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NATIONAL INSTITUTE OF OCEANOGRAPHY

WORMLEY, GODALMING, SURREY

The Design of a Servo-Controlled Winch for R. R. S. Discovery

with Compensation for Ship Motion

(INTERIM REPORT)

by

C. H. CLAYSON

N. I. O. Internal Report No. A. 40

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(Interim Report)

Summary

This report includes some preliminary considerations in the design of a servo control system for an electrical hydrographic winch which compensates for ship motion in the following manner: a heave velocity signal, obtained from a gyro-stabilised vertical accelerometer and integrator, is used to control the wire velocity relative to the ship, thereby cancelling the effect of the ship's heave on the motion of the wire relative to the Earth and consequently removing the fluctuations in cable tension which would normally result from the ship's motion.

Recordings of ship motion were made on R.R.S. DISCOVERY for the purpose of specifying the winch motor ratings and the analysis of these recordings is presented, together with a discussion of the servo system in general and predictions of performance of the proposed system.

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1. Introduction

The measurement of vertical profiles of such features as temperature and salinity may be hindered by the vertical motion (heave) of the ship, which will cause fluctuations in the depth of the measuring instruments (e.g. Temperature-Salinity-Depth recorder or water bottles) below the mean water surface level. Thus, the detection and, in particular, the measurement of any fine detail in the vertical profiles, with a scale of less than the peak to peak vertical displacement of the ship, may be rendered impossible: this will also depend upon the spatial and temporal resolution of the sensors, of course.

In order to prevent this undesirable transmission of the ship's motion to the instruments, one must isolate the instruments and cable from the ship motion. This requirement is facilitated, to some extent, by the mass inertias of the instruments and cable and by hydrodynamic drag forces. However, it is impracticable to obtain sufficient compliance in a spring accumulator to provide adequate decoupling because of the large static extension which would result. For example, to reduce the effect of a 7·5 second period, 0·2g peak sinusoidal heave by 90% the compliance of a linear spring accumulator would need to be such that the static extension was 414ft (126m). Since the static tension in the wire at the sea surface may vary from about 100lb to more than 1800lb according to the length of wire out, a simple linear spring is impracticable. The type of non-linear spring characteristic required is shown in figure 1. A more practical alternative is to use an active device with servo control.

Since a new electric winch was being designed for R.R.S. Discovery, the author was asked to explore the possibility of controlling this to provide the necessary stabilisation of the sensors. In the proposed system the ship's heave is measured, as near as possible to the final sheave, with a gyro-stabilised vertical accelerometer whose output is integrated to give a heave velocity signal. Small corrections for the roll and pitch can be added, if required, to allow for the separation of the accelerometer from the final sheave. The velocity signal is used to control the wire velocity so as to cancel the effect of the ship's heave. The motor control system includes a wire velocity feedback loop. In actual practice, it will usually be required that the cable should have a constant mean vertical velocity, in which case an appropriate constant control signal is added to the integrated accelerometer output signal. The cable tension will, ideally, not fluctuate, since the cable and instruments ideally move with constant, or zero, vertical velocity relative to the Earth.

After consideration of simplicity, cost and speed of response, thyristor control of a d.c. motor was decided upon, with the capacity for regenerating into the supply so as to allow rapid reversing of the motor. The mean cable tension is balanced by some mean armature current through the motor and, because of the considerable rotational inertia of the system, the motor has to provide accelerative torque at both crests <u>and</u> troughs of the ship's motion. Ideally, if the motor system has sufficient torque and speed ranges and a sufficiently rapid response, a high degree of compensation can be achieved.

The cable velocity signal, which is fed back in the servo system, is obtained from a d.c. permanent magnet tachogenerator which is driven by a

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sheave near the motor - thus, any movement of any mechanical accumulator included between the winch and the final sheave is not sensed by the tacho: this was thought desirable in view of the possibility of instability occurring if the accumulator, with its associated transient response, were included in the servo loop. It should have little effect on the stabilisation because of the near-constant tension in the wire. For small variations in the amount of cable payed out, the cable speed is linearly related to the motor speed, although the ratio (cable speed/motor speed) will vary appreciably, in small steps, according to the number of layers of cable on the spool. In the worst case, this step variation will be 4% when nearly 6000m of 1/4" armoured cable are payed out. The layout of the winch is shown in the appended drawing. (fig. 2). The drive is connected to either spool by the dog clutches on the worm shafts. Helical 1:1 gears are used to transmit the motor output to each clutch with minimum noise. Each spool holds approximately the same number of turns per layer.

When the change from one layer to the next occurs, the motor speed will have to change, at the most, by about 4% over the course of about one turn of the spool, in order to maintain constant cable velocity. This requirement does not, however, result in very high accelerations. Even for the high spool speed of 160rpm, the angular acceleration need only be $\cdot 036 \mathrm{rad/sec}^2$ for 4% speed change over one revolution. This is very small in comparison with the accelerations needed to compensate for the ship's motion.

Although velocity control, as opposed to displacement or acceleration control, has this advantage that the motor speed may be directly controlled, there are disadvantages when the mean velocity is required to be zero, i.e. when the winch is to be used to maintain apparatus at a constant depth. Imperfections in the system will result in a slow creep which will be below the threshold of the velocity correcting system. This creep may result from inching of the winch drive and from any non-linearity of the control system. The result is that a limit is put upon the maximum duration of observations at a particular depth, unless the winch operator is prepared to perform manual corrections from time to time. Displacement control would, however, introduce difficulties in the servo system and it was felt that velocity control was likely to prove to be the most satisfactory system.

Other possible sources of error, not peculiar to velocity control, are:

a) saturation at the limits of the drive motor performance
b) cross over effects at reversals of cable velocity due to static frictiom and to the dead zone of the motor-control system
c) operation of any mechanical accumulators
d) limitations in speed of response, due to mechanical inertia and electrical drive circuitry, resulting in amplitude and phase errors of the cable velocity relative to the ship
e) transient effects due to power supply disturbances, etc.

In order to be able to specify the motor ratings required, it was necessary to carry out a series of recordings of the heave motion of R.R.S Discovery, using a sensor situated as near as possible to the proposed location of the sheave. The ship's pitch and roll were also recorded, in case it was found necessary to make corrections for the disparity in the motions at the sensor and at the proposed sheave location. It was then possible, using these recordings together with estimated values of the motor inertia and other parameters, to investigate the result of varying the overall mechanical gearing ratio upon the requirements of motor torque and speed. Because of limitations on price, space and generating capacity, it was necessary to keep the motor as small as possible. Indeed, as the motor size increases, the winch performance may not improve pro rata, owing to the increase in motor inertia. Because of this, it was found to be preferable to impose transient overloads upon a small motor, rather than to use a larger motor, since the r.m.s. torque requirement (proportional to the heat dissipated in the motor armature windings) is considerably less than the peak torque requirement.

2. Ship Motion Recordings

A number of half hour records were made of vertical acceleration, vertical velocity, roll and pitch angles and shipborne wave recorder output, on R.R.S. Discovery's Cruise 22 in June 1968. Both analogue recordings (ultra-violet and pen recorder) and digitised recordings (0·5 second cycle on data logger) were made. The sea conditions were various, with winds between force 4 and 7 in the neighbourhood of the St. Kilda group of islands and the ship was either hove to or lying to. The sensor unit was an N.I.O. heave, pitch and roll buoy gyro unit, which was situated in one of the forward protected corners of the boat deck. The heave velocity signal was obtained by integrating the heave accelerometer output with an operational amplifier integrator having a time constant of 9 seconds.

