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AUGUST 1983

THEORETICAL STUDY, DESIGN AND ANALYSIS OF AN IMPROVED SHUTTLE CAR CABLE REEL

Contract J0318086
Jeffrey Mining Machinery Division
Dresser Industries, Inc.

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An hydraulic-powered reel system to minimize peak tension transients in shuttle car trailing cables has been theoretically designed and analyzed using a computer. A single spring-loaded take-up sheave stores or gives up cable to maintain tension within specified limits. Controls have been incorporated to shut down the vehicle before all cable is removed from the reel and to de-energize the cable in the event of a reel/cable jam. Full-scale hardware configuration drawings sufficient for manufacture of a test system have been completed.

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FOREWORD

This report was prepared by Jeffrey Mining Machinery Division, Dresser Industries, Inc., under United States Bureau of Mines Contract J0318086. The contract was initiated under the Coal Mine Health & Safety Research Program. It was administered under the technical direction of the Pittsburgh Research Center, with Mr. MR Yenchek acting as Technical Project Officer. Mr. William R. Mundorf was the Contract Administrator. This report is a summary of all work completed under the contract during the period of August 1981 to August 1983 and was submitted by the authors in August 1983.

Program Management by Jeffrey Mining Machinery Division was under the direction of PW Meisel and RJ Gunderman. Contract administration by Jeffrey Mining Machinery Division was performed by GF Woodward.

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EXECUTIVE SUMMARY

1.1 Project Objectives

The objective of this contract was to design an hydraulic-powered cable reel system that minimizes peak tension transients in the shuttle car trailing cable, shuts down the vehicle before all the cable is pulled off the reel, and provides electrical protection in the event of a reel/cable jam. A detailed listing of system design requirements is provided as Section 2.9. Engineering detailed drawings and system sketches adequate to manufacture a test system have been completed.

1.2 Conclusions and Recommendations

This system is feasible and will provide a substantial savings (at the mine level) in cable handling and vehicle down time (due to cable-related problems).

The system, as designed, is suitable for high-seam vehicles. A high-seam vehicle was chosen because it could accept the added height of a retrofit system better than the lower-seam vehicle. Higher vehicles also have a higher vehicle speed and cable reel inertia, which is the worst case for cable reel drive systems.

The computer model shows the system as designed is stable but more marginal than would be ideal. It also shows system stability would increase when adapted to lower vehicles.

Because the actual mechanical components cannot duplicate the computer model, it is recommended a prototype be fabricated, installed on a high-seam vehicle and tested underground to prove the principle. The system could then be designed into the frame of a lower vehicle with certainty of success. However, if the system is designed and tested on a lower-seam vehicle, it may not be stable on a higher vehicle.

SECTION II

DESIGN CONSIDERATIONS

2.1 Typical Existing System Performances

Typical existing cable reel systems are based on the use of a fixed displacement pump and motor and a two-pressure "cable reel" valve. This valve consists of two relief valves and a flow-sensing valve, which can shift and allow the valve to operate at either of the two pressure settings.

Oil provided by the pump is routed through the "cable reel" valve to the motor. If the motor is turning at a speed at which it will not allow all of the oil flow available from the pump to pass through it, the excess oil is diverted to the tank through the "cable reel" valve. The pump and motor are sized so that this is always the case.

If the pump is driving the motor, e.g., when the vehicle is reeling in cable, the "cable reel" valve is in its higher pressure setting. This allows adequate torque to drive the reel to pick up cable while overcoming inefficiencies in the hydraulic system. If the pump and motor are running in opposite directions, e.g., when the vehicle is reeling out cable, the motor acts like a pump; and both pump and motor flows are routed through the "cable reel" valve back to the tank. In this mode, the valve shifts to the lower pressure setting to minimize the torque required to drive the motor. Normally, this setting is only high enough to insure that there is adequate tension on the cable to keep it from fouling the vehicle's wheels while backspooling, and to adequately brake the reel as the vehicle brakes.

"Cable reel" valves currently on the market are of two types. The first, and by far the most common, is the high-pressure idle system. In this system, the tension on the reel is determined by the high-pressure relief when the vehicle is at rest. This enables the reel to have maximum torque available to accelerate, should the vehicle start moving toward the cable (pay-in). It has a drawback, however, in that the cable tension must overcome the inefficiencies in the hydraulic drive, the high-pressure setting, and the inertia of the reel when the vehicle starts to accelerate away from the cable (pay-out). Once this acceleration starts, however, the low-pressure relief is brought on line, and tension is determined only by the lower hydraulic setting and the inertia effects.

The second type of "cable reel" valve is the low-pressure idle system. With this, not as much tension in the cable is required in the pay-out mode when first accelerating. There are still the hydraulic inefficiency and acceleration factors, but the system pressure is reduced. When accelerating in the opposite direction (pay-in), however, the low-pressure setting must be high enough to start the reel turning, at which point the higher relief pressure can take over to accelerate the reel. This makes it necessary to have a relatively high low-pressure setting and causes higher pay-out tensions in all but the accelerating mode.

In testing the previously-discussed systems, it was found that sharp tension spikes were found in both, when traveling from pay-in to pay-out modes (passing the tie point). The spikes were caused by a number of factors listed below, in order of importance:

1. Cable reel inertia (acceleration/deceleration factors).

2. Location of tie point relative to car (acceleration/deceleration factors).

3. Inability of hydraulic systems to anticipate tie point. (No matter how quickly they react, they still lag the event.)

4. Cable reel pressure settings.

Figures 1 through 3 illustrate the kind of performance exhibited by current production systems, using a Jeffrey 4015 Shuttle Car, tramming past the tie point at a speed of 6 MPH and a passing distance of 5 feet. Note that while pay-out tension levels varied substantially, the pay-in and peak torques were almost identical.

These are typical examples, although a wide range of tests were run using different vendors' "cable reel" valves, low-efficiency gear and high-efficiency piston motors. While these tests were conducted prior to the start of this contract, it is worth noting there was very little variation in different systems' performance.

2.2 Reduction of Tension Transients in the Cable

Review of the data represented in Figures 1 through 3 makes it evident that steady-state operating tensions in all cases are not unreasonable. Even the single-pressure relief valve averaged only a little over 300 pounds in the worst steady-state condition. The maximum allowable tension deemed practical for a #4-3 conductor cable was 390 pounds. The rationale for this conclusion was:

650 lbs.	- Average ultimate strength of splice in a single conductor of #4 wire
1950 lbs.	- Three times above for 3-conductor cable
975 lbs.	- 1950 ÷ 2 = estimated yield strength
585 lbs.	- 975 X 0.6 = estimated fatigue strength
390 lbs.	- 585 ÷ 1.5 safety factor

However, peak tensions generated by acceleration forces and reel inertia prove to be impractical for the passive-type drive systems on existing machines. In an effort to optimize inertia variations and tension variations caused by the layering of cable on the reel with a conventional cable drive system, the following table was generated (Figure 4):

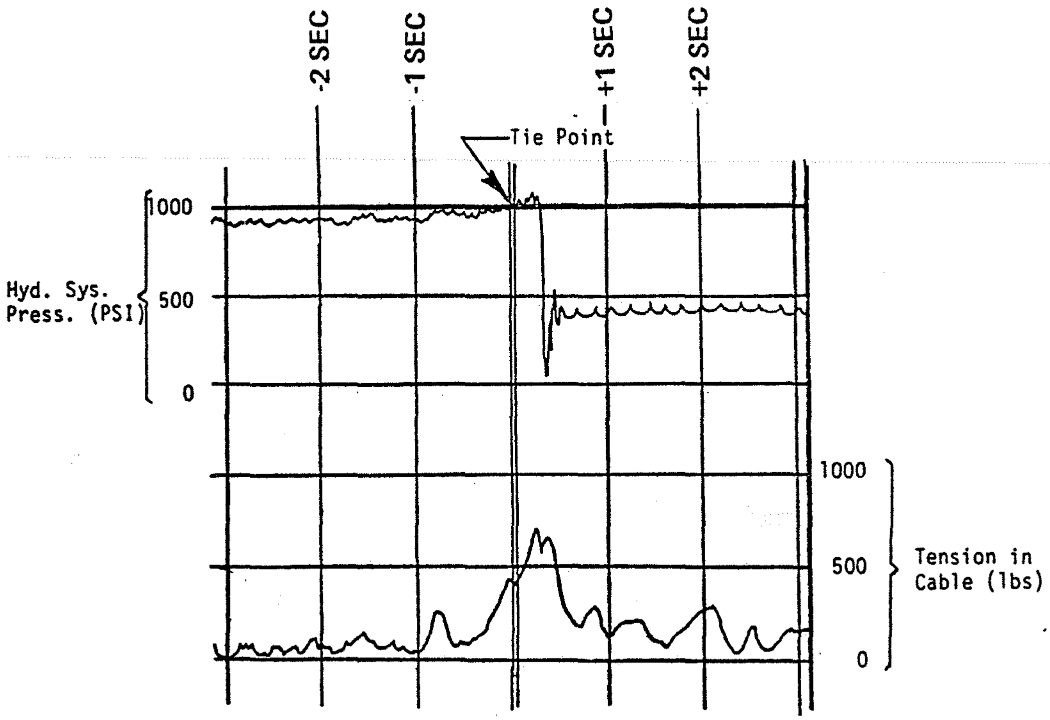


FIGURE 1 TYPICAL HIGH PRESSURE IDLE SYSTEM PERFORMANCE

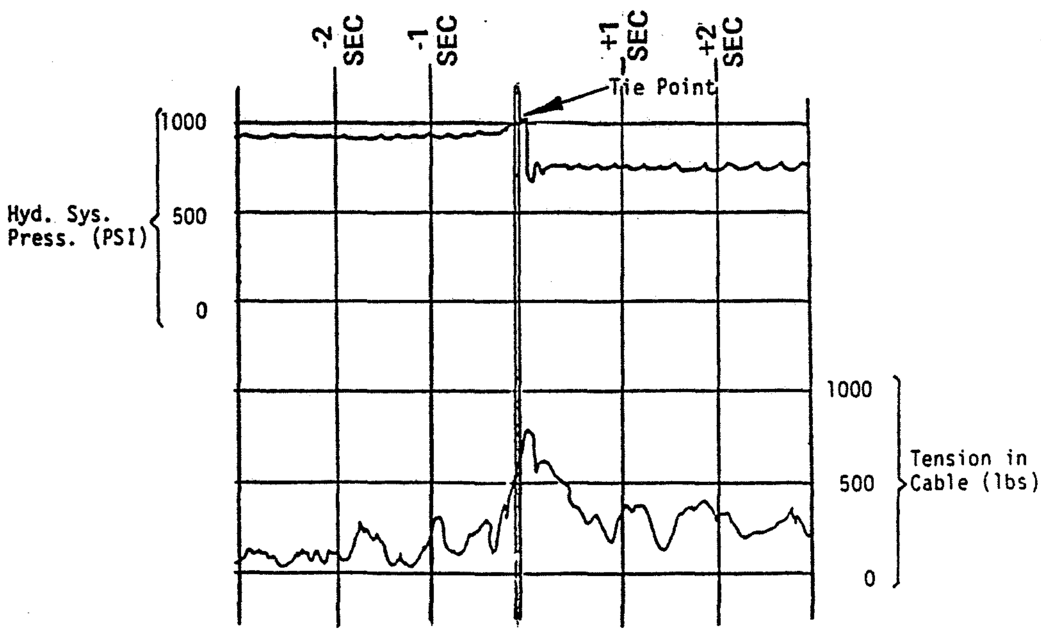


FIGURE 2 TYPICAL LOW PRESSURE IDLE SYSTEM PERFORMANCE

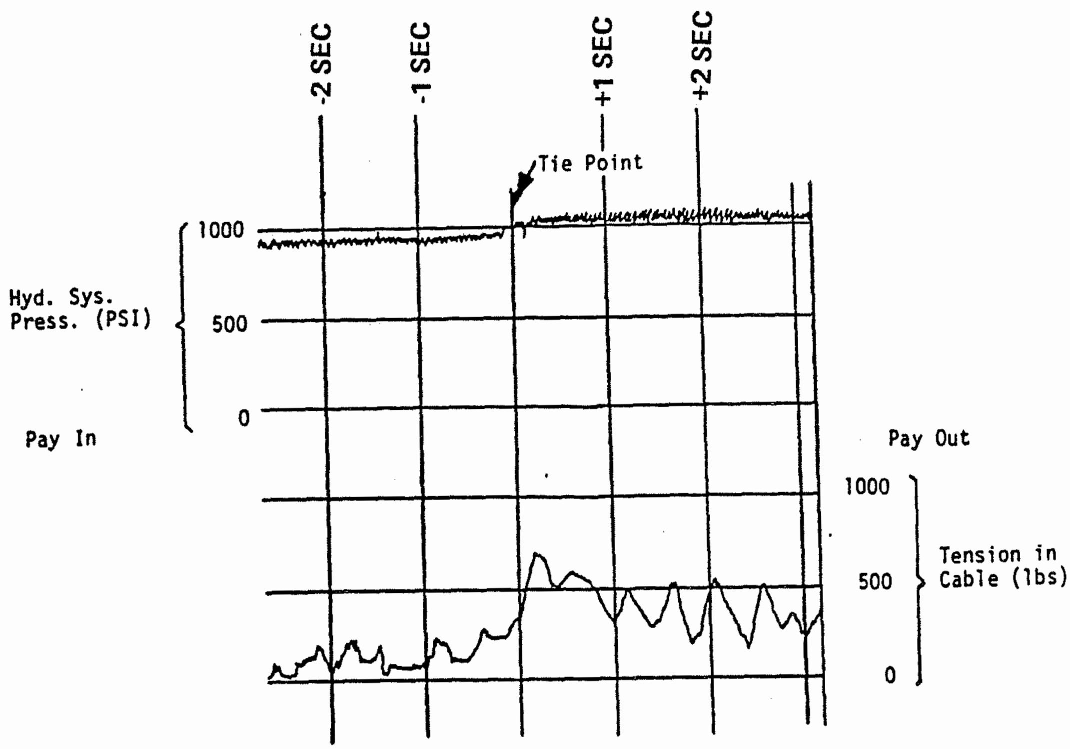


FIGURE 3 SINGLE PRESSURE SYSTEM PERFORMANCE

CABLE REEL COMPARATIVE DATA

	REEL O.D. IN.	REEL I.D. IN.	REEL CAPACITY IN.^{3}	I	Reeling In: CABLE TENS O.D. LBS	Reeling In: CABLE TENS I.D. LBS	CABLE TENS TIE OFF SPIKE LBS	Reeling Out: CABLE TENS O.D. LBS	Reeling Out: CABLE TENS I.D. LBS	ACCL RUN HI PRES FT/SEC^2	ACCL STALL HI PRES FT/SEC^2	ACCL STALL LO PRES FT/SEC^2	HIGH PRES SETTING PSI	LOW PRES SETTING PSI	CABLE SPEED I.D. MPH
Acceptable limit			14000 min.		390 max	390 max	390 max	390 max	390 max	7 min	4.5 min				8.8 min
	40"	24"	14074	52.6	133#	222#	353#	115#	192#	4.2	2.5	1.5	800	500	11.5
	40"	24"	14074	52.6	167#	278#	388#	161#	269#	5.3	3.1	2.2	1000	700	11.5
	40"	24"	14074	52.6	200#	334#	421#	207#	345#	6.3	3.7	2.8	1200	900	11.5
	38"	20.5"	14074	45.7	176#	326#	378#	170#	315#	6.1	3.6	2.5	1000	700	9.9
	38"	20.5"	14074	45.7	194#	359#	396#	194#	359#	6.7	4.0	2.8	1100	800	9.9
	37"	18.5	14074	42.9	180#	371#	370#	174#	348#	6.5	3.8	2.7	1000	700	8.9
	37"	18.5"	14074	42.9	199#	397#	393#	200#	399#	7.1	4.2	3.0	1100	800	8.9
	37"	18.5"	14074	42.9	195#	389#	390#	194#	389#	7.0	4.1	3.0	1078	779	8.9
	36"	16.5"	14074	39.7	186#	405#	371#	180#	392#	7.0	4.1	2.9	1000	700	7.9
	36"	16.5"	14074	39.7	176#	385#	361#	167#	364#	6.7	3.9	2.7	950	650	7.9
	34.5"	13"	14074	35.0	155#	411#	325#	134#	356#	6.4	3.7	2.3	800	500	6.2

☐ Indicates figure outside allowable limits

FIGURE 4 CABLE REEL COMPARATIVE DATA

Reel capacity is sized to carry 700 feet of #4 GGC cable, maximum tension was derived earlier, and acceleration factors were those determined necessary to keep most of the slack out of the cable, as the vehicle travels through a typical cycle.

The following four figures (Figures 5 through 8) graphically show the relative speed between trail cable and backspooler, as the vehicle trams through various mine features. Figure 5 shows the relative velocity when passing the tie point (with the cable tied off at different passing distances from the car). Figure 6 shows straight-ahead acceleration. Figure 7 shows a 6 MPH turn with the backspooler on the outside corner, while Figure 8 shows the opposite turn with the backspooler on the inside corner.

The nominal acceleration criteria listed on Figure 4 was determined as being the minimum necessary to fulfill all of the necessary acceleration requirements, with only a minimum of slack. In the case of Figure 5, it is necessary that the mine locate the tie point approximately 10 feet back in a cross cut, if backspooling (estimate 16 feet passing distance). Note also from Figure 4 that a 37 inch O.D. by 18.5 inch I.D. reel is about optimum (ignoring the fact that the cable must also overcome the inertia of the reel when backspooling or any time the vehicle is accelerating in the pay-out mode).

Because the reel must accelerate against the driving force of the pump, substantial tension transients exist (as exemplified by the first three figures). Tension spikes are also caused by the inability of the reel to accelerate quickly enough to keep up with actual requirements. This causes slack, followed by snapping of the cable when the reel does catch up. These last tension spikes are difficult to measure and vary greatly with conditions. Estimates have been made and some testing done on other contracts, and it appears that spikes in the 1000-1500 pound range are not uncommon.

2.3 System Performance Requirements

The primary function of this new type of cable-handling system is to keep cable tensions within specified limits throughout the operational cycle. The maximum tension should be as low as possible and definitely not exceed the 390 pound limit derived earlier. The minimum tension should, if possible, not drop below that required to keep the cable from fouling the vehicle's tires. This amounts to 115 pounds for a #4 cable with a backspooler height of five feet and a 25 foot distance between backspooler and tire (see Figure 9). It should be capable of operating on a vehicle with a speed of six MPH and should store as much cable as existing shuttle car systems.

2.4 Retrofit to Existing Machines

Part of the objective of this project is to produce a system that could be fitted to an existing shuttle car with a minimum of

