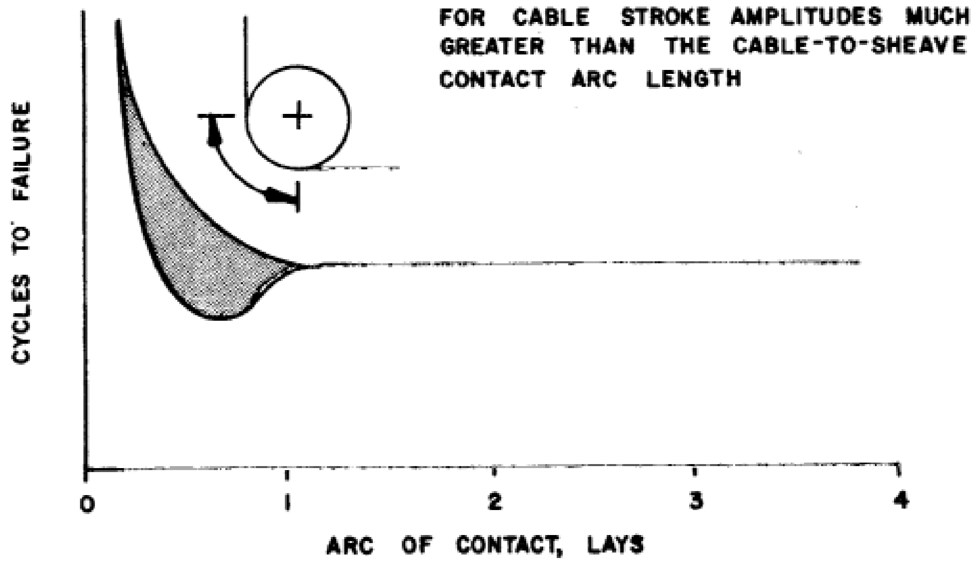
6.5 Effects of Cable Wrap Angles on Sheaves

All of the bending-induced changes in stresses and relative motions among the cable components as described above take place in the immediate vicinity of the cable-to-sheave tangent point. Because of the internal friction within the cable structure, the affected portion of the cable is relatively short. In other words, portions of the cable which are a short distance away from the sheave, or portions of a cable which are on the sheave a short distance away from a sheave tangent point, experience no changes in internal stresses or motions and are not influenced by the bending of other portions of the cable.

If the arc of contact between the cable and sheave exceeds approximately one lay length, then there will be a certain portion of cable in contact with the sheave which, having undergone stress changes in the vicinity of one tangent point, will experience no further changes in its state of stress until it approaches the second tangent point. Thus, for typical deployment and retrieval operations, the bending fatigue life of a cable is not influenced by the wrap angle on a sheave as long as at least one lay length of the cable is in contact with the sheave. (See the following Section 7.0 for special considerations regarding cable wrap angles on sheaves.)

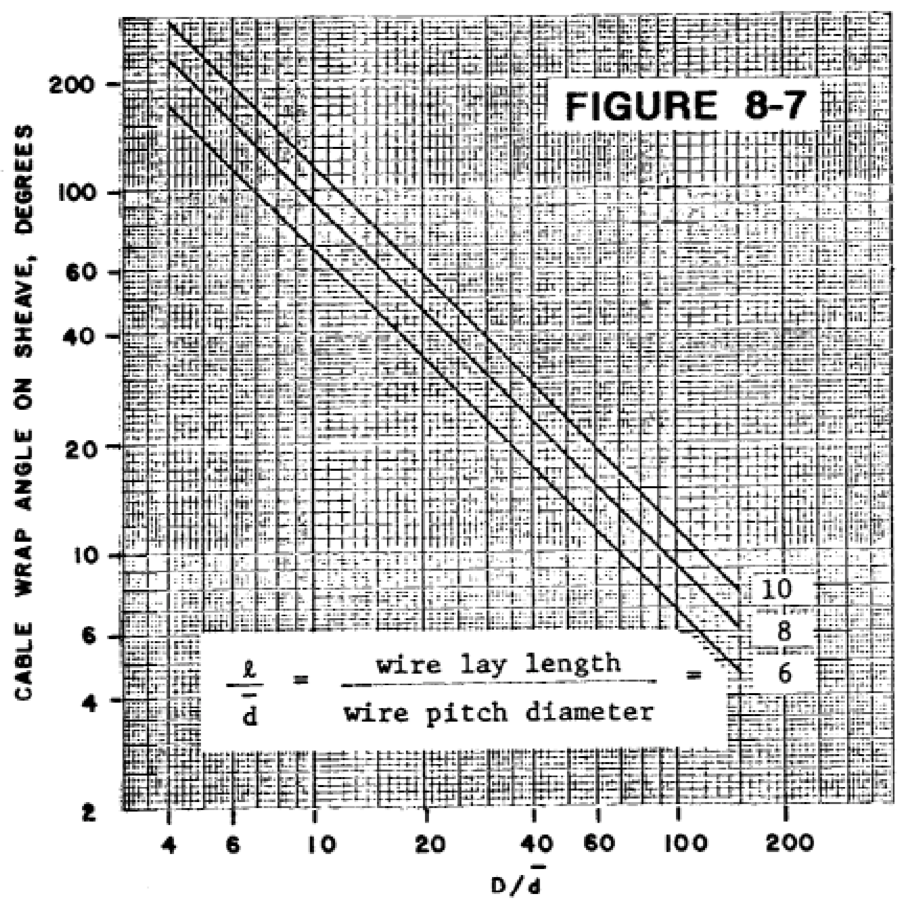
Cables which are deployed and retrieved through a series of fairlead sheaves will have a bending fatigue life which will be the same regardless of the cable wrap angles on the sheaves, at least for contact arcs equal to one or more lay lengths of the strength members. For a contact arc of less than one lay length, the bending fatigue damage produced by a sheave is typically less, but there are notable exceptions to this rule. Depending on the specific cable design, the sheave-to-cable diameter ratio, and the operating tension, a cable contact arc of one-half lay may be more damaging than a longer contact arc. This behavior is shown graphically in Figure 8-6.

Figure 8-7 shows the cable wrap angles on a sheave required to produce an arc of contact equal to one lay length for various sheave diameters. This graph assumes that the cable has a external strength member and that the outer layer has a lay length equal to six, eight, or ten times its pitch diameter (a helix angle of 27.64, 21.44, or 17.44 degrees, respectively):



EFFECT OF SHEAVE CONTACT ARC ON CABLE BENDING FATIGUE LIFE

FIGURE 8-6



WRAP ANGLE OF A CABLE ON A SHEAVE CORRESPONDING TO A CONTACT ARC OF ONE LAY LENGTH OF THE OUTER LAYER STRENGTH MEMBERS

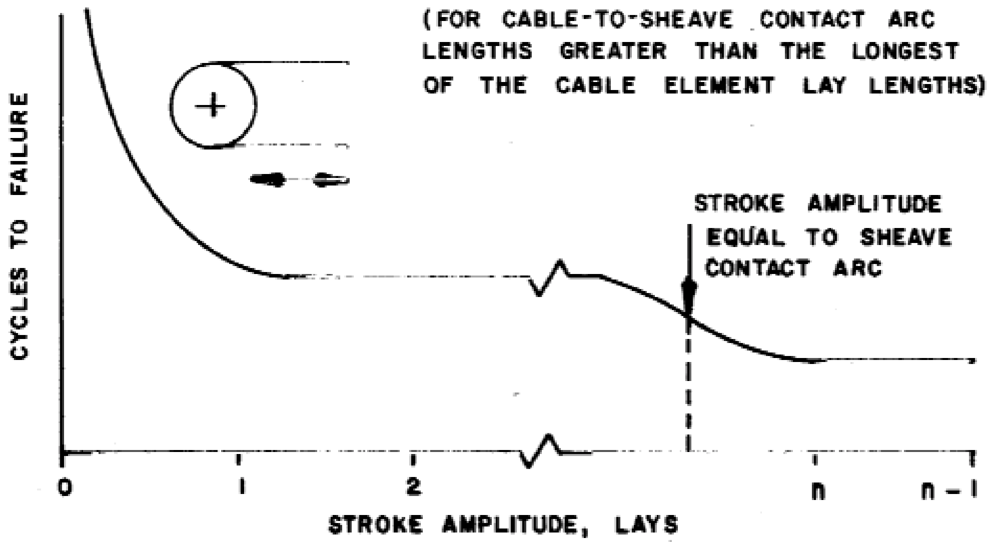
One of the important conclusions which can be drawn from these considerations is that a sheave diameter should not be arbitrarily reduced just because a cable happens to have a relatively small wrap angle on that sheave. A single, small deflection sheave or roller can produce more cable damage than all of the other sheaves in the fairlead system. Even worse is the replacement of a sheave with a series of small rollers in the interest of saving space. This procedure can quickly destroy a cable which supports any significant tensile load.

6.6 Effects of Cable Stroke Amplitude Fatigue Life

The above discussions assume that the cable is being deployed and retrieved with a stroke amplitude that is quite large. In this case, the cable experiences two straight-bent-straight bending cycles at each sheave, one during deployment and another during retrieval.

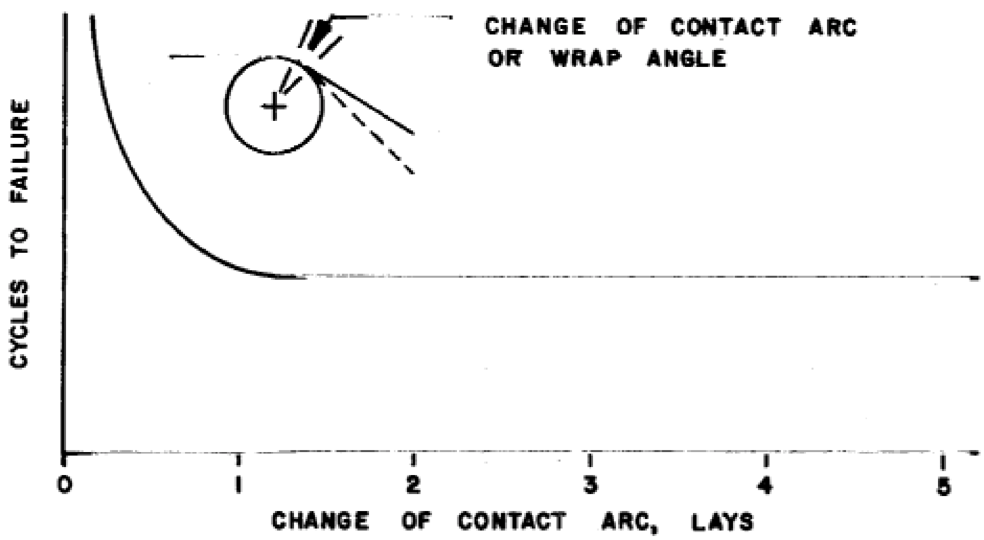
If the cable stroke amplitude is diminished to less than the cable-to-sheave contact length, then the cable will experience only one straight-bent-straight bending cycle during a complete deployment and retrieval sequence, and the cable life in the system will be essentially doubled. This behavior is shown in Figure 8-8. For the case of a large cable-to-sheave wrap angle, the cable fatigue life will remain the same with decreasing stroke amplitude until the stroke amplitude equals approximately one lay length of the outer strength members. For even shorter stroke amplitudes, the cable fatigue life will increase further.

Consider the case of a tow cable which makes contact with an overboarding sheave as shown in Figure 8-9. Ship motions will cause the cable to experience continuous flexing at the sheave. For relatively calm conditions, the length of cable involved in the flexing may be less than one lay length of the outer strength members, and the cable fatigue life may be quite good. However, if the bending zone of the cable approaches one lay length, then a condition of full bending will be experienced, and the cable may rapidly accumulate fatigue damage. Figure 8-7 may be used to determine the change of cable wrap angle corresponding to a bending zone of one lay length of the outer strength member.



EFFECT OF CYCLING STROKE AMPLITUDE ON CABLE BENDING FATIGUE LIFE

FIGURE 8-8



EFFECT OF BENDING AMPLITUDE AT OUTBOARD SHEAVE ON CABLE BENDING FATIGUE LIFE

FIGURE 8-9

7.0 MOTION COMPENSATION SYSTEMS

If a cable is to be used under severe dynamic conditions, some type of motion compensation system may be required to decouple the motions of the host vessel from those of the payload. Two common types of systems are the bobbing boom and the ram tensioner.

7.1 Bobbing Boom Systems

In the bobbing boom system, the cable passes over a sheave that is located at the end of an articulated boom. The position of the boom is controlled by a hydraulic ram and pressurized accumulator system, and the boom bobs up and down in response to vessel motions so as to maintain a more or less constant cable tension. This type of system is least damaging to the cable since only a short cable section experiences repeated bending of the type shown in Figure 8-9. By periodically changing the position of the cable by only a few feet, it is possible to distribute the cable wear so as to prolong cable life.

7.2 Ram Tensioner Systems

In a ram tensioner, the cable passes over one or more sheaves that are connected to a hydraulic ram and pressurized accumulator system. The ram provides a pneumatic spring which acts to maintain the cable tension at a more or less constant value as the cable strokes in and out in response to vessel motions. In this type of system, special consideration must be given to cable wrap angles and sheave spacing.

Obviously, the system should employ as small a number of sheaves as possible to minimize the bending fatigue damage to the cable. Furthermore, these sheaves should be spaced as far apart as necessary to assure that no single section of the cable comes into contact with more than one sheave during each heave cycle. Should a section of cable pass over two sheaves with each heave cycle, the total bending fatigue life of the cable will obviously be one-half of that which could be achieved if the sheaves were further apart.

Another important factor which influences the total achievable cable fatigue life is the arc of contact between the cable and each sheave in the fairlead system. (It should be noted that the previous discussion of cable contact arc effects applied to typical deployment and retrieval operations and not to motion compensation.) During active motion compensation, if the cable stroke amplitude is less than the cable arc of contact with a sheave, then each heave cycle will produce one straight-bent-straight cable bending cycle.

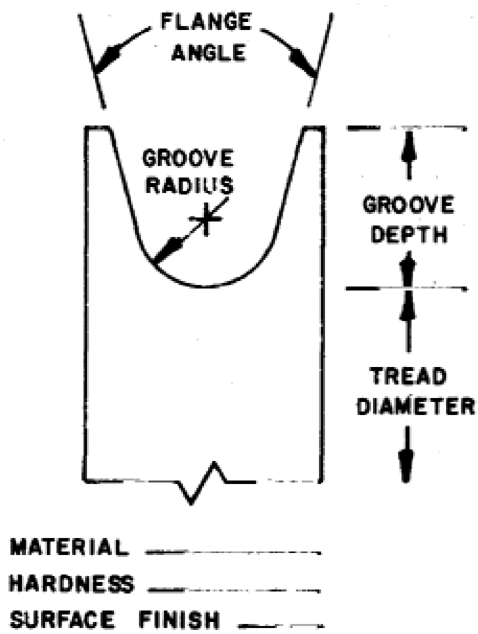
If, on the other hand, the amplitude of cable motion should exceed the length of the cable arc of contact on the sheave, then a section of the cable will pass onto, completely around, and off of the sheave as the cable strokes in one direction, and it will return to its original position as the vessel completes one heave cycle. In this case, a section of the cable will receive two straight-bent-straight bending cycles during each heave cycle.

Thus, a cable wrap angle of 180 degrees will allow cable motions (heave amplitudes) twice as large as could be accommodated with a wrap angle of 90 degrees before each heave cycle produces two cable bending cycles. In the long term, a motion compensation system which employs larger cable wrap angles will accumulate fewer cable bending cycles and will enjoy a longer cable service life.

8.0 SHEAVES FOR CABLES

The sheaves in the cable handling system should be as large in diameter as practical to maximize cable service life. In addition, the sheave grooves must be smooth, and the groove diameter should be the same as the cable diameter when measured at zero tension. A sheave groove which pinches the cable or which fails to support the cable properly will diminish the cable bending life. A cable should never be used on a sheave grooved for a cable of larger diameter. It is also important that all sheaves are properly aligned so that the cable experiences little or no fleet angle. Any sheave misalignment will cause cable wear due to rubbing on the sheave flanges. Figure 8-10 shows the various factors to be considered during sheave design.

Sheave flanges for many applications typically have an included angle of approximately 30 degrees. This configuration provides adequate cable support and will accommodate small cable fleet angles without causing unusual cable wear. (The fleet angle is the departure angle between cable and the plane of the sheave.) In some situations, however, it may not be possible to avoid a large fleet angle. For example, a vehicle tether cable may experience large out-of-plane motions at an overboarding sheave due to changes in the relative positions of the vehicle and the support vessel. In this situation, a large flange angle may be required to prevent the cable from coming into contact with a potentially sharp edge at the top of the sheave flange.



SHEAVE DESIGN PARAMETERS

FIGURE 8-10

A sheave groove depth of one cable diameter is usually satisfactory for many applications. However, deeper grooves may be needed to assist in reeving a cable through a handling system or to assure that the cable does not come out of the grooves during extreme operating conditions. If a cable has even a small fleet angle at a sheave, special attention must be paid to both the groove depth and the flange angle. Otherwise, the cable may come into contact with the top edge of the flange. For example, a sheave flange half angle of 15 degrees does not imply that a fleet angle of 15 degrees can be accommodated. The actual maximum fleet angle will be a function of both the sheave-to-cable diameter ratio (Did) and the groove depth. The larger the Did ratio, the greater the required groove depth to avoid cable contact with the top edge of the flange.

The sheave diameter which is appropriate for a given system depends on the details of the cable design, the severity of the operating tensions, the desired cable service life, and the consequences of a cable failure. Some cable designs and operational requirements demand large sheaves, while others allow smaller sheaves to be used. In selecting sheave sizes it should be remembered that small, increases in sheave diameter can produce dramatic increases in cable life.

Plastic lined or nylon sheaves may offer some advantage for steel wire armored electromechanical cables in terms of cable wear and wire-to-sheave contact stress. However, in situations where a highly loaded cable repeatedly passes back-and-forth over a sheave in a motion compensation system, especially when transmitting large amounts of electrical power, it may be advantageous to avoid the use of nonmetallic sheaves or sheave liners because of cable heating considerations. A metal sheave can act as a heat sink to reduce cable heating under these conditions.

9.0 CABLE REEVING CONFIGURATIONS

In the simplest system, the cable is deployed directly from its storage drum without passing over any sheaves or through any guide rollers. Other systems require relatively complex reeving configurations where the cable must pass over a number of sheaves. Regardless of the simplicity or complexity of the system, safety considerations should not be neglected. Whenever possible, personnel walkways should be designated away from the cable system to avoid having the cable pass near or through a commonly used walkway. An equally important consideration is the recoil path the cable will have in the event of a catastrophic failure. Cable recoil can inflict serious damage and injury at locations well away from the normal route of the cable. Where necessary, barricades should be erected to absorb the energy of a recoiling cable.

Cable systems vary not only in their complexity, but also in their frequency of use. Some systems require a cable to be deployed and retrieved relatively infrequently, while other systems may subject the cable to nearly continuous load and flexure cycling. In any case, there are a number of system design guidelines which will improve cable service life.

Of course, the number of sheaves in the system should be kept to a minimum whenever cable flexure life is a concern. Also, the greater the number of desired cable flexure cycles, the greater will be the required sheave diameters and operating safety factors.

All components of a cable handling systems should be arranged so as to minimize the cable fleet angles at the drum and sheaves. A large fleet angle can lead to cable mis-spooling on the drum or cable wear due to rubbing against adjacent wraps on a drum or against sheave flanges. A large fleet angle is also detrimental because it can produce a small-radius cable bend at a sheave flange in a plane perpendicular to the plane of the sheave. These small-radius bends are potentially as damaging to the cable as small sheaves.

Whenever possible, the cable routing from the drum and through the various sheaves should be chosen so as to eliminate reverse bending of the cable. A cable which is bent in the same direction over two sheaves will have considerably better service life than if it is subjected to a reverse bend over the same two sheaves. While it is sometimes impossible to avoid reverse bending of a cable, the consequences of this reeving configuration must be recognized.

It is a common misconception that a cable can be routed in a large-radius arc over a series of small-diameter rollers without affecting the performance which would otherwise be obtained by use of a single sheave of the same radius. While this arrangement may be acceptable for cable tensions near zero, the small rollers may severely damage the cable at normal operating tensions. Each roller wilt subject the cable to a severe bending condition even though the cable wrap angle at each roller may be very small. Thus, it is always advantageous to eliminate guide rollers whenever possible in favor of sheaves having the proper geometry.

Finally, it is important that a moving cable not come into contact with any stationary structure. In addition to possible abrasive damage to the jacket or strength member, steel armor wires may experience sufficient frictional heating to form a very thin layer of untempered martensite on the outer surface at the contact location. Martensite is very hard and brittle, and it will develop small cracks as soon as the cable is subjected to any significant tension or bending. These cracks will then propagate rapidly through the remainder of the wire cross section to produce premature wire failures.

10.0 CABLE WINDING ON DRUMS

For those systems in which the cable tension is always quite low at the drum (systems employing some type of traction winch), the cable life will probably not be influenced to a great extent by the details of the drum design. However, for those systems which require the cable to be wrapped on the drum under high tensions, the drum can be a major source of cable damage.

The factors which affect cable life are the drum diameter, the number of cable layers, the type of grooving, and the uniformity of winding. The influence of these factors can vary from one installation to another. However, the cable will usually benefit from a large drum diameter, proper grooving, and the use of a level-wind system to achieve smooth winding.

If a cable must be wound on a drum in multiple layers and must also sustain a significant tension load, cable damage may occur due to crushing of the bottom layers on the drum, localized pinching and bending due to uneven winding at the drum flanges, or "cutting in" where the outer wrap of cable becomes buried within the inner wraps in response to a high tension load. The potential for crushing of the inner cable layers can be minimized by using a large diameter drum which reduces both the radial force developed by the cable and the required number of cable layers. The drum should also be properly grooved. The use of riser and filler strips at the drum flanges will reduce the potential for localized cable damage at the flanges where the cable rises from one layer to the next. Also, the potential for cutting in of a cable can be reduced if the cable is wound on the drum so that the inner layers have a high tension.

Proper winding of a cable will usually require some type of level-wind system to guide the cable onto the drum. Synchronization of the level-wind mechanism with the cable lead on the drum is critical to avoid mis-spooling. Also, care must be taken in the design of the level-wind sheaves or rollers to assure that they are not a source of premature cable damage.

If no level-wind system is used, the first fixed-position sheave must be positioned far enough away from the drum to limit the cable fleet angles (usually to less than 1-1/2 degrees). Excessive fleet angles can produce mis-spooling on the drum and also cable wear due to rubbing on sheave flanges or against adjacent cable wraps on the drum.

To allow the cable tension to be maintained at a low value on the drum, some type of traction winch can be used. A common system uses a double-drum capstan as discussed elsewhere in this handbook. In this system, the cable passes over a pair of grooved drums in a series of half wraps. One or both of the drums is driven electrically or hydraulically, and the friction between the cable and the drum grooves allows a significant tension gradient to be developed, thereby allowing the cable to be wound on the storage drum at a low tension. (An electromechanical or fiber optic cable should not be used on a single, flat-faced type of capstan such as is often used for mooring ropes.)

11.0 CABLE VOID FILLERS

It is important for any electromechanical cable to be properly void filled to minimize the change in diameter during tensile loading and the associated constructional elongation. However, for any cable which is to be operated over sheaves, the void fillers used within the core must not be of the type which remain liquid, no matter how high the viscosity may appear to be. Repeated cycling of the cable over a sheave will cause such void fillers to be milked away from the sheave contact zone due to the increased pressures produced by cable contact with the sheave. The void filling material will then accumulate just beyond the sheave contact zone and will produce bulging of the cable or even total rupture of the core jacket. Elastomeric void fillers, such as DPR, are much preferred.

Also, as discussed earlier, the cable void-filling material should not “glue” the core elements together so as to restrain element motions during cable bending. The consequence will be an amplification of the strain experienced by each core element and premature failure of core conductors or optical fibers.

12.0 CABLE TERMINATIONS

The ideal cable termination restrains the cable core and external strength members in such a manner as to duplicate the stress distribution in the cable elements which would be present in an undisturbed continuation of the original cable. Compression of the cable core by the strength members when the cable is under tension prevents the core tram slipping longitudinally inside the cable and allows the core to extend through the strength member termination without affecting the stress distribution in the strength members.

Four basic types of cable terminations are in common use. The drum-grip termination is simplest in concept. It consists of a wide sheave having either a flat face or a helical, conformal groove upon which are wrapped several turns of cable. The friction between the cable strength member and the drum face provides a means for transferring the stress in the cable to the grip. A portion of the cable tension is transferred to the drum for each wrap of cable, and the low-tension end of the cable is anchored with a suitable secondary termination which can accommodate the lower tension level. The drum grip is particularly effective for steel-wire armored cables, and it may be used to provide a means of easily reterminating a cable in the field.

To achieve a high termination strength efficiency using a drum grip, the same drum geometry requirements as mentioned for sheaves must be met; i.e. a large drum-to-cable diameter ratio, a groove diameter equal to the cable diameter at zero tension, and a small fleet angle. (Termination efficiency is defined as the ratio of the terminated cable breaking strength to the mid-span breaking strength achievable with ideal terminations, expressed as a percent.) Drum-grip terminations with high strength efficiency are usually large in diameter and are relatively heavy.

Termination efficiencies of near 100 percent are achievable with drum grips for steel wire armored cables without external jackets. However, jacketed cables can encounter problems when terminated with drum grips. If the coefficient of friction between the cable strength member and the jacket is less than the coefficient of friction between the jacket and the face of the drum grip, the strength members may slip inside the jacket. Then, upon repeated load cycling, the entire load will eventually appear at the secondary termination resulting in cable failure at that location. (If the secondary termination is capable of handling the entire load, then the drum grip is superfluous.) This same jacket slippage problem can occur in systems utilizing traction sheaves, and total jacket delamination may be the final result.

The resin-filled .socket termination is a proven technology used successfully with steel wire armored cables, and termination strength efficiencies of 100 percent are commonly achieved. However, for the case of cables having non-metallic fiber strength members, high strength efficiency can be routinely achieved only with very small cables with breaking strengths of a few thousand pounds. For larger cables, strength efficiencies of as little as 60 percent are often encountered.

A factor contributing to the low strength efficiency of resin terminations when used on Kevlar fiber strength members is the fact that, unlike steel wires which can yield under tension and allow all wires to share the load, Kevlar fibers fail without yielding. Thus, careful preparation of the Kevlar before pouring the resin in the socket is essential for good fiber load sharing.

External compression-type terminations apply radial compression over some length of the cable and transfer the stress in the cable tension elements to some type of external termination. Woven wire mesh grips ("Chinese finger" grips), single-layer and double-layer helical wire grips, and split-pipe grips fall into this termination category: They are quite effective on steel strength member cables and may work well on externally jacketed cables if the coefficient of friction between the jacket and the strength members is high enough. If this is not the case, the termination and a section of the jacket may pull off of the cable at a rather low tension.

In general, external, compression-type terminations are not suitable for cables with multiple layers of non-metallic fiber strength members, especially cables designed with low internal friction to provide good bending fatigue life. In this case, the friction between layers may not be sufficient to allow load transfer to take place from layer to layer so as to provide uniform loading of all fibers.

Cables with non-metallic fiber strength members may be successfully terminated with special splicing techniques. The TMT Braid-Splice Termination uses a separate fiber eye assembly which is spliced into the end of the cable, and it can be used even with cables fabricated with low-friction fiber finishes. This splice requires some rearrangement of the geometry of the strength member fibers at the end of the cable to form a special braided geometry. Although time consuming to install, this termination is light in weight and typically provides a strength efficiency of 100 percent (tensile test specimens break mid span). The splice geometry must be carefully engineered for each specific cable to provide uniform core compression over the length of the splice to avoid damage to the cable core.

13.0 CABLE FAILURE MECHANISMS AND RETIREMENT CRITERIA

Cables used in the ocean have several modes of failure which may occur if the cable does not first encounter accidental damage such as entanglement with propellers or slipping off of the sheaves of the handling system. One common mode of failure is tensile overload due to snap loading induced by the dynamics of ship motions, such as during deployment or retrieval of a payload. Snap loading can induce cable tensions many times greater than the nominal operating tension, and it can produce slack loops and the potential for hockling and kinking.

Internal cable failure mechanisms include:

(1) Abrasive wear of strength member components

(2) Fatigue failure of metallic strength members

(3) Circumferential migration of non-metallic strength members

causing cable cork screwing

(4) Breakage or shorting of electrical conductors

(5) Breakage of optical fibers

(6) Thermal damage of the cable due to resistive heating and/or heating due to continual cycling over sheaves within motion compensation equipment

The potential for cable failure due to resistive heating must not be underestimated. The conflicting requirements of neutral buoyancy, small diameter, high strength, and high power capability often result in cables which operate at elevated temperatures. Usually, once the cable is underwater, the heat dissipation into the water is sufficient to keep the internal temperatures within acceptable limits. When the cable is in air or is wound on a drum, severe heating problems often occur.

Cables with non-metallic fiber strength members are particularly susceptible to thermal damage because of the low thermal conductivity of the strength member and the cable jacket. One means of partially relieving this problem is to impregnate the fiber with a grease or other thermally conductive material. However, the lubrication effect of the grease can increase the probability of fiber migration and corkscrewing of contrahelically served strength members.

Since many cables are retired from service following, rather than prior to, some cable failure, it is desirable to limit the cable damage to a localized area. The failure of communication or control system elements in a cable may scrub a mission, but the payload can usually be retrieved by means of the cable strength member. If an electrical or optical failure occurs near one end of the cable, and particularly if it is due to an external cause rather than general internal wear, cutting off the damaged section and retermination of the cable is a reasonable approach. This technique also applies when opens in the power conductors cause loss of power to the vehicle.

A serious type of cable failure is shorting of the power conductors when the system does not have adequate safeguards to prevent additional cable damage. Cables particularly susceptible to thermal heating damage during short circuits are those which use several power conductors in parallel to achieve the required conductor cross-sectional area for one phase of a circuit. For example, if three power conductors are used in parallel to carry 15 amperes and are protected by a single 15 ampere circuit breaker, shorting of one of the three conductors to a return conductor at a damage site can cause that conductor to carry most of the current with little current being tarried by the two remaining power conductors. This type of fault may not trip the circuit breakers and may allow the insulation on one conductor to be thermally damaged along the entire length of the cable between the power source and the short circuit. This damage is likely to force early retirement of the cable.

Power systems should be designed to accommodate shorts and opens in power conductors without causing any additional local damage such as arcing at the location of the cable short circuit. This approach can prevent additional damage and will allow a failure analysis to be performed on the damaged section. The addition of conductive blocking compounds and drain wires to the cable core allows the use of ground-fault detector circuits at the power source. These circuits disconnect the power to the cable upon detection of electrical leakage above a predetermined level to either sea water or the cable drain wires. This system prevents power surges from passing through a shorted section of cable and heating the entire length of the power conductors sufficiently to thermally damage the conductor insulation.

The importance of cable failure analysis cannot be over emphasized. After any failure, a section of cable including the failure location should be saved for analysis. The end toward the payload should be marked, and a cable map prepared showing the location of the failed section in relationship to the handling system sheaves. The cause of failure, if known, the sea state, and other operating conditions should be recorded.

It is important to determine whether a failure is due to externally or internally induced cable damage. If externally induced, then an examination of the operational procedures is in order. If internally induced, the cable may be worn out or have design deficiencies which make it unsuitable for use under existing conditions. A change in operational procedure may reduce the cable stresses to a level which will allow the cable to perform satisfactorily.

14.0 SPECIAL CONSIDERATIONS FOR WIRE AND NON-METALLIC ROPES

The preceding portions of this chapter are directed at various types of cables which combine power and/or data transmission elements with flexible metallic or non-metallic strength members. Much of this information can be applied to metallic and nonmetallic ropes, as well. However, there are a number of special considerations for ropes which deserve additional discussion. The remainder of this chapter is directed at ropes used in the ocean environment.

15.0 TYP1CAL ROPE CONFIGURATIONS

Wire ropes typically have one or more layers of helically wrapped strands, each of which is made up of one or more layers of helically wrapped wires. The core of the wire rope, if any, may be a natural or synthetic fiber rope, a strand similar to one of the main rope strands, or an independent wire rope core (IWRC).

Many wire rope materials and designs are available with a variety of physical characteristics in terms of strength, flexibility, torque and rotation balance, abrasion resistance, and corrosion resistance. Examples of common wire rope configurations appear in Figure 8-11.

It is recognized that the term “wire” is often used in the field to identify a wire rope. However, for the purpose of this chapter, the term “wire” will apply to an individual metal wire that is used as a component of a more complex rope structure.

Ropes made of high-modulus, non-metallic fibers are a fairly recent development, arid their designs continue to be refined as test and field performance data are accumulated. Many of these ropes are similar in geometry to wire ropes, while others incorporate special design features to enhance their strength and bending-fatigue performance.

16.0 ROPE TORQUE

With the exception of 3-strand torque balanced ropes and certain ropes with multiple layers of strands, most metallic and non-metallic ropes develop a significant torque when loaded in tension.

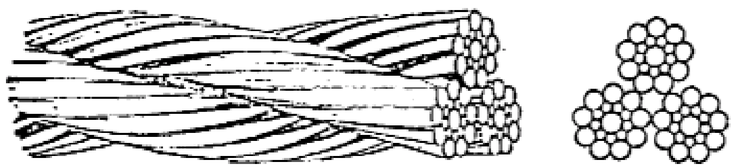
The torque produced by a rope is due to both the helix of the main strands and the helix of the individual strength members (wires or non-metallic fiber bundles) within the strands. In a regular lay construction (which has opposite helix directions for these rope components), the net rope torque is lower than for a Lang lay construction (which has the same helical direction for all components).

The torque produced by 6-strand, fiber-core wire ropes can be approximated by:

$$\text{Regular Lay Rope Torque} = 0.55\frac{d}{L/d}T \quad \text{(inch - pounds)}$$

$$\text{Lang Lay Rope Torque} = 0.91\frac{d}{L/d}T \quad \text{(inch - pounds}$$

where: d = nominal rope diameter, inches
L = rope lay length, inches
T = rope tension, pounds.



3 x 19 Searl Right Regular Lay

6 x 19 Searl Right Regular Lay, IWRC

18 x 7 Non Rotating, Fiber Core

TYPICAL WIRE ROPE CONSTRUCTIONS

FIGURE 8-11

For many working wire ropes, the ratio L/d is in the range of 6.25 to 6.5.

Whenever two ropes are connected together, they should be of similar diameter and construction, and if not, they should have similar torque versus tension characteristics. Otherwise, the connection will rotate as the two ropes seek the same torque value. This rotation will adversely affect the strength and fatigue properties of both ropes, especially ropes constructed from high modulus, non-metallic fibers.

17.0 ROPE HOCKLING AND KINKING

The discussion early in this chapter about hockling and kinking of cables is applicable to ropes, as well. However, since most ropes develop considerably more torque than do most cables, the potential for rope hockling and kinking is much greater. The reader is encouraged to review the earlier discussion of this subject.

18.0 ROPE ROTATION

Many ropes exhibit huge amounts of rotation if one end is allowed to turn. Generally, the rotation is in the direction to loosen the outer layer of strands and increase the lay length of this strand layer.

If a rope has more than one layer of strands, any rotation will typically have a major effect on the load distribution among the strand layers. For example, if a specimen of 19x7 spin resistant wire rope is allowed to rotate freely as it is pulled to failure, the breaking load will be approximately 30 percent lower than that of a specimen pulled to failure with the ends restrained to prevent rotation.

Rotation of a regular lay wire rope tends to tighten the outer wires of each strand, while rotation of a Lang lay rope tends to loosen the outer wires. This stress redistribution within each strand can adversely affect the rope breaking strength and fatigue performance. Furthermore, the wire looseness in a Lang lay rope can lead to “secondary bending” of the outer wires as the rope passes over a sheave. This secondary bending can lead to premature wire fatigue failures.

The use of a swivel with a rope is sometimes required to decouple the rope from a spinning load. However, all ropes provide the best performance if used with the ends restrained from rotation.

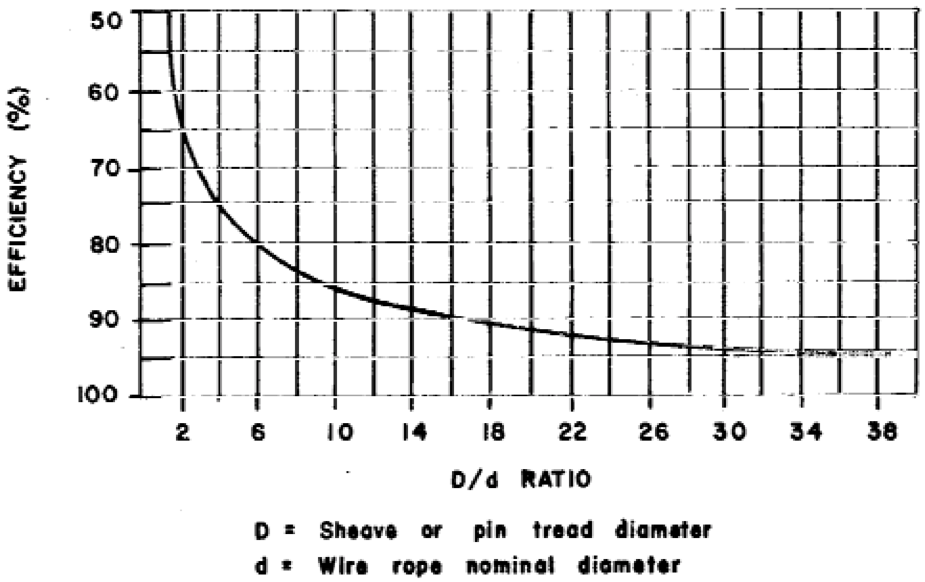
19.0 ROPE BEHAVIOR IN BENDING

The discussions presented earlier in this chapter about the behavior of cables in bending also apply generally to ropes. The reader is encouraged to review this information to develop a basic understanding of rope behavior. The following comments amplify this information as it applies specifically to ropes.

20.0 ROPE STRENGTH REDUCTION DUE TO BENDING

As in the case of cables of all types, ropes also exhibit a reduction in breaking strength when pulled to failure while wrapped around a sheave or pin. The strength reduction is greater with smaller bending diameters. It is also greater if the rope is moving over a rotating sheave than if it is stationary over a non-rotating sheave.

Approximate strength efficiencies achievable with 6x19 and 6x37 Class ropes appear in Figure 8-12. In this case, the rope is assumed to be loaded symmetrically around the sheave or pin, with no rope motion relative to the sheave or pin.



APPROXIMATE STRENGTH EFFICIENCE OF WIRE ROPE WHEN BENT OVER SHEAVES OR PINS OF VARIOUS SIZES

FIGURE 8-12

21.0 SHEAVES FOR ROPES

An earlier section of this chapter discussed the design of sheaves for use with various types of cables. This same information also applies to sheaves used with wire and non-metallic ropes. However, there are a few additional considerations which apply specifically to ropes.

Since the diameter tolerance for a new wire rope can be as great as five percent over the nominal rope diameter, sheaves for wire ropes are generally designed to have a groove diameter approximately five percent larger than the nominal rope diameter to avoid pinching the rope.

To achieve the longest possible operating life for both wire ropes and sheaves, steel sheaves should be hardened to avoid wear and changes of groove shape. Unhardened sheaves can become "corrugated" so as to develop a wear pattern which can accelerate the wear of the rope.

Non-metallic sheaves (for example, nylon sheaves) and sheaves with plastic lined grooves can provide increased rope life in some cases. However, as discussed below, the corresponding change in wire rope failure modes may necessitate a modification of the rope retirement criteria. Also, non-metallic or plastic lined sheaves may be disadvantageous for installations where either a wire rope or a non-metallic rope is used with a ram-tensioner motion compensation system. The repeated bending of the rope over the sheaves can lead to excessive rope temperatures, since non-metallic sheaves are unable to function as a heat sink. In this case, metallic sheaves may be required to minimize temperature build up in the rope.

22.0 ROPE FAILURE MECHANISMS AND RETIREMENT CRITERIA

The tension-induced radial forces exerted by the outer strands of a rope produce high localized contact loading between adjacent strands. In a wire rope, these contact forces produce very high contact stresses in the individual wires. The contact stresses are often responsible for the initiation and propagation of fatigue cracks and the eventual fatigue failure of the rope. This is the dominant failure mechanism for ropes subjected to cyclic-tension loading.

In the case of a wire rope with an independent wire rope core (IWRC), the internal contact stresses often cause the core to break up so that it ceases to contribute to the rope strength. Then, not only is the strength contribution of the core lost, but the IWRC continues to be a source of high contact stresses for the wires in the main strands and accelerates fatigue failure of the entire rope. For some applications, a rope with an IWRC provides poorer long term performance than a rope with a fiber core, even though the latter has a lower initial breaking strength.

When a wire rope is bent over a sheave or drum, the change in curvature produces bending stresses in the individual wires, and the relative motions among the strands produce internal wear and variations in the load distribution among the strands. (See an earlier section of this chapter for a detailed discussion of cable behavior in bending.) It is interesting to note, however, that unlike a simple beam in bending, the maximum bending stresses in a rope do not occur in the wires furthest from the center of rope curvature. Instead, they occur in the wires adjacent to the rope core. If a rope should be subjected to bending over a very small sheave at a very low tension, it will eventually fail due to the accumulation of wire breaks in the interior of the rope. The insidious nature of this fatigue damage has led to unexpected rope failures.

However, in a great majority of common wire rope applications, the location of the wire fatigue failures is on the surface of the rope where the wires contact a steel sheave or other rope wraps on a drum. These failures are often erroneously attributed to bending stresses, but they are actually the result of the high contact stresses in the wires at the surface contact locations.

If a non-metallic or plastic-lined sheave is substituted for a steel sheave, the surface contact stresses will be essentially eliminated, and the rope may enjoy an improved fatigue life. In this case, the wire failure locations may change to the strand-to-strand or strand-to-core contact sites. Although the rope life in terms of cycles to failure may be improved by the elimination of steel sheaves, the change in rope failure mode may have an important effect on the rope retirement criteria.

For example, if a rope used on steel sheaves is retired on the basis of the number of visible broken wires (for example, six broken wires per lay), this criterion may no longer be valid with non-metallic sheave grooves since the rope may fail internally, not externally. In many applications, the ability to assess the condition of a rope through broken-wire counts may be much more important than the potential rope life improvement offered by non-metallic sheaves. Serious accidents may occur unless the rope retirement criteria are carefully matched to the rope failure mechanisms for any particular application.

23.0 WIRE ROPE FATIGUE DATA

The usable life of a wire rope in a particular application may be limited by one or more of the following factors:

1. Metal Fatigue
2. Internal or External Abrasion
3. Internal or External Corrosion
4. Damage due to cutting, kinking, crushing, bird caging, or tensile overload

With respect to metal fatigue, wire rope performance depends upon the rope construction and material, the sizes of any sheaves or drums over which the rope must operate, the operating tension, and the type of lubrication applied to the rope.

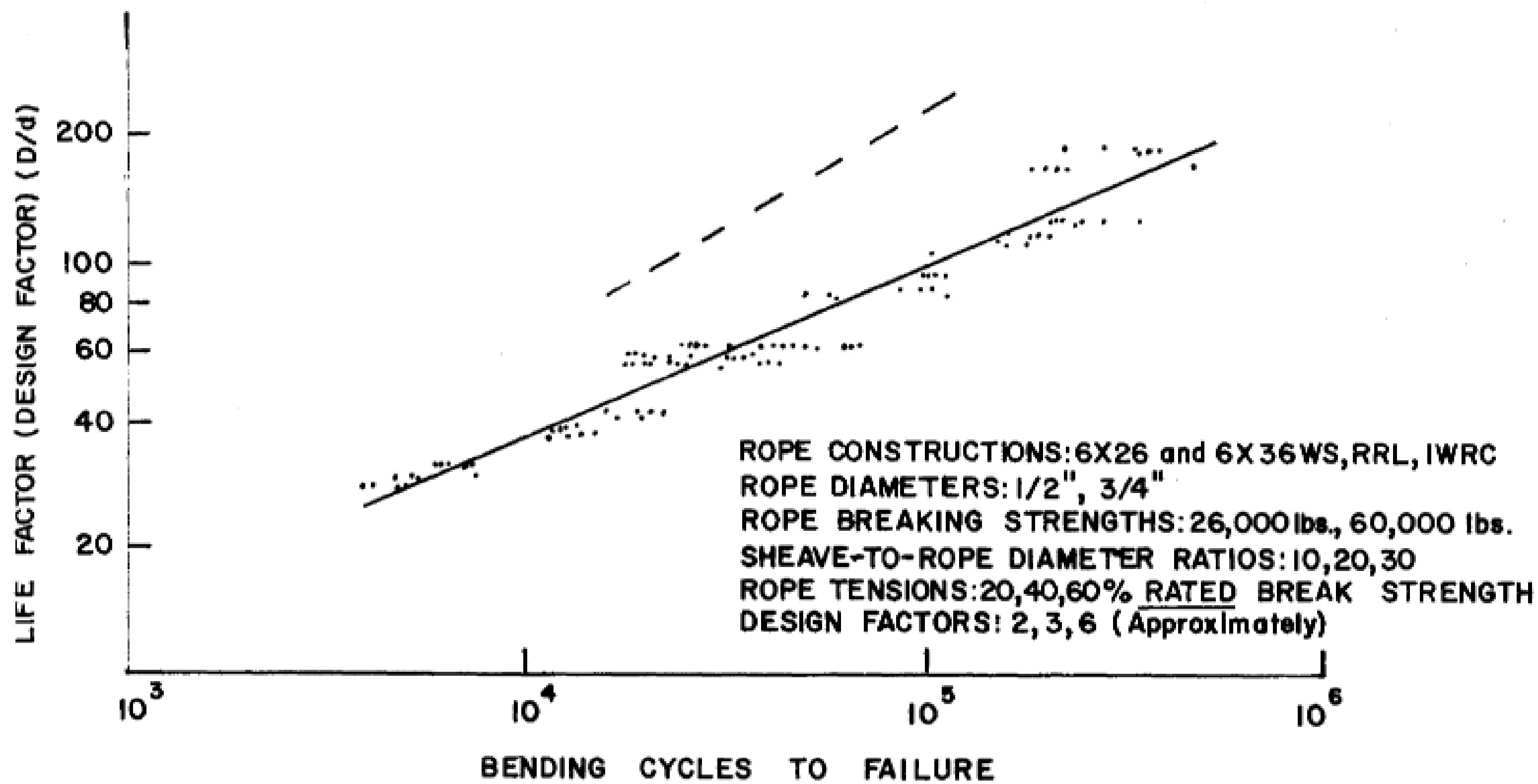
For most rope applications, there is no optimum sheave size or Design Factor. (The Design Factor is the ratio of the rope breaking strength to the operating tension and is sometimes referred to as the Safety Factor.) Unless the rope must be retired for reasons other than metal fatigue, the rope life will benefit from larger sheaves and higher Design Factors. Considerations of system size, weight, and cost will establish the practical limits for these parameters.

