Cable Payout System

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This paper describes the development of cable machinery and control equipment capable of laying modern submarine cable systems. The necessity for reliable continuous operation is established, and the problems involved in gripping armorless cable are pointed out. The concept of a linear cable engine is introduced, and the evolution of a track design is followed through several model stages.

The various test programs employed to establish feasibility, prove-in component reliability, and check performance of the final machine are discussed. A control philosophy is presented and the development of a control system is described. Final design, construction, and testing of the cable machinery and its control equipment are covered briefly.

I. INTRODUCTION

In essence, a cable payout system consists of stowage for the cable, machinery for exerting a braking action on the cable to regulate the payout rate and instrumentation for determining the correct payout rate. In the first attempt to lay a telegraph cable across the English Channel, the stowage and braking functions of a payout system were provided by a reel mounted on the after deck of a tug and fitted with a brake. For cable systems more than a very few miles in length, the size of the reel soon became unreasonable and less obvious approaches had to be taken.

Stowage of cable in cylindrical holds or tanks was employed early in the development of the ocean cable laying art, and seems to have been a natural evolution from the common nautical practice of tiering chain in chain lockers or of coiling whale line in tubs. Similarly, the drum-type cable engines employed for many years as braking devices in handling telegraph cables appear to have been an early adaptation of the anchor windlass.

While a combination of tank stowage with drum machinery was adequate for telegraph systems, it leads to problems when systems with rigid repeaters are laid, since the ship must be stopped and turns taken off the drum while each rigid repeater is passed overboard. Analytical work in the field of cable mechanics has shown that adequate control of slack requires continuity of operation. It was therefore necessary to develop continuous methods of handling rigid repeaters. Drum-type cable engines can be fitted with auxiliary devices for handling rigid repeaters continuously at slow or moderate speeds, and cable engines consisting of multiple V-sheaves have been designed for handling rigid repeaters at slow speeds by means of bypass ropes. For continuous handling of rigid repeaters at full cable laying speeds, however, a linear or straight line cable engine is essential.

II. LINEAR CABLE ENGINE DESIGN REQUIREMENTS

Modern submarine cable systems, in addition to having large rigid repeaters, differ from previous systems in another aspect which is important from the cable machinery viewpoint, namely that the cable is of the so-called armorless type with the strength member in the center.[1] The strength member is composed of very high-strength steel, and is surrounded by the delicate transmission structure consisting of polyethylene and thin sheet copper.

The cable tension which results from the weight of the length of cable suspended between a ship and the ocean bottom must be transferred to the ship by frictional shear forces exerted on the cable surface. In the case of drum engines the normal forces on the surface required for the development of the frictional forces result from the wrapping of the cable around the drum. In the linear cable engine, the curved surface is absent and the cable tension does not result in any normal force on the cable's surface. The cable must therefore be squeezed between the tracks of the cable engine.

The cable engine design requirements, from the viewpoint of the cable system, can be summarized as follows: the cable engine must handle cable and repeaters continuously at a steady speed, in a straight line, gripping the cable so as to transfer the tension to the ship without damaging the transmission structure and passing repeaters without excessive shock.

III. DESIGN APPROACH

The development approach adopted for the linear cable engine was strongly influenced by two very firmly held principles. First, the cable engine and associated equipment was to be designed to handle a mechanically optimum cable and repeater system; in other words, the cable system designers would not be asked to compromise their designs in order to ease machinery problems. Secondly, the entire design would be based on achieving the highest possible reliability so that there would be a very high probability of completing a lay (say 2000 nm) in one continuous operation without any interruption due to cable engine failure. Because of the first of these requirements, it was natural to design the cable engine from the inside out; that is, to commence by considering the problem of gripping a straight cable in such a way as to transfer the highest possible cable tension to the ship without damage to the transmission structure. The moving track which accomplished the gripping was then to be designed to accommodate repeaters. A system of sprockets and shafting to support and drive the track would be added next, and addition of a framework and a drive and control system would complete the design. At each stage in its evolution the design was evaluated for reliability, and an extensive testing program was planned to prove-in each element.

3.1 *Cable Gripping*

The squeezing forces required to develop cable tension must be applied in a symmetrical manner to avoid any tendency to separate the dielectric from the center conductor. Analysis, described in Appendix A, showed that the cable would have to be gripped along at least four lines, equally spaced around its circumference, if regions of negative pressure on the center conductor interface were to be avoided. The double V-block design which is shown in Fig. 1 gave the desired cable gripping configuration.

Because of the nature of the resulting pressure distribution in the dielectric, the frictional shear forces which can be transmitted across the various unbonded interfaces within the cable are directly proportional to the radii of the interfaces. This means that the stretching of the center conductor which occurs under high tension cannot be accommodated by slip of the entire cable, because slip will occur between the center conductor and the dielectric before the outer jacket slips in the grooved blocks of the cable engine. For this reason, a linear cable engine which is to handle cable without interface bonds must have a so-called "shear-limiting" feature which limits the shear force applied to a unit length of the cable. The allowable value of shear force per unit length depends in turn on the squeeze applied to the cable.

Experimental work on cable samples showed that when the cable was gripped between the double V-blocks no permanent distortion of the

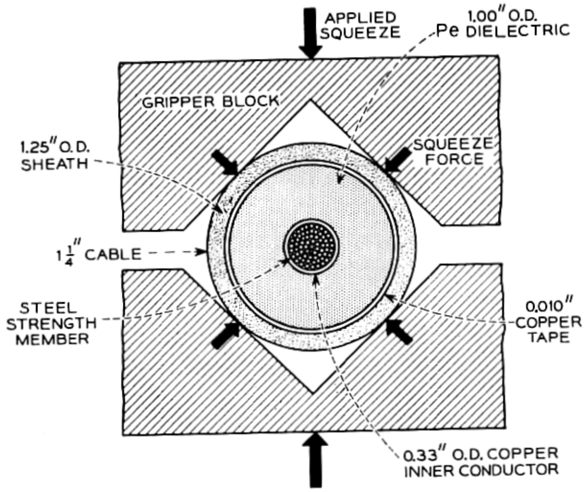


Fig. 1 — Double V-block cable gripping.

cross section would occur if the squeeze force did not exceed the value which gave a contact force of 500 pounds per inch along each contact. A working value of one-half the maximum or 250 pounds was used for design purposes. With this value of squeeze force and a coefficient of friction between copper and polyethylene of approximately 0.16, a tension decay rate of 40 pounds per inch of cable could be obtained without causing slip on the center conductor interface, which had a circumference of approximately one inch. Since the nominal tensile strength of the cable was to be approximately 16,000 pounds, this meant that the cable would have to be gripped over a length of 400 inches, or approximately 33 feet, if it was to be held up to breaking tensions without any internal slip. Thus the length of the gripping region of the cable engine was set by cable characteristics.

3.2 *Quarter-Scale Model*

The development of the linear cable engine involved several phases. In the first phase a one-quarter scale model was built to test the basic concept. At this stage in the development the cable gripping problem was not completely understood, and there was considerable optimism regarding the development of a cable having bonds on the various interfaces to transmit shear stresses. It was also thought that the proposed tracks would have to be forced apart by some sort of cam to allow the repeater to pass through. For these reasons the initial quarter-scale model did not include the shear limiting feature, but was fitted with a pair of traveling cams which were clutched into the main drive system so as to spread the tracks apart, leaving a pocket to accommodate the repeater. The tracks which gripped the cable were pressed together by rollers mounted on saddles, the saddles in turn being mounted on compliant air bags. The quarter-scale model, shown in Fig. 2, was operated for several months in carrying out tests of various stowage arrangements and overboarding devices. Tests with this machine indicated that model repeaters would pass through with no difficulty and that the cam system was not necessary.

3.3 *Full-Scale Track Tests*

During the evaluation of the quarter-scale engine it was decided that shear limiting would be provided in the final engine, since it was certain that reliable chemically bonded interfaces could not be produced in time.

Before proceeding with the design of a mchine incorporating shear limiting, the shear limiting concept was checked out on a bench-type test rig which gripped a length of cable several feet long between blocks mounted on rollers between squeezing units. Axial motion in the direction of applied cable tension was controlled by the shear limiting units. Various spring configurations were tried and an ordinary coil spring

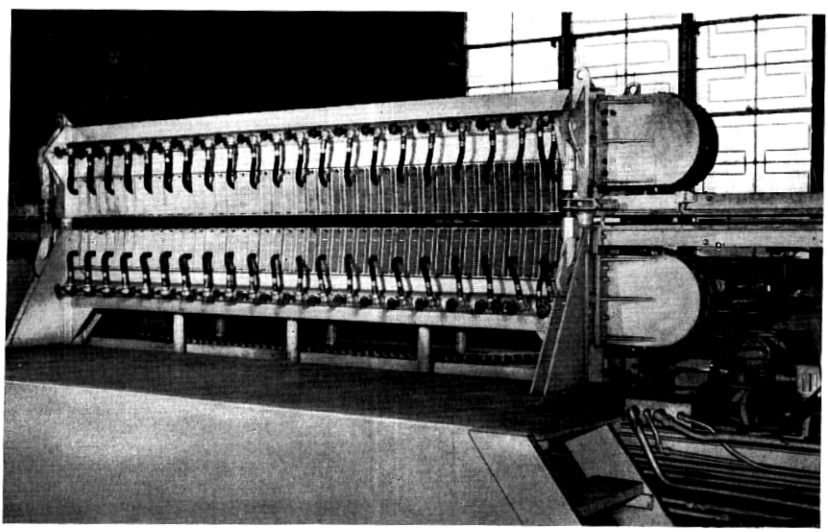


Fig. 2 — Quarter-scale model.

mounted with a preload in a cartridge was found to be the most satisfactory. The shear limiters function as shown schematically in Fig. 3. The observed deflection is essentially parabolic, as predicted by theory.

In order to allow for shear limiting motion, it was necessary to leave gaps between the individual members of the track. The track also had to articulate for envelopment of the repeater. Because the track moves with the cable, it was necessary for the track to move at high speed with respect to the elements which applied the squeezing force. After an investigation of possible sliding contacts, it was decided that rollers on the track elements would be necessary to handle the large squeezing forces and high speeds.

A track test rig was designed to check out the track as conceived at that stage of its development. Fig. 4 shows this machine, which was full size as regards width of track, but was only a few feet in gripping length. In addition, the test rig involved only one track, the opposing track being simulated by a series of simple roller carriages traveling on a rigid base. The moving track was backed up by an articulated belt, and squeeze was applied by air springs as shown in Fig. 5. This test rig was run for many hundreds of miles and at intervals a longitudinally split half repeater was passed through.

3.4 *Design and Construction Contractor*

At this point Western Gear Corporation was selected to do final design and to manufacture the linear cable engine. Its representatives joined with engineers of Bell Laboratories and Bergen Research Engineering

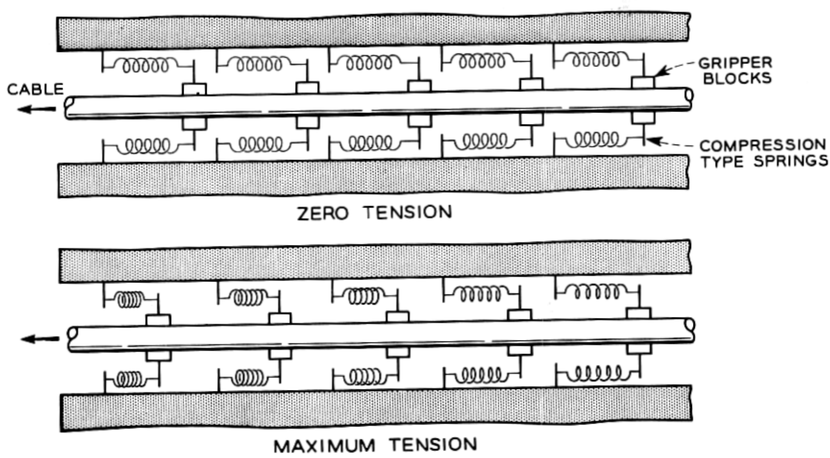


Fig. 3 — Shear limiting.