Apart from trouble with the data logger, the recordings were routine; each digitised record consisted of 3000-3600 samples per channel (25 to 30 minute duration). The apparatus had been calibrated with the ship berthed at Aberdeen. The accelerometer was calibrated statically and dynamically. The static calibration (tilting through $\pm 36^\circ 50'$ in N-S/E-W planes, giving -0·2g change in acceleration) did not give consistent results. The dynamic calibration (swinging arm of 18" radius and 12 second period, giving a 0·0128g sinusoid) was therefore, accepted. The velocity channel was calibrated by the same method (0·785ft/sec sinusoid). The shipborne wave recorder was calibrated by the normal procedure. The roll and pitch channels were calibrated statically, using the engraved markings.

3. Analysis of the recordings

The wave recordings were analysed by the standard technique (M.J. Tucker, 1963. Proc. Inst. Civ. Engrs. 26, pp. 305-316) and the results are given in Table 1 below.

Table 1 - Wave Data for Recordings. Serial no: 22/01-22/08
(22/06 scrapped because of D/L fault)

Parameter \ Record No.	22/01	22/02	22/03	22/04	22/05	22/07	22/08
Approx. wind Speed (Knts)	17	17	27	27	33	32	25
True Wind Direction	-	-	190°	195°	265°	200°	200°
Ship State	Hove to	Lying to	Hove to	Lying to	Hove to	Hove to	Lying to
Estimated Wind Force	4	4	$5\frac{1}{2}$	$5\frac{1}{2}$	7	$6\frac{1}{2}$	6
T_z sec	8·12	7·83	5·64	5·88	6·72	7·59	8·37
T_c sec	7·00	6·43	5·15	4·97	5·84	6·14	6·34
S.B.W.R. Correction factor	1·27	1·30	1·90	1·77	1·51	1·33	1·24
H_1 feet	8·0	8·3	7·4	7·2	14·7	15·4	12·5
H_2 feet	7·5	7·8	6·4	6·9	14·2	14·9	11·8
H_s feet	5·9	6·3	8·0	7·3	12·6	11·9	9·0
ϵ	0·51	0·57	0·41	0·53	0·49	0·59	0·65

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Note on Table 1. S.B.W.R. Correction factor was calculated for depth of pressure transducers on this cruise (8·4 ft., KD = 21 ft.) and for periods T_z. From the values of T_z and ϵ, it appears that 22/01, 22/02, 22/07 and 22/08 had dominant swell, with rather more high frequency components in the latter two records. The records 22/03, 22/04 and 22/05 seem to represent a sea in the course of generation, with a fairly narrow range of short periods (or a fully arisen sea from which the long period waves have dispersed). These surmises are generally in agreement with the visual observations made at the time of the recordings.

The acceleration, velocity, roll and pitch records were analysed in several ways. The roll and pitch rates were found to be so small under these condition (less than 0·03rad/sec and 0·1rad/sec respectively) that they were of little importance in view of the close proximity of the gyro unit and the proposed sheave location. Two factors of great importance in the specification of the motor are the maximum torque which it will have to produce and the maximum speed at which it will have to run. The maximum torque is related to the cable tension and to the maximum downwards acceleration of the ship (this being the worse case for the motor, with the ship at a crest and a transition from paying out the cable to hauling it in). The instantaneous torque, T, is given in terms of the ship's acceleration upwards, $\ddot{z}$, by

$$T = \frac{W r_{spool} - J\ddot{z}/r_{spool}}{\eta_{mech} \quad R} \qquad (1)$$

where W is the cable tension, r_{spool} is the radius of the outermost layer of cable on the spool, J is the total effective rotational inertia referred to the spool, and η_{mech}, R are, respectively, the overall mechanical efficiency and the step down ratio of the gearing between motor and spool. J may be separated into three basic components, these being (a) the inertia of the spool and its worm wheel (estimated to be 32 slug feet^2 for the larger spool and 31·4 slug feet^2 for the smaller spool, (b) the effective inertia of the motor and worm shaft assembly, referred to the spool (estimated to be $8{\cdot}7\eta R^2/32{\cdot}2$ slug feet^2 for the motor frame size and mechanical components envisaged) and (c) the inertia of the cable on the spool, which will vary: see Tables II, IV and Appendices A and C.

The motor speed N r.p.m., is simply given by

$$N = \frac{(\bar{v} - \dot{z})\,R}{2\pi r_{spool}} \times 60 \qquad \text{r.p.m.} \qquad (2)$$

where $\bar{v}$ is the mean hauling velocity
and $\dot{z}$ is the ship's velocity upwards.

The greatest value of $-\ddot{z}$ was found to be 5·84 ft/sec^2 in force 7 conditions and 4.16 ft/sec^2 in force $5\frac{1}{2}$. The investigation was largely concentrated on these two conditions as it was thought that force 7 represents the worst conditions in which one would use the winch and that one might pay out up to 2000m of cable under such conditions. Force $5\frac{1}{2}$ represents the worst conditions under which one might may out up to 4000m cable. Graphs were drawn showing the probability of exceeding a given peak acceleration, for all the records, and, by fitting normal distributions to these curves, it was possible to estimate the peak acceleration likely over a given length of time, for given wind force. It should be pointed out, at this stage, that the records may not be truly representative and that the ship's response to waves will, in any case, vary quite appreciably according to her loading. Therefore, the analyses of the records, although carried out accurately, can only serve as a guide to the likely requirements. In any case, the very important parameter η_{mech} can not be estimated with very much certainty; also, the cable tension is affected by drag on the instruments which may vary widely on different occasions and with wire angle. For the purpose of the calculations, the load on the lower end of the cable was taken to be 150lbs, this being the sum of the weight of instruments in water and of any hydrodynamic drag force on them.

The predicted peak accelerations, for 8 hour and 4 hour periods, were $7{\cdot}5\text{ft/sec}^2$ and $7{\cdot}2\text{ft/sec}^2$ for force 7 conditions: $5{\cdot}7\text{ft/sec}^2$ and $5{\cdot}5\text{ft/sec}^2$ for force $5\frac{1}{2}$ conditions. These periods correspond approximately to the durations of stations with casts to 4000m and 2000m respectively. In view of the statements made above, the worst accelerations to be expected are $5{\cdot}7\text{ft/sec}^2$ with 4000m out and $7{\cdot}2\text{ft/sec}^2$ with 2000m out.

Similarly, the maximum peak velocities recorded were 16·8ft/sec in force 7 and 5·3ft/sec in force $5\frac{1}{2}$; from the graphs of probability of exceeding a given peak velocity, the greatest velocities to be expected during a station are 6·1ft/sec with 4000m out and 18·5ft/sec with 2000m out.

Harmonic analysis of the acceleration and velocity records for record 22/05 (fig. 5), showed that the dominant period of the ship's motion was 7·2 seconds and that there was very little motion with periods greater than 10 seconds or less than 4 seconds. Since the spectrum is so narrow, one can approximate the heave velocity by a sinusoid slowly modulated in amplitude, such that the velocity and acceleration are in quadrature and a single cycle can be approximated by the expression $\dot{z} = \hat{v} \sin \omega t$. Therefore the instantaneous motor power for 100% compensation can be estimated. The motor speed, N, is given by equation (2) where $\dot{z} = \hat{v} \sin \omega t$, with t = 0 at a trough of the ship's motion. The motor torque, T, is given by equation (1) where $\ddot{z} = \omega \hat{v} \cos \omega t$. The instantaneous horse power, P, is then given by

$$P = \frac{2\pi\, TN}{550} \qquad (3)$$

If $\hat{v} \gg \bar{v}$ and $J\, \omega \hat{v} < W\, r^2_{spool}$, the maximum instantaneous power occurs at a phase angle given, approximately, by

$$\omega t = \cos^{-1}\left[\frac{-W\, r^2_{spool} \pm \sqrt{(W\, r^2_{spool})^2 + 8(J\omega\hat{v})^2}}{4\, J\omega\hat{v}}\right] \qquad (4)$$