To evaluate rope bending fatigue performance under ideal laboratory conditions, a series of tests was undertaken with the following test parameters:

Rope Diameters - 1/2 and 3/4 inch
Rope Constructions - 6x26 and 6x36 Warrington Seale, Right Regular Lay, IWRC
Rope Material - Extra Improved Plow Steel
Sheave-to-Rope Diameter Ratios - 10, 20, and 30
Design Factors - 2, 4, and 6 (approximately)

All four ropes were manufactured from the same lot of steel by the same manufacturer, and all had the same type of lubrication. Four rope specimens were cycled to failure for each possible combination of rope diameter and construction, design factor, and sheave size. Furthermore, the specimen selection and test sequence were totally randomized to allow statistical analysis of the test results. The test results appear in Figure 8-13.

The tests were conducted with a rope cycling stroke amplitude sufficient to allow a rope section four lay lengths long to pass onto, around, and completely off of the sheave with each machine stroke. Thus, this rope section experienced two straight-bent-straight bending cycles with each complete machine cycle (two strokes). In Figure 8-13, the bending cycles to failure (complete parting of at least one complete rope strand) are plotted as a function of a dimensionless parameter defined as the Life Factor. The Life Factor is the product of the Design Factor and the Sheave-to Rope Diameter Ratio (Did). In this case, the sheave diameter, D, is the pitch diameter measured at the rope center line.



WIRE ROPE BENDING FATIGUE DATA

FIGURE 8-13

Even though the test conditions varied widely in terms of the combinations of sheave size and rope tension, all of the test results are fairly well normalized when presented in terms of the Life Factor. This approach allows estimates of rope life for other combinations of sheave size and rope tension not specifically included in the test program.

It should be noted, however, that wire ropes of other sizes, constructions, material strengths, and manufacturing lots will produce different test results. However, only a few tests are required to establish the position and slope of the Life Factor curve for any specific rope.

Experience has shown that the bending fatigue life of a wire rope diminishes with increasing rope diameter. For example, the dashed line in Figure 8-13 corresponds to the approximate fatigue life of a 3-1/2-inch diameter rope tested during another program. Other tests of 1-3/4-inch diameter ropes produced results between those of the curves presented in Figure 8-13. Because of these variations of fatigue life with rope size as well as with rope construction, care must be exercised in using the results of tests of one rope to predict the fatigue life of a different rope.

24.0 MATHEMATICAL MODELING

An important accomplishment in recent years has been the development of mathematical models for use in the design and analysis of ropes and cables. Commercially available computer programs can be used to analyze rope and cable designs prior to manufacture for the purpose of predicting performance in both tension and bending. These programs not only minimize the requirement for prototype fabrication and testing, but they also provide insight into failure mechanisms and means for improving overall rope or cable performance.

An example of such a program is CABLE SOLVER 1 developed by Tension Member Technology for the analysis of steel wire armored electromechanical and fiber optic cables. This program takes into consideration the compressibility and nonlinear stress versus strain behavior of various cable materials, and the accuracy of the mathematical model has been validated through extensive laboratory testing.

The program accepts as input:

(1) Details of the actual or assumed cable geometry

(2) Data regarding certain physical properties of the various materials in the structure

(3) A choice of analysis strategy including either applied tension or strain, end constraint (fixed end, free end, or induced rotation), cable diameter reduction methodology, core model, and external hydrostatic pressure

The program provides the following output:

(1) The initial as-manufactured or as-designed cable geometry including helix angles, lay lengths, exact coverages, radii of curvature, compactions, cusp fill, and diameters of the various element layers

(2) The deformed geometry of the loaded cable including all of the above parameters

(3) The strain, stress, and tension experienced by each layer of elements, as well as the torque contribution of each layer, and the layer weight

(4) The overall cable characteristics including strain, tension, torque, rotation, ideal breaking strength, average core pressure, and weight.

Similar programs are also available for the analysis of cables having non-metallic fiber strength members and for both metallic and non-metallic ropes. CABLE SOLVER is not the only set of computer programs commercially available as this chapter is being written, and other programs are likely to become available as the sophistication of cable design continues to advance, especially in the area of fiber optics and with the development of new cable materials.

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CHAPTER 9

EQUIPMENT LOWERING MECHANICS

H. O. Berteau

1.0 INTRODUCTION

Improper design considerations while lowering or retrieving equipment at sea can result in substantial cable damage including rupture followed by the total loss of a valuable payload.

The hydrodynamic behavior of the payload at the end of the lowering cable is often unknown or ignored. Yet the shape and weight of the payload has considerable bearing on the cable performance. For example, a payload with a large drag area can substantially add to cable static tension when the cable is hauled in. Conversely, while paying out the payload may fall slower than the cable itself thereby creating a slack cable condition. This slack condition may result in a subsequent snap load, when the lowering is stopped and the payload impacts the cable, or in a kink in the wire,, or both. Payload spinning, kiting and tumbling can also obviously impair an orderly lowering.

Recurrent causes of cable damage include: loading the cable beyond its yield point (or breaking strength), fatigue failure, and kinking. In this section the mechanisms which create these detrimental conditions are first reviewed. These potential failure mechanisms include: quasi-static tensile loads, wave induced dynamic loads, zero load (slack cable), and impact (snap) loads.

The mathematical concepts which are used to predict and quantify these causes of cable failure are also reviewed and their use illustrated with a few typical examples.

In the fast part of this chapter specific design recommendations are made for improving the payload hydrodynamic behavior. Finally, the operational limits such as maximum length of cable paid out or allowable payout rates are discussed.

2.0 MECHANISMS CAUSING LOWERING CABLE DAMAGE

One way to obtain measurements of oceanographic parameters at great depths is to lower sensing instrument packages with electromechanical cable. Of necessity these cables are kept to small, workable sizes but because of the long lengths deployed their immersed weight often results in very high tension levels. Vessel motion, due to wave action, introduces additional cyclic loads which can, and often do cause, cable deterioration due to flexure fatigue. Kinks can occur in the cable during zero load conditions. These problems often result, at best, in loss of electrical signal due to short or open circuits and at worst in complete failure of the cable and total loss of the instrument package.

While hanging free from the ship, the tension in the cable is the sum of the static load, due to cable and instrument immersed weight, and the dynamic load, due to cable and attached equipment inertia and hydrodynamic resistance. Most of the time static and dynamic effects occur simultaneously. However, for the sake of clarity it will be helpful to consider them separately.

2.1 Static and Quasi-Static Tensile Loads

By and large, cables are designed and built to resist fair amounts of tensile loads. As their working life progresses their original strength is reduced by corrosion, abrasion and normal wear and tear. If the tensile loads come close to the actual strength of the cable, permanent cable damage or even total failure will occur. To prevent this form of failure it is necessary to understand and quantify the mechanisms of tensile loading.

The first factor of tensile loading is plain weight. The weight that the cable must support at its ship end is made of two parts: 1) the weight in water of the payload and 2) the weight in water of the cable itself. Whereas the payload immersed weight remains constant, the immersed weight of the cable increases with the length of cable paid out. In many situations the payload weight is but a small fraction of the cable weight.

The second factor of cable tensile loading is due to hydrodynamic resistance (drag). If, on a calm day, the lowering winch is turning at a constant rate, the resulting steady state motion of the payload and lowering cable through the water will produce a quasi-static loading which, depending on the direction of motion (up or down), will add to or subtract from the static loading due to cable and payload weight. A few words on the nature of hydrodynamlc resistance will help understand how cable and payload drag interact and combine to drastically change the static loading due to weight only.

Simply stated, hydrodynamic resistance is the force experienced by a body when moving through a fluid. This resistance is due to a combination of viscous and pressure effects. These two effects are concurrent. Their relative magnitude depends, however, on the nature of the flow past the body. As long as the flow remains smooth, or laminar, shear stresses predominate and the resistance, or drag, is essentially due to the friction of the fluid on the bodies immersed surface (skin friction drag). On the other hand, when a combination of fluid speed and body shape (blunt bodies) result in a wake past the body the drag force is then essentially due to the pressure difference between the upstream and downstream sides of the body (pressure drag).

To illustrate the point, the force needed to tow a small but long and neutrally buoyant fishing line aft of a sail boat is essentially due to friction drag on the line. On the other hand the force experienced by someone towing a fully submerged bucket from a short rope aft of the same boat is essentially pressure drag.

In applications involving lowering and hauling equipment to and from the sea floor it is fair to say that the hydrodynamic resistance experienced is the sum of the friction drag on the cable and of the pressure drag on the equipment or payload at the end of the cable.

Cable drag is directly proportional to cable length, whereas equipment drag remains essentially constant. When hauling in, the hydrodynamic resistance will increase the static load due to cable and equipment weight. Its maximum contribution of course is at the beginning of the haul when the cable is longest. Methods to calculate drag forces are reviewed in the next section.

2.2 Wave Induced Dynamic Loads

Next we will consider the dynamic loads imparted to the cable as the ship heaves, rolls, and pitches in rough seas.

After lowering the payload to a certain depth (say 2,000 meters) let us secure (stop) the winch. If the cable tension at the head sheave could then be read and displayed, the record would show large fluctuations around a mean. This mean would of course be the immersed weight of the cable paid out and attached equipment. Deviations from the mean are due to dynamic forces imparted on the cable by the motion of the head sheave. As the cable and attached equipment are pulled towards the surface or allowed to plunge back into the sea the cable and the equipment experience both drag and inertia forces.

As previously mentioned, the drag forces are caused by cable and payload instantaneous speed. The inertia forces are caused by cable and payload instantaneous change of speed. Both forces are concurrent. Drag forces reach a maximum when the speed is largest, inertia forces are greatest at the time of maximum acceleration - usually when cable and payload are at rest, at the beginning of a new motion cycle. Here again it may be instructive to briefly look at the nature of the inertia forces. If at some instant the cable and the equipment are hanging still from the ship (zero speed) and at some later but proximate instant cable and payload are pulled upwards at some speed by a ship roll, the tension at the sheave increases. This tension increase is caused by the "inertia" of the cable and equipment which "resent" and resist the instantaneous upward pull.

In general the inertia force can be defined as the force required to change the speed of a body. lts magnitude equals the product of the body mass by the change of speed experienced per unit of time (acceleration).

Fully immersed bodies do trap and entrain a certain amount of water in their motion. This entrained water undergoes the same acceleration as the body itself. The effect is as if the mass of the body had been increased. In fact the actual mass to be accelerated, called the body virtual mass, is the sum of the body mass and of the mass of the entrained water. As a result in the increase in mass, the force needed to accelerate a body in water may be much larger than in air. For example the starting load to accelerate an elevator from rest would be much larger if the elevator was fully submerged (neglecting buoyancy effects). Formulas to calculate inertia forces are presented in the next section.

Now let us go back to our ship and let the winch run again, hoisting the equipment back to the surface at some constant hauling speed. The hydrodynamic resistance due to this additional speed will, at least in the beginning when cable weight reduction is not significant, increase the tension mean and therefore also the tension peaks previously experienced when heaving to, with the winch secured.

The instantaneous tension is now the algebraic sum of four simultaneously occurring effects, namely:

- o the drag due to hauling speed
- o the drag due to wave induced motion
- o the inertia forces to accelerate (or decelerate) the cable and the equipment.

This time varying, wave induced, tension results in cyclic stresses which can cause the wires and/or the conductors of a cable to fail in fatigue.

It is a well-known fact that the number of fatigue cycles to total failure dramatically decreases as the cyclic tension increases. In instrument lowering applications, because of the long lengths of cable required, the tension can reach a very large fraction of the cable strength. Under these conditions, only a few hundred cycles of repeated stresses can severely damage the cable (see Reference 1).

Keeping the wave induced loads and their time of application small will prevent accelerated fatigue cable deterioration.

2.3 Zero Load. Slack Conditions

Zero load can be the prelude to catastrophe. A slack cable can easily jump out of a sheave, can kink, or it can be subjected to severe snap loading. The payloads attached at the free end of the cable may force the cable to unlay and turn on itself. If the cable is allowed to become slack at some later time it will relieve some of the stored torsional energy by forming one or a number of twisted loops at the point of slack. When tension is reapplied the loops are pulled tight, the armor wires and the conductors are then severely bent thus permanently damaging the cable at the point of kink.

Understanding the mechanisms leading to slack conditions is a first step towards the prevention of their occurrence.

If a body heavier than water is allowed to free fall to the sea floor, it will first accelerate and gain speed. As speed increases so does the hydrodynamic resistance on the body. Sooner or later the drag will equal the pull of gravity and the body will continue to fall at a constant maximum speed called "terminal velocity." This being accepted, let us consider what happens as the equipment is lowered to the bottom.

Assuming the sea to be flat calm and the winch to pay out at some constant and reasonable speed, then the equipment will descend smoothly at the payout speed. But if this speed is increased beyond the equipment's own terminal velocity then the cable will override the equipment and form a slack loop probably full of kinks.

One might be tempted to think that paying the cable at a rate less than the equipment terminal velocity would prevent slack conditions to occur anywhere along the cable. This is not always true. As evidenced in an example presented in the next section, in certain cases a length can be reached where the combined drag on the cable and the equipment entirely negates the gravity pull. The cable will then again become slack, this time at the shipboard end.

Now let us assume a situation where the winch is secured but the ship is rolling heavily. On a down roll the head sheave may well reach speeds high enough to momentarily create slack conditions either at the sheave, or at the equipment end, or at any point in between. Of course such high speeds can also be obtained when paying out from a rolling ship.

Methods for determining conditions of zero tension and points of occurrence will be briefly reviewed in the next section.

2.4 Snap Loads

Cable tension, as we have seen, is the algebraic sum of the external forces acting on the cable, namely the static force due to weight and the dynamic forces of inertia and hydrodynamic resistance.

This dynamic force can be either compressive or tensile. When the compressive component exceeds the static tensile force the cable goes slack. The payload is then allowed to travel on its own until the cable catches it again. Severe snap loads, as high as ten times the immersed weight of the payload (Reference 2) are then imparted to the cable.

It may again be instructive to describe the mechanism which produces snap loads in some detail. Let us assume that an up roll is pulling hard and fast on the cable. The steel cable is rather stiff, having a high modulus of elasticity. The payload has a large virtual mass.

It is heavy and its ugly shape entrains a lot of water. The upper end of the cable moves with the ship. Because of its inertia the payload does not move appreciably yet. The cable is forced to stretch and because of its stiffness the pull on the payload increases at a rapid rate. As a result the payload starts to move faster acquiring upward speed and momentum.

Now comes the down roll. The pull of the cable on the payload diminishes and vanishes as soon as the distance between payload and cable shipboard end equals the relaxed (no load) length of the cable. The payload then starts to travel on its own. It still has considerable momentum and keeps on going upwards, slowing down until the pull of gravity stops it. It then reverses direction of motion. It starts to fall acquiring downwards speed and momentum.

In the meantime the cable is still going down following the ship down roll and giving plenty of time for the payload to gain considerable downwards momentum. Now comes the next up roll. The cable rushes back to the surface. When the distance between the upper end of the cable and the payload position again equals the unstretched length of the cable, the cable starts to pull on the payload. The great force necessary to rapidly stop and reverse the direction of the payload constitutes the snap load.

lf properly timed, that is if the wave frequency is such as to permit the procedure to repeat itself the cable will be subjected to a series of snap loads and probably will break.

A simple mathematical model to predict the occurrence of snap loads and quantify their magnitude is presented in the next section.

3.0 PREDICTING CABLE LOADS

This section will present the formulas and certain simple analytical methods which will permit a reasonable prediction of both the static and dynamic cable loads.

3.1 Immersed Weight. Static Load

The weight of a fully immersed object equals the weight of the object in air less the weight of the water displaced by the object. If the two weights are equal the object is said to be neutrally buoyant. If the air weight of the object is less than the weight of the water displaced the buoyant object will want to come back to the surface.

Example 9.1

What is the static load at the ship due to 2,000 meters of 1/2 inch 3x19 wire rope supporting a cylinder of cast iron 4 feet high by 2 feet in diameter.

Use:	Weight in water of 1/2" 3x19	=	.341 lb/foot
	Water density	=	64 lbs/cu foot
	Cast iron density	=	450 lbs/cu foot
	One meter	=	.28 feet

Solution

Air weight of cylinder	=	π x 4x450	=	5655
Weight of water displaced	=	π x 4x64	=	- 804
Immersed weight of cylinder			=	4851
Immersed weight of cable	=	2000x3.28x.341	=	2237
Static load at ship end of cable			=	7088 lbs

3.2 Hydrodynamic Resistance – Quasi-Static Load

The hydrodynamic resistance of a fully submerged object moving at a constant speed "V" (ft/sec) can be estimated using the formula:

$$D = 1/2^{\rho}\, C_D\, AV^2 \qquad (9.1)$$

where D is the hydrodynamic resistance or drag (lbs)

ρ is the water mass density = 2 slugs/cu.ft.

C_D	is the drag coefficient
A	is the object area used to empirically derive the drag coefficient (sq-ft)

Pressure drag coefficients for various body shapes (spheres, cylinders, plates, etc...) have been widely published in the literature (References 3 and 4). Longitudinal drag coefficients for cables and long cylinders have also been extensively studied. Published values vary from .02 for rough cylinders to .0025 for smooth cylinders (see Figure 9-1).

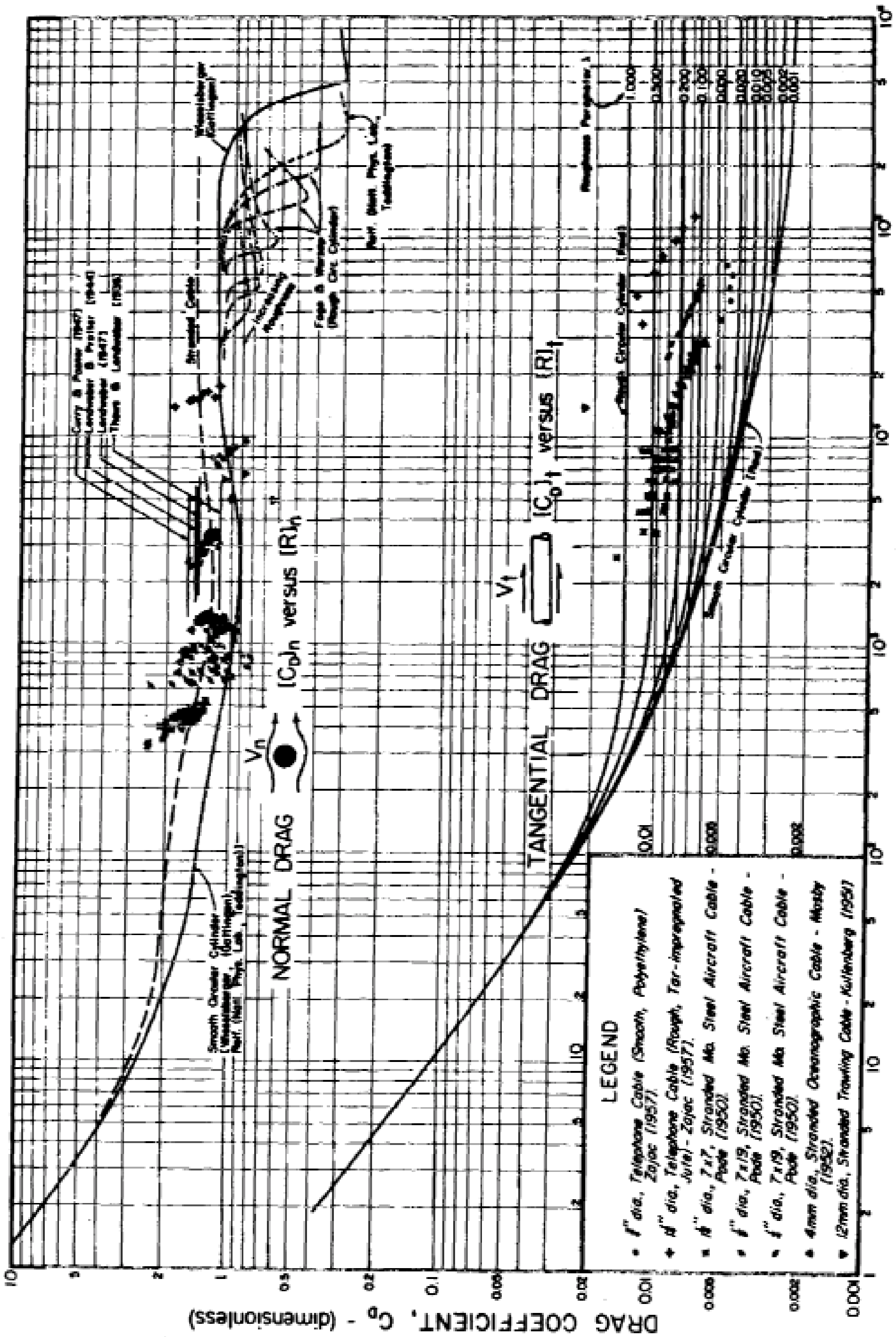
The following example illustrates the use of formula (9.1).

Example 9.2

A biological sampler is lowered to a depth of 2,000 meters with the help of a 3/8 inch 3x19 wire rope. It is then hauled back at a constant speed of 100 meters/mm. Find the tension at the upper end of the cable immediately after the starting transient, using the following characteristics:

Immersed weight of cable	=	.191 lb/ft
Drag coefficient of cable	=	.01
Shape of sampler	=	cone, base up, filled with water
Diameter = 6 ft; Height = 5 ft		
Immersed weight of Sampler = 200 lbs	=	Dry weight = 320 lbs
Drag coefficient of sampler	=	1.0

Also, find the percent increase due to drag over the plain static load.

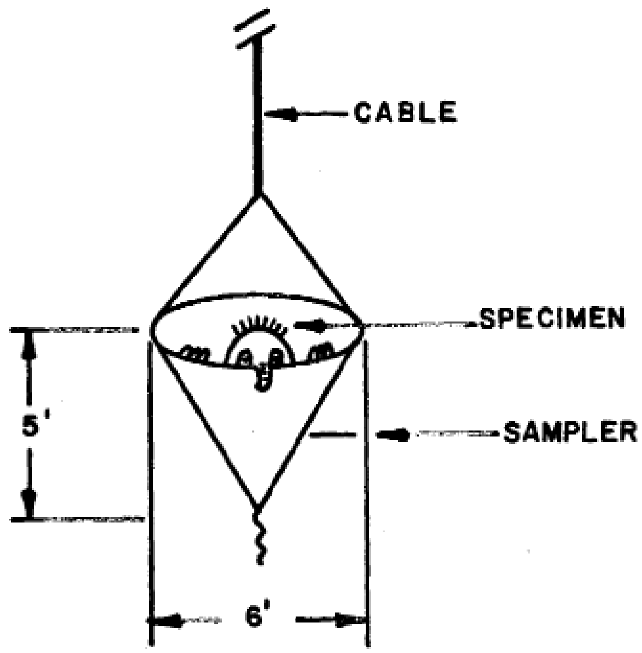


FROM 'BUOY ENGINEERING' by H.O. BERTEAUX. (WILEY & SONS) Page 101

FIGURE 9-1

DRAG COEFFICIENT vs REYNOLDS NUMBER FOR FLOWS NORMAL AND TANGENTIAL TO SMOOTH AND ROUGH CIRCULAR CYLINDERS

Solution



Total immersed weight of cable = .191 x2000x3.2 = 1253 lbs

Weight of sampler = 200 lbs

Static tension = 1453 lbs

Hauling speed = 100x3.28/60 = 5.46 ft/sec

Skin area of cab $= \pi \times \frac{3}{8} \times \frac{2000}{12} \times 3.28$

= 644 sq-ft

Cable drag = 0.01 x 644 x 5.46 x 5.46 = 192 lbs

Cross section of sampler	= π x 3 x 3	
	= 28.3 sq.ft.	
Sampler drag	= 1.0 x 28.3 x 5.46 x 5.46 =	844 lbs
Total drag	=	1036 lbs
Tension at cable upper end	=	2489 lbs
Percent increase due to drag	= 1036/1453 = .713 or 71.3%	

If "s" is the length of cable paid out, the quasi-static tension T(s) at the ship end can be found using the following expression:

$$T(s) = W_p + W_L s + 1/2\rho C_c \pi D_c s V|V| + 1/2\rho\, C_p A_p V|V| \qquad (9.2)$$

where

W_p = immersed weight of payload (lbs)

WL = immersed weight of cable per unit of length (lbs/ft)

C_c = cable longitudinal drag coefficient

D_c = cable diameter (ft)

C_p = payload normal drag coefficient

A_p = payload normal cross section (sq-ft)

V = constant cable speed, hauling being positive and lowering negative

$|V|$ = absolute value of V

Cable drag is both a function of speed and length. If the amount of cable paid out and the speed of lowering are large enough, the combined cable and payload drag can become as large as the cable and payload immersed weight. The tension in the cable then becomes zero.

For a given lowering speed V, the length of cable necessary to produce a slack condition can be found by setting T(s)=O in (9.2) and solving for s.

3.3 Terminal Velocity. Zero Load

At terminal velocity the immersed weight of the object "W" equals the drag on the object, a condition which is expressed by:

$$W = \frac{1}{2}\rho C_D A V_T^2$$

Therefore the terminal velocity VT of the object is given by:

$$V_T = \sqrt{\frac{2\,W}{\rho\, C_D A}} \tag{9.3}$$

Example 9.3

Find the terminal velocity of:

1. The biological sampler described in Example 9.2 using a nose down drag coefficient $C_D = 0.2$.
2. The 2000 meters of 3/8" 3x1 9 wire rope combined with the sampler

Use $\rho = 2$ slugs/cu.ft

Solution

1. Terminal velocity of the sampler.

From previous computations,

Immersed weight of sampler	= 200 lbs.
Cross section	= 28.3 sq-ft

$$\text{Terminal velocity} = \sqrt{\frac{200}{0.2 \times 28.3}} = 5.94 \text{ ft/sec}$$

(1.81 m/sec)

or 109 meters/min

2. Terminal velocity of cable and sampler combined.

From previous computations,

Immersed weight of cable	=	1253 lbs.
Skin area of cable	=	644 sq ft
Drag coefficient of cable	=	0.01
Terminal velocity	=	

$$\sqrt{\frac{1253 + 200}{644 \times 0.01 + 0.2 \times 28.3}} = 10.95 \text{ ft/sec}$$

(3.34m/sec)

This example shows that a payout rate in excess of 109 meters/mm would cause a stack condition in the wire rope lower end. Similarly a downwards speed in excess of 11 ft/sec, which could be easily obtained by a combination of payout rate and ship down roll, would produce a slack condition at both ends of the 1,000 meters length of cable.

3.4 <u>Virtual Mass. Inertia Load</u>

As previously discussed, in order to accelerate a body immersed in water not only must the body be accelerated but also a certain amount of water close to or ahead of the body. As a result the force F needed to accelerate the body in water is greater than the force F required to accelerate the same body in vacuum. This can be expressed by:

$$F^1=(m+m^1)a>F=ma$$

where m is the body mass

m^1 is the added mass of the entrained water

a is the acceleration.

The added mass is usually computed using

$$M^1 = cm\ \rho\ (Vol) \qquad (9.3)$$

where

C_m is the added mass coefficient, ρ is the water mass density (slugs/fl^3) and Vol is the volume of water displaced by the immersed body (cu-fl).

Added mass coefficients for bodies of different shape (sphere, cylinders, plates, etc.) have been empirically determined for linear and oscillating accelerations. Published values of added mass coefficients pertinent to cable lowering problems can be found in Reference 4.

The virtual mass m_v is the sum of the body mass m and of the added mass m^1.

$$m_v = m + m^1$$

Example 9.4

1. Find the virtual mass of the biological sampler previously discussed.
2. Find the inertia force on the lower end of the cable and the percent increase over the static load at that end if the sampler is accelerated towards the surface from rest to a speed of 8 ft/sec (146 meters/minute) in (a) 8 seconds, (b) 2 seconds.

Use $\rho = 2$ slugs/cu ft

and $C_M = 1.5$

Solution

1. Virtual mass of sampler.

Volume of sampler	=	$1/3\pi \times 3 \times 3 \times 5$	= 47.12 cu-ft
Mass of water in sampler	=	47.12×2	= 94.24 slugs

Added mass	=	1.5x47.12x2	=	141.36 slugs
Mass of sampler structure	=	320/32	=	10 slugs
Virtual mass			=	245.6 slugs

2. Inertia force.

Case a. The prudent operator brings the load to full speed in 8 seconds.

The average acceleration is then

$$\frac{8\text{ ft/sec}}{8\text{ sec}} = 1\text{ ft/sec}^2$$

The average inertia force is then 245.6x1 = 245.6 lbs

The percent increase over the 200 lbs of static load due to the immersed weight of the sampler is then

$$\frac{245.5}{200} = 1.23$$

or 123% increase.

Case b. The "other" operator brings the load to full speed in two seconds.

The average acceleration is then

$$\frac{8\text{ ft/sec}}{2\text{ sec}} = 4\text{ ft/sec}^2$$

The resulting inertia force is then 245.6x4 = 982 lbs

The percent increase is

$$\frac{(982)}{200} = 4.92$$

or 492%. This is almost five times the immersed weight of the sampler at rest.

3.5 All Forces Considered. Steady State Peak Tensions

When the hauling and lowering of equipment is done in a rough sea way the tension is no longer time independent, and inertia as well as drag forces must be considered.

To find, under these conditions, the tension in the cable at the shipboard end it is practical to first assume a zero hauling speed (winch secured). Assuming the travel path of the payload and the cable to be vertical (or nearly so) the dynamic tension T(s,t) at the head sheave can then be evaluated with the help of Morisson's equation in one direction, namely

$$T(s,t)= W\rho + W_L s+1/2\ \rho(C_c\ \pi D_c s+C_p A_p)\ V\,|\,V\,|+(m+m^l)\frac{dV}{dt} \qquad (9.5)$$

where

s is the length of cable paid out

V is the vertical component of the head sheave speed at time t.

M is the mass of the cable and payload

m^l is the added mass of the cable and payload

and dV/dt is the vertical acceleration of the head sheave at time t.

Expression (9.5) implicitly stipulates that cable and payload rigidly follow the head sheave motion. In other words it treats the cable as a rigid bar. Despite this oversimplification expression (9.5) can be profitably used to calculate maxima of expected cable tension other than snap load.

To this end one first derives the expression of head sheave vertical speed and acceleration as a function of ship geometry, wave amplitude and frequency. These expressions are then introduced in (9.5) and values T(s,t) are computed at discrete time intervals over a full wave period. The time of maximum dynamic tension occurrence can then be found by inspection.

The next step is to add the quasi-static contribution of drag due to hauling speed. A computation is made of the sheave velocity at time of maximum T(s,t). The hauling speed is then added to this particular sheave velocity and the tension due to drag is then computed. Next the acceleration of the sheave is found for the time of maximum T(s,t) and the corresponding inertia force is also computed.

The instantaneous maximum tension is then the sum of the total drag force, the inertia force, and the immersed cable and payload weight.

This simple computing procedure is best implemented with the help of a computer. Reference 5 presents in detail a derivation of head sheave speed and acceleration due to ship heave and roll and a program to evaluate peak tensions due to combined hauling and ship motion.

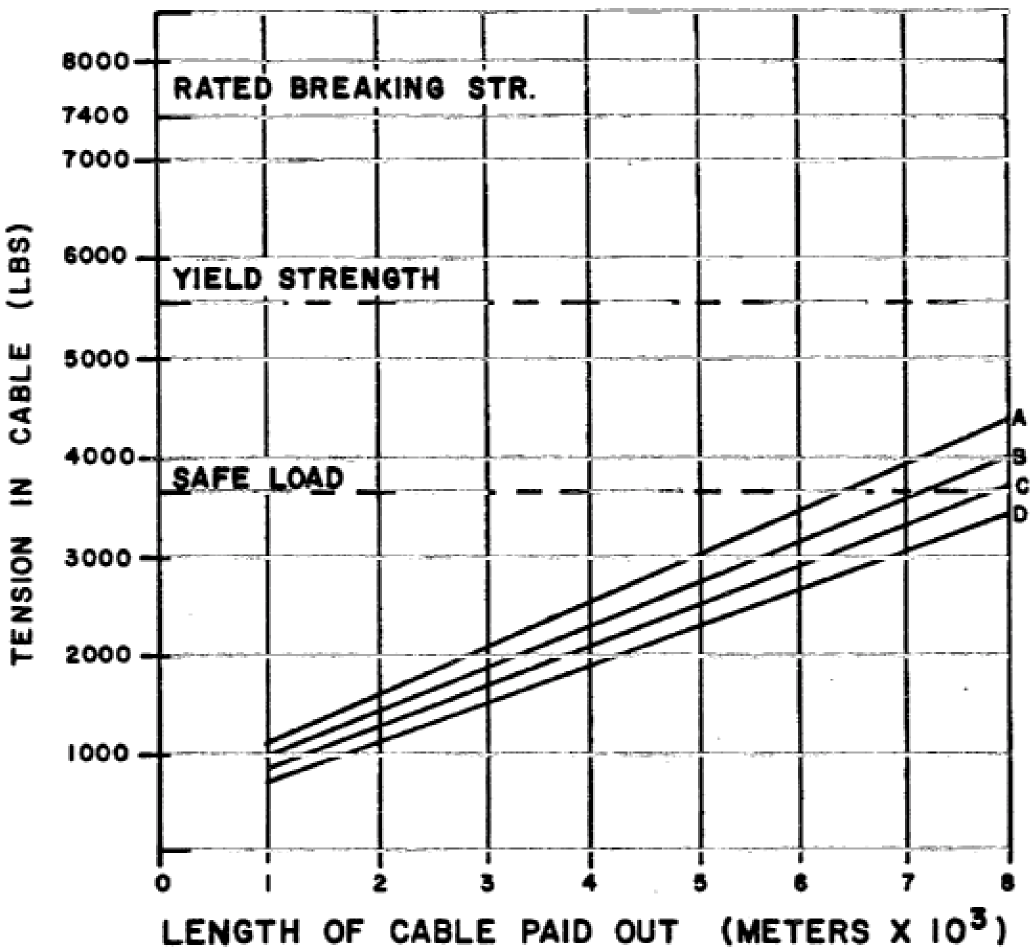
Once these calculations have been performed for a specific vessel the results obtained using this technique should be condensed and presented in a form easy to read. As an example, Figures 9-2 and 9-3 show the peak tensions calculated while hauling a CTD instrument package from the R/V ATLANTIS II under flat calm and sea state 3 conditions.

3.6 Snap Loads

A simple spring mass model (see Figure 9-4) can be used to predict the occurrence of snap loads and compute the ensuing cable tensions. In this model (Reference 2) the following assumptions are made:

- The motion of the payload is entirely vertical (one degree of freedom system).
- The mass of the cable is assumed to be a small fraction of the equipment mass. This would be the case for rather short lengths of cable (hundreds of meters instead of thousands), or if the cable is light (Kevlar line for example), or if the payload entrains a lot of water.
- The cable acts as a linear spring, the tension "T" being directly proportional to the cable elongation "ΔL" i.e.

$$T = k \, \Delta L \tag{9.6}$$



SEA STATE 0
SHIP - ATLANTIS II
PERIOD = N.A.
HEAVE AMPLITUDE = 0.0
ANGLE OF ROLL = 0.0

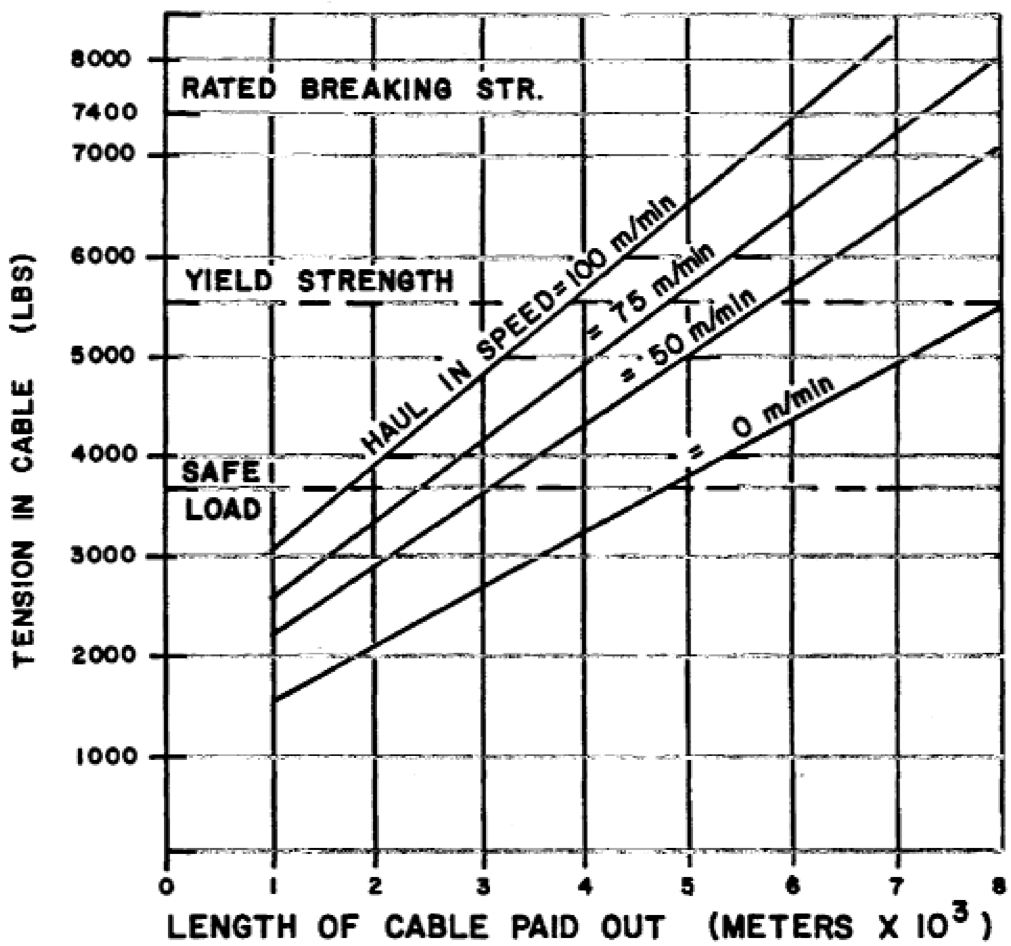
HAUL IN SPEED
A = 100 m/min
B = 75 m/min
C = 50 m/min
D = 0 m/min

CABLE CHARACTERISTICS =
WT/1000' = 145 lb.
DIAMETER = .303"
DRAG COEFF. = .01
RBS = 7400 lb.

INSTRUMENT CHARACTERISTICS =
IMMERSED WT. = 350 lb.
DRAG CONSTANT = 9.72 ft^2
VIRTUAL MASS = 21.0 SLUGS

FIGURE 9-2

PEAK TENSION AT HEAD SHEAVE vs LENGTH OF CABLE PAID OUT



SEA STATE 3
SHIP - ATLANTIS II
PERIOD = 8 SECONDS
HEAVE AMPLITUDE = 3 FT.
ANGLE OF ROLL = 15 DEGREES

CABLE CHARACTERISTICS =

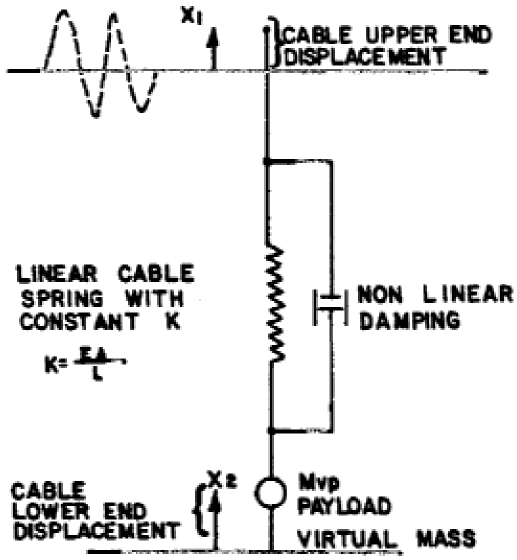
WT/1000' = 145 lb.
DIAMETER = .303"
DRAG COEFF. = .01
RBS = 7400 lb.

INSTRUMENT CHARACTERISTICS =

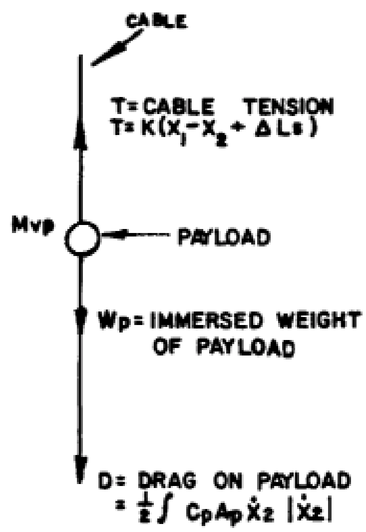
IMMERSED WT. = 350 lb.
DRAG CONSTANT = 9.72 ft^2
VIRTUAL MASS = 21.0 SLUGS

FIGURE 9-3

PEAK TENSION AT HEAD SHEAVE vs LENGTH OF CABLE PAID OUT



CABLE/PAYLOAD SPRING MASS MODEL



PAYLOAD
FREE BODY DIAGRAM

FIGURE 9-4

In the elastic range of cable elongation the spring constant k is given by

$$k = \frac{EA}{L}$$

where E is the cable modulus of elasticity (psi)

A is the cable metallic area (sq.in)

L is the cable unstretched length (ft)

k will then be expressed in lbs/ft.

- Equipment drag is non linear and of the form

  $D = 1/2\, \rho\, C_p A_p\, V \,|V|$

- Equipment drag is assumed much larger than the cable drag.

- The vertical displacement of the cable upper end can be described by an explicit function of time (such as a sinusoid).

Applying Newton's law to the payload mass m_{v_p} m_{vp} (see Figure 9.4) yields the following equation of motion:

$$k(X_1 - X_2 + \Delta L_s) - W_p - 1/2\, \rho\, C_p A_p \dot{X}_2 \,|\dot{X}_2| = m_{vp} \ddot{X}_2$$

where

K = cable spring constant (lb/ft)

X_1 = displacement of cable upper end (ft)

X_2 = displacement of cable lower end (ft)

ΔL_S = cable elongation under pure static loading (ft)

Wp = immersed weight of payload (lbs)

m_{vp} = virtual mass of payload (slugs)

X_2 = instantaneous speed of payload (ft/sec)

X_2 = instantaneous acceleration of payload (ft/sec^2)

Noting that $K\,\Delta L_S = W_p$, this equation of motion reduces to

$$k(X_1 - X_2) - 1/2\,\rho C_p A_p X_2 \,|\, X_2 \,| = m_{vp} X_2 \qquad (9.7)$$

The instantaneous cable tension is then, at the payload end, given by

$$T = W_p + k(X_1 - X_2) \qquad (9.8)$$

The motion of the payload mass is governed by equation (9.7) as long as T>O.

If T, as given by (9.8) equals zero, then the payload is no longer pulled by the cable and a new equation of motion will prevail. Applying Newton's law to the payload in free flight yields

$$-W_p - 1/2\rho\, C_p A_p\, \dot{X}_2 \,|\, \dot{X}_2 \,| = m_{vp} X_2 \qquad (9.9)$$

The system can be assumed to be initially at rest. At time $t = 0$, the upper end of the cable starts moving upwards. The ensuing motion of the payload is then found by integrating equation (9.7) using suitable numerical integration techniques. The author has found Euler's algorithm to be satisfactory provided the time increments are kept small.