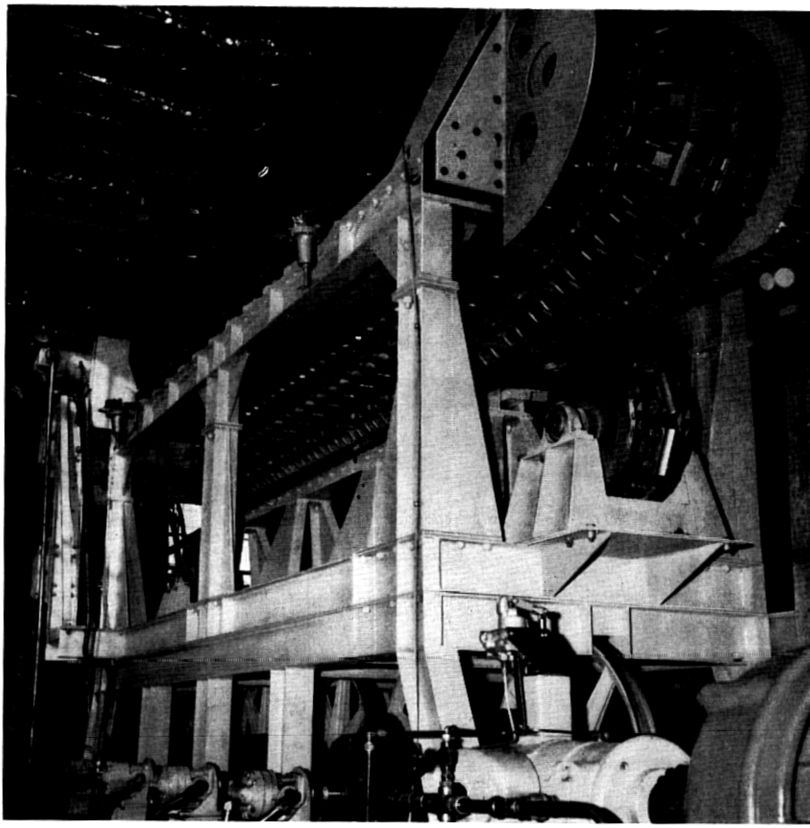


Fig. 4 — Track test apparatus.

Corporation in planning and evaluation of the test program for the track test rig. After observing the tests and studying the test results, the three groups concurred in the opinion that the rubber rollers and their bearings would be the limiting factors in designing a squeeze-type engine to have sufficient reliability. It was also concluded that the air bags which provided the compliant backing for the belts were causing an intolerable pressure increase when repeaters passed through the machine and that the solution of this problem by increasing the number of air bags in each stack would lead to a stability problem.

3.5 *Linear Cable Engine Design Studies*

Upon completion of the test program and associated studies, preliminary design work on a prototype linear cable engine was started. Initially,

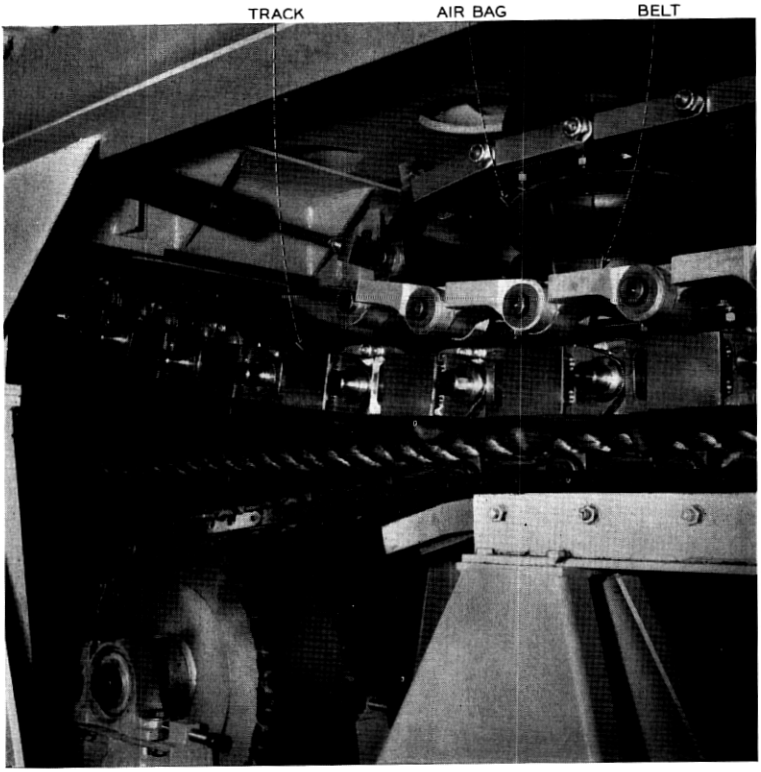


Fig. 5 — Track test apparatus; detail showing articulated belt and air springs.

effort was directed along two parallel lines: first, attempts to resolve the problems inherent in the squeeze-type linear cable engine and second, paper and model studies of alternative linear cable engine concepts proposed by Bell Laboratories.*

A nitrogen-accumulator-backed oil-hydraulic squeeze system, proposed as a substitute for the air bags, was studied. These studies showed that a combination of hydraulic cylinders with linkages could be designed to maintain the squeeze pressure essentially constant as the repeater entered the machine.

The question of transients incidental to repeater passage at high speeds was resolved by means of a test rig based on a large lathe. A cam mounted on the face plate subjected a cylinder and linkage assembly to the displacements and velocities incidental to the passage of a repeater

* These involved other gripping methods which would avoid the problem of applying high squeezing forces to a moving track.

at speeds up to 10 knots. After many hours of operation of this test rig it was concluded that the hydraulic cylinders with their linkages and nitrogen accumulators would be a satisfactory substitute for the air bags used in the previous models.

Discussion of the rubber roller problem with many rubber manufacturers led to the conclusion that the loads and speeds involved in the proposed linear cable engine would be marginal for available rubber compounds. Because our particular operating conditions and duty cycle were unique, a test program was required to establish reliability for the rubber rollers. The rubber compounds which appeared to be most suitable were made up in sample rollers, and these were run for several thousands of hours on rotating drums. The rollers were pressed against the drums intermittently to simulate the duty cycle of rollers passing through the squeeze zone on the linear cable engine. On the basis of the results of these tests, it was concluded that the rollers could be depended upon to function reliably for at least 2000 nm and possibly for several times this figure.

Initial design studies of track configurations capable of passing around sprockets of reasonable size and enveloping repeaters indicated that approximately 3000 antifriction bearings would be required to support the necessary rubber rollers. Because of the space restrictions and the load and speed requirements, the B_{10} life* of the bearings could not be made high enough to give an acceptably large probability of no bearing failures in 2000 miles of operation. Thus it was necessary to assure ourselves that a bearing which had failed by the usual bearing industry criterion would not lead to a catastrophic failure.

The Timken Roller Bearing Company had recommended a bearing identical with that used in the front wheels of compact automobiles for this application. Its engineers felt that because of the heat treatment given to the bearing races an initial fatigue failure would not propagate rapidly and thus lead to an early catastrophic failure. In order to test this hypothesis a roller bearing test program was planned. The Timken Company produced initial fatigue failues in a group of 20 bearings in their laboratory. Prototype rubber rollers were mounted on these bearings and run at prototype loads, speeds, and duty cycles for thousands of hours. The rollers were then disassembled and the bearings photographed. The photographs were compared with photographs taken at the time of initial fatigue failure. In most cases there was no discernible increase in the spalled area of the initial failure. In one or two cases the area did increase noticeably, but there was no indication that catastrophic failure would

* Life exceeded by 90 per cent of a population of bearings.

have occurred within many thousands of hours. Details of the failure propagation study are given in Appendix B.

3.6 *Track Side Chains*

The proposed track was to consist of a series of roller carriages mounted on roller chains which were in turn carried on sprockets. From exploratory work it was concluded that it was not feasible to lubricate modern roller chains for high-speed operation in a salt water atmosphere without exposing the cable to contamination by the lubricating medium. A special precision pintle-type chain, having oil-impregnated sintered-metal bearings, was therefore developed. Since there are no rotating rollers in this type of chain and each link merely oscillates through a small angle with respect to its neighbor, it was possible to seal in the lubricating medium by means of rubber shear seals which could tolerate the small angle of rotation. Because of the lack of rollers, there is of course a sliding motion of the links with respect to the sprocket. It was possible to provide lubrication for this motion by means of a very viscous type of grease which would not be thrown off the sprocket onto the cable. A test section of the new type of chain was manufactured and was found to operate satisfactorily under the prototype loads and speeds.

3.7 *Roller Carriage Design*

The roller carriages were to be connected to the side, or tension, chains by means of shear limiters which would allow essentially free axial motion of the roller carriage with respect to the chain under a certain axial load, thus limiting the shear force per unit length applied to the cable. In addition, it was necessary to design the roller carriages so that the squeezing forces applied to the cable through the rollers were not transmitted to the chains, since this would put large frictional side loads on the shear limiters. It was also desirable that the two chains be connected transversely by a rigid member. The roller carriage design which met all of these requirements was, of course, somewhat complicated. Fig. 6 is a photograph of a wooden model of the design. The model included three links of the pintle chain on each end of the roller carriage. Fig. 7 shows the relative orientations of the roller carriages, the side chains, and the articulated metal belts which apply the squeezing forces to the roller sides of the carriages.

3.8 *Final Design of Linear Cable Engine*

Upon completion of the development testing which proved-in the bearings, rollers, shear limiter, pintle chain, and the hydraulic squeezing

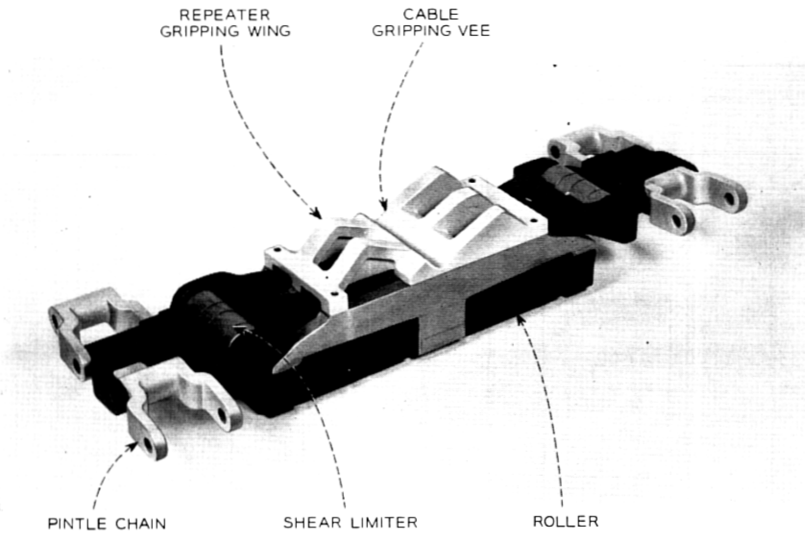


Fig. 6 — Model of roller carriage.