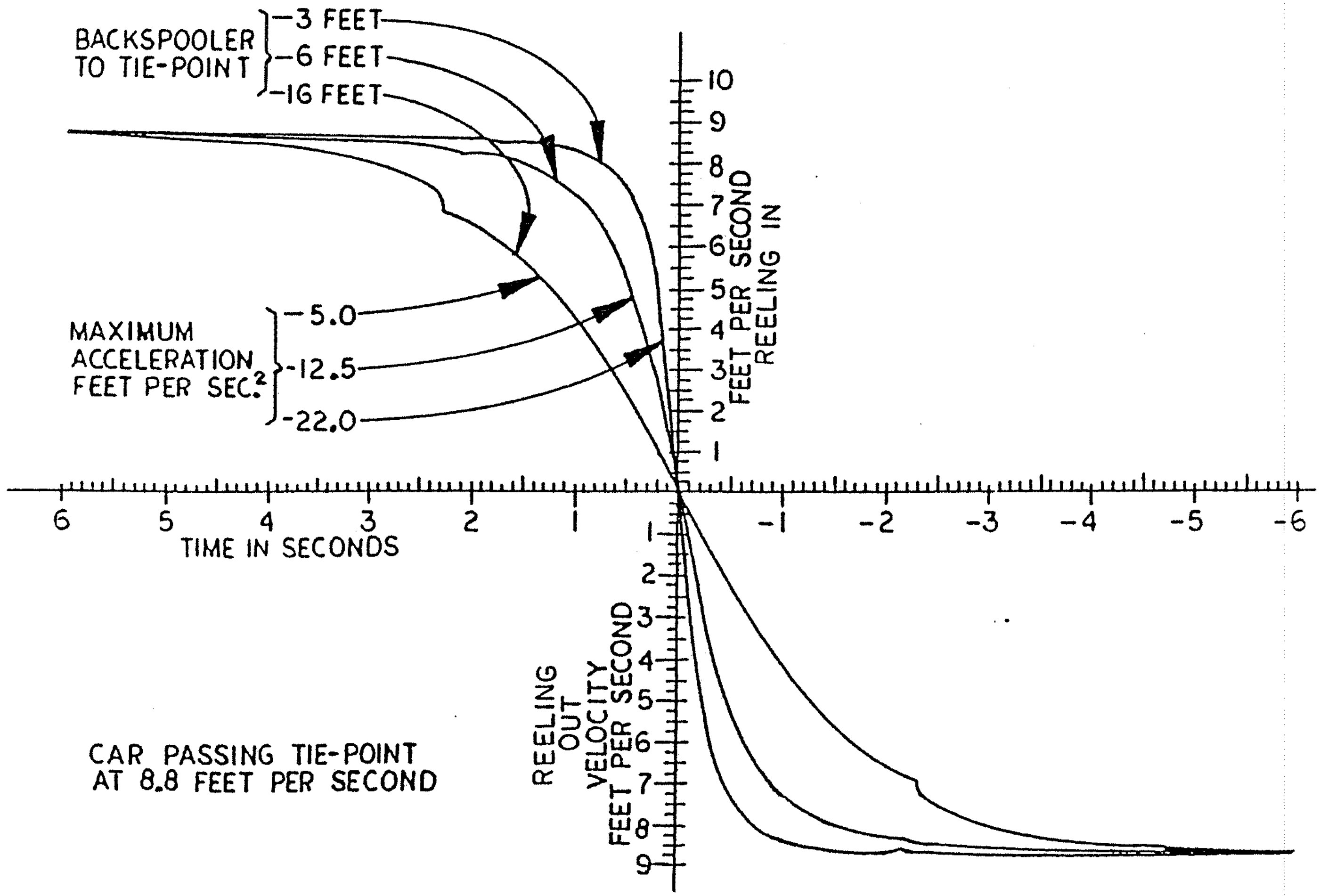


FIGURE 5 CABLE VELOCITIES AT TIE POINT

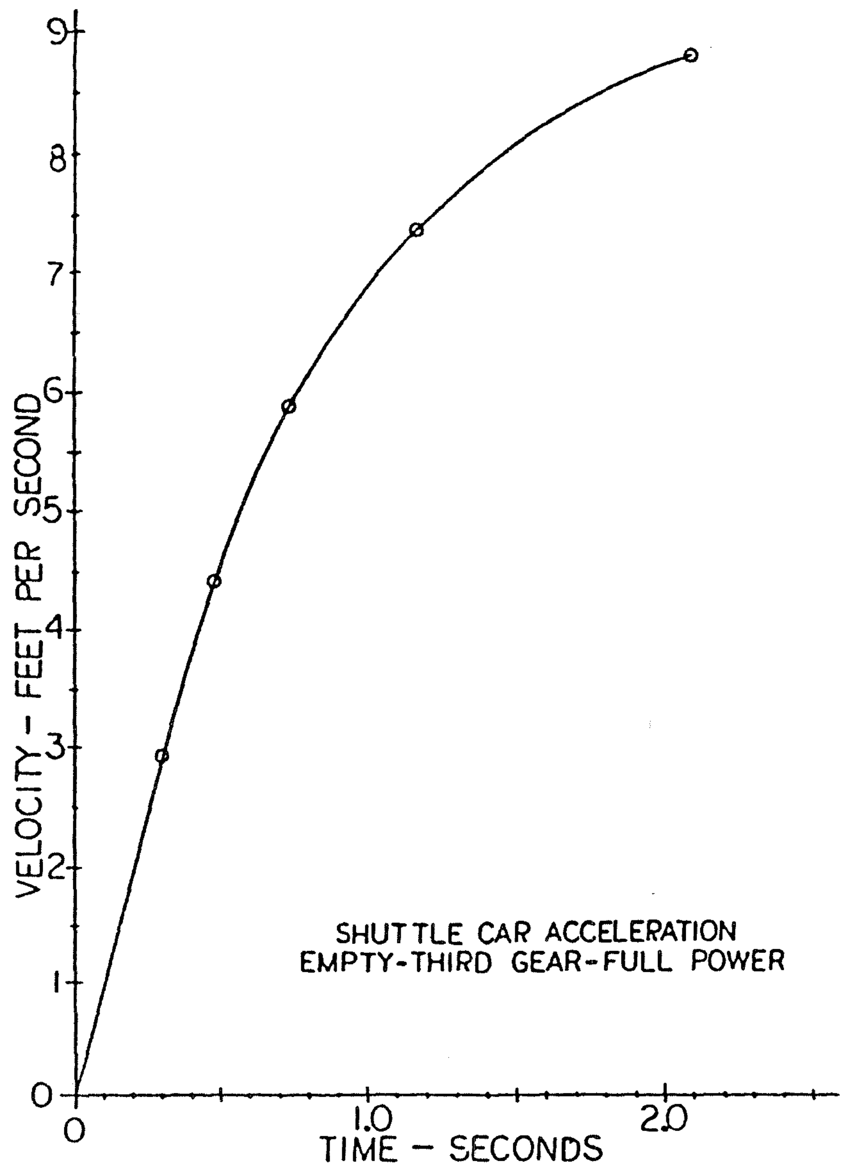


FIGURE 6 CABLE VELOCITIES - STRAIGHT ACCELERATION

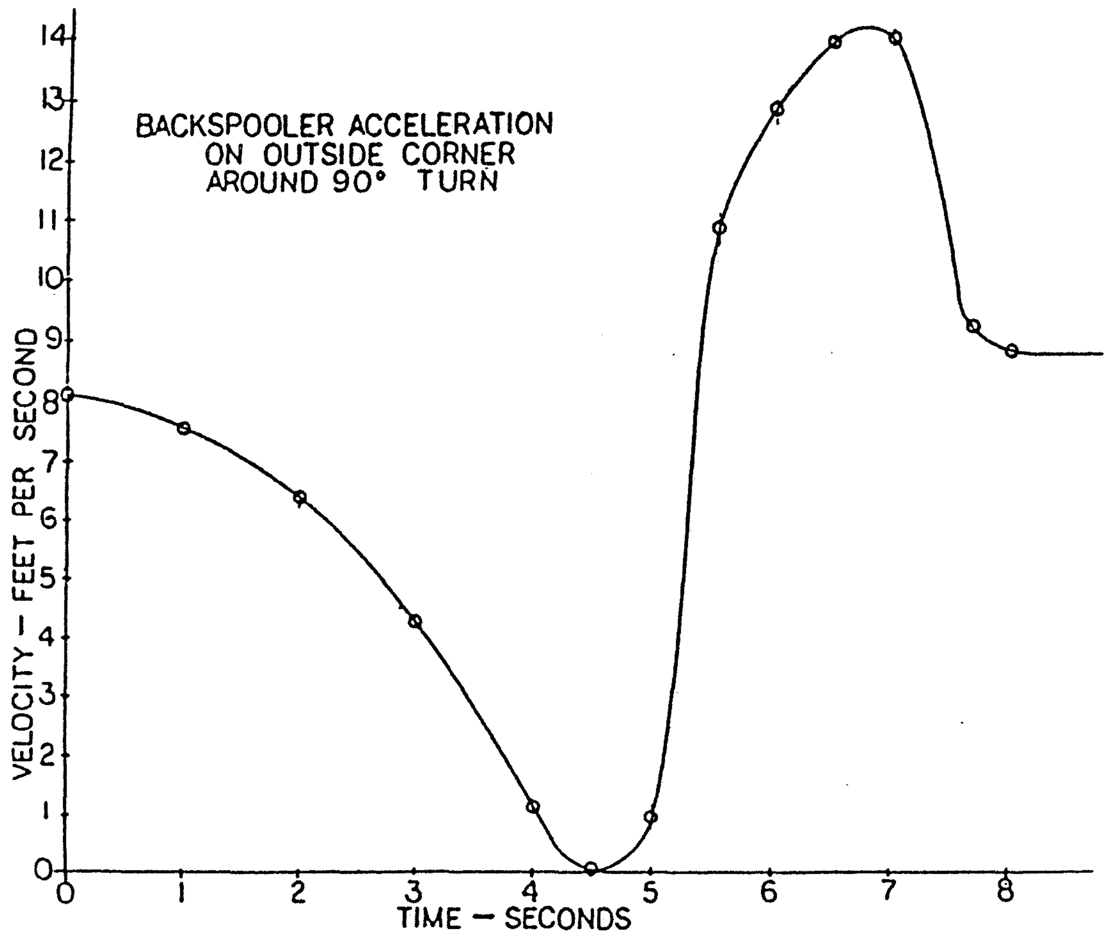


FIGURE 7 CABLE VELOCITIES - BACKSPOOLER ON OUTSIDE OF CORNER

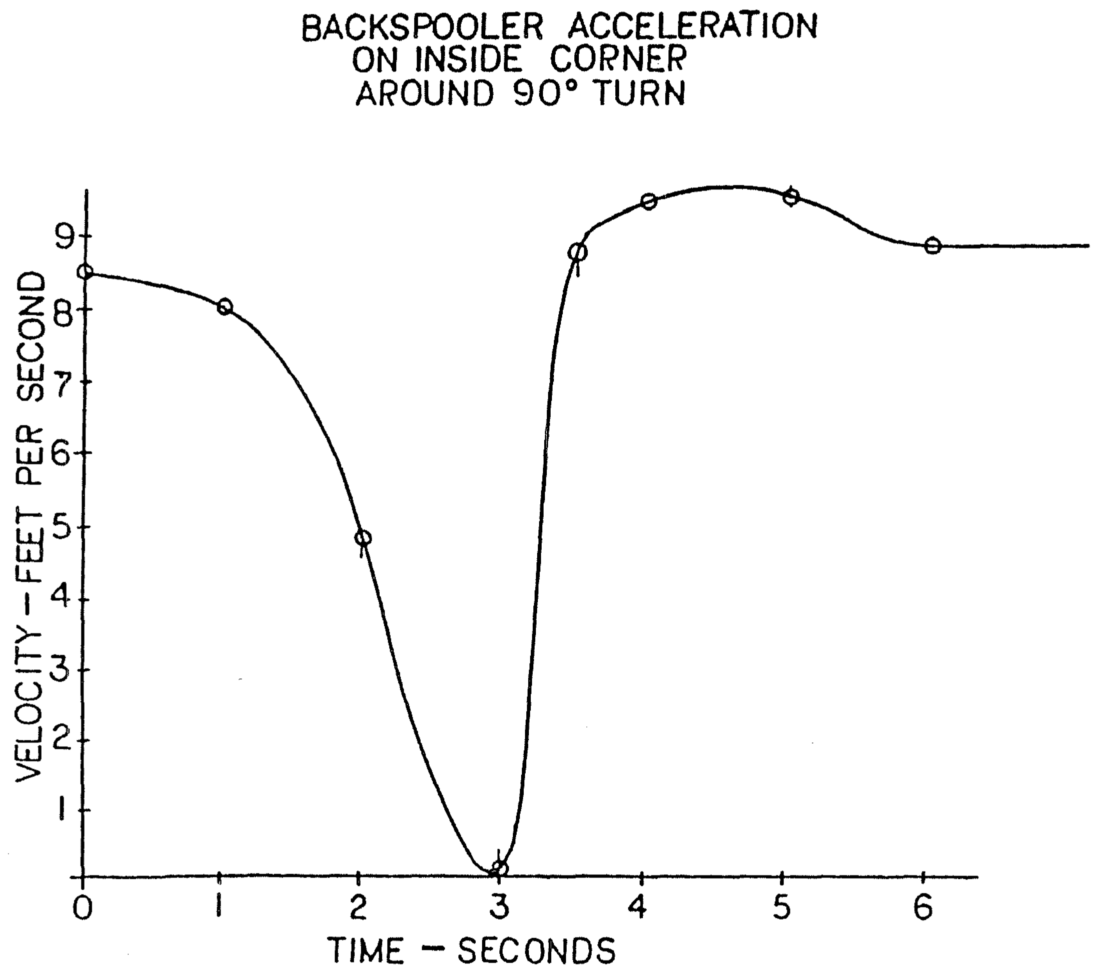


FIGURE 8 CABLE VELOCITIES - BACKSPOOLER ON INSIDE OF CORNER

CATENARY CABLE TENSION

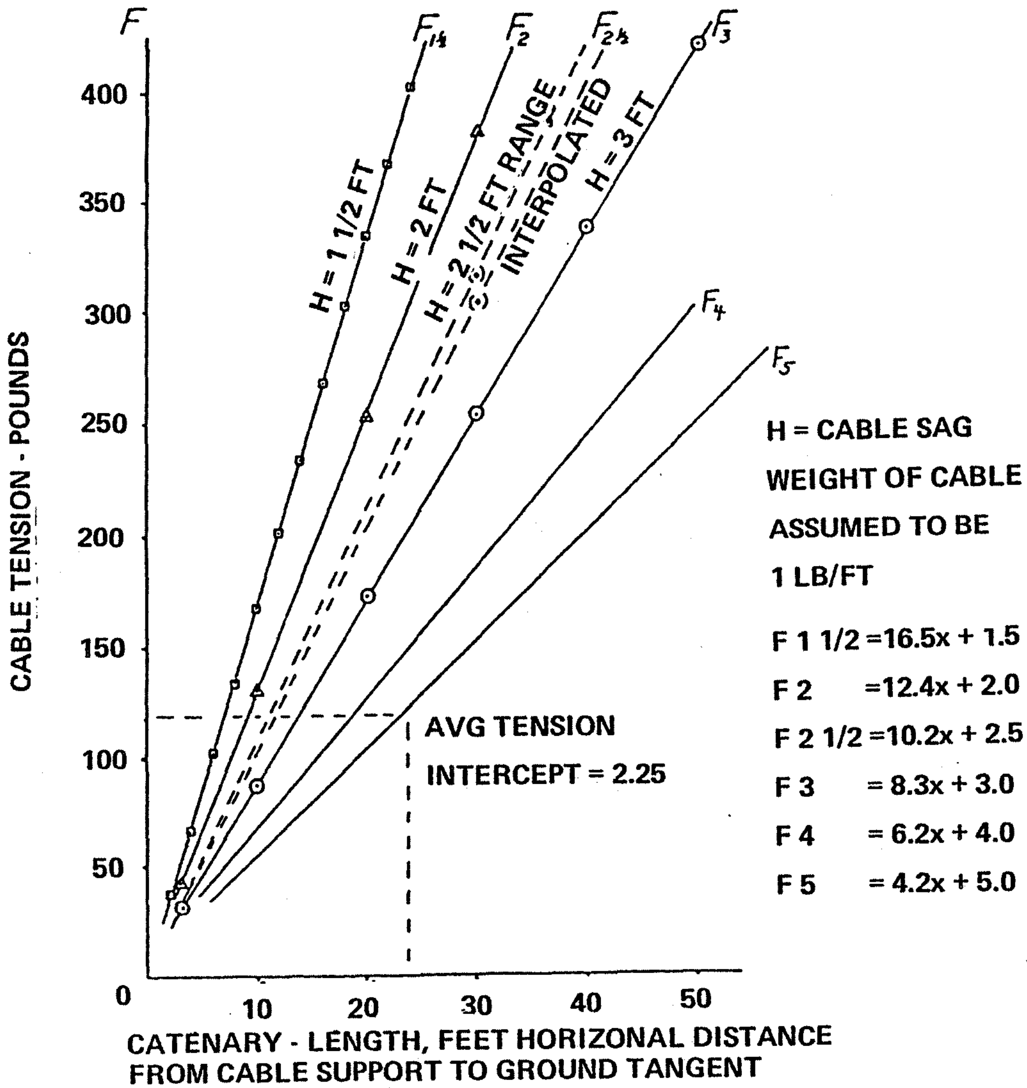


FIGURE 9 CATENARY CABLE TENSION

modification. A high-seam machine was chosen because higher-seam machines in general travel faster, have more cable reel inertia to contend with (because of cable size and physical constraints imposed by the design), and normally are less critical for height, thus allowing room for retrofit in the field.

The system should also be simple enough so that it could be developed and tested in a couple of years as opposed to a long-term development program.

2.5 Mine Survivability

Componentry selected should be compatible with the underground mine environment. Mechanical components should be rugged or well protected; hydraulic components should operate at low pressures and be relatively dirt-tolerant; electrical circuitry should be kept to a minimum and be enclosed in permissible enclosures.

Efforts should be made to allow access to all components in the system with minimal effort.

The system should be designed to ingest a reasonable quantity of dirt with minimal effects on performance.

2.6 Maintenance Should be Obvious and Simple Enough to Insure Consistent System Performance

Wear points should be kept to a minimum to eliminate wear in such places as bushed joints and hence limit the need to adjust because of wear in these points. System adjustments should be made obvious and, if possible, eliminated.

In general, "brute force" (simple, unsophisticated) type systems should be used in place of sophisticated, less obvious types of systems.

Electrical systems should either be of the "brute force" (simple, unsophisticated) type or be made of easily-replaceable circuit modules for troubleshooting.

Hydraulic systems should not contain small orifices or componentry drastically affected by dirt unless adequate filtration is guaranteed.

2.7 Various Concepts Considered

It became apparent after looking at the results of the conventional cable reel drive tests that the reasons for the tension transients in the cable were caused mostly by the acceleration/deceleration inertia effects of the reel/vehicle system. Originally variances in the cable reel radius were suspected to be the cause of this problem. Furthermore, in an effort to limit these inertia effects, the inertia had to be drastically reduced or a system would have to be incorporated that could power the reel in either direction.

Techniques of reducing and/or eliminating reel inertia were reviewed but were found to be unsatisfactory. Some of those reviewed were:

1. Stuffing box, patented by Jamison, et al (Patent No. 3,990,551). This technique effectively eliminated reel inertia but is susceptible to problems caused by cable entanglement and dirt buildup.

2. A similar arrangement to the stuffing box, but with an integral cone in the box to cause the cable to lie in an orderly manner. This approach was unsatisfactory because of cable twisting, dirt buildup and physical-size constraints.

3. A spinning roll and type of cable storage device. This would enable the cable to remain at rest. Cable twisting and sharp cable bending radii rendered this method unsatisfactory.

The alternate method of providing a bi-directional mechanical drive for the reel requires a feedback loop to sense direction of cable movement, to provide a signal for the drive to operate. Also, because of the time lag required by a feedback loop and the size of componentry required to accelerate the reel, some type of a cable storage device is necessary when reel speed and relative cable velocity are not precisely matched. It is also possible to use this storage device as the input for a feedback signal and to maintain a certain range of tensions on the cable.

A concentric-type stuffing box with a low-power electronic logic was considered (Figure 10) but was discarded because of the physical size and complex electronics required to make the system functional.

A spring-loaded cable storage device that would store cable around the O.D. of the cable reel was also considered but dropped because of physical constraints, as shown in Figure 11.

2.8 Final Design Concept

The final design concept concluded to be the most practical, within the confines of a typical shuttle car, is that shown in Figure 12. This uses the existing cable reel and level wind assembly for cable storage, but the logic driving the motor has been changed to make it drive in either direction.

A slack chamber is added to the existing vehicle and protrudes about 10 inches above the vehicle frame on the side opposite the operator. This should not be a serious problem on a larger vehicle, since 12 inch sideboards are common. The spring-loaded take-up device serves to store or give up cable over short-term transients, maintain tension