Briefly stated, in this algorithm the acceleration of the payload over the time increment ΔT is given by:

$$\ddot{X}_2 = \text{Sum of the forces}/m_{vp}$$

The speed is then simply

$$\dot{X}_2 = \dot{X}_{2\,\mathbf{(t-\Delta T)}} + \ddot{X}_2\,\Delta T$$

where $X_{2(t-\Delta T)} + \ddot{X}_2\,\Delta T$

Similarly the displacement is then

$$X_2 = X_{2(t-\Delta T)} + \ddot{X}_2\,\Delta T$$

The tension is computed for each time interval, using (9.8). If it becomes positive then equation (9.7) prevails again. Speed and displacements when switching to a new equation are of course those computed in the time increment immediately preceding the switch over.

Here again this computing procedure is best implemented with the help of a computer.

Example 9.5

To illustrate the use of this technique let us consider the response of a particular payload/cable system with characteristics as follows:

- **Cable characteristics**

Type:	= 3x19 wire rope
Size	= .375 inch
Length	= 3000 ft
Immersed weight	= 3000 x .191 = 573 lbs
Modulus of elasticity	= 18,000,000 psi
Metallic area	= .1 sq. in
Strength	= 14,800 lbs

- **Payload characteristics**

Type	= heavy instrument package

Weight in air	=	4200 lbs
Weight in water displaced	=	2200 lbs
Added mass	=	12 slugs
Normal area	=	3.14 sq. ft
Drag coefficient	=	1.0

- **Input**

  The vertical displacement of the cable upper end is assumed to be given by

$$X_1 = 7 \sin \frac{2\pi}{4} t$$

ie. Displacement amplitude = 7 ft

Period = 4 secs

Solution

After transient, the response of the system--as calculated by a computer program implementing the technique just described--is as shown on Figure 9-5. From this figure one can see that the vertical displacement of the cable lower end varies from -3.5 ft to+14.8 ft whereas the upper end goes from -7 to +7 ft.

The peak tension obtained after the period of slack is 8,100 lbs or four times as much as the static load (2,000 lbs).

Under static load the comfortable cable safety factor is 14,800/200 = 7.4.

Under snap load conditions, this safety factor reduces to a mere 14,800/8100 = 1.83.

3.7 Advanced Cable Dynamics

The purpose of this section was to introduce the basic principles which govern cable dynamics. External forces acting on the cable--immersed weight, hydrodynamic drag, and inertia--have been reviewed. Formulas to calculate their magnitude have been given. How these forces interact to produce tension peaks and, equally important, slack conditions in the cable has been explained with the help of simple mathematical models. These introductory concepts will enable the reader to quantify the impact that payload weight and shape as well as hauling speed and ship motion may have on cable tension. They will help predict extreme conditions which after all are the most important ones.

Of course there is much more to the science of cable dynamics. Models treating the cable as a continuum in which deformation waves travel and dissipate have been proposed. Others treat the cable as a multiple degree of freedom system made of a number of point masses connected by linear and nonlinear springs and damping elements. Cable response to deterministic and random input has been investigated both in the time and the frequency domain. Readers interested in this field are referred to the bibliography at the end of this chapter.

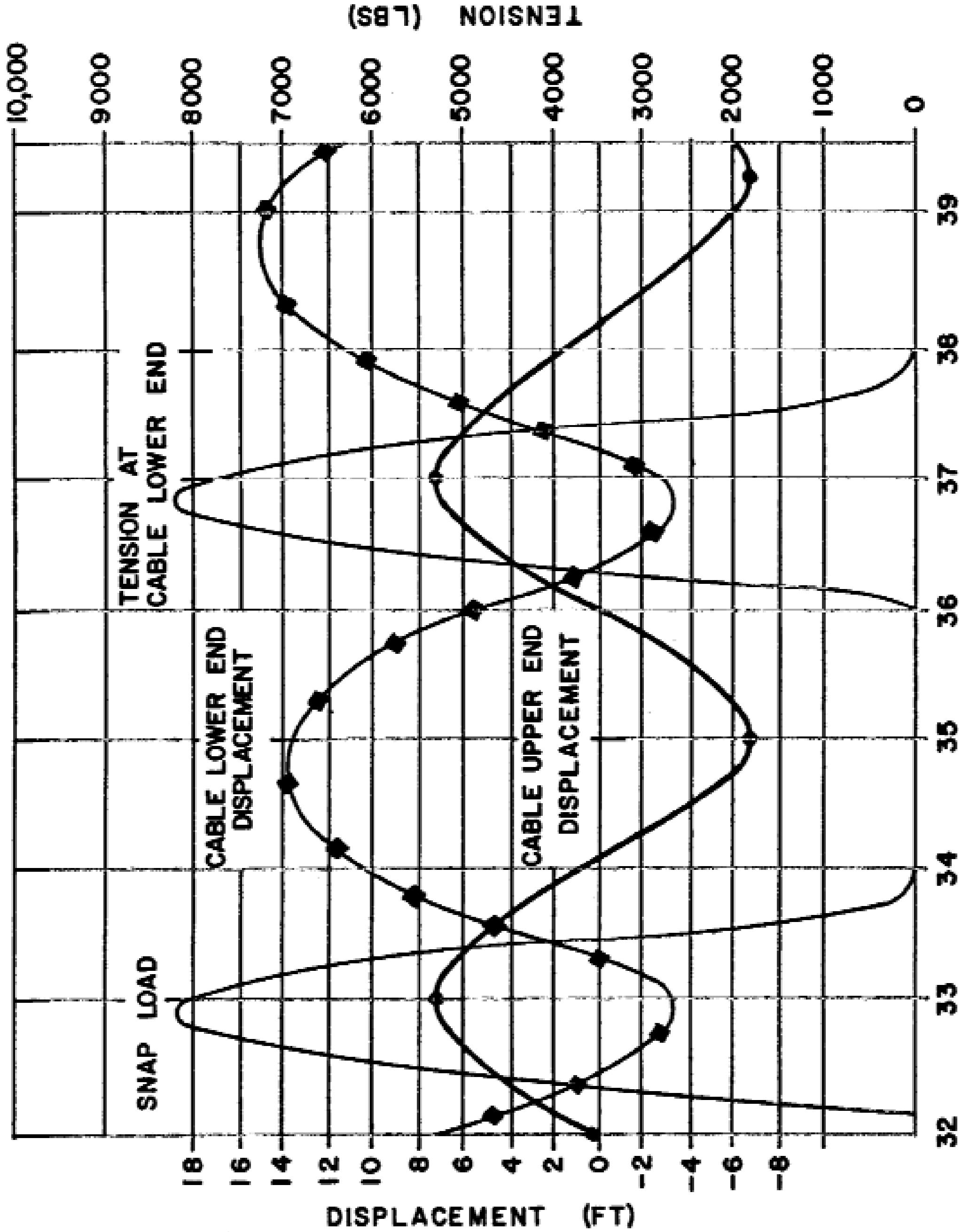
4.0 RECOMMENDATIONS

In this section certain recommendations will be made to improve lowering mechanics and increase cable life expectancy.

4.1 Equipment Design Considerations

Equipment handled underwater is subjected to hydrodynamic forces not present when handled ashore. If the equipment has poor hydrodynamic characteristics it will impart high and unnecessary loads to the cable or cause it to kink and fail. To reduce or better yet suppress these detrimental conditions, underwater payloads should be designed with the following considerations in mind.

4.1.1 Stability. If the payload was permitted to free fall it should do so in a vertical path. In addition it should not tumble, flutter, or spin. A payload falling sideways, or kiting, will pull the cable at large angles from the vertical perhaps causing large cable bites and slack conditions. A tumbling or fluttering payload will jerk the cable at the point of



CABLE LOWER END DISPLACEMENT AND TENSION AS A FUNCTION OF CABLE UPPER END DISPLACEMENT.
(IMMERSED WEIGHT OF PAYLOAD EQUALS 2000 LBS)

FIGURE 9-5

attachment which may fail due to repeated bending. A spinning load can force hundreds of turns in the cable which will kink at the first opportunity.

To fall in a plumb, orderly way, the payload must be statically and dynamically stable. The payload is statically stable if it has a natural tendency to return to an upright steady state vertical flight. The payload is dynamically stable if it returns to its steady state upright vertical flight with oscillations of decaying amplitudes.

Investigating the static stability of a submerged object falling at some constant speed is straightforward and relatively easy. The step-by-step procedure involves:

- assume an initial tilt angle from the vertical
- resolve the drag forces into normal and tangential components
- compute the moments with respect to the body center of gravity induced by the buoyancy and drag forces
- sum the moments

If the resultant moment tends to reduce the initial tilt angle then the object is statically stable at that angle. lf not it will have a tendency to capsize. The process must then be repeated for increasing initial tilt angles.

If an object is found to be statically unstable over a large range of tilt angles (say from 0 to 45°), then it is unfit for cable deployment. Its configuration must be altered until a proper combination of weight distribution and drag righting moment is found for all tilt angles considered. An example of such sensitivity study can be found in Reference 6.

Predicting the dynamic stability of free falling objects involves the simultaneous solution of six nonlinear partial differential equations. The mathematics required for this solution certainly go beyond the scope of this discussion. Interested readers should again consult the bibliography at the end of this chapter.

The following practical considerations if systematically implemented, can greatly improve the flight stability of cable lowered equipment and instrument packages.

- Weight Distribution. Packages should not be too heavy. Their center of gravity should be as low as possible and in all cases well below the center of buoyancy so as to provide a good righting moment.

- Shape. When placing instrumentation on a frame or equipment in some packaging form an effort should be made at reducing the top and bottom drag areas. A slender package will have much less drag as it travels vertically through the water than a fat, chubby one. Furthermore, it will entrain much less water and its added mass will be small.

- Symmetry. Vertical axisymmetry will greatly enhance flight stability. The payload weight should be distributed evenly around the vertical axis. If not the center of gravity will be off the center line and the package will not hang vertically from its point of cable attachment. The payload shape should also be axisymmetrical so that fluid induced forces cancel each other. As demonstrated in tank tests, an acoustic pinger strapped on the outside of an instrument package frame causes the package to tilt and kite sideways as it sinks.

- Spin. In certain cases, equally distributed appendages can have the proper shape or inclination to induce a torque on the lowered equipment. This torque will force the equipment to spin, It is often possible to observe the spin of a load at the beginning (or the end) of its lowering. If detected, the condition causing the load to spin should be corrected.

- Control Surface. Control surfaces can sometimes be used to advantage to stabilize an otherwise tumbling payload. Vertical fins located in the upper part of the package can provide a good righting moment, however, off center loads equipped with vertical fins will steadily kite sideways. Furthermore, if one or several fins are bent, the load will spin. Horizontally mounted circular flaps are known to be very effective for stabilizing blunt cylinders. They are less sensitive to off-center loading. Their drawback is to reduce the cylinder terminal velocity.

4.1.2 Terminal Velocity. As previously explained to maintain tension in the lowering cable the speed of the payload fall must always exceed the speed of the cable. A payload with a small terminal velocity will therefore impose limits on the payout rate and/or the sea state in which the lowering operation can take place. If such operations are repetitive - as in the case of oceanographic profiling instrumentation - the ship time consumed in performing the lowering operation or in waiting for favorable weather becomes prohibitively expensive.

Achieving a reasonable fast terminal velocity should therefore be an important design consideration. The equipment designer should not hesitate at clamping some lead or steel blocks at the bottom of the payload to increase its weight. Doubling the instrument weight in most cases would have but a small effect on the lowering cable safety factor. Reducing the drag area and profiling the bottom of the equipment package will also increase the terminal velocity.

4.2 Equipment Handling Considerations

Now that the equipment has been properly shaped and trimmed, an investigation should be made of the operational limits necessary for its orderly and safe deployment.

4.2.1 Depth Limits. Maximum tension occurs at the head sheave. As previously outlined this tension depends on the length of cable paid out, the weight and shape of the payload, the prevailing sea state and the hauling speed.

Whatever the actual condition of use may be, this tension should not be permitted to exceed a value corresponding to a safety factor of two for most applications, and in no case larger than the yield strength of the cable (about 75% of cable breaking strength for most data logging cables). To help plan safe lowering, predictions of tension levels should therefore be readily available. If for example one had graphs of peak tension versus cable length for different hauling speeds and sea states of the type shown in Figures 9-2 and 9-3, then the maximum allowable cable length could be explicitly and rapidly established. In this case, the maximum length that the cable can (or should) have for a given sea state and a given hauling speed can be easily found from the intersection of the particular tension curve with the safe load (50% of RBS) line or the yield strength (75% of RBS) line.

4.2.2 Winch Speeds Limits. Critical and/or repetitive lowering operations should certainly avoid slack cable conditions. Calculations of winch speeds which would cause the cable to become slack should be made as a function of sea state and length of cable paid out. These predictions should be made for every type of equipment lowered. They should be available in a convenient and tabular form. Limits on payout rates should then be set accordingly.

Measurements of oceanographic packages hydrodynamic behavior have been made both on scale models and on actual instruments being lowered from rolling ships (Reference 7). Cable slack conditions followed by severe snap loads at the cable lower end have been observed and reported (Reference 8).

4.3 Motion Compensation. Limits on deployment depths to avoid high dynamic stresses and on payout rates to prevent slack conditions should be considered temporary measures. The alternative is to consider motion compensators which can greatly reduce or suppress the undesirable effects of ship motion. The following considerations on motion compensation were prepared by J.D. Bird (Reference 9).

Motion compensating handling systems for shipboard applications can be categorized in at least two different ways. The first is a mechanical classification that is descriptive of the basic hardware utilized as the primary compensating element:

- Ram tensioners (the term tensioner is retained here since these units were first commonly used in tensioning applications).

- Bobbing booms, and

- Controllable winches.

The second category is a control law classification that is based upon the primary input signal used in the compensation strategy. The two major strategies are:

- Tension activated, and
- Motion activated

Two or more of these basic techniques can also be combined into more complicated systems where certain characteristics of each are desired. One example might be the high frequency rashness of a ram plus the larger amplitude capability of a controllable winch.

The three basic hardware approaches are shown schematically in Figure 9-6. In evaluating alternative hardware approaches, a number of parameters can be examined that will facilitate fair comparisons between approaches that differ significantly in their method of operation. These include total weight, deck space required, power consumption, complexity, cost, cable wear and fatigue during the compensation process, and frequency response. Most of these are fairly straightforward and self explanatory. Frequency response is an important control system parameter that is a complex function of several other parameters including torque available for control, effective inertia at the load, total system compliance and control actuator response. The effective inertia and system compliance define a natural frequency, above which it is difficult to. achieve effective control response. The torque available is one measure of the limit on the rate at which the control can be applied to the system. The servo-control activator can usually be chosen to have response characteristics above these other limits. The effective inertia of the system is an aggregate measure of the inertias of all moving components reflected to a common point such as the drum or load. Reducing the effective inertia increases the natural resonant frequency and responds faster to limited applications of torque. Low effective system inertia is therefore one of the most important characteristics of the hardware that results in systems with better frequency response and wider overall band width.

The Ram Tensioner is a hydraulic cylinder with a sheave or sheaves attached at the end of the piston. The cylinder can be mounted in any orientation that permits the cable to be fair lead from the winch, around the ram sheaves and to the overboarding sheave. As the ram piston is extended the cable is hauled in at the overboarding sheave. Cable is payed out when the piston is retracted. By making multiple passes around the ram sheaves, the cable compensation amplitude can be several times the piston stroke. Ram tensioners have a relatively low effective inertia, however they have fixed maximum amplitude, and subject the cable to relatively high wear and fatigue.

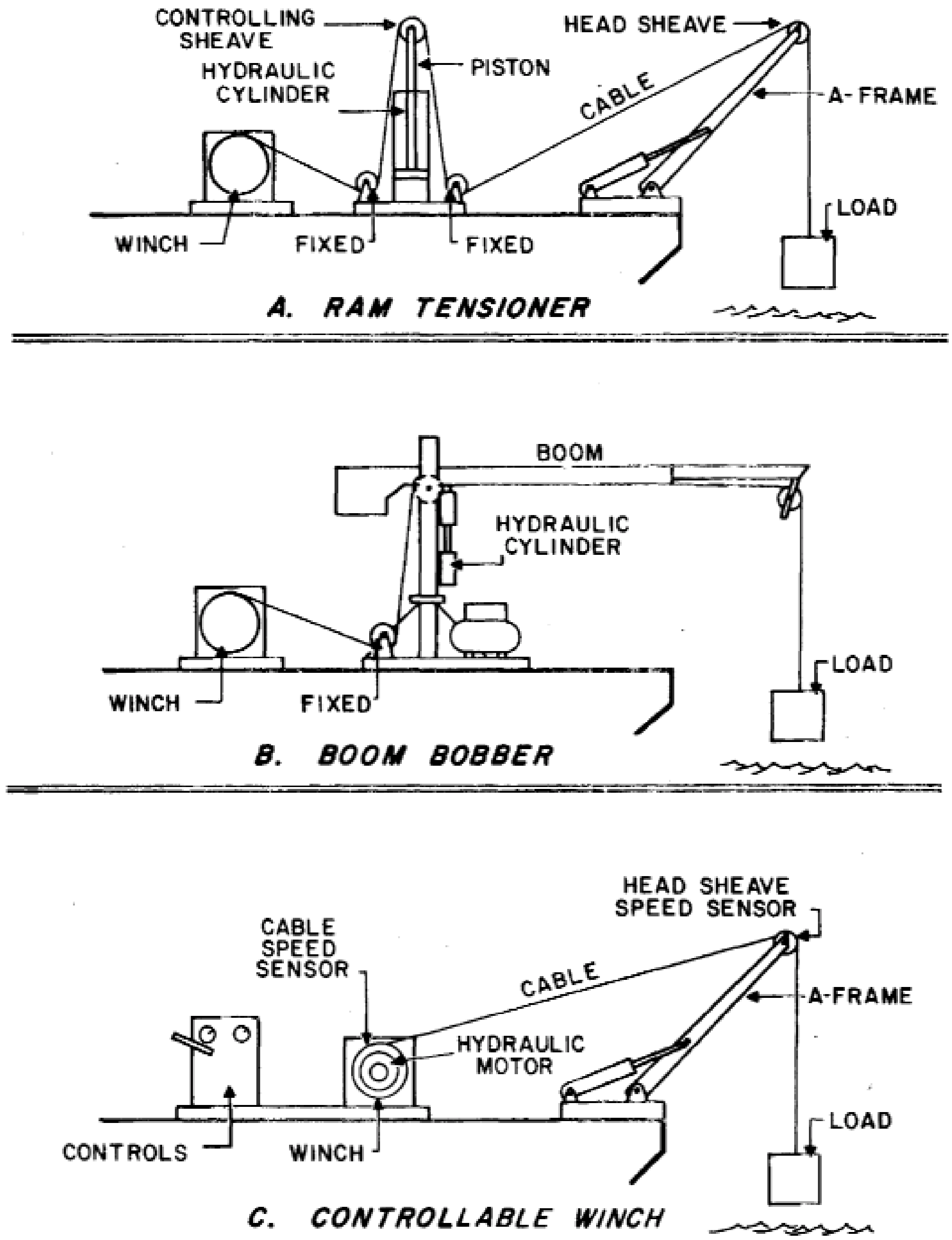


FIGURE 9-6

MOTION COMPENSATION HARDWARE ALTERNATIVES

The Boom Bobber is a cantilevered arm free to pivot at one end with an overboarding sheave fixed at the other end and a hydraulic cylinder located somewhere along its length to support the weight of the arm and the cable suspended payload. As the piston is extended or retracted, the overboarding sheave is raised and lowered respectively. Boom bobbers have a relatively high effective inertia due to the required mass of the moving boom structure and therefore have poor frequency response in servo-controlled applications. Like the ram tensioner, they also have limited compensation amplitude. With careful reeving, however, cable wear can be held to a minimum.

Controllable winches are mechanically the simplest of the three approaches since, presumably, a winch is required in the system for normal cable handling. Winches have medium to low effective inertia depending upon the particular design. Since the effective inertia of the drive motors at the winch drum is increased by the drive gear ratio squared, high speed motor drives generally have higher effective inertias than slower speed, direct drive motors. A major advantage of the controllable winch is that the amplitude of compensation is limited only by the length of the suspension cable. Cable wear with a compensating winch is moderate, when compared to the ram tensioner and the boom bobber.

Current literature tends to group motion compensation strategies as either “Active” or “Passive.” This may be an unfortunate choice of terms. Active systems are thought of as ones that add energy while passive systems do not. Active systems are thought of as possessing feedback elements while passive systems do not. For these reasons, passive systems are considered to be inherently stable, but this may not always be the case. If a passive system has a spring-mass resonance near the peak of the ship motion spectrum, responses can grow uncontrollably.

Systems which are basically classified as passive, such as boom bobbers, often contain rather complex servo-control systems to maintain nominal tension bias to compensate for a changing suspended load such as increasing cable length. For these reasons, the more appropriate and less ambiguous grouping of “Tension Activated” and “Motion Activated” compensation system is proposed here.

Tension activated systems respond to changes in wire rope tension by hauling in or paying out the line in such a manner as to reduce these loads. The most common examples of this type of system are boom bobbers and ram tensioners supported by hydraulic accumulators (Figure 9-7a). There are some basic characteristics associated with tension activated systems. They exhibit an effective spring constant which is in series with the spring-mass system of the cable and payload. In order to minimize the change in tension for a given ship displacement, this effective spring constant must be relatively low. The natural frequency of the cable-payload system is generally above the significant heave spectrum of the ship, except for very deep casts. The addition of another spring often aggravates the problem by moving the system's natural resonant frequency nearer to the ship's heave frequency. The solution to this dilemma is to provide some damping (a natural byproduct of hydraulic oil moving through piping and accumulators) and to soften the spring sufficiently to lower the natural frequency below the ship's range of significant heave energy. The resulting soft spring constant provides poor position control, since small changes in tension result in large displacements.

Tension activated systems actually do very little to directly control the position of the payload. Some researchers have experienced reductions in motion of payloads with properly tuned tension activated systems, but this was primarily due to damping and a shift in the systems natural resonance away from the predominant frequency of the ship motion energy. The primary function of tension activated systems is to reduce the magnitude of tension fluctuations in the suspension cable. For this purpose, they are relatively effective.

Motion activated systems (Figure 9-7b) , on the other hand, deal with the problem at its source. If the upper end of the suspension cable can be held stationary in inertial space, the unwanted energy cannot be transmitted to the payload. This point is particularly well made by Clifford L. Trump in his paper, "Effects of Ship's Roll on the Quality of Precision CTD Data" (Reference 10). Motion activated systems, however, do not utilize the rather easily monitored tension input. Measurement of the vertical motion of the suspension point requires elaborate instrumentation. For this reason, motion activated systems generally have more complex servocontrol elements and multiple feedback loops. System stability becomes more of a concern when high gains are used to provide the required frequency response and accuracy.

Finally, a third category of motion compensators must also be considered. These are "Tension-limiting" devices such as shock absorbers and slip clutches. They are, however, probably

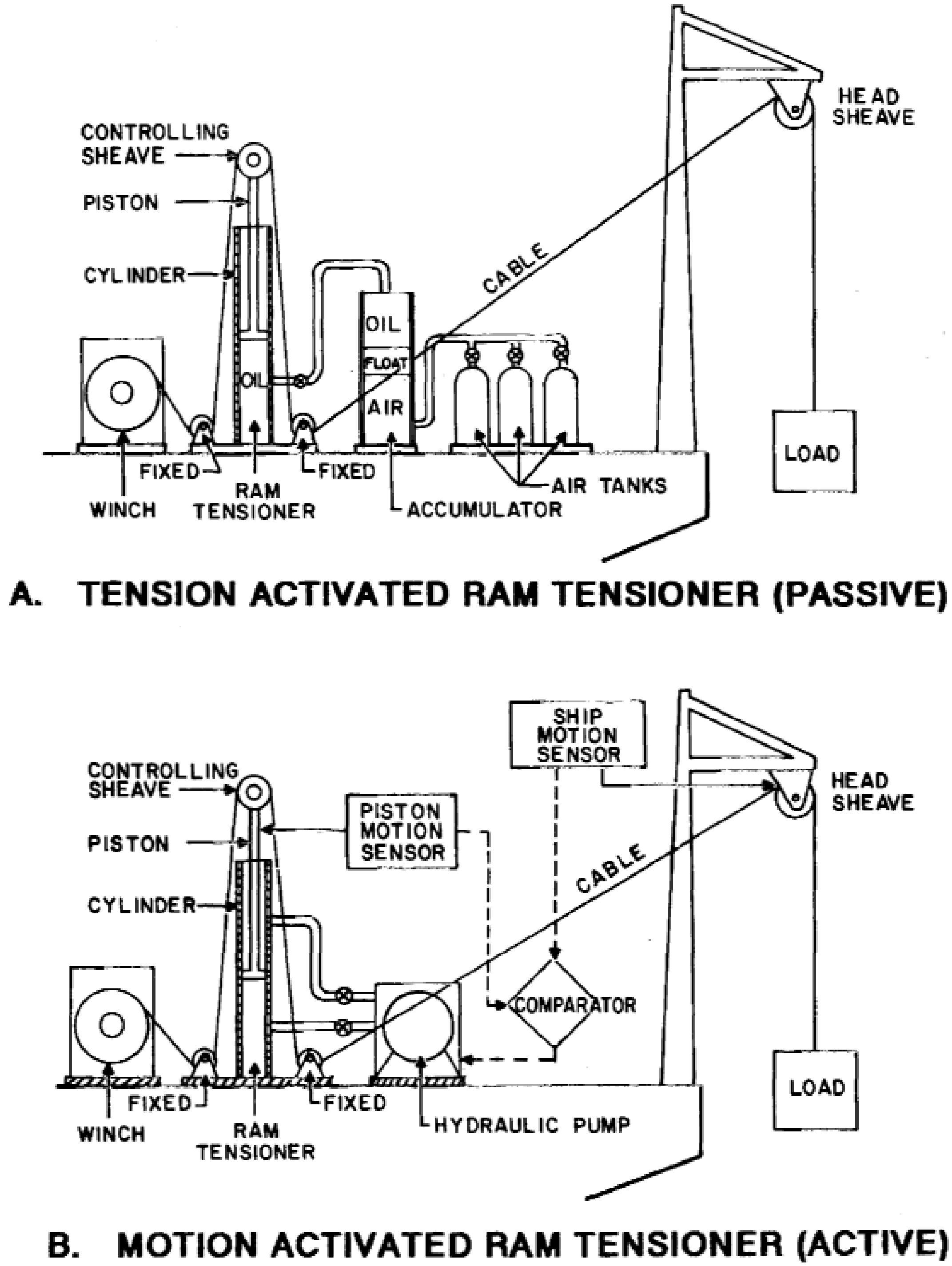

CONTROLLING
SHEAVE
PISTON
CYLINDER
OIL
OIL
FLOAT
AIR
CABLE
HEAD
SHEAVE
LOAD
FIXED
FIXED
WINCH
RAM
TENSIONER
ACCUMULATOR
AIR TANKS
A. TENSION ACTIVATED RAM TENSIONER (PASSIVE)
SHIP
MOTION
SENSOR
CONTROLLING
SHEAVE
PISTON
PISTON
MOTION
SENSOR
CYLINDER
CABLE
HEAD
SHEAVE
COMPARATOR
LOAD
FIXED
FIXED
WINCH
RAM
TENSIONER
HYDRAULIC PUMP
B. MOTION ACTIVATED RAM TENSIONER (ACTIVE)

FIGURE 9-7 A & B

best categorized as a subset of Tension activated devices although they do little or nothing during normal operations and only begin to function during overload conditions to serve, much in the capacity of an electrical fuse, to prevent the cable from breaking under excessive overloads.

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CHAPTER 10

SINGLE DRUM WINCH DESIGN

Michael Markey

PREFACE

Third Addition - Chapter 10

In the eleven years since the second edition appeared the single-drum of "load-drum" winch has seen notable evolution. The natural opposite to a "load-drum" is a "storage drum" as with Traction Winch systems. Storage "reels" do not qualify as winches, and will not be covered here.

A "new" rating definition has become more widely used. "Deck Lift" calls out the maximum "air-weight" which a winch can deploy. The number has been available on layer-by-layer drum charts, but it is being recognized as another important defining parameter.

Two interpretations of "deck-lift" require clarification. The more obvious is the winch drum's available output line pull at the full layer. This is where loads are handled at the deck and through the surface plane. Many drum proportions result in a lesser payload capacity being available with some amount of wire paid out. "Live Load" (sum of payload + drag) deducts the wire weight from drum pull. This non-linear value can reach a "minimum which is substantially less than the full-drum pull. If a payload is to be lowered below the depth where this "minimum" occurs, the winch will theoretically be unable to recover it -- baring drive overload availability. The conservative definition of "deck-lift" would use this minimum "live-load" value based on full depth deployment. However, since builders prefer to present their machines in the best light, the full-drum value seems likely to prevail.

AC-Variable Frequency winch drives have come of age, demonstrating enough reliability to become the cost-effective electric option over SCR-DC. Availability of modular planetary gear reducers has enlarged the layout window for the winch designer.

Hydraulic winches have kept pace, with the wider range of hydraulic components and the increased flexibility of hydraulic controls - with or without electronic and digital presence. Increased availability of large displacement hydraulic motors has changed the look of many winches. A small lower-cost motor plus a planetary unit makes an interesting option to a direct large motor, and can also simplify or eliminate conventional gear trains.

Off-the-shelf Programmable Logic Controllers (PLC) and higher-level digital drives in combination with precision spooling have provided the ability to offer "layer-compensated" output. This gives the same pull and speed at each layer of wire -- thus matching two of the features which have so far justified the cost, complexity and wire bends of the traction type machines.

Fiber-optic cables and multi-function umbilical cables have demanded large diameter drum cores and sheave suites, along with the provision of more elaborate slip-ring packages. Fiber optic cables are gaining in their ability to be stored at higher working tensions, and thereby reducing the low-tension storage advantage of the traction type winch systems.

A new method of spooling certain types of softer or less-circular tension members is worth consideration. Experience with the high performance soft lines used aboard the growing class of escort and ship-assist tugs has proved that the "open-lay" or "universal spooling" concept can greatly reduce line "pull-down" problems. These large braided lines are soft when slack, oval shaped around bends, and very slippery. It remains to be seen whether a fast fairlead traverse rate with wide gaps between turns and many cross-overs between layers will be a benefit with particular oceanographic tension members which have so far proved hard to spool cleanly.

A number of winches sized to carry 5000 and 6000 meters of wire have provided "enough" reach, while being smaller, lighter and less costly than the older standard of 10,000 meters. The practice of carrying around "one's spare wire" has become merely an option, in the face of limited space and economic constraints.

A semi-production class of portable winch systems with perhaps a maximum capacity of 3000 meters of 0.322" CTD cable (or its inevitable replacement...) could be a useful tool aboard the large and mid-size ships. One logical approach for this service would be a modularized electro-hydraulic suite with a single

460/3/60 “fleet-standard” electrical cable plug and receptacle configuration. In the interest of package location and maintenance access, it may be desirable to have the hydraulic power pack as a separate module which could be attached to the winch, or remote. An important feature would be drum interchangeability. If shorter cable lengths were to be used, spare drums with larger barrel diameters could be provided. Storage space for hoses and power cables, proper lifting eyes and fork-lift sockets, and perhaps frameworks to accept “shipping plywood” panels would add value to such systems.

Smaller yet are the 1000 and 2000 meter machines for the new class of inshore and near-shore boats. (see Photos # 6 & 7) As demands for environmental data proliferate, more jurisdictions and enterprises which include a beach will need scientific data of the same quality and quantity as the blue-water boats provide. Weight limits aboard 30 ft to 80 foot boats are leading to aluminum construction, and with the potential numbers of this type vessel, short-delivery semi-stock winch families are being developed. Often these near-shore boats see shorter and steeper waves than do the big ships. These conditions exacerbate the ship-motion and slack-wire problems, and require maximum winch controllability, along with careful cable-lead arrangements.

One frequently-requested feature has remained illusive. “Why not a “dithering” winch drum for motion-compensation?” After costly failed attempts, the fundamentals of rotational inertia appear to remain stubbornly in place. Instead, separate hardware such as the “nodding boom” and the “ram tensioner” appear to be the better way to help a winch move a payload at a constant rate relative to the earth as the ship’s overboard sheave dances in the bight of the wire.

Electronics have taken over the “cable-data” sub-systems. The mechanical tachometer and the wheel counter have taken their rightful places in the museums. (Except possibly for the smallest and most basic machines...) Line speed and count signals are provided by proximity sensors or optical encoders. Line tension signals from moving lines originate with strain-gages on sheave axle pins. Often these signals originate from sheaves which “ride” the traversing fairlead heads. It is important to provide a conductor-protecting slack-loop”, such as teflon-lined stainless armored hydraulic hose between the head and a fixed J-box on the winch.

The wire data is available on PC screens or dedicated PLC windows, as well as analog meters, and the numbers are readily routed to the vessel's own computers, and then onward to anywhere on or off the planet. Analog meters retain their natural advantage of providing an "instant visual impression" which is frequently useful to a busy operator. (See photo # 11)

The possibility of programming a complete winch time-motion "sequence" remains available, but this feature has seldom been used. Not everything which is possible is worth doing.

Chapter 10 Single-Drum or "Load-Drum" Research Winches

1.0 BASICS OF OPERATION

The single-drum research winch has been the data-gathering mainstay of sea-going wire handling and storage since the activity passed the hand-line and bucket stage.

When discussing these machines, the oceanographic community applies an interesting mix of English and Metric units. Wire diameters have been usually expressed in inches, although millimeters are creeping in. The length of wire payed out is commercially and scientifically voiced in meters, while the Navy generally calls out feet. Line pulls are most often in pounds, but kilograms and kiloNewtons are here.

Speeds are referred to in meters per minute, but many of us still design in feet per minute. Horsepower remains the term of capability, but kilowatts loom.

To convert kilowatts into horsepower, multiply kw by 1.341

To convert meters to feet, multiply meters by 3.2808

To convert millimeters to inches, multiply mm by 0.0394
(25.38 mm = 1 inch)
To convert kilograms to pounds, multiply kg by 2.205
To convert kiloNewtons to pounds, multiply kN by 224.82
(4.448 kiloNewtons equals 1000 pounds)

Winch builders do their best to speak in the terminology which the customer brings to the table.

Operators are beginning to call out the air-weight "DECK-LIFT" which a winch will handle -- most logically at the outer layer of the available wire. This "spec. item" is useful in that it can help shape drum proportions, and clearly tells the operator one very important performance boundary value.

Research winches handle wire, cable and umbilical sizes from 1/8" on up, with a wide application plateau at 0.680" and 0.681", and with no real end in sight.

Lengths of wire vary from a few hundred feet to as much as 43,000 feet. Empty winch weights range from 300 lb. to 80,000 lb. (the side-by-side two drum unit aboard the icebreaker “Nathaniel Palmer”.)

The term “Single Drum Winch” can be a misnomer, since two or three “Load Drums” can be combined on a single structure, in side-by-side or in “waterfall” arrangement. When each drum has its own independent drive the terms “dual winch” or “triple winch” are appropriate. The alternative is to employ a single drive and clutch-in the one desired drum.

These and other variations maintain the largely custom nature of this equipment. Efforts have been made to “standardize” a family of machines -- often the effort comes from a regulatory or government body. Each manufacture’s ingenuity, design approach and market focus has continued to confound the “standardizers. Having said that, each builder has a core of his own “good practice” which will be repeated as much as possible to help control engineering cost.

With smaller winches, a “quantity-built” design becomes feasible, although often “custom” variations will be required and the winch returns to custom or semi-custom status.

A primary advantage of a “Load Drum” winch is the directness of the cable’s path, when compared to a traction machine. At the winch, the cable bends itself continuously in the same direction with small radius changes as the drum fills or empties. Overboard sheaving is common to both types of winch.

With proper selection of sheave and drum barrel diameters, and the general application of the “Lebus” grooved shell, Load-Drum spooling is a relatively gentle process on the wire, and results in good wire life. Photographs # 8,9 & 10 show spooling the way it is intended to be. While factory or dockside drum loadings are “artificial” environments with controlled hold-back tension, the designer’s true delight arrives when the at-sea deployments behave in the same precision way.

It is worth noting that “Lebus” suggests that its shells are at their best up to approximately 16 layers. Few winch designs allow the width implied by such a shallow drum, and in fact the “Lebus” offsets remain visible and effective through many more layers. The forced “Lebus” crossovers will often migrate from their initial 180 degree locations, while continuing to exert wire control.

2.0 WINCH DRUMS -- DESIGN CHARTS -- CAPACITIES -- PROPORTIONS

A fundamental characteristic of the “Load Drum” Winch is the change of working radius which occurs as the drum fills and empties. Available line pull is the greatest at the barrel layer, whereas the greatest speed is available at the full drum. The spreadsheets which illustrates this radius-dependent output are the subject of this section.

2.1 Drum Charts

Many forms of spreadsheet can be written to show each layer’s geometry and the pulls and speeds available, once the winch’s design parameters have been selected. Pages 5, 6, 7, & 8 illustrate data for winches with three electric drives and one hydraulic drive. Page 5 shows a small 7-1/2 hp, AC-VF, semi-production winch. Page 6 shows a 50 hp SCR/DC machine with a single gear ratio. Page 7 shows a 75 hp SCR/DC unit with gear shift between two gear ratios, and page 8 shows a 100 hp, gear-shift AC/hydraulic machine.

DRUM CAPACITY AND PERFORMANCE CHART

MARKEY RESEARCH WINCH DRUM & PERFORMANCE CHART
MMCo. Type = COM-07-X2CL   Done For
MMCo. SERIAL @ 17708

DATE 02-01-2000
DRIVE TYPE = AC Variable Frequency   7-1/2 HP, A.C. V-F

Cable Dia	0.322	Fce Width	21.000	Wt.in H20	0.144 lb.	Fast rpm	2,520
Flang Dia	26.750			Ult.Stngth	10,000 lb. free end	Slow rpm	1,800 Base
Barel Dia	12.190			Deploy	6,562 feet	RPMratio	1.400
Flng Dpth	7.280			Barel Pull	3,950 lb.	BaseFreq	60 Hertz
Act Layers	22.609			Line Speed	52 ft/min (@ barrel)	Max Freq	84 Hertz
USE LAYERS	21.000						

Layer Number	Pitch Diam.	Feet/ Wrap	Feet/ Layer	Feet ON	Feet OFF	H20 Wt. of Wire	BASE MOTOR RATING 60 Hz: PULL	Live Load	Ft/ Min	Meters/ Min	FREQUENCY ENHANCED 84 Hz: PULL	Live Load	Ft/ Min	Meters\ Min	Layer Number
24	27.324	7.153	454	7,947	-931	-134	1,975	2,109	113	34.4	1,339	1,473	158	48.2	24
23	26.680	6.985	444	7,493	-487	-70	2,022	2,092	110	33.6	1,371	1,442	154	47.0	23
22	26.036	6.816	433	7,049	-54	-8	2,072	2,080	108	32.8	1,405	1,413	151	45.9	22
21	25.392	6.648	422	6,616	368	53	2,125	2,072	105	32.0	1,441	1,388	147	44.8	21
20	24.748	6.479	411	6,194	779	112	2,180	2,068	102	31.2	1,479	1,366	143	43.6	20
19	24.104	6.310	401	5,783	1,180	170	2,238	2,068	100	30.4	1,518	1,348	139	42.5	19
18	23.460	6.142	390	5,382	1,570	226	2,300	2,074	97	29.5	1,560	1,334	136	41.4	18
17	22.816	5.973	379	4,992	1,949	281	2,365	2,084	94	28.7	1,604	1,323	132	40.2	17
16	22.172	5.805	369	4,613	2,318	334	2,433	2,100	92	27.9	1,650	1,317	128	39.1	16
15	21.528	5.636	358	4,244	2,676	385	2,506	2,121	89	27.1	1,700	1,314	125	38.0	15
14	20.884	5.467	347	3,886	3,023	435	2,583	2,148	86	26.3	1,752	1,317	121	36.8	14
MID 13	20.240	5.299	336	3,539	3,359	484	2,666	2,182	84	25.5	1,808	1,324	117	35.7	13 MID
12	19.596	5.130	326	3,203	3,685	531	2,753	2,223	81	24.7	1,867	1,337	113	34.6	12
11	18.952	4.962	315	2,877	4,000	576	2,847	2,271	78	23.9	1,931	1,355	110	33.4	11
10	18.308	4.793	304	2,562	4,305	620	2,947	2,327	76	23.1	1,999	1,379	106	32.3	10
9	17.664	4.624	294	2,257	4,598	662	3,054	2,392	73	22.2	2,071	1,409	102	31.1	9
8	17.020	4.456	283	1,964	4,881	703	3,170	2,467	70	21.4	2,150	1,447	98	30.0	8
7	16.376	4.287	272	1,681	5,153	742	3,295	2,552	68	20.6	2,234	1,492	95	28.9	7
6	15.732	4.119	262	1,409	5,415	780	3,429	2,650	65	19.8	2,326	1,546	91	27.7	6
5	15.088	3.950	251	1,147	5,666	816	3,576	2,760	62	19.0	2,425	1,609	87	26.6	5
4	14.444	3.781	240	896	5,906	850	3,735	2,885	60	18.2	2,533	1,683	84	25.5	4
3	13.800	3.613	229	656	6,135	883	3,910	3,026	57	17.4	2,651	1,768	80	24.3	3
2	13.156	3.444	219	427	6,354	915	4,101	3,186	54	16.6	2,781	1,866	76	23.2	2
1	12.512	3.276	208	208	6,562	945	4,312	3,367	52	15.8	2,924	1,979	72	22.1	1

MjM disc #185
A:17708.cht

PERFORMANCE CHART

MARKEY RESEARCH WINCH DRUM & PERFORMANCE CHART   DATE= JUL-17-97
MMCo. Type DESH-5   Done For   DRIVE TYPE   DC Elect 50 hp

Cable Dia	0.322	Fce Width	28.000	Wt.in H20	0.139 lb.	Fast rpm	1,800
Flang Dia	40.000			Ult.Stngth	11,000 lb.	Slow rpm	1,200
Barel Dia	18.000			Deploy	19,674 feet	RPMratio	1.500
Flng Dpth	11.000			Barel Pull	12,000 lb.	Sftratio	3.070
Act Layers	34.161						
USE LAYERS	36.000						

Layer Number	Pitch Diam.	Feet/ Wrap	Feet/ Layer	Feet ON	Feet OFF	H20 Wt. of Wire	PULL	Live Load	Ft/ Min	Meters/ Min	PULL	Live Load	Ft/ Min	Me…
34	39.574	10.360	891	22,160	-1,595	-222	5,556	5,777	220	67.2	3,704	3,926	330	10
MAX 33	38.930	10.192	877	21,269	-718	-100	5,648	5,748	217	66.1	3,765	3,865	325	9
DESIGN 32	38.286	10.023	862	20,392	144	20	5,743	5,723	213	65.0	3,828	3,808	320	9
31	37.642	9.855	848	19,530	991	138	5,841	5,703	210	63.9	3,894	3,756	314	9
30	36.998	9.686	833	18,683	1,824	254	5,943	5,689	206	62.8	3,962	3,708	309	9
29	36.354	9.517	819	17,850	2,643	367	6,048	5,681	202	61.7	4,032	3,665	304	9
28	35.710	9.349	804	17,031	3,447	479	6,157	5,678	199	60.6	4,105	3,626	298	9
27	35.066	9.180	790	16,227	4,236	589	6,270	5,681	195	59.5	4,180	3,591	293	8
26	34.422	9.012	775	15,438	5,011	697	6,387	5,691	192	58.4	4,258	3,562	287	8
25	33.778	8.843	761	14,663	5,772	802	6,509	5,707	188	57.3	4,339	3,537	282	8
24	33.134	8.674	746	13,902	6,518	906	6,636	5,730	184	56.2	4,424	3,518	277	8
23	32.490	8.506	732	13,156	7,249	1,008	6,767	5,759	181	55.1	4,511	3,504	271	8
22	31.846	8.337	717	12,425	7,966	1,107	6,904	5,797	177	54.0	4,603	3,495	266	8
21	31.202	8.169	703	11,708	8,669	1,205	7,046	5,842	174	52.9	4,698	3,493	261	7
MID 20	30.558	8.000	688	11,005	9,357	1,301	7,195	5,894	170	51.9	4,797	3,496	255	7
SCOPE 19	29.914	7.831	674	10,317	10,030	1,394	7,350	5,956	167	50.8	4,900	3,506	250	7
18	29.270	7.663	659	9,644	10,689	1,486	7,512	6,026	163	49.7	5,008	3,522	244	7
17	28.626	7.494	645	8,985	11,334	1,575	7,681	6,105	159	48.6	5,120	3,545	239	7
16	27.982	7.326	630	8,340	11,964	1,663	7,857	6,194	156	47.5	5,238	3,575	234	7
15	27.338	7.157	616	7,710	12,579	1,749	8,042	6,294	152	46.4	5,362	3,613	228	6
14	26.694	6.988	601	7,095	13,180	1,832	8,236	6,404	149	45.3	5,491	3,659	223	6
13	26.050	6.820	587	6,494	13,767	1,914	8,440	6,526	145	44.2	5,627	3,713	218	6
12	25.406	6.651	572	5,907	14,339	1,993	8,654	6,661	141	43.1	5,769	3,776	212	6
11	24.762	6.483	558	5,335	14,896	2,071	8,879	6,808	138	42.0	5,919	3,849	207	6
10	24.118	6.314	543	4,778	15,439	2,146	9,116	6,970	134	40.9	6,077	3,931	201	6
9	23.474	6.145	529	4,235	15,968	2,220	9,366	7,147	131	39.8	6,244	4,025	196	5
8	22.830	5.977	514	3,706	16,482	2,291	9,630	7,340	127	38.7	6,420	4,129	191	5
7	22.186	5.808	500	3,192	16,981	2,360	9,910	7,550	124	37.6	6,607	4,246	185	5
6	21.542	5.640	485	2,693	17,466	2,428	10,206	7,778	120	36.6	6,804	4,376	180	5
5	20.898	5.471	471	2,208	17,937	2,493	10,521	8,028	116	35.5	7,014	4,521	175	5
4	20.254	5.302	456	1,737	18,393	2,557	10,855	8,299	113	34.4	7,237	4,680	169	5
3	19.610	5.134	442	1,281	18,834	2,618	11,212	8,594	109	33.3	7,475	4,857	164	4
2	18.966	4.965	427	840	19,261	2,677	11,593	8,915	106	32.2	7,728	5,051	158	4
1	18.322	4.797	413	413	19,674	2,735	12,000	9,265	102	31.1	8,000	5,265	153	4