4. Selection of Gear ratio, R

The selection of the most suitable gear ratio is complicated considerably by the variations which will occur in the values of r_{spool}, J_c and W, according to the amount of cable payed out and on the spool. In the calculations, the length of cable between the winch and the final (outboard) sheave was assumed to be small and no allowance was made for friction at the various sheaves in the cable run. The figures for the overall efficiency of the gearing, η_{mech}, are possibly underestimates and were based on discussions with Clarke Chapmans, F. Pierce and R. Dobson. The moment of inertia of the cable J_c was calculated by the method given in Appendix A, both for ideal and imperfect spooling. Only the cases of the 1/4" armoured British Ropes cable and of 4mm wire rope were considered. The use of 6mm wire rope on the longer spool will result in rather better performance than with the 1/4" armoured cable, assuming that 6000m lengths are used in both cases. If the spool is used with its full complement of 8000m of 6mm wire rope, the performance will be poor with more than 2km of rope out and temporary overloading of the motor will be necessary to recover the rope if it is all payed out.

Table II - Spool data for 1/4" armoured cable

Length of Cable on spool m		0	1000	2000	3000	4000	5000	6000
Ideal spooling (a)	r_{spool} ft	·5104	·6727	·7991	·9358	·9976	1·106	1·196
	J_c slug ft^2	0	3·7	9·3	18·0	24·9	36·4	52·7
	Wr_{spool} lb ft	930	1039	1011	923	706	474	179
Imperfect spooling (b)	r_{spool} ft	·5104	·6979	·8437	·9687	1·094	1·177	1·281
	J_c slug ft^2	0	3·8	9·9	18·4	31·8	42·5	58·0
	Wr_{spool} lb ft	930	1078	1067	955	775	505	192
Length of cable overboard m		6000	5000	4000	3000	2000	1000	0

The motor under consideration had a rated full load torque of 90 ft lb and could be overloaded to a maximum torque of 135 ft lb. The performance available, using this maximum torque and for a maximum motor speed of 2500 r.p.m., is shown below for various values of R, both for ideal (a) and imperfect (b) spooling, for the worst case for acceleration (ship at crest). (See Table III overleaf).
The performance with 4mm wire rope on the smaller spool is given in Tables IV and V.

The acceleration figures given in Table III are satisfactory at 2000m out but fall short of the required 5·7ft/sec^2 at 4000m out. The minimum value of R (assuming η_{mech} = 0·75) to give 5·7ft/sec^2 with 4000m out is 17·1, which would result in a rather inadequate velocity performance with 2000m out (15·3ft/sec). Now, the mass inertia of the cable and instruments assists the motor when it cannot provide sufficient acceleration due to torque limitations. This can be seen as follows:-

(a) with the ship at a crest (motor required to accelerate cable upwards relative to ship), insufficient motor acceleration results in the cable accelerating downwards relative to the sea: consequently, the cable tension drops and the motor has a greater torque surplus for providing acceleration.

(b) with the ship at a trough (motor required to accelerate cable downwards relative to ship), insufficient motor acceleration results in upwards acceleration of the cable relative to the sea: consequently, the cable tension rises and the total accelerative torque on the spool increases.

(continued on page 9)

Table III   Available performance with 1/4" cable

(A) R = 14•5, η_{mech} = 0•77, J = 75•9 + J_c (slug ft^2)							
Length on spool m	0	1000	2000	3000	4000	5000	6000
(a) $\ddot{z}$ max ft/sec^2	3•86	3•96	4•65	5•82	7•70	10•2	12•4
(b) $\dot{z}$ max ft/sec	9•21	12•1	14•4	16•9	18•0	20•0	21•6
(b) $\ddot{z}$ max ft/sec^2	3•86	3•76	4•32	5•67	7•43	9•96	12•6
(b) $\dot{z}$ max ft/sec	9•21	12•6	15•2	17•5	19•7	21•2	23•1
(B) R = 15•5, η_{mech} = 0•76, J = 81•5 + J_c (slug ft^2)							
(a) $\ddot{z}$ max ft/sec^2	4•13	4•35	5•10	6•26	8•26	10•5	12•5
(a) $\dot{z}$ max ft/sec	8•62	11•4	13•5	15•8	16•8	18•7	20•2
(b) $\ddot{z}$ max ft/sec^2	4•13	4•18	4•83	6•16	7•85	10•3	12•8
(b) $\dot{z}$ max ft/sec	8•62	11•8	14•2	16•4	18•5	19•9	21•6
(C) R = 16•5, η_{mech} = 0•75, J = 87•4 + J_c (slug ft^2)							
(a) $\ddot{z}$ max ft/sec^2	4•32	4•67	5•45	6•64	8•56	10•7	12•7
(a) $\dot{z}$ max ft/sec	8•10	10•7	12•7	14•8	15•8	17•5	19•0
(b) $\ddot{z}$ max ft/sec^2	4•32	4•54	5•24	6•55	8•20	10•6	13•0
(b) $\dot{z}$ max ft/sec	8•10	11•1	13•4	15•4	17•4	18•7	20•3
Length overboard m	6000	5000	4000	3000	2000	1000	0

Table IV Spool data for 4mm wire rope

	Length on spool	m	0	1000	2000	3000	4000	5000	6000
(a) Ideal Spooling	r_{spool}	ft	·5066	·6088	·7111	·7907	·8702	·9384	1·007
	J_c	slug ft^2	0	1·4	3·3	5·8	8·8	12·4	16·6
	Wr_{spool}	lb ft	416	432	425	384	325	246	151
(b) Imperfect Spooling	r_{spool}	ft	·5066	·6247	·7297	·8347	·9134	·9928	1·058
	J_c	slug ft	0	1·4	3·5	6·1	9·5	13·4	18·0
	Wr_{spool}	lb ft	416	444	436	406	342	260	159
	Length overboard	m	6000	5000	4000	3000	2000	1000	0

Table V Available performance with 4mm wire rope $J = 54{\cdot}7 + J_c$, $R = 10{\cdot}5$

$\eta_{mech} = 0{\cdot}78$

Length on spool	m	0	1000	2000	3000	4000	5000	6000
(a) $\ddot{z}$ max	ft/sec^2	6·39	7·31	8·35	9·44	10·7	12·0	13·5
(a) $\dot{z}$ max	ft/sec	12·6	15·2	17·7	19·7	21·7	23·4	25·1
(b) $\ddot{z}$ max	ft/sec^2	6·39	7·36	8·40	9·61	10·9	12·3	13·8
(b) $\dot{z}$ max	ft/sec	12·6	15·6	18·2	20·8	22·8	24·7	26·4
Length overboard	m	6000	5000	4000	3000	2000	1000	0

(Continued from page 7)

However, neither inertia nor hydrodynamic drag forces can assist the motor very much when it reaches its speed limits (± 2500 r.p.m.) Any change in load tending to increase the cable speed from the limiting values set within the servo system will result in changes in the motor torque (within the limits of the motor torque range) to offset these load changes, since the servo system is a cable velocity control system. For this reason, it was thought preferable that any inadequacies of the motor should result in insufficient torque rather than in insufficient speed. Therefore a 15·5:1 overall gear ratio was adopted: this being the maximum ratio which should cope with most velocity peaks, whilst retaining a nearly adequate acceleration performance.

Another important reason for having a high velocity capability is to reduce the time spent when it is required to retrieve the cable rapidly for any reason. This situation, the fast hauling mode, will be discussed further below.