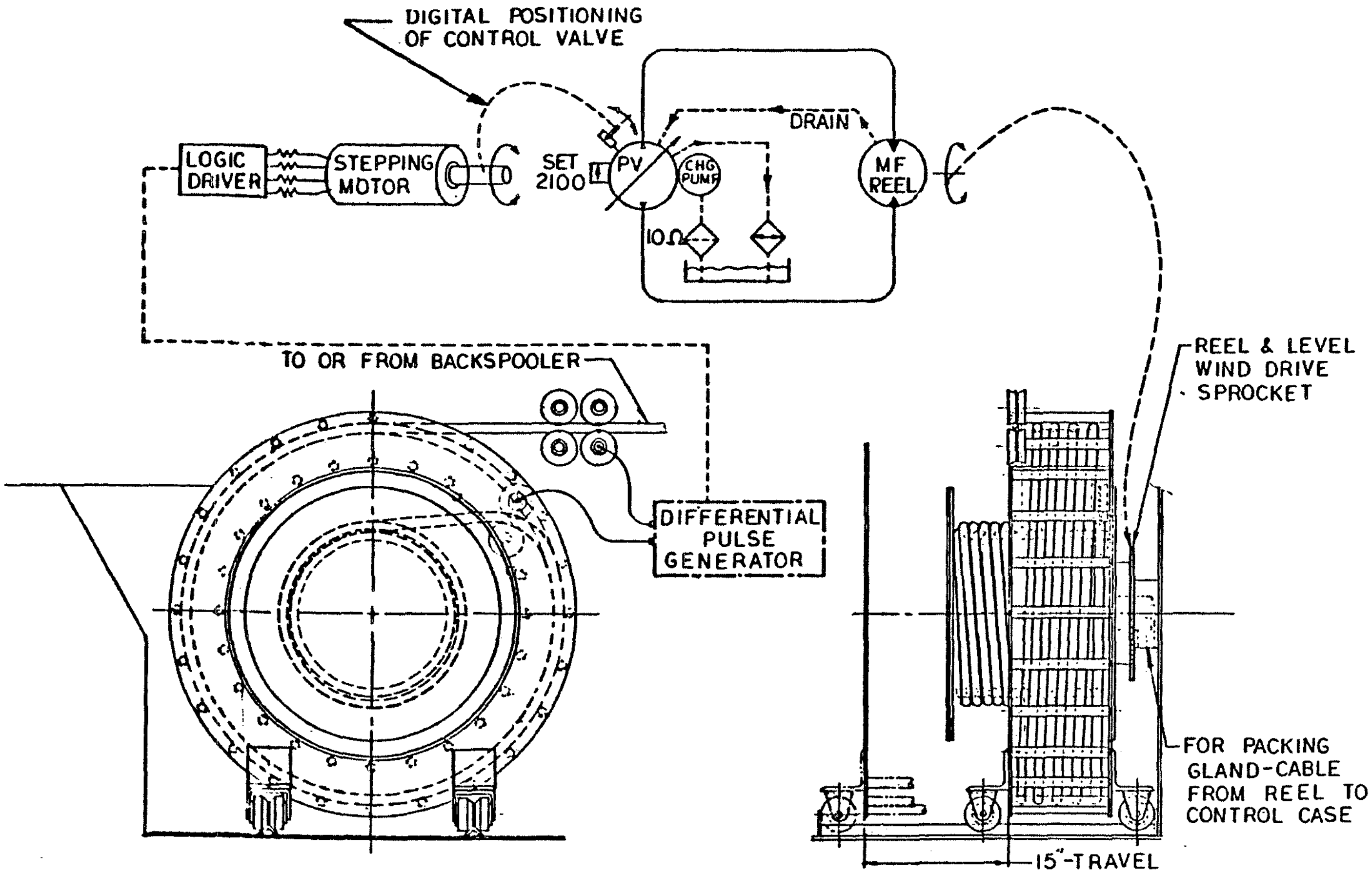


FIGURE 10 CONCENTRIC CABLE CAGE CONCEPT

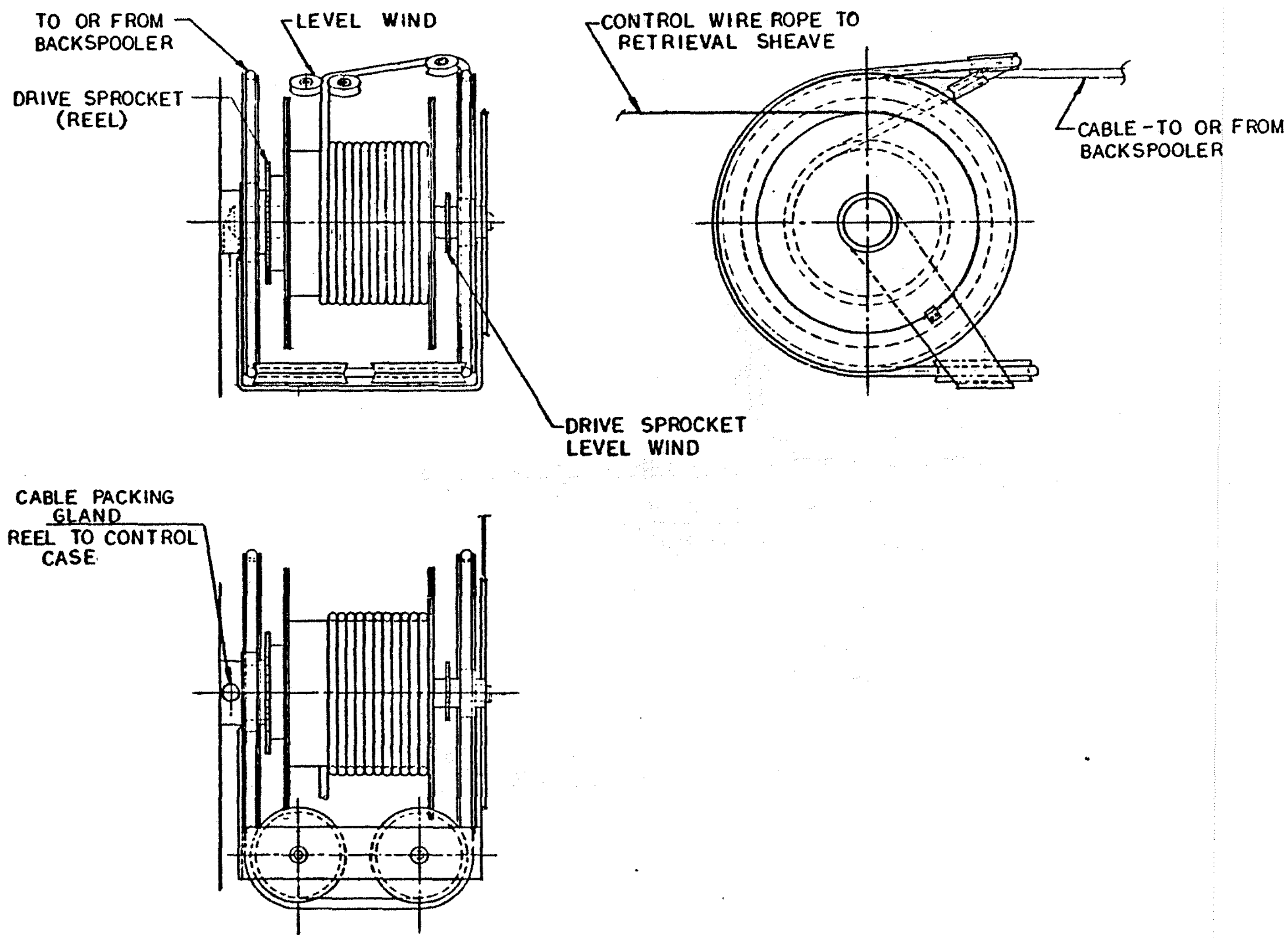


FIGURE 11 WRAP AROUND TAKE-UP CONCEPT

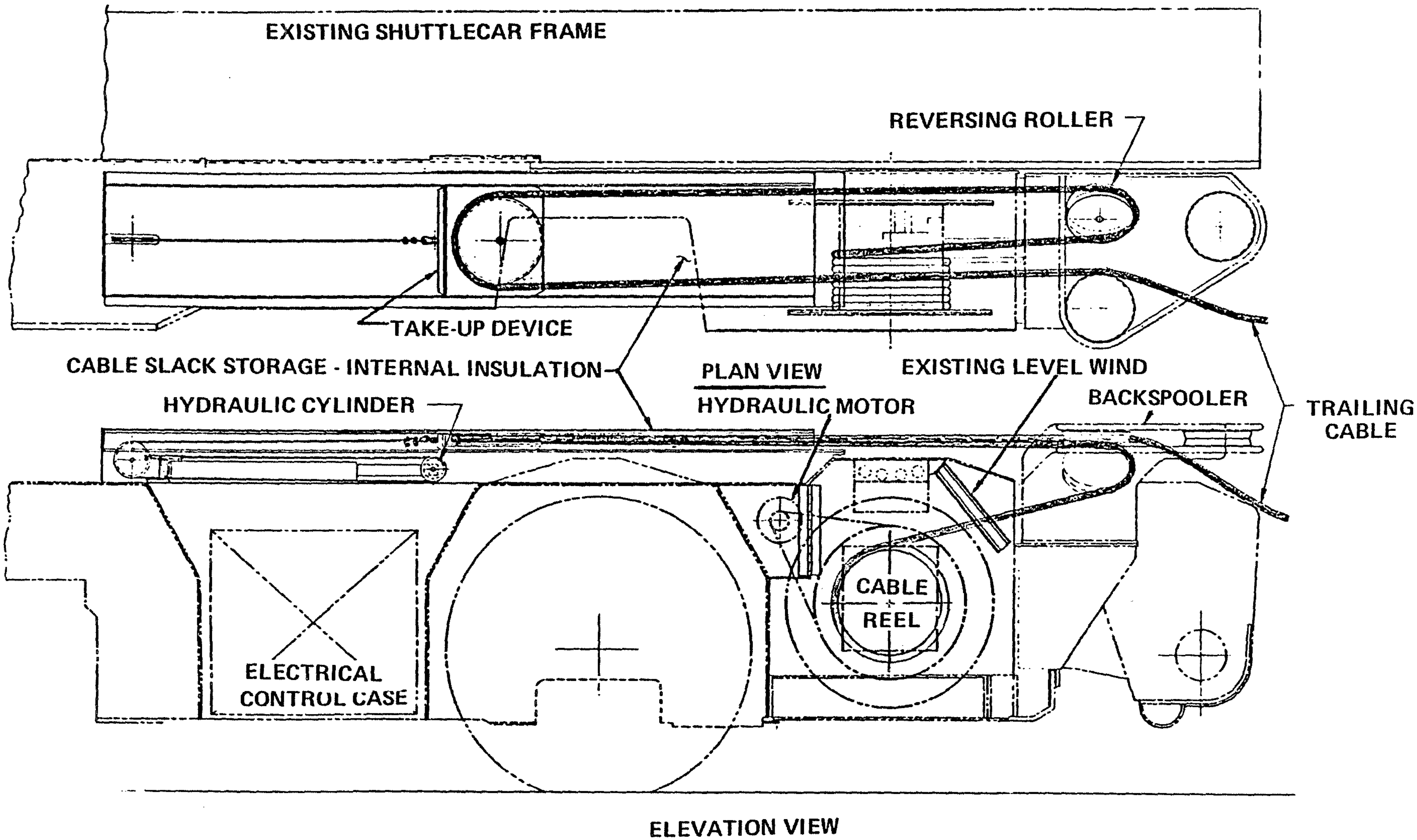


FIGURE 12 GENERAL ARRANGEMENT OF FINAL DESIGN

within specified limits, and provide a sensing device for the primary feedback signal.

This device requires that the cable travels through two additional 180° bends; to maximize cable life, large diameter sheaves are used. If the actual field cable life approaches the theoretical reduced life due to the sheaves, great overall life improvements will have been made.

2.9 Tentative Performance Specifications - Improved Shuttle Car Cable Reel System

General

The cable reel system shall be capable of:

1. Satisfactory performance for 1,000 hours of normal operation in underground bituminous coal mines on a basis of two operating shifts and one maintenance shift per day, when routine maintenance instructions are followed.

2. Being repaired or replaced at the underground site of operations by a crew of two men with hand tools and a powered scoop, within the time span of one maintenance shift.

Provisions shall be made to guard against the addition of hydraulic fluid from a general supply by:

1. Use of special containers for hydraulic fluid.

2. Other means which assure that the hydraulic fluid cleanliness level remains consistent with proper operation.

Operating Capabilities

The reel system shall be required to maintain cable tension at any set value between 50 and 300 pounds pull, and in such a manner that in normal operation of the shuttle car in an underground coal mine entry no part of the deployed cable will acquire a velocity greater than one foot per second, nor experience an acceleration greater than two feet per $second^2$. Pay-out and rewind cable tension shall be adjustable. Normal shuttle car operation shall include but not be limited to the following:

1. Maximum acceleration from standstill to full speed of the shuttle car or three feet per second, whichever is greater.

2. Crash stops (defined as maximum practical deceleration to full stop) from any speed up to the maximum of six miles per hour.

3. Rapid reversal of direction of motion of the shuttle car from 10 percent of maximum speed in one direction to 10 percent of maximum speed in the opposite direction.

4. Rapid reversal as in 3, above, but with a two-minute period at standstill between reversals.

5. Drifting to a standstill from any speed up to maximum with traction drive power off.

6. Drifting as in 5, above, but with no power to the car (circuit breaker tripped at power center), assuming that parking brake stops vehicle within specified distance (one ft./mph).

7. Drifting downgrade from a standstill to any speed up to maximum with traction drive power off, assuming that parking brake stops vehicle within specified distance (one ft./mph).

8. Passing the cable tie point at a distance of six feet or more at any speed up to maximum in either direction.

9. Roll, pitch, and yaw motions at any speed which may be induced by irregularities in the mine floor or by the operator's steering efforts.

10. Maintaining up to six mph vehicle centerline ground speed with the reel located on either the inside or outside of the vehicle turn radius.

The reel system shall provide storage for sufficient cable to permit a working radius of not less than 600 feet from the cable tie point.

The reel system shall be required to produce a level wind of cables containing splices which increase the major diameter of the cable by no more than 12 percent.

The reel system shall clean the cable during the rewind to the extent necessary to assure satisfactory operation within the assigned probability limit.

The reel system shall provide means of cooling the cable or assuring by other means that at no point in the reeled cable or in the system, when operating at maximum-rated conditions, will component temperatures or hot-spot temperatures exceed 105 degrees centigrade (221^{o}F), or the temperature rise above ambient exceed 70 degrees centigrade (158^{o}F), whichever is smaller.

SECTION III

DETAIL DESIGN

3.1 General Design Layout

The cable drive mechanism is shown in Figure 12. The first cable interface with the vehicle is the backspooler which serves the same function as on any cable reel system. The two horizontally-mounted rollers are located at the extremes of the vehicle envelope. This allows 270 degree range of horizontal cable feed. On higher shuttle cars, the backspooler is usually mounted on the conveyor boom and feeds directly to the level wind/cable reel assembly. In this configuration, the backspooler is mounted on the vehicle frame, high enough to clear the tire without restricting overall working machine height.

The slack take-up is a spring-loaded pulley. The "pulley" can travel four feet either side of its static position, allowing eight feet of cable release or take-up. The hydraulic spring is the primary factor in determining cable tension. Other factors include:

a. The inertia to accelerate the take-up follower.

b. Follower moving friction.

c. Roller friction.

(This assumes the take-up never reaches its travel limits, in which case the cable tension would be drastically affected.)

Since the friction forces are small and the weight of the take-up follower is light, little tension variations will be seen.

Besides maintaining a controlled cable tension and temporary surge storage, the follower also provides the primary signal to the reel drive system. By sensing take-up location relative to the central "static" position, the system can determine reel drive direction. The further the take-up is off center, the faster the reel turns, trying to get back to the center position.

From the take-up, the cable is fed through a reversing roller. The additional 180 degree cable bend is theoretically not necessary, as the cable reel and level wind could be reversed. However, from a practical standpoint, retrofit requires the reversing roller. If the reversing roller were eliminated:

1. slack storage would be less or the storage chamber would interfere with the miner boom;

2. some means would be required to center the cable on the reel (in the plan view); and

3. a device is still required to sense cable velocity at the reel.

Sensing velocity was found necessary as a secondary feedback signal to eliminate system instability.

The conventional reel drive has been replaced by a fixed displacement/bi-rotational motor and a variable displacement pump. The motor drives the reel in either direction. Figure 13 is a schematic of the entire system.

3.2 Slack Take-up Device

Because of vehicle size, cable tension required the use of an hydraulic spring rather than a mechanical one. In order to keep the cable tension within acceptable limits, a mechanical spring would be longer than the vehicle.

The hydraulic spring has a 4:1 mechanical advantage between the cylinder and take-up pulley. This is achieved by a cable/pulley arrangement. A cam governing the action of the reel drive hydraulic pump is attached to the cylinder rod. Figure 14 shows the cylinder and cam mechanism which is mounted directly under the take-up pulley. Under the cylinder and cam is the 320 PSIG precharged gallon accumulator. The hydraulic spring rate at the take-up pulley is 2.1 pounds/inches. (The ideal figure from our later computer simulations is 2.5 pounds/inches.) There is also some non-linearity associated with this spring rate. When maximum storage of cable is achieved, the spring rate is 1.75 pounds/inches. As the cable is released, spring rate increases to a maximum of 2.5 pounds/inches. Figure 15 shows cam stroke, cable tension (used in the computer model) and the actual cable tension dictated by the hydraulic spring, as a function of the cylinder stroke. The law requires take-up chamber insulation. An ultra-high molecular weight (UHMW) polymer was selected because of insulating properties and lubricity.

The inherent low natural frequency of the take-up system and its tendency to cause instability make it necessary to minimize the take-up roller weight. It should be noted, this system will not remain stable if the take-up pulley mass exceeds 25 pounds.

3.3 Reel Drive Mechanism

The cable reel drive consists of a closed loop hydraulic circuit with a variable displacement pump and a fixed displacement motor. See Figure 16, 17 and 18. The units selected are relatively dirt-tolerant. The pump has a direct manually-activated swash plate, which eliminates lag and stroke time associated with other pumps having servo controls.

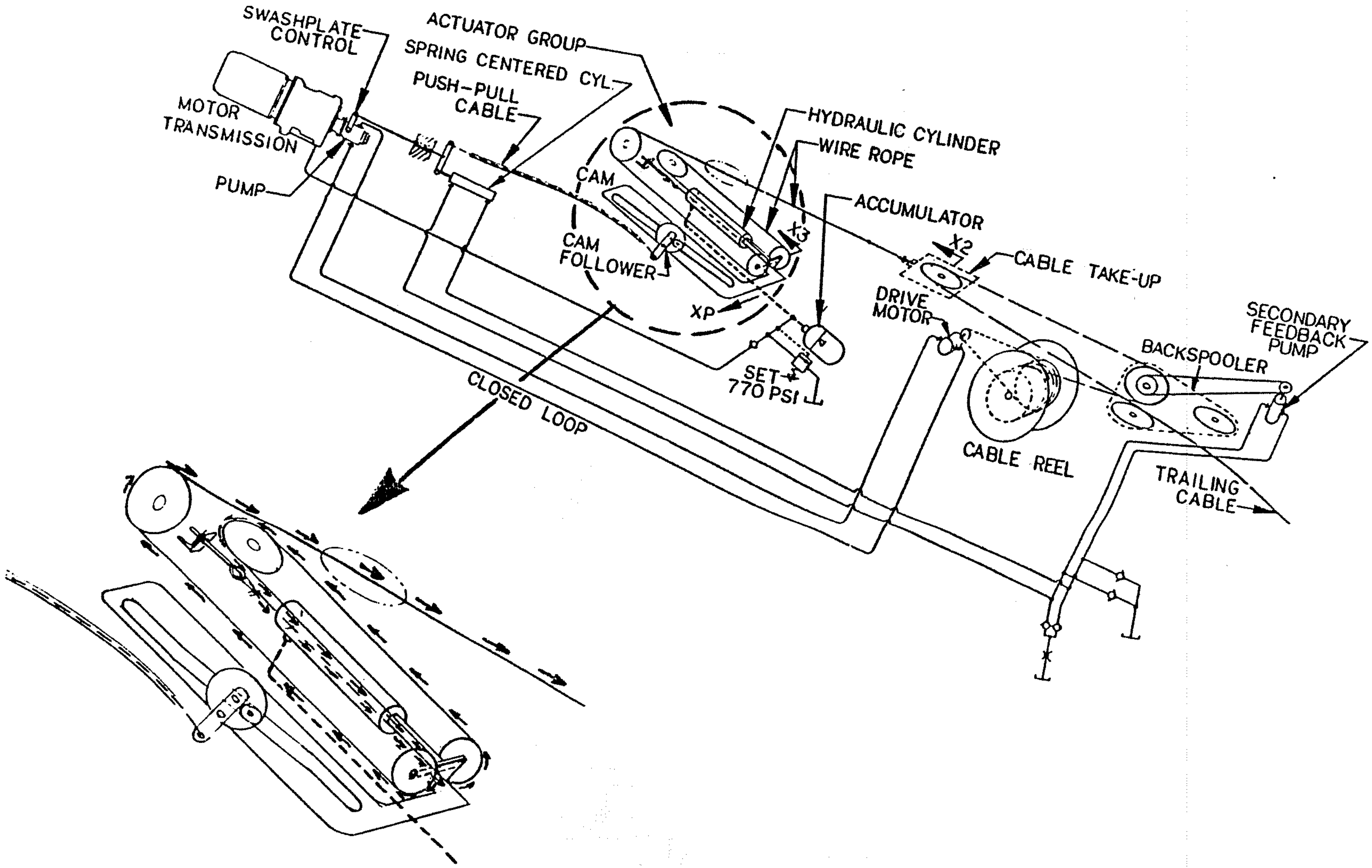


FIGURE 13 SCHEMATIC OF FINAL DESIGN CONCEPT

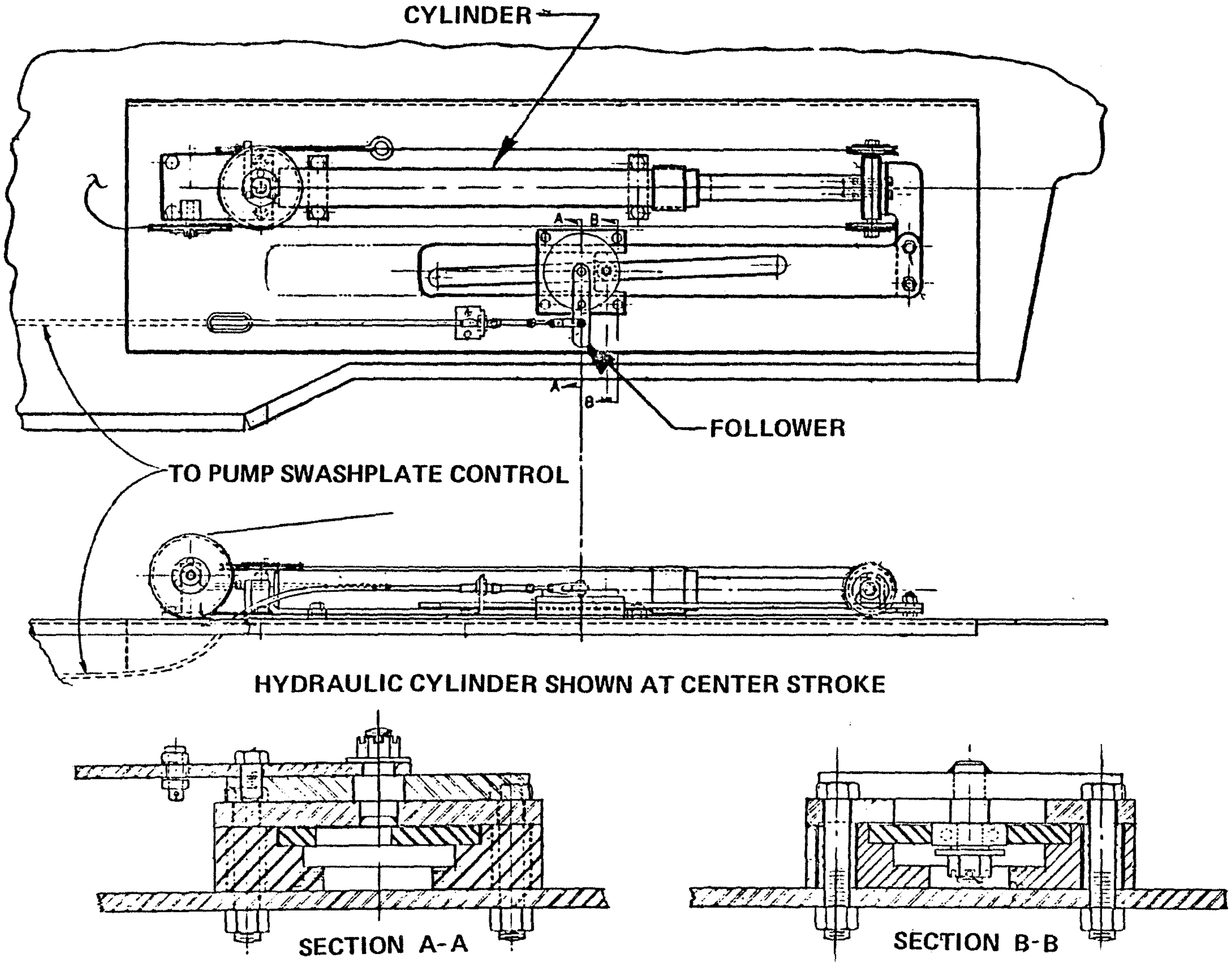
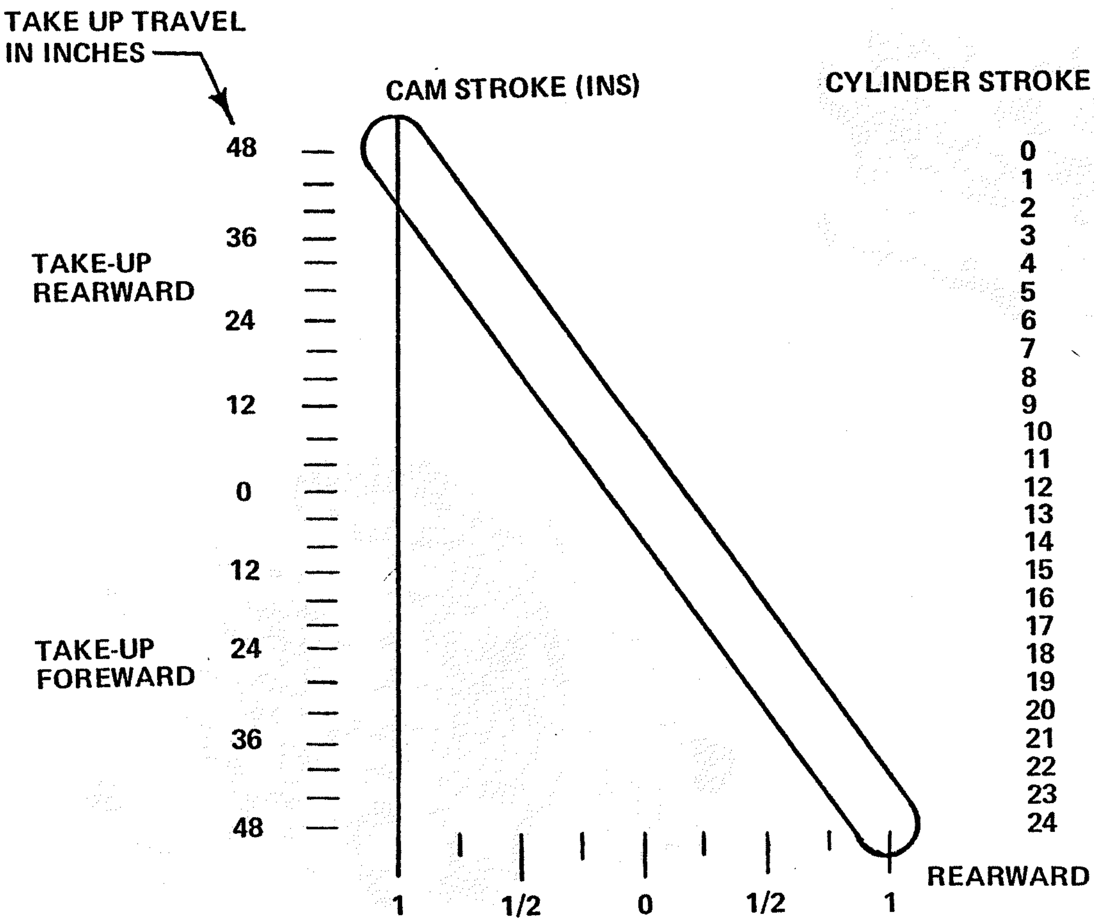