Base Motor Rating____________ Field-Weakened Rating___

/ RESEARCH WINCH DRUM & PERFORMANCE CHART
Type DESH-5 S/N 16665 Done For

DATE= December 15, 1995
DRIVE TYPE 75HP AC/SCR DC Electric

Wire Diameter = 0.322 in.
Ultimate Strength of Wire = 10,000 lbs

Length of Wire = 10,000 meters
Layers Used = 37

Drum Size: 18" Barrell Diameter X 38" Face Width X 44" Flange Diameter

							HIGH PULL GEARING RANGE: Base Motor Rating			HIGH PULL GEARING RANGE: Field Weakended Rating			HIGH SPEED GEARING RANGE: Base Motor Rating			HIGH SPEED GEARING RANGE: Field Weakended Rating				
er er	Pitch Diam. (METERS)	Meter/ Wrap	Meter/ Layer	Meter ON	Meter OFF	H20 Wt. of Wire (LBS)	PULL (LBS)	Live Load (LBS)	Meter/ Min	PULL (LBS)	Live Load (LBS)	Meter/ Min	PULL (LBS)	Live Load (LBS)	Meter/ Min	PULL (LBS)	Live Load (LBS)	Meter/ Min	Layer Number	
39	1.109	3.485	402	11,298	-896	-391	5,275	5,666	107	3,293	3,684	162	3,531	3,922	160	2,204	2,595	243	39	MAXIMUM CAPACITY
38	1.093	3.433	397	10,896	-499	-218	5,354	5,572	105	3,343	3,560	160	3,584	3,802	157	2,237	2,455	239	38	
37	1.076	3.382	391	10,499	-109	-47	5,436	5,483	104	3,393	3,441	158	3,638	3,686	155	2,271	2,319	236	37	DESIGNED CAPACITY
36	1.060	3.330	385	10,109	276	120	5,519	5,399	102	3,446	3,325	155	3,694	3,574	152	2,306	2,186	232	36	
35	1.044	3.279	379	9,724	655	286	5,606	5,320	100	3,500	3,214	153	3,752	3,467	150	2,342	2,057	228	35	
34	1.027	3.228	373	9,345	1,027	448	5,695	5,247	99	3,555	3,107	151	3,812	3,364	148	2,380	1,931	225	34	
33	1.011	3.176	367	8,973	1,394	608	5,787	5,179	97	3,613	3,005	148	3,874	3,265	145	2,418	1,810	221	33	
32	0.995	3.125	361	8,606	1,755	766	5,882	5,117	96	3,672	2,906	146	3,937	3,172	143	2,458	1,692	218	32	
31	0.978	3.074	355	8,245	2,110	921	5,981	5,060	94	3,734	2,813	143	4,003	3,082	141	2,499	1,578	214	31	
30	0.962	3.022	349	7,890	2,459	1,073	6,083	5,009	93	3,797	2,724	141	4,071	2,998	138	2,542	1,469	211	30	
29	0.946	2.971	343	7,541	2,802	1,223	6,188	4,965	91	3,863	2,640	139	4,142	2,919	136	2,586	1,363	207	29	
28	0.929	2.919	337	7,198	3,140	1,370	6,297	4,927	89	3,931	2,561	136	4,215	2,845	134	2,631	1,261	203	28	
27	0.913	2.868	331	6,860	3,471	1,515	6,410	4,895	88	4,001	2,487	134	4,290	2,776	131	2,678	1,164	200	27	
26	0.897	2.817	325	6,529	3,796	1,656	6,526	4,870	86	4,074	2,418	131	4,368	2,712	129	2,727	1,071	196	26	
25	0.880	2.765	319	6,204	4,116	1,796	6,648	4,852	85	4,150	2,354	129	4,450	2,654	127	2,778	982	193	25	
24	0.864	2.714	313	5,884	4,429	1,933	6,774	4,841	83	4,229	2,296	127	4,534	2,601	124	2,830	898	189	24	
23	0.847	2.662	308	5,571	4,736	2,067	6,904	4,838	82	4,310	2,244	124	4,621	2,555	122	2,885	818	185	23	
22	0.831	2.611	302	5,264	5,038	2,198	7,040	4,842	80	4,395	2,197	122	4,712	2,514	120	2,942	743	182	22	MID-SCOPE
21	0.815	2.560	296	4,962	5,334	2,327	7,182	4,854	78	4,483	2,156	119	4,807	2,480	117	3,001	674	178	21	
20	0.798	2.508	290	4,666	5,623	2,454	7,329	4,875	77	4,575	2,121	117	4,905	2,452	115	3,062	609	175	20	
19	0.782	2.457	284	4,377	5,907	2,578	7,482	4,904	75	4,671	2,093	115	5,008	2,430	112	3,126	549	171	19	
18	0.766	2.405	278	4,093	6,185	2,699	7,642	4,943	74	4,771	2,072	112	5,115	2,416	110	3,193	494	168	18	
17	0.749	2.354	272	3,815	6,457	2,817	7,809	4,991	72	4,875	2,057	110	5,227	2,409	108	3,263	445	164	17	
16	0.733	2.303	266	3,543	6,723	2,934	7,983	5,049	71	4,984	2,050	107	5,343	2,410	105	3,336	402	160	16	
15	0.717	2.251	260	3,277	6,983	3,047	8,165	5,118	69	5,097	2,050	105	5,465	2,418	103	3,412	365	157	15	
14	0.700	2.200	254	3,017	7,237	3,158	8,356	5,198	67	5,216	2,059	103	5,593	2,435	101	3,492	334	153	14	
13	0.684	2.149	248	2,763	7,485	3,266	8,556	5,290	66	5,341	2,075	100	5,727	2,461	98	3,575	309	150	13	
12	0.668	2.097	242	2,515	7,727	3,372	8,765	5,394	64	5,472	2,100	98	5,867	2,495	96	3,663	291	146	12	
11	0.651	2.046	236	2,273	7,964	3,475	8,986	5,511	63	5,610	2,135	95	6,014	2,540	94	3,755	280	143	11	
10	0.635	1.994	230	2,036	8,194	3,575	9,217	5,642	61	5,754	2,179	93	6,169	2,594	91	3,851	276	139	10	
9	0.618	1.943	224	1,806	8,418	3,673	9,461	5,788	60	5,906	2,233	91	6,333	2,659	89	3,953	280	135	9	
8	0.602	1.892	218	1,582	8,637	3,769	9,718	5,949	58	6,067	2,298	88	6,505	2,736	87	4,061	292	132	8	
7	0.586	1.840	213	1,363	8,849	3,861	9,989	6,128	56	6,236	2,375	86	6,686	2,825	84	4,174	313	128	7	
6	0.569	1.789	207	1,151	9,056	3,952	10,276	6,325	55	6,415	2,464	83	6,878	2,927	82	4,294	342	125	6	
5	0.553	1.737	201	944	9,257	4,039	10,580	6,541	53	6,605	2,566	81	7,082	3,043	80	4,421	382	121	5	
4	0.537	1.686	195	743	9,451	4,124	10,903	6,779	52	6,806	2,682	79	7,298	3,174	77	4,556	432	117	4	
3	0.520	1.635	189	549	9,640	4,207	11,245	7,039	50	7,020	2,814	76	7,527	3,321	75	4,699	493	114	3	
2	0.504	1.583	183	360	9,823	4,286	11,611	7,324	49	7,248	2,962	74	7,771	3,485	72	4,852	565	110	2	
1	0.488	1.532	177	177	10,000	4,364	12,000	7,636	47	7,491	3,128	71	8,032	3,669	70	5,014	651	107	1	

Base Motor Rating ———— Field Weakended Rating
<-------------- HIGH PULL GEARING RANGE -------------->

Base Motor Rating ———— Field Weakended Rating
<-------------- HIGH SPEED GEARING RANGE ------------->

MARKEY RESEARCH WINCH DRUM & PERFORMANCE CHART — DATE= 10/11/97

MMCo. Type DUSH-5 — Done For — DRIVE TYPE = Hydraulic

Cable Dia	0.322	Fce Width	38.000	Wt.in H20	0.144 lb.		
Flang Dia	44.000			Ult.Stngth	10,000 lb.		
Barel Dia	18.750			Deploy	32,808 feet		
Flng Dpth	12.625	Air Gap	0.487	Barel Pull	11,000 lb.	Sftratio	1.318
Act Layers	39.208	Gap % /100	0.013	Line Speed	165 ft/min (@ barrel)		

	Layer Number	Pitch Diam.	Meter/ Wrap	Meter/ Layer	Meters ON	Meters OFF	H20 Wt. of Wire	PULL	Live Load	Ft/ Min	Meters/ Min	PULL	Live Load	Ft/ Min	Met… Mi…
	40	44.188	3.526	411	11,762	-1,351	-195	4,748	4,942	382	116.5	3,602	3,797	504	153
	39	43.544	3.475	405	11,351	-946	-136	4,818	4,954	377	114.8	3,655	3,792	497	151
	38	42.900	3.423	399	10,946	-547	-79	4,890	4,969	371	113.1	3,710	3,789	489	149
	37	42.256	3.372	393	10,547	-155	-22	4,965	4,987	366	111.4	3,767	3,789	482	146
	36	41.612	3.320	387	10,154	232	33	5,042	5,008	360	109.7	3,825	3,792	474	144
	35	40.968	3.269	381	9,768	613	88	5,121	5,033	354	108.0	3,885	3,797	467	142
	34	40.324	3.218	375	9,387	988	142	5,203	5,060	349	106.3	3,947	3,805	460	140
	33	39.680	3.166	369	9,012	1,357	195	5,287	5,092	343	104.6	4,011	3,816	452	137
	32	39.036	3.115	363	8,643	1,720	248	5,374	5,127	338	102.9	4,078	3,830	445	135
	31	38.392	3.064	357	8,280	2,077	299	5,464	5,165	332	101.2	4,146	3,847	438	133
	30	37.748	3.012	351	7,923	2,428	350	5,558	5,208	327	99.5	4,217	3,867	430	131
	29	37.104	2.961	345	7,572	2,772	399	5,654	5,255	321	97.8	4,290	3,891	423	129
	28	36.460	2.909	339	7,227	3,111	448	5,754	5,306	315	96.1	4,366	3,918	416	126
	27	35.816	2.858	333	6,888	3,444	496	5,857	5,362	310	94.4	4,444	3,948	408	124
	26	35.172	2.807	327	6,555	3,771	543	5,965	5,422	304	92.7	4,526	3,983	401	122
	25	34.528	2.755	321	6,229	4,092	589	6,076	5,487	299	91.0	4,610	4,021	394	120
	24	33.884	2.704	315	5,908	4,407	635	6,191	5,557	293	89.4	4,698	4,063	386	117
	23	33.240	2.652	309	5,593	4,716	679	6,311	5,632	288	87.7	4,789	4,109	379	115
MID	22	32.596	2.601	303	5,284	5,019	723	6,436	5,713	282	86.0	4,883	4,160	372	113
	21	31.952	2.550	297	4,981	5,316	766	6,566	5,800	276	84.3	4,982	4,216	364	111
	20	31.308	2.498	291	4,683	5,607	807	6,701	5,893	271	82.6	5,084	4,277	357	108
	19	30.664	2.447	285	4,392	5,893	849	6,842	5,993	265	80.9	5,191	4,342	350	106
	18	30.020	2.395	279	4,107	6,172	889	6,988	6,100	260	79.2	5,302	4,414	342	104
	17	29.376	2.344	273	3,828	6,445	928	7,142	6,214	254	77.5	5,419	4,490	335	102
	16	28.732	2.293	267	3,555	6,712	966	7,302	6,335	249	75.8	5,540	4,573	328	99
	15	28.088	2.241	261	3,288	6,973	1,004	7,469	6,465	243	74.1	5,667	4,663	320	97
	14	27.444	2.190	255	3,027	7,228	1,041	7,644	6,604	237	72.4	5,800	4,759	313	95
	13	26.800	2.139	249	2,772	7,477	1,077	7,828	6,751	232	70.7	5,939	4,863	306	93
	12	26.156	2.087	243	2,523	7,720	1,112	8,021	6,909	226	69.0	6,086	4,974	298	90
	11	25.512	2.036	237	2,280	7,957	1,146	8,223	7,077	221	67.3	6,239	5,093	291	88
	10	24.868	1.984	231	2,042	8,189	1,179	8,436	7,257	215	65.6	6,401	5,222	284	86
	9	24.224	1.933	225	1,811	8,414	1,212	8,661	7,449	210	63.9	6,571	5,359	276	84
	8	23.580	1.882	219	1,586	8,633	1,243	8,897	7,654	204	62.2	6,750	5,507	269	82
	7	22.936	1.830	213	1,367	8,846	1,274	9,147	7,873	198	60.5	6,940	5,666	262	79
	6	22.292	1.779	207	1,154	9,054	1,304	9,411	8,107	193	58.8	7,140	5,837	254	77
	5	21.648	1.727	201	946	9,255	1,333	9,691	8,358	187	57.1	7,353	6,020	247	75
	4	21.004	1.676	195	745	9,450	1,361	9,988	8,627	182	55.4	7,578	6,217	239	73
	3	20.360	1.625	189	550	9,639	1,388	10,304	8,916	176	53.7	7,818	6,430	232	70
	2	19.716	1.573	183	361	9,823	1,414	10,641	9,226	171	52.0	8,073	6,659	225	68
	1	19.072	1.522	177	177	10,000	1,440	11,000	9,560	165	50.3	8,346	6,906	217	66

<- HIGH PULL / LOW SPEED ->   <- LOW PULL / HIGH SPEED

At MMCo. we choose to locate the barrel layer (#1) at the bottom of the chart, and work upwards, since this gives a visual analog to what one sees when looking at an actual drum as it is filling.

The normal "Load Winch" is based on having single-value drum torques and drum revs/min available across all layers, at each "point" on the drive's performance envelope.

The conservative approach to a Drum Chart is to assume that the wires stack vertically atop each other. It is clear that "cannon-ball" stacking does take place, with each wire laying in the valley between its two lower supporters. However, at the layer "cross-over" points, the wires ARE radially aligned. When a Lebus shell is used, two cross-overs occur at 180 degrees apart on each turn. Thus vertical stacking makes sense for the spreadsheet, and gives conservative drum volumes, with flange margin. The left portion of a typical chart defines the geometry and the summations of wire footage and weight as the drum fills. "Inverse" data columns show footage and wire-weight payed out. It should be noted that these two columns depend upon filling the drum to its rated capacity. If a drum is loaded only half way, all the chart rating data changes since less wire weight applies at each layer. New charts can be quickly prepared for partial drum loads.

The important portions of each chart are the data columns shown at the "right." The format will vary depending upon the number of gear ratios available, as well as how many motor speed/pull ratings are defined for the winch drive.

For each "rating point" on the performance map, the drum output pull and speed are linear with radius change. We show the line speeds in both feet/min and in meters/min. The more interesting data column at each rating point is that known as "LIVE LOAD."

2.1.1 <u>"Live Load"</u>

Live Load is the simple subtraction of the accumulated wire weight from the available drum line pull at each layer. Considering the wire's weight as parasitic leaves a non-linear summation of forces which is much more useful to the winch operator than the basic drum pull value.

Live Load is also definable as the sum of the payload's in-water weight, it's drag, and the drag of the cable. Small acceleration loads are usually negligible.

Drag values can be estimated as non-linear functions of the payload's speed through the water, as can the cable's drag under a variety of conditions. In practice, this level of "science" is usually overkill, and in any case not accurately available until the winch is designed, built and deployed with real payloads. The Live Load sum is an early design value, and gives the operator an adequate guide.

Scanning a Live Load data column will show that the values usually trend through a shallow "minimum." Where that occurs depends upon the drum proportions. Taken literally, the lowest value of Live Load is the greatest "weight + drag" which can be payed out to full wire depth and successfully recovered. With a marginal winch drive or a unusually high load, hoisting speed can usually be reduced to shift available energy over from "drag" to "weight coming up."

Electric systems mask the importance of the minimum Live Load by providing built-in overload capability. Simple hydraulic systems reach their relief valve pressure settings and just stall. The elaborate hydraulic circuits are more flexible, as will be discussed later.

With multi-ratio gearing and multi-speed drives, the "Live Load" data will change from rating point to rating point out of all proportion to the drive rating differences. The wire weight remains the same, and seeking higher line speeds can reduce Live Load so much that the faster operating points become useful only for high-speed payout.

2.2 Drum Proportions

2.2.1 Drum "Barrel" or "Core" Selection

A prospective winch owner should be prepared to define the primary wire by diameter and length. Projecting future applications over a thirty-plus year winch life is an awkward but necessary exercise, and often leads to the provision of "easily" interchangeable drums and quick-change fairlead drive components. The initial projected wire parameters will define the drum and the rest of the winch.

The diameter ratio between the drum barrel and the wire (D/d) is critical. The wire or cable builders normally provide a minimum D/d as part of their specification, and this value can be used to size the barrel -- hoping that the winch will never need to handle a “stiffer” or larger product.

At least four reality factors suggest that the winch should provide a somewhat larger barrel than the initial wire’s minimum.

a) Loads become larger over the years.
b) New payloads are appearing rapidly, with requirements for differing wires, EM cables or Fiber Optic cables.
c) Wire life is improved when it is spooled at a larger radius than that which the wire builder considers minimum.
d) A larger barrel results in fewer wire layers. Spooling benefits from keeping the number of layers as small as practical.

Three contravening factors mitigate against selection of a larger-than-minimum drum. These are space, weight and cost !

D/d ratios vary enormously for different winch applications. Escort tugs can wrap 4” diameter braided “Spectra” or “Plasma” type soft ropes onto a 30” dia. barrel, for a 7-1/2 to 1 D/d. Steel towing wires should see 15 to 1, but are often fudged down to 13 to 1 with the assumption that the barrel layer is seldom used.

The UNOLS “standard” 0.322” CTD cable shows a 12” minimum diameter, or 37 to 1. One 0.680” EM cable shows a 28” minimum diameter, for 41 to 1. A typical 0.681” fiber-optic cable shows a 48” minimum diameter, for 70 to 1. This illustrates the need to know or assume the intended tension member at the start of the design process.

2.2.2 Drum Width and Flange Dimensions

With the barrel diameter established, the designer will balance face width against flange diameter, based upon experience and what “looks right.” Occasionally available space will dictate a long and shallow drum. This requires a stiffer drum structure and fairlead assembly to avoid deflections. At the opposite extreme, MMCo. has provided wire rope anchor windlasses where a tall and narrow drum was demanded by the overboarding layout and where a fairleader was not practical.

Excessive drum depth increases the flange steel's tendency to creep into a permanently deformed "hour-glass" or cone shape. If the face width at the outside is greater than that at the barrel, spooling suffers. Contrary to common sense, the small research wires require stiffer drum flanges than do those for large towing wires which see much higher pull ratings. Good practice calls for heavy duty flanges, with clean-up cuts on the inside to maintain square and parallel drum ends.

If a sample were taken of many research winches built by the many good builders, we suspect that there would be a rather narrow grouping of empirical values which constitute rational drum proportions.

Flange margin should be provided, beyond what the design Drum Chart shows as the outer layer's diameter. Spooling can become irregular for various reasons, with wire piling up at one end -- not desirable, but it happens. A better reason is to allow for future installation of a larger wire. For 0.322" cable, if the design drum chart shows a 41" o.d. for the outer layer, it would be reasonable to call out a 44" diameter flange.

3.0 "LEBUS" GROOVED SHELLS

The Lebus shell is essentially a grooved cylinder, manufactured from either steel, aluminum or fiberglass, that is designed to assure the proper seating of a specific rope or cable and the proper movement and spacing of that wire between the flanges of the winch drum. In order to effectively use the Lebus shells the winch drum must have flanges that are perpendicular to the barrel or core of the smooth winch drum. The shells, when delivered are split for easy installation on the winch or take-up spool and attachment can be accomplished by either welding or bolting the shell in place. Where winch systems utilize more than one size of wire or cable during their operational life, it is recommended that the bolt-on technique be used. Except in special situations, most research winches NEED this shell. If spare interchangeable drums are provided, each may require its own shell.

The unique twin half-wire-width crossover feature positively locates the barrel layer of wire, and allows the inter-wrap air gap to be much closer than does a smooth barrel. A "Lebus-spooled" air gap can be in the order of 1-1/2 to 2% of the "load-settled" slack wire diameter. If a deck engineer must "hammer" the first layer onto a bare drum with a stick and a maul, a design air gap of 6% to 9% is necessary. This reduces drum capacity and unsettles the fairleading.

Good practice requires that a sample of the ACTUAL cable be sent to the "Lebus" firm for incremental diameter checks at varying loads, both increasing and decreasing. For small wires a 30 foot sample works well. Given a tabulation of the diameter versus load, and the degree to which the sample's final diameter remains below its design nominal size, the winch designer can determine how may wraps should be spooled onto each layer.

For reasons which "Lebus" can explain, the winch builder should allow face width for a number of wraps which is an INTEGER PLUS ONE HALF. When calculating his fairlead drive, he must recognize that the number of drum turns is 1/2 fewer than the number of Lebus grooves. e.g., 100-1/2 grooves give 100 turns. The section covering fairleaders will expand on the implications.

4.0 WINCH PERFORMANCE

4.1 Line Pull Ratings versus Wire Strength

Given the likelihood that a research winch will need to handle increasing payloads over its decades of service, and the equal likelihood that a now-standard cable, e.g. the UNOLS pool 0.322" EM cable, will eventually give way to a more capable tension member, the question of "how much pull" is not simple. At one boundary, a ship's available winch drive power may force the balancing of pull versus speed -- recognizing that there must be enough line pull to get the work done, or "don't bother." Speed is the secondary parameter.

With adequate power available, the breaking strength of the present and potential future wires enters the picture. One arbitrary approach would be to design the winch drive and scantlings to break the basic wire at the barrel layer. The drum geometry and its Chart will show whether sufficient "Deck Lift" is available with the drum filled. Often a winch will be given the increased ability to break the wire "up the stack" in order to increase the pulls available higher up, (less wire out) where most of the work may be done.

Dipping, towing and core pull-out all effect the thought process, as does the possibility of a load or wire becoming snagged. It is clearly impractical to carry wire that an R/V's propulsion can't break, (as opposed to a tug..,) and it is equally necessary that the winch and the entire rigging system be tough enough to break any intended wire without mechanical damage.

RATED LINE PULL is best determined by close consideration of the projected payloads, experience, and the operator's instinct, as passed through the hands of the system specifier.

Multi-range winches will often be run most of their lives in the higher-speed-and-reduced-pull-range, with the "cable-parting" grunt-range reserved for special situations.

4.2 Line Speed

Hoisting speeds are often a case of "more is better." Assuming proper speed control to allow gentle handling of sampling nets, etc., minimizing time on station calls for all the hoisting speed the ship's power supply will allow, without increasing the drag forces beyond reason. The multi-range winch package recognizes the fact that most loads are not at the pull-maximum, and provides higher speeds at reduced load, in a variety of ways.

Lowering speed is a function of how fast the payload will fall through the water; a widely variable number. Many operators have set 100 M/min as a nominal maximum. Slack wire is a high-level "No-No", since it allows the wire to hackle or kink. Wire breakage (and another package insurance claim), serious loss of wire strength, or at least a spooling mess results.

From a design standpoint, most winches will payout as fast as they will hoist. It becomes the responsibility of the man on the joystick, with the help of a good tension metering system, to avoid excessive payout speeds. As will be noted later, a computer can take over part of the control task, but SEAMANSHIP cannot be eliminated. The winch driver should be a "top-hand."

5.0 WINCH DRIVES

Steam is a magnificent fluid for powering a winch. With the re-powering of the "Atlantis II" decades ago, and the electrification of its 10"x10" integrated two-cylinder steam trawl winch, that era ended. The present alternatives are electrical drives or hydraulic drives.

5.1 Power Determination

With the pulls and speeds selected, the basic "drum output power" comes from the two well-known relationships:

$$\text{H.P. out} = \frac{\text{Line Pull (lb.) x Line Speed (ft/min)}}{33{,}000}$$

$$\text{H.P. out} = \frac{\text{Output torque (in-lb.) x Output Speed (rev/min)}}{63{,}000}$$

$$\text{H.P. input} = \text{H.P. out x overall winch efficiency}$$

Mechanical efficiencies for machines with spur or planetary reducers and fairleaders will range between 80% and 85%. If worm gearing is involved, an efficiency of 70% to 75% is appropriate. Each manufacturer will have his own design values, ranging from hopeful to conservative.

5.1.1 Electric Power Rating

With a known input power requirement, the electric winch motor can be called out. Conservative practice would always round the nameplate rating UPWARD, to provide reserve, and allow for the fact that all machinery ages. If the nameplate power is close to the calculated need, it may be acceptable to "push" the assumptions and round the electric motor rating DOWNWARD. The "new-standard" squirrel-cage motors used with variable-frequency drives are NEMA-B in design, and provide a 1.0 service factor.

The power rating of an electric winch is usually the nameplate rating on the winch motor itself, without reference to the electrical losses back to the buss.

5.1.2 Hydraulic Power Rating

a) PRESSURE MAKES MOTOR TORQUE, WHICH CREATES LINE PULL.

b) VOLUME MAKES MOTOR SPEED, WHICH CREATES LINE SPEED.

Hydraulic drives require the designer and the operator to initially almost ignore "horsepower", and to think in the two separate quantities of "torque" and "speed."
"Horsepower" eventually brings these two back together at the upstream pump drive, but all the intermediate calculations are based on the two separate factors.

A useful preliminary "Rule Of Thumb" says that "1 h.p. into a pump will raise the pressure of 1 g.p.m., by 1,500 psi.

The ratio between "drum power out" and "pump power demand" can be as "good" as 1.6 to 1, or as inefficient as 2.0 to 1. This is a lively issue among hydraulic marketing people. The "Power Rating" of a hydraulic system is usually called out as the rating of the motor or engine used to drive the hydraulic pump.

5.2 Electric Drives

Excepting only the smallest and simplest of applications, the single-speed or even two-speed push-button "START-STOP" drive is not useful. Smooth and predictable variable speed control is an irreducible minimum.

5.2.1 Direct Current (D.C.)

The older motor-generator type of D.C. drive has followed the steam engine to the museum, as has the constant-voltage "street-car-controller". With "Silicon Controlled Rectifiers" (SCR), the benefits of DC-driven winches can be provided on AC-powered vessels. 100% rated design torque can be maintained down to slow speeds, and by providing higher-than-rated amperage for short times, torque-boost is available. This can provide short time pull increases, such as might be required when pulling a core.

By the process of reducing the motor's field current, higher speeds above "rated" are available, with reduced torque output. An analogy exists between this ability and the "H.P. Limiter" type of hydraulic pump control.

Two important characteristics limit the utility of D.C. drives. Stalled operation is not available since the commutator bars will shortly be destroyed. Carbon-brush wear is a continuing maintenance chore. At voltage-controlled slow speeds, the heat generated will require external blower ventilation.

An occasional winch design will require multiple motors. DC motors have the ability to "lean on the load" together, and divide the work nicely.

5.2.2 Alternating Current (A.C.)

Electronic controls have come of age and provide good reliability. Several levels of "Variable-Frequency" drives are available, with the distinctions better left for the electrical specialist to explain. The terms "Vector" and "Half-Vector" are among those which can mean different things to different drive suppliers. Motors are of the "squirrel-cage" type, although they require specific construction and cooling details to tolerate the artificial "choppy" AC sine-waves which the V.F. drives provide.

The "Variable-Frequency" drive operates by taking in the ship's standard AC power, rectifying it into D.C., and then electronically creating artificial A.C. power at essentially any desired frequency. The squirrel-cage motor responds to the frequency and produces the desired speed.

It is possible to provide 100% rated torque at zero or stall speed, with proper cooling required if the time at stall is "more than a little." The drives are not limited to the source's 60 Hz, but can provide motor frequencies up to 120 Hz. For winch applications, enhanced frequencies of 80 to 90 Hz. will keep the higher motor speeds within reason. Above the "base frequency -- usually 60 Hz), the torque available falls off as frequency and motor speed increase.

At full load or in overload conditions, the torque at enhanced frequency will be below the theoretical “HP=k” curve of torque vs. speed, by the ratio of the “square of the frequencies.” As a worst-case example, a heavily-loaded motor with a 60 Hz base (knee) rating will provide only 25% of base-rated torque when operating at 120 hz., as opposed to the theoretical 50% or the more probable 40% to 45% available at light loading.

Paying out a load is always more demanding than hoisting, in that the descending payload is generating energy which must be absorbed in order to maintain speed control. On a ship with ample AC power capacity a “regenerative” winch drive can pass the retarding energy upstream into the buss system. The alternative is termed “dynamic braking” and requires resistor “banks” with proper cooling to dissipate the incoming energy. For long casts the grids must rate for 100% of the “power.” This system is a better choice for marginally powered ships, or for self-contained winch packages which must operate on ships-of-opportunity. The winch builder and his electrical people can best aid a winch customer by assisting early in the operational definition, to reach the right overall system.

5.2.3 CHART OF TORQUE vs. SPEED, Hydraulic Compared to A.C. Electric Variable Frequency

The following plot is an idealized comparison of a hydraulic "closed-loop" drive with a 2-to-1 "horsepower limiter" type control, to a typical variable-frequency electronically controlled drive with the frequency enhanced beyond the "base" 60 Hz. Both types provide 100% rated torque from "zero" (or "creep") speed up the the "base" or "knee" point. Both types show a fall-off of available torque, as the drive is "pushed" or "enhanced" above that base point.

With the hydraulic system, the provision of addition pump input power will allow a "corner h.p." drive, where there is no fall-off of torque. With an H.P. Limiter hydraulic design, it is important to select a pump with enough displacement to provide the maximum flow rate, at the reduced pressure.

Electrically the light-load maximum motor speed will usually limit how far the frequency is enhanced. Motor and gear noise increases quickly above 1,800 rpm.

5.3 Hydraulic Drives

Several distinct types can be defined.

a) Closed-loop hydrostatic transmissions, with a dedicated circuit for each winch. Piston type pumps and motors are used, with variable displacement at the pump or at both the pump and motor.

b) Open-loop systems, with variable displacement pump and a wider selection of motor type.

c) "Header-&-Branch" systems, with pressure-compensated pump(s) supplying several pieces of equipment via accurate "proportional valves" controlling each branch line.

d) Simple vane pump(s) and motor(s) with a simple throttling valve for direction and speed control.

5.3.1 Hydrostatic Transmissions

The "best" approach from the individual winch viewpoint, this dedicated drive with remote electrical control of pump flow direction and volume can be very accurate. Pumps for this service include a small "charge pump" and perhaps a second small "servo pump. It is tempting to utilize this auxiliary flow and pressure for other functions, such as auto-brake release, but a small separate electrically driven brake control pump will better serve in the long term. If charge pressure is not maintained, the main pump will lose stroke and sag down in volume and the winch will slow for no apparent reason. A separate charge-pressure gage is a useful auxiliary at each control station.

By adding a variable displacement winch motor, the output speed can be raised considerably, although with reduced line pulls. Here again, the "H.P.= k" hyperbola above a base condition is the theoretical result. Electronic digital control can blend operation of three pumps and three motors into a seamless system, such as that required for a "Traction Winch" package.

When designing a closed-loop with a charge pump, it must be remembered that the charge pressure is seen on the return side of the motor, with a corresponding reduction in the motor's pressure differential (& therefore torque output and winch line pull.)

One vital capability of any variable-displacement pump is to provide a FIRMLY DEFINED AND FIXED zero-displacement condition. Hydraulic Power Units (HPU) are often left operating during portions of a deployment when the winch drum is to be stopped. If the pump is even fractionally “on-stroke” the winch will slowly creep, without alerting the crew. (We have participated in the hour’s-long task of manually re-spooling a winch room full of slack wire, hand over hand. This is to be avoided.)

Motors and pumps have defined catalog efficiencies; often separated into “torque efficiency”, (pull) and “volumetric efficiency” (speed). Hydraulic drives will almost always require higher ship’s primary power than will electric drives. The energy required to push the fluid through the pipes and hoses is the main difference, and it is both variable and awkward to estimate. These fluid losses add directly to the electrical and mechanical losses.

With a “unit-mounted” and short-plumbed winch package, piping losses are small and testable. A fixed winch with a remote power supply is at the mercy of the piping crew at the installing shipyard. There is a non-literal but illustrative “joke” which claims that NINE ELBOWS EQUALS ONE PLUG. Formed stainless tubing with long smooth radii and ample diameters (for low fluid flow velocities) is the most power-efficient and the most costly approach. Anti-vibration hoses should be installed at pumps, motors and primary valves.

Fluid velocity is a good measure of a piping system’s design. The relationship is another of the standard formulae:

“V” (ft/sec) = <u>“Q” (ft-cubed per sec)</u>
divided by “A” (flow-area in square feet)

Often-used target values of fluid speed are: 15 ft/sec for supply lines (w. 20 fps being tolerable)
10 ft/sec for return lines
5 ft/sec for drain lines

Both legs of a closed loop can be the high pressure side, (even if the only “payout-direction” load is created by an auxiliary warping head...) and must be piped symmetrically. Reservoirs can be small, since they supply only the charge pump flow, but individual applications require caution in this area. A small tank may require heat-exchanging, where a larger tank may radiate enough heat to reach a balance.

5.3.2 Open-Loop Piston Circuits

The piston pump's output returns to the reservoir rather than returning directly to the pump's suction port, as in a closed loop. The pump suction is from the reservoir. Charge pumps are usually not involved. Return piping can be of lighter scantling, and the filtration is handled in a different manner. Reservoirs should be large; sized for two or three minutes of the maximum flow rate.

5.3.3 "Header & Branch Circuits

Many R/V's have large numbers of hydraulic motors and cylinders doing a variety of tasks. The opportunity to run dead-ended "pressure" "return" & "drain" header piping is appealingly simple.

Each functional drive element connects to all three headers. Control is via various types of valving, usually termed "proportional." Valves are available with
integral flow-limiting, direction control, and accurate flow metering for functional speed control. One pump set, (often with a back-up set for redundancy) uses pressure-compensation to hold maximum pressure at zero flow in the supply header until a branch line is opened.

It is not logical or economic to assume that every winch or cylinder will operate simultaneously. Coring winches are inactive when the Anchor Windlass is in use. The challenge to the operator and architect is to define how many and which functions might operate simultaneously. Until this decision is made, the capacity of the pumping system cannot be determined.

5.3.4 Simple Vane Pump System
Low cost vane pumps can provide fluid to several machines, by using individual manual throttling and reversing valves which have their pressure ports open to the tank port in the off position. The "tank" port piping can be connected to the pressure port of the next valve downstream, and finally back to the tank. Thus an "open-series-circuit" is created. Valve selection must include the ability to accept system pressure at the "tank" port. Only one of the several machines on such a series-circuit can normally be operated at a time, but this is not a hindrance if it fits the operating pattern.

Another circuit option is to provide a branching series of six-way selector valves which can route a single pump's flow to a number of working locations.

When two vane pumps are piggy-backed, with the same or differing flow capacities, the opportunity exists to provide two distinct flow values. One pump is fitted with an "unloading" valve which "dumps" to tank at a selected signal pressure and continues at a low-loss threshold pressure in the order of 50 psi. The second pump has a normal higher pressure relief valve.

As long as the overboard load is below the dump pressure, the combined flow provides the maximum speed. As the applied load creates the unload signal pressure the flow and machine speed drop, and at the maximum pressure, stalling takes place. High speed and high pull can thus be provided; not simultaneously, but with lower input power.

A number of "two-speed" hydraulic motors are available, with equal or differing displacements. As an alternative two winch motors can be valved & piped for either "Series-Parallel" or "One-Motor-Two-Motor" operation to provide a "high-low" two-range winch response.

<u>Unexpected speed changes can surprise and endanger a crew when sudden load changes occur.</u>

These combinations can provide a four-range winch drive. When controlled by a manual reversing valve with good throttling characteristics, a low cost and quite flexible system can be provided. The "seaman" on the valve is totally responsible for the accuracy of winch motion in the critical load locations.

6.0 WINCH CONTROLS

Variations in control system range from an operator standing next to a winch with his hand on a manual reversing valve, to that same operator in a cab or the wheelhouse using a PC with a touch-screen or a mouse which manipulates the entire ship-suite of overboard handling equipment.

Maintenance considerations suggest that control systems be design as simple as possible, however, many tasks demand complex controls. The massive "machine" which rides the R/V "Atlantis" A-Frame to safely deploy and recover "Alvin" does not lend itself to simplicity. A-Frames and booms require their own controls that add to the required operator skill level and to his "busyness."

Most control systems now are blends of electronic, electro-mechanical, electro-hydraulic and hydro-mechanical elements, with air controls also making their contribution. The common features of a proper control sub-system, whether primarily electric, hydraulic, or air, are:

6.1 Precision

The three load locations demanding the greatest precision are at the water-plane, at the deck, and at or near the sea-bed, such as when coring. Whether the man-machine contact device is a potentiometer knob, a joystick or a PC mouse, the load must be locatable at extremely slow creep speeds, with very fine position increments. Ideally, the control should be able to place the payload onto an egg, without cracking the shell.

Most winch systems include a spring-set automatic "parking" brake. An important element of precise control is the timing of this brake's set and release action. Present technology provides a variety of signals which will allow a drive's torque to build up gently to pull slack, overcome static friction and the payload's weight and only then release the brake. Similar delicacy of timing is needed in setting the brake.

Electric winch drives have their advocates, utilizing the arsenal of electronic and computer-based controls. A seemingly equal number of designers prefer to apply electric and electronic controls to the strokers of variable displacement hydraulic pumps. The choice can hinge upon the rest of the equipment on the ship. Traction winch systems have all been hydraulic, and regularly provide precise action.

6.2 Smoothness

The old multi-step DC control systems provided coarse steps, using five levels of grid resistance. Commutator-type lever switches provided up to 16 points.

Motor Generator Set drives could provide variable speed control. Modern SCR-DC and AC-VF drives are stepless and completely smooth between creep and the maximum design speed. The higher-level hydraulic control packages have the similar ability, with perhaps somewhat more initial tuning and periodic maintenance required. (The inside of an electric wire is a "very clean" environment, whereas achieving and maintaining a clean hydraulic system has often been a challenge.)

6.3 Functional Clarity

A variety of guides and standards are available which purport to define the semantics of control definition and labeling. In reality the variety of functions appears to stay ahead of the "standards" and requires thought and excellent communications between the manufacturer and the operator.

One control which can be cited as almost universal is the bright red "Emergency Stop" or "E-Stop." Almost every panel provides one oversized button which is pressed (or bashed) to abort an operation in the minimum time. Even here, different systems will require properly sequenced detail actions within a control package, to achieve the quickest shutdown without shocking the payload and wire, or leaving the control box in a non-resettable condition.

6.4 Multiple Stations

If for no other reason than permitting a technician to operate a winch during maintenance, it is desirable to fit a "local" or "at-winch" set of controls. Many working deployments are managed from a weather-protected cab which is located to provide the operator a clear view of both the winch being used and the overboard point. Aboard smaller vessels, this sheltered station may be in the wheelhouse itself.

Semantically, a station which is away from the winch itself is best referred to as a "remote" location. "Local" implies that the man is at the machine and able to react immediately to any malfunctions which might occur.

When winch machinery is inside or below deck, the operator visibility of the overboard point takes priority. Video cameras and monitors should clearly show the state of the

wire spooling onto the winch drum, and instruments monitoring the drive-condition should be available to the operator.

Station selection has two main formats. The least desirable provides a “Give-It-To-Me” button at every station, accompanied by lamps which define the controlling point. This arrangement sets up the possibility for two would-be operators conflicting with each other.

Except in special situations the better approach is to provide a single station-selector -- again with lamps to indicate the active station. This may require additional walking or communication, but it gives control to one person. The “E-Stops” at all stations must remain active.

7.0 FAIRLEADING AND SPOOLING

A major feature of a winch’s success is how smoothly the “line” is spooled onto the drum. An even and level “lay” minimizes line wear and adds to the accuracy of the instrumentation by providing a known number of wraps per layer. For any tension member which retains an essentially round cross section as it wraps, the “Lebus” proprietary drum shell can make the major difference between a smooth spooling to be admired and a “hill-and-valley” spooling; possibly tolerable, but likely demanding frequent fairlead adjustments, to avoid wear-inducing piling, and in the worst case, “back-&-overs” which can lock the line into place. If a machine must handle a variety of lines, the interchangeable drum, each with a grooved shell is often the right answer.

If the winch location provides a 20-to-1 or greater ratio between the drum’s face width and the distance from the drum center-line exit-tangent to the nearest (& centrally aligned) fixed guide point, a mechanical fairleader theoretically should not be needed at all. However, in almost every case reality trumps theory, and a fairleader is indispensable.

7.1 Fairlead Drives

Two opposing fairleader design approaches have attracted their own adherents. The simpler design mechanically drives a "diamond screw" from the winch drum hub. The "shuttle" (cam follower, butterfly, etc.), which rides in the bi-directional long-pitch screw groove, pushes the traversing fairlead head back and forth. A simple hand-wheel and jaw clutch allows for initial head positioning and any adjustment that may be needed.

The other style is based around a single-thread lead-screw which is separately powered and reversed by various forms of servo-drive and switching, and which uses various forms of cable-position sensors to tell the drive where it should place the fairlead head.

As is normal, each type has its own plusses and minuses. For this description, we elect to use the term "Putter" for the diamond screw type, and "Chaser" for the servo-drive type.

7.1.1 Diamond Screw "Putter" Type

To succeed, this drive must be designed with a exact knowledge of the line's loaded diameter -- "Lebus" normally tests samples of actual wires or cables to provide this data. The relationship between the face width of the winch drum and the "turn-around" stroke of the diamond screw must be correctly laid out. The exact practice is hard-won knowledge and is often held as proprietary.

In effect, the fairlead head is going to be "Put" the wire in perhaps 3000 exact locations (100 wraps by 30 layers). This requires a high degree of numerical precision in the drive chain ratio, and benefits from the application of a custom pitch-matched Lebus shell.

Bare-drum spooling (i.e., without a grooved shell,) can work well, however the inter-wrap air-gap is determined by the crew person defining the barrel layer with a stick and a hammer, and the designer must guess at a wider air gap when determining the number of wraps per layer. The "Lebus" shell allows a closer air-gap and determines the exact number of wraps on the barrel layer. That same value will apply to each layer, if the intent is met.

For most research winch applications, one or two different line types will cover years of its work. A pair of interchangeable drums and a quick-change set of drive sprockets will maintain the needed matching. Working experience with the "Lebused Putter" fairlead has been good to excellent, and maintenance is reduced primarily to normal bearing and guide lubrication.

7.1.2 Servo "Chaser" Type

This form of level-winding is intended to accommodate any size of line onto the same drum, since its sensors are designed to tell the servo drive where the line is, and the separate screw drive then places the traversing head at that location. In practice, they are frequently also matched up with a "Lebus" shell for a known line diameter.

Impressive design ingenuity has been observed on many variations of the servo fairleader. Excellent results have often been achieved. Since this author's firm does not use the "Chaser" fairlead, the only reasonable comment is to note the increased number of parts involved and the apparent complexity of the overall approach.

7.2 Rollers and Sheaves

Each research wire, cable and umbilical has its own bending-radius requirement. If a long & centrally-aligned line lead is available and the cable's direction change at each end of the drum can be less than 3 to 5 degrees, a simple set of vertical and horizontal guide rollers may be adequate. With large roller diameters and the small wrap angle (a "kiss-bend"), the low-cost roller head has sometimes been sufficient. With a rollers-only fairlead head, all cable parameter instrumentation must be taken from an off-winch sheave.

The more frequent design solution mounts one or more sheaves directly on the traversing head. One arrangement provides two guide sheaves on either side of a central "metering sheave." The cable payout and tension sensors are incorporated with the central sheave. Looking at a side elevation, the tension sensor axle-pin sees an always-vertical direction vector as the wire exit angle varies up and down.

A pair of adjustable-angle rollers beyond the downstream guide sheave insures that the line enters the sheave suite in a straight line. These rollers can be adjusted to cover a range of up or down lead directions to the overboard point. If deck space requires the winch to be close to the first guide sheave, the entire fairlead assembly can be designed with a built-in up or down angle.

7.3 "Open-Lay" A Speculation

Many cables and umbilicals are not firmly round in cross-section. The attempt to spool these in the normal manner can be frustrating and can cause a deployment to fail. Such failures can result from the cable becoming out of step with the fairlead and creating an "over-and-back," or if the cable lay becomes "loose" the exiting lead can "pull-down" through many layers and lock up.