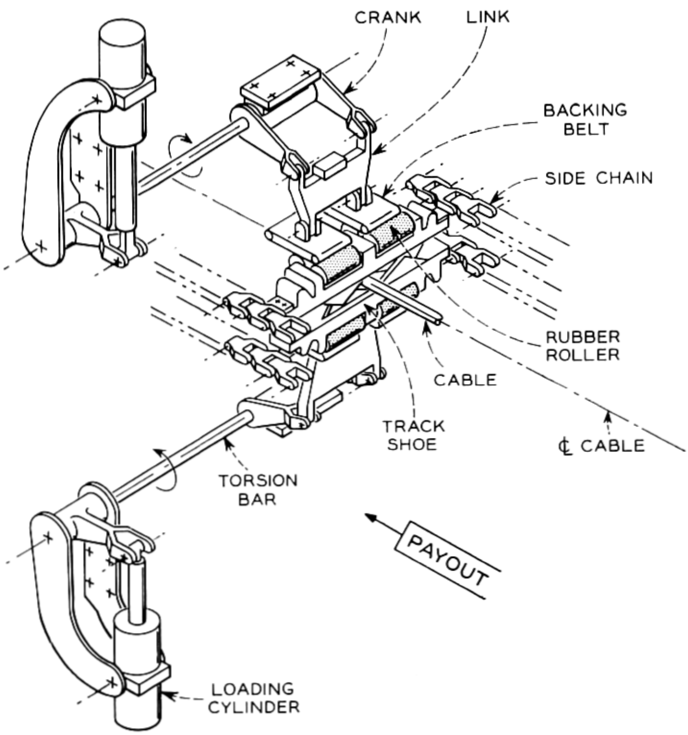
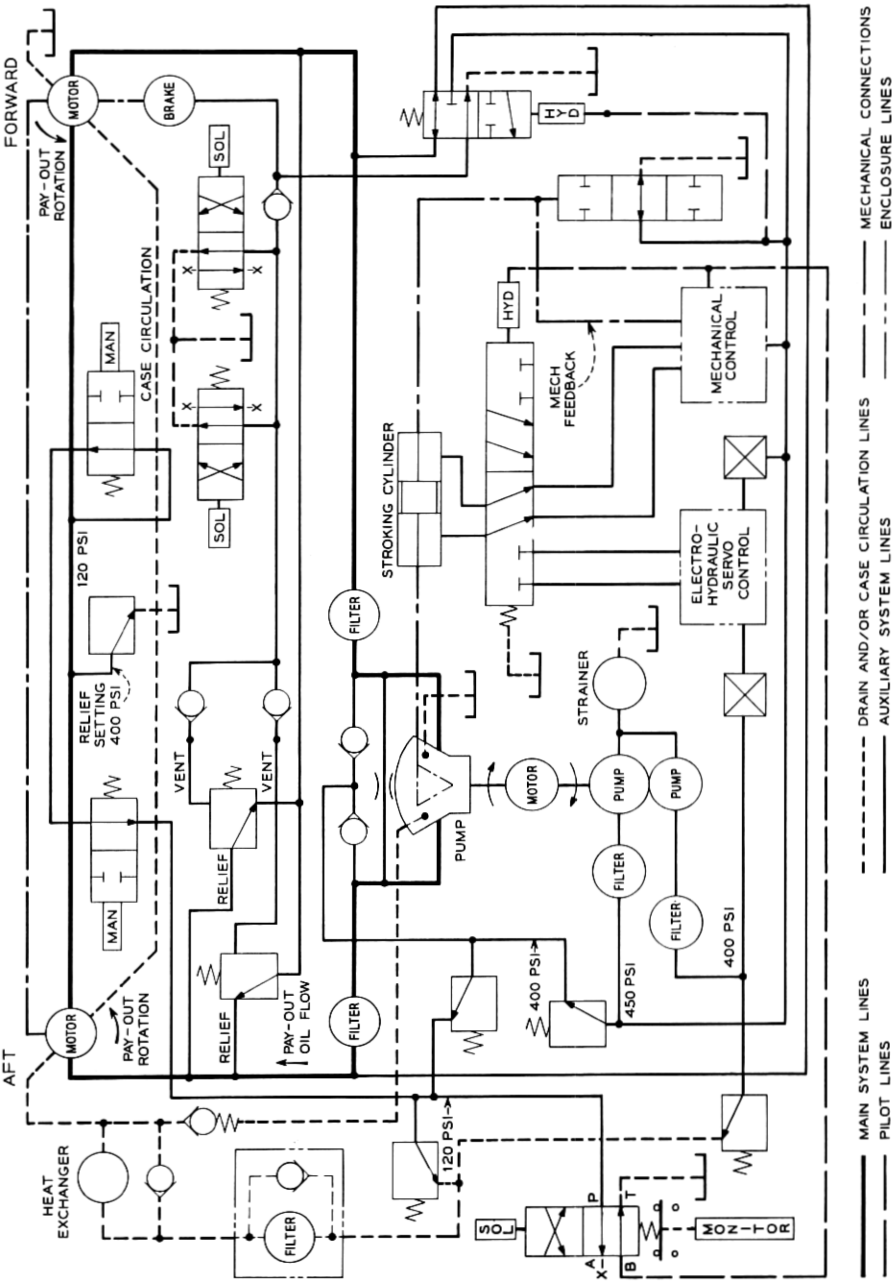


Fig. 7 — Roller carriage assembly detail.


FORWARD
PAY-OUT ROTATION
MOTOR
BRAKE
SOL
CASE CIRCULATION
MAN
DRAG (PICKUP)
CABLE TENSION
120 PSI
RELIEF SETTING 400 PSI
VENT
RELIEF
DRAG (PAYOUT)
PAY-OUT OIL FLOW
AFT
HEAT EXCHANGER
FILTER
HYD
MECH FEEDBACK
MECHANICAL CONTROL
STROKING CYLINDER
ELECTRO-HYDRAULIC SERVO CONTROL
STRAINER
PUMP
400 PSI
450 PSI
P
T
A
B
MONITOR
MAIN SYSTEM LINES
PILOT LINES
DRAIN AND/OR CASE CIRCULATION LINES
AUXILIARY SYSTEM LINES
MECHANICAL CONNECTIONS
ENCLOSURE LINES

units, the entire design was reviewed for comparison with some of the other types of linear cable engines under consideration. Although it was felt that at least one of the other approaches was probably capable of yielding a much lighter, smaller and cheaper machine, the schedule did not allow for the additional study and testing work needed to prove-in the alternate design. The development schedule, in conjunction with the encouraging results of the endurance tests on the bearings and rollers, therefore led to a decision to drop further study of alternative designs and to proceed with the final design of a squeeze-type linear cable engine.

Functionally, the linear cable engine was to consist of two major subsystems, the drive system and the squeeze system. The drive system included the track which grips the cable and moves with it, the sprockets, sprocket shafts, gear boxes, and a hydraulic system for driving the track. The squeeze system was comprised of the articulated belts between which the tracks are squeezed, the hydraulic units for applying the squeeze force to the track and the associated piping, accumulators, and hydraulic pumps for supplying oil pressure and nitrogen back-up pressure to the squeezing units.

3.8.1 *Drive System*

Because of rolling resistance of the rubber rollers and drag of the bearing seals, it was anticipated that there would be several thousand pounds of track drag. This meant that at low cable tensions it would be necessary to drive the cable engine in the payout direction, while at high cable tensions, on the other hand, it would be necessary to exert a braking effort on the track. To avoid pulling the slack, required to envelope the repeater, out of the return side of the chain and introducing it into the squeezed portion, it was necessary to drive the outboard sprocket when pushing cable out and to drive the inboard sprocket when braking. This called for a drive system consisting of a hydraulic pump with two motors in series, one attached to the sprocket shafts at each end of the machine. The replenishing oil required to prevent cavitation of the pump was introduced into the loop in such a way as to cause the two motors to pull against each other and thus keep the track under tension in the squeezing zone. A hydraulic schematic of the system is shown in Fig. 8.

3.8.2 *Squeeze System*

Fig. 9 is an artist's rendering of the linear cable engine as envisioned at this stage in its development. The machine is shown built into the deck of a ship, with a repeater enveloped by the track. The tracks in turn are backed up by the articulated "belts" which appear light-colored

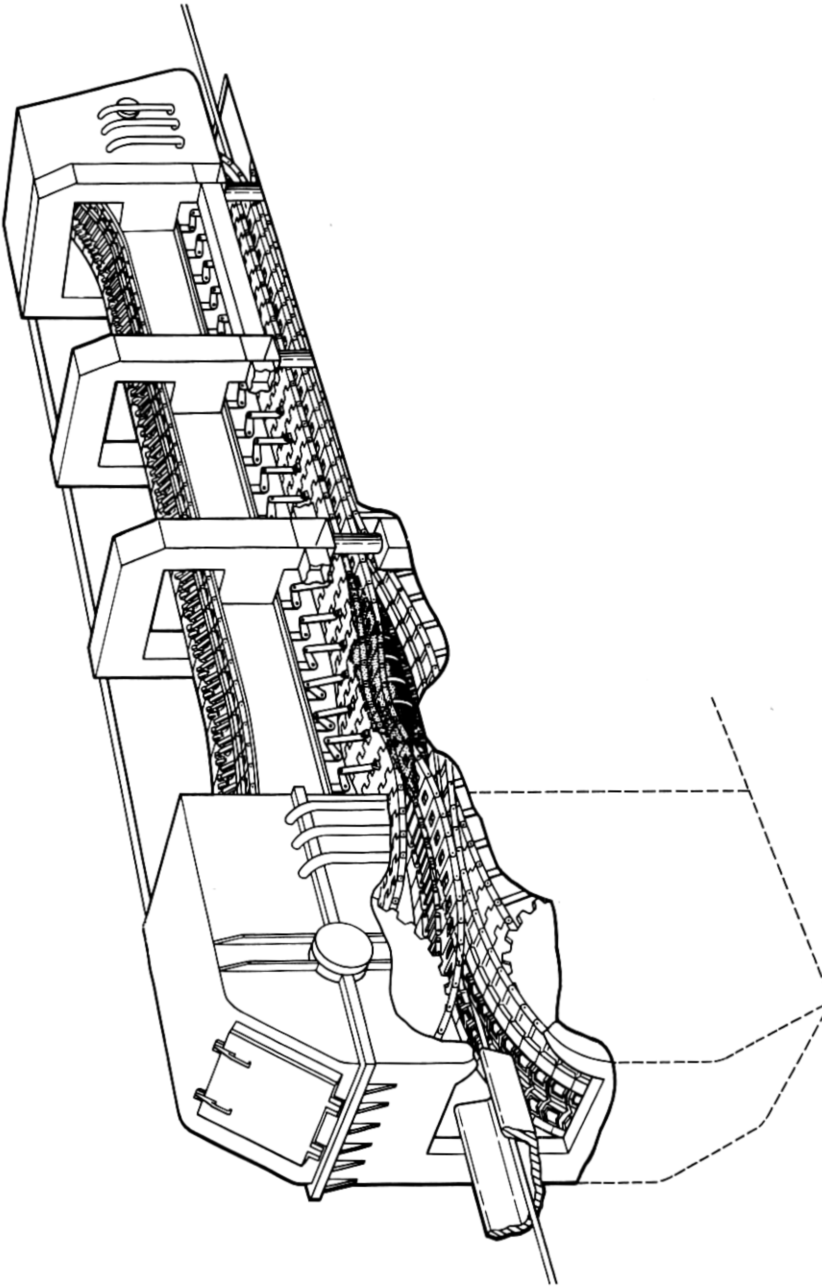


Fig. 9 — Linear cable engine, artist's rendering.

in the figure. Details of the proposed hydraulic squeezing systems are given in Figs. 7 and 10. For reliability, the gripping region of the cable engine was to be divided into seven zones, each of which had its own system of hydraulic valves and accumulator. The seven sections were to be supplied from a dual hydraulic power supply, as shown in the schematic.