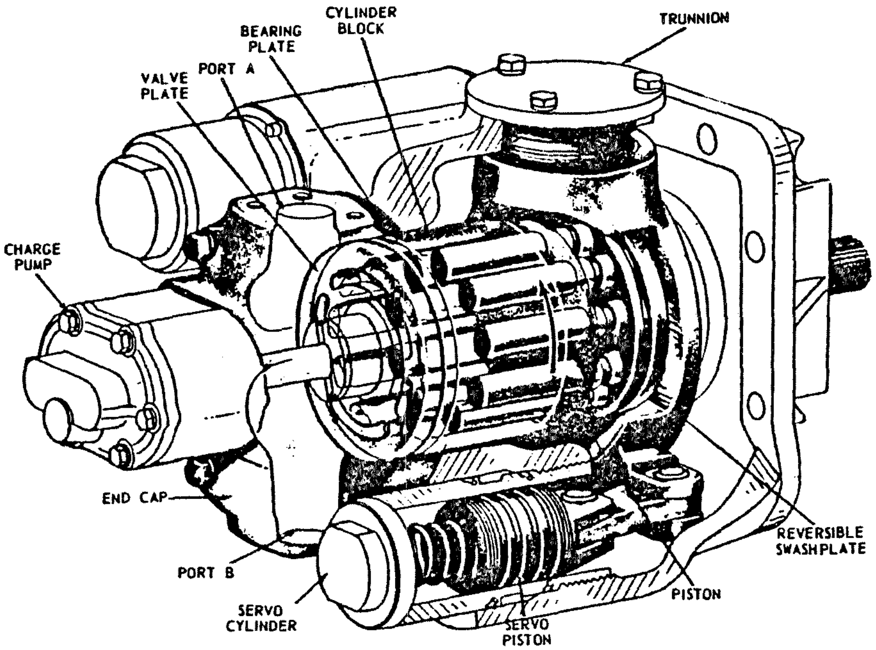
FIGURE 14 TAKE-UP SPRING AND CAM ARRANGEMENT

PULL ON CABLE AND CAM POSITION

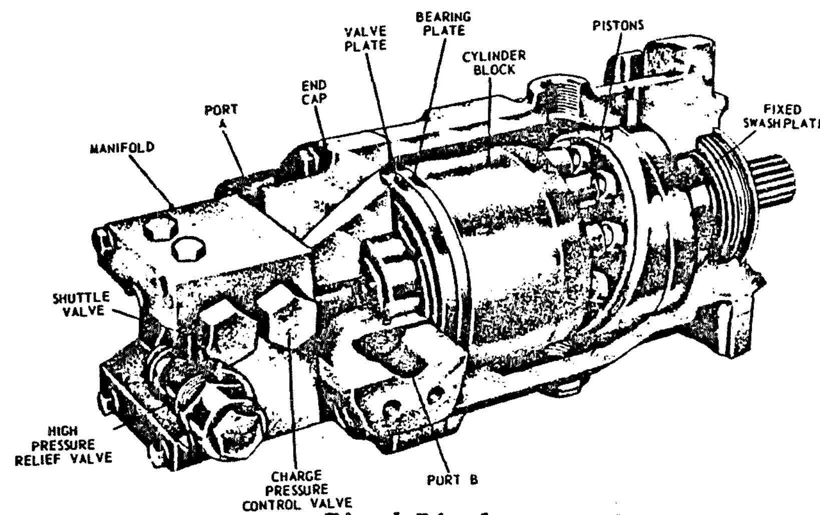


CYLINDER STROKE (INS)		PULL ON CABLE MODEL ACTUAL
0	--	235 - 226
1	--	230 - 219
2	--	225 - 213
3	--	220 - 207
4	--	215 - 201
5	--	210 - 196
6	--	205 - 191
7	--	200 - 186
8	--	195 - 181
9	--	190 - 177
10	--	185 - 172
11	--	180 - 168
12	--	175 - 164
13	--	170 - 160
14	--	165 - 156
15	--	160 - 153
16	--	155 - 149
17	--	150 - 146
18	--	145 - 142
19	--	140 - 139
20	--	135 - 137
21	--	130 - 134
22	--	125 - 131
23	--	120 - 128
24	--	115 - 125

FIGURE 15 TAKE-UP LOCATION/CABLE TENSION/CAM RELATIONSHIPS



Variable Displacement Pump



Fixed Displacement Motor

FIGURE 16 AXIAL PISTON PUMP AND MOTOR

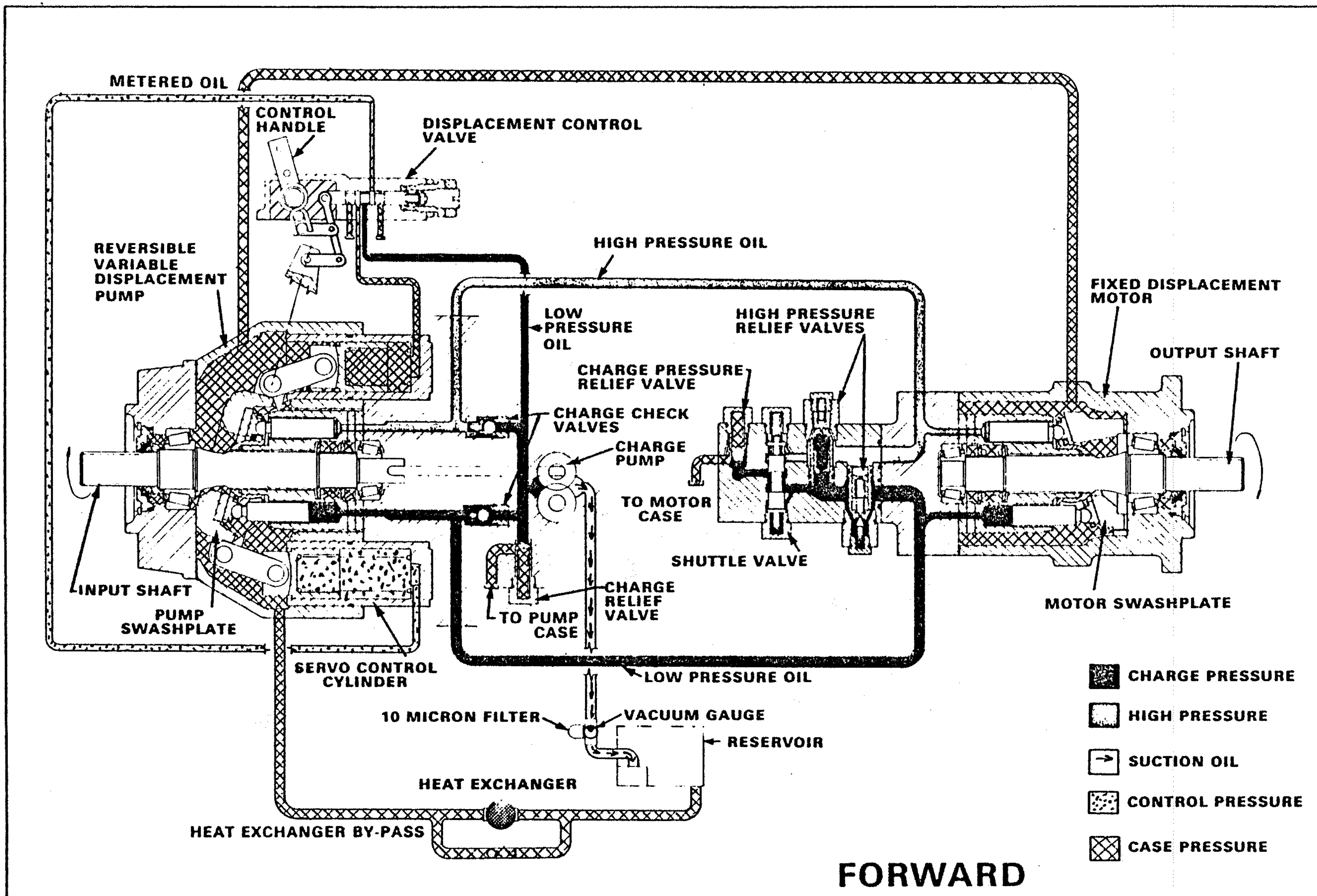


FIGURE 17 TYPICAL CLOSED LOOP HYDRAULIC CIRCUIT

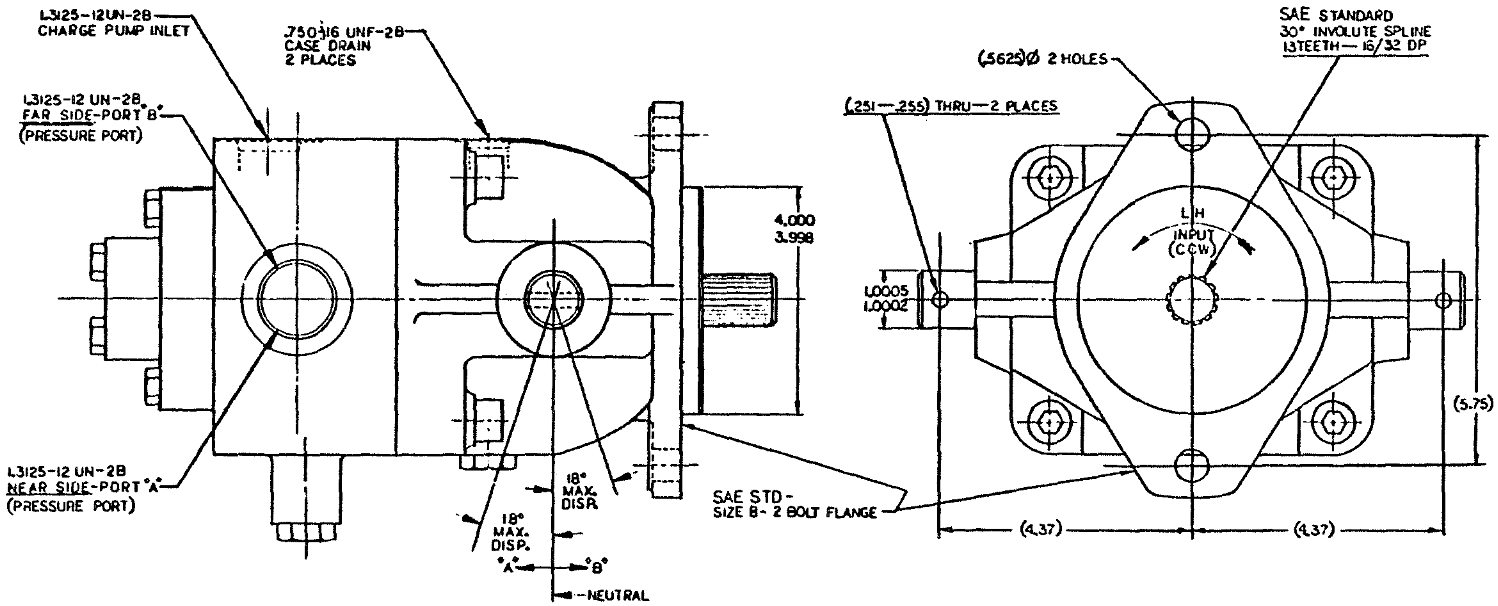


FIGURE 18 MANUALLY OPERATED PISTON PUMP

The motor has a 4.26 cubic $inch^3$/rev. displacement and is coupled to the cable reel through a 5.85 ratio chain and sprocket. The motor was selected because the motor shaft is well supported, therefore an overhung bearing load adapter is not required. The key to making this system work is to minimize the amount of oil in the closed loop and to use steel tubing rather than hose. Both of these factors have a large impact on the accumulator affect of the oil. In our analysis, with the pump located at one end of the vehicle and the motor at the other, lines must be 1/2 inch diameter steel tubing or system instability would result. The steel tubing generates some line losses during high-speed operation, however not enough to cause problems.

In addition, the hydraulic-closed loop must have a 10 micron filter between the tank and the charge pump. Jeffrey's 4015 Shuttle Car's hydraulic tank should be large enough to dissipate the additional heat caused by this system; on the other machines, an oil cooler may be necessary. Maximum operating pressure should be 2500 PSI.

3.4 Cam and Primary Feedback System

The primary feedback signal is generated by the take-up pulley location. By sensing which direction this moves off-center, it is possible to determine whether the reel should reel in cable or pay it out. This signal is generated by a cam and follower. The cam is bolted to the piston end of the cylinder and the follower is fixed to the vehicle frame (see Figure 14). The cam design was selected because it can easily be changed or modified to vary the output/input signals relative to each other. Also the system gain and the profile of that gain can be varied during testing.

The final cam design used in our computer simulations was a constant slope cam. (See Figure 15.) The cam must be capable of producing 170 pounds of force on the cable. This corresponds to 71 pounds required at the swash plate control arm to vary the swash plate angle at relief pressure.

The cam and push/pull cable was chosen over electrical or hydraulic systems for the following reasons. An electrical system would require several electrical enclosures. The design requires proportional pump output electrical componentry which would become large and awkward. Hydraulics were not selected,as the mechanical approach seems simpler and more reliable. Also a hydraulic (master cylinder) system would require additional maintenance. If problems arise with the mechanical system, a master/slave hydraulic system could be used.

3.5 Speed-Sensing Roller and Secondary Feedback System

To stabilize the cable reel drive system while maintaining adequate driving horsepower (gain), it is necessary to have a secondary feedback loop. This second loop feeds in a negative signal (to the primary signal) proportional to the speed of the cable feeding onto or off of the reel. These two signals provide a fast pump reaction while

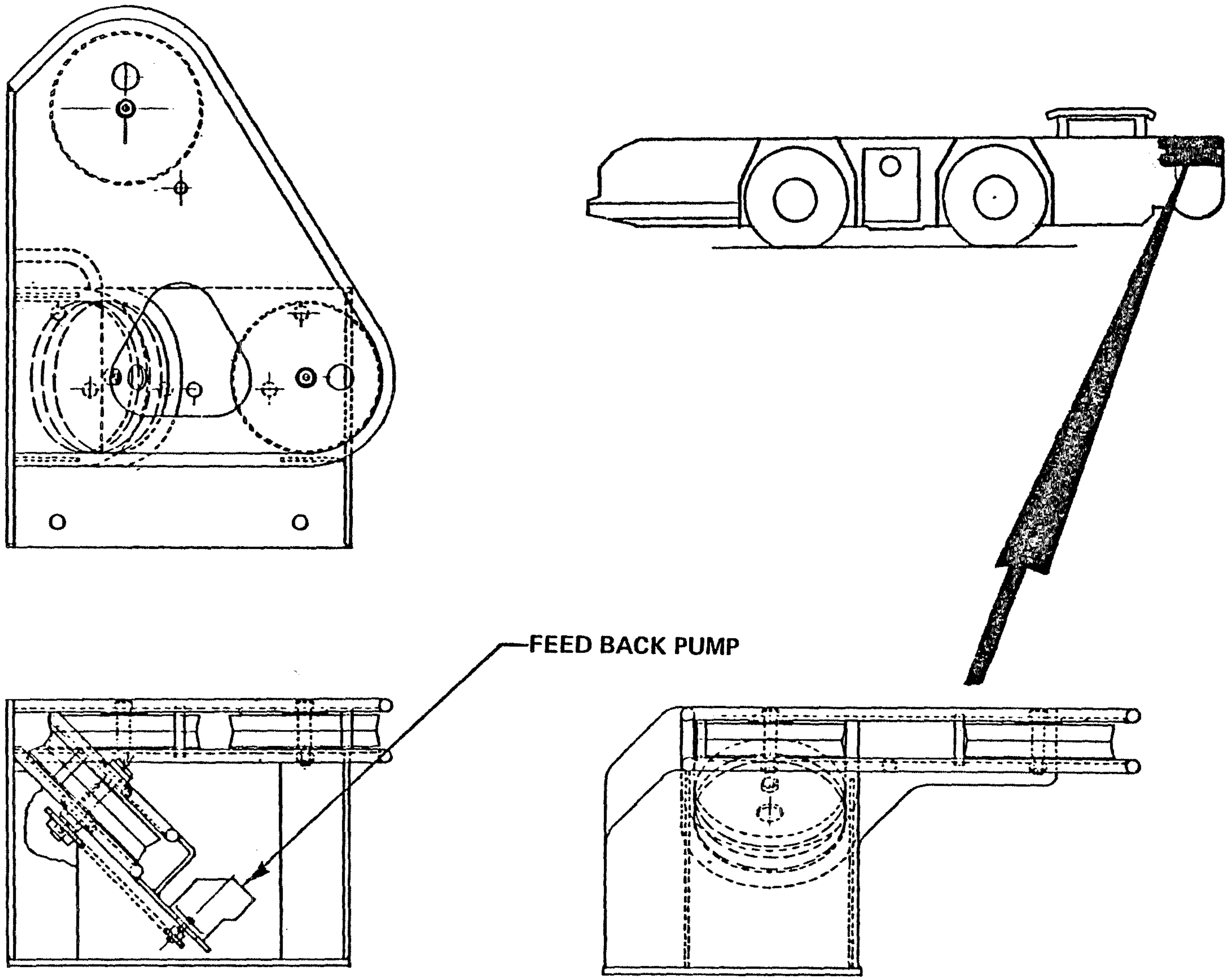


FIGURE 19 BACKSPOOLER AND SECONDARY FEEDBACK PUMP

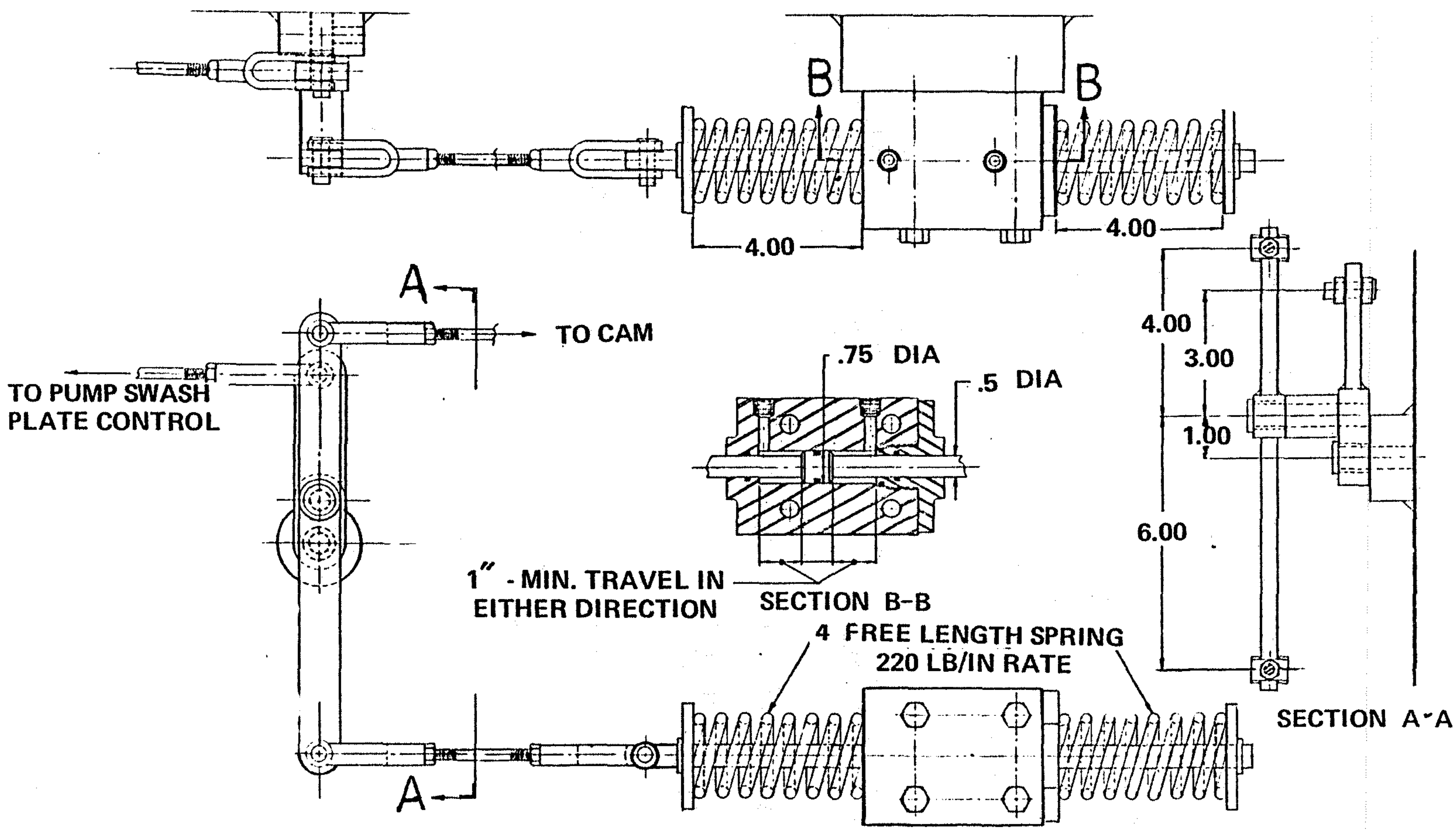


FIGURE 20 SECONDARY FEEDBACK CYLINDER AND RATIOING LEVERS

maintaining system stability. When the follower goes off center and as the cable speed builds up, the take-up's contribution to the feedback signal is reduced, thereby stabilizing the system.