One simple solution to both problems might be borrowed from the work boat fleet which is increasingly using "soft" "Spectra" and "Plasma" type braided lines in ship-assist and ship-escort assignments. Working with a bare (non-Lebus) drum, the traverse rate of a level-winder's head can be sped up by the substitution of a smaller driven sprocket at the diamond screw. By tripling the rate above the normal, an "open-lay" is achieved whereby the "air-gap" is twice the nominal width of the line. This is sometimes called "universal spooling."

This is readily illustrated by the reader spreading his fingers and laying one hand atop the other. A large number of cross-overs take place and because of this "under-bridging", pull-down is prevented.

With many cross-overs, the angular change at each becomes small. The down-side to this layout is an increase in contact pressure at each cross-over. For some cables this may not be a problem. For others it may rule out the concept. Recall that for a regular "wire-lay" there is always one crossover per wrap, (or two with a Lebus shell). They are at a flatter relative angle with more length spreading the contact force.

It will be intriguing to see if such an "open" or "universal" lay can alleviate current and future spooling problems for particular forms of tension member. Where it will function, the need for grooving and fairlead precision go away, and cost would be reduced.

8.0 INSTRUMENTATION

The research winch provides three of the cable data parameters which computer collector systems require -- Line Tension, Amount Of Line Payed Out, and Line Speed. In some cases an operator may wish to know the approximate angle at which the wire enters the water. Winch drive power demands (speeds, amps, pressures, etc.) and condition indicators such as various system temperatures can also be added.

While the three primary cable values can be taken from the winch drive package, r.p.m., amps or psi are only rude indicators of drum speed and torque, and a layer-compensation chart must be used. A much more reliable approach is to utilize a direct-reading cable sheave as the sensor "drive." The "measuring sheave" can ride the traversing fairlead head, or it can be a separate downstream sheave -- preferably with a known wrap angle.

8.1 Signal Generation

8.1.1 Cable Tension

Strain-gauged sheave axle-pins are a specialist product requiring close cooperation between the winch builder and the pin supplier. The signals which sense the bending of the pin under varying cable tensions are very low-level. To improve reliability many pins now provide integral amplification electronics in an enclosure at the pin's end.

Knowing a tension sheave's cable wrap angle is a major advantage. While it is possible to use an overboard sheave for this purpose, the load vector seen by the pin is constantly changing. "Rube Goldberg" wire followers and potentiometers can be used to compensate, but this choice is far down the list.

8.1.2 Cable Length Payed Out

The distance that the wire travels in one sheave revolution is readily translatable into feet or meters paid out. The addition of time results in the line speed in feet or meters per minute. One basic method involves a pair of marinized proximity sensors. (A pair is necessary in order to sense the wire's direction or travel.) These can react to machined spots on a sheave's ribs, but much better resolution is provided by providing a slotted disc with more "interruption" points.

The best results are provided by employing fully housed "Optical Encoders" with step-up gearing from the sheave's hub. Very high accuracy's can result.

8.1.3 Cable Speed

Once the length-reading hardware is in place, the count pulses allow production of a cable velocity signal to be an electronic "no-brainer."

8.1.4 Cable Exit Angle

While criticizing vector-correction angle sensors as part of an overboard sheave's tension-measuring action, such hardware may be needed for the different purpose of defining the angle at which the wire is either leaving the ship or entering the water. The fully deployed overboard frame is likely the preferred reference point. A wide variety of sensing tools can be envisioned, and each could work well in flat water or perhaps in long swells. Given the Sea States in which the ships and crews are being asked to do science, this angular information would seem of secondary reliability. On the other hand it is a risk to bet against the potential power of gyros, lasers and computers.

8.2 Cable Slippage

The length and velocity information is contaminated by line slippage relative to the measuring sheave throat. One observed remedy has been to use a "rubber" or similar material to improve the grip on the cable. This generates a "wearing surface" involving maintenance, and with wear, small changes in the working pitch diameter of the cable can result. This change biases the data quality.

Most research winches do utilize “Lebus” shells matched to known cable diameters. One builder provides a three-piece measuring sheave with a hard “T-1” (or similar) steel center plate and an demountable outer sheave side plate. The o.d. of the replaceable center plate places the pitch line of the cable at a known and convenient diameter. Typical pitch circumferences are one-meter, 1-1/2 meter, two meter, etc. Such initial “natural” selections can ease the system designer’s thinking process.

Since the cable’s diameter is known, the center plate’s thickness can be ground to insure that the machined throat sides provide a “light kiss grip” on the cable. Watching the length meter return to “zero” after a multi-thousand meter cast is normal and gratifying, since slippage control has been achieved.

8.3 Displays

PC screens can be utilized to directly provide the winch operator with the information about what is going on with his overboard cable. Additional information such as that sent from payload “pingers” can expand his awareness. The PC approach has so far been awkward for on-deck local control stations.

Presently the preference remains for dedicated digital and analog displays of the three primary line values. A variety of digit types and sizes are available, and waterproofing is available. Particularly with the tension value, the addition of a naturally damped analog dial can provide the operator with a more useful sense of the overboard load than attempting to focus on a “jumping” string of digits.

These electronic displays provide connection points for the standard variety of electronic cabling to move the data onward to a ship’s logging computer. Integral key-pads can facilitate the setting of various alarms such as high or low tension, near-surface load location, etc. These settings can talk back to the winch drive and provide automatic slow-down or other responses.

9.0 FITTING THE WINCH TO THE SHIP, AND PORTABILITY

9.1 The Basic Suite

Most mid and large R/V's are outfitted with a basic suite of large, medium and small winches to bracket the predicted requirements of the scientists. The smaller boats often omit the mid-sized machine. This equipment-matching is major part of cruise planning.

Ideally, with a new ship program, the winch people are brought in at the layout stage. This permits engineered and correct relationships between the winch line leads, the overboarding frames or cranes, and often the sheaves that must take the cables "from here to there." When a winch is within a working space or even below deck, this early involvement upgrades from a convenience to being critical. The ability to customize large winches to their available spaces has been an important selection factor in many instances.

A ship's basic winch suite can be welded down to suitably muscular deck insert plates. Alternatively structural subbases can be shipyard-furnished to accept hold-down bolts at the winch sills. The "turntable ring" base is the logical extension, where a winch can be aimed directly toward more than one overboard point.

9.2 Ship-Of-Opportunity Winches

In an environment where vessels are increasingly marketing their available time to a "customer-base" of marine scientists, more situations are arising where a science team will have its own packages, cable, and perhaps the full winch and drive package. Many ships outfit their working decks with a 24" grid of tapped and plugged bolt-down sockets as an accommodation means. The precision of this grid spacing has been known to vary, and the machine may require orientation other than orthogonal to the ship. The simple addition of a "subplate" by a shipyard or the institution's own shop can resolve both situations.

The unit-mounted or modularized winch package which provides its own power umbilical with a standardized plug works with the receptacle outlets that are a part of most modern R/V's. Electro-hydraulic winch packages lend themselves to this approach, whether totally self contained, or with a separate

HPU incorporating all the connecting hoses and the prime power "cord." Where an electric starter panel is required, it can be separate or fully waterproofed (NEMA-4) and integrated. The less costly dripproof panel requires a properly sheltered location.

Ship Of Opportunity winches should ideally provide integral data signal sources and display components, with standard porting to allow passing information to the ship's PC.

10.0 CONSTRUCTION

Applications exist for absolutely minimal winches intended to last for only one cruise or project. The opposite requirement is for a machine intended to perform to design specs for 30 or 40 years, with the probability of seeing service on the "next" ship, after the initial vessel is retired. Winches can be "throw-away" quickies, or full capital investments. Individual builders normally find their own "comfort zone" along this spectrum, so that a buyer will have a reasonable idea of where to go to match his needs.

The author's firm chooses to emphasize the "Hell-For-Stout" approach, although this may involve serious engineering to "build in lightness." This imposes the duty to provide machinery which can be maintained easily and which is worth a re-powering when the initial drive have lived its life. Record keeping must be complete, so that wearing parts remain available for the decades involved -- not waiting on the shelf, but quickly produced from the as-built detail drawings.

Certain basic practices lend themselves to long life. Steel fabrications which are stress relieved before machining, hold their shapes as the machining cuts are made. Integrally designed gear housings allow full design control and compactness as compared with bolted down commercial reducers. A major decision is whether to line-bore all the bearing fits or to use separate pillow blocks with shims or "chock-fast". Line boring insures repeatable reassembly during the machine's life and eventual refreshing. Anti-friction bearings are preferred over bronze bushings, at all primary shafts, including the outboard winch A-frame. Where drums are interchangeable, each drum should be fitted with its own outboard steel anti-friction pillow block, machined to defined dimensions for a positive fit.

10-39

Lubrication provisions are critical. Research winches often operate slowly for extended periods -- an auxiliary lube oil pump will maintain gear and bearing lubrication, where simple oil-bath lubrication might not. Linkage grease fittings which are inaccessible will not be serviced! The right approach is to provide heavy duty brake-type hoses which bring lube points out to ganged blocks which are handy for the crew person with the grease gun. Gear housings should have marine ball-type drain valves to allow regular condensate drain-off. Jaw-type gear-range or free-spool clutches require less maintenance than the friction type, and as seldom used elements, they are best located within the gear housing. Heavy transparent "windows" into the gear housing should be provided to aid clutch engagement and confirm lubrication. External mechanisms such as drum brakes should be heavily constructed to withstand the full input of the largest crew person, utilizing the longest "cheater bar." (Which should never be needed, given proper maintenance.) All fittings should be stainless or otherwise non-corrosive, and winch surfaces should be sandblasted, inorganic zinc coated, and top-coated with one of the many epoxy or equal coatings.

11.0 "R.F.Q." INPUT INFORMATION

No standardized form or questionnaire can properly lead to the right Research Winch being produced. Engineering communication at all stages is required to assure that result. There are basic parameters that serve well to open discussions. These include:

a) Diameter and characteristics of the primary wire or cable, including the manufacturer's bending radius recommendation.
b) Length of the longest cable envisioned, over the winch service life.
c) Planned payload's in-water weight, and something about drag characteristics, both hoisting and lowering.
d) Hoisting speed required
d) Type of power preferred -- electric or hydraulic.
e) Type and amount of ship's power, for permanent installations.
f) Type of ship's hydraulic circuitry

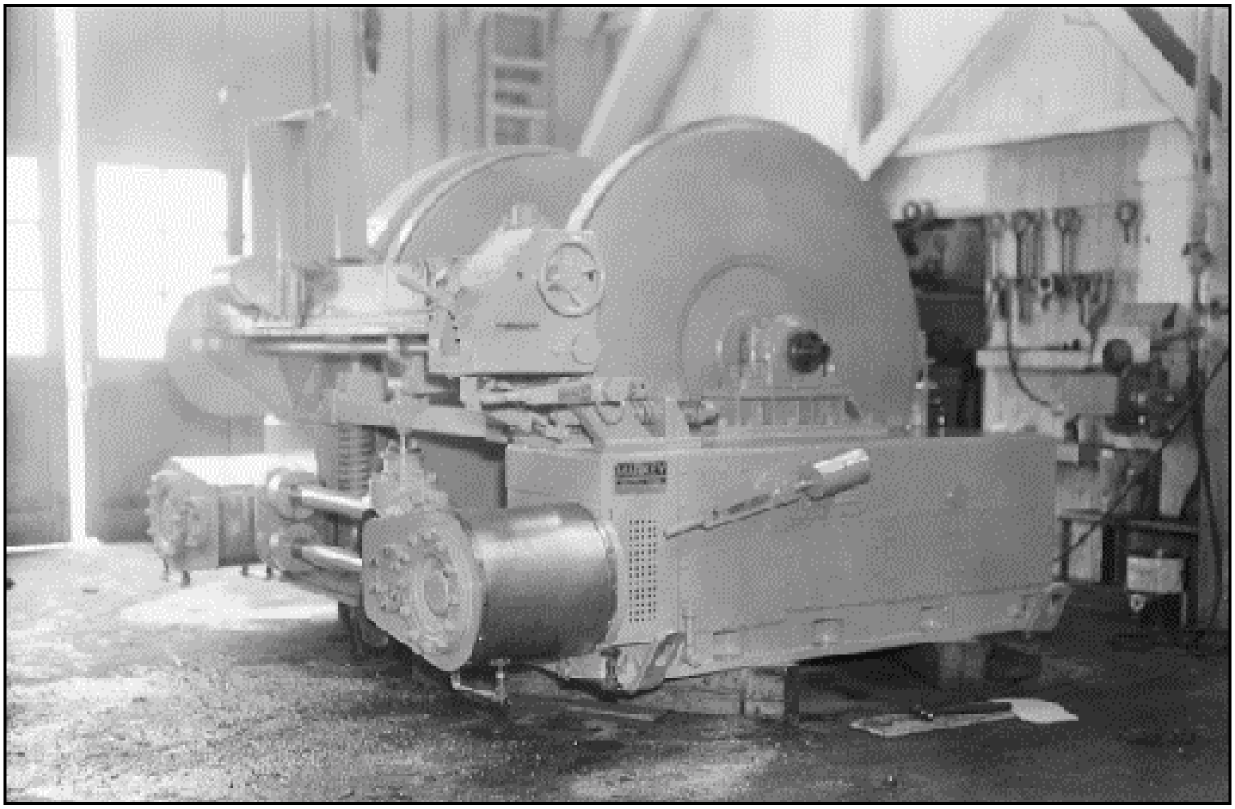
g) Vessel details, such as intended winch location, orientation, location of the overboard points, etc.
h) Number of control locations required
i) Degree of instrumentation required
j) Weight-critical circumstances which might require aluminum or part-aluminum construction.

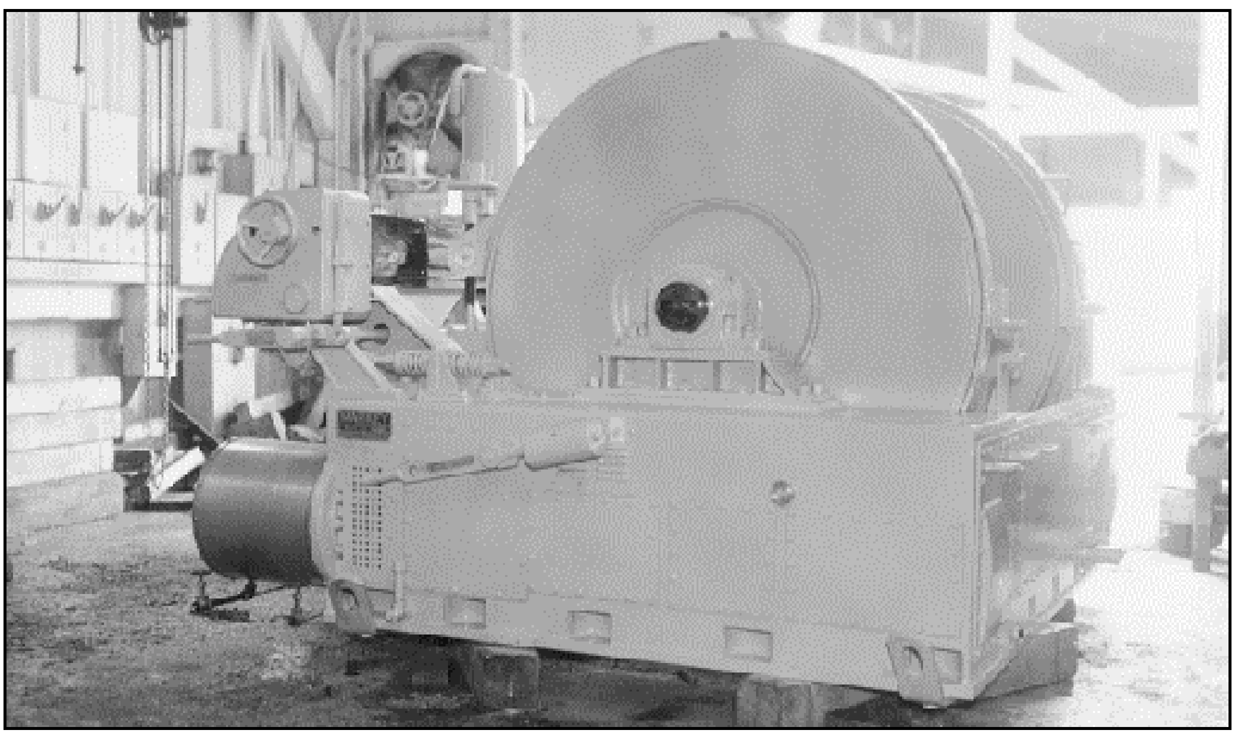
It is not necessarily easy to gather up this much information about a new winch requirement. The buyer will often be caught in a web of conflicting preferences generated by his own people and by potential users of the ship. The necessary compromises are a normal feature of the preliminary definition "give-and-take."

12.0 CONCLUSION

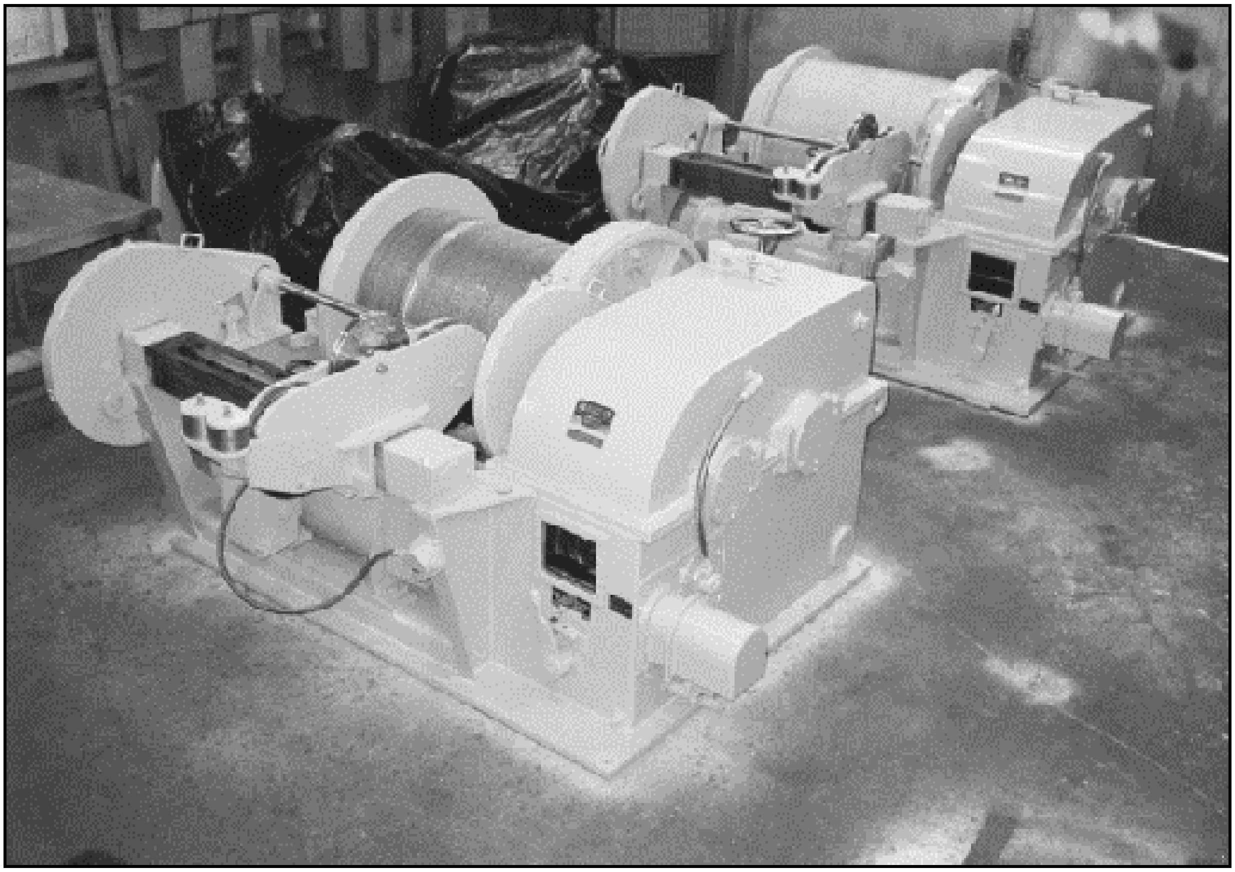
Tethered and free-swimming vehicles are adding to the kit of tools available to the marine scientist, as are buoy arrays of all types, manned submersibles, and a list of others which one can only be certain will grow and improve. The author is of the opinion that there will remain an active requirement to handle all manner of lines, wires, cables, umbilicals, etc. on powered drums. It appears equally certain that the basic tool called the "Winch" will also continue to evolve in all directions -- absorbing ever increasing degrees of electronics, computer power, and material science.

Our final opinion is that just as the Oceanographic Community remains a very diverse group of individuals, (as does the larger "waterfront" tribe), the Research Winch branch of the sea-going deck machinery tree will remain primarily custom in nature. Experience and continuity will continue to be important attributes for the owner, the architect, and the machinery designer and manufacturer.

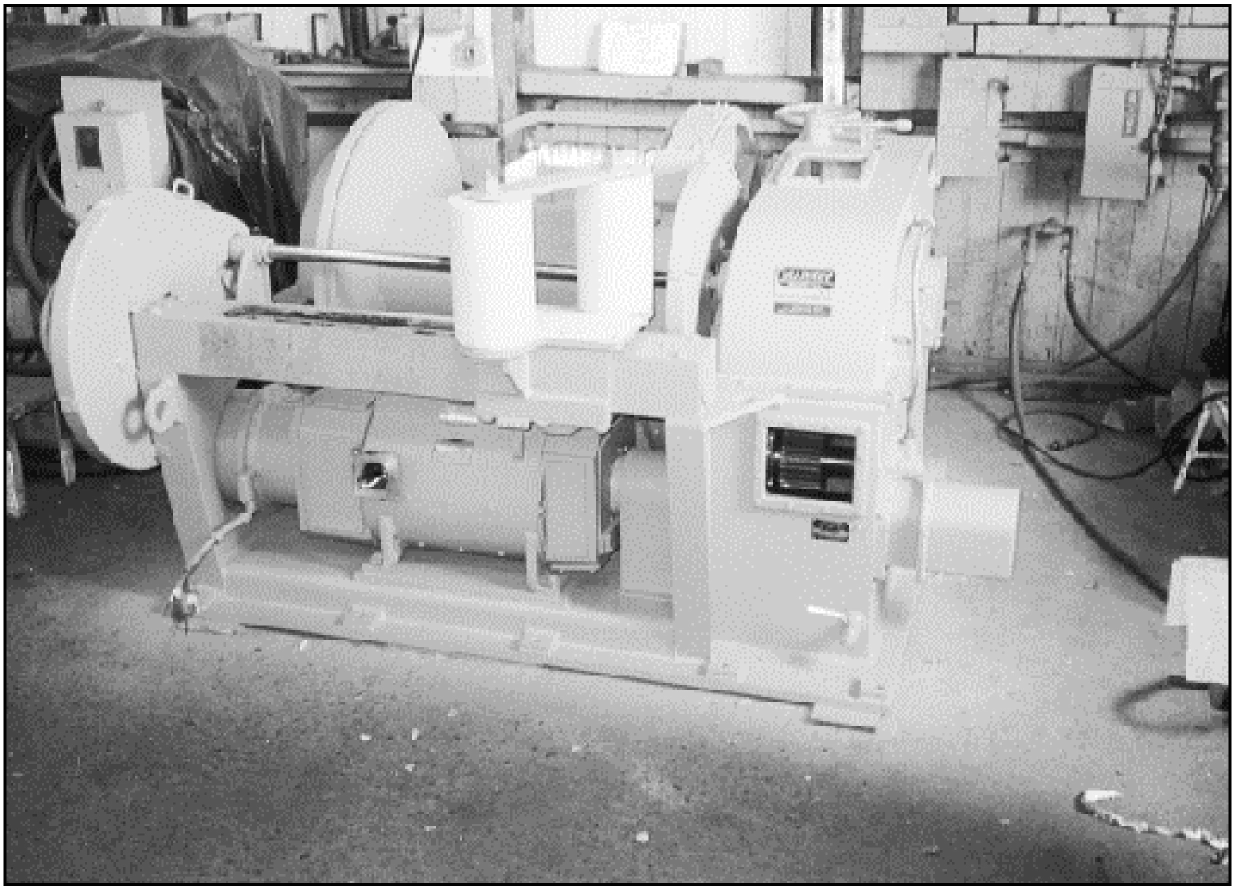




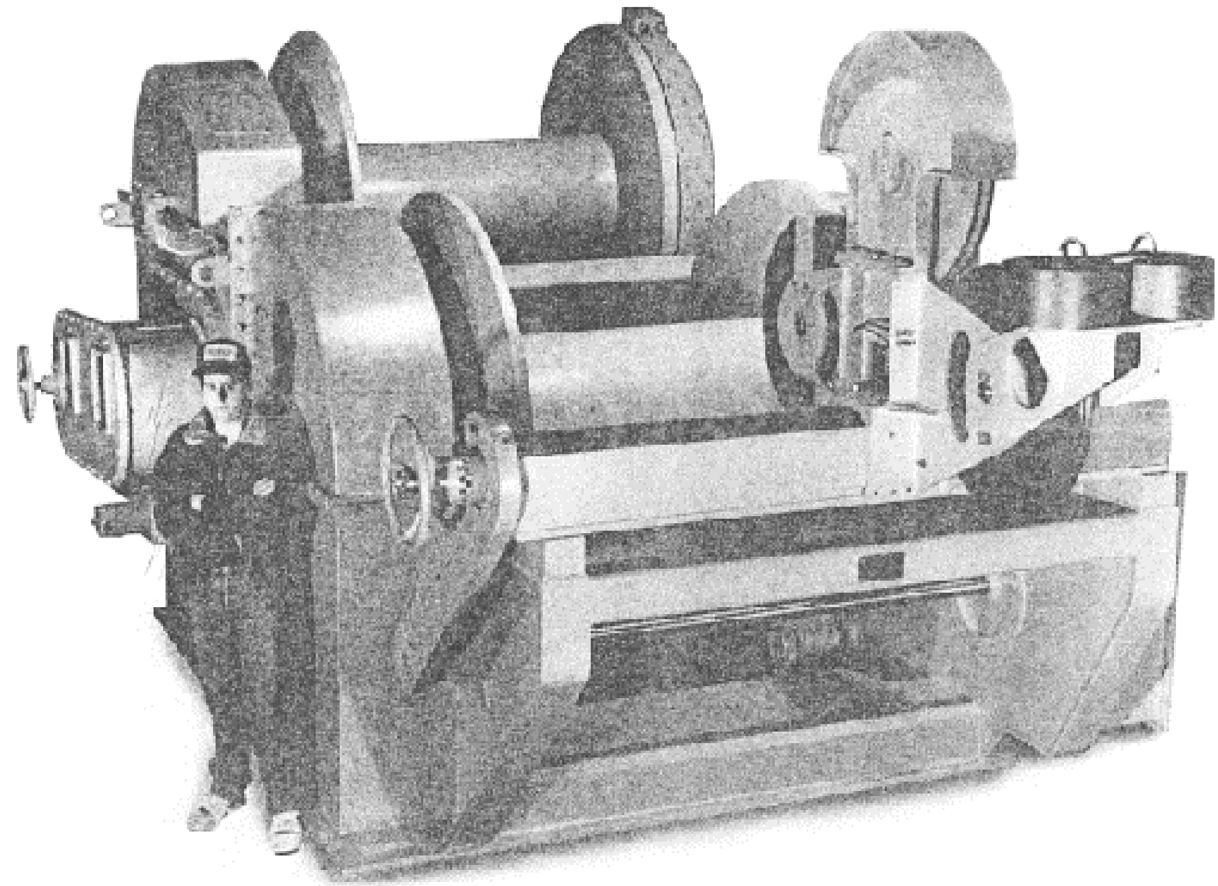
The last U.S.-built steam powered Research Trawl Winch, this DSSH-10 went aboard the new "AtlantisII" to utilize the steam from her Skinner Uniflow engines. The entire integrated engine was designed from scratch. When the ship was repowered, the con-rods were removed and a 200 h.p. electric-plus-gear drive was coupled to the crank disks.



Two DESH-5 winches with 75 h.p. SCR-DC drives are shown spooled with 3 x 19 wire and 0.322 cable, ready to go. The three-sheave fairlead heads show clearly. The extended enclosures coaxle with the input shaft are tachometer generators with zero-lash couplings, which are critical to the DC drive performance. “Atlantis” and “Ron R. Brown” received these 15,000 lb. machines.

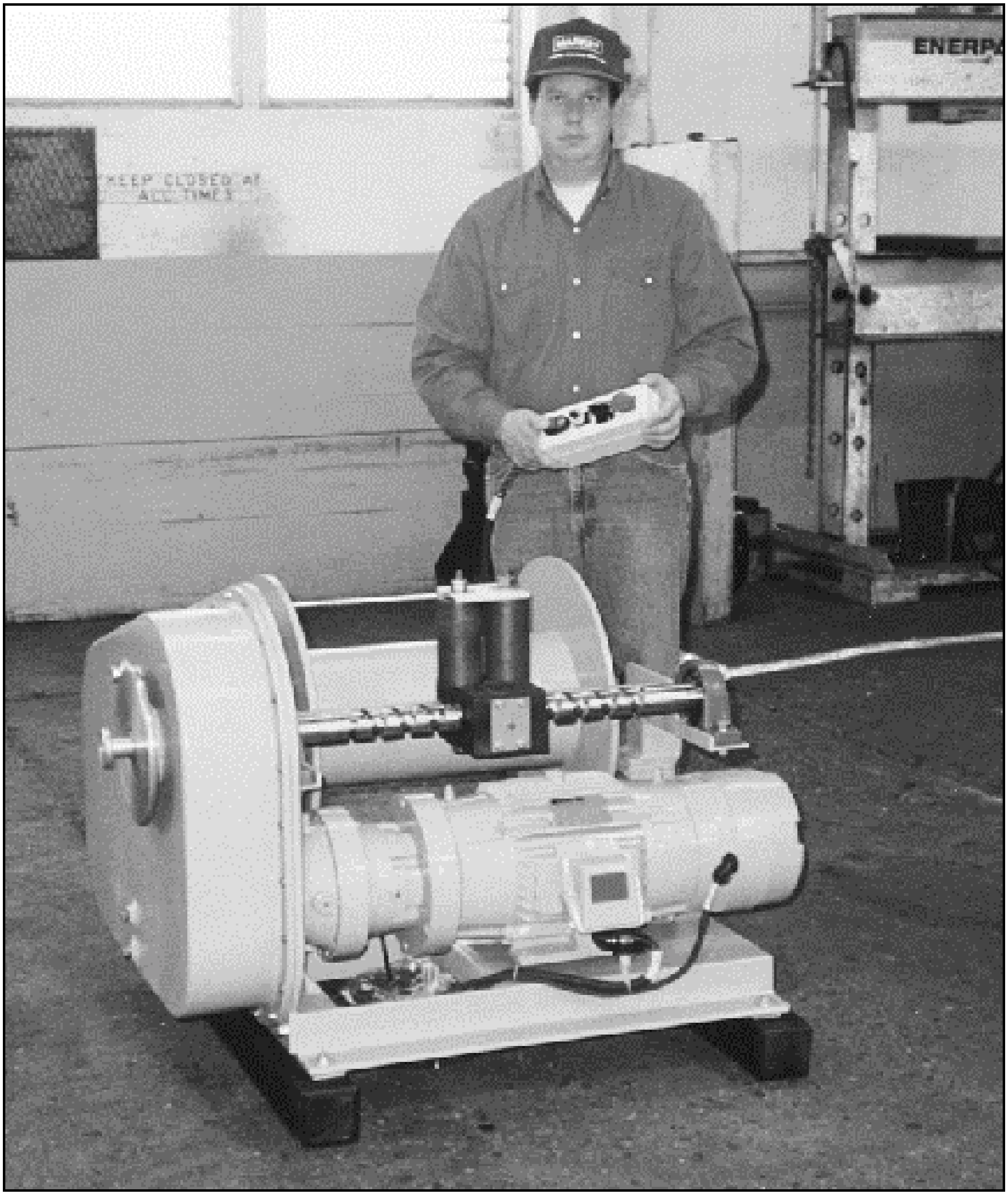


A 75 h.p. SCR-DC DESH-5 destined for “Knorr” illustrates the simpler fairlead, with nylatron guide rollers. Instrument signals were taken from off-winch sheaves. The armored feed hose from the Tuthill lube pump is visible above the tach-generator, as are the two large gear housing inspection windows.

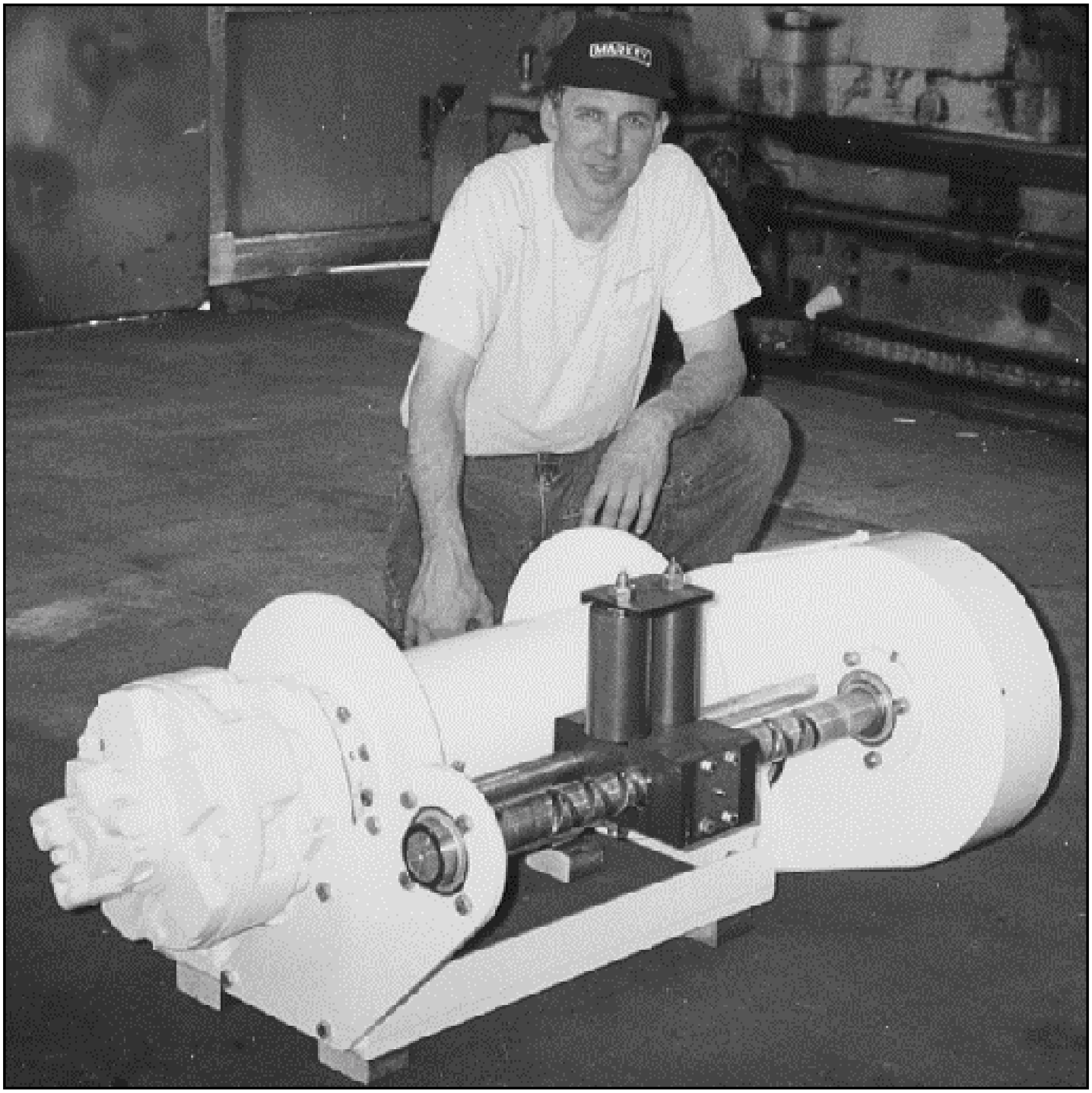


This 150 h.p. Type DESH-9-11 AC-Hydraulic two-drum “waterfall” winch is installed below aboard “Thomas A. Thompson”. (AGOR-24) The larger low drum handled 0.680 EM cable while the smaller high drum attempted to spool 9/16” 3 x 19 wire. This machine weighed 98,000 lb., with both wires

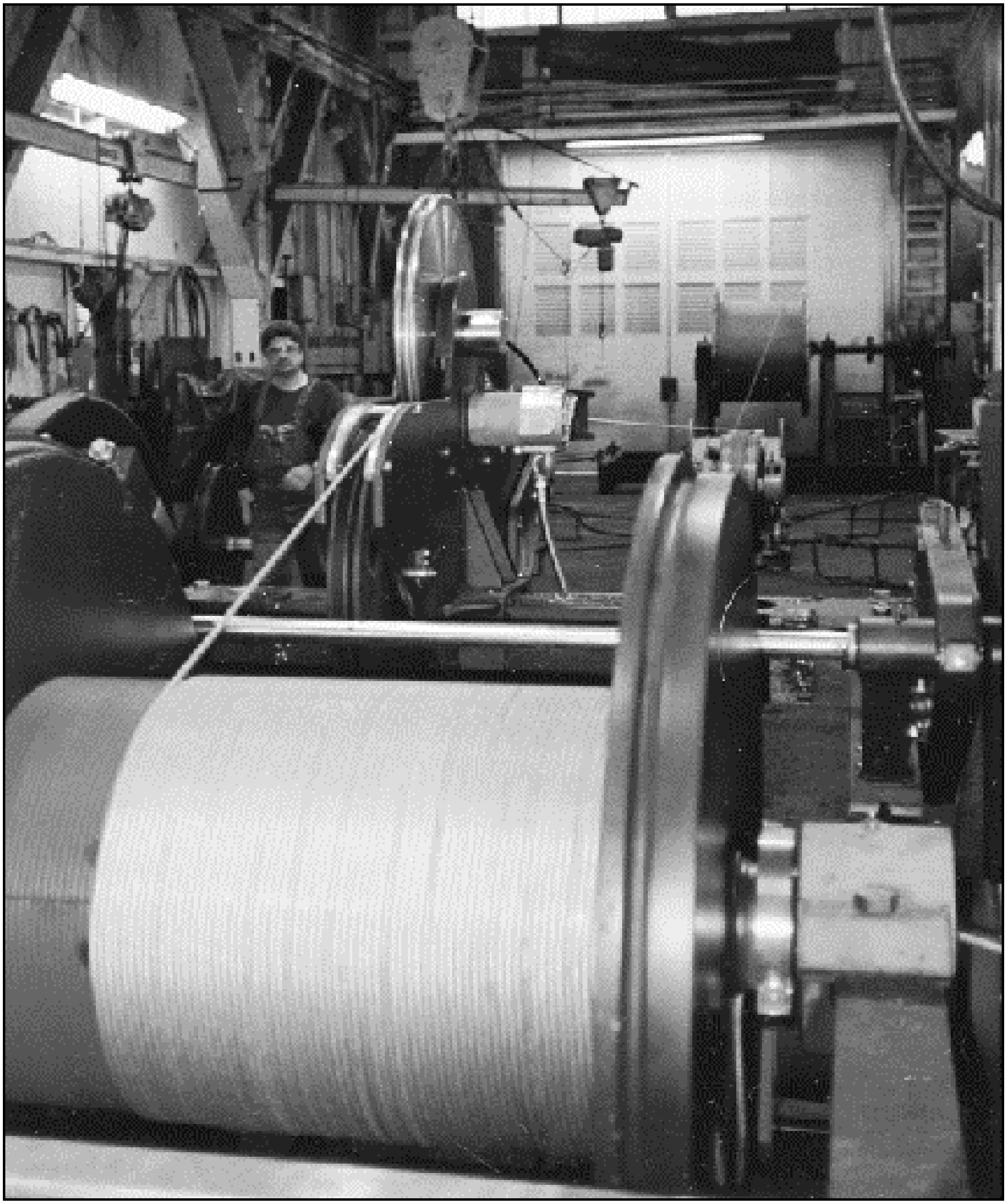
The later three AGORs (Revelle, Atlantis and Brown) were fitted with dual-storage-winch plus traction winders. Both storage drum barrels were of 48” diameter, and one of the counterbalanced 90” fairlead sheaves was 48” to handle 0.681” fiber optic cable in the future. The 3 x 19 wire drum had a 30” dia. fairlead sheave. These three components and the H.P.U. are set up for test in the winch-room orientation.



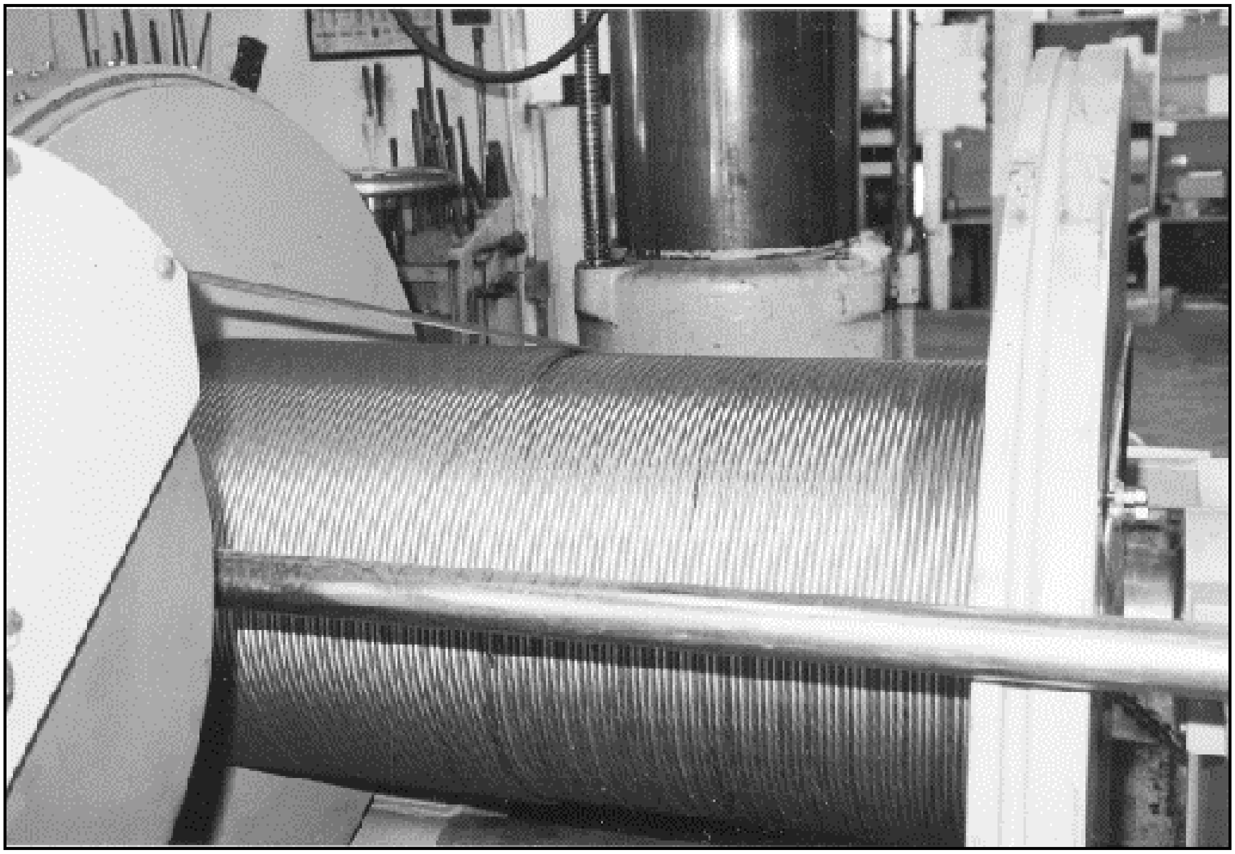
From the large to the small. This 5 h.p. COM-5 (Compact CTD) winch is of aluminum construction with a weight of 850 lb., plus the AC-Variable Frequency control panel. Designed from up to 2000 meters of 0.322” E.M. these are being “batch-produced” as a semi stock line, developed with “Sea Bird.” The multi-width drive chain and the fairlead drive chains run in oil within the cast aluminum housing.