IV. CONSTRUCTION

During the final stages of the design a contract was negotiated between Transoceanic Cable Ship Company, a subsidiary of the American Telephone and Telegraph Company, and Western Gear Corporation for the construction of the linear cable engine and other machinery to be described below. This contract called for manufacture in the Seattle, Washington, area with assembly at a newly acquired facility at Everett, Washington.

The completed linear cable engine is shown in Figs. 11 and 12. Fig. 13 is a close-up of the tracks and belts.

V. TEST PROGRAM

In addition to the large amount of development testing which was used to prove-in various components and subsystems, two other major testing programs were carried out to insure reliability of the linear cable engine. The first of these was in the area of production testing. As a supplement to a 100 per cent inspection of all machined parts, extensive use was made of X-rays for castings and Magnaflux for the large number of aluminum forgings in the track. Many special production tests were made on subassemblies. For example, a special test rig was made and calibrated for checking the bonding of the rubber rollers to their shafts. Another example was the production testing of the roller carriages. After assembly of the rollers with the Timken tapered roller bearings, each carriage was put on a test rig which pressed it under the full prototype load against a drum which rotated with a peripheral speed of 800 feet per minute. Temperature measurements were made and each carriage was run until the temperatures at several points stabilized. This check was intended to detect bearings with excess axial preload and faulty seals.

The final category of testing involved the completed machine. This program in turn consisted of two categories: first, performance testing in which it was determined that the machine met all of the specification requirements as to speed, tension, etc., and second, an endurance run of approximately 500 miles.

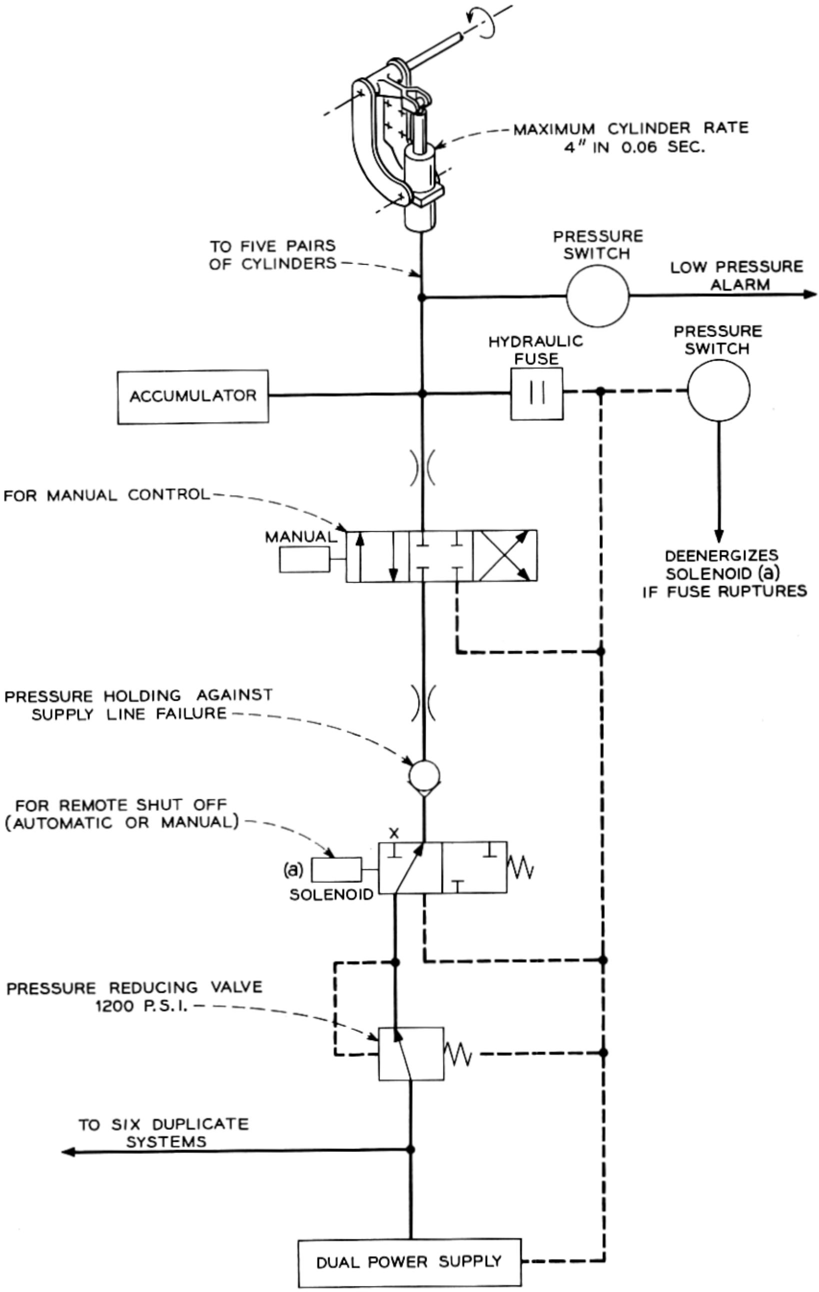


Fig. 10 — Schematic of hydraulic squeezing system.

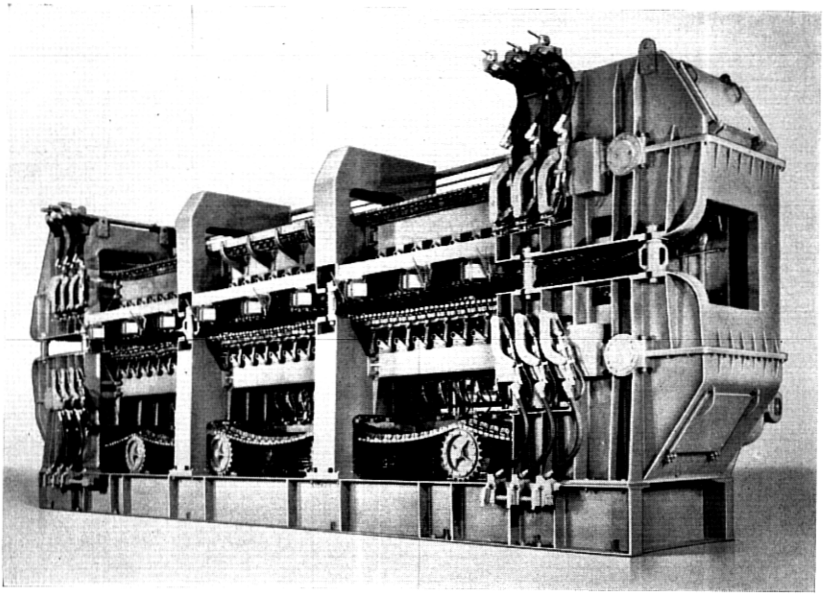


Fig. 11 — Linear cable engine.

Fig. 12 — Linear cable engine.

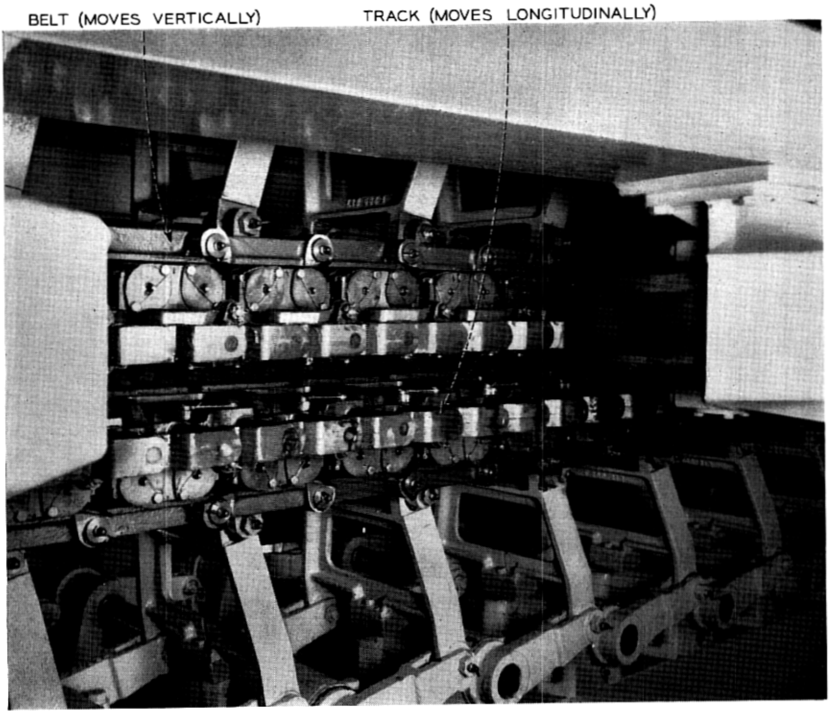


Fig. 13 — Linear cable engine; detail showing tracks and belts.

For the performance evaluation and the endurance test, the completed cable machinery was set up in such a manner that the linear cable engine could be operated against one of the drums of the bow cable machinery which will be described below. A 1000-foot length of $1\frac{1}{2}$-inch diameter "Spring Lay" mooring line was spliced into a closed loop and was run through the linear engine and over one of the drums. Large horizontal sheaves were used to complete the loop. The machine was then run at various speeds and tensions up to a maximum of 8 knots speed and 8000 pounds tension, and a wooden repeater having the size and shape of the prototype repeater was introduced into the inboard end of the machine and run through to check the articulation of the track.

The linear cable engine was designed to pull up to tensions of 16,000 pounds at low speed in the pickup direction. Pickup performance was checked by picking up at high tension with the machine in low gear. Next the brakes were set and a 40,000-pound static pull applied to the linear cable engine without slip or damage.

After successful completion of the performance checkout, the linear cable engine was run continuously day and night for 500 miles at 8 knots with the wooden repeater being sent through at intervals corresponding to the 20-mile repeater spacing. In the final test the linear cable engine was mounted on angle blocks and run inclined at an angle of $22\frac{1}{2}°$. Principal characteristics of the linear cable engine are summarized in Table I.

VI. OTHER CABLE HANDLING EQUIPMENT

In addition to the linear cable engine which was designed for steady-state high-speed laying, it was necessary to provide drum-type cable engines for handling grapnel rope, buoy moorings, and various kinds of cable encountered in repair work on telephone, telegraph, and other cable systems. Drum cable engines are essentially large-diameter winches in which tensions are built up in several turns of cable. In the case of an ordinary deck winch the necessary back tension is applied by a seaman who pulls on the hawser on the low-tension side of the winch. For a drum-type cable engine, a so-called "drawoff-holdback gear" serves this function.

The double-drum bow cable engine was also designed to handle rigid repeaters at low speeds during repair operations. This requirement called for large-diameter, wide drums and special auxiliary equipment. The 12-foot diameter specified for the drums allowed some margin over the minimum called for by the repeater rigid length, gimbal angle, and cable bending radius; the three-foot width allowed for five turns of heavy shallow-water cable on either side of a repeater.

As each turn of cable comes onto a drum, it tends to fall alongside the

Table I — Linear Cable Engine Capabilities

Cable diameter	
armorless cable (double-vee gripping)	1.20″ min.–1.80″ max.
armored cable	1.20″ min.*
Repeater dimensions	
diameter	14″ max.
length	12′ max.
Speed	
high gear	9 knots max.
low gear	2 knots max.
Tension	
high gear	8,000 lbs max.
low gear	16,000 lbs max.