The actual hardware used to provide this secondary feedback signal is shown in Figure 19. The reversing roller drives a small, low-leakage pump with a 2.5 speed-up ratio. The pump is piped into a spring-centered cylinder. (See Figure 13.) The orifice (in the pressure-to-tank line) should be variable and sized by test. The cylinder has a two-inch stroke, 0.75-inch bore, 0.5 inch-rod and equal effective area on each end. (See Figure 20.) Two centering springs are incorporated into the cylinder, so that 900 PSI moves the cylinder rod one inch either side of center. The pressure-to-tank line orifice should be adjusted to develop a pressure of 900 PSI, with 63 cubic inches/minimum flow (six mph cable velocity). Response time will be 0.47 seconds (two-inch travel), and maximum output force will be 220 pounds. Since 113 pounds is required at relief pressure, this gives a 2:1 safety factor. Torque required at the reversing roller to drive the secondary feedback pump is 107 inches/pounds, or approximately 18 pounds' tension on the cable.

The two feedback signals are fed back to the feedback linkage, as shown in Figure 20. These levers are arranged to give the proper mechanical output corresponding to the computer analysis. Mechanical output can be changed by varying the lengths of these levers during tests.

3.6 Failures and Protective Devices

Two separate areas were looked at for cable protection. The first relates to an "end-of-cable" warning and shut-off device and the second relates to cable jams or system malfunctions.

The end-of-cable devices are located in the cable reel. The reel shown (Figure 21) is the Jeffrey 2015 Shuttle Car, and these warning devices will require modification to alternate reels. The spring-actuated plunger shown in Figure 21 is held in the retracted position by the eighth lap of cable. As the reel unwinds beyond this point, the plunger is extended by the spring, and it strikes a hammer which, in turn, strikes a gong each revolution of the reel. If the operator trams an additional 31 feet, the reel will turn six times, striking the gong six times, and finally uncovering the end of cable and shut down the power. The limit switch interrupts the ground check circuit, cutting power to the machine. As the machine trams the opposite direction (reeling cable back in), the extended plunger on the warning system still strikes the hammer, but the hammer is designed to fold in this direction and allow the plunger to pass. Cable tension is enough to reset the warning device on the eighth lap of cable.

Anti-jam sensors (shown in Figure 22) are limit switches which sense that the cable take-up pulley has reached the end of its travel.

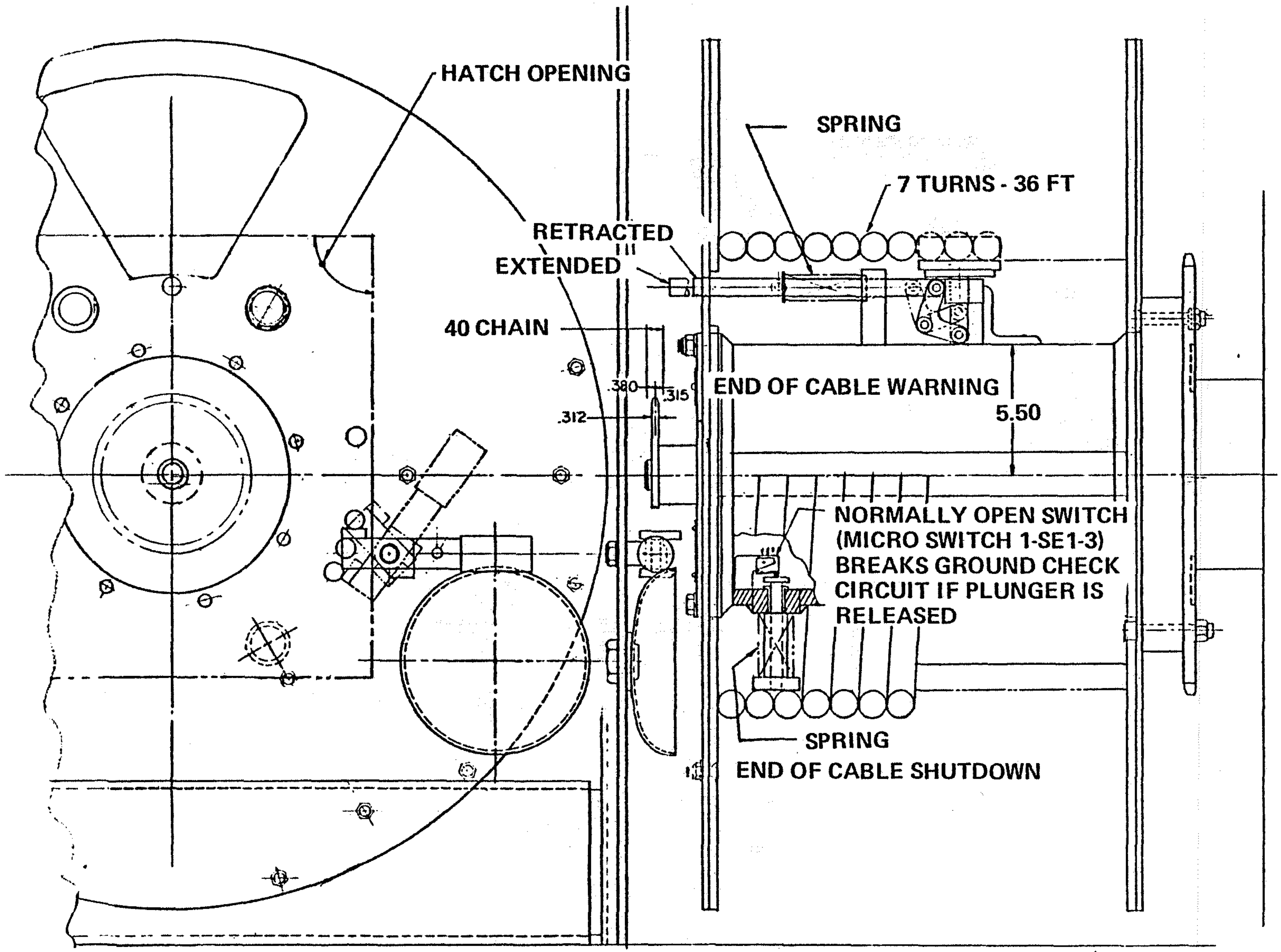


FIGURE 21 END OF CABLE WARNING AND SHUT DOWN

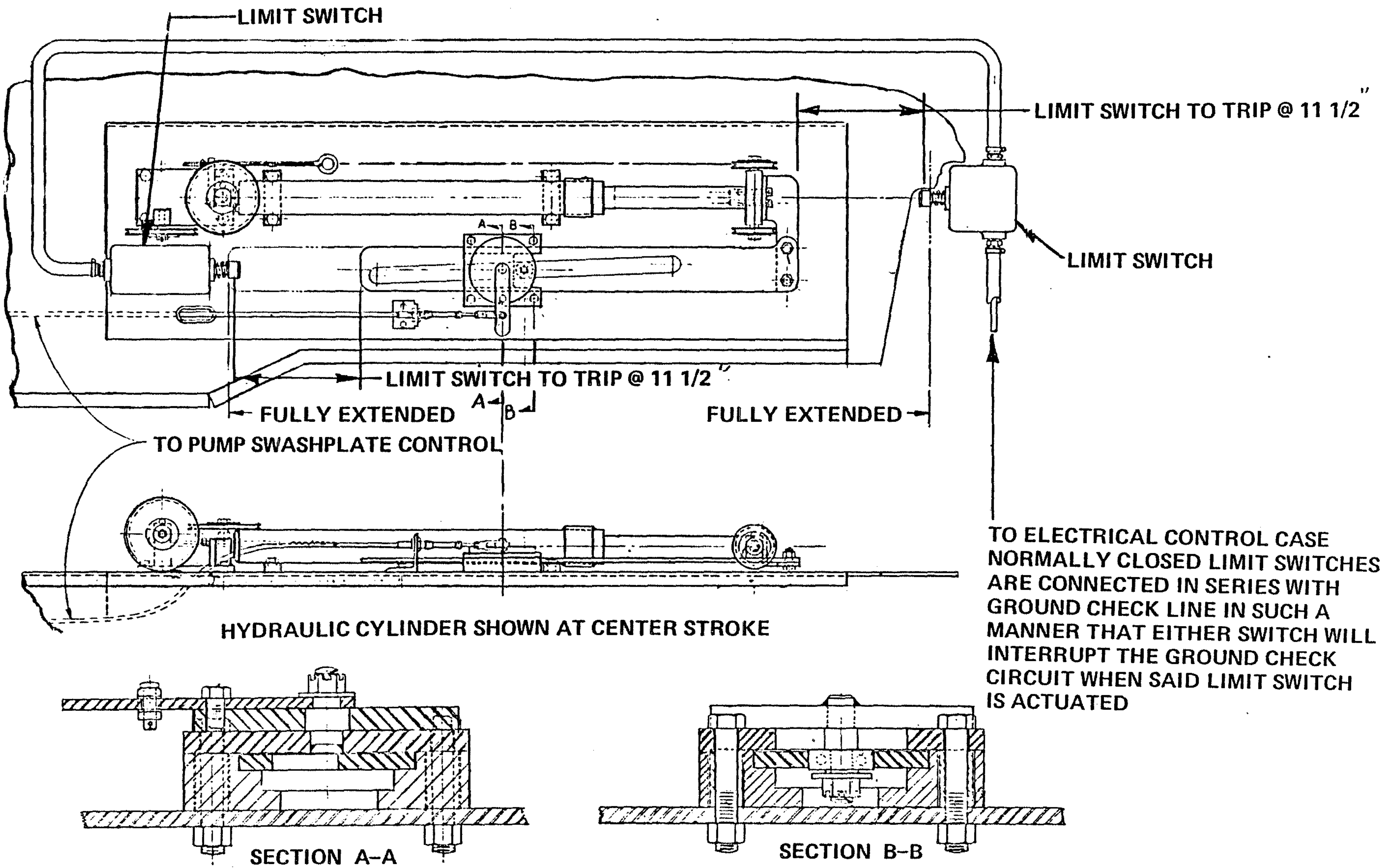


FIGURE 22 CABLE JAM SENSING SWITCHES

Tension spikes have the capability of building in the cable at this point. The sensors also interrupt the ground check circuit, cutting power to the machine.

Possible modes of system failure and the results are as follows:

1. The reel or reel-drive mechanism jams.

   a. If the car is reeling in cable, the take-up takes up slack until contacting the shut-down switch.

   b. If the car is reeling out cable, the take-up gives up cable until contacting the shut-down switch on the opposite end.

2. The cable becomes tangled on the reel and starts wrapping the wrong direction.

   a. Since rotation in either direction on the reel results in reeling in cable, slack will be taken out of the take-up until the shut-down switch is contacted.

3. Jam between the cable and the take-up pulley.

   a. The take-up will move directly with the motion of the cable until a shut-down switch is contacted reeling out.

   b. The take-up will center itself and stabilize the system, and slack will develop in the cab outboard of the back-spooler reeling in.

4. Jam between take-up and take-up guide.

   a. The take-up will not be able to sense movement of the cable, and the cable will either go slack if the car is reeling in cable or pull tight against the relief pressure of the reel-drive system when paying out cable. For this reason, the drive system relief should be set no higher than necessary to drive the system (approximately 2,500 PSI).

5. A jam at the backspooler.

   a. When reeling in or out, the drive system will seek its mid-position. If reeling in cable, the cable outboard of the backspooler will go slack and the operator should be aware of it.

   b. If reeling out cable, the cable will be stretched until it unjams or breaks,as the sensing system will not detect this type of jam. This is, however, the most unlikely jam to occur.

SECTION IV

ANALYTICAL TEST RESULTS

4.1 Computer Model

The analytical dynamic model used was designed for CSSL-IV (continuous system simulation language - version four) software. This is the dynamic system modeling program used by Control Data Corporation, the computer network used. The final model diagram (Figure 24) was developed with the support of Dr. Donald Houser (a professor in dynamic systems) of the Ohio State University Engineering Department. Model parameters used are identified on pages 44 through 47.

The model is straightforward, except the pump/motor closed hydraulic loop. Equations for this system are from E.O. Doebelin's book, Hydraulic Conduits and Machinery, pages 395 to 411 (Variable Displacement Pump Systems).

4.2 Single-Loop Feedback System

The original model did not include the secondary feedback loop (represented by the line beginning with WM (top right) and ending with $\frac{d}{\chi}$ (top center in Figure 24).

All variations tried will not be discussed, as over 200 different cases were used; and in all cases, when sufficient power was provided (to meet system requirements), instability resulted. As parameters were varied outside the limits of practicality, stability could be achieved with the reel empty or full but never in both situations, simultaneously.

Figures 25-28 show a typical example of output from a single-loop feedback system using a six mph step input. These figures are of the same case, but plot different parameters versus time. Two separate low-natural frequency systems are the primary cause of output instability.

The first system is the take-up. The system spring rate is determined by the allowable cable force variations and the amount of cable storage required. The mass of this system (all moving parts) is 25 pounds, the lowest possible for a viable system. Because of the low spring rate and relatively high mass, this system would vibrate at a low speed if allowed to cycle independently.

The second system is the cable reel and drive. In this system, there is a stiffer spring (caused by the bulk modulus of the oil and piping in the hydraulic-closed loop) and a higher (and in this case variable) cable reel mass. The natural frequency is low (but varying due to

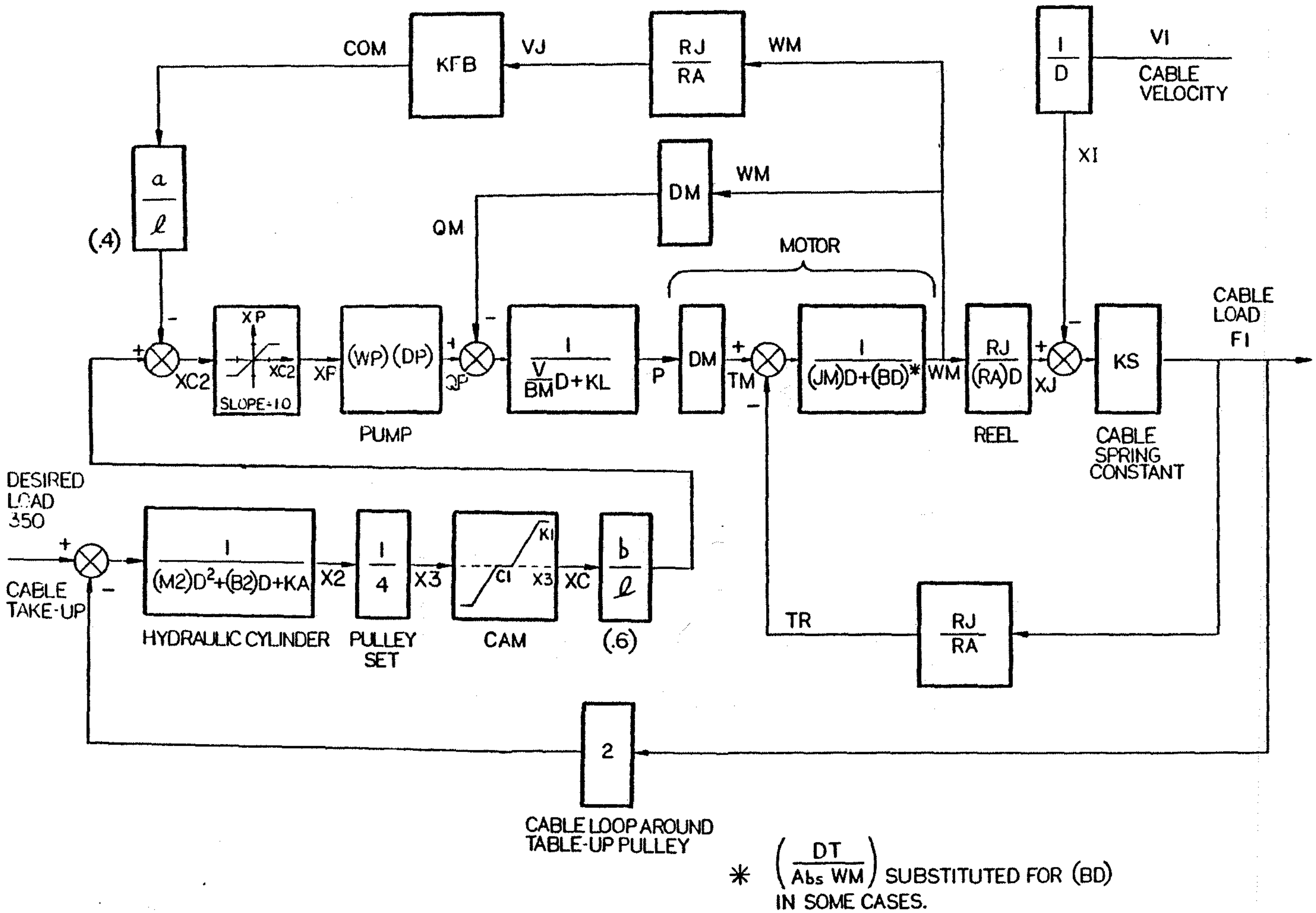


FIGURE 24 COMPUTER MODEL (BLOCK DIAGRAM)

SYMBOLS FOR CABLE REEL ANALYTICAL MODEL

M2	MASS OF TAKE-UP ($\frac{25}{386}$ = 0.06477) (LB-SEC^2/IN)
D	D/DT
B2	DAMPING IN TAKE-UP (0.178) (LB SEC/IN)
KA	STIFFNESS OF ACCUMULATOR REFLECTED TO X2 DISPLACEMENT (LBS/IN)
X2	TAKE-UP DISPLACEMENT (IN) (POSITIVE WINDING IN CABLE)
X2DOT	VELOCITY OF TAKE-UP PULLEY (IN/SEC)
X2DDOT	ACCELERATION OF THE TAKE-UP PULLEY (IN/SEC^2)
X3	CAM DISPLACEMENT (HYDRAULIC PISTON DISPLACEMENT) (IN)
C1	CAM NULL LENGTH (IN) - (NOTE: THE VALUE IS NOW ZERO)
K1	CAM SLOPE
XC	CAM OUTPUT DISPLACEMENT (IN)
A,B,L	LINKAGE LENGTHS ON MECHANISM COMBINING CAM DISPLACEMENT WITH VELOCITY FEEDBACK (IN)

SYMBOLS FOR CABLE REEL ANALYTICAL MODEL (CONT'D)

XC2	COMBINED UNCLIPPED OUTPUT OF CAM DISPLACEMENT AND FEEDBACK VELOCITY
XP	CAM FOLLOWER DISPLACEMENT (CLIPPED) TO SWASHPLATE (IN) . MAXIMUM PUMP FLOW IS ONE.
WP	PUMP SPEED (RADIANS/SEC)
DP	DISPLACEMENT OF PUMP (IN^3/RADIAN)
QP	PUMP OIL FLOW (IN^3/SEC)
V	VOLUME OF OIL WITHIN SYSTEM (IN^3)
BM	BULK MODULUS OF HYDRAULIC OIL (USING 100,000 LBS-IN^2)
KL	OIL LEAKAGE (IN^3/SEC/PSI)
P	OIL PRESSURE (LB/IN^2)
P1O	INITIAL OIL PRESSURE AT TIME O (PSI)
PDOT	RATE OF CHANGE OF HYDRAULIC MOTOR OIL PRESSURE (PSI/SEC)
PI	PRESSURE THAT WOULD OCCUR IF NO RELIEF VALVE PRESENT (PSI)
KLM	OIL LEAKAGE IN MOTOR (IN^3/SEC-PSI)

SYMBOLES FOR CABLE REEL ANALYICAL MODEL (CONT'D)

KLP	OIL LEAKAGE IN PUMP (IN^3/SEC PSI)
DM	DIAPLACEMENT OF THE MOTOR (IN^3/RADIAN)
TM	TORQUE ON THE MOTOR CAUSED BY CABLE (LB IN) (CCW IS POSITIVE)
TMO	TORQUE AT THE MOTOR, AT TIME O, CAUSED BY TRO PLUS THE VISCOUS DRAG CAUSED BY THE INITIAL VELOCITY OF THE MOTOR AND REEL (LB-IN)
JM	CABLE REEL POLAR MOMENT OF INERTIA (LB-IN-SEC^2)
BD	DAMPING IN PUMP-MOTOR AYATEM ($\frac{\text{LB IN SEC}}{\text{RADIAN}}$)
WM	MOTOR SPEED (RADIAN/SEC) (POSITIVE CCW)
WMDOT	ANGULAR ACCELERATION OF THE MOTOR (RAD/SEC^2)
WMO	INITIAL MOTOR SPEED (RAD/SEC) (CCW IS POSITIVE)
QM	MOTOR OIL FLOW (IN^3/SEC)
RJ	CABLE REEL INITIAL (IN)
RA	RATIO OF REEL SPROCKET TO MOTOR SPROCKET ($-\frac{76}{13}$ = 5.846)
VJ	WIND IN VELOCITY OF THE CABLE AT THE REEL (IN/SEC)

SYMBOLS FOR CABLE REEL ANALYTYCAL MODEL (CONT'D)

KFB	VELOCITY FEEDBACK CONSTANT
COM	INPUT TO THE FEEDBACK ADDING MECHANISM WHICH COMES FROM THE VELOCITY OF THE MOTOR (IN)
XJ	REEL DISPLACEMENT AT CABLE DIAMETER (IN) (CCW IS POSITIVE)
XJO	INITIAL DISPLACEMENT (STRETCH) IN CABLE (IN)
WJ	ANGULAR VELOCITY OF THR THE CABLE REEL (RAD/SEC)
XI	CABLE DISPLACEMENT AT BACK SPOOLER (IN) (WINDING IN IS POSITIVE)
VI	RELATIVE VELOCITY OF CABLE TO BACK SPOOLER (IN/SEC) (WINDING IN IS POSITIVE)
VIO	INITIAL VELOCITY OF CABLE PAST THE BACK SPOOLER (IN/SEC)
XIO	INITIAL CABLE DISPLACEMENT AT THE BACK SPOOLER (IN)
KS	SPRING CONSTANT OF CABLE (CONSIDERS CATENARY) (LB/IN)
FL	PULL LOAD ON CABLE (LBS) (CABLE IN TENSION IS POSITIVE)
TR	TORQUE ON THE REEL CAUSED BY CABLE, REFLECTED TO MOTOR (LB-IN)
TRO	INITIAL TORQUE ON THE REEL, AT TIME O, CAUSED BY INITIAL TENSION IN THE CABLE, REFLECTED TO MOTOR (LB-IN)
FP2	SUM OF THE EXTERNAL FORCES ACTING ON THE TAKE-UP PULLEY. (POSITIVE WHEN IN THE POSITIVE X2 DIRECTION)
KIT	FACTOR FOR STEP INCREMENT IN CABLE VELOCITY FOR WINDING IN (IN/SEC)
VIT	STEP INCREMENT IN TELATIVE VELOCITY BETWEEN THE CABLE AND BACK SPOOLER, EQUAL IN MAGNITUDE TO KIT (IN/SEC), IT OCCURSAT THE TIME INDICATED IN THE ARGUMENT OF THE STEP FUNCTION.