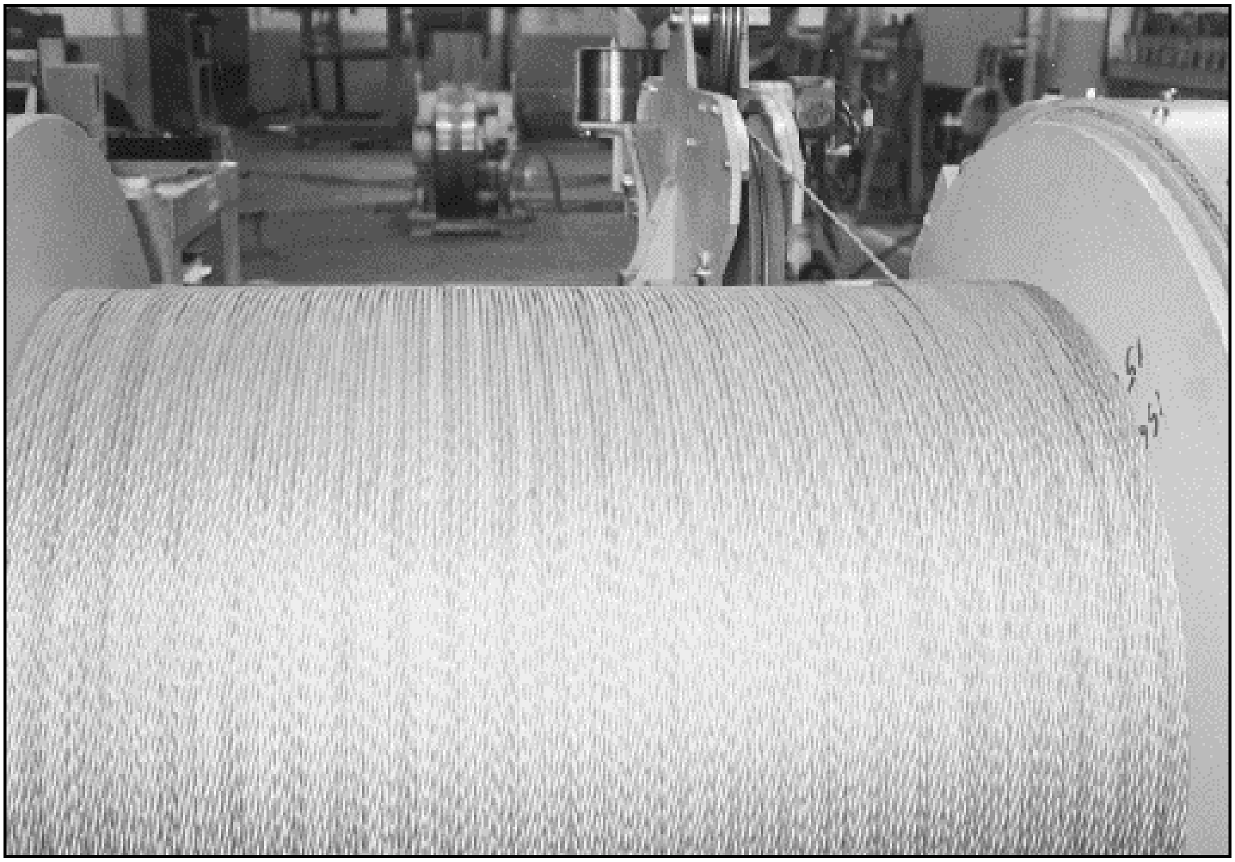
From small to smaller. This 350 lb. coaxial-drive hydraulic COM-4 was custom designed to mount on the cabin top of a 40 ft inshore yacht-type “R/V”. Drum Capacity was for 400 meters of 0.322” or 600 meters of 0.257”.



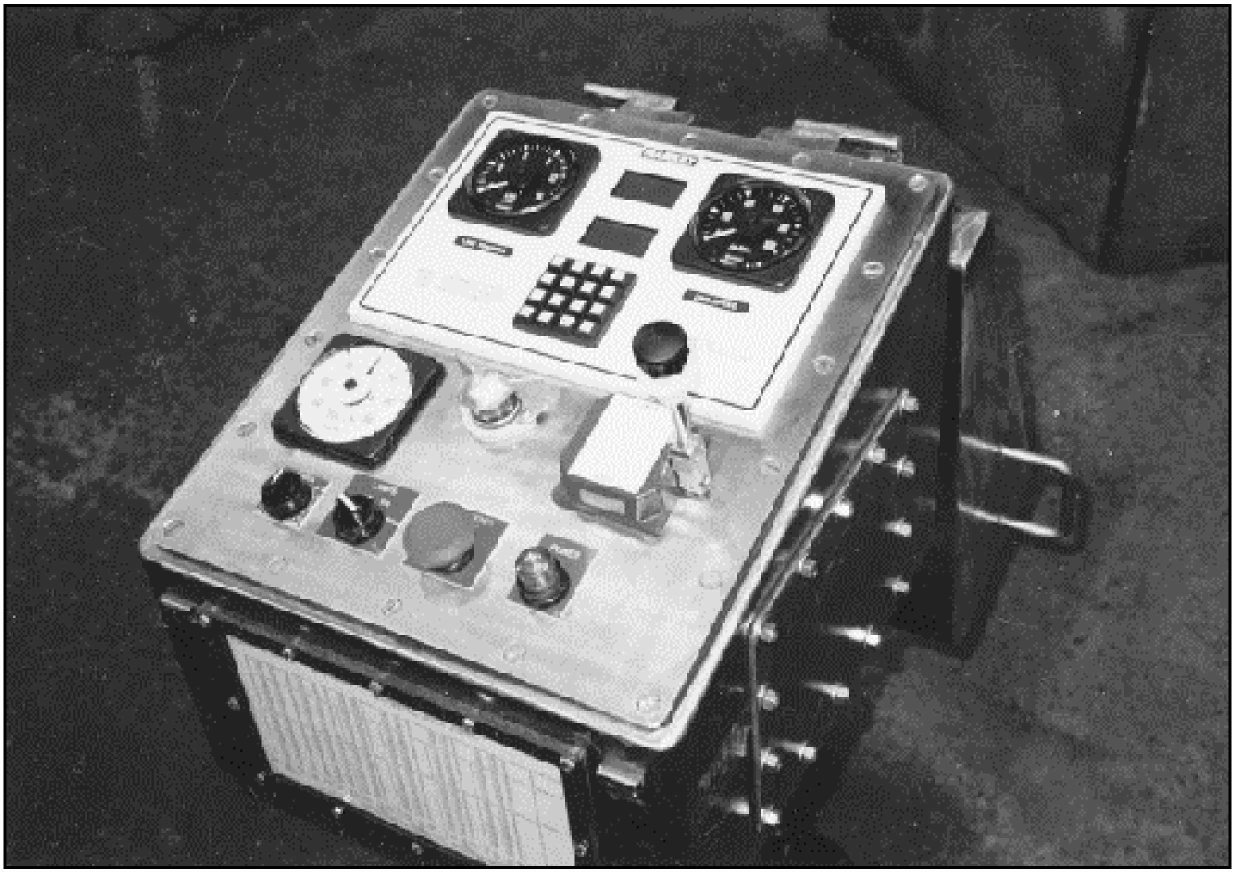
0.322” cable being factory spooled onto the first layer of a DESH-5. The shipping reel is feeding through a factory-owned “Traction retarder” with air-cooled disk braking. The winch’s instrumentation is used to maintain the desired tension.



Spooling is well along the 2nd of perhaps 30 layers. Note the drum flange thickness, and its sling-groove to facilitate drum interchange.



Even 3 x 19 torque-balanced wire rope can be cleanly spooled --at least under factory conditions. The traction retarder is visible beyond the winch’s three-sheave fairleader.



A typical electric winch operating panel for console installation. The in-house instrumentation display provides analog meters for speed and tension, a digital display for amount of wire out, and an additional digital unit for keypad setting of alarms, etc. The joystick controller and the system ammeter share the middle row with the bronze marine light. The large red “E-Stop” is on the bottom row.

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Michael J. Markey, p.e. 2000

Chapter 11

DOUBLE DRUM TRACTION WINCH SYSTEMS FOR OCEANOGRAPHIC RESEARCH

James Stasny

DOUBLE DRUM TRACTION WINCH SYSTEMS FOR OCEANOGRAPHIC RESEARCH

The traction winch, as defined in marine deep water applications, is the primary component of a system designed to provide a significant tractive or load-bearing effort to subsea cable or umbilical. Conventional traction winch systems utilize two sheaves with multiple cable grooves to apply this tractive effort via elliptically reeving cable around the two sheaves. Although traction winches have been around for many years, earlier systems were designed for use with wire ropes in applications where high line pulls tended to knife or bury the outer layer of wire rope into previous layers of rope on the drum. The concept of using a traction winch to extend the life of the cable other than by preventing knifing was usually not considered and in fact some earlier traction winch systems earned the reputation of "cable eaters".

Advancements in instrument packages and vehicles to perform more complex, intervention tasks at greater depths have placed greater demands on the cables and umbilicals linking them to the surface. Cable tensions in these applications will frequently approach 50% of the cable breaking strength and, as a result, larger minimum bend diameters must be maintained throughout the system while handling greater loads to obtain the best performance and extend the life of the cable. The need for optimizing cable and umbilical performance and the introduction of new cable designs to meet these demands have resulted in an ever increasing challenge for handling systems which allow optimum cable performance and maximize cable life.

1.0 TRACTION WINCH SYSTEM ADVANTAGES

Sheave diameter is determined by the cable's minimum bend diameter which increases as resulting loads approach 100% of the cable breaking strength. Large diameter traction winch sheaves permit cable to be reeved in a single layer and to further avoid being subjected to unacceptable bend diameters. Traction winch system sheaves absorb high line pull loads by allowing the cable to work in a formed groove at the required minimum bend diameter. This single-layer effect is significant when compared with high line pull conditions in a conventional drum winch in which cable works on itself as a result of multiple layer wraps.

In this case, the cable will, at cross-over points, be subjected to a bend diameter equal to that of the actual cable diameter, resulting in possible damage to the cable. In addition, under extremely high tension, the cable may also pull down with a force great enough to penetrate existing cable layers on the winch drum.

The effects of multiple grooving, single cable layering and applied torque from powered traction sheaves are combined to absorb high line tension, translating to low tension as cable exits the inboard traction sheave. The cable is then stored in multiple layers on a storage winch under low line tension in a range equal to approximately 10 to 15% of maximum operating load throughout the entire cable scope. Consistent, low storage tensions also improve accuracy and repeatability when levelwinding the cable. As a comparison, storage tension on a conventional drum winch can vary between cable layers in response to varying dynamic tow loads.

2.0 THEORY OF OPERATION

Traction winches are a friction drive device. By maintaining a sufficient arc of contact, an adequate coefficient of friction, and appropriate back tension on the inboard end of the cable or rope, controlled movement of the cable will be affected by powering the traction sheaves.

We can use the formula: $T_1=T_2e^{\mu\beta}$ to calculate the total developed line pull.

B = included angle formed by arc of cable contact with sheave (radians)

μ = coefficient of friction

e = 2.718

T_1 = high tension leg

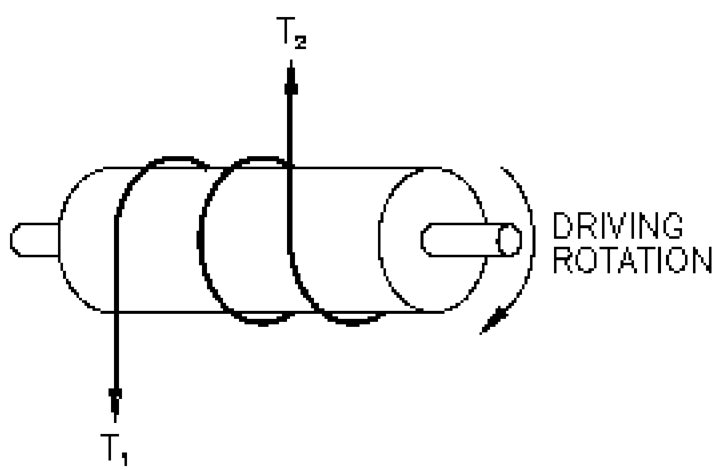
T_2 = low tension leg

Note that if T_2 , the low tension leg is brought to zero then T_1 also goes to zero.

Another important point regarding traction winch design is in addition to the coefficient of friction, traction depends on the arc of contact and not the diameter of the sheave. If the arc of contact is two half wraps or 360 degrees, traction would be the same with a 50 inch diameter sheave as with a 100 inch diameter sheave. It makes no difference that the length of rope in contact is twice as long with the larger sheave.

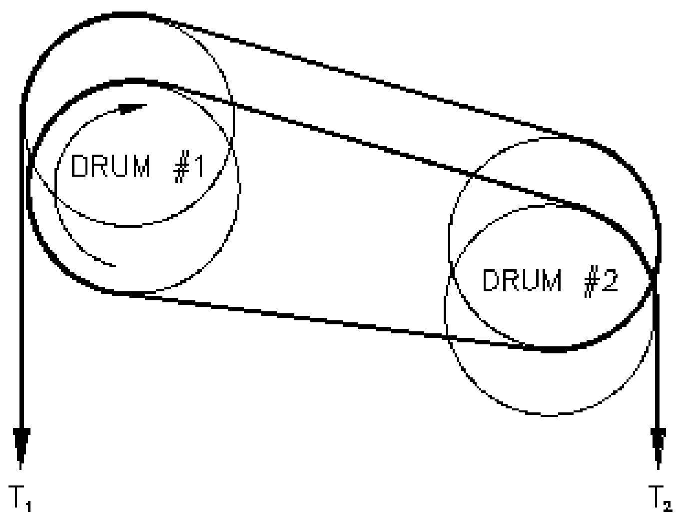
Figure 1 shows the most common form of traction device is the single drum capstan used extensively on ships for handling mooring lines. The drawback to handling long lines is the axial movement of the line across the face of the drum. This causes frictional wear and/or rotation of the line. These are best used when low line to drum friction coefficients are encountered and the line can slide axially across the drum face.

Figure 1



The double drum traction winch with a companion storage winch is commonly used with long lengths of cable. In most cases, both drums are powered and have multiple grooves. The cable is reeved from drum to drum and groove to groove providing the axial movement without producing the axial friction inherent in a single drum capstan.

Figure 2



Earlier double drum traction winches were designed to generate high line pull when using a conventional winch was not feasible. They did not always treat the cable very well. Many systems were designed using a single drive powering both of the sheaves. The sheaves were simple offset one half of the cable diameter to help with the transition of the cable from groove to groove. This created several forms of abuse to the cable. Both of the sheaves are forced to turn at the same speed however due to machining tolerances and groove wear, different surface velocities on the two sheaves occur. This requires that the cable slip on one of the sheaves. If the friction is high enough, the cable may not be able to slip and extremely high tension can occur within the wraps and armor wires can be broken.

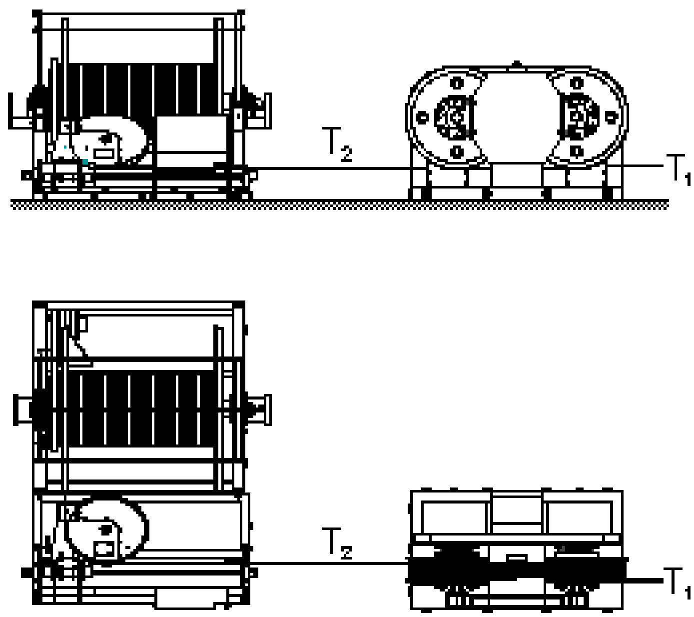
The problem with simply offsetting the sheaves by one half a cable diameter is that while it may appear that a smooth transition is taking place in fact the cable is subjected to a sharp bend as is leaves the sheave and a rolling or twisting of the cable also occurs.

Another form of abuse can occur between the traction winch and storage winch. Using a storage winch and conventional guide roller levelwind sometimes subjected the cable to severe bending due to the close proximity of the storage winch and traction winch.

3.0 ADVANCES IN TRACTION WINCH SYSTEM DESIGNS / OPTIMUM USE AND CONSERVATION OF DECK SPACE

The modern traction winch system is a multi-component system designed to extend cable life while optimizing available deck space. Current traction winch system designs includes a dual sheave traction winch, storage winch, right-angle levelwind and hydraulic power unit.

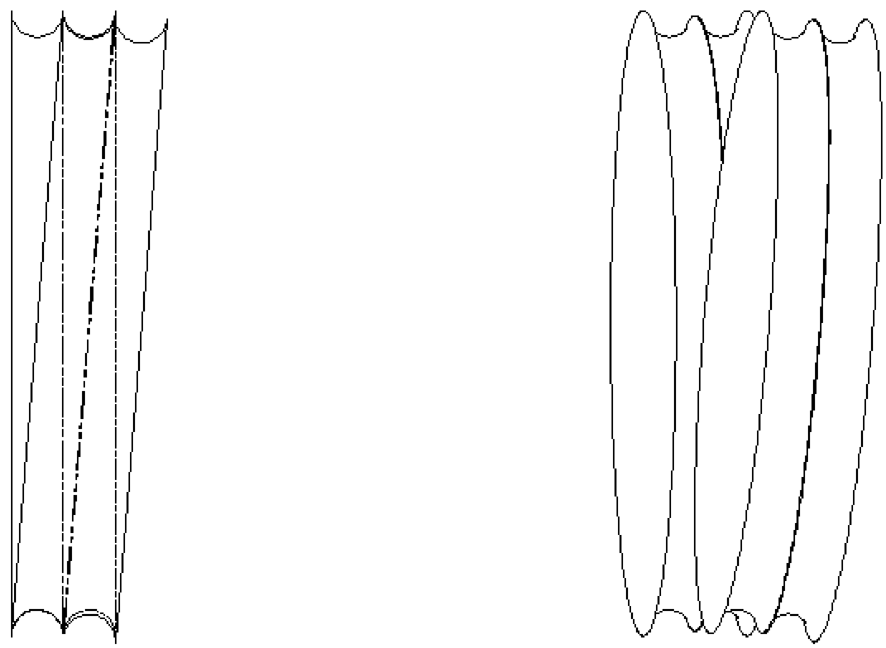
Figure 3



Advances in traction winch systems include designs featuring cantilevered traction winch sheaves with exceptionally high line pull ratings. The cantilevered sheaves allow clear passage of cable on the top and bottom of the traction sheaves. This arrangement allows cable to simply be lifted off the traction winch sheaves without removing cable from or through any other handling system components. The cantilever design also permits greater freedom in configuring the traction winch system in a range of horizontal to vertical positions to optimize available deck space.

By canting the sheaves in relation to each other, each sheave is positioned to facilitate a smooth transfer of cable between grooves of the two sheaves. This relative positioning eliminates twisting and/or chafing of the cable that might be associated with groove transfer. Figure 4 shows two views; the left view is looking aft or overboard from the storage winch. The right view is the same set of sheave viewed slightly from the side.

Figure 4



A properly designed traction winch system also includes independently driven traction sheaves. Independent sheave drives compensate for sheave machining variations and sheave groove wear by providing the same relative surface speed.

Since today's umbilicals are becoming more sophisticated and expensive, the emphasis should be on protecting the umbilical. In the past, some designers used hardened groove material to minimize or eliminate groove wear. This not only lowered the coefficient of friction but also increased wear on the armor wires. By using softer materials and allowing independent sheave speeds, groove wear is insignificant and armor wire life is greatly enhanced.

4.0 LEVELWINDS

In the event that overboarding devices are located in close proximity to the winch, the resulting fleet angles may require a significant cable wrap angle around levelwind guide rollers. As a general rule, guide rollers do not maintain the cable manufacturer's minimum bend diameter while accommodating these various fleet angles.

For example, in order to maintain the cable manufacturer's minimum bend requirements, a 40-inch minimum bend diameter could be maintained in a conventional drum winch with levelwind guide rollers only if the rollers had an outside diameter equal to 40 inches.

A right angle levelwind provides the dual benefit of maintaining the required cable minimum bend diameter while further optimizing deck space.

The levelwind is used to fairlead cable from the traction winch through a sheave assembly integrated to the levelwind and onto the storage winch drum at a right angle. The levelwind sheave maintains the manufacturer's minimum bend diameter of the cable under low tension with no change in fleet angle. As a result, the use of a right angle levelwind sheave has essentially eliminated the need for guide rollers in a levelwind.

5.0 APPLICATION SUITABILITY OF A TRACTION WINCH SYSTEM

When considering the use of a traction winch system, the following factors should be examined:

a) Optimum operating depth
b) Length of cable required
c) Cable composition
d) Cable breaking strength
e) Minimum bend diameters of cable at 0%, 25% and 50% of cable breaking strength
f) Total weight of cable and payload in seawater
g) Drag coefficient, if applicable
h) Anticipated line pull and line speed requirements
i) Variations in dynamic loads resulting from operations in high sea states or frequent changes in ship speeds

j) Winch drum brake requirements

Although a number of the items listed above are a function of operating depth, priority should be assigned to the optimum line pull requirements and/or winch braking capacity relative to the cable breaking strength. The duty cycle of the particular application at hand should also be examined to determine the relationship between total operational time vs. time spent at maximum line pull when approaching 25% to 50% of cable breaking strength.

Cable length should be reviewed not only from the standpoint of the total cable weight in seawater and the resulting impact on line pull, but also the levelwinding ability of cable, particularly at longer lengths. Storage behavior properties of cable under high tension should also be reviewed.

It should be re-emphasized that high line pull alone should not necessarily be viewed as the sole criterion in the consideration of a traction winch system. As operational line pull and/or winch braking capacity approaches 25 to 50% of cable breaking strength, the comparison becomes significant.

6.0 ECONOMIC JUSTIFICATION OF TRACTION WINCH SYSTEMS

Associative umbilical and payload costs will require winch and handling system to be further refined with increasing attention to cable preservation by adapting to the specific needs of individual customers.

As the ocean industry continues to work at greater depths, vehicle/umbilical design and subsequent capital expenditures will continue to advance in support of more complex operations. Increasing applications for fiber optic cable, as an example, can result in typical costs ranging from $30.00 US per meter for standard 3 fiber-3 power 17mm diameter armored cable to $130.00 US per meter for 6 fiber-38mm diameter Kevlar jacketed umbilicals. Depending on cable length and related costs per meter, traction winch systems may represent a small percentage investment relative to the total cable cost.

Traction winch systems can extend the life of a cable and may allow cable to be used at tensions considerably higher than normal.

As umbilical lengths increase, cable heating concerns require greater awareness. Traction winch systems may allow a cheaper alternative in managing heat problems by allowing cable to be stored on larger drums with fewer layers under low tension.

7.0 BASIC DESIGN CRITERIA FOR A TRACTION WINCH SYSTEM

In the past, a general rule of thumb for sheave diameters was to use a ratio of sheave diameter to cable diameter of 40:1. With the advent of deeper tow systems and the desire to explore all ocean depths, cable lengths have increased and with it the loads placed on the cables. It is not uncommon to load 12,000 meters of .680 inch coaxial or power optic cable onto a winch system. The in-water weight of the cable added to the payload and drag coefficients now have these cables operating at loads as high as 50% of breaking strength.

Simply increasing the size of the cable leads to diminishing returns as a major portion of the load is the weight of the cable. This has lead winch designers and cable experts to look for ways of extending the safe operating loads of existing cables. Cable testing using a larger bend ratio of 80:1 indicated that cable life was acceptable when these larger sheaves were used with high tension loads. Another interesting find was that some fiber optic cables suffer failures from storage at high tensions. Traction winch systems using low tension storage offer a solution to this problem.

When calculating the power required for a traction winch system it can be assumed that the power required by the winch system is shared by the traction winch and the storage winch. In other words, if a total of 100 horsepower is required to lift a load, then 90 could be used by the traction winch and 10 by the storage winch. The same holds true for line pull. In a system with a total line pull of 20,000 pounds, the traction winch could provide 18,000 pounds of line pull and the storage winch 2,000 pounds of line pull.

When designing a traction winch system, the drive train for the storage winch is especially critical. If the torque characteristics and response time of the drive train are not optimized, then dangerous conditions can exist. Slack forming between the traction and storage winch, excessive line pull from the storage winch during payout and insufficient line pull by the storage winch during haul-in are all typical results.

8.0 WINCH POWER CALCULATIONS

A simple formula for calculating power required by a winch is:

Horsepower = (line speed in Ft/Min x line pull in pounds)/33,000

Example: A typical deep tow traction winch system requires a line pull of 20,000 pounds at a speed of 200 feet per minute. The mechanical horsepower at the winch system is (200 x 20,000)/33,000 = 121.21 HP. Typical efficiencies of modern hydraulic winch systems is approximately 70% so 121.21/.70= 173 HP. The closest electric motor to this size is 200 HP. Final design would attempt to provide slightly higher performance to utilize all of the available power.

One of the benefits of a traction winch system is that performance is the same regardless of the amount of cable paid out since line speed and line pull is not affected by the amount of cable paid out as it is with a conventional single drum winch.

Most traction winch systems are either electro-hydraulic or diesel-hydraulic. It is possible to design an all electric system however a minimum of three electric motors would be required and packaging difficulties could make the overall design unwieldy.

9.0 DRUM AND GROOVE DESIGN

Early traction winch designs primarily used a single drive powering both of the traction sheaves. Because of machining tolerances and varying cable elongation, the cable was forced to slip on one of the sheaves. This caused either accelerated wear on the sheave or the cable, whichever was softer. With today's expensive cables, it is not an option to sacrifice the cable to save the sheave. Independent, balanced torque drives for the traction sheaves alleviated this problem. Softer sheave material resulting in a higher coefficient of friction and sheaves that can turn at different speeds from each other greatly eliminate the wear problems. Some early designs used “V” or modified “V” grooves to increase the coefficient of friction when using very hard sheave material. This caused deformation of the cables and with today’s fiber optic cable is unacceptable.

When designing a traction system, the designer must first determine what the maximum line pull of the system needs to be. Next, what is the desired or maximum storage tension? A coefficient of friction needs to be assumed – usually greased steel on steel. The Machinery Handbook, 24th Edition, gives a coefficient of friction for lubricated steel on steel of .16 and just to be conservative, we will use .10 in the sample calculation. This tends to be the worst case and by default the safest assumption.

Now to determine the number of grooves in the sheaves, we use the following equation: $\ln(T_1/T_2)/\mu=\theta$

Where:

ln=natural log

T_1= high tension leg, in this case 20,000 pounds

T_2= low tension leg, in this case 2,000 pounds

μ = coefficient of friction, in this case .10

θ = sheave radians of contact

$\ln(20{,}000/2{,}000)/.10 = 23.025$ radians

or 23.025/6.28 = 3.66 total wraps around the sheaves. Rounding up this indicates that a minimum 4 grooves in each sheave need to have 180 degrees of contact.

10.0 CONCLUSION

Traction winch systems should be considered as an economically viable cable handling option when operational line pull approaches 25% to 50% of cable breaking strength and/or brake ratings meet or exceed cable breaking strength. As operating depths and related cable lengths increase, traction winch systems are particularly attractive as these operating parameters converge.

Ocean industry users should look beyond acceptance of the traction winch concept to ensure that the selected traction winch design addresses the issues of sheave construction, sheave relative position and sheave drives.

The hallmark of a properly designed traction winch system is the ability to apply operating performance that is consistent throughout the cable scope, regardless of length.

CHAPTER 12

Alan Driscoll **Reviewed by J. F. Bash/Walter Paul**

1.0 WIRE ROPE DATA

Bright or AMGAL MONITOR AA Torque-Balanced Rope

Size Inches	Construction (Seale)	Wt.in Air - lbs/ft	Wt.in Water lbs/ft	Approx. Elastic Limit	Breaking Load lbs.	0.2 % Yield Strength lbs	Max. Length ft
3/16	3x19Seale	.0586	.0509	3,000	4,000	3,500	50,000
1/4	3x19 "	.0997	.0867	3,063	6,750	5,900	45,000
5/16	3x19 "	.153	.133	7,725	10,300	9,100	30,000
3/8	3x19 "	.220	.191	11,100	14,800	13,000	50,000
7/16	3x19 "	.304	.264	15,000	20,000	17,600	42,000
1/2	3x19 "	.392	.341	19,275	25,700	22,600	98,000
9/16	3x19 "	.492	.428	24,375	32,500	28,600	77,000
5/8	3x19 "	.602	.523	30,225	40,300	35,500	62,000
3/4	3x19 "	.879	.764	43,350	57,800	50,900	43,000
7/8	3x19 "	1.21	1.05	58,500	78,000	68,600	32,000
1	3x19 "	1.56	1.36	75,450	100,600	88,500	24,000
1 1/8	3x19 "	1.96	1.70	93,000	124,000	109,000	19,000
	Seale FW						
1/2	3x46	.417	.362	19,275	25,700	22,600	98,000
9/16	3x46	.517	.449	24,375	32,500	28,600	77,000
5/8	3x46	.631	.548	30,225	40,300	35,500	62,000
3/4	3x46	.903	.785	43,350	57,800	50,900	43,000
7/8	3x46	1.27	1.10	58,500	78,000	68,600	32,000
1	3x46	1.64	1.43	75,450	100,600	88,500	24,000
1 1/8	3x46	2.07	1.80	93,000	124,000	109,000	19,000
1 1/4	3x46	2.60	2.26	118,500	158,000	139,000	15,500
1 3/8	3x46	3.10	2.69	141,000	188,000	165,000	12,900
1 1/2	3x46	3.69	3.21	166,500	222,000	195,000	10,800
1 5/8	3x46	4.43	3.85	198,750	265,000	233,000	9,200
1 3/4	3x46	5.12	4.45	228,000	304,000	267,000	8,000

3 x 7 TYPE 304 STAINLESS STEEL ROPE

Size	Minimum Breaking Strength lbs.	Approx. Elastic Limit lbs.	Min. 0.2% Yield Strength lbs.	Weight lbs./ft.	Area Sq. In.
5/32	2,800	2,100	2,460	.0406	.01096
11/64	3,300	2,470	2,900	.0491	.01326
3/16	3,900	2,920	3,500	.0578	.01561
7/32	5,000	3,750	4,500	.0745	.02011

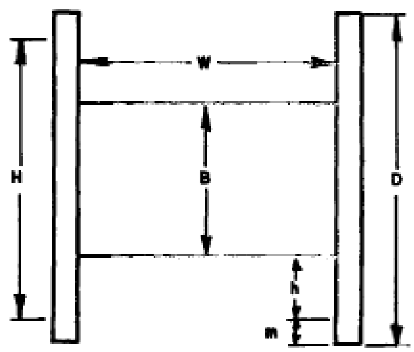
3 x 19 TYPE 304 STAINLESS STEEL ROPE

Size	Minimum Breaking Strength lbs.	Approx. Elastic Limit lbs.	Min. 0.2% Yield Strength lbs.	Weight lbs./ft.	Area Sq. In.
11/64	3,500	2,620	3,100	.0512	.01394
3/16	4,000	3,000	3,500	.0592	.01611
7/32	5,400	4,050	4,750	.0803	.02184

3 x 19 TENELON STAINLESS STEEL (*for non-magnetic applications)

Size	Minimum Breaking Strength lbs.	Approx. Elastic Limit lbs.	Min. 0.2% Yield Strength lbs.	Weight lbs./ft.	Area Sq. In.
3/8	12,700	8,900	11,200	.221	.06015
7/16	17,200	12,900	15,100	.299	.08114
1/2	22,000	16,500	19,400	.388	.10529
9/16	28,000	21,000	24,100	.487	.13255

* This product for non-magnetic applications. To improve its resistance to Stress Corrosion, the wires may be galvanized.



Let D = Diameter of Head in Inches.
B = Diameter of Barrel in Inches.
h = Depth of Cable in Inches.
W = Width between Flanges in Inches.
d = Diameter of Cable in Inches.
L = Length of Cable in Feet.
m = margin.

To Compute Length of Cable in Feet for any Reel or Drum:

L = Factor x W x h x (B + h)

A table of factors for ropes to the maximum oversize tolerance (as shown previously in this handbook) is presented below.

Nominal Rope Dia.	Factor	Nominal Rope Dia.	Factor
1/4	3.728	2 1/4	.0469
5/16	2.432	2 3/8	.0421
3/8	1.689	2 1/2	.0380
7/16	1.241	2 5/8	.0345
1/2	.9498	2 3/4	.0314
9/16	.7505	2 7/8	.0287
5/8	.6079	3	.0264
3/4	.4222	3 1/8	.0243
7/8	.3102	3 1/4	.0225
1	.2375	3 3/8	.0208
1 1/8	.1876	3 1/2	.0194
1 1/4	.1520	3 5/8	.0181
1 3/8	.1256	3 3/4	.0169
1 1/2	.1055	3 7/8	.0158
1 5/8	.0899	4	.0148
1 3/4	.0775	4 1/4	.0131
1 7/8	.0675	4 1/2	.0117
2	.0594	4 3/4	.0105
2 1/8	.0526	5	.0095

The Formula can be readily derived:

(1) Length of Coil of Middle Layer

$$= \frac{\pi}{12}\left(B + \frac{H - B}{2}\right)$$

Number of Coils $= \frac{W}{d}$

Number of Layers $= \frac{H - B}{2d}$

$$L = \frac{\pi}{12}\left(B + \frac{H - B}{2}\right) \times \frac{W}{d} \times \frac{H - B}{2d}$$

$$= \frac{\pi W(H + B)(H - B)}{48d^2}$$

(2) Volume of Drum in Cubic Inches

$$= W\left(\frac{\pi H^2}{4} - \frac{\pi B^2}{4}\right)$$

$$L = \frac{W}{12d^2}\left(\frac{\pi H^2 - \pi B^2}{4}\right) = \frac{\pi W}{48d^2}(H^2 - B^2)$$

$$= \frac{\pi W(H + B)(H - B)}{48d^2}$$

$$L = \frac{\pi W(H + B)(H - B)}{48d^2} = \frac{W(H+B)(H-B)}{15.28d^2}$$

$$= \frac{.06545\ W(H + B)(H - B)}{d^2}$$

$$= \frac{.2618\ Wh\ (B + h)}{d^2}$$

$$\text{Let Factor} = \frac{.2618}{d^2}$$

then

$$L = \text{Factor} \times W \times h \times (B + h)$$

When the diameter of the rope is not full oversize or when strand is to be reeled, the actual product diameter should be used with the formula

$$L = \frac{.2618\ Wh\ (B + h)}{d^2}$$

to determine capacities.

When shipping rope on reels, the reels should not be completely filled. A margin (m) should be left to protect the rope.

This Formula is based on the assumption that: the rope is oversize and does not flatten when coiled; and that it is in perfectly uniform layers with no meshing of the coils. These factors vary with size and construction of the cable and with the dimensions of the reel or drum. As these variables tend to offset each other, this method of computing reel and drum capacities has proved to be reliable.

ELECTRO-MECHANICAL CABLE DATA

All values are nominal unless otherwise stated.

CABLE TYPE	DIAMETER		CABLE B.S.	WEIGHT	WIRE COUNT & B.S.		STRETCH	dcR	CAP.	MAX. TEMP.	SHEAVES
	FRACTION	INCHES	(lbf)	(lb/kft)	(out/in)	(lbf)	ft/kft/klbf	(Ω/kft)	(pF/ft)	(°F)	(Inches)
1-H-100A	1/10	.101	1,000	19	18/12	43/43	—	25.2	40	300	6
1-H-125A	1/8	.123	1,500	27	18/12	64/64	—	25.2	35	300	7
1-H-125K	1/8	.123	1,500	28	18/12	64/64	—	25.2	50	500	7
1-H-181A	3/16	.185	3,900	63	15/12	198/127	4.0	9.8	50	300	12
1-H-181D	3/16	.185	3,900	65	15/12	198/127	4.0	9.8	55	420	12
1-H-181K	3/16	.185	3,900	65	15/12	198/127	4.0	9.8	51	500	12
1-H-181M	3/16	.187	3,900	68	15/12	198/132	4.0	12.5	45	600	12
4-H-181A	3/16	.186	3,300	60	18/18	143/76	5.1	26.0	50	300	10
1-H-203A	13/64	.203	4,500	79	16/10	211/211	3.2	6.9	60	300	12
1-H-203D	13/64	.203	4,500	79	16/10	211/211	3.2	6.9	65	420	12
1-H-203K	13/64	.203	4,500	80	16/10	211/211	3.2	6.9	65	500	12
1-H-220A	7/32	.223	5,500	92	18/12	211/211	2.6	4.5	56	300	12
1-H-220D	7/32	.223	5,500	95	18/12	211/211	2.6	4.5	68	420	12
1-H-220K	7/32	.223	5,500	95	18/12	211/211	2.6	4.5	58	500	12
1-H-220M	7/32	.223	5,500	97	18/12	211/211	2.6	5.2	52	610	12
1-H-226K (MP35N)	7/32	.222	5,000	99	18/12	207/207	2.6	7.7	45	500/450	12
1-H-226K (31MO)	7/32	.223	4,800	96	18/12	189/189	2.6	7.7	37	500/350	12
1-H-281A	9/32	.288	10,000	153	18/12	352/352	1.6	2.9	55	300	16
1-H-281K	9/32	.288	10,000	158	18/12	352/352	1.6	2.9	54	500	16
1-H-314A	5/16	.316	11,200	183	18/12	426/426	1.3	2.9	47	300	17
1-H-314D	5/16	.316	11,200	188	18/12	426/426	1.3	2.9	55	420	17
1-H-314K	5/16	.316	11,200	190	18/12	426/426	1.3	2.9	48	500	17
1-H-314M	5/16	.318	11,200	193	18/12	426/426	1.3	3.3	42	610	17
7-H-314A	5/16	.325	9,600	180	18/18	426/225	1.9	16.6	58	300	17
1-H-375A	3/8	.375	14,600	253	18/12	595/595	1.0	2.9	39	300	20
1-H-375D	3/8	.375	14,600	260	18/12	595/595	1.0	2.9	45	420	20
1-H-375K	3/8	.375	14,600	261	18/12	595/595	1.0	2.9	45	500	20
3-H-375A	3/8	.372	13,500	240	20/16	486/397	1.2	7.1	43	300	18
4-H-375A	3/8	.372	13,500	239	20/16	572/301	1.4	10.0	41	300	18
7-H-375A	3/8	.372	12,800	243	18/18	572/301	1.4	10.0	66	300	20
1-H-422A	7/16	.414	17,800	307	18/12	727/727	0.8	2.9	35	300	23
1-H-422D	7/16	.414	17,800	316	18/12	727/727	0.8	2.9	40	420	23
1-H-422K	7/16	.414	17,800	317	18/12	727/727	0.8	2.9	40	500	23
7-H-422A	7/16	.426	18,300	314	18/18	766/397	0.9	11.0	57	300	23
7-H-422D	7/16	.426	18,300	324	18/18	766/397	0.9	10.0	63	420	23
7-H-422K	7/16	.426	18,300	326	18/18	766/397	0.9	10.0	54	500	23
7-H-464A	15/32	.462	18,300	326	24/24	539/335	0.9	10.0	43	300	20
7-H-464D	15/32	.462	18,300	333	24/24	539/335	0.9	10.0	45	420	20
7-H-464K	15/32	.462	18,300	347	24/24	539/335	0.9	10.0	40	500	20
7-H-472A	Slammer	.472	22,200	379	18/18	929/486	0.8	10.0	47	300	26
7-H-472D	Slammer	.472	22,200	386	18/18	929/486	0.8	10.0	50	420	26
7-H-472K	Slammer	.472	22,200	394	18/18	929/486	0.8	10.0	45	500	26
7-H-520A	17/32	.522	26,000	462	20/16	958/778	0.6	10.0	42	300	26
7-H-520D	17/32	.522	26,000	467	20/16	958/778	0.6	10.0	46	420	26

Courtesy of the Rochester Corporation

3.0 DATA ON ROPES FROM INDUSTRIAL SYNTHETIC FIBERS

3.1 INTRODUCTION

In Chapter 3 High Strength Synthetic Fiber Ropes are described. High strength ropes, or more specifically *Ropes Made from High Performance Synthetic Fibers,* have found significant applications in oceanographic work. However in the majority of oceanographic and general ship operations ropes from “industrial” fibers are used. Industrial synthetic fibers[1], also called “conventional” fibers, are nylon, polypropylene, copolymers of polypropylene and polyethylene, and polyester[2]. Compared to wire ropes, all fiber ropes are more stretchable and flexible, do not corrode, but are sensitive to cutting, abrasion, in most cases extended sunlight exposure, and high temperatures which can fuse or melt the fibers (for example at contact areas over which a moving and tensioned rope is bent). Compared to ropes from high performance fibers, ropes produced from industrial fibers are weaker, more flexible, and can stretch from twice to over ten times as much before failure. This section highlights pertinent properties of ropes made from industrial synthetic fibers and compares them with fiber ropes made from high performance synthetic fibers and with wire ropes. This section also lists Safe Use Guidelines as well as rope data for the most frequently used rope types, made from industrial synthetic fibers, taken from fiber rope specifications issued by the *Cordage Institute*.

3.2 GENERAL PROPERTIES OF INDUSTRIAL FIBER ROPES

Strength, Size, Weight, and Cost: Comparison with High Performance Ropes and Wire Ropes

Ropes of the Same Size: Ropes from industrial fibers, depending on fiber material and rope construction, have between 10 and 64 percent of the strength of high performance fiber ropes of the same diameter. The ropes can be heavier or lighter than their high strength counterparts depending on the fiber materials used. All fiber ropes weigh in air only 10 to 20 percent of a wire rope of the same diameter, and at most 4 percent of the wire rope weight when submerged in the ocean.

Ropes of the Same Strength: Ropes from industrial fibers have a 40 to 70 percent larger diameter, and two to three times the weight and fiber mass of equally strong high performance ropes. Depending on the fiber material and rope construction equally strong (but larger) ropes from conventional synthetic fibers have 65 to 91 percent of the dry weight of wire rope, and 0 to 26 percent of the wet wire rope weight when submerged in the ocean[3].

Cost: The purchase price per pound of rope from conventional synthetic fibers is significantly lower than that of rope produced with high performance fibers. Polypropylene ropes are the least expensive, but are larger than nylon, polyester, and polyethylene-polypropylene copolymer ropes of the same strength. However in a number of applications

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[1] Industrial fibers are defined as fibers having an average tenacity of 7.0 and 15.0 grams/denier, high performance fibers have an average tenacity of 20 to 24 grams per denier. (Cordage Institute CI 2003 Guideline “Fibers for Cable, Rope and Twine”, Wayne, PA, 2001). Grams/denier is a textile term similar to breaking stress in solid materials. It is calculated by dividing the breaking load of a fiber or rope through its weight per unit length. A denier is the weight of a 9,000 meter long textile fiber, yarn, or rope. Fibers and yarns are measured in denier. 1 gram/denier is approximately 20,000 psi, dependent on fiber density.

[2] The industrial synthetic fibers have almost completely replaced the natural Manila fiber, which was the only largely available and used cordage fiber for oceanographic and marine applications until the 1950s and 1960s. In this 3rd edition of the Handbook the table for Manila fiber rope has been omitted.

[3] The diameter, strength and weight data of 6 x 19 wire rope with wire rope and fiber rope cores are used for comparison.

the high performance ropes can be more cost effective, in particular when the cost of smaller winch sizes and rope storage reels or storage containers are considered.
Larger Rope Size, Problems and Advantages: Storage space and weight of larger ropes becomes a problem in applications where space is a premium or handling crew size is limited[4]. However the larger rope size can be also of considerable advantage. The larger fiber mass provides much more cushion against abrasion damage. The strength drop due to similar wear damage is proportionally a lot less in a larger rope, since less of the load carrying rope cross section area is destroyed. The reduced relative wear results in increasing service life and lowers the risk of operation. Exchanging a high performance rope with equal strength than an established rope from industrial fibers has caused serious accidents due to the much faster wear of some of the considerably thinner ropes made from high performance fibers and high strength industrial fibers, this is a dangerous practice[5].

Fiber and Rope Stretch:

Fiber Stretch: All currently available high performance fibers suitable for ropes have a low range of stretch a break and at maximum working load. Industrial synthetic fibers suitable for ropes have considerably higher working and breaking elongation, see Table 3-1.

Fiber Material:	High-Performance	Nylon	Polyester	Polypropylene	PE-PP Co-Polymer
Elongation at Break [%]	2.8 – 4.6	15 - 28	7 – 12	18 – 22	14 - 18
Elongation at max. Work Load [%]	1 – 2	7 - 12	5 – 10	9 – 13	5 - 10

Note: Through variation of spinning conditions lower and higher stretching fibers of a specific material are available.