Apart from the peak torque and speed requirements, the r.m.s. torque and horsepower ratings are obviously of great importance in the design of the motor and of its control gear. In fact, the mean horsepower in the automatic mode is quite low by comparison with that resulting under fast hauling conditions, although the instantaneous horsepower may reach very high values.

The r.m.s. accelerations were calculated from the recordings and were $1{\cdot}32 ft/sec^2$ in force $5\frac{1}{2}$ and $2{\cdot}09 ft/sec^2$ in force 7. From these figures, it was possible to calculate the r.m.s. accelerative torques and hence the overall r.m.s. torques. For the force $5\frac{1}{2}$ case, with R = 15•5 and in the neighbourhood of 4000m of 1/4" cable out, the overall r.m.s. motor torque is between 87 and 91 ft lbs, depending upon the spooling. The corresponding figures for force 7, with about 2000m of 1/4" cable out, are 63 to 68 ft lbs. The r.m.s. speeds were also calculated and were 299 to 285 r.p.m. (force $5\frac{1}{2}$, 4000m out) and 539 to 491 r.p.m. (force 7, 2000m out). The resulting horsepowers are 5•0 and 6•4 H.P. for the two cases, whilst the maximum instantaneous horsepowers may be as much as 5 or 6 times these figures.

5. Fast hauling and paying

When the cable is payed out, the maximum cable velocity relative to the sea will normally be limited by hydronamic drag, since the axial drag on the cable attains the value of the (cable weight + 150 lbs load) at a cable speed which is always lower than the maximum winch velocity capability, for all the cables.

From Kullenberg's figures (ref: 2), we have

$$\text{Drag} = \cdot 0037\ (\pm\ 0{\cdot}0005 \text{ st'd dev'n})\ \phi\, v^2$$
$$\text{in Kg/metre length} \qquad (5)$$

where ϕ = cable diameter in mm.
v = cable velocity in m/sec.

This expression gives the following longitudinal drags for the 1/4", 6mm and 4mm cables, with v in ft/sec.

1/4",	Drag	=	$4{\cdot}81\ (\pm\ 0{\cdot}65)\ v^2$	lb/Km length
6mm,	Drag	=	$4{\cdot}55\ (\pm\ 0{\cdot}61)\ v^2$	"
4mm,	Drag	=	$3{\cdot}03\ (\pm\ 0{\cdot}41)\ v^2$	"

The range of velocity over which these figures will apply is not stated, but Kullenberg's experiment was carried out for v = 5•9 ft/sec. It will be assumed that they are applicable at up to 15ft/sec, this being the maximum hauling speed with any of the cables since it was decided, for safety, to limit the motor speed to 1400 r.p.m. when fast hauling. Equating these expressions for the drag to the cable weights in water, 274, 225 and 112 lbs/Km length, respectively, gives limiting pay out velocities of 7•55, 7•54 and 6•33 ft/sec, which correspond to motor speeds of (at the most) 2190, 2189 and 1253 r.p.m.

In addition to the hauling speed limit of 1400 r.p.m., which will be set automatically when the winch operator changes from the automatic to the fast handling mode, a red warning light will be used to indicate to the winch operator when the motor torque exceeds 90 ft lb and he should attempt to keep within this limit at all times. The third limit applying to fast handling conditions is the safe working load of the cable.

The cable tension will be presented to the operator in the normal way. The safe working loads used in the theoretical evaluation of the fast hauling performance were 2040 lb for the 1/4" armoured cable, 1800 lb for 6mm wire rope and 900 lb for 4mm wire rope; these all represent safety factors of about 2½.

On the basis of these limits, graphs were drawn showing the variations in the maximum safe hauling speeds against the length of cable out. The hauling times were estimated from these graphs, for the case of ideal spooling, and are tabulated below. The figures for 6000m of 6mm wire rope should be similar to those for the 1/4" armoured cable.

Table VI Fast hauling times, speeds and loads.

(A) 6000m of 1/4" cable on longer spool, normal limits.

Depth (m)	Hauling time to surface (mins)	Cable velocity ft/sec	Motor speed (r.p.m.)	Motor torque % of FLT	Motor Horsepower
6000	107	2•92	847	98•2	14•3
5000	78	1•53	337	100	5•8
4000	45	2•04	378	100	6•5
3000	24	3•34	528	100	9•1
2000	11½	6•16	914	100	15•7
1000	5	10•46	1400	99•2	23•8
500	2	10•97	1400	63•0	15•1
0	0	11•31	1400	16•9	4•1
(B) 6000m of 1/4" cable on longer spool, 110% torque limit					
6000	72	2•92	847	98•2	14•3
5000	54	2•98	656	110	12•4
4000	36	3•33	617	110	11•6
3000	21	4•36	689	110	13•0
2000	11½	7•00	1039	110	19•6
1000	5	10•46	1400	99•2	23•8
500	2	10•97	1400	63•0	15•1
0	0	11•31	1400	16•9	4•1
(C) 4000m of 1/4" cable on longer spool, normal limits.					
4000	29	4•83	1400	81•6	19•6
3000	20½	6•36	1400	98•7	23•7
2000	12½	7•56	1400	94•1	22•6
1000	6	8•85	1400	70•7	17•0
500	3	9•27	1400	45•7	11•0
0	0	9•44	1400	14•1	3•4

Table VI (continued)
(D) 6000m of 4mm wire rope on shorter spools, normal limits

Depth (m)	Hauling time to surface (mins)	Cable Velocity ft/sec	Motor Speed (r.p.m.)	Motor torque % of FLT	Motor Horsepower
6000	59	2·07	410	61·9	4·3
5000	39	3·54	583	74·3	7·4
4000	25½	4·99	704	86·9	10·5
3000	16½	6·75	856	96·6	14·2
2000	9	8·83	1017	100	17·4
1000	4	13·09	1400	99·5	23·9
500	2	13·67	1400	69·2	16·6
0	0	14·05	1400	20·5	4·9

Because the hauling time for 6000m of 1/4" cable was rather excessive, the figures for a 110% torque limit are also given. The result is a reduction by 35 minutes, but it would be wise to allow this overloading only in calm conditions.

Graphs were also plotted to show the variation in torque and output power with time and approximate values of r.m.s. torque and mean horsepower over the total hauling periods were obtained graphically. It will be seen that the r.m.s. torques are quite low excepting the cases for 6000m of 1/4" cable, but that the mean horsepower is quite high in the case of 4000m of 1/4" cable.

Table VII R.M.S. Torque, Mean H.P. and work done

Case	R.M.S. torque % FLT	Mean Horsepower	Total work done $\times 10^6$ ft lb	Work done against gravity $\times 10^6$ ft lb	Work done against drag $\times 10^6$ ft lb
A	97·6	8·9	31·4	19·1	12·3
B	104·5	13·9	33·0	19·1	13·9
C	80·8	19·4	18·6	9·2	9·4
D	79·0	10·8	21·0	9·5	11·5

It is interesting to note that, in the cases of the 4km length of 1/4" cable and of the 6mm cable, more than half of the work done is dissipated in the form of drag.

The above figures were all calculated for the case of zero heave motion of the ship. Now, in the presence of heave, but with the heave compensating input disconnected the motor will attempt to maintain the cable speed set by the operator and, since the mean torque is held to 100% f.l.t. or less by the operator, there should be a torque surplus of about 45 ft lb available at the motor to balance any changes in cable tension due to acceleration of the cable and instruments or due to extra drag on the cable and instruments. The safety factor in the cable loading will of course vary and it is up to the winch operator to reduce speed of hauling if the minimum safety factor becomes too small.