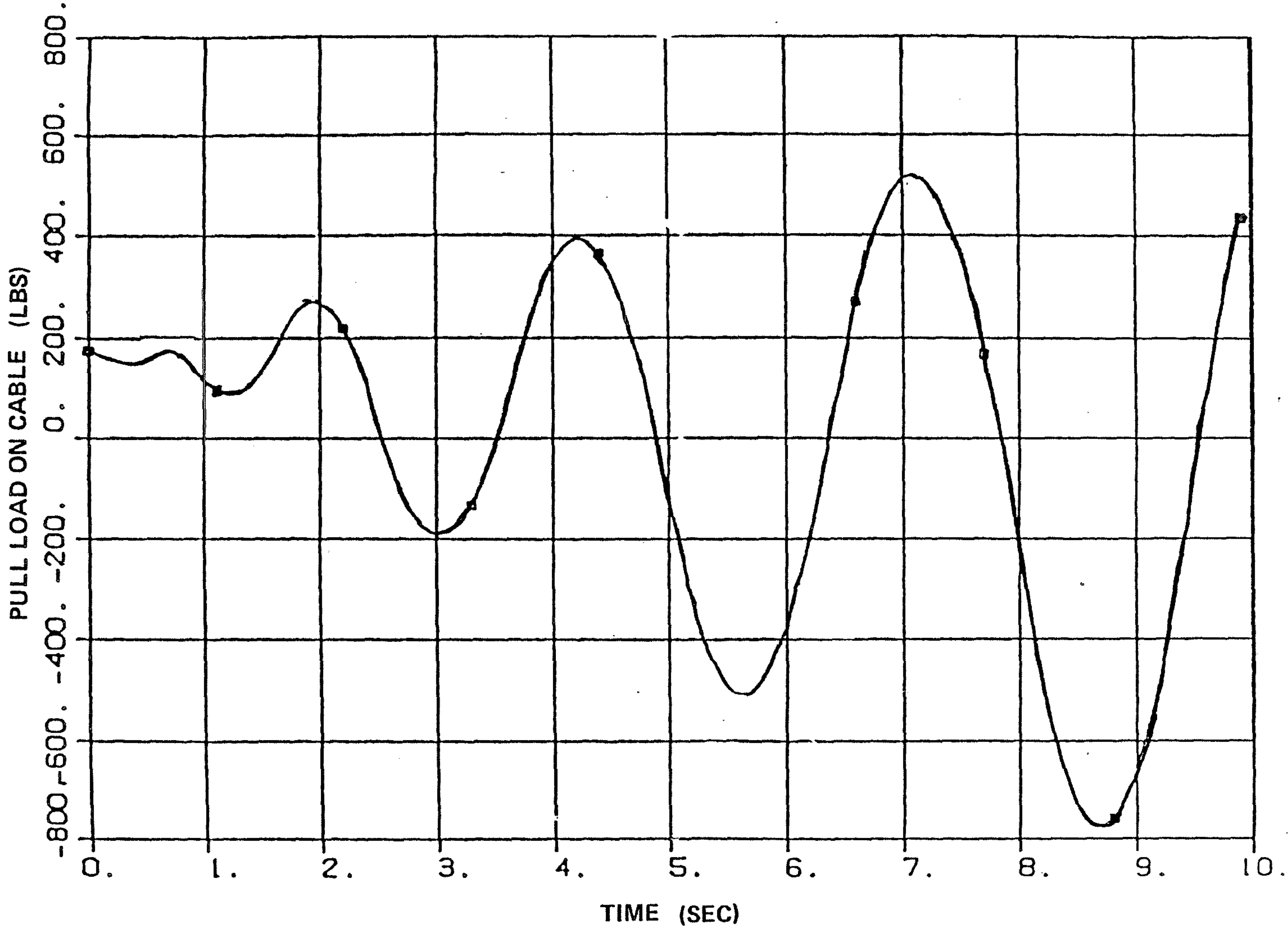


FIGURE 25 UNSTABLE OUTPUT (CABLE TENSION)

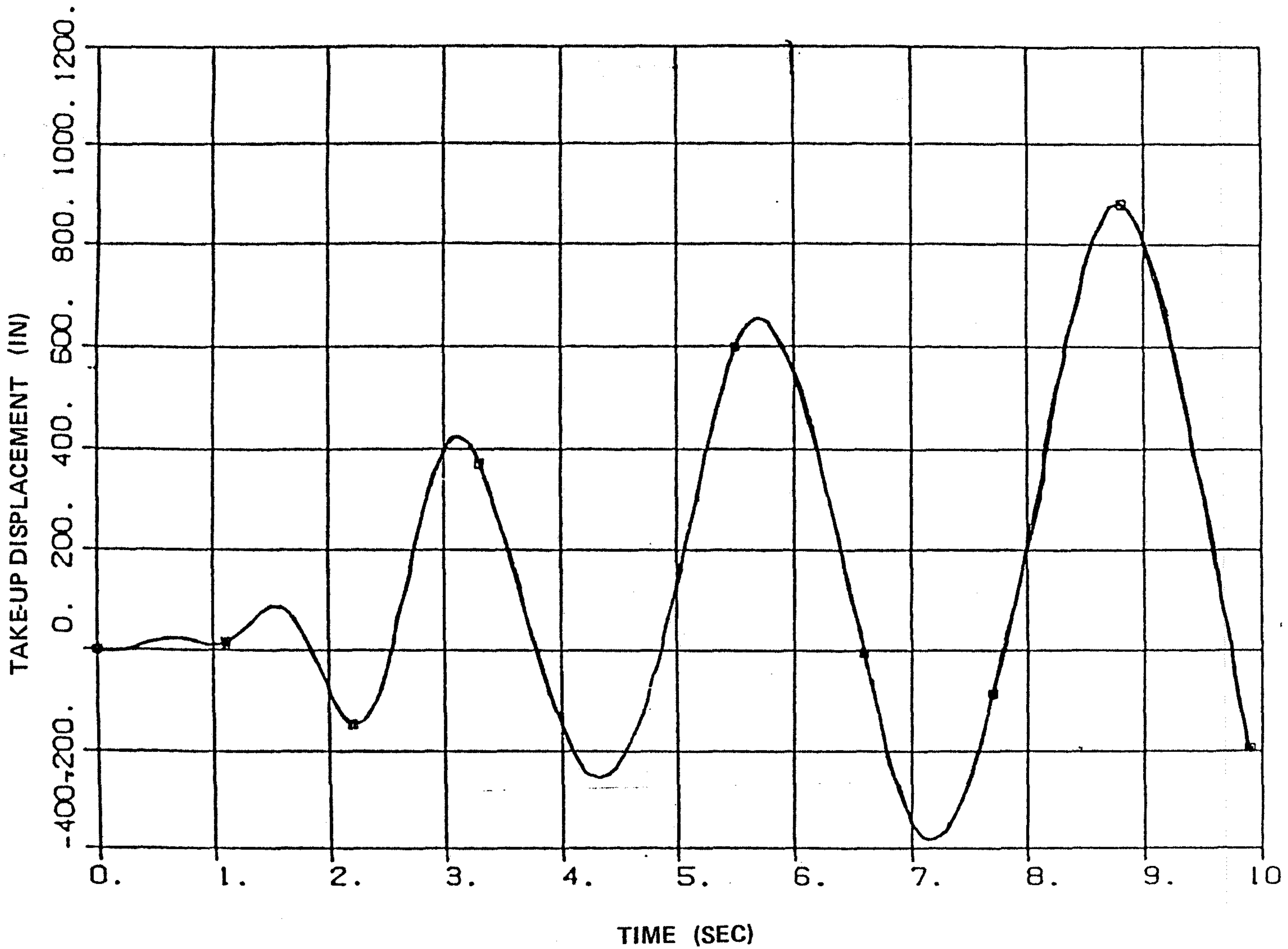


FIGURE 26 UNSTABLE OUTPUT (TAKE-UP MOVEMENT)

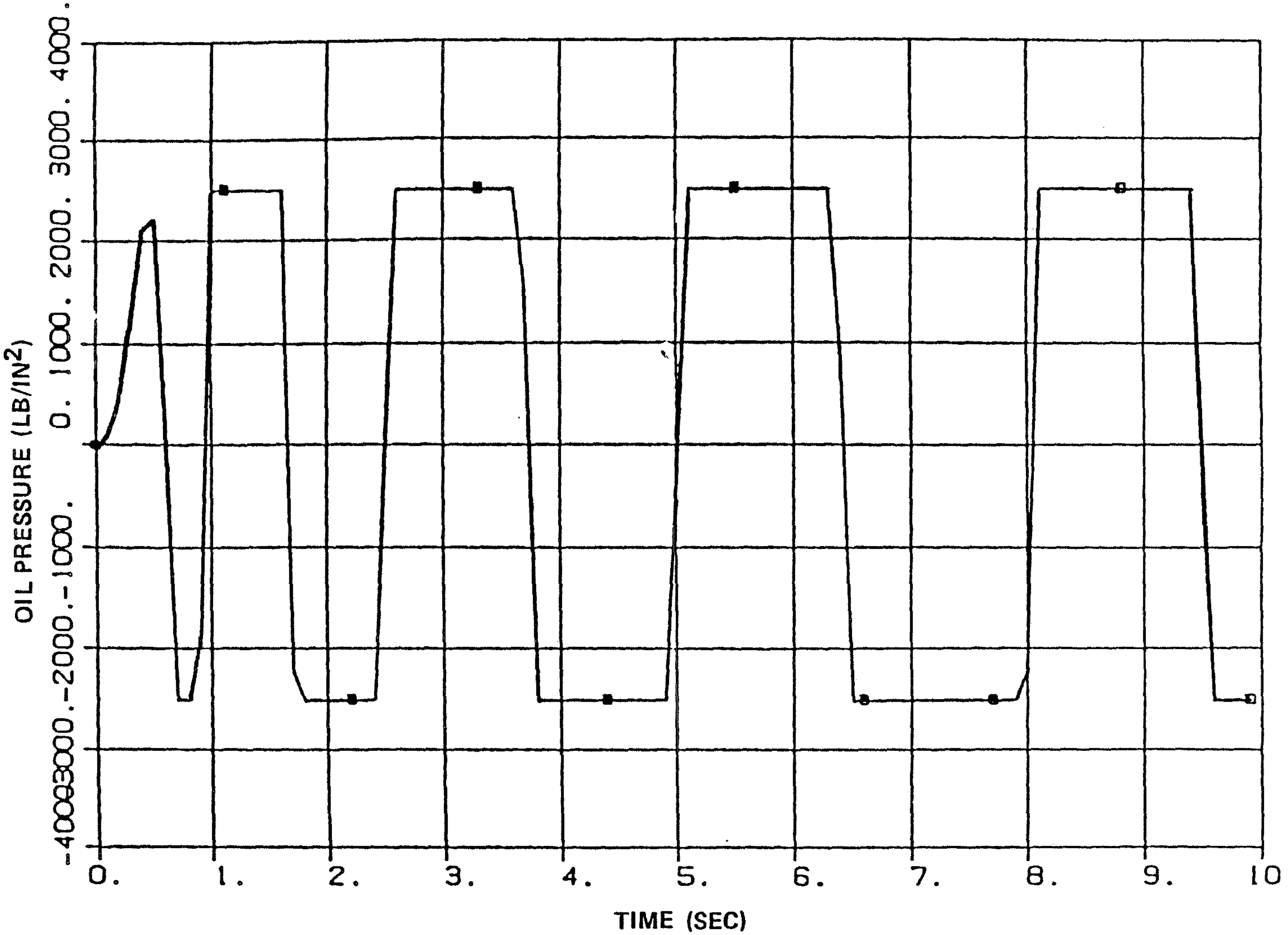


FIGURE 27 UNSTABLE OUTPUT (CLOSED LOOP PRESSURE)

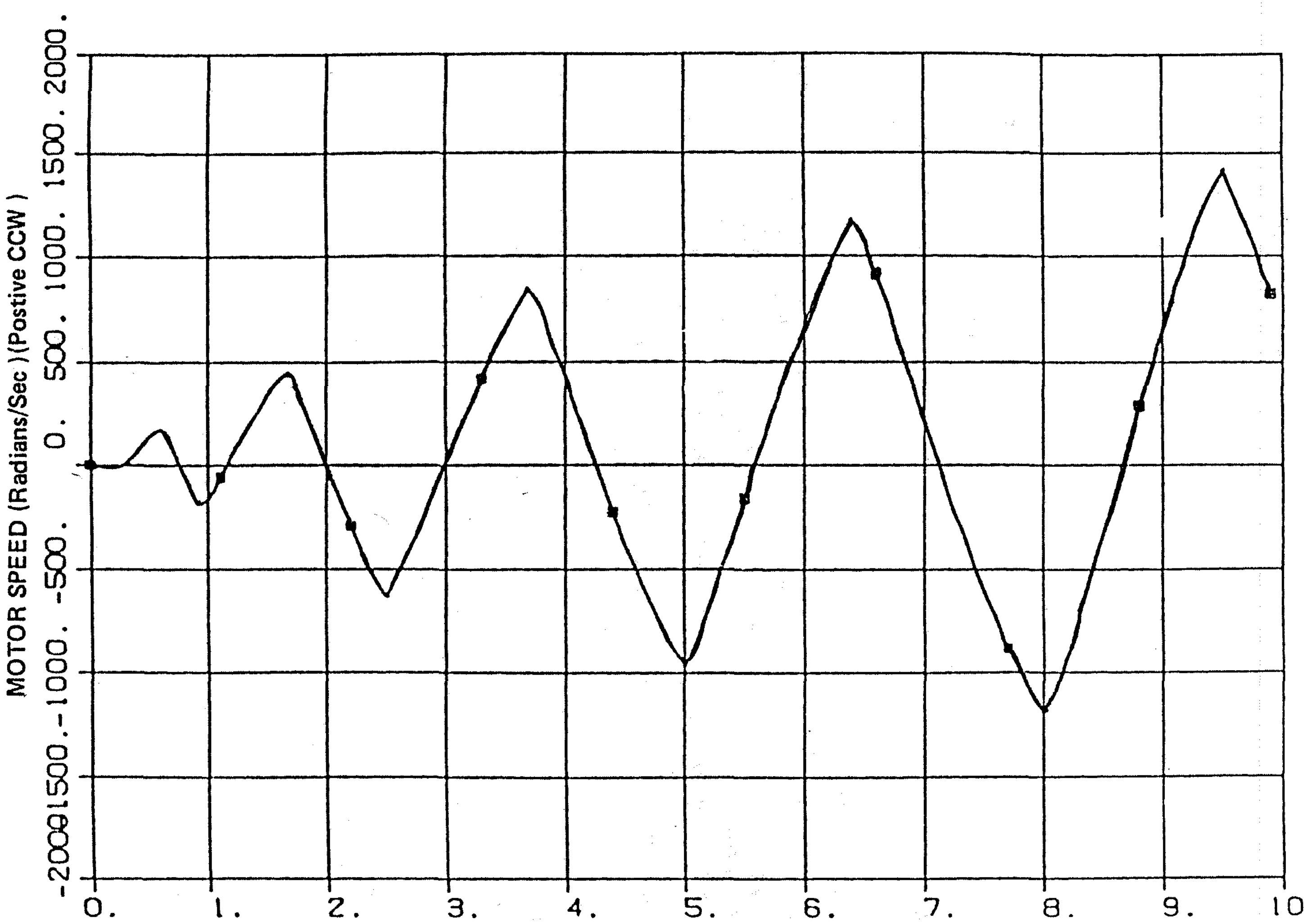


FIGURE 28 UNSTABLE OUTPUT (REEL DRIVE MOTOR SPEED)

cable reel inertia). Since we have little control over the reel inertia, we have minimized the "spring" effect of the hydraulic loop. This is accomplished by limiting the volume of oil in the loop and by increasing the stiffness of the piping (using steel tubes in place of hoses).

In addition to these easily-excitable, low-natural frequency systems, the overall cable reel drive is a low-friction device. This does not help in the overall stability, since friction tends to be a stabilizing factor. Horsepower requirements are dictated by vehicle performance parameters. This is a fixed figure based on a fixed reel inertia.

The horsepower and its rate of application form the basic system gain. The friction is the damping. Because of their interaction, the low-natural frequency systems tend to vibrate in harmony, with the gain (drastically larger than the damping), giving the system a boost each time the systems' cycle. This causes the unstable output shown on Figures 25-28.

4.3 Systems with Secondary Feedback Loops

Substantial time was spent trying to achieve a stable single-feedback loop system. This approach would not work, and a secondary signal had to be incorporated. The secondary signal reduces the gain of the overall drive system, while maintaining its ability to accelerate.

Several variations were modeled, sensing cable and reel velocities, accelerations, and the quantity of cable stored on the reel. The results show sensing the cable velocity at the cable reel and using that in connection with the primary follower feedback signal gives the best result. The amplitude of the secondary feedback system was varied extensively until maximum stability with a full and empty reel was achieved. Key parameter response with the empty cable reel is shown on Figures 29-32, while Figures 33-36 show key parameters with the cable reel full. (All are with a six mph step input.)

This system requires more time to stabilize when full than we would like. However, a stability factor review shows the system is only stable by a factor of two, which means if any single factor increases by two or decreases by one-half, the system will be unstable. This also applies to the net result of all system factors/non-linearities, etc. System designers do not like this factor less than three; therefore, it is <u>extremely</u> important that the mechanical components mimic the computer model. The final numerical values for the parameters used are listed in Figure 37.

Assuming the componentry imitates the model, the performance of the system and cable tensions would vary in accordance with Figure 38, as the vehicle performs various maneuvers.

The computer input and output data is shown on Figures 39-42 for stable full and empty reel cases.

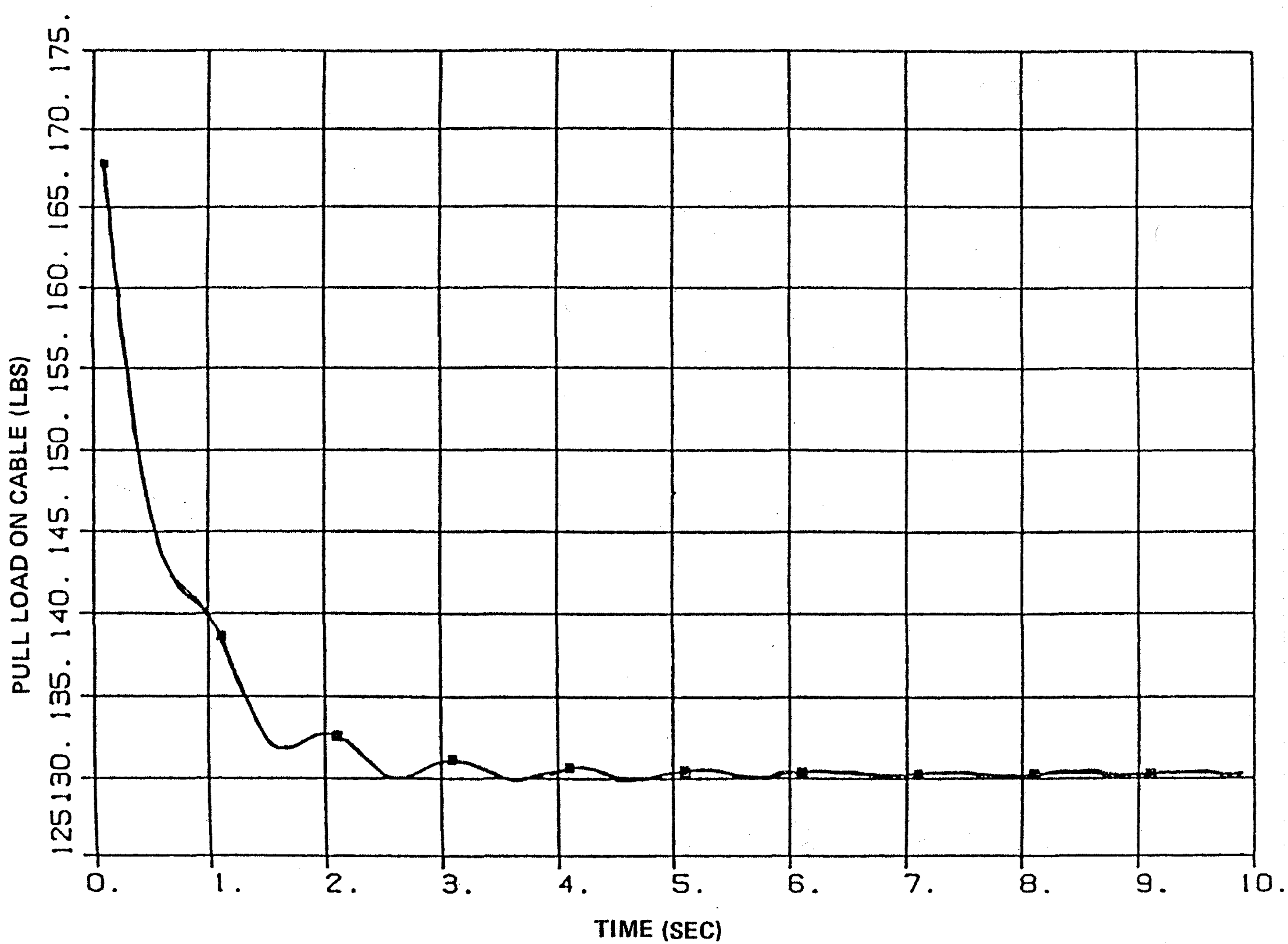


FIGURE 29 STABLE OUTPUT - EMPTY REEL (CABLE TENSION)