Table 3-1: Elongation at Break and at Maximum Working Load for High Performance and Industrial Textile Fibers

Rope Stretch: Rope stretch depends on both the fiber elongation and the rope construction[6]. Depending on the number of steps and the amount of twists applied the rope construction can significantly increase the rope elongation beyond the fiber elongation (as high as two to three times). Table 3-2 lists strength, stretch, and weight data for wire rope and ropes from industrial fibers. The values of elongation at break are for new ropes, the

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[4] Strong towing hawsers made of low density high performance and industrial fibers became light enough that one or two people can handle and move around the towlines of modern tractor tugs without undue strain. These ropes are used instead of a long wire rope.
[5] A replacement of a lower strength rope with a higher strength rope should be made to result in the same strength loss after comparable use for both ropes. Or a high performance rope has to be sized with a higher strength than the rope from industrial fibers it replaces. Goal is to retain a comparable residual breaking strength under similar depth of surface wear on both ropes after a given service period.
[6] The rope construction is needed to build a coherent mostly cylindrical rope construction in a number of steps. The large number of fibers have to be bundled and controlled through twisting operations to form yarns and multi-plies, which in turn are twisted together to form rope strands (a nylon or polyester rope of 1.375 inch diameter contains about 1 million fibers). Rope strands are either twisted to form a 3-strand twisted or laid rope, plaited together to form an eight-strand rope, or braided together to form a 12-strand braid or a double braided rope construction. Other rope constructions have been developed, such as a parallel fiber core with a cover braid, the cable laid construction for large twisted ropes, wire-lay constructions with a cover braid, and the 12 by 12 braided construction for large 12-strand braided ropes.

stretch values at maximum working load are for used, heavily cycled ropes. In use ropes become permanently longer and lose about half of their initial elongation due to compacting of their fibrous structure and reduction in fiber stretch (similar to work hardening in metals). Exception: Nylon rope shortens and swells in hot and wet storage due to shrinkage, which increases its stretch, counteracting the elongation reduction under loading in use.

	Breaking Strength Range [lbs]	Elongation at break [percent]	Stretch at max. Working Load [%]	Weight In air [lbs/ft]	Weight In sea water [lbs/ft]
Wire Rope	71,000-100,000	1-2	0.5-1	1.85	1.61
Ropes from Industrial Synthetic Fibers					
Ropes from Nylon (Polyamide) Fibers	25,000-32,000	30-50	11-20	0.26	0.025
Ropes from Polyester Fibers	25,900-46,500	12-35	3-8	0.32-.35	0.06-0.07
Ropes from Polypropylene Monofilaments	12,600-17,600	20-33	7-16	0.17-.18	Floats
Polypropylene-Polyethylene Copolymer Ropes	16,600-22,300	14-20	4-8	0.17-.18	Floats

Note: The wide stretch range of ropes from the same material is due to the typical structural stretches of different rope constructions, combined with the variations in fiber stretch behavior due the fiber processing variations. For specific rope constructions from a given material a much narrower elongation range is observed and can be specified.

Table 3-2: Breaking Strength, Elongation at Break and at Maximum Working Load for One-Inch Diameter Wire Ropes and Ropes from Industrial Fibers

3.3 SELECTION CRITERIA FOR ROPES FROM INDUSTRIAL FIBERS

Ropes to Absorb Dynamic and Shock Loads:

Where significant dynamic loads have to be absorbed, ropes with high working stretch (9 percent and up) are needed. Lifting and lowering lines for ROVs and AUVs, open ocean towing rope, fall arrest and safety ropes, surface buoy mooring ropes for deep ocean sites, and ropes to moor ships at sea, require significant elongation under load in order to lessen shock effects and reduce fatiguing oscillating loads. Nylon fiber is the most suitable and often only possible material for ropes in these applications.

Ropes for General Oceanographic Work and Ship Use:

Low to modest working stretch (2-10 percent) is required for ropes used for deck work support, docking lines in calm water, ropes for subsurface buoy moorings, and large diameter mooring lines for deep water tension leg oil platforms. Polyester is the most suitable material for most of these applications, and polypropylene ropes are used where their light weight (buoyant in sea water) is of advantage. Where higher strength combined with light weight is needed, ropes from polyethylene-polypropylene copolymer fibers are suggested[7].

Abrasion and Wear:

Synthetic fiber ropes are threatened by wear and cuts, the threat could be from rough decks and rope hardware surfaces (e.g. chocks and sheaves), contact with certain types of sea floor, contact with rust particles, cuts by certain fish species in particular on thin ropes. Despite these threats fiber ropes can have remarkable survival records when selected and handled properly and within safe working load limits, and when contact surfaces are smooth and without sharp edges. Deep water buoy moorings have survived up to seven years in exposed positions with only modest strength loss, and some docking lines have shown similar longevity. Keeping working loads low and avoiding abrasive contact and slack conditions with subsequent snap loading in sea states are examples of maintaining favorable working conditions for all fiber ropes and many other load transferring or load supporting materials. The different rope fibers react differently to external wear. Water works as abrasion reducing lubricant in polypropylene ropes, and increases abrasion for nylon ropes not treated with a protective surface finish. Polyester ropes have in general the best resistance again abrasion.

Special rope constructions are offered with increased wear resistance. These include mooring and utility ropes with rope strands produced with an external layer or cover braid of more abrasion resistant polyester over a mainly load carrying and less costly core of polypropylene or polyethylene–polypropylene copolymer. Another way to improve resistance to surface wear is the covering of nylon and polyester rope with a coating of polyurethane.

Braided or Twisted Rope Constructions

Braided or eight strand plaited rope constructions are usually completely torque and rotation free under load, since they are normally composed of an equal number of left-hand and right-hand twisted yarns or strands. The braided or plaited rope construction is preferred for oceanographic buoy mooring ropes and lift lines[8]. Twisted ropes are built from three or sometimes four equally sized rope strands with the same twist direction, the strands are turned or twisted together in the opposite twist direction to form the twisted rope. It is not possible to have exactly opposing torque of strands and rope, so twisted ropes have a tendency to rotate under tension, and developing permanent kinks or hockles (to absorb torque differences between strands and rope) in particular when experiencing sudden load release. Kinked ropes are severely compromised and weakened. In order to prevent the formation of kinks and hockles “wire-lay” rope constructions are available, which are covered with a sturdy outer braided jacket. The braided jacket greatly reduces or eliminates the risk of kink formation.

3.4 FIBER ROPE SPECIFICATIONS BY THE CORDAGE INSTITUTE

In the United States the most commonly used fiber ropes were covered by less than a dozen military specifications for many years, complimented by applicable textile test methods and test specifications issued by other federal agencies. The rope specifications were usually prepared for the government by the Technical Committee of the *Cordage*

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[7] Where ropes with lowest working stretch (1-2 percent), light weight, and highest strength are needed, high performance fiber ropes are best suited. If rope weight is unimportant, wire rope could be selected instead.

[8] Twelve-strand braids can be supplied as rather loose structures for increased strength. Such ropes can be easily shortened (squished) under no load, which can invite snagging or increased inter-strand abrasion. This is found in the top portion of taut surface buoy moorings due to frequent slack-snap conditions from wave action. A more compact and coherent twelve-strand rope structure is achieved with reduced strength and increased constructional stretch. In general within each material the lowest stretching rope construction is twelve-strand single braid, more elongation is found in double braided rope constructions, while plaited and twisted rope constructions have the highest rope stretch.

Institute, cooperating with technical representatives of federal and military agencies. The Cordage Institute represents the rope and twine industry in the United States. Around 1995 the Cordage Institute became the issuing organization for rope specifications[9]. Its Technical Committee has currently (year 2001) over fifty fiber rope related specifications and guidelines issued or prepared for circulation. The specifications include the main rope constructions from different fiber materials, guidelines for safe use and retirement, and a number of related test standards. Table 3-3 lists the specifications for ropes from industrial fibers, which are most frequently used in oceanographic and ship handling and towing work. Data sheets taken from specifications of the most commonly used ropes from industrial fibers are listed in on the following pages[10], preceded by general Safe Use Guidelines for fiber ropes.[11]. This information is compiled to help rope buyers and users to make the most suitable choice for their application [12].

Table 3-3: List of Specifications for Fiber Ropes from Industrial Fibers

Specification # *)	**Fiber Material(s)**	**Rope Construction(s)**
C! 1301	Polypropylene	3-strand twisted + 8 strand plaited
CI 1302A	Rope strands with outer layer from Polyester, core layers from polypropylene	3-strand twisted
CI 1302B	Rope strands with outer layer from Polyester, core layers from polypropylene	8 strand plaited
CI 1303	Nylon	3 strand twisted + 8 strand plaited
CI 1304	Polyester	3 strand twisted + 8 strand plaited
CI 1305	Polyester	12-strand braid
CI 1310 **)	Nylon	Double braided or 2 in 1 braid
CI 1311 **)	Polyester	Double braided or 2 in 1 braid
CI 1312	Nylon	12-strand braid
Cit1900	Polyethylene-polypropylene co-polymer	3-strand twisted + 8-strann plaited
CI 1901	Polyethylene-polypropylene co-polymer	12-strand braid

Notes: *) Individual specifications are in various stages of development some are still preliminary, some under revision. A mandatory review of each specification is conducted every five years. For the latest status contact the Cordage Institute

**) These specifications are for a high strength version of the double braided rope construction. There is a commercial lower strength rope of this type available under CI 1306 in polyester, CI 1307 in nylon

3.5 Safe Use of Ropes and Safety Factor Selection

The selection of the most suitable fiber rope with sufficient strength for a specific task can be challenging. System loads must be known before selection of an adequate rope can be made. Rope load elongation behavior and weight are inputs for system simulation, and

[9] The Technology Transfer Act of the National Institute of Standards shifted the authority to issue specifications from government agencies to the private sector wherever possible.

[10] Data sheets from the specifications for highly specialized ropes and twines such as life safety (fall arrest) ropes, and ropes with reduced recoil, are not included.

[11] Rope specifications and guidelines are available from the Cordage Institute. Contact on the web: www.ropecord.com, by phone 510/971-4854. The assistance by the Cordage Institute's Technical Director Mr. Gale Foster, and the permission to publish specification excerpts and guidelines in this Handbook is thankfully acknowledged.

[12] There are many trade names for different rope types. Competing firms offer the same rope product under different trade names. Large fiber manufacturers introduced trade names for groups of industrial and high performance fibers, rope producers followed with trade names for polyolefin fibers produced in their factories. The rope specifications help in selecting the most suitable rope candidate for a given task.

several iterative steps may be required to result in workable design solutions. Critical applications warrant tests which simulate the planned use. Expected average and maximum loads drive the rope strength selection, required stretch the fiber material, and the operating conditions a preferred rope construction. Table 3-4 provides Safe Rope Use Information issued by the Cordage Institute. It contains instructions for rope users and rope system designers. The selection of the best design factor (= breaking strength/maximum working load) for a given application requires good judgement and experience, and may need expert advice.

3.6 Physical Properties and Safe Working Load Limits of Selected Rope Types

After selecting the Design Factor (see Table 3-4) and the rope fiber material and rope construction, the correct rope size can be selected through the Safe Working Load data provided in the following Tables 3-5 to 3-14. Tables 3-5 to 3-15 form part of the Cordage Institute rope specifications listed in Table 3-3.

Table 3-4: CI 1401; Safe Rope Use Guidelines, Cordage Institute

1. Purpose

This Appendix is provided to emphasize the critical importance of selecting cordage and rope and related hardware by considering the many variables that exist when they are used as a single strength member or in a system.

The most important factor is to be aware of the critical conditions of use, including the degree of exposure to risk of personnel and property, if the rope were to break. It is always best to consult a rope engineer in the design and selection process.

2. Diameter and Size Number Values

Size is determined by linear density; diameter is given as a nominal value; that is, it may vary slightly. If a specific diameter is specified, linear density and minimum breaking strength values may be different from the tables. Size number is given as a reference.

3 Minimum Breaking Strength, Hardware and Terminations

Minimum Breaking Strength (MBS) is based on data from a number of rope manufacturers and represents a value of 2 standard deviations below the mean, as established by regression analysis. Hardware such as deck fittings, bits, cleats, fairleads, chocks, thimbles and shackles, should have greater strength than the rope itself. Consequently, the hardware should be rated at least 1.3 times the MBS of the rope to be used. The MBS is based on rope being terminated by eye splices.

4. Working Load Limit (WLL)

The Working Load Limit of a rope is determined by dividing the Minimum Breaking Strength (MBS) by the Design Factor. MBS ÷ Design Factor = WLL. Because of the wide variety of rope use, the many factors affecting rope behavior, and the many degrees of risk to life and property that can be encountered, it is not realistic to have a fixed Design Factor or Working Load Limit. To provide guidelines, a range of Design Factors and Working Load Limits are provided in the standards for rope in good condition, with standard splices, and used under normal service conditions. Normal service is considered to be use under modest dynamic loading, not over the WLL. This means that the load must be handled slowly and smoothly to minimize dynamic effects.

Nearly all rope in use is subject to dynamic loading to some degree. Loads over the WLL should be considered as shock loading, which can stress and damage fibers and result in early rope failure, even when handling loads below the WLL. **The load applied to the rope should not exceed the working load limit.**

5. Dynamic Loading

Whenever a load is picked up, stopped, moved or swung, there is an increased force due to dynamic loading. The more rapidly or suddenly such actions occur, the greater this increase will be. In extreme cases, the force put on the rope may be two, three, or even more times the normal load involved. For instance, when picking up a tow on a slack line or using a rope to stop a falling object. Therefore, in all such applications as towing lines, lifelines, safety lines, climbing ropes, etc., Design Factors[13] must reflect the added risks involved.

Users should be aware that dynamic effects are greater on a low-elongation rope, such as manila, than on a high-elongation rope, such as nylon, and greater on a shorter rope than on a longer one. The range of Design Factors reflect provision for modest dynamic loads.

6. Design Factors

Users must determine the Design Factor, as they are the only ones who can assess actual service conditions and establish operating procedures. Design Factors are shown in the standards from 5 to 12 for normal service and modest dynamic loading.

Design factors at the low end of the suggested range should only be selected with expert knowledge of conditions and professional estimate of risk, based on critical conditions of use listed below.

Design Factors at the high end of the range or a larger rope should be selected for critical conditions of use.

Critical Conditions of Use

1. Small ropes are more easily damaged by cutting, abrasion, UV
2. Loads are not accurately known.
3. Operators are poorly trained.
4. Operation procedures are not well defined and/or controlled.
5. Inspection is infrequent.
6. Abrasion, cutting surfaces and dirt are present.
7. Shock loads or extreme dynamic loads are likely to occur.
8. Temperatures higher than fiber limits are present.
9. Hazardous chemicals are present.
10. Ropes are kept in service indefinitely.
11. Tension is on the rope for long periods.
12. Rope will be subject to sharp bends, or used over pulleys or surfaces with too small a radius.
13. Knots are used. Knots reduce rope strength by up to 50% in industrial fiber ropes, up to 75 percent in high performance fiber rope.
14. Death, injury, or loss of valuable property may result if rope fails

7. Snapback Safety Warning

A dangerous situation occurs if personnel are in line with a rope under excessive tension. Should the rope fail, it may recoil with considerable force. Death may result. Persons must be warned against standing in line with the rope or in its bight.

8. Special Applications

The Design Factor ranges can be extended in applications where a thorough engineering analysis of all conditions of use has been made by qualified professionals. In such cases, breaking strength, elongation, energy absorption, cyclic loading and other factors, including operating procedures, have been evaluated to allow the selection of a Design Factor best suited to the actual conditions of use.

[13] Design factor = rope strength/maximum working load

Table 3-5

CI 1301, Polypropylene Fiber Rope January 2001

3-Strand Twisted and 8-Strand Plated Construction

Physical Properties and Working Load Limits (WLL)

Nominal[(1)] Diameter		Size[(2)] No.	Linear Density[(3)]		Min Breaking Strength[(4)]		Working Load Limit (WLL) for Design Factor[(5)] of:			
							12		**5**	
Inch	(mm)		Lbs/100 ft	(ktex)	Lbs.	(daN)	lbs	(daN)	lbs	(DaN)
3/16	(5)	5/8	0.65	(9.7)	650	(289)	54	(24)	130	(58)
1/4	(6)	3/4	1.15	(17.1)	1,125	(500)	94	(42)	225	(100)
5/16	(8)	1	1.80	(26.8)	1,710	(761)	143	(63)	342	(152)
3/8	(10)	1-1/8	2.60	(38.7)	2,430	(1,081)	203	(90)	486	(216)
7/16	(11)	1-1/4	3.50	(52.1)	3,150	(1,401)	263	(117)	630	(280)
1/2	(12)	1-1/2	4.60	(68.5)	3,780	(1,681)	315	(140)	756	(336)
9/16	(14)	1-3/4	5.90	(87.8)	4,590	(2,042)	383	(170)	918	(408)
5/8	(16)	2	7.20	(107)	5,580	(2,482)	465	(207)	1,116	(496)
3/4	(18)	2-1/4	10.4	(155)	7,650	(3,403)	638	(284)	1,530	(681)
7/8	(22)	2-3/4	14.2	(211)	10,350	(4,604)	863	(384)	2,070	(921)
1.0	(24)	3	18.0	(268)	12,825	(5,705)	1,069	(475)	2,565	(1,141)
1-1/16	(26)	3-1/4	20.9	(304)	14,400	(6,405)	1,200	(534)	2,880	(1,281)
1-1/8	(28)	3-1/2	22.8	(339)	16,000	(7,117)	1,333	(593)	3,200	(1,423)
1-1/4	(30)	3-3/4	27.6	(411)	19,350	(8,607)	1,613	(717)	3,870	(1,721)
1-5/16	(32)	4	30.4	(452)	21,150	(9,408)	1,763	(784)	4,230	(1,882)
1-1/2	(36)	4-1/2	39.4	(586)	27,350	(12,165)	2,279	(1,014)	5,470	(2,433)
1-5/8	(40)	5	46.0	(685)	31,950	(14,211)	2,663	(1,184)	6,390	(2,842)
1-3/4	(44)	5-1/2	53.0	(789)	36,900	(16,413)	3,075	(1,368)	7,380	(3,823)
2.0	(48)	6	69.0	(1,027)	46,800	(20,817)	3,900	(1,735)	9,360	(4,163)
2-1/8	(52)	6-1/2	78	(1,161)	52,650	(23,419)	4,388	(1,952)	10,530	(4,684)
2-1/4	(56)	7	88	(1,310)	59,400	(26,421)	4,950	(2,202)	11,880	(5,284)
2-1/ 2	(60)	7-1/2	107.	(1,592)	72,000	(32,026)	6,000	(2,669)	14,400	(6,405)
2-5/8	(64)	8	120.	(1,786)	80,500	(35,806)	6,708	(2,984)	16,100	(7,161)
2-3/4	(68)	8-1/2	141.	(2,098)	94,500	(42,034)	7,875	(3,503)	18,900	(8,407)
3	(72)	9	153.	(2,277)	102,600	(45,636)	8,550	(3,803)	20,520	(9,127)
3-3/4	(80)	10	186.	(2,768)	121,500	(54,043)	10,125	(4,504)	24,300	(10,809)
3-1/2	(88)	11	223.	(3,319)	144,000	(64,051)	12,000	(5,338)	28,800	(12,810)
4	(96)	12	272.	(4,048)	171,900	(76,461)	14,325	(6,372)	34,380	(15,292)
4-1/4	(104)	13	315	(4,688)	198,000	(88,070)	16,500	(7,339)	39,600	(17,614)
4-1/2	(112)	14	360	(5,357)	223,200	(99,279)	18,600	(8,273)	44,640	(19,856)
5	(120)	15	420	(6,250)	256,500	(114,091)	21,375	(9,508)	51,300	(22,818)
5-5/16	(128)	16	517	(7,054)	287,100	(127,702)	23,925	(10,642)	57,420	(25,540)
5-5/8	(136)	17	531.	(7,902)	319,500	(142,114)	26,625	(11,843)	63,900	(28,423)
6	(144)	18	603	(8,974)	358,200	(159,327)	29,850	(13,277)	71,640	(31,865)

(1) Diameters are approximate, reference for a rope size is its linear density or weight per unit length.

(2) Size is the approximate rope circumference in inches

(3) Linear Density is considered standard. Tolerances are: ±10% for diameters 3/16" – 5/16" inclusive; ±8% for diameters 3/8"- 9/16" inclusive; ±5% for 5/8" diameter and up. The SI unit of linear density is Ktex (kilotex). 1 kilotex = 1 gram per meter

(4) New rope Minimum Breaking Strength was based on data from a number of manufacturers and represents a value of 2 standard deviations below the mean, established by regression analysis

(5) For Design Factor selection , refer to Table 3-4, *CI 1401 Safe Use Guidelines*. Working Load Limit is determined by dividing the new rope Minimum Breaking Strength by the selected Design Factor.

Table 3-6

Polyester/Polyolefin Fiber Rope Standard; CI 1302A- January 2001

3- Strand Twisted Construction

Physical Properties and Working Load Limits (WLL)

Nominal[(1)] Diameter		Size[(2)] No.	Linear Density[(3)]		Min Breaking Strength[(4)]		Working Load Limit (WLL) for Design Factor[(5)] of:			
							12		**5**	
Inch	(mm)		Lbs/100 ft	(ktex)	Lbs.	(daN)	lbs	(daN)	lbs	(DaN)
3/16	(5)	5/8	0.65	(9.7)	650	(289)	54	(24)	130	(58)
1/4	(6)	3/4	1.15	(17.1)	1,125	(500)	94	(42)	225	(100)
5/16	(8)	1	1.80	(26.8)	1,710	(761)	143	(63)	342	(152)
3/8	(10)	1-1/8	2.60	(38.7)	2,430	(1,081)	203	(90)	486	(216)
7/16	(11)	1-1/4	3.50	(52.1)	3,150	(1,401)	263	(117)	630	(280)
1/2	(12)	1-1/2	4.60	(68.5)	3,780	(1,681)	315	(140)	756	(336)
9/16	(14)	1-3/4	5.90	(87.8)	4,590	(2,042)	383	(170)	918	(408)
5/8	(16)	2	7.20	(107)	5,580	(2,482)	465	(207)	1,116	(496)
3/4	(18)	2-1/4	10.4	(155)	7,650	(3,403)	638	(284)	1,530	(681)
7/8	(22)	2-3/4	14.2	(211)	10,350	(4,604)	863	(384)	2,070	(921)
1	(24)	3	18.0	(268)	12,825	(5,705)-	1,069	(475)	2,565	(1,141)
1-1/16	(26)	3-1/4	20.4	(304)	14,400	(6,405)	1,200	(534)	2,880	(1,281)
1-1/8	(28)	3-1/2	22.8	(339)	16,000	(7,117)	1,333	(593)	3,200	(1,423)
1-1/4	(30)	3-3/4	27.6	(411)	19,350	(8,607)	1,613	(717)	3,870	(1,721)
1-5/16	(32)	4	30.4	(452)	21,150	(9,408)	1,763	(784)	4,230	(1,882)
1-1/2	(36)	4-1/2	39.4	(586)	27,350	(12,165)	2,279	(1,014)	5,470	(2,433)
1-5/8	(40)	5	46.0	(685)	31,950	(14,211)	2,663	(1,184)	6,390	(2,842)
1-3/4	(44)	5-1/2	53.0	(789)	36,900	(16,413)	3,075	(1,368)	7,380	(3,823)
2	(48)	6	69.0	(1,027)	46,800	(20,817)	3,900	(1,735)	9,360	(4,163)
2-1/8	(52)	6-1/2	78.0	(1,161)	52,650	(23,419)	4,388	(1,952)	10,530	(4,684)
2-1/4	(56)	7	89.0	(1,310)	59,400	(26,421)	4,950	(2,202)	11,880	(5,284)
2-1/2	(60)	7-1/2	108	(1,592)	72,000	(32,026)	6,000	(2,669)	14,400	(6,405)
2-5/8	(64)	8	121	(1,786)	80,500	(35,806)	6,708	(2,984)	16,100	(7,161)
2-3/4	(68)	8-1/2	142	(2,098)	94,500	(42,034)	7,875	(3,503)	18,900	(8,407)
3	(72)	9	153	(2,277)	102,600	(45,636)	8,550	(3,803)	20,520	(9,127)
3-3/4	(80)	10	187	(2,768)	121,500	(54,043)	10,125	(4,504)	24,300	(10,809)
3-1/2	(88)	11	224	(3,319)	144,000	(64,051)	12,000	(5,338)	28,800	(12,810)
4	(96)	12	272	(4,048)	171,900	(76,461)	14,325	(6,372)	34,380	(15,292)
4-1/4	104)	13	315	(4,688)	198,000	(88,070)	16,500	(7,339)	39,600	(17,614)
4-1/2	(112)	14	360	(5,357)	223,200	(99,279)	18,600	(8,273)	44,640	(19,856)
5	(120)	15	420	(6,250)	256,500	(114,091)	21,375	(9,508)	51,300	(22,818)
5-5/16	(128)	16	474	(7,054)	287,100	(127,702)	23,925	(10,642)	57,420	(25,540)
5-5/8	(136)	17	532	(7,902)	319,500	(142,114)	26,625	(11,843)	63,900	(28,423)
6	(144)	18	603	(8,974)	358,200	(159,327)	29,850	(13,277)	71,640	(31,865)

(1) Diameters are approximate, reference for a rope size is its linear density or weight per unit length.

(2) Size is the approximate rope circumference in inches

(3) Linear Density is considered standard. Tolerances are: ±10% for diameters 3/16" – 5/16" inclusive; ±8% for diameters 3/8"- 9/16" inclusive; ±5% for 5/8" diameter and up. The SI unit of linear density is Ktex (kilotex). 1 kilotex = 1 gram per meter

(4) New rope Minimum Breaking Strength was based on data from a number of manufacturers and represents a value of 2 standard deviations below the mean, established by regression analysis.

(5) For Design Factor selection, refer to Table 3-4, *CI 1401 Safe Use Guidelines*. Working Load Limit is determined by dividing the new rope Minimum Breaking Strength by the selected Design Factor.

(6) Rope strands have an outer layer of Polyester for abrasion protection, and a core of Polypropylene.

Table 3-7

Polyester/Polyolefin Fiber Rope Standard; CI 1302B- January 2001
8- Strand Plaited Construction

Physical Properties and Working Load Limits (WLL

Nominal(1) Diameter Inch (mm)		Size(2) No.	Linear Density(3) Lbs/100 ft (ktex)		Min Breaking Strength(4) Lbs.	(daN)	Working Load Limit (WLL) for Design Factor(5) of: 12 lbs (daN)	5 lbs (DaN)
3/8	(16)	2	9.56	(142)	7,450	(3,314)	621 (276)	1,490 (663
3/4	(18)	2 1/4	13.5	(201)	10,260	(4,564)	855 (380)	2,052 (913)
7/8	(22)	2 3/4	18.1	(270)	13,500	(6,005)	1,125 (500)	2,700 (1,021)
1	(24)	3	23.6	(351)	16,963	(7,545)	1,414 (629)	3,393 (1,509)
1 1/16	(26)	3 1/4	26.3	(392)	18,675	(8,307)	1,556 (692)	3,735 (1,661)
1 1/8	(28)	3 1/2	29.2	(434)	19,945	(8,872)	1,662 (739)	3,989 (1,774)
1 1/4	(30)	3 3/4	35.5	(528)	23,940	(10,649)	1,995 (887)	4,788 (2,130)
1 5/16	(32)	4	38.5	(573)	25,650	(11,409)	2,138 (951)	5,130 (2,282)
1 1/2	(36)	4 1/2	49.7	(740)	32,850	(14,612)	6,570 (2,922)	2,738 (1,218)
1 5/8	(40)	5	57.9	(862)	36,990	(16,453)	3,083 (1,371)	7,398 (3,291)
1 1/8	(44)	5 1/2	65.8	(979)	41,400	(18,415)	3,450 (1,535)	8,280 (3,683)
2	(48)	6	84.2	(1,253)	52,200	(23,219)	4,350 (1,935)	10,440 (4,644)
2 1/8	(52)	6 1/2	94.6	(1,407)	58,500	(26.021)	4,875 (2,168)	11,700 (5,204)
2 1/4	(56)	7	104	(1,554)	64,440	(28,663)	5,370 (2,389)	12,888 (5,733)
2 ½	(60)	7 1/2	128	(1,906)	78,750	(35,028)	6,563 (2,919)	15,750 (7,006)
2 5/8	(64)	8	140	(2,083)	85,680	(38,110)	7,140 (3,176)	17,136 (7,622)
2 7/8	(68)	8 1/2	166	(2,477)	101,430	(45,116)	8,453 (3,760)	20,286 (9,023)
3	(72)	9	179	(2,668)	108,270	(48,158)	9,023 (4,013)	21,654 (9,632)
3 5/16	(80)	10	216	(3,210)	129,780	(57,726)	10,815 (4,811)	25,956 (11,545)
3 5/8	(88)	11	257	(3,826)	153,900	(68,445)	12,825 (5,705)	30,780 (13,691)
4	(96)	12	311	(4,632)	185,580	(82,546)	15,465 (6,879)	37,116 (16,509)
4 5/16	(104)	12	360	(5,350)	213,480	(94,956)	17,790 (7,913)	42,696 (18,991)
4 5/8	(112)	14	411	(6,113)	243,000	(108,086)	20,250 (9,007)	48,600 (21,617)
5	(120)	15	497	(7.095)	281,160	(125,060)	23,430 (10,422)	56,232 (25,012)
5 5/16	(128)	16	535	(7,960)	314,100	(139,712)	26,175 (11,643)	62,820 (27,942)
5 5/8	(136)	17	592	(8,810)	341,010	(151,681)	28,418 (12,640)	68,202 (30,336)
6	(144)	18	665	(9,895)	377,010	(167,694)	31,418 (13,975)	75,402 (33,539)

Notes:

(1) Diameters are approximate, reference for a rope size is its linear density or weight per unit length.
(2) Size is the approximate rope circumference in inches
(3) Linear Density is considered standard. Tolerances are: ±10% for diameters 3/16" – 5/16" inclusive; ±8% for diameters 3/8"- 9/16" inclusive; ±5% for 5/8" diameter and up. The SI unit of linear density is Ktex (kilotex). 1 kilotex = 1 gram per meter
(4) New rope Minimum Breaking Strength was based on data from a number of manufacturers and represents a value of 2 standard deviations below the mean, established by regression analysis.
(5) For Design Factor selection, refer to Table 3-4, *CI 1401 Safe Use Guidelines*. Working Load Limit is determined by dividing the new rope Minimum Breaking Strength by the selected Design Factor.
(6) Rope strands have an outer layer of Polyester for abrasion protection, and a core of Polypropylene

Table 3-8

Nylon (Polyamide) Fiber Rope Standard: CI 1303-01 Review Draft #2 April 2001
3-Strand Twisted and 8-Strand Plaited Constructions

Physical Properties and Working Load Limits (WLL)

Nominal(1) Diameter		Size(2) No.	Linear Density(3)		Min Breaking Strength(4)		Working Load Limit (WLL) for Design Factor(5) of:			
							12		**5**	
Inch	(mm)		Lbs/100 ft	(ktex)	Lbs.	(daN)	lbs	(daN)	lbs	(DaN)
3/16	(5)	5/8	.89	(13.2)	880	(391)	73	(33)	176	(78)
1/4	(6)	3/4	1.57	(23.4)	1,486	(661)	124	(55)	297	(132)
5/16	(8)	1	2.45	(36.5)	2,295	(1,021)	191	(85)	459	(204)
3/8	(10)	1-1/8	3.55	(52.8)	3,240	(1,441)	270	(120)	648	(288)
7/16	(11)	1-1/4	4.8	(71.4)	4,320	(1,922)	360	(160)	864	(384)
1/2	(12)	1-1/2	6.3	(93.8)	5,670	(2,522)	473	(210)	1,134	(504)
9/16	(14)	1-3/4	8.0	(119)	7,200	(3,203)	600	(267)	1,440	(641)
5/8	(16)	2	9.9	(147)	8,910	(3,963)	743	(330)	1,782	(793)
3/4	(18)	2-1/4	14.3	(213)	12,780	(5,685)	1,065	(474)	2,556	(1,137)
7/8	(22)	2-3/4	19.5	(290)	17,280	(7,686)	1,440	(641)	3,456	(1,537)
1	(24)	3	25.3	(377)	22,230	(9,888)	1,853	(824)	4,446	(1,978)
1-1/16	(26)	3-1/4	28.7	(427)	25,200	(11,209)	2,100	(934)	5,040	(2,242)
1-1/8	(28)	3-1/2	32.2	(479)	28,260	(12,570)	2,355	(1,048)	5,652	(2,514)
1-1/4	(30)	3-3/4	39.7	(591)	34,830	(15,492)	2,903	(1,291)	6,966	(3,098)
1-5/16	(32)	4	43.0	(650)	38,250	(17,041)	3,188	(1,418)	7,650	(3,403)
1-1/2	(36)	4-1/2	57.0	(848)	48,600	(21,617)	4,050	(1,801)	9,720	(4,323)
1-5/8	(40)	5	67.3	(1,002)	57,375	(25,520)	4,781	(2,127)	11,475	(5,104)
1-3/4	(44)	5-1/2	78.0	(1,161)	66,150	(29,424)	5,513	(2,452)	13,230	(5,885)
2	(48)	6	100	(1,488)	84,600	(37,630)	7,050	(3,136)	16,920	(7,526)
2-1/8	(52)	6-1/2	113	(1,682)	95,400	(42,434)	7,950	(3,536)	19,080	(8,487)
2 -1/4	(56)	7	127	(1,890)	107,100	(47,638)	8,925	(3,970)	21,420	(9,936)
2-1/ 2	(60)	7-1/2	157	(2,336)	131,400	(58,447)	10,950	(4,871)	26,280	(11,689)
2-5/8	(64)	8	173	(2,575)	144,000	(64,051)	12,000	(5,338)	28,800	(12,810)
2-3/4	(68)	8-1/2	208	(3,095)	171,000	(76,061)	14,250	(6,338)	34,200	(15,212)
3	(72)	9	226	(3,363)	185,400	(82,466)	15,450	(6,872)	37,080	(16,493)
3-3/4	(80)	10	275	(4,093)	224,100	(99,680)	18,675	(8,307)	44,820	(19,936)
3-1/2	(88)	11	329	(4,896)	267,300	(118,895)	22,275	(9,908)	53,460	(23,779)
4	(96)	12	400	(5,953)	324,000	(144,115)	27,000	(12,010)	64,800	(28,823)
4-1/4	(104)	3	460	(6,846)	369,000	(164,131)	30,750	(13,678)	73,800	(32,826)
4-1/2	(112)	14	525	(7,813)	418,500	(186,149)	34,875	(15,512)	83,700	(37,230)
5	(144)	15	610	(9,078)	480,600	(213,771)	40,050	(17,814)	96,120	(42,754)
5-5/16	(128)	16	685	(10,194)	532,800	(236,989)	44,400	(19,749)	106,560	(47,398)
5-5/8	(136)	17	767	(11,414)	589,500	(262,210)	49,125	(21,851)	117,900	(52,442)
6	(144)	18	870	(12,947)	660,600	(293,835)	55,050	(24,486)	132,120	(58,767)

Notes:

(1) Diameters are approximate, reference for a rope size is its linear density or weight per unit length.

(2) Size is the approximate rope circumference in inches

(3) Linear Density is considered standard. Tolerances are: ±10% for diameters 3/16" – 5/16" inclusive; ±8% for diameters 3/8"- 9/16" inclusive; ±5% for 5/8" diameter and up. The SI unit of linear density is Ktex (kilotex). 1 kilotex = 1 gram per meter

(4) New rope Minimum Breaking Strength was based on data from a number of manufacturers and represents a value of 2 standard deviations below the mean, established by regression analysis.

(5) For Design Factor selection, refer to Table 3-4, *CI 1401 Safe Use Guidelines*. Working Load Limit is determined by dividing the new rope Minimum Breaking Strength by the selected Design Factor.

Table 3-9

Polyester (PET) Fiber Rope Standard: CI 1304-01 Review Draft #2 Jan. 2001
3-Strand Twisted and 8-Strand Plaited Constructions

Physical Properties and Working Load Limits (WLL)

Nominal[1] Diameter	Size[2] No.	Linear Density[3]	Min Breaking Strength[4]		Working Load Limit (WLL) for Design Factor[5] of:	
					12	5
Inch (mm)		Lbs/100 ft (ktex)	Lbs.	(daN)	lbs (daN)	lbs (DaN)
3/16 (5)	5/8	1.10 (16.4)	765	(340)	64 (28)	153 (68)
1/4 (6)	3/4	1.95 (29.0)	1,315	(585)	110 (49)	263 (117)
5/16 (8)	1	3.05 (45.4)	2,050	(912)	171 (76)	410 (182)
3/8 (10)	1-1/8	4.35 (64.7)	2,900	(1,290)	242 (107)	580 (258)
7/16 (11)	1-1/4	5.9 (87.8)	3,915	(1,742)	326 (145)	783 (348)
1/2 (12)	1-1/2	7.7 (115)	5,085	(2,262)	424 (188)	1,017 (452)
9/16 (14)	1-3/4	9.8 (146)	6,435	(2,862)	536 (239)	1,287 (572)
5/8 (16)	2	12.0 (179)	7,825	(3,481)	652 (290)	1,565 (696)
3/4 (18)	2-1/4	17.2 (256)	11,200	(4,982)	933 (415)	2,240 (996)
7/8 (22)	2-3/4	23.4 (348)	15,225	(6,772)	1,269 (564)	3,045 (1.354)
1 (24)	3	30.4 (452)	19,775	(8,796)	1,648 (733)	3,955 (1,759)
1-1/16 (26)	3-1/4	34.2 (509)	22,225	(9,868)	1,852 (824)	4,445 (1,977)
1-1/8 (28)	3-1/2	38.5 (573)	24,800	(11,031)	2,067 (919)	4,960 (2,206)
1-1/4 (30)	3-3/4	46.5 (692)	29,800	(13,255)	2,483 (1,105)	5,960 (2,651)
1-5/16 (32)	4	51.0 (759)	32,500	(14,456)	2,708 (1,205)	6,500 (2,891)
1-1/2 (36)	4-1/2	67.0 (997)	42,200	(18,771)	3,517 (1,564)	8,440 (3,754)
1-5/8 (40)	5	78.0 (1,161)	49,250	(21,906)	4,104 (1,826)	9,850 (4,381)
1-3/4 (44)	5-1/2	91.0 (1,354)	57,000	(25,354)	4,750 (2,113)	11,400 (5071)
2 (48)	6	117 (1,741)	72,000	(32,026)	6,000 (2,669)	14,400 (6,405)
2-1/8 (52)	6-1/2	133 (1,979)	81,000	(36,029)	6,750 (3,002)	16,200 (7,206)
2 -1/4 (56)	7	149 (2,217)	90,500	(40,254)	7,542 (3,355)	18,100 (8,051)
2-1/ 2 (60)	7-1/2	184 (2,738)	110,000	(48,928)	9,167 (4,077)	22,000 (9,786)
2-5/8 (64)	8	203 (3,021)	121,000	(53,821)	10,083 (4,484)	24,200 (10,764)
2-3/4 (68)	8-1/2	243 (3,616)	144,000	(64,051)	12,000 (5,338)	28,800 (12,810)
3 (72)	9	264 (3,929)	156,000	(69,389)	13,000 (5,782)	31,200 (13,878)
3-3/4 (80)	10	323 (4,807)	188,500	(83,845)	15,708 (6,987)	37,700 (16,769)
3-1/2 (88)	11	387 (5,759)	225,000	(100,080)	18,750 (8,340)	45,000 (20,016)
4 (96)	12	470 (6,994)	270,000	(120,096)	22,500 (10,008)	54,000 (24,019)
4-1/4 (104)	3	547 (8,140)	310,000	(137,888)	25,833 (11,491)	62,000 (27,578)
4-1/2 (112)	14	630 (9,376)	355,000	(157,904)	29,583 (13,159)	71,000 (31,581)
5 (144)	15	732 (10,894)	410,000	(182,368)	34,167 (15,197)	82,000 (36,474)
5-5/16 (128)	16	825 (12,278)	459,000	(204,163)	38,250 (17,014)	91,800 (40,833)
5-5/8 (136)	17	925 (13,766)	508,500	(236,181)	42,375 (18,848)	101,700 (45,236)
6 (144)	18	1,050 (15,626)	567,000	(252,202)	47,250 (21,017)	113,400 (50,440)

Notes:

(1) Diameters are approximate, reference for a rope size is its linear density or weight per unit length.

(2) Size is the approximate rope circumference in inches

(3) Linear Density is considered standard. Tolerances are: ±10% for diameters 3/16” – 5/16” inclusive; ±8% for diameters 3/8”- 9/16” inclusive; ±5% for 5/8” diameter and up. The SI unit of linear density is Ktex (kilotex). 1 kilotex = 1 gram per meter

(4) New rope Minimum Breaking Strength was based on data from a number of manufacturers and represents a value of 2 standard deviations below the mean, established by regression analysis.

(5) For Design Factor selection, refer to Table 3-4, *CI 1401 Safe Use Guidelines*. Working Load Limit is determined by dividing the new rope Minimum Breaking Strength by the selected Design Factor

Table 3-10

Polyester (PET) Fiber Rope Standard: CI 1305 Draft Jan. 2001
12-Strand Braid Construction

Physical Properties and Working Load Limits (WLL)

Nominal(1) Diameter		Size(2) No.	Linear Density(3)		Min Breaking Strength(4)		Working Load Limit (WLL) for Design Factor(5) of:			
							12		5	
Inch	(mm)		Lbs/100 ft	(ktex)	Lbs.	(daN)	lbs	(daN)	lbs	(DaN)
3/8	(10)	1-1/8	4.35	64.8	3,870	1,721	323	143	744	344
7/16	(11)	1-1/4	5.9	87.9	5,175	2,302	431	192	1,035	460
1/2	(12)	1-1/2	7.7	225	6,700	2,980	558	248	1,340	596
9/16	(14)	1-3/4	9.8	146	8,350	3,714	696	310	1,670	743
5/8	(16)	2	12.1	180	10,375	4,615	865	385	2,075	923
3/4	(18)	2-1/4	17.4	259	14,850	6,605	1,238	550	2,970	1,321
7/8	(22)	2-3/4	23.6	352	19,800	8,807	1,650	734	3,960	1,761
1	(24)	3	30.8	459	25,650	11,409	2,138	951	5,130	2,282
1-1/16	(26)	3-1/4	34.8	518	28,980	12,890	2,415	1,074	5,796	2,578
1-1/8	(28)	3-1/2	39.0	181	32,400	14,412	2,700	1,201	6,480	2,882
1-1/4	(30)	3-3/4	48.0	715	39,600	17,614	3,300	1,468	7,920	3,523
1-5/16	(32)	4	52.8	786	42,850	19,060	3,571	1,588	8,570	3,812
1-1/2	(36)	4-1/2	68.8	1,025	55,350	24,620	4,613	2,052	11,070	4,924
1-5/8	(40)	5	80.5	1,199	64,450	28,667	5,371	2,389	22,890	5,733
1-3/4	(44)	5-1/2	93.0	1,385	74,150	32,982	6,179	2,748	14,830	6,596
2	(48)	6	121	1,802	96,300	42,834	8,025	3,570	19.260	8,567
2-1/8	(52)	6-1/2	136	2,026	108,000	48,038	9,000	4,003	21,600	9,608
2 -1/4	(56)	7	152	2,264	120,600	53,643	10,050	4,470	24,120	10,729
2-1/ 2	(60)	7-1/2	188	2,800	148,950	66,253	12,413	5,521	29,790	13,251
2-5/8	(64)	8	207	3,083	163,800	72,858	13,650	6,072	32,760	14,572
2-3/4	(68)	8-1/2	248	3.694	195,300	86,869	16,275	7,239]	39,060	17,374
3	(72)	9	270	4,022	212,400	94,476	17,700	7,873	42,480	18,895
3-3/4	(80)	10	329	4,900	256,500	114,090	21,375	9,508	51,300	22,818
3-1/2	(88)	11	292	5,839	310,100	135,710	25,425	11,209	61,020	27.142
4	(96)	12	475	7,075	369,000	164,130	30.750	13,678	73,800	32,826
4-1/4	(104)	3	550	8,192	425,700	189,350	35,475	15,779	85,140	37,870
4-1/2	(112)	14	631	9,399	486,000	216,175	40,500	18.014	97.200	43,235
5	(144)	15	737	10,978	567,000	252,200	47,250	21,017	113,400	50,440
5-5/16	(128)	16	831	12,378	635,400	282,626	52,950	23,552	127,080	56,525
5-5/8	(136)	17	930	13,852	706,500	314,251	58,875	26,188	141,300	62,850
6	(144)	18	1,055	15,714	798,300	355,085	66,525	29,590	159,660	71,017

Notes:

(1) Diameters are approximate, reference for a rope size is its linear density or weight per unit length.

(2) Size is the approximate rope circumference in inches

(3) Linear Density is considered standard. Tolerances are: ±10% for diameters 3/16" – 5/16" inclusive; ±8% for diameters 3/8"- 9/16" inclusive; ±5% for 5/8" diameter and up. The SI unit of linear density is Ktex (kilotex). 1 kilotex = 1 gram per meter

(4) New rope Minimum Breaking Strength was based on data from a number of manufacturers and represents a value of 2 standard deviations below the mean, established by regression analysis.