For example, with 6km of 1/4" cable out and a mean hauling rate of 2•92 ft/sec (see Table VI A), the maximum vertical acceleration of the ship upwards without torque limiting is 16•2 ft/sec^2: the maximum vertical velocity of the ship upwards without torque limiting, due to increased drag, is 3•85 ft/sec. These limitations are summarised in Tables VIII A to D for the mean hauling rates given in Tables VI A to D. The weight of the instruments in air is taken to be 200 lbs.

Table VIII (A) 6000m of 1/4", mean hauling velocities as in Table VI A.

Depth m	Cable safety factor at 150% f.l.t.	Maximum ship acceleration upwards ft/sec^2	Maximum cable velocity though water ft/sec	Maximum ship velocity upwards ft/sec	Maximum instantaneous horsepower
6000	1•61	15•8	6•77	3•85	21•8
5000	2•12	13•6	5•92	4•39	8•7
4000	2•52	14•0	6•21	4•17	9•7
3000	2•95	15•2	7•10	3•76	13•6
2000	3•15	19•8	9•66	3•50	23•5
1000	3•49	29•5	14•52	4•06	36•0
500	3•66	69•9	21•23	10•26	36•0
(B) 6000m of 1/4", mean hauling velocities as in VI B.					
6000	1•61	15•8	6•77	3•85	21•8
5000	2•12	13•6	5•92	2•94	16•9
4000	2•52	14•0	6•21	2•88	15•8
3000	2•95	15•2	7•10	2•74	17•7
2000	3•15	19•8	9•66	2•66	26•7
1000	3•49	29•5	14•52	4•06	36•0
500	3•66	69•9	21•23	10•26	36•0
(C) 4000m of 1/4", mean hauling velocities as in VI C.					
4000	1•61	29•9	9•86	5•03	36•0
3000	2•12	21•7	9•82	3•46	36•0
2000	2•52	27•6	11•58	4•02	36•0
1000	2•95	54•3	16•28	7•43	36•0
500	3•09	99•2	23•57	14•30	36•0

A/

Table VIII (D) 6000m of 4 mm, mean hauling velocities as in VI D

Depth m	Cable safety factor at 150% f.l.t.	Maximum ship acceleration upwards ft/sec^2	Maximum cable velocity through water ft/sec	Maximum ship velocity upwards ft/sec	Maximum instant-aneous horse-power
6000	1·03	44·4	8·65	6·58	10·5
5000	1·23	32·5	8·54	5·00	15·0
4000	1·44	30·7	8·88	3·89	18·1
3000	1·60	28·4	10·02	3·27	22·0
2000	1·76	30·8	12·16	3·33	26·2
1000	1·90	39·7	17·38	4·29	36·0
500	2·11	70·5	23·74	10·07	36·0

These figures show that the increased drag due to ship motion upwards will be the limiting factor and that the motor torque will limit quite frequently unless the mean hauling velocity is reduced quite appreciably. The mechanical accumulator will, of course, operate when the cable tension varies and will, to some extent, reduce the severity of this effect. Table VIII D shows how small the safety factor for the wire rope can become unless the winch operator reduces speed.

6. Summary of winch motor performance

The motor size and gearing have been selected to provide adequate torque and speed performance to deal with up to 4000m of 1/4" British Ropes cable out, with 150 lb load (in water) including drag on load, in force 5 - 6 conditions with a hauling rate of 1 ft/sec. In force 7 conditions, the performance should be adequate with up to 2000m of this cable out. However, a number of assumptions had to be made regarding mechanical efficiencies and, since the performance is critically dependent on these efficiencies, the actual performance may differ considerably from that estimated above. Also, the ship motion may vary considerably for given wind conditions, so that it was not thought worth while to make any more detailed analyses than those given above. Since the estimates of efficiences are rather conservative, the actual performance may be superior to that calculated above.

7. The Performance of the Servo System

The requirements of the system were that it should compensate for the ship's motion to such an extent that any residual vertical motions of the submerged instruments relative to their desired position(s), whether stationary or moving with constant velocity, should not exceed 3 feet peak-to-peak amplitude, in conditions up to force 7. This implies a system accuracy of about 90%. A simplified diagram of the control system is shown in figure 9. The principal "built-in" time constants of the system result from the inertias of the motor and of the spools, but the armature circuit of the motor is also of importance. It will be shown that, owing to variation in the inertia of the cable on the spool, J_c, and in the effective radius of the spooled cable, r_{spool}, the transient response of the system will vary in accordance with the lengths of cable payed out and on the spool.

The servo system consists essentially of the motor package, which uses both tachogenerator and IR feedback to obtain good performance over wide torque and speed ranges, and a cable velocity feedback loop with inputs for mean velocity v and for the compensating ship velocity v_s. The motor speed feedback and the cable velocity feedback are, in a sense, at odds with each other, since the ratio of cable velocity/motor speed varies. One might question the necessity for having motor speed feedback at all but, in fact, its presence is quite convenient from the point of view of the system as a whole and it safeguards the motor to some extent.

Motor control loops

The IR compensating loop improves the speed regulation, with varying load torque, whilst the tacho feedback improves both regulation and the accuracy of the output speed. The use of IR compensation allows a lower tacho loop gain and this may be desirable for stability reasons. IR compensation by itself can only have a limited effectiveness as it relies on the constancy of the total flux per pole ϕ and of the brush voltage drop V_b. Due to armature reaction, the leading pole tips may become saturated at high armature currents thereby reducing ϕ and, consequently, reducing the generated back e.m.f. E. Similarly V_b displays a non-linear relation with armature current I.

The effect of this can be seen as follows (see figure 10), in the steady state case.

Armature circuit voltage $V = E + I(r_a + r) + V_b$ (6)

where $E = k\phi N_o$ and k is a motor constant, r_a is the armature resistance, r is the compensating resistance and N_o is the motor speed.

Now $V = k_c (V_i + IrA)$ (7)

where V_i is the input voltage and is proportional to the desired speed N_i (tacho loops open)

$$\text{i.e.} \quad V_i = A'k' N_i \tag{8}$$

From equations 6, 7 and 8, we have

$$N_o = \frac{A'k_c k'}{k\phi} N_i - \frac{I\left[r_a - r(k_c A - 1)\right] + V_b}{k\phi} \tag{9}$$

$$\text{and, since } k''\phi I = T_L + \frac{2\pi N_o d}{60} \tag{10}$$

$$N_o = \frac{k''\phi A'k_c k' N_i - T_L\left[r_a - r(k_c A - 1)\right] - k''\phi V_b}{kk''\phi^2 + \frac{1}{30}\pi d\left[r_a - r(k_c A - 1)\right]} \tag{11}$$

Clearly, the speed regulation can be improved by making $r_a - r(k_c A - 1) \rightarrow 0$, but the accuracy of the N_o/N_i ratio (approximately equal to $A'k_c k'/k\phi$) will vary because of the dependence of ϕ upon I at high values of I.

Example: For the motor to be used,

$$k\phi = 0{\cdot}180 \text{ volt/r.p.m.}$$
$$r_a = 0{\cdot}24\Omega \text{ (including compole resistance)}$$
$$k''\phi = 1{\cdot}27 \text{ ft lb/amp}$$
and
$$d \simeq {\cdot}07 \text{ ft lb sec.}$$

Without IR compensation, the speed would drop 93•7 r.p.m. when full load torque (90 ft lb) was applied. If the power dissipated in r on full load (I = 71 amps) were limited to 10 watts, the gain k_cA required for a no load - full load speed drop of 5 r.p.m. (for example) would be 114•3. Note that, if $(k_cA - 1)r > r_a$, instability could result.

When used in conjunction with tachogenerator feedback of the motor speed, the contribution of IR feedback becomes less important as the speed rises.