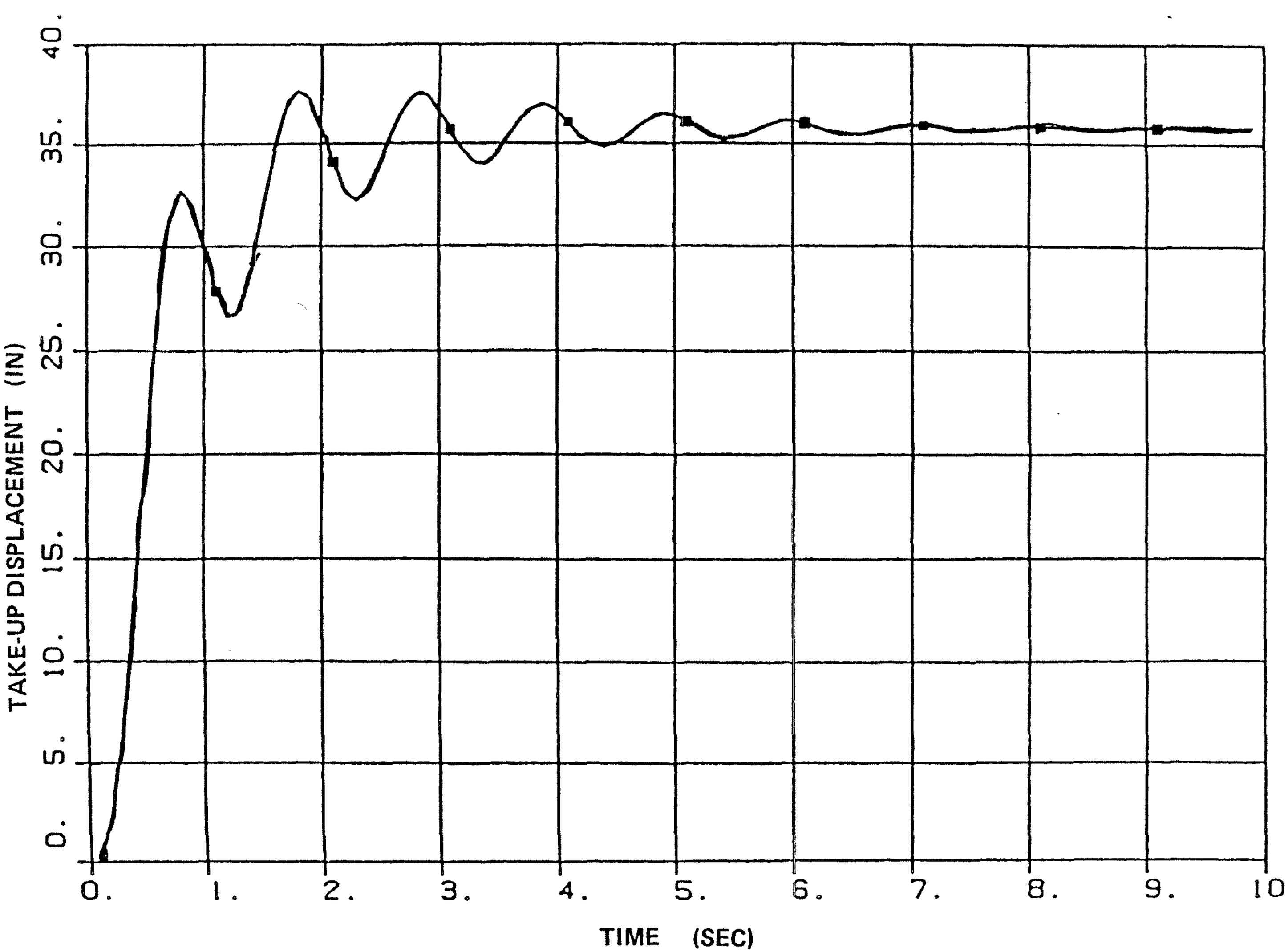


FIGURE 30 STABLE OUTPUT - EMPTY REEL (TAKE-UP MOVEMENT)

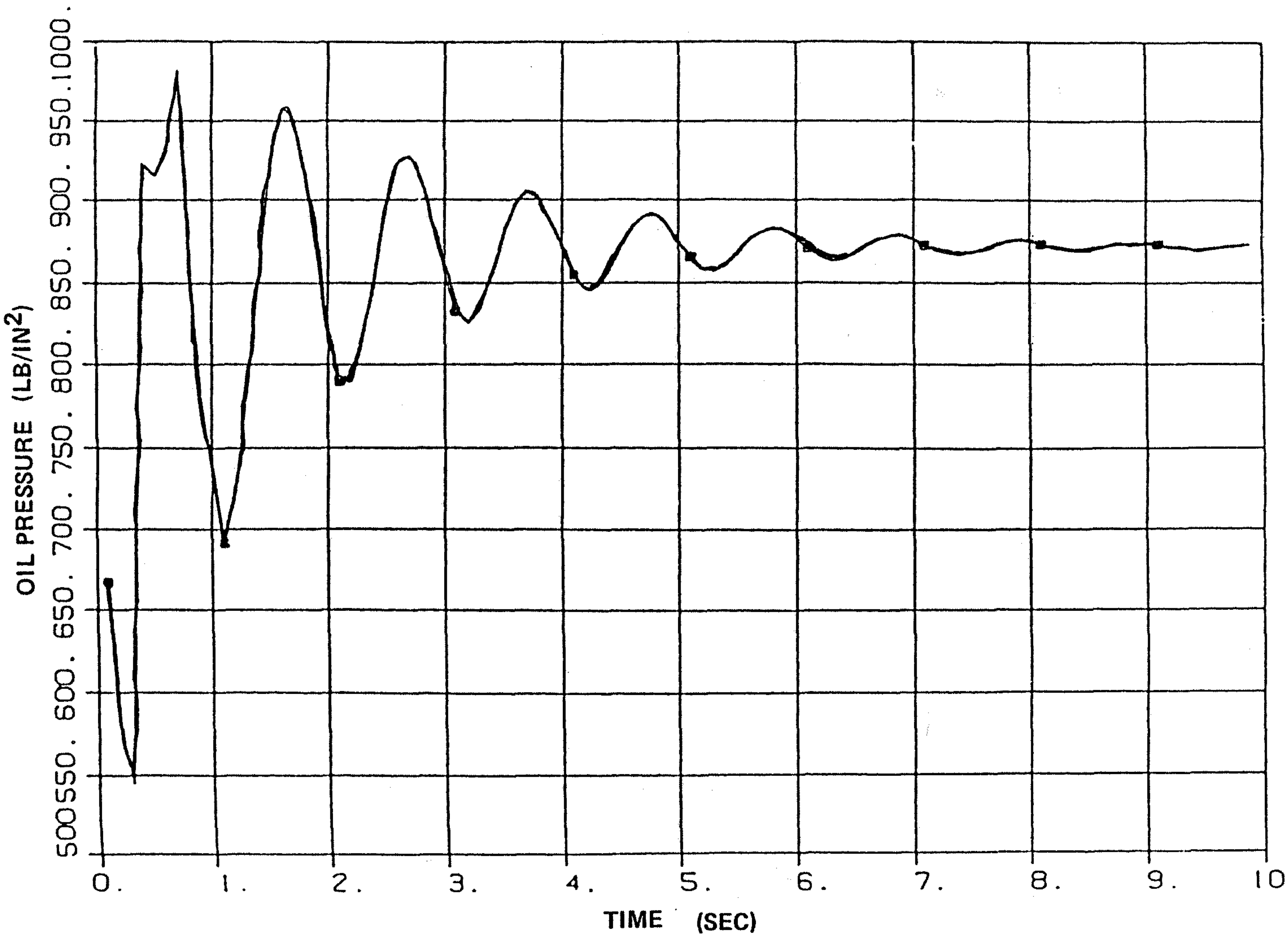


FIGURE 31 STABLE OUTPUT - EMPTY REEL (Closed Loop Pressure)

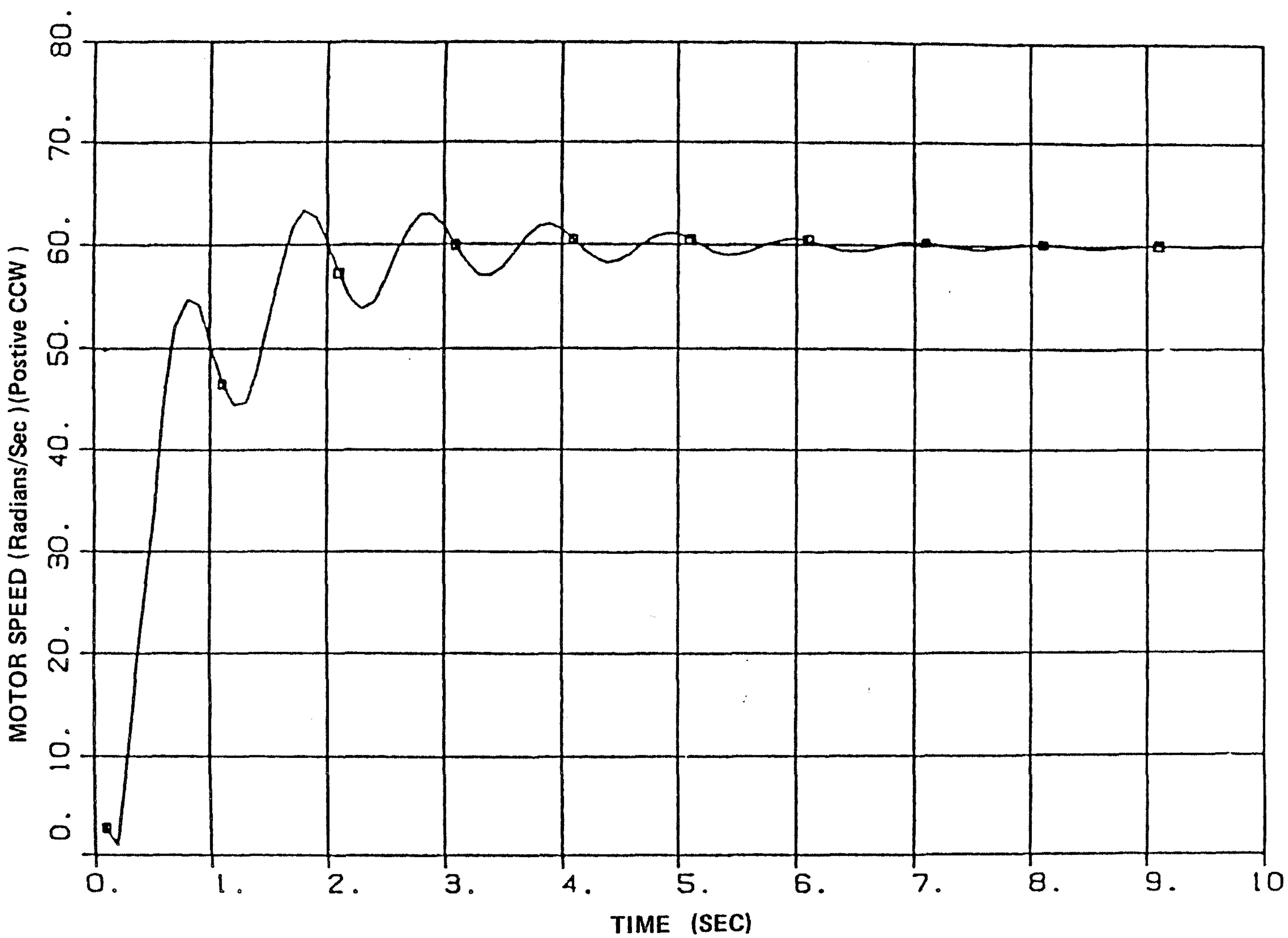


FIGURE 32 STABLE OUTPUT - EMPTY REEL (REEL DRIVE MOTOR SPEED)

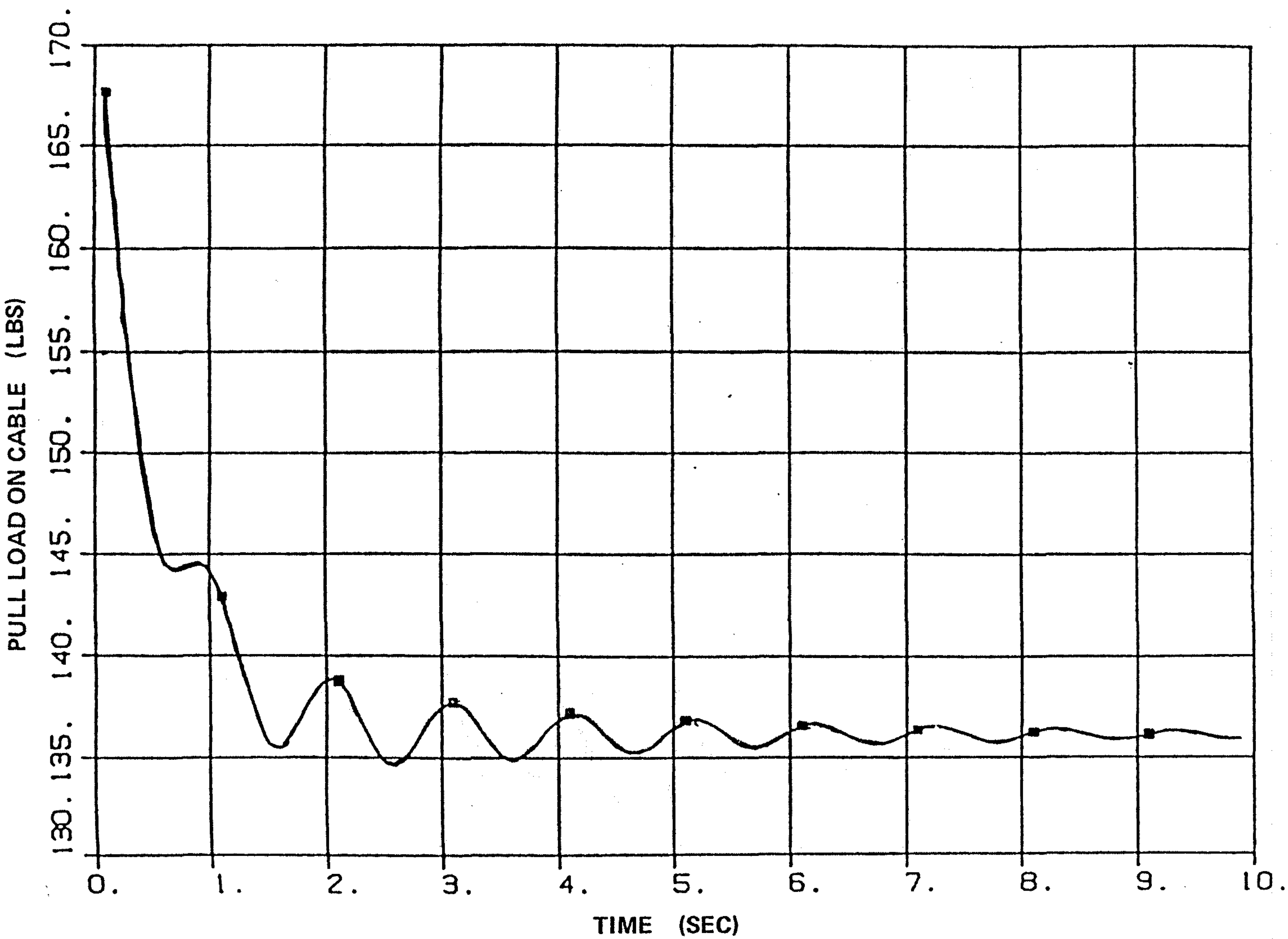


FIGURE 33 STABLE OUTPUT - FULL REEL (CABLE TENSION)

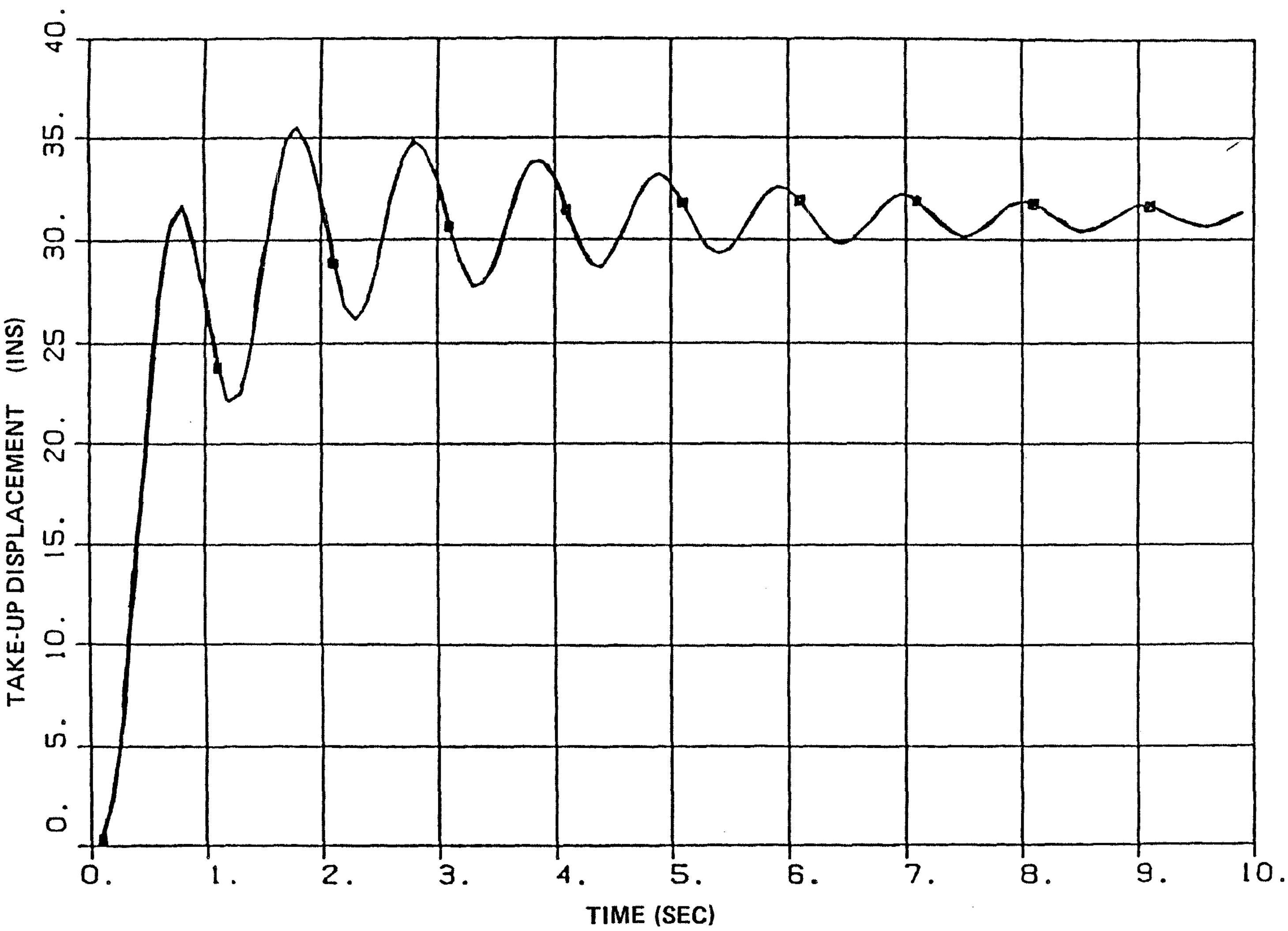


FIGURE 34 STABLE OUTPUT - FULL REEL (TAKE-UP MOVEMENT)

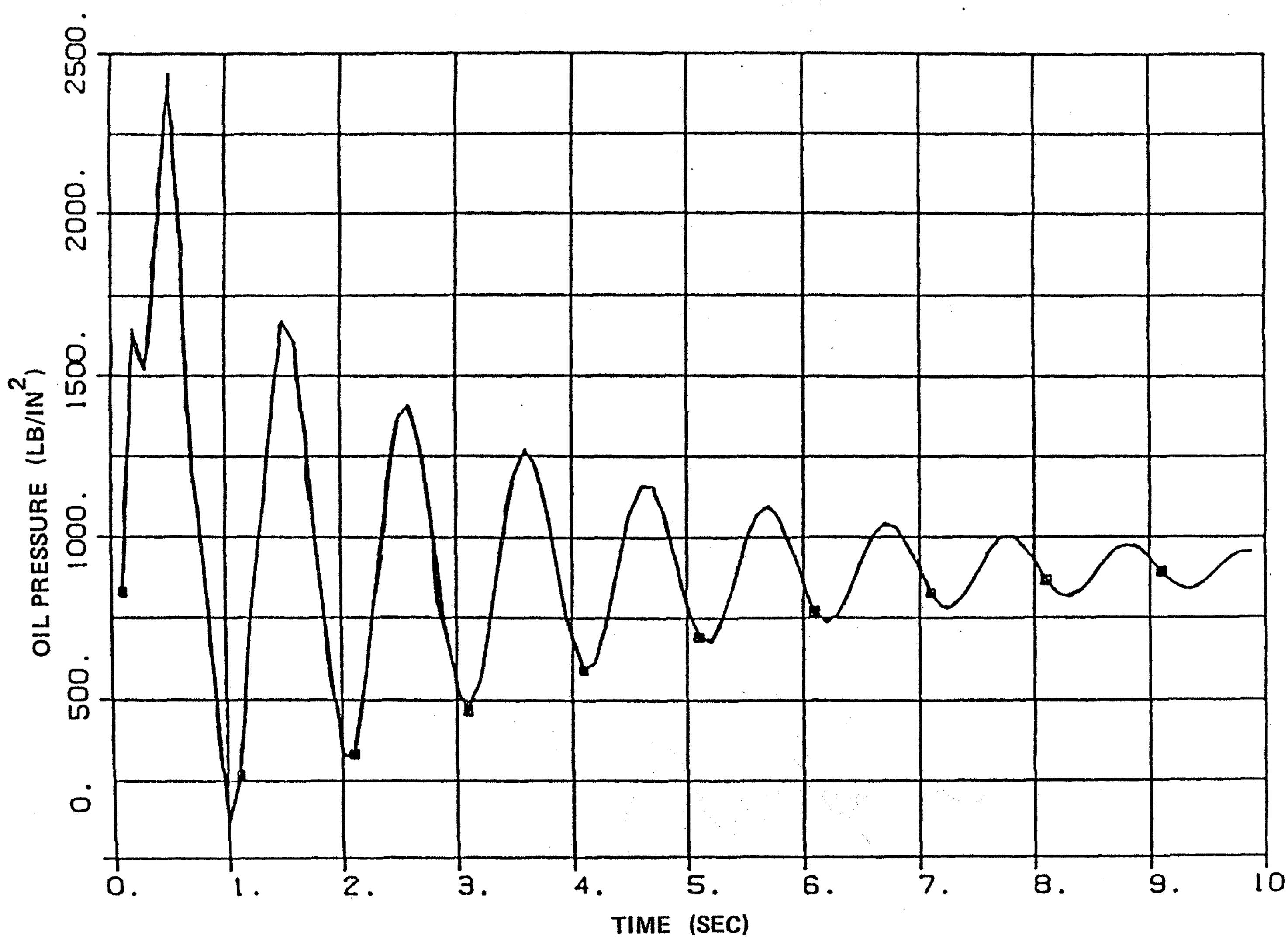


FIGURE 35 STABLE OUTPUT - FULL REEL (CLOSED LOOP PRESSURE)