* Diameters greater than 1.80″ contacted by "wings" on gripper blocks.

preceding turn; in other words, the cable "spools" itself across the drum. Drum cable engines are therefore normally fitted with so-called "fleeting knives" which plow the cable aside one cable diameter for each rotation of the drum to maintain the several turns of cable in an axially stationary position on the drum surface. To avoid rough treatment of the cable by a stationary knife, the cable engine was fitted with fleeting rings which rotate freely about axes inclined slightly with the axis of each drum. The rings are so oriented that they are further apart at the top of the drum, where the cable comes on, and closer together at the bottom so that, whether the machine is paying out or picking up, the oncoming cable falls freely onto the drum surface and then is pushed axially by contact with the inclined ring as the drum rotates. The several turns of cable, therefore, lie against each other and against one fleeting ring or the other, depending on whether cable is being picked up or payed out.

The two drums are carried on antifriction bearings on stationary shafts which are cantilevered toward the center of the ship from opposite sides of a closed rectangular box frame. This type of mounting has two distinct advantages. First, the end of the "dead" shaft carries the stationary mounting for one fleeting ring. Since this structure is closed in by expanded metal panels, there are no moving parts which can be contacted by a seaman walking between the drums. Second, the fact that the drums face each other facilitates transfer of cable or rope from one drum to another and decreases the size of the deck opening required.

Each drum is driven by an independent direct-current electric drive system consisting of a heavy duty mill-type motor, a separately excited generator driven by one of the main propulsion turbines, and the necessary amplidynes, excitation generators and control equipment. In the payout gear ratio, the engine is capable of paying out or picking up cable at tensions up to 16,000 pounds at speeds up to 8 knots. In two other gear ratios, tensions of up to 100,000 pounds can be handled at lower speeds. The drum speed and direction of rotation can be controlled manually from a console near the engine or remotely from the cable control center. In addition to the dynamic braking inherent in the electrical drive, the drums have mechanical shoe brakes which can be applied hydraulically or by hand.

In order to avoid bending the cable to a small radius and subjecting it to concentrated loads, a linear drawoff-holdback gear was designed. The linear design also facilitates handling of rigid repeaters and of rigid connectors in grapnel rope. A hydraulic drive system is used to allow operation at constant tension when functioning as drawoff or holdback gear and at constant speed if it should be necessary to use the gear as a cable hauler when loading cable.

Cable tension is measured by means of a dynamometer which deflects the cable and weighs the resulting component of cable tension. Earlier dynamometers consisted of large-diameter wheels or sheaves with load cells or other devices for indicating tension. The dynamometers for C.S. *Long Lines* substitute sliding of the cable on a curved "table" for the rotation of a large sheave. The load is weighed by a conventional load cell arrangement.

To allow handling of repeaters without causing the overriding turns on the drum which would result if an attempt were made to fleet large-diameter repeaters and cable together, the drawoff-holdback gear and dynamometer for each drum were arranged to traverse. Auxiliary traversable fleeting knives were also provided. When a repeater approaches the drum, the several turns of cable are fleeted away from the active fleeting ring by the traversable fleeting knife, which is then retracted. The drawoff-holdback gear or dynamometer is then traversed to direct the oncoming lead of cable onto the drum in such a manner that the repeater clears the cable on the drum. Several turns of oncoming cable are then wound onto the drum between the repeater and the active fleeting ring as the turns on the other side come off. Further rotation of the drum causes the turns to "spool" along the drum until they again contact the fleeting ring to complete the cycle.

Fig. 14 is an artist's rendering of one drum of the bow cable engine

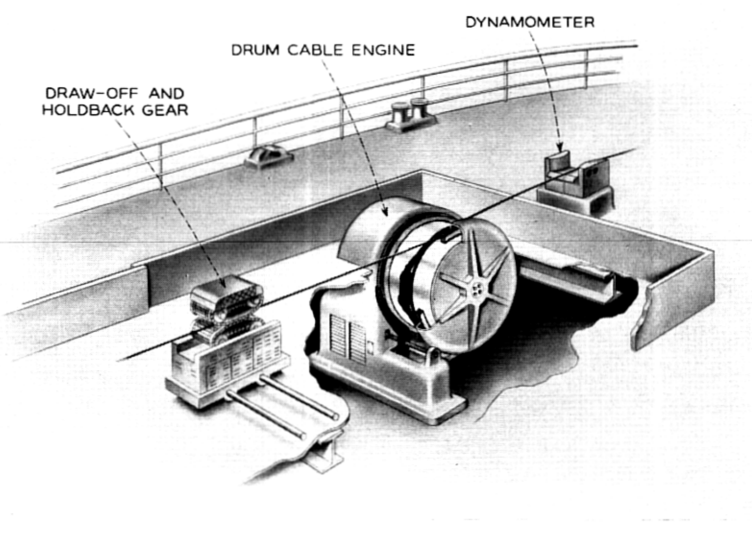


Fig. 14 — Cable engine drum with dynamometer and drawoff-holdback gear.

with its associated dynamometer and drawoff-holdback gear. Fig. 15 is a photograph of the actual drums as set up for test at the plant of the manufacturer. The fleeting rings can be seen in Fig. 16, which shows the starboard drum as installed in C.S. *Long Lines*.

6.1 *Cable Control Center*

On previous cable ships it has been the practice to control each cable engine from a nearby station so that the operator could maintain a watch on the engine and make speed adjustments by hand. Information needed for calculation of the required payout speed was collected from various sources throughout the ship. The required payout speed was then calculated, generally on the bridge, and payout speed or tension orders were then relayed to the engine operator. To integrate the cable laying operations and to permit a degree of automation, the concept of a "cable control center" has been introduced. This center was to be a room, located beneath the ship's bridge, containing all the instrumentation relating to the cable laying operation. It was also to be a control station from which any of the cable engines could be operated. Local

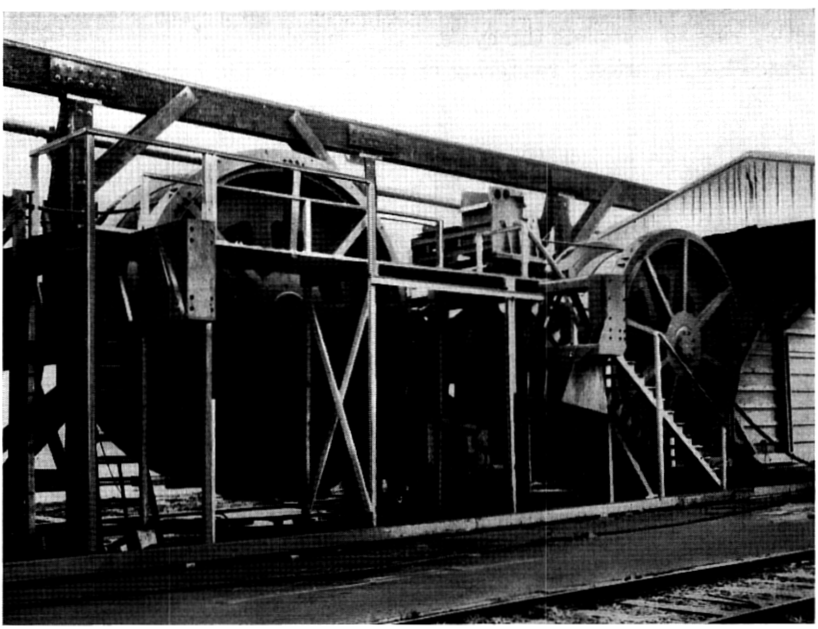


Fig. 15 — Cable engine drums set up for manufacturer's tests.

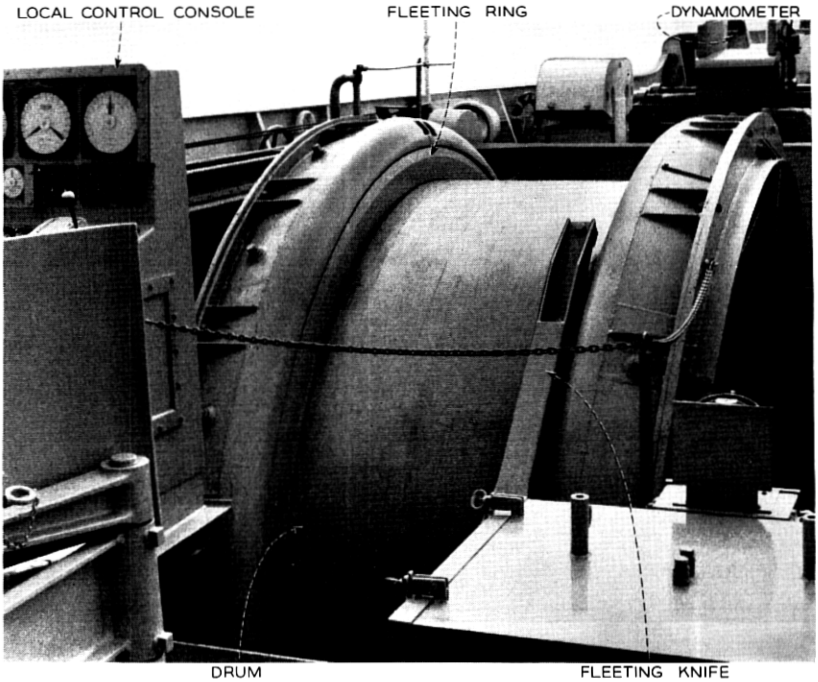


Fig. 16 — Cable engine drum installed in C.S. *Long Lines*.

control stations at each engine were still to be retained for periods when engine observation was necessary, such as during start-up or for cable repair operations. Fig. 17 shows the cable payout control console in the cable control center on C.S. *Long Lines*.

6.2 *Cable Engine Payout Control — Control Modes*

Submarine cable should be laid with just sufficient slack to enable conformance with the ocean bed, and to avoid residual tension in the cable after laying. Under these ideal conditions the cable tension at the ship during steady-state laying conditions would be essentially equal to what is usually referred to as the wh factor — that is, the product of the weight per unit length of the cable in sea water and the depth of the water. The tension at the ship will increase above this value if residual tension exists in the cable on the sea bed and conversely will be less than wh if sufficient excess slack is payed out to result in a significant longitudinal velocity of the cable relative to the water. Severe ship's motion

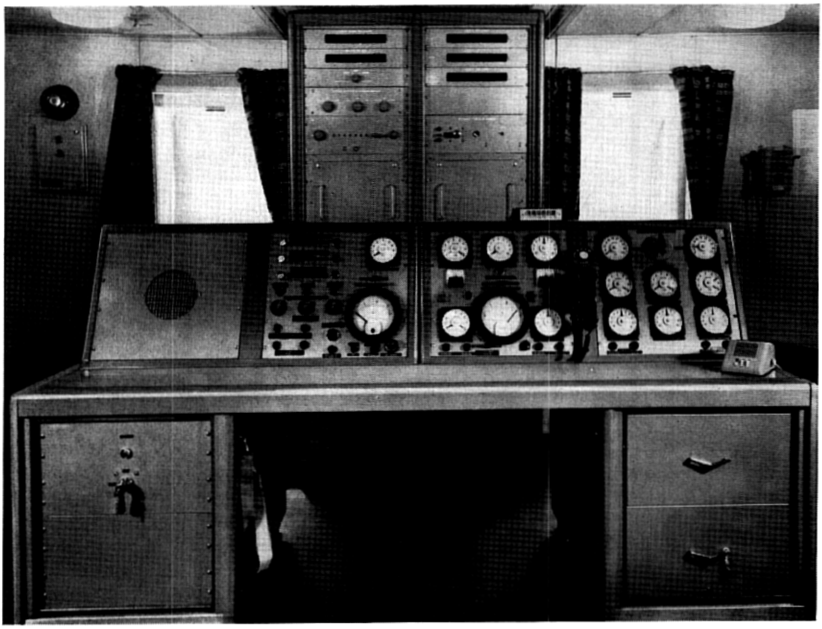


Fig. 17 — Cable payout control console.

will introduce transients into the steady-state laying conditions and will cause undesirable tension surges if the cable is payed out at a fixed speed. These tension surges can be eliminated by paying out at a fixed tension.