When the motor speed tacho loop in figure 10 is closed, in the steady state we have $(f(s=0) = 1)$, putting $r_a - r(k_cA - 1) = r'(= 12{\cdot}71\text{m}\Omega)$,

$$N_o = \frac{k''\phi A'k_ck' N_i - T_Lr' - k''\phi V_b}{k''\phi A'k_ck' + kk''\phi^2 + \frac{1}{30}\pi dr'} \qquad (12)$$

A practical value for k' (motor speed tacho constant) is 0•002: this gives a 5 volt output at maximum motor speed, which is compatible with the solid state amplifiers to be used. Therefore, if we put $k_c = 114{\cdot}3$ and A = 1, such that the IR compensating voltage is summed directly with V_i, the steady state accuracy is given approximately by

$$\frac{A'}{A' + 0{\cdot}786} \times 100\%$$

A d.c. gain of A' = 10 should be adequate. This further impoves the speed regulation by a factor of about 12•7 and gives a steady state accuracy of 92•9% for N_o/N_i.

When the wire speed tacho loop in figure 10 is also closed, in the steady state we have (g(s=0) = 1)

$$v_o = \frac{2\pi r_{sp}\left[k''\phi A'k_ck'Pk_tv_i - T_Lr' - k''\phi V_b\right]}{(60R + Pk_t2\pi r_{sp})\,k''\phi A'k_ck' + 60Rkk''\phi^2 + 2\pi dr'R} \qquad (13)$$

For the values of $k''\phi$, $k\phi$, k_c, etc: given above, the steady state accuracy v_o/v_i is given, approximately, by

$$\frac{v_o}{v_i} = \frac{18{\cdot}2\, r_{sp}\, Pk_t}{18{\cdot}2\, r_{sp}\, Pk_t + 188R}$$

For the worst case, R = 15•5 and $r_{sp} = 0{\cdot}5104$, and the steady state accuracy is

$$\frac{9{\cdot}30\, Pk_t}{9{\cdot}30\, Pk_t + 2914}$$

so that, for 95% accuracy, Pk_t must be 5950. Since k_t is, for practical reasons, limited to 0·2 volt sec/ft (5 volt output from wire speed tacho at 25 ft/sec wire velocity), P must be 29,750 r.p.m./volt and the amplifier gain, Pk', between the two summing junctions is 59·5, which is a reasonable figure.

The complete expression for v_o, as a function of v_i, T_L and V_b is

$$D\, v_o = H_v\, v_i - H_t\, T_L - H_b\, V_b \tag{14}$$

where

$$D = 2\pi R(sJ + d)(r' + sL) + 60Rk''\phi k_c A'k'f + 60Rkk''\phi^2 + 2\pi r_{sp}\, k''\phi k_c A'k'Pk_t g \tag{15}$$

$$H_v = 2\pi r_{sp} k''\phi k_c A'k'Pk_t \tag{16}$$

$$H_t = 2\pi r_{sp}\, (r' + sL) \tag{17}$$

$$H_b = 2\pi r_{sp}\, k''\phi \tag{18}$$

There are two poles in the response which are given by the roots of the denominator expression D. The positions of these poles, and hence the nature of the response, are dependent upon R, J and r_{sp}, which vary according to the amount and type of wire on the spool in use. The pole locations are also dependent upon d, which may vary considerably with gearbox oil temperature etc, apart from the other parameters which may vary to a lesser extent.

The parameter values used in the steady state calculations above result in the following expression for D. The value of L was taken to be 6·8mH: this comprises the total motor armature circuit inductance (6·6mH) plus estimated bus bar inductance (0·2mH)

$$D = \cdot 0427\, JR\, s^2 + (\cdot 00299R + \cdot 0799JR)s + (174 \cdot 4f + 13 \cdot 7)R + 108{,}500\, r_{sp} g$$

Putting f(s) = g(s) = 1 for the moment, this expression has roots which, for the longer spool, are at −1·014 ± j444 (spool empty) and at −0·983 ± j522 (6000m of 1/4" cable on spool). For the shorter spool, the root locations vary between −0·991 ± j447 and −0·978 ± j547. These figures represent a very lightly damped response in all cases. Additional damping can be introduced conveniently by making f(s) a function of the form

$$f(s) = 1 + \alpha s$$

Alternatively or additionally, one could make g a function of s, e.g. $g(s) = 1 + \beta s$.

Ideally, the system should be in a nearly critically damped state under all conditions. This state of affairs could be achieved by making α a function of the amount of wire on the spool, such that the equality $(\cdot 00299R + \cdot 0799JR + 174 \cdot 4\,\alpha R)^2 = 4 \times \cdot 0427JR(188 \cdot 1R + 108{,}500\, r_{sp})$ was maintained as J and r_{sp} varied. If anything, the system design should err on the side of underdamping as it is usually found that viscous damping is in excess of the expected figure. Also, when one is interested in speed of response to a step function, an underdamped system can reach and remain within a given tolerance band about the steady state value faster than a critically damped system. Similarly, the phase lag in response to sinusoidal excitation is, for an underdamped or critically damped response, given by

$\tan^{-1}\left[2\gamma\omega/(\gamma^2 + \delta^2 - \omega^2)\right]$, where $-\gamma \pm j\delta$ are the roots of the denominator, so that underdamping is beneficial, especially if γ is not very much greater than ω. The response to a sinusoid of frequency ω, normalised relative to the d.c. response, is given by

$$\left| \frac{v_o}{v_i} (j\omega) \right| = \frac{\gamma^2 + \delta^2}{\sqrt{(\gamma^2 + \delta^2 - \omega^2)^2 + 4\gamma^2 \omega^2}}$$

which, for the case of critical damping ($\delta = 0$), simplifies to $\gamma^2/(\gamma^2 + \omega^2)$. Thus for 95% response at the dominant ship motion period of 7·2 seconds, γ must be 3·8 radians/sec.

Now if α is adjusted to give critical damping for all conditions, γ varies between -444 and -547 radians/sec, so that the response at a period of 7·2 seconds is almost equal to the steady state accuracy.

The values of α required to give critical damping are shown in Table IX.

Table IX Variation in time constant α with depth

(A) 1/4" cable, R = 15·5				
Length of cable out (m)	6000	4000	2000	0
for critical damping (sec)	0·0969	·1268	·1529	·1876
(B) 4mm wire rope R = 10·5				
α for critical damping (sec)	0·1389	·1686	·1901	·2216

These time constants could be reduced to a more desirable order by using $g(s) = 1 + \beta s$ compensation, instead of $f(s) = 1 + \alpha s$ compensation. This has a further advantage that the required values of β do not vary as much.

Table X Variation in time constant β with depth

(A) 1/4" cable, R = 15·5				
Length of cable out (m)	6000	4000	2000	0
β for critical damping (sec)	·00473	·00395	·00382	·00391
(B) 4mm wire rope R = 10·5				
β for critical damping (sec)	·00463	·00400	·00369	·00371

Whilst it would be possible to monitor r_{sp} and to provide exact adjustment of α or β automatically, using suitable law potentiometers, the additional complexity and reduced reliability which would result are not desirable. However, examination of Table X shows that, for most purposes, near ideal damping could be achieved using fixed time constants of, say, β = 3·9mS and β = 3·8mS for the cases of 1/4" cable and 4mm wire rope, respectively. The roots of D then lie at $-366{\cdot}6 \pm j250{\cdot}9$ and $-367{\cdot}2 \pm j254{\cdot}6$ for the worst cases with 6000, of 1/4" cable and 4mm wire rope out respectively. This is not an excessive amount of underdamping, bearing in mind that the winch would only be used in fairly calm conditions with more than 4,000m of cable out.

In the absence of more detailed and accurate information regarding viscous damping coefficients etcetera, it is proposed to make initial adjustments on the basis of the above time constants.