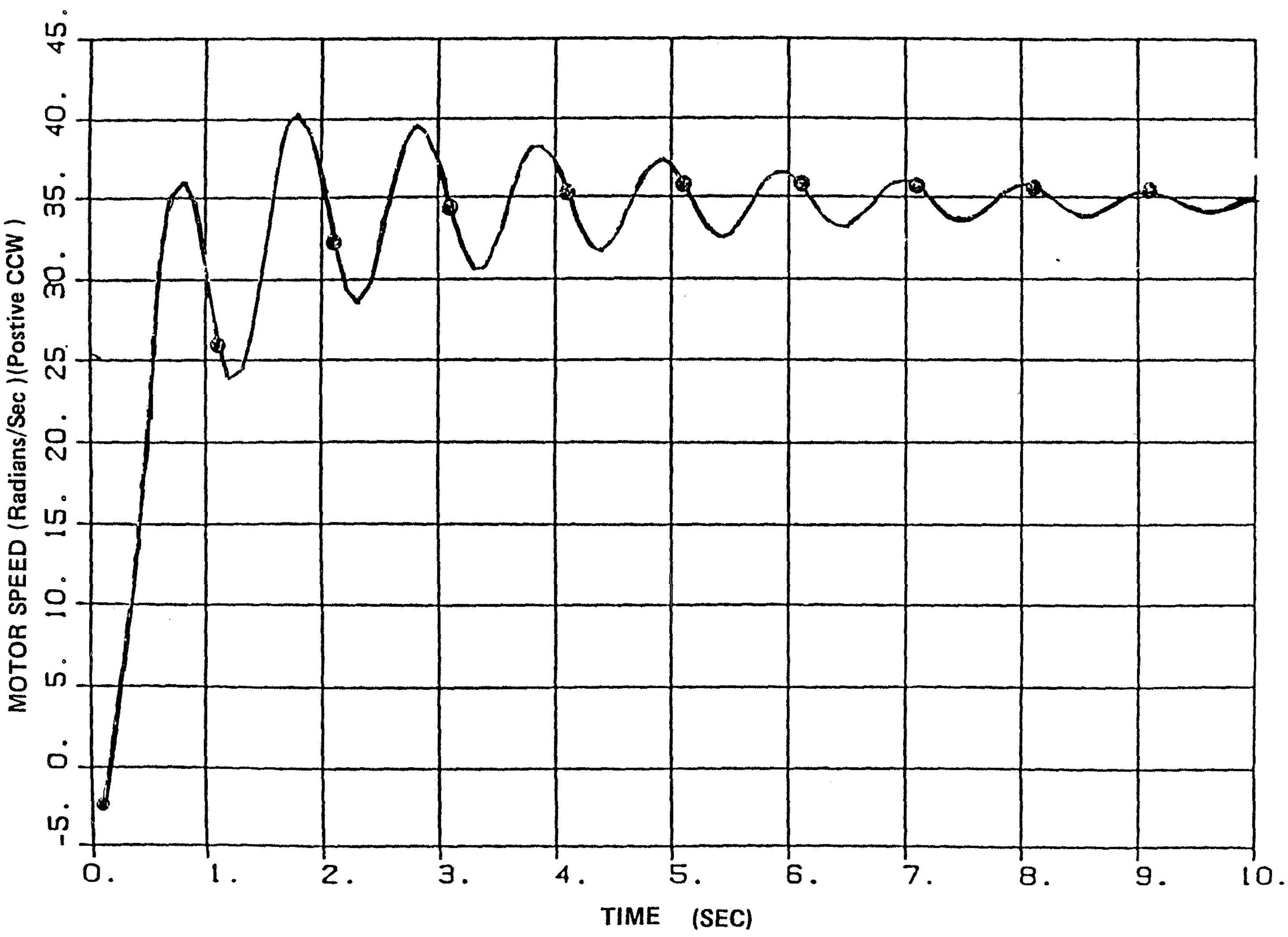


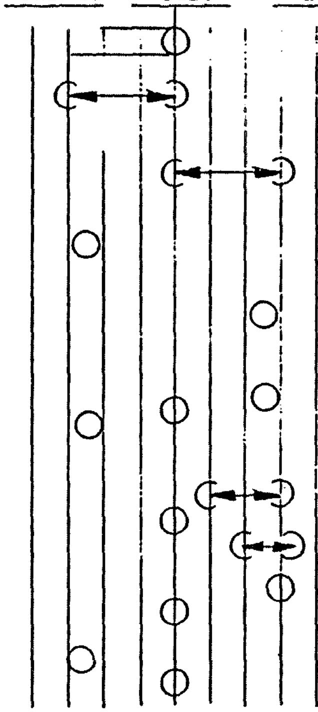
FIGURE 36 STABLE OUTPUT - FULL REEL (REEL DRIVE MOTOR SPEED)

CABLE PARAMETER VALUES – AS SHOWN ON COMPUTER

M2	.0648 LB-SEC^2/IN
B2	0.178 LB-SEC/IN
KA	2.5 LBS/IN
KI	0.3332
WP	198.97 RAD/SEC
DP	0.366 IN^3/RAD
V	60 IN^3
BM	100,000 LBS/IN^2
DM	0.678 IN^3/RAD
JM	1.46 LBS IN SEC (FIRST LAYER) 10.71 LBS IN SEC (FULLY LAYERED)
BD	$6.2 \dfrac{\text{LBS IN SEC}}{\text{RAD}}$
RJ	9.85 IN (FIRST LAYER) 17.0 IN (FULLY LAYERED)
RA	5.846
KS	0.6917
VI	105.6 IN/SEC (/ MPH)
KIT	105.6 IN/SEC
KFB	.03
VIO	1.0 IN/SEC
KLM	.00077 IN^3/SEC PSI
KLP	.00077 IN^3/SEC PSI
XIO	0.0 IN

FIGURE 37 FINAL VALUES OF CONSTANTS USED IN MODEL

CALCULATED SHUTTLE CAR CABLE REEL RESPONSE



CONDITION	4Ft. Rearwd.	Take-up Ctr.	4Ft. Fwd.	Fl(lbs)	Pull(lbs)	Rotation	Torque
Car Stopped, but powered				175	175	Stop	Res.
Start from stop and paying out cable				215 to 220	155 to 160	CW	Res.
Start from stop and reeling in cable				130 to 135	190 to 195	CCW	Aid
Constant velocity Straight & Paying Out				215 to 220	155 to 160	CW	Res.
Constant velocity, Straight & Reeling in cable				130 to 135	190 to 195	CCW	Aid
Passing Tie point, reel in, to pay out				190 175 220	235 to 190 175 115 to 160	CCW Stop CW	Aid Res. Res.
90 degree turn approaching tie point with reel on outside of turn radius				160 to 130 175 120 to 125	100 to 70 175 180 to 185	CCW Stop CCW	Aid Res. Aid
Max. velocity reeling in & coming to stop				130 to 135 175	190 to 195 175	CCW Stop	Aid Res.
Max. velocity paying out & coming to stop				215 to 220 175	155 to 160 175	CW Stop	Res. Res.

NOTE: Assumes 60 lb. pull of cable to overcome friction

FIGURE 38 CALCULATED SHUTTLE CAR CABLE REEL RESPONSE

```
*** CONTINUOUS SYSTEM SIMULATION LANGUAGE-VERSION FOUR (CSSL-IV) ***
      *** SUPPORTED BY SIMULATION SERVICES, CHATSWORTH, CA. ***
           *** CDC CYBERNET ECZ    CSSL-IV VERSION 4.2 ***
                *** DATE 82/09/14.   TIME 14.36.53. ***


PROGRAM XC FEEDBACK LOOP WITH LEVER
   INITIAL
       CONSTANT  BD=6,2, BM= 100000.,B2=0,178,DM=0.678,DP=0.366
       CONSTANT  JM=1.46,KFB=0.03,KA=2.5,KIT=100.,KLM=0.00077
       CONSTANT  KLP=0.00077,KS=0.6917,K1=.3332,M2=0.0648,RA=5.846
       CONSTANT  RJ=9.85,V=60.,V10=1.,WP=198.97,X10=0.0
       CONSTANT  L=10.0,A=4.0,PMAX=2500.0,PMIN=-2500.0
       B=L-A
       WMO=RA*VIO/RJ
       XJO=175./KS
       TRO=-175.*RJ/RA
       TMO=-TRO+BD*WMO
       PIO=TMO/DM
       XPO=(PIQ*(KLM+KLP)+(WMO*DM))/(WP*DP)
       XCO=(XPO+KBF*WMO*A/L)*L/B
       X30=-XCO/K1
       X20=-4.*X30
   END INITIAL
   DYNAMIC
      CINTERVAL  CI=0.1
          DERIVATIVE CABLE
             PDOT=(BM/V)*(WP*DP*XP-DM*WM-(KLM+KLP)*P)
             P1=INTEG(PDOT,PIO)
             P=BOUND(PMIN,PMAX,P1)
             WMDOT=(TR+DM*P-BD*WM)/JM
             WM=INTEG(WMDOT,WMO)
             WJ=WM/RA
             VJ=WJ*RJ
             XJ=INTEG(VJ,XJO)
             VIT=KIT*STEP(0.0.T)
             VI=VIO+VIT
             XI=INTEG(VI,XIO)
             F1=KS*(XJ-XI)
             TR=-RJ*F1/RA
             FP2=-2.*F1+350.
             X2DDOT=(FP2-KA*X2-B2*X2DOT)/M2
             X2DOT=INTEG(X2DDOT,0.0)
             X2=INTEG(X2DOT,X20)
             X3=-X2/4
             COM=RJ*KFB*WM/RA
             XC=-K1*BOUND(-12.,12.,X3)
             XC2=B*XC/L-A*COM/L
             XP=1.0*BOUND(-1.,1.,XC2)
          END DERIVATIVE
    TERMT(T.GE.10.)
          PREPAR T,F1,X2,WM,P,DGT,PDOT,XP
   END DYNAMIC
   TERMINAL
   END TERMINAL
END PROGRAM
```

FIGURE 39 INPUT DATA FOR EMPTY REEL CASE

*** CSSL-IV VERSION 4.2   82/09/14.   14.43.19. ***

CABLE3 REVERIFY WITH CASE1 RERUN

T	F1	X2	WM	P	XP
0.	1.750000E+02	1.105573E-01	5.935025E-01	0.	-6.474347E-03
5.000000E-01	1.459395E+02	2.035561E+01	3.269622E+01	9.142592E+02	3.562902E-01
1.000000E+00	1.400011E+02	3.005374E+01	5.045434E+01	7.367136E+02	4.819520E-01
1.500000E+00	1.323952E+02	3.162793E+01	5.275952E+01	9.248023E102	5.140215E-01
2.000000E+00	1.327443E+02	3.588016E+01	6.025761E+01	8.174688E+02	5.749446E-01
2.500000E+00	1.302551E+02	3.388633E+01	5.661529E+01	8.957703E+02	5.489371E-01
3.000000E+00	1.310216E+02	3.678094E+01	6.171945E+01	8.551581E+02	5.904087E-01
3.500000E+00	1.300889E+02	3.458326E+01	5.784217E+01	8.779931E+02	5.589630E-01
4.000000E+00	1.305183E+02	3.667579E+01	6.150231E+01	8.701786E+02	5.895437E-01
4.500000E+00	1.301784E+02	3.498283E+01	5.854914E+01	8.698846E102	5.646394E-01
5.000000E+00	1.303398E+02	3.641379E+01	6.103480E+01	8.754702E+02	5.859013E-01
5.500000E+00	1.302493E+02	3.527063E+01	5.905439E+01	8.673606E+02	5.688081E-01
6.000000E+00	1.302723E+02	3.618440E+01	6.063277E+01	8.765087E+02	5.825652E-01
6.500000E+00	1.302820E+02	3.547406E+01	5.940904E+01	8.674319E+02	5.718048E-01
7.000000E+00	1.302499E+02	3.602026E+01	6.034728E+01	8.759495E+02	5.801340E-01
7.500000E+00	1.302914E+02	3.560859E+01	5.964246E+01	8.683654E+02	5.738093E-01
8.000000E+00	1.302464E+02	3.591413E+01	6.016434E+01	8.748399E+02	5.785283E-01
8.500000E+00	1.302901E+02	3.569181E+01	5.978519E+01	8.694271E+02	5.750828E-01
9.000000E+00	1.302499E+02	3.585036E+01	6.005461E+01	8.738669E+02	5.775596E-01
9.500000E+00	1.302852E+02	3.574002E+01	5.986955E+01	8.703960E+02	5.757864E-01

FIGURE 40 TABULATED OUTPUT FOR EMPTY REEL CASE

```
  PROGRAM XC FEEDBACK LOOP WITH LEVER
     INITIAL
         CONSTANT  BD=6.2,BM=100000.,B2=0.178,DM=0.678,DP=0.366
         CONSTANT  JM=1.46,KFB=0.03,KA=2.5,KIT=100.,KLM=0.00077
         CONSTANT  KLP=0.00077,KS=0.6917,K1=.3332,M2=0.0648,RA=5.846
         CONSTANT  RJ=9.85,V=60.,V10=1.,WP=198.97,XI0=0.0
ENQUIRE  CONSTANT  L=10.0,A=4.0,PM
//
NET 024086
PLEASE SIGN ON--KD
82/09/15. 14.27.10.
EASTERN CYBERNET CENTER SN123 NOS       1.3/477.769/8AD
USER NAME:  85965
PASSWORD

TERMINAL:    136,TTY
RECOVER/ CHARGE: RECOVER, 143

RECOVERY COMPLETE.
  LAST COMMAND    =CALL
  JOB STATUS      =OUTPUT AVAILABLE
  NEXT OPERATION  =CONTINUE
PRESS RETURN KEY TO CONTINUE.

PO=(PIO*(KLM+KLP)+(WMO*DM))/(WP*DP)
         XCO=(XPO+KBF*WMO*A/L)*L/B
         X30=-XCO/K1
         X20=-4.*X30
     END INITIAL
     DYNAMIC
        CINTERVAL  CI=0.1
            DERIVATIVE CABLE
               PDOT=(BM/V)*(WP*DP*XP-DM*WM-(KLM+KLP)*P)
               P1=INTEG(PDOT,PIO)
               P=BOUND(PMIN,PMAX,P1)
               WMDOT=(TR+DM*P-BD8WM)/JM
               WM=INTEG(WMDOT,WMO)
               WJ=WM/RA
               VJ=WJ*RJ
               XJ=INTEG(VJ,XJO)
               VIT=KIT*STEP(0.0,T)
               VI=VIO+VIT
               XI=INTEG(VI,XIO)
               F1=KS*(XJ-XI)
               TR=RJ*F1/RA
               FP2=-2.*F1+350.
               X2DDOT=(FP2-KA*X2-B2*X2DOT)/M2
               X2DOT=INTEG(X2DDOT,0.0)
               X2=INTEG(X2DOT,X20)
               X3=-X2/4
               COM=RJ*KFB*WM/RA
               XC=K1*BOUND(-12.,12.,X3)
               XC2=B*XC/L-A*COM/L
               XP=1.0*BOUND(-1.,1.,XC2)
            END DERIVATIVE
      TERMT(T.GE.10.)
            PREPAR T,F1,X2,WM,P,DOT,PDOT,XP
```

FIGURE 41 INPUT DATA FOR FULL REEL CASE

*** CSSL-IV VERSION 4.2 82/09/15. 14.34.03. ***

FD BK JM=10.7,RJ=17,M2=.0648

T	F1	X2	WM	P	XP
0.	1.750000E+02	6.405819E-02	3.438824E-01	0.	-8.798372E-03
5.000000E-01	1.460250E+02	2.074221E+01	2.218555E+01	2.440852E+03	2.625164E-01
1.000000E+00	1.442228E+02	2.691005E+01	3.067057E+01	1.076663E+02	2.746948E-01
1.500000E+00	1.356774E+02	2.862029E+01	3.131736E+01	1.674073E+03	3.376023E-01
2.000000E+00	1.389041E+02	3.175604E+01	3.601117E+01	3.383999E+02	3.305334E-01
2.500000E+00	1.347891E+02	2.924372E+01	3.213932E+01	1.375485E+03	3.400786E-01
3.000000E+00	1.375399E+02	3.279079E+01	3.709176E+01	5.333196E+02	3.445423E-01
3.500000E+00	1.351512E+02	2.920761E+01	3.218294E101	1.192083E+03	3.367519E-01
4.000000E+00	1.369154E+02	3.293452E+01	3.717238E+01	6.810525E+02	3.489126E-01
4.500000E+00	1.355588E+02	2.931443E+01	3.238583E+01	1.064830E+03	3.350102E-01
5.000000E+00	1.365301E+02	3.277270E+01	3.690831E+01	7.858699E+02	3.500398E-01
5.500000E+00	1.358574E+02	2.955996E+01	3.274230E+01	9.783857E+02	3.348428E-01
6.000000E+00	1.362825E+02	3.249250E+01	3.651653E+01	8.555029E+02	3.497070E-01
6.500000E+00	1.360507E+02	2,986352E+01	3.315542E+01	9.231690E+02	3.355987E-01
7.000000E+00	1.361321E+02	3.218428E+01	3.610405E+01	8.982554E+02	3.486959E-01
7.500000E+00	1.361634E+02	3.016597E+01	3.355496E+01	8.909484E+02	3.367730E-01
8,000000E+00	1.360502E+02	3.189649E+01	3.572768E+01	9.216710E+02	3.474457E-01
8.500000E+00	1.362198E+02	3.043386E+01	3.390234E+01	8.747513E+02	3.380400E-01
9.000000E+00	1.360143E+02	3.165192E+01	3.541286E+01	9.320348E+02	3.462077E-01
9.500000E+00	1.362395E+02	3.065331E+01	3.418294E+01	8.690090E+02	3.392163E-01

FIGURE 42 TABULATED OUTPUT FOR

FULL REEL CASE

SECTION V

SYSTEM APPLICATION TO ALTERNATE VEHICLES

5.1 Modifications Required Because of Simple Hardware Differences Only

Alternate designs (in particular, those relating to the end of cable shutdown and warning and pump drive) will be required to fit the various manufacturer's vehicles. Because of the many different designs in the field, these will have to be addressed on a vehicle-by-vehicle basis but should have no major affect on the overall system performance.

No attempt has been made to review alternate designs within the scope of this contract.

5.2 Modifications Affecting Major Design Parameters

In general, changes to major parameters should be avoided. However, a review and computer model should be generated to determine system stability if major parameters are changed. Following are some general guidelines allowing changes in major parameters which should improve stability.

A. A reduction in the diameter and inertia of the cable reel will improve stability, although cable reel drive ratios would have to be changed to increase the rotational velocity of the reel and make up for the decreased minimum effective cable reel radius. The closed loop hydraulic system pressure must then also be dropped in proportion to the change in inertia. (While decreasing the reel size is possible, increasing it is not.)

B. An increase in the spring rate of the take-up pulley will improve stability, although it will cause larger variations in operational cable tensions. (This is accomplished by decreasing the size of the accumulator or increasing the displacement of the cylinder.)

C. A decrease in vehicle operational speed. The system is sized to a vehicle speed of six mph. If the vehicle operates with a maximum speed of four and one-half mph, it is possible to decrease the reel drive speed by 25 percent (although no change in the closed loop pressure is possible). It would also be possible to cut down some on the take-up storage capability (probably also 25 percent), but the slope of the cam would have to be changed to ensure full cam output at the extremes of the take-up pulley travel.

5.3 Physical Limitations

When fitting this system onto a low vehicle, some sacrifices would have to be made in height or the system would have to be set into the vehicle frame with the trail cable straddling the wheel on the oscillating axle. Take-up travel would also be limited by the available space between the tires on the cable reel side of the car. Some trade-offs would be obtained in system stability as the reel is also a good deal smaller and the driving torque could be reduced. Enough variables would change, however, to require modeling the system before attempting to test any hardware. Pump drive facility and space is also limited, therefore this system may not be easily installed on low vehicles.

SECTION VI

COST CONSIDERATIONS

6.1 Initial Cost

It must be recognized this system (being more complex than existing vehicle cable reel systems) would probably add $6,000 to the vehicle cost. However, if it saved one cable a year, pay-back would be in three years (based on a $2,000 cable cost).

Several maintenance and cost-saving modifications could be made after testing, bringing the system cost down to about $5,000 (possibly less on smaller cars). Among these are:

A. An elimination of the cam and use of simpler (but less flexible) linkage system.

B. Down-sizing the driving pump and motor.

C. Simplification of the secondary feedback linkage and feedback ratioing levers.

6.2 Cable Handling System Operating Cost

Because of the system complexity and medium pressure hydraulics, some operating expenses are inevitable. However, since all components are oversized, these costs should be minimal. In addition, the actual sub-systems are simple enough for the typical underground mechanic to troubleshoot and downtime should not be excessive.

Since existing cable reel values are not trouble-free, it is not expected that this system would require any more maintenance than existing systems, however, component costs may be slightly higher.

6.3 Possible Cable Cost Savings

One cable saved a year was mentioned earlier, but the actual savings are unknown. Cable damage is caused by many different factors, only some of which are related to the on-board cable handling.

Examples of damage preventable by a 100 percent effective cable reel drive system are:

A. Damage caused by tension spikes passing the tie point.

B. Damage caused by the cable sawing against the rib.

C. Damage caused by cable whipping.

D. Damage caused by cable jams.

Examples of damage which cannot be protected against by cable handling system design are:

A. Other vehicles tramming over cable.

B. Faulty splices/water penetration.

C. Cable overheating and baking.

D. Damage caused by improper clamping at tie point.

No study could be found that addressed the ratios of these various types of damage. Therefore, it is difficult to determine the total cost savings possible. If 50 percent of the cable damage is assumed to be handling system-related and a typical shuttle car requires the replacement of its cable two to six times per year (based on our shuttle car trial history), it is possible to save one to three cables or $2,000 to $6,000 per year in direct cable costs and a considerably higher amount in downtime related to cable repairs and replacement.

If cables are replaced after six splices, and each splice takes 45 minutes of lost production (actually 30 minutes per splice, but assume that half the time the second car is blocked and cannot run), the total lost production per cable is four and one-half hours. If it is further assumed that cables are replaced on a maintenance shift with no resulting downtime and that typical mine production is about 130 tons per hour if no lost time is encountered (the last figure is estimated, based on high-seam production of about 800 tons per six hours, working at the face), production losses per cable amount to 585 tons, or $17,550, at $30/ton. The above figures are all very conservative but show that significant savings can be achieved with the savings of only one cable over the life of the machine. In reality, savings could run between $19,550 and $58,650 per year.