Because of the relationship among cable tension, payout rate, ship's speed, depth of water, and cable characteristics, a particular value of tension will yield a slack percentage which depends on the values of the other parameters. Because of uncertainties in some of these values and because of the sensitivity of payout rate to small changes in tension, it is not feasible to calculate the required value of tension from measured values of the other variables. A practical method, however, is to set up the steady-state laying conditions on the basis of cable engine speed as a result of ship speed and desired slack, to observe the resultant tension at that particular depth of water, and then to use this value of tension if ship's motion becomes excessive. Under tension control, the cable mileage payed out would be compared with distance steamed and the tension adjusted to increase or decrease the payout rate as required to produce the desired percentage of slack on the bottom.

From these considerations the following basic modes of operation were selected:

(a) Manual speed control, to obtain a desired amount of slack to cover the contour of the sea bed. Because the linear cable engine can actually push cable out, it is possible to lay slack even in very shallow water where the tensions approach zero.

(b) Manual tension control, to adjust the tension to give the average speed of the engine which results in the desired amount of slack. The instantaneous speed of the engine varies to reduce short-term tension variations.

(c) Automatic slack control, in which the amount of required slack is calculated and set into a computer which controls the cable engine average speed to maintain the required amount of slack despite changes in ship speed. The fast-acting constant tension mode can still be retained to operate within the slow acting automatic slack control, and hence the effect of ship motion on the cable tension can be reduced.

The control system design was based on these principles and also incorporated a fail-safe feature which was intended to ensure that if a component or power source should fail during the operation, control would revert without interruption to one of the simpler modes. The primary control of the engine was therefore mechanical speed control operating from a lever connected by a mechanical linkage to the stroke of the hydraulic pump which drives the engine.

Improving on this to enable a more precise adjustment of speed is the closed-loop speed control system in which a handwheel, operating through clutches and gear reductions, drives a dc-excited potentiometer, the output of which is compared with a tachometer signal from the cable engine to yield an error signal capable of operating the pump stroke control via a servo valve and linkage. The basic tension control was constructed in a similar manner, but in this case the feedback signal is obtained from a load cell arranged to measure cable tension. Again the error signal adjusts engine speed via the servo valve. Fig. 18 is a block diagram of the cable engine control system.

The manual inputs for these three control arrangements are all in close proximity to the cable engine, and remote operation of the latter two systems is obtained by means of control motors activated by push buttons on the console in the cable control center. When remote operation is desired, the motors are clutched into the systems in place of the handwheels. All necessary information regarding the behavior of the engine or cable is transmitted to the cable control center by selsyn signals, or as dc voltages, and displayed on the cable payout control console as shown in Fig. 17.

Once control of the engine, in either the speed or tension modes, is

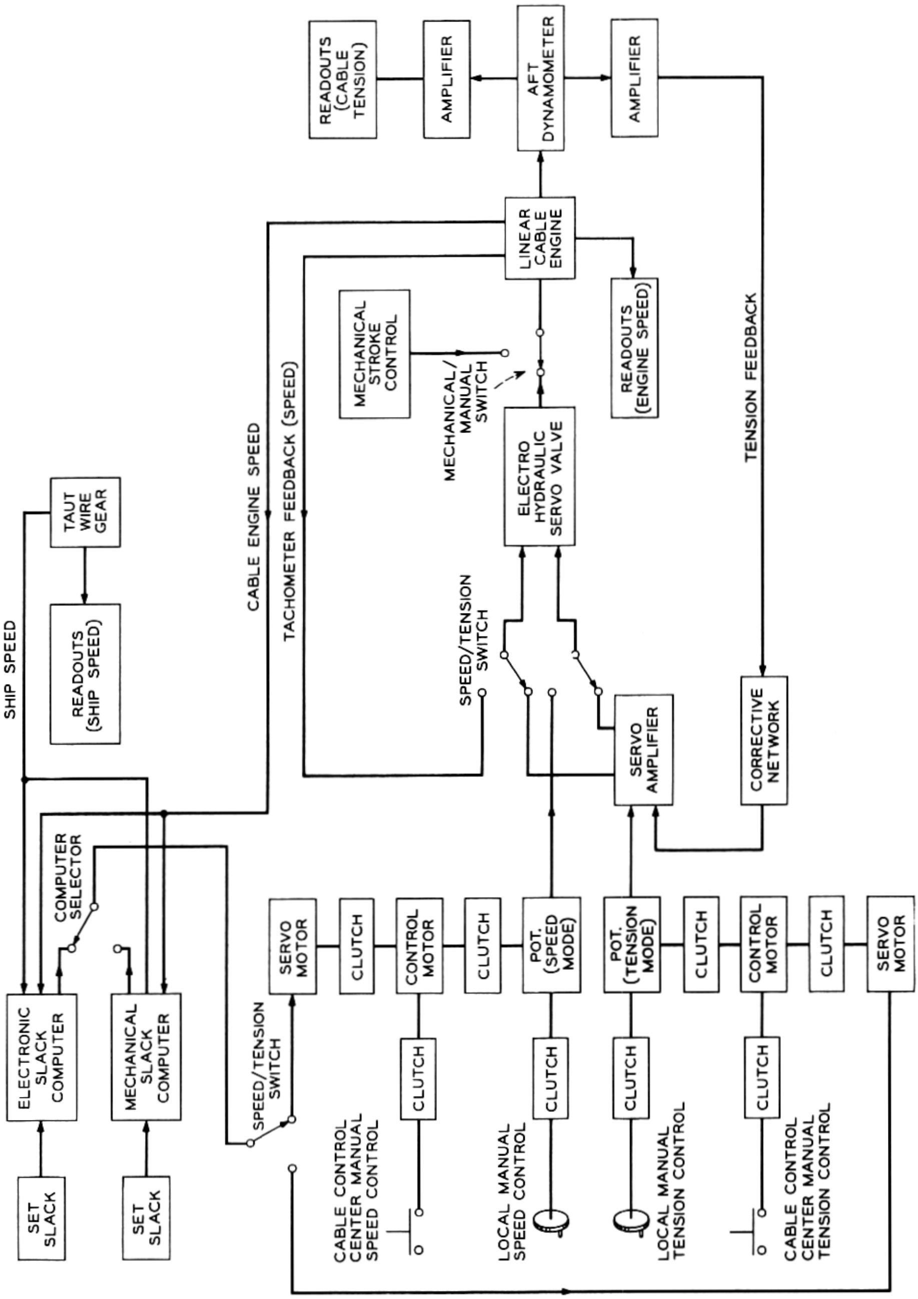


Fig. 18 — Cable engine control system.

transferred to the cable control center, the automatic slack mode can be superimposed on the system. Now the operator is merely required to set the required slack as the input to a computer, which then operates the speed or tension control potentiometers via a further series of servo motors, clutches and gear reductions.

The computer is, for reasons of simplicity and reliability, merely a refinement of the cone-and-cylinder type of mechanical computer used for many years on cable ships and miniaturized for convenience. As a standby a proprietary electronic computer is also available in the cable control center, and control could be readily transferred to this if necessary.

A separate unit was provided in the cable control center for the sole function of providing a continuous indication of cable mileage and taut wire mileage. This unit, which is shown in Fig. 17, operates independently of the other systems and is actuated by small pulse generators attached to each engine. The pulse generating units transmit one pulse for each fathom of cable or wire to an assembly of stepping switches. The accumulated count is displayed on "Nixie" readouts. Remote repeaters with identical readouts are also provided in the test room, on the bridge, and on the forecastle. This type of presentation was selected to avoid the noise of counting relays in the test room and on the bridge, and to allow all units to be zero-set or reset simultaneously from one location.

VII. CONCLUSION

The cable engines and control system were checked out during several short crew-training cruises soon after delivery of the ship. Cable and repeaters were payed out by the linear cable engine at speeds up to 8 knots in all control modes. The drum engines successfully handled armored cable, armorless cable, repeaters, grapnel rope, chains, shackles and other hardware.

Note Added in Proof

As of May 15, 1964, the linear cable engine has payed out approximately 8200 nm of SD cable systems.

APPENDIX A

Cable Gripping

Consider an armorless coaxial submarine cable having a high-strength steel central member surrounded by a coaxial transmission structure consisting of thin copper sheet and polyethylene dielectric. An axial tensile load on the strength member can only be dissipated by axial shear forces applied to the external surface and transmitted through the transmission structure to the central strength member. The magnitude of the maximum allowable axial shear force per unit length will be limited because the shear stresses in the copper and polyethylene, and particularly on the interfaces, must be kept to a minimum. For a given allowable maximum shear force per unit length, the linear cable engine which makes most efficient use of gripping length is that which applies a uniform shear force per unit length to the cable.

Assume that a linear cable engine can be designed so that the tension in the cable is dissipated uniformly in a gripping length λ. The tension distribution will then be as shown in Fig. 19. As a basis for the design of a cable engine which would achieve this tension decay, it is of interest to calculate the displacement of the cross section of the center conductor at the high-tension end of the machine.

The axial strain in the center conductor is given by

$$\epsilon = T/EA \qquad (1)$$

where

T = local tension

E = Young's modulus

A = cross-sectional area of the strand.

If the axial strain is integrated from the point in the cable at which the tension is zero, we find that the cross section at the high-tension end will move, with respect to the assumed stationary cross section at the zero tension point, an amount given by

$$u = \tfrac{1}{2}\epsilon_{\max}\lambda \qquad (2)$$

or

$$u = T_o\lambda/2AE. \qquad (3)$$

With a knowledge of the cable properties the maximum displacement could be calculated from (3). To avoid becoming involved in the details of the cable design it is more convenient to work directly with (2) and the fact that the strength member will be composed of "improved plow steel." Since this type of steel has a tensile strength of 300,000 psi and its Young's modulus is 30 million psi, the maximum strain in the strength member will reach 1 per cent under the ultimate load. For lower tensile

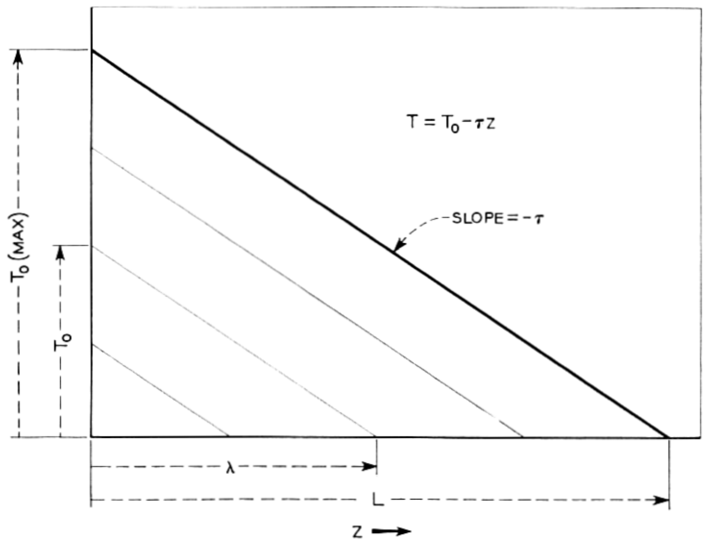


Fig. 19 — Uniformly dissipated cable tension distribution.

loads the maximum strain in the cable will be given, approximately, by

$$\epsilon_{\max} = \frac{T_o}{T_{o(\text{ultimate})}} \cdot \epsilon_{(\max)(\text{ultimate})} \cdot \tag{4}$$

Equation (1) can now be put into the form

$$u = \frac{1}{2} \frac{T_o}{T_{o(\text{ultimate})}} \cdot \epsilon_{(\max)(\text{ultimate})} \cdot \lambda \tag{5}$$

and the total displacement can be evaluated for a given operating tension, ultimate tension, and gripping length. The gripping length, in turn, is determined by the allowable shear force per unit length.