It has already been mentioned that the ship motion correction signal spectrum is quite sharply peaked so that it should be possible to reduce errors due to the time lag of the servo system by providing a phase advance circuit in the v_s input. Of course, the servo system phase lag is a function of the amount of cable on the spool but, once again, near ideal correction can be achieved (particularly in the range 0 - 4000m out) by the use of fixed time constants of 3·9mS and 3·8mS respectively. These could be realised, in theory, by means of the imperfect heave integrator response if operation were limited to a single frequency but, since the integrator imperfections produce dispersion of opposite sense to that required, it would be preferable to use a high pass filter network after the integrator. The imperfect integrator, having transfer function $\frac{1}{s + a}$, gives an output $\frac{1}{\omega\sqrt{1 + (\frac{a}{\omega})^2}}$ $\sin(\omega t + \tan^{-1}(\frac{a}{\omega}))$ for an input of $\cos \omega t$. For 95% output at $\omega = 2\pi/7{\cdot}2$, 'a' must be ·287 rad/sec, resulting in a phase advance of ·318 radian at this frequency. This is about two orders of magnitude greater than the phase lag of the servo system at this frequency. The largest practical value of 'a' is about 0·01 which would result in a phase advance of ·0115 radians which is still in excess of the required phase advance. Furthermore, the phase advance is proportional to ω^{-1}, approximately, and not proportional to ω as desired. This means that the nett phase error would increase rapidly with frequency away from $\omega = 2\pi/7{\cdot}2$. Some form of compensating network h(s) is required to reduce the integrator phase lead to a suitable value. Such a network could be of the form shown in figure 11. The transfer function for this network is given by

$$h(s) = \frac{1 + bs^{-1}}{1 + (b/(N + 1))s^{-1}} \quad \text{where } b = \frac{1}{CR_3}$$

For N = 9,

$$h(s) \simeq 1 + 0{\cdot}9bs^{-1} - 0{\cdot}09b^2s^{-2} \quad \text{for } b << 1$$

and the phase lag is given by

$$\tan^{-1}\left[\frac{{\cdot}9\omega CR_3}{(\omega CR_3)^2 + {\cdot}1}\right]$$

Thus, if the integrator time constant is 10 seconds, the required value of CR_3, for $\frac{2\pi}{7{\cdot}2} \times 3{\cdot}6$ m rad nett phase advance, is $CR_3 = 9{\cdot}345$ seconds. The nett phase advance over the range $\omega = 2\pi/9$ to $2\pi/5$ rad/sec then varies from $\frac{2\pi}{9} \times 7{\cdot}9$ to $\frac{2\pi}{5} \times 2{\cdot}4$ m rad. These deviations from the ideal figure of 3·6 m rad are not important.

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The analysis of the servo system given above has, for reasons of brevity, used a number of simplifications which have not always been pointed out. For example, the possibility of saturation of the loop amplifiers and other components has not been considered. Nor have other non-linear effects such as back-lash and the dead zone of the motor/controller. In an ideal system, saturation would not occur but in practice the system may saturate if the input signal can not be followed. The ship motion study and the resulting specification of the motor/controller unit capabilities should ensure that saturation of this unit only occurs at infrequent intervals. It is, therefore, important to ensure that the rest of the servo system does not unnecessarily restrict the overall performance. Of course, if a step input is applied to the system, saturation of some of the components is inevitable. However, under steady state conditions of fast hauling (with the v_s input switched out) or with a given sinusoidal v_s input and slow hauling rate, none of the components should saturate. Since the latter case is the more severe, it will be examined more fully.

From the preliminary motor data given by Clarke Chapmans:

Maximum supply voltage	=	460 V
Full load current	=	71 A
$k\phi$	=	$\cdot 180$ V/r.p.m.; $k''\phi = 1{\cdot}27$ ft lb/A
$r_a + r$ (hot)	=	$\cdot 253\Omega$

∴ At 150% full load current, max r.p.m. is 2395 r.p.m. (assuming 1 volt drop at each brush)

At 2500 r.p.m., max current is 33·6 A (assuming 0·75 volt drop at each brush)

At 2500 r.p.m., max torque is 42·6 ft lb.

The motor/controller unit is therefore, capable of meeting the peak instantaneous speed and torque requirements.

Consider now the performance with a velocity input $v_i = \bar{v} + v_s \sin \omega t$ and assume that T_L is constant and that V_b can be neglected.

i.e. $D\, v_o = H_v\, v_i - H_t\, T_L$ from equation 14

with $v_i(s) = \dfrac{\bar{v}}{s} + v_s \cdot \dfrac{\omega}{s^2 + \omega^2}$

Also, let $f(s) = 1$ and $g(s) = 1 + {\cdot}0036s$ and let k_t, P, k', A', k_c, r' $k''\phi$, $k\phi$ and d take the values assumed above.

Then, for 4000 of 1/4" cable out ($R = 15{\cdot}5$, $J = 90{\cdot}8$ and $r_{sp} = {\cdot}7991$), we have

$$D = 0{\cdot}32913\,(s^2 + 1029{\cdot}4s + 272{,}287)$$
$$H_v = 86{,}732$$
$$H_t = {\cdot}0638\,(1 + {\cdot}535s)$$

It is clear that the effect of load torque is very small (about 10^{-4} ft/sec error at 150% f.l.t.) so that, for an approximate treatment, this may be neglected.

The inputs to the summing junction preceding P are, therefore,

$$k_t\, v_i(s) = {\cdot}2\left(\frac{\bar{v}}{s} + v_s \frac{\omega}{s^2 + \omega^2}\right)$$

and $k_t \, g(s) \, v_o(s) \quad = \quad \cdot 2(1 + \cdot 0039s) \dfrac{H_v \, v_i(s)}{D}$

$$\therefore \quad \frac{N_i(s)}{P} \quad = \quad \cdot 2 \left\{ \frac{s^2 + 2\cdot 012s + 8{,}858}{s^2 + 1029\cdot 4s + 272{,}287} \right\} v_i(s)$$

The steady state terms in $N_i(t)/P$ have the following magnitudes

d.c.	$0\cdot 00651 \, \bar{v}$
at $\omega = \frac{2\pi}{7\cdot 2}$;	$0\cdot 00651 \, v_s$

If $\bar{v}$ = 1 ft/sec and v_s = 6•1 ft/sec., it follows that the inputs to the summing junction do not instantaneously exceed 1•42 and that the output from the junstion does not exceed 46•2 millivolts. The $k'N_i$ input to the next junction does not exceed 2•750 volts. The $k'f \, N_o$ input is given by

$$k'f \, N_o(s) \quad = \quad \frac{k'f \; 60R}{2\pi r_{sp}} \qquad \frac{H_v \, v_i(s)}{D}$$

(neglecting T_L and V_b as before)

For $v_i(t)$ = 1 + 6•1 sin($2\pi t$/7•2), this input has a maximum value of 2•546 volts. Now the phase shift between N_o and N_i at the v_s frequency is 0•00023 radian, so that the input to amplifier A' does not exceed 0•204 volt. V_i does not, therefore, exceed 2•04 volts and, since rAI can not exceed 0•212 volts at 150% full load current, the input to the controller k_c will be well below the 4•02 volt limit to its linear range.

From the above step by step examination of the voltages along the top line in figure 10, it is clear that saturation will not occur at any point under these conditions of R, J, r_{sp}, $\bar{v}$, and v_s. Similar analyses for 2000m of 1/4" cable out and v_s = 15 ft/sec give

$k'N_i$	<	4•994 volts
$k'fN_o$	<	4•625 volts
V_i	<	3•69 volts

For 4000m of 4mm wire rope out, v_s = 6•1

$k'N_i$	<	2•109
$k'fN_o$	<	1•953 volts
V_i	<	1•56 volts

and for 2000m of 4mm wire rope out, v_s = 15

$k'N_i$	<	3•901
$k'fN_o$	<	3•613
V_i	<	2•88

These figures are all satisfactory. Furthermore it can be inferred that the system will not overload under fast hauling conditions since the velocities will be less than those given in the above examples.