(5) For Design Factor selection, refer to Table 3-4, *CI 1401 Safe Use Guidelines*. Working Load Limit is determined by dividing the new rope Minimum Breaking Strength by the selected Design Factor

Table 3-11

Nylon (Polyamide) Fiber Rope Standard: CI 1310-01 Review Draft #2 January 2001
Double Braid High Performance Construction

Nominal(1) Diameter	Size(2) No.	Linear Density(3)		Min Breaking Strength(4)		Working Load Limit (WLL) for Design Factor(5) of:			
						12		**5**	
Inch (mm)		Lbs/100 ft	(ktex)	Lbs.	(daN)	lbs	(daN)	lbs	(DaN)
¼ (6)	¾	1.65	24.6	2,000	890	167	74	400	178
5/16 (8)	1.0	2.6	38.7	3,150	1,401	263	117	630	280
3/8 (10)	1-1/8	3.7	55.1	4,400	1,957	367	163	880	391
7/16 (11)	1-1/4	5.1	75.9	6,000	2,669	500	222	1,200	534
1/2 (12)	1-1/2	6.6	98.2	7,800	3,469	650	289	1,500	694
9/16 (14)	1-3/4	8.4	125	9.900	4,404	825	367	1,980	881
5/8 (16)	2	10.4	155	12,200	5,427	1,017	452	2,440	1,085
3/4 (18)	2-1/4	15.0	223	17,350	7,717	1,446	643	3,470	1,543
7/8 (22)	2-3/4	20.4	304	23,400	10,408	1,950	867	4,680	2,082
1 (24)	3	26.6	396	30,250	13,455	2,521	1,121	6,050	2,691
1-1/16 (26)	3-1/4	30.0	446	34,000	15,123	2,833	1,260	6,800	3,025
1-1/8 (28)	3-1/2	33.6	500	37,800	16,813	3,150	1,401	7,560	3,363
1-1/4 (30)	3-3/4	41.5	618	46,450	20,661	3,871	1,722	9,290	4,132
1-5/16 (32)	4	45.7	680	51,000	22,685	4,250	1,890	10,200	4,537
1-1/2 (36)	4-1/2	59.7	888	66,000	29,357	5,500	2,446	13,200	5,971
1-5/8 (40)	5	70/0	1,042	77,300	34,383	6,442	2,865	15,460	6,877
1-3/4 (44)	5-1/2	81.0	1,205	89,300	39,721	7,442	3,310	17,860	7,944
2 (48)	6	106	1,577	116,300	51,730	9.692	4,311	23,260	10,346
2-1/8 (52)	6-1/2	120	1,786	130.700	58,135	10,892	4,845	26,140	11,627
2 -1/4 (56)	7	143	1,994	145,800	64,852	12,150	5,404	29,160	12,970
2-1/ 2 (60)	7-1/2	165	2,456	179,300	79.753	14,942	6,646	35,860	15,951
2-5/8 (64)	8	181	2,694	196,600	87,448	16,383	7,287	39,320	17,490
2-3/4 (68)	8-1/2	217	3,229	232,600	103.460	19,383	8,622	46,520	20,692
3 (72)	9	237	3,572	251,700	111,956	20,975	9,330	50,340	22,391
3-3/4 (80)	10	288	4,286	299,700	133,307	24,975	11,109	59,940	26,661
3-1/2 (88)	11	345	5,134	357,300	158.927	29,775	13,244	71,460	31,785
4 (96)	12	420	6,250	427,700	190.241	35,642	15,853	85,540	38,048
4-1/4 (104)	3	488	7,262	494.600	219,998	41,217	18,833	98.920	44,000
4-1/2 (112)	14	561	8,349	558,400	248,376	46,533	20,698	111,680	49,675
5 (144)	15	656	9,763	648,000	288,230	54,000	24,019	129,600	57,646

Notes:

(1) Diameters are approximate, reference for a rope size is its linear density or weight per unit length.

(2) Size is the approximate rope circumference in inches

(3) Linear Density is considered standard. Tolerances are: ±10% for diameters 3/16" – 5/16" inclusive; ±8% for diameters 3/8"- 9/16" inclusive; ±5% for 5/8" diameter and up. The SI unit of linear density is Ktex (kilotex). 1 kilotex = 1 gram per meter

(4) New rope Minimum Breaking Strength was based on data from a number of manufacturers and represents a value of 2 standard deviations below the mean, established by regression analysis.

(5) For Design Factor selection, refer to Table 3-4, *CI 1401 Safe Use Guidelines*. Working Load Limit is determined by dividing the new rope Minimum Breaking Strength by the selected Design Factor

Table 3-12

Polyester (Pet) Fiber Rope Standard: CI 1311-01 Review Draft #1 January 2001

Double Braid High Performance Construction

Nominal[(1)] Diameter Inch (mm)	Size[(2)] No.	Linear Density[(3)] Lbs/100 ft	(ktex)	Min Breaking Strength[(4)] Lbs.	(daN)	Working Load Limit (WLL) for Design Factor[(5)] of: 12 lbs	(daN)	5 lbs	(DaN)
¼ (6)	¾	2.0	29.8	1,935	861	161	72	387	172
5/16 (8)	1.0	3.1	46.1	2,975	1,323	248	110	595	265
3/8 (10)	1-1/8	4.5	67.0	4,275	1,002	356	158	855	380
7/16 (11)	1-1/4	6.1	90.8	5,725	2,546	477	212	1,145	509
1/2 (12)	1-1/2	8.0	119	7,425	3,303	619	275	1,485	661
9/16 (14)	1-3/4	10.1	150	9,225	4,103	769	342	1,845	821
5/8 (16)	2	12.5	186	11,250	5,004	938	417	2,250	1,001
3/4 (18)	2-1/4	17.9	266	16,000	7,117	1,333	593	3,200	1,423
7/8 (22)	2-3/4	24.4	363	21,600	9,608	2,800	801	4,320	1,922
1 (24)	3	31.9	475	28,100	12,499	2,342	1,042	5,620	2,500
1-1/16 (26)	3-1/4	36.0	538	31,700	14,100	2,642	1,175	6,340	2,820
1-1/8 (28)	3-1/2	40.4	601	35,550	15,813	2,951	1,313	7,083	3,151
1-1/4 (30)	3-3/4	49.8	741	43,750	19,460	3,646	1,622	8,750	3,892
1-5/16 (32)	4	55.0	819	48.250	21,462	4,021	1,788	9,650	4,292
1-1/2 (36)	4-1/2	71.8	1,069	62,100	27,622	5,175	2,301	12,420	5,524
1-5/8 (40)	5	84.0	1,250	72,000	32,026	6,000	2,669	14,400	6,405
1-3/4 (44)	5-1/2	97.7	1,454	82,800	36,829	6,900	3,069	16,560	7,366
2 (48)	6	128	1,905	107,550	47,838	8,963	3,987	21,510	9,568
2-1/8 (52)	6-1/2	144	2,143	119,700	53,243	9,975	4,437	23,940	10,649
2 -1/4 (56)	7	1161	2.396	132,300	58,847	11,025	4,904	26,460	11,769
2-1/ 2 (60)	7-1/2	199	2,961	161,100	71,657	13,425	5,071	32,220	14,331
2-5/8 (64)	8	220	3,274	177,300	78,675	14,775	6,572	35,460	15,773
2-3/4 (68)	8-1/2	263	3,914	210,600	93,675	17,550	7,806	42,120	18,735
3 (72)	9	287	4,271	227,700	101,281	18,975	8,440	45,540	20,256
3-3/4 (80)	10	350	5,209	275,400	122,498	22,950	10,208	55,080	24,500
3-1/2 (88)	11	419	6,235	324,900	144,516	27,075	12,043	64,980	28,903
4 (96)	12	510	7,590	393,300	174,940	32,775	14,578	78,660	34,988
4-1/4 (104)	3	593	8,825	450,000	200,160	37,500	16,680	90,000	40,032
4-1/2 (112)	14	682	10,149	510,000	226,981	42,525	18,915	102,060	45,396
5 (144)	15	798	11,878	594,000	264,211	49,500	22,018	118,800	52,842

Notes:

(1) Diameters are approximate, reference for a rope size is its linear density or weight per unit length.

(2) Size is the approximate rope circumference in inches

(3) Linear Density is considered standard. Tolerances are: ±10% for diameters 3/16" – 5/16" inclusive; ±8% for diameters 3/8"- 9/16" inclusive; ±5% for 5/8" diameter and up. The SI unit of linear density is Ktex (kilotex). 1 kilotex = 1 gram per meter

(4) New rope Minimum Breaking Strength was based on data from a number of manufacturers and represents a value of 2 standard deviations below the mean, established by regression analysis.

(5) For Design Factor selection, refer to Table 3-4, *CI 1401 Safe Use Guidelines*. Working Load Limit is determined by dividing the new rope Minimum Breaking Strength by the selected Design Factor

Table 3-13

CI 1312 – 12-Strand Braided Nylon (Polyamide) Standard

Physical Properties and Working Load Limits (WLL)

Nominal(1) Diameter	Size(2) No.	Linear Density(3)		Min Breaking Strength(4)		Working Load Limit (WLL) for Design Factor(5) of:			
						12		5	
Inch (mm)		Lbs/100 ft	(ktex)	Lbs.	(daN)	lbs	(daN)	lbs	(DaN)
3/8 (10)	1-1/8	3.6	64.8	4,175	1,857	348	155	835	371
7/16 (11)	1-1/4	4.9	87.9	5,650	2,414	417	210	1,130	503
1/2 (12)	1-1/2	6.4	115	7,350	3,271	613	273	1,470	654
9/16 (14)	1-3/4	8.1	146	9,330	4,125	778	346	1,866	830
5/8 (16)	2	10.0	180	11,500	5,118	958	426	2,300	1,024
3/4 (18)	2-1/4	14.4	259	16,500	7,343	1,375	612	3,300	1,469
7/8 (22)	2-3/4	19.5	352	22,275	9,912	1,856	826	4,455	1,982
1 (24)	3	25.4	459	28,750	12,794	2,396	1,066	5,750	2,559
1-1/16 (26)	3-1/4	28.7	518	32,000	14,240	2,667	1,187	6,400	2,848
1-1/8 (28)	3-1/2	32.2	581	35,875	15,964	2,990	1,330	7,175	3,193
1-1/4 (30)	3-3/4	39.7	715	43,775	19,480	3,650	1,623	8,755	3,896
1-5/16 (32)	4	43.6	786	47,500	21,138	3,958	1,761	9,500	4,228
1-1/2 (36)	4-1/2	56.8	1.025	61,400	27,323	5,117	2,277	12,280	5,465
1-5/8 (40)	5	66.5	1,199	71,300	31,729	5,942	2,644	14,260	6,346
1-3/4 (44)	5-1/2	76.8	1,365	81,700	36,357	6,808	3,030	16,340	7,271
2 (48)	6	100	1,802	105,600	46,992	8,800	3,916	21,120	9,398
2-1/8 (52)	6-1/2	112	2,026	117,850	52,443	9,821	4,370	23,570	10,489
2 -1/4 (56)	7	126	2,264	131,350	58,451	10,946	4,871	26,270	11,690
2-1/ 2 (60)	7-1/2	155	2,800	162,675	72,390	13,556	6,033	32,535	14,479
2-5/8 (64)	8	171	3,083	177,450	78,965	14,788	6,580	35,490	15,793
2-3/4 (68)	8-1/2	205	3,694	210,000	93,450	17,500	7,788	42,000	18,690
3 (72)	9	223	4,022	226,100	100,615	18,842	8,385	45,220	20,123
3-3/4 (80)	10	272	4,900	267,000	118,815	22,250	9,901	53,400	23,673
3-1/2 (88)	11	324	5,839	317,600	141,332	26,467	11,778	63520	28,266
4 (96)	12	392	7,075	383,100	170,613	31,950	14,218	76,680	34,123
4-1/4 (104)	3	454	8,192	444,100	197,625	37,008	16,469	88,820	39,525
4-1/2 (112)	14	521	9,399	502.500	223,613	41,875	18,634	100.500	44,723
5 (144)	15	609	10,978	583,200	259,524	48,600	21,627	116,640	51,905
5-5/16 (128)	16	687	12,378	655,200	291,564	54,600	24,297	131,040	58,413
5-5/8 (136)	17	770	13,852	727,200	323,604	60,600	26,967	145,440	64,721
6 (144)	18	875	15,714	818,100	364,055	68,175	30,338	163.620	72,811

Notes:

(1) Diameters are approximate, reference for a rope size is its linear density or weight per unit length.

(2) Size is the approximate rope circumference in inches

(3) Linear Density is considered standard. Tolerances are: ±10% for diameters 3/16" – 5/16" inclusive; ±8% for diameters 3/8"- 9/16" inclusive; ±5% for 5/8" diameter and up. The SI unit of linear density is Ktex (kilotex). 1 kilotex = 1 gram per meter

(4) New rope Minimum Breaking Strength was based on data from a number of manufacturers and represents a value of 2 standard deviations below the mean, established by regression analysis.

(5) For Design Factor selection, refer to Table 3-4, *CI 1401 Safe Use Guidelines*. Working Load Limit is determined by dividing the new rope Minimum Breaking Strength by the selected Design Factor

Table 3-14

High Performance (Co-Polymer) Polypropylene-Polyethylene Fiber Rope Cl 1900

3-Strand Twisted, and 8-Strand Plaited Construction January 2001

Physical Properties and Working Load Limits (WLL)

Nominal[1] Diameter	Size[2] No.	Linear Density[3]		Min Breaking Strength[4]		Working Load Limit (WLL) for Design Factor[5] of:			
						12		5	
Inch (mm)		Lbs/100 ft	(ktex)	Lbs.	(daN)	lbs	(daN)	lbs	(DaN)
3/16 (5)	9/16	.68	10.1	880	391	73	33	176	78
1 /4 (6)	3 /4	1.20	17.8	1,520	676	127	56	304	138
5/16 (8)	1	1.87	27.9	2,300	1,023	192	85	460	205
3/8 (10)	1-1/8	2.70	40.2	3,280	1,459	273	122	656	292
7/16 (11)	1-1/4	3.64	54.2	4,250	1,890	364	158	850	378
1/2 (12)	1-1/2	4.78	71.2	5,100	2,268	425	189	1,020	454
9/16 (14)	1-3/4	6.14	91.3	6,200	2,758	517	230	1,240	552
5/8 (16)	2	7.5	111	7,530	3,349	628	279	1,506	670
3/4 (18)	2-1/4	10.8	161	10,300	4,581	858	382	2,060	916
7/8 (22)	2-3/4	14.8	220	14,000	6,227	1,167	519	2,800	1,245
1 (24)	3	18.7	279	17,300	7,695	1,442	641	3,460	1,539
1-1/16 (26)	3-1/4	21.2	316	19,400	8,629	1,617	719	3,880	1,726
1-1/8 (28)	3-1/2	23.7	353	21,600	9,608	1,800	801	4,320	1,922
1-1/4 (30)	3-3/4	28.7	427	26,100	11,609	2,175	967	5,220	2,322
1-5/16 (32)	4	31.6	471	28,600	12,721	2.383	1,060	5,720	2,544
1-1/2 (36)	4-1/2	41.0	610	36,900	16,413	3,075	1,368	7,380	3,283
1-5/8 (40)	5	47.8	712	43,100	19,171	3,592	1,598	8,620	3,834
1-3/4 (44)	5-1/2	55.1	820	49,800	22,151	4,150	1,846	9,960	4,430
2 (48)	6	71.8	1,068	63,200	28,111	5,267	2,343	12,640	5,622
2-1/8 (52)	6-1/2	81.1	1,207	71,100	31,625	5,925	2,635	14,220	6,325
2 -1/4 (56)	7	91.5	1,362	80,200	35,673	6,683	2,973	16,040	7,135
2-1/ 2 (60)	7-1/2	111.3	1,656	97,200	43,235	8,100	3,603	19,440	8,647
2-5/8 (64)	8	125	1,857	109,000	48,483	9,083	4,040	21,800	9.697
2-3/4 (68)	8-1/2	147	2,182	128,000	56,934	10,667	4,735	25,600	11,387
3 (72)	9	159	2.368	139,000	61,827	11,583	5,152	27,800	12,365
3-3/4 (80)	10	193	2,879	164,000	72,947	13,667	6,079	32,800	14,589
3-1/2 (88)	11	232	3,451	194,000	88,291	16,175	7,191	38,800	17,258
4 (96)	12	283	4,210	232,000	103,194	19,333	8,599	46,400	20,639
4-1/4 (104)	3	328	4,875	267,000	118,762	22,250	9,897	53,400	23,752
4-1/2 (112)	14	374	5,572	301,000	133,885	25,083	11,157	60,200	26,777
5 (144)	15	437	6,500	364,000	153,901	28,833	12,825	69,200	30,780
5-5/16 (128)	16	493	7,336	388,000	172,582	32,333	14,382	77,600	34,416
5-5/8 (136)	17	552	8,218	431,000	191,709	35,917	15,976	86,200	38,342
6 (144)	18	627	9,333	484,000	215,283	40,333	17,940	96,800	43,057

Notes:

(1) Diameters are approximate, reference for a rope size is its linear density or weight per unit length.

(2) Size is the approximate rope circumference in inches

(3) Linear Density is considered standard. Tolerances are: ±10% for diameters 3/16" – 5/16" inclusive; ±8% for diameters 3/8"- 9/16" inclusive; ±5% for 5/8" diameter and up. The SI unit of linear density is Ktex (kilotex). 1 kilotex = 1 gram per meter

(4) New rope Minimum Breaking Strength was based on data from a number of manufacturers and represents a value of 2 standard deviations below the mean, established by regression analysis.

(6) For Design Factor selection, refer to Table 3-4, CI 1401 *Safe Use Guidelines*. Working Load Limit is determined by dividing the new rope Minimum Breaking Strength by the selected Design Factor

Table 3-15

High Performance (Co-Polymer) Polypropylene/Polyethylene Fiber Rope CI 1901

12-Strand Braid Construction

January 2001

Physical Properties and Working Load Limits (WLL)

Nominal(1) Diameter	Size(2) No.	Linear Density(3)		Min Breaking Strength(4)		Working Load Limit (WLL) for Design Factor(5) of:			
						12		**5**	
Inch (mm)		Lbs/100 ft	(ktex)	Lbs.	(daN)	lbs	(daN)	lbs	(DaN)
1 /4 (6)	3 /4	1.22	18	1,630	725	136	60	326	145
5/16 (8)	1	1.91	28	2,480	1,103	207	92	496	221
3/8 (10)	1-1/8	2.76	41	3,500	1,557	292	130	700	311
7/16 (11)	1-1/4	3.71	55	4,600	2,046	383	171	920	409
½ (12)	1-1/2	4.88	73	5,500	2,446	458	204	1,100	489
9/16 (14)	1-3/4	6.25	93	6,700	2,980	558	248	1,340	596
5/8 (16)	2	7.63	114	8,100	3,603	675	300	1,620	721
¾ (18)	2-1/4	11.0	164	11,100	4,937	925	411	2,220	987
7/8 (22)	2-3/4	15,1	224	15,000	6,672	1,250	556	3,000	1,334
1 (24)	3	19,1	284	16,600	8,273	1,550	689	3,720	1,655
1-1/16 (26)	3-1/4	21,6	322	20,900	9,296	1,742	775	4,180	1,859
1-1/8 (28)	3-1/2	24,1	360	23,300	10,319	1,933	860	4,640	2,064
1-1/4 (30)	3-3/4	29,3	435	28,100	12,499	2,342	1,042	5,620	2,500
1-5/16 (32)	4	32,2	480	30,700	13,655	2.558	1,138	6,140	2,731
1-1/2 (36)	4-1/2	41,8	622	39,700	17,659	3,308	1,472	7,940	3,532
1-5/8 (40)	5	48,8	726	46,300	20,594	3,858	1,716	9,260	4,119
1-3/4 (44)	5-1/2	56,2	836	53,500	23,797	4,458	1,983	10,700	4,759
2 (48)	6	73,1	1,088	67,900	30,202	5,658	2,517	13,580	6,040
2-1/8 (52)	6-1/2	82.7	1,239	76,300	33,938	6,358	2,828	15,260	6,788
2 -1/4 (56)	7	93.3	1,388	86,100	38,297	7,175	3,191	17,220	7,659
2-1/ 2 (60)	7-1/2	113	1,688	104,400	46,437	8,700	3,870	20,880	9,287
2-5/8 (64)	8	127	1,893	116,700	51,908	9,725	4,326	23,340	10.382
2-3/4 (68)	8-1/2	149	2,224	137,000	60,938	11,417	5,078	27,400	12,188
3 (72)	9	162	2,414	148,800	66,186	12,400	5,516	29,760	13,237
3-3/4 (80)	10	197	2,934	176,200	78,374	14,683	6,531	35,240	15,675
3-1/2 (88)	11	236	3,518	208,800	92,874	17,400	7,740	41,760	18,575
4 (96)	12	288	4,291	249,300	110,889	20,775	9,241	49,860	22,178
4-5/16 (104)	13	334	4,969	287,100	127,702	23,925	10,642	57,420	25,540
4-5/8 (112)	14	382	5,679	323,600	143,937	26,967	11,995	64,720	28,787
5 (120)	15	445	6,625	372,000	165,466	31,000	13,789	74,400	33,093

Notes:

(1) Diameters are approximate, reference for a rope size is its linear density or weight per unit length.

(2) Size is the approximate rope circumference in inches

(3) Linear Density is considered standard. Tolerances are: ±10% for diameters 3/16" – 5/16" inclusive; ±8% for diameters 3/8"- 9/16" inclusive; ±5% for 5/8" diameter and up. The SI unit of linear density is Ktex (kilotex). 1 kilotex = 1 gram per meter

(4) New rope Minimum Breaking Strength was based on data from a number of manufacturers and represents a value of 2 standard deviations below the mean, established by regression analysis.

(5) For Design Factor selection, refer to Table 3-4, CI 1401 *Safe Use Guidelines*. Working Load Limit is determined by dividing the new rope Minimum Breaking Strength by the selected Design Factor

4.0 CHAIN DATA

ENGINEERING SPECIFICATIONS

CROSBY PROOF COIL - SPECTRUM 3 CHAIN

Trade Size Inches	Size Material Inches	Working Load Limit/lb.	Nominal Inside Length/in.	Nominal Inside Width/in.	Max.(in) Length 100Links	Wt./100 ft.in lbs.
3/16	.218	850	.95	.40	99	40
1/4	.281	1,450	1.00	150	104	73
5/16	.343	2,200	1.10	.50	114	110
3/8	.406	3,050	1.23	.62	128	159
1/2	.531	5,150	1.50	.81	156	275
5/8	.656	7,900	1.87	1.00	194	408
3/4	.781	11,150	2.12	1.12	220	581

CROSBY HIGH TEST - SPECTRUM 4 CHAIN

1/4	.281	2,600	.82	.39	86	77
5/16	.343	3,900	1.01	.48	105	117
3/8	.406	5,400	1.15	.56	121	165
7/16	.468	7,200	1.29	.65	135	220
1/2	.531	9,200	1.43	.75	150	282
5/8	.656	12,750	1.79	.90	186	422
3/4	.781	18,500	1.96	1.06	205	615

CROSBY SPECTRUM 7 CHAIN

1/4	.281	3,600	.82	.39	86	77
5/16	.343	5,400	1.01	.48	105	117
3/8	.406	7,500	1.15	.56	121	165
7/16	.468	10,000	1.29	.65	135	220
1/2	.531	12,750	1.43	.75	150	282
5/8	.656	19,000	1.79	.90	186	422

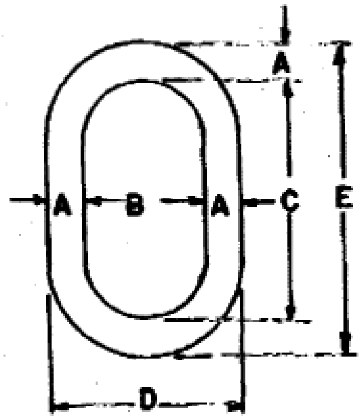
CROSBY ALLOY - SPECTRUM 8 CHAIN

					(Links Per/ft)	
1/4	.280	4,100	.85	.42	14 1/8	75
5/16	.343	5,100	1.00	.47	12	92
3/8	.390	7,300	1.14	.53	10 1/2	145
1/2	.515	13,000	1.43	.69	8 3/8	256
5/8	.640	20,300	1.74	.83	6 7/8	404
3/4	.765	29,300	2.04	.97	5 7/8	575
7/8	.875	39,900	2.32	1.11	5 1/8	730

*Courtesy of Crosby Group

5.0 MARINE HARDWARE DATA

MASTER LINK

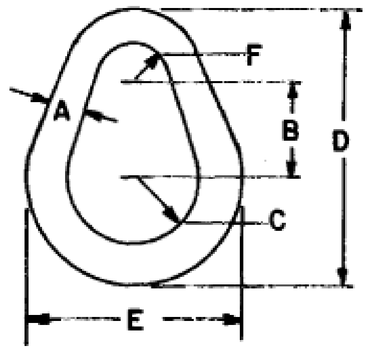


STOCK DIA. A	B	C	D	E	WEIGHT EACH	SAFE LOAD* SINGLE PULL POUNDS
1/2	2.50	5.00	3.25	6.00	.81	3,250
5/8	3.00	6.00	4.25	7.25	1.5	4,400
3/4	2.75	5.50	4.25	7	2	7,000
1	3.50	7.00	5.50	9	4.6	16,500
1 1/4	4.38	8.75	6.88	11.25	9.2	25,000
1 1/2	5.25	10.50	8.25	13.50	15.7	35,500
1 3/4	6.00	12.00	9.50	15.50	24.5	44,500
2	7.00	14.00	11.00	18.00	38.1	57,500
† 2 1/4	8.00	16.00	12.50	20.50	54.8	67,000
† 2 3/4	9.50	16.00	15.00	21.50	87.7	100,000

* Minimum Ultimate Strength Six times safe working load.
† Contact Crosby Group Office on Availability.

Courtesy The Crosby Group

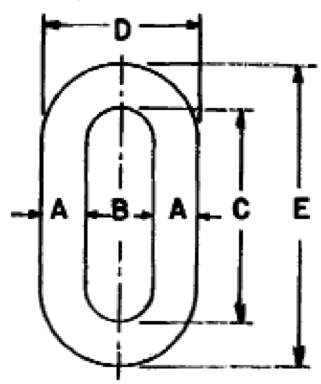
SLING LINK



STOCK DIA. A	B	C	D	E	F	WEIGHT EACH	SAFE LOAD* SINGLE PULL POUNDS
3/8	1.13	.75	3.00	2.25	.38	.23	1,800
1/2	1.50	1.00	4.00	3.00	.50	.53	2,900
5/8	1.875	1.25	5.00	3.75	.63	1.1	4,200
3/4	2.25	1.50	6.00	4.50	.75	1.9	6,000
7/8	2.63	1.75	7.00	5.25	.88	2.9	8,300
1	3.00	2.00	8.00	6.00	1.00	4.3	10,800
1 1/4	4.00	2.50	10.25	7.50	1.25	8.5	16,750
1 3/8	4.13	2.75	11.00	8.25	1.38	11.3	20,500

*Minimum ultimate strength six times safe working load.

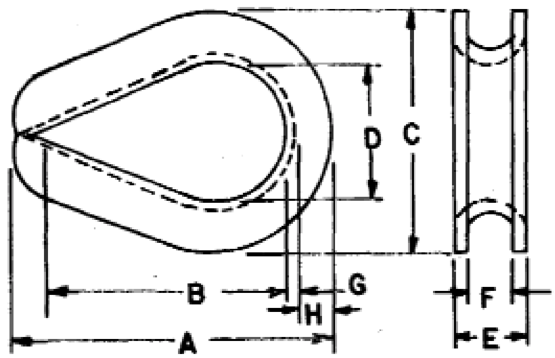
END LINK



STOCK DIA. A	B	C	D	WEIGHT EACH	SAFE LOAD* POUNDS
5/16	.50	1.75	1.13	.14	2,500
3/8	.56	1.88	1.31	.22	3,800
1/2	.75	2.38	1.75	.48	6,500
5/8	1.00	3.25	2.13	.92	9,300
3.4	1.13	3.50	2.63	1.37	14,000
7/8	2.00	5.13	3.75	2.75	12,000
1	2.25	5.75	4.25	3.6	15,200
1 1/4	2.50	7.00	5.00	7	26,400
1 3/8	2.75	7.75	5.50	10	30,000

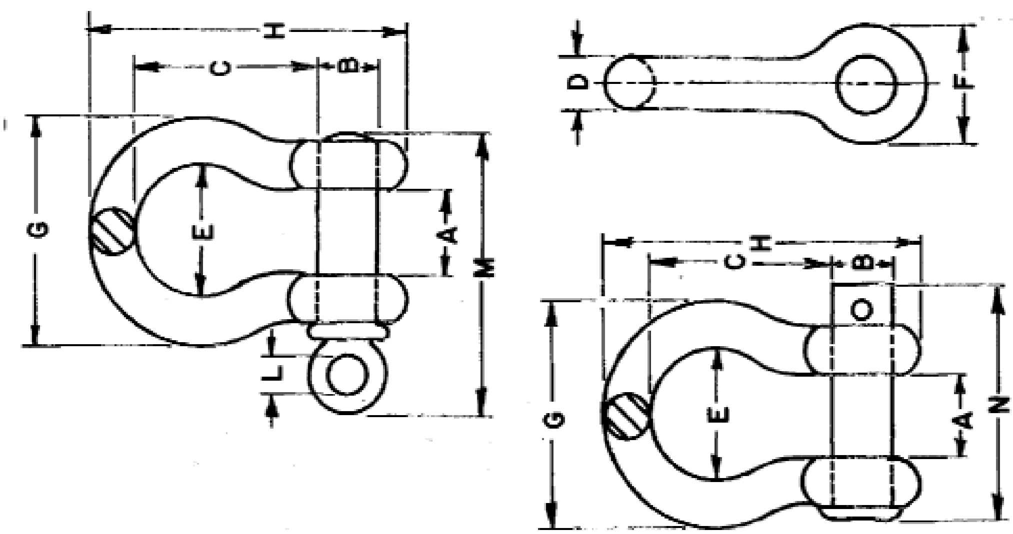
*Ultimate Load Five Times Safe Working Load.

HEAVY WIRE ROPE THIMBLES



ROPE DIA.	A	B	C	D	E	F	G	H	WT.PER 100
1/4	2.19	1.63	1.50	.88	.41	.28	.06	.23	.5
5/16	2.50	1.88	1.81	1.06	.50	.34	.08	.28	14
3/8	2.88	2.13	2.13	1.13	.63	.41	.11	.34	25
7/16	3.25	2.38	2.38	1.25	.72	.47	.13	.38	36
1/2	3.63	2.75	2.75	1.50	.81	.53	.14	.41	51
9/16	3.63	2.75	2.69	1.50	.88	.59	.14	.41	51
5/8	4.25	3.25	3.13	1.75	.97	.66	.16	.50	75
3/4	5.00	3.75	3.81	2.00	1.22	.78	.22	.66	147
7/8	5.50	4.25	4.25	2.25	1.38	.94	.22	.75	185
1	6.13	4.50	4.94	2.50	1.56	1.06	.25	.88	300
1 1/8 - 1 1/4	7.00	5.13	5.88	2.88	1.81	1.31	.25	1.13	410
1 1/4 - 1 3/8	9.06	6.50	6.81	3.50	2.19	1.44	.38	1.13	834
1 3/8 - 1 1/2	9.00	6.25	7.13	3.50	2.56	1.56	.50	1.13	1200
1 5/8	11.25	8.00	8.13	4.00	2.72	1.72	.50	1.38	1625
1 3/4	12.19	9.00	8.50	4.50	2.84	1.84	.50	1.31	1800
1 7/8 - 2	15.13	12.00	10.38	6.00	3.09	2.09	.50	1.50	2600
2 1/4	17.13	14.00	11.88	7.00	3.63	2.38	.63	1.63	4300

*Courtesy The Crosby Group


H
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SCREW PIN ANCHOR SHACKLES

S.W.L. TONS	SIZE D	A	B	C	E	F	G	H	L	M	N	TOLERANCE + or - C	TOLERANCE + or - A	WT. EA. LBS
1/3	.19	.38	.25	.88	.69	.56	.98	1.47	.13	1.13	--	.06	.06	.05
1/2	.25	.50	.31	1.13	.78	.69	1.28	1.88	.16	1.44	1.34	.06	.06	.12
3/4	.31	.53	.38	1.22	.84	.81	1.47	2.13	.19	1.72	1.59	.06	.06	.18
1	.38	.66	.44	1.44	1.03	.97	1.78	2.53	.22	2.06	1.88	.13	.06	.3
1 1/2	.44	.72	.50	1.69	1.16	1.06	2.03	2.91	.25	2.34	2.13	.13	.06	.49
2	.50	.81	.63	1.88	1.31	1.19	2.31	3.28	.31	2.72	2.38	.13	.06	.74
3 1/4	.63	1.06	.75	2.38	1.69	1.56	2.94	4.22	.38	3.41	2.91	.13	.06	1.44
4 3/4	.75	1.25	.88	2.81	2.00	1.88	3.50	5	.44	4.03	3.44	.25	.06	2.16
6 1/2	.88	1.44	1.00	3.31	2.28	2.13	4.03	5.75	.50	4.63	3.84	.25	.06	3.37
8 1/2	1.00	1.69	1.13	3.75	2.69	2.38	4.69	6.50	.56	5.31	4.53	.25	.06	5.3
9 1/2	1.13	1.81	1.25	4.25	2.91	2.63	5.16	7.31	.63	5.88	5.13	.25	.06	7
12	1.25	2.03	1.38	4.69	3.25	3.00	5.75	8.13	.69	6.44	5.50	.25	.06	9.6
13 1/2	1.38	2.25	1.50	5.25	3.63	3.31	6.38	9.03	.75	7.13	6.13	.25	.13	12.6
17	1.50	2.38	1.63	5.75	3.88	3.63	6.88	9.88	.81	7.66	6.50	.25	.13	17.3
25	1.75	2.88	2.00	7.00	5.00	4.31	8.50	11.88	1.00	9.19	7.75	.25	.13	27.8
35	2.00	3.25	2.25	7.75	5.75	5.00	9.75	13.38	1.13	10.34	8.75	.25	.13	41.1
50	2.50	4.13	2.75	10.50	7.25	6.00	12.25	17.38	1.38	12.97	10.94	.75	.13	83.5
75	3.00	5.00	3.25	13.00	7.88	6.50	13.88	20.88	--	--	13.25	.75	.13	119

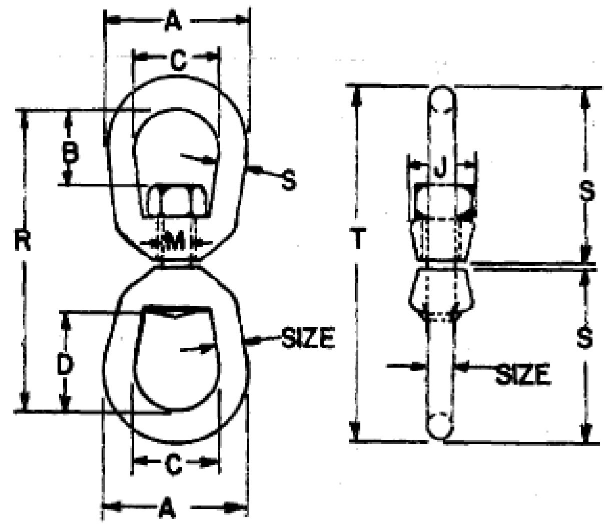
Proof Load 2.2 times safe working load.

Ultimate Strength 6 times safe working load through 50 ton capacity.

Ultimate Strength 5 times safe working load on 75,100 and 130 ton capacity.

*Courtesy The Crosby Group

REGULAR SWIVEL

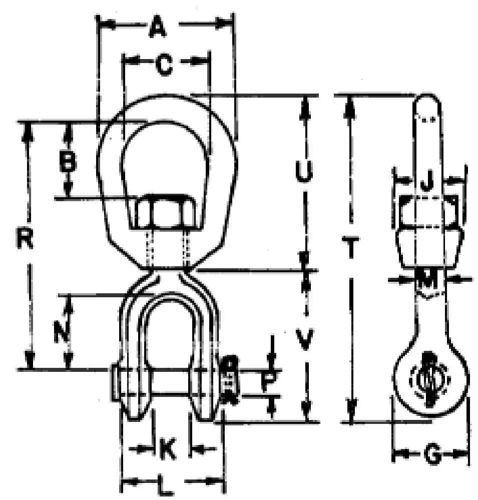


SIZE	SAFE WORK LOAD POUNDS	A	B	C	D	J	M	R	S	T	WEIGHT LBS/EA.
1/4	850	1.25	.69	.75	1.06	.69	.31	2.94	1.69	3.44	.19
5/16	1,250	1.63	.81	1.00	1.25	.81	.38	3.56	2.06	4.19	.31
3/8	2,250	2.00	.94	1.25	1.50	1.00	.50	4.31	2.50	5.06	.68
1/2	3,600	2.50	1.31	1.50	2.00	1.31	.63	5.44	3.19	6.44	1.25
5/8	5,200	3.00	1.56	1.75	2.38	1.50	.75	6.56	3.88	7.81	2.25
3/4	7,200	3.50	1.75	2.00	2.63	1.88	.88	7.19	4.31	8.69	3.5
7/8	10,000	4.00	2.06	2.25	3.06	2.13	1.00	8.38	5.00	10.13	5.4
1	12,500	4.50	2.31	2.50	3.50	2.38	1.13	9.63	5.75	11.63	8.8
1 1/8	15,200	5.00	2.38	2.75	3.75	2.56	1.25	10.38	6.25	12.63	12
1 1/4	18,000	5.63	2.69	3.13	3.69	3.00	1.38	11.13	6.75	13.63	16
1 1/2	45,200	7.00	4.19	4.00	4.19	4.00	2.25	17.13	10.00	20.13	19

Ultimate Tensile Strength Five Times Safe Working Load.

*Courtesy The Crosby Group

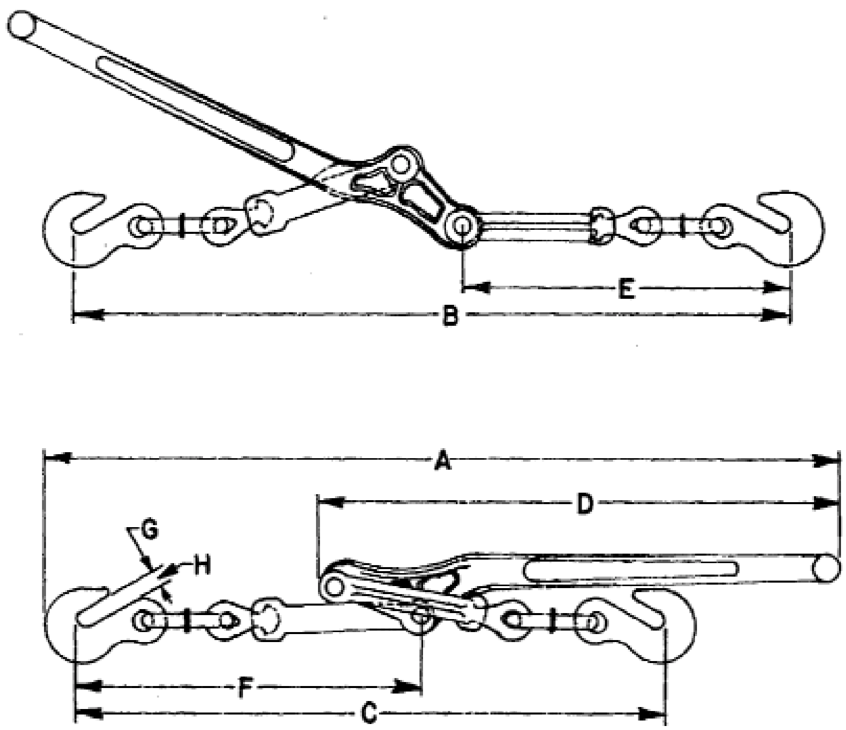
JAW END SWIVEL



SIZE	SAFE WORK LOAD LBS.	A	B	C	G	J	K	L	M	N	P	R	T	U	V	WT.LBS EACH
1/4	850	1.25	.69	.75	.69	.69	.47	1.03	.31	.88	.25	2.63	3.38	1.69	1.69	.22
5/16	1,250	1.63	.81	1.00	.81	.81	.50	1.13	.38	.88	.31	2.94	3.88	2.06	1.81	.31
3/8	2,250	2.00	.94	1.25	1.00	1.00	.63	1.41	.50	1.06	.38	3.63	4.75	2.50	2.25	.56
1/2	3,600	2.50	1.31	1.50	1.31	1.31	.75	1.75	.63	1.31	.50	4.50	6.06	3.19	2.88	1.25
5/8	5,200	3.00	1.56	1.75	1.63	1.50	.94	2.06	.75	1.50	.63	5.31	7.31	3.88	3.44	2.13
3/4	7,200	3.50	1.75	2.00	1.88	1.88	1.13	2.53	.88	1.75	.75	6.06	8.31	4.31	4.00	3.5
7/8	10,000	4.00	2.06	2.25	2.13	2.13	1.19	2.75	1.00	2.06	.88	7.00	9.53	5.00	4.53	5.3
1	12,500	4.50	2.31	2.50	2.63	2.38	1.75	3.72	1.13	2.81	1.13	8.56	11.69	5.75	5.94	9.8
1 1/8	15,200	5.00	2.38	2.75	2.63	2.56	1.75	3.72	1.25	2.81	1.13	8.84	12.19	6.25	5.94	14
1 1/4	18,000	5.63	2.69	3.13	3.13	3.00	2.06	4.31	1.50	2.81	1.38	9.44	13.13	6.75	6.38	17
1 1/2	45,200	7.00	4.19	4.00	5.63	4.00	2.88	6.00	2.25	4.44	2.25	14.74	20.84	10.0	10.84	49

Ultimate Tensile Strength Five Times Safe Working Load.

*Courtesy The Crosby Group



MODEL	A	B	C	D	E	F	G	H CHAIN SIZE	TAKE UP
7-1	24.19	21.5	17.25	16.0	10.0	10.5	0.5	5/16-3/8	4.25
A-1	28.0	24.63	20.13	18.5	11.63	11.56	0.63	3/8 -1/2	4.5
C-1	30.5	28.25	23.5	19.5	13.5	13.25	0.75	1/2 -5/8	4.75
L-1	35.75	31.0	26.5	23.25	15.5	14.13	0.94	5/8 -3/4	4.5

LOAD BINDER

*Courtesy The Crosby Group

6.0 METRIC CONVERSIONS

METRIC EQUIVALENTS OF
STANDARD ROPE SIZES

DIAMETER Inches	Millimeters
1/64	.40
1/32	.79
3/64	1.2
1/16	1.6
5/64	2.0
3/32	2.4
7/64	2.8
1/8	3.2
5/32	4.0
3/16	4.8
7/32	5.6
1/4	6.3
5/16	7.9
3/8	9.5
7/16	11.1
1/2	12.7
9/16	14.3
5/8	15.9
11/16	17.5
3/4	19.0
13/16	20.6
7/8	22.2
15/16	23.8
1	25.4
1 1/16	27.0
1 1/8	28.6
1 3/16	30.2
1 1/4	31.7
1 3/8	34.9
1 7/16	36.5
1 1/2	38.1
1 5/8	41.3
1 11/16	42.9
1 3/4	44.4
1 13/16	46.0
1 7/8	47.6
1 15/16	49.2
2	50.8
2 1/16	52.4
2 1/8	54.0
2 1/4	57.1
2 1/2	63.5
2 3/4	69.9
3	76.2
3 1/4	82.5
3 1/2	88.9

METRIC EQUIVALENTS OF
WEIGHT IN TONS

SHORT TONS	SHORT TONS
1	.90
2	1.81
3	
4	2.72
5	3.62
6	5.44
7	6.35
8	7.25
9	8.16
10	9.07
11	9.97
12	10.88
14	12.70
15	13.60
20	18.14
25	22.67
30	27.21
35	31.75
40	36.28
45	40.82
50	45.35
55	49.89
60	54.43
65	58.96
70	63.50
75	68.03
80	72.57
90	81.64
100	90.75
115	104.32
125	113.39
130	117.93
135	122.47
140	127.00
150	136.07
165	149.68
200	181.43
250	225.79
265	240.40
300	272.15
325	294.83
350	317.51

METRIC CONVERSIONS
ENGLISH TO METRIC

To Find	Multiple	By
microns	mils	25.4
centimeters	inches	2.54
meters	feet	0.3048
kilometers	miles	1.609344
gram	ounces	28.349523
kilogram	pounds	4.5359237×10^{-1}
liters	gallons (U.S.)	3.7854118
liters	gallons (Imp.)	4.546090
milliliters (cc)	fl. Ounces	29.573530
sq. centimeters	sq. inches	6.4516
sq. meters	sq. feet	9.290304×10^{-2}
sq. meters	sq. yards	8.3612736×10^{-1}
milliliters (cc)	cu. Inches	16.387064
cu. meters	cu. feet	2.8316847×10^{-2}
cu. meters	cu. yards	7.6455486×10^{-1}

Temperature conversion

$$°F = 9/5\,(°C) + 32$$
$$°C = 5/9\,(°F - 32)$$

7.0 ENGINEERING UNITS

ENGINEERING UNITS

1 Horsepower	=	33,000 foot pounds per minute
		550 foot pounds per second
		746 watts
		.746 kilowatts
1 Horsepower Hour	=	.746 kilowatt hours
		1,980,000 foot pounds
		2,545 heat units (B.T.U.)
1 Kilowatt	=	1,000 watts
		1.34 horsepower
		737.3 foot pounds per second
		44,240 foot pounds per minute
		55.9 heat units (B.T.U.) per minute
1 Kilowatt Hour	=	1,000 watt hours
1.34 horsepower hours		
		2,654,200 foot pounds
		3,412 heat units (B.T.U.)
1 British Thermal Unit	=	1,055 watt seconds
		778 foot pounds
		.000293 kilowatt hour
		.000393 horsepower hour
1 Watt	=	1 joule per second
		.00134 horsepower
		3,412 heat units (B.T.U.) per hour
		.7373 foot pounds per second
		44.24 foot pounds per minute

INDEX

A

B

C

D

P

R

S