Fig. 20 is a plot of normal stress on the center conductor of an armorless cable having bonded interfaces and subjected to a tension T on a V-groove sheave of radius R and V-groove angle φ. In the special case where the V-groove angle equals 180°, the two normal reaction forces on the cable surface coalesce into one and we have an ordinary cylindrical drum. The calculations were actually made for this case, and the 90° situation was obtained by superposition.

For both the 90° V-groove and the plain cylindrical drum there are negative normal stresses on the center conductor in certain regions. Unless the cable has a bonded interface with sufficient strength to tolerate

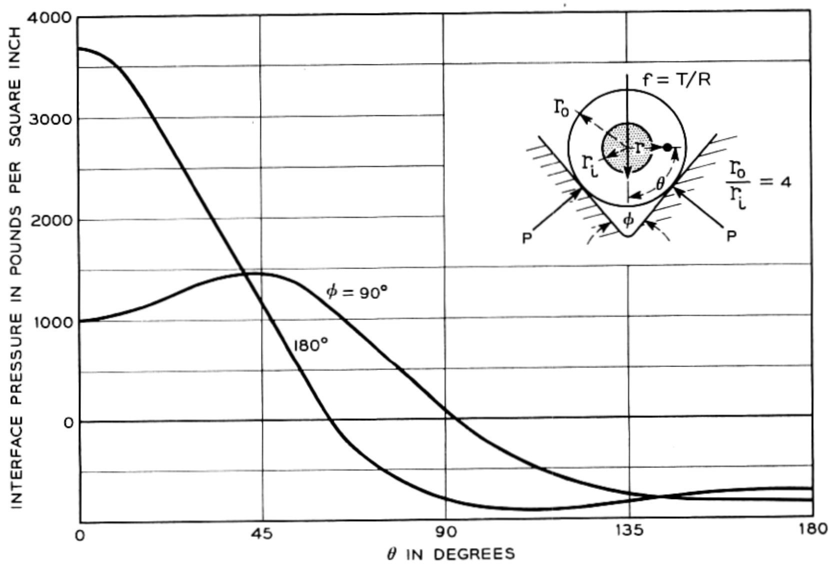


Fig. 20 — Center conductor normal stress — one V-block.

the tensile stresses or there is a shrink pressure on this interface equal to the maximum tensile stress, there will be separation and of course the calculated stress distributions will no longer apply.

Fig. 21 depicts the center conductor normal stress distribution for the case of two symmetrically located V-blocks which represent the gripper blocks of a linear cable engine. In this case the radius of the equivalent V-groove sheave has gone to infinity and the finite cable tension can give no normal component of force. The two blocks must therefore be squeezed together. It is to be noted that the normal stress on the center conductor interface does not become tensile anywhere around the circumference. This type of gripping will therefore not cause radial separation of an armorless cable with unbonded interfaces, a polyethylene dielectric and a ratio of outside radius to center conductor radius of four.

The cable gripping forces which are shown as concentrated in Fig. 21 will actually cause slight flat spots to develop on the cable and the contact stress distribution will be semielliptical, as shown in Fig. 22. The pressure distribution around the outer periphery of the cable can be represented by a Fourier series which will have a constant term given by the average pressure distribution which, in turn, is simply the sum of the four contact forces divided by the outside circumference. In addition there will be a series of sinusoidal terms. If the center conductor is as-

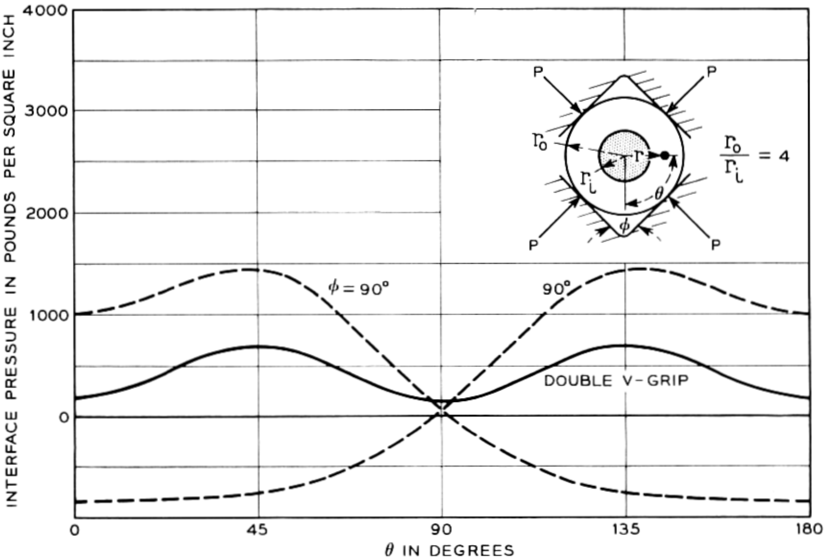


Fig. 21 — Center conductor normal stress — two V blocks.

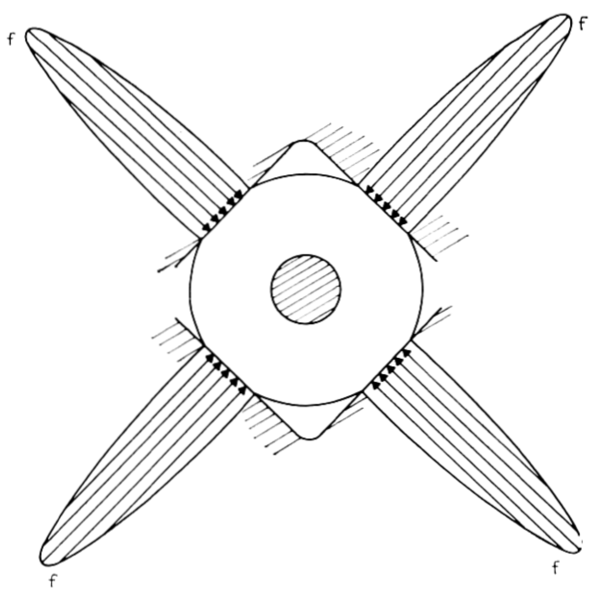


Fig. 22 — Contact stress distribution — two V-blocks.

sumed to be infinitely rigid and the polyethylene incompressible, the average pressure term will be transmitted undiminished to the center conductor. The various sinusoidal terms will be attenuated in some fashion, but in any case their contribution to the average pressure on the center conductor interface will be zero. The average pressure on the center conductor interface will therefore be equal to the average pressure on the external circumference of the cable.

The frictional shear stress which can be developed on any cylindrical interface will be given by the product of the coefficient of friction upon that interface and the normal pressure. The shear force per unit length which can be transmitted across the interface can be obtained by integrating the frictional shear stress around the circumference. If the coefficient of friction is independent of pressure, the frictional shear force at the condition of impending slippage will be given by the product of the coefficient of friction, the circumference, and the average interface pressure. For the center conductor interface we have

$$\tau_i = 2\pi r_i \mu_i P_{i(\mathrm{av})} \tag{6}$$

and for the outer surface

$$\tau_o = 4\mu_o f \tag{7}$$

where

$$P_{i(\mathrm{av})} = P_{o(\mathrm{av})} \tag{8}$$

$$P_{o(\mathrm{av})} = 4f/2\pi r_o \tag{9}$$

τ = slip value of frictional shear force per unit length

r = interface radius

μ = coefficient of friction

P = pressure

and

f = compressive force per unit length for the area of contact with each of the four plane faces of the gripper

and the subscripts "i" and "o" refer to inner and outer, respectively. Combining (6), (7), (8), and (9) yields

$$\tau_i/\tau_o = \mu_i r_i/\mu_o r_o\,. \tag{10}$$

Equation (10) indicates that slip will occur on the center conductor interface for values of shear force per unit length which will not cause the outer jacket to slip in the V-blocks unless we have

$$\mu_0/\mu_i \leqq r_i/r_o \tag{11}$$

Since the coefficient of friction on the center conductor interface will be approximately 0.16 and the ratio of outside to inside radius will be approximately four, inequality (11) calls for an external friction coefficient of less than 0.04. It was not considered feasible to maintain this value of coefficient of friction between the outer surface of the cable and the gripping elements of the cable engine. A shear limiting feature, which allowed axial motion of the gripper block at a shear force per unit length which would not cause slip on the center conductor interface, was therefore included in the track design.

Equations (6), (8), and (9) can be combined to give

$$\tau_i = 4\mu_i(r_i/r_o)f \tag{12}$$

or, for our case

$$\tau_i = \mu_i f \tag{13}$$

since the outside-to-inside radius ratio is four.

Experimental work on gripping of cable in double V-blocks has established 500 pounds per inch as the maximum allowable contact force. A design value of 250 pounds per inch was used in setting the allowable shear force per unit length. Substitution of this value and of the previously mentioned coefficient of friction of 0.16 into (13) gives

$$\tau_{(\text{shear limit})} \leqq \tau_{i(\max)} = 40 \text{ lb/in.} \tag{14}$$

The τ notation is used for the shear limiter setting because the effect of an ideal shear limiter is equivalent to slip of the jacket in a gripper block. The active gripping length is then

$$\lambda = \frac{T_o}{\tau_{(\text{shear limit})}} \tag{15}$$

and (3) and (5) can be written

$$u = \frac{T_o^2}{2AE\tau_{(\text{shear limit})}} \tag{16}$$

and

$$u = \frac{T_o^2 \epsilon_{(\max)(\text{ultimate})}}{2T_{o(\text{ultimate})}\,\tau_{(\text{shear limit})}} \tag{17}$$

respectively. For the cable in question we have

$$T_{o(\text{ultimate})} = 16{,}000 \text{ lb}$$

$$\epsilon_{(\max)\ (\text{ultimate})} = 0.01$$

and

$$\tau_{(\text{shear limit})} = 40 \text{ lb/in.}$$

whence

$$u = \frac{T_o^{\,2}}{128 \times 10^6} \text{ in.} \tag{18}$$

for T_o in pounds.

If this cable were gripped and tensioned in such a manner as to achieve the linear decay of Fig. 19 for all tensions up to the nominal ultimate, (18) would give

$$u_{(T_o=16{,}000 \text{ lb})} = 2 \text{ in.}$$

For smaller working tensions the required shear limiter travel decreases parabolically and

$$u_{(T_o=8{,}000 \text{ lb})} = \tfrac{1}{2} \text{ in.}$$

APPENDIX B

*Bearing Reliability**

The bearing industry defines the life of bearings as that period of service which is limited by fatigue phenomena. The fatigue-limited life terminates in what will be referred to as the "initial failure" of a bearing. It is characterized by fatigue of the material due to repeated stress under rotation resulting in flaking and ultimately in spalled areas on the rollers and/or races of the bearing. This initial failure will not make a bearing inoperable for cable engine application but will merely tend to increase the noise level and possibly the rotating torque to a small degree.

When a bearing is operated after initial failure the spalled areas increase in size until ultimately the bearing is incapable of performing its design function. It may "lock-up" so that the cup and cone cannot rotate with respect to each other, or the rollers and cage may disintegrate, allowing large radial displacements of cup with respect to cone. In either of these cases the bearing will be said to have failed catastrophically.

Since castrophic failure is a result of running an initially failed bearing

* Furnished by G. J. Levenbach and T. N. Grogean.

beyond initial failure, it is necessary to discuss the fatigue life of bearings as a basis for discussion of the subsequent catastrophic failure.