Effects due to mechanical backlash will not be considered here. Mechanical backlash in the gear train will only occur in force 7 conditions when the pull on the cable is insufficient to provide reversing torque with the ship at a trough. The motor then has to provide some reversing torque and so both sides of the gear teeth will be loaded alternately. Considerable assumptions were made in deriving a figure for the viscous damping factor.* Static friction can be included in T_L, with the reservations that stiction may be important and that the static friction always opposes any motion (similarly V_b opposes current).

An important feature of the system which has been reduced to "black box" status for the purposes of the above analysis is the controller. Obviously dynamic control by means of SCRs with a 3-phase supply results in step variation of the voltage supplied to the motor because this voltage can only be varied at intervals of, on the average 6·67mS(for a 3-phase supply) or 3·33mS (for a 6-phase supply). Then, the average delay inherent in the controller will be one half of these periods i.e. 3·33mS (3-phase) or 1·67mS (6-phase). The controller constant k_c is, therefore, a very complicated function and, since the delay time involved should be small compared with the armature circuit time constant, it was, for the purpose of this investigation, neglected. The effect of a delay τ is to make k_c of the form

$$k_c(s) = 114{\cdot}3\ e^{-\tau s}$$

which, for small τs, can be approximated to

$$k_c(s) = 114{\cdot}3(2 - s\tau)/s + s\tau)$$

Examination of the expression for D (equation 15) shows that the delay in k_c could result in instability of the system. This could be corrected by appropriate adjustment of β, or, preferably, by inserting a phase advance network in series with the controller. However, since the exact performance of the controller is not known at present, this feature of the system can not be profitably investigated at this stage. Further details will be included in a later report.

The mechanical design of the winch has not been discussed in this report; the preliminary design was done by R. Dobson of N.I.O. From the servo system point of view, it is obviously important to keep the inertias of the moving parts low. The quality of the gearing and of the lubrication are also important, in view of the continual reversing that will occur in the automatic mode. The spooling gear will be an important factor in determining the cable inertia. The nature of the safety protection devices is also of great importance because of the remoteness of the control points from the winch unit.

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* However, when β is suitably adjusted for critical damping, the viscous damping terms are made completely insignificant.

9. Conclusions

From measurements of the heave motion of R.R.S. Discovery it has been possible to specify the peak torque and speed requirements for an electric drive to provide acceptable compensation for this motion. The fast hauling performance which should result has been calculated and found to be acceptable. Analysis of the servo system showed that stable operation and accuracy can be achieved by the incorporation of two phase compensating networks into the system. The accuracy of compensation of ship motion should be approximately 95% under linear operating conditions. When working at the limits of the specified conditions (2000m of cable out in Force 7, or 4000m out in Force $5\frac{1}{2}$), occasional saturation of the system will occur and the mechanical accumulator will operate. The errors resulting from such saturation will normally be small and it would not be economical to increase the motor ratings to reduce the probability of their occurring. For this preliminary study, several simplifications and assumptions have been made and are pointed out in the text. It is hoped to publish a more complete report when the system has been tested in practice, as opposed to in theory.

10. Acknowledgements

The author wishes to thank members of the staffs of N.I.O. and Clarke Chapmans for their assistance and Mr. John Cant of the National Institute for Research in Dairying for his important contribution while working at N.I.O. during July to September 1968 on this project for his M.Sc.

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2. KULLENBERG, B., 1951. On the shape and the length of the cable during a deep-sea trawling. Reports of the Swedish Deep-Sea Expedition. Vol.II Zoology. No. 2, pp. 31-44.

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APPENDIX A

Estimation of inertias of the winch spools, etc.

Estimated moment of inertia of the motor	=	$7{\cdot}5$ lb ft^2
Estimated moment of inertia of the worm shaft, helical gears, brake drum, etc	=	$1{\cdot}2$ lb ft^2
Estimated moment of inertia of the winch spool assy (long)	=	$4{\cdot}2$ lb ft^2
(short)	=	$9{\cdot}0$ lb ft^2

These figures are all referred to the motor shaft.

The inertias of the cable or wire rope on the spools were estimated as follows.

(a) Ideal spooling

Ideal spooling is defined to be that in which the cross sectional packing density is greatest (figure A(a)). For 1/4" cable, there is a maximum of 95 turns/layer - for 4mm rope, there is a maximum of 94turns/layer.

The length of a turn in the Nth layer is

(1/4" cable) $0{\cdot}97751 + (N-1) \times 0{\cdot}034553$ metre

(4mm wire rope) $0{\cdot}97013 + (N-1) \times 0{\cdot}021766$ metre

The inertia was calculated as if the spooled cable was a homogeneous solid of density $(\frac{\pi}{4} + 0{\cdot}126)$ times the mean cable density - except for the innermost and outermost layers, which have a packing density of $(\frac{\pi}{4} + {\cdot}063)$. The mean (1/4") cable density was taken to be 30·36 lb/100 yds: this is the British Ropes' figure - a sample of cable was weighed and gave the result $29{\cdot}8 \pm 0{\cdot}2$ lb/100 yds as a check. Mean 4mm rope density taken to be 11·10 lb/100 yds.

For a given length of cable on the spool (e.g. 3000m of 1/4") the number of layers (23·200) and hence the moment of inertia (580·4 lb ft^2) were calculated. The results are tabulated in Tables II and IV.

(b) Imperfect spooling

In the case of "imperfect" spooling, the cable was assumed to be spooled in a 90° cross sectional matrix (fig. A(b), instead of the 60° matrix assumed for ideal spooling.

In this case, the length of a turn in the Nth layer is given by

(1/4" cable) $0{\cdot}97751 + (N - 1) \times 0{\cdot}039898$ metre

(4mm wire rope $0{\cdot}97013 + (N - 1) \times 0{\cdot}025133$ metre

The moments of inertia in this case are simply calculated using a packing density of $\frac{\pi}{4}$.

APPENDIX B

Preliminary Motor Design Figures

Compole inductance	0·188	mH
" leakage inductance	1·2	mH
" resistance	0·04	Ω
Armature inductance	4·33	mH
" slot leakage inductance	0·47	mH
" end turn leakage inductance	0·44	mH
" resistance	0·16 0·20	Ω cold Ω hot
Total flux density/pole	1·25	M lines /$\square$"
Full load current	71	amps } (150% of these allowed)
" " torque	90	ft lb }
Maximum speed	2500	r.p.m.
Friction and Windage	220	W at 1400 r.p.m.
I^2R loss at full load	1210	W
Inertia of armature ass^y	7·5	lb ft^2

$$\therefore \text{ Back e.m.f. constant } k\phi = \frac{\text{torque constant}}{0\cdot 1173 \times 60} = 0\cdot 180 \text{ V/r.p.m.}$$

Maximum supply voltage = 460 V.

APPENDIX C

Wire data

1/4" diameter British Ropes Limited Armoured single insulated conductor cable

Weight in air	332 lbs/km length
Weight in sea water (S.G. 1·03)	274 lbs/km length
Breaking strain	2 tons

6mm diameter Wire rope

Weight in air	259 lbs/km length
Weight in sea water	225 lbs/km length
Breaking strain	2 tons

4mm diameter Wire rope

Weight in air	121·4 lbs/km length
Weight in sea water	112 lbs/km length
Breaking strain	1 ton

Load on wire taken to be 150 lb in sea water