B.1 *Initial Fatigue Failure*

The following discussion is based on the assumption that the bearings are run under ideal conditions. We assume proper lubrication, the absence of foreign materials, isolation from the atmosphere, and proper installation and handling.

With these assumed running conditions, the probability that no initial fatigue failure will occur in a bearing running for a given time is contained in a formula given by Palmgren:[2]

$$\mathrm{Log}_{10} \frac{1}{S} = (L_s)^b \log_{10} \frac{1}{0.9} \tag{19}$$

where

S = probability of a bearing exceeding a life L_s

L_s = life measured as a fraction of the life exceeded by 90 per cent of population of bearings (so-called B_{10} life)

b = shape constant in the Weibull distribution.

Palmgren's formula is based on extensive experimental evidence that the fatigue failure time distribution can be described by a Weibull distribution.

For a sample of n bearings the probability, S_n, that all of them exceed L_s is derived from (19) and takes the form

$$S_n = \exp\left[-(L_s)^b \cdot \frac{n}{a}\right] \tag{20}$$

where a is a conversion constant given by

$$a = \frac{\log_{10} e}{\log_{10} (0.9)^{-1}} = 9.5$$

and e is the base of the Naperian logarithm.

The shape constant b is a property of a specific type of bearing.[3] Palmgren notes that for commonly used bearing steels b has values in the range of 1.1 to 1.5.

Data obtained from 30 bearings fatigue-failed by Timken Roller Bearing Company can be used to estimate the shape factor b. When the data listed in Table II are plotted on Weibull probability paper the sample distribution should appear as a straight line.[4] The slope of the line

TABLE II — INITIAL FAILURE PATTERN OF TEST LOT OF 30 LM11700 SERIES TIMKEN BEARINGS (TIMKEN TEST NUMBER 151.3-G)

Number of bearings exceeding life L_s	29	23	16	10	3
L_s	0.64	2.80	5.00	9.0	9.90

gives an estimate of $b = 1.45$. Using (20), this can be used to calculate the probability that no bearing failure will occur in h hours of operation. This probability is plotted in Fig. 23.

The expected number of failures (n') in h hours can be determined by multiplying the probability of a bearing failure p by the total number of bearings n and noting that the probability of a bearing fatigue-failing is $p = 1 - S$.

$$n' = pn = (1 - S)n = \left[1 - \exp\left\{-(L_s)^b \cdot \frac{1}{a}\right\}\right] n. \qquad (21)$$

Equation (21) is plotted in Fig. 24 showing the expected number of failures vs hours of operation.

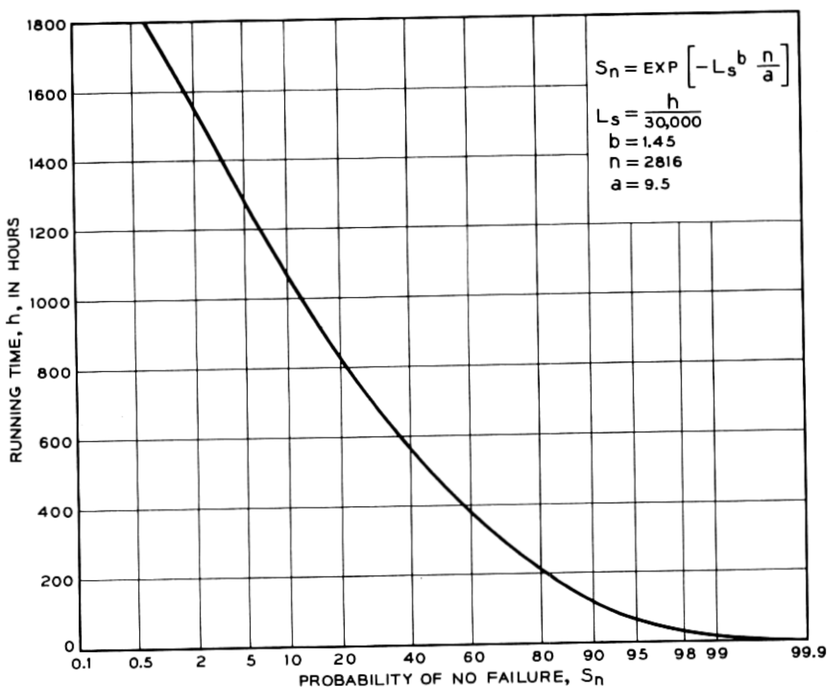


Fig. 23 — Probability of no bearing failure in l hours.

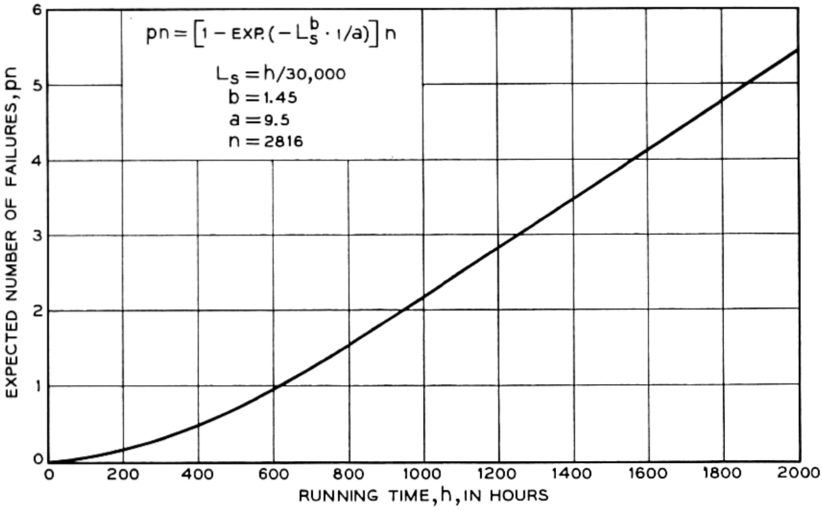


Fig. 24 — Expected number of failures in h hours.

Another useful result is the probability that there will be N or more bearing failures in a given number of hours. Basically this can be described by a binomial process. The binomial formula gives the probability of N failures of a total of n bearings. For easy computation, this can be approximated by using the pn in the previous discussion as the parameter in the Poisson distribution. The resulting relation shows

$$P = e^{-pn} \sum_{i=N}^{n} \frac{(pn)^i}{i!} \tag{22}$$

where P is the probability that there will be N or more failures in a total number n with pn the expected number of failures. Equation (22) is plotted in Fig. 25 for P vs N for various values of h.

It is seen from the previous discussion that there is a high probability that there will be initial failures in relatively short operating times. For example, in 300 hours, which is roughly the time for one ocean crossing, Fig. 23 shows that there is only 70 per cent probability that there will be no bearing failures; Fig. 24 shows that the probable number of failures is approximately 1; and Fig. 25 shows that there is a 5 per cent probability that there will be 2 or more.

With these facts known for bearings run under ideal conditions, it is necessary to accept the fact that there will be initially failed bearings in the cable engine during extended running periods. The possibility of propagation to catastrophic failure is therefore of interest.

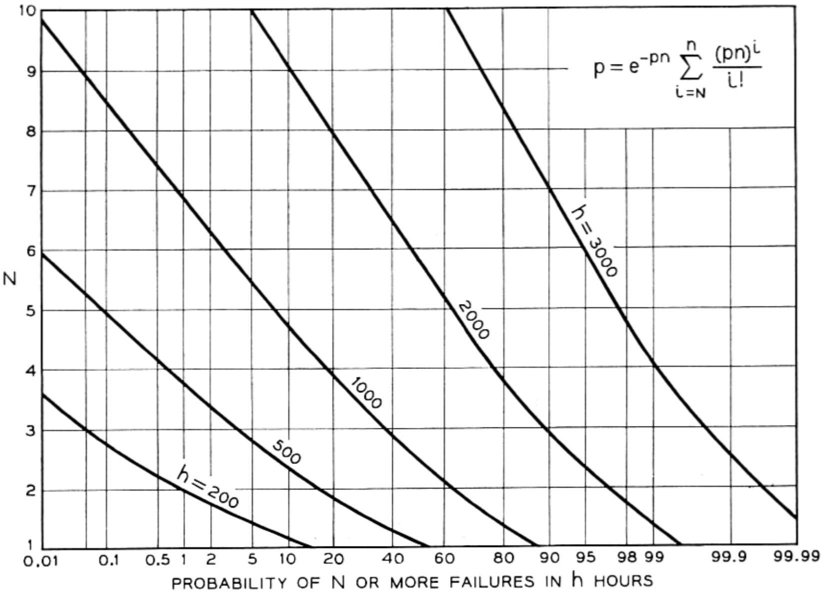


Fig. 25 — Probability of N or more failures for various hours of operation.

B.2 *Failure Propagation*

The bearing industry has not been greatly concerned with catastrophic failure; consequently, little information is available. At the request of Bell Laboratories, Timken Roller Bearing Company ran accelerated life tests on four bearings of the type considered here. The four bearings were run on their laboratory fatigue test machines at high speed under heavy load with oil lubrication. Two of the four bearings experienced what might be considered catastrophic failure at the end of 3.4 and 4.0 B_{10} life. The other two were still operable at the end of 4.7 B_{10} life.

In addition to this accelerated test result, Timken supplied Bell Laboratories with twenty bearings which had been run to the point of initial fatigue failure with oil lubrication. Eight of the twenty bearings were run under prototype loads at prototype speed by Western Gear Corporation and lubricated every 400 hours with Shell Darina A-X. The test was terminated after 3020 hours with little change in running properties. Pictures taken before and after the test indicate that the failure did not propagate to any considerable degree.

The remaining twelve fatigue-failed bearings were tested at Bergen Research Engineering Corporation. In this case they were initially greased with Shell Darina A-X and run continuously without further lubrication. At 5500 hours the driving torque on one roller had increased significantly, and the test rig was shut down. Inspection indicated that the grease had become dry and hard in both bearings of the offending roller. The remaining rollers were regreased and the test resumed.

At 5850 hours, it was observed that several seals had been forced out against the ends of the rollers, apparently because dried grease had blocked the vent fittings. The test was therefore terminated at 5850 hours and all twelve bearings were returned to the Timken Roller Bearing Company for examination. Photographs taken before and after the 5850-hour failure propagation test were compared. Although there had been a considerable increase in spalled areas in some cases none of the bearings could be said to have failed catastrophically.

B.3 *Interpretation of Results*

Because of the small amount of failure propagation data available, it is not possible to develop a failure distribution theory similar to that which applies to the initial fatigue failure. It might, however, be of interest to look at the results under the somewhat tenuous assumption that the times to catastrophic failures are exponentially distributed.

(a) Assuming the accelerated life test of the four bearings terminated at 4.0 B_{10}, which is conservative, we estimate the mean life for catatrophic failure to be[5]

$$\theta_a = 231{,}000 \text{ hours} = 7.7\ B_{10}\,.$$

(b) Assuming one failure in the eight-bearing test to have occurred at 3020 hours provides an estimate of

$$\theta_b = 24{,}000 \text{ hours} = 0.8\ B_{10}\,.$$

(c) Assuming one failure in the 12-bearing test to have occurred at 5500 hours provides an estimate of

$$\theta_c = 66{,}000 \text{ hours} = 2.2\ B_{10}\,.$$

Taking the lowest estimate at its face value would yield a probability that an undetected initial fatigue failure would not propagate in the next 300 hours to the point of catastrophic failure as

$$\exp\,(-t/\theta) = \exp\,(-300/24{,}000) = 0.99.$$

This result is conservative in that there was actually no catastrophic failure in the eight-bearing test.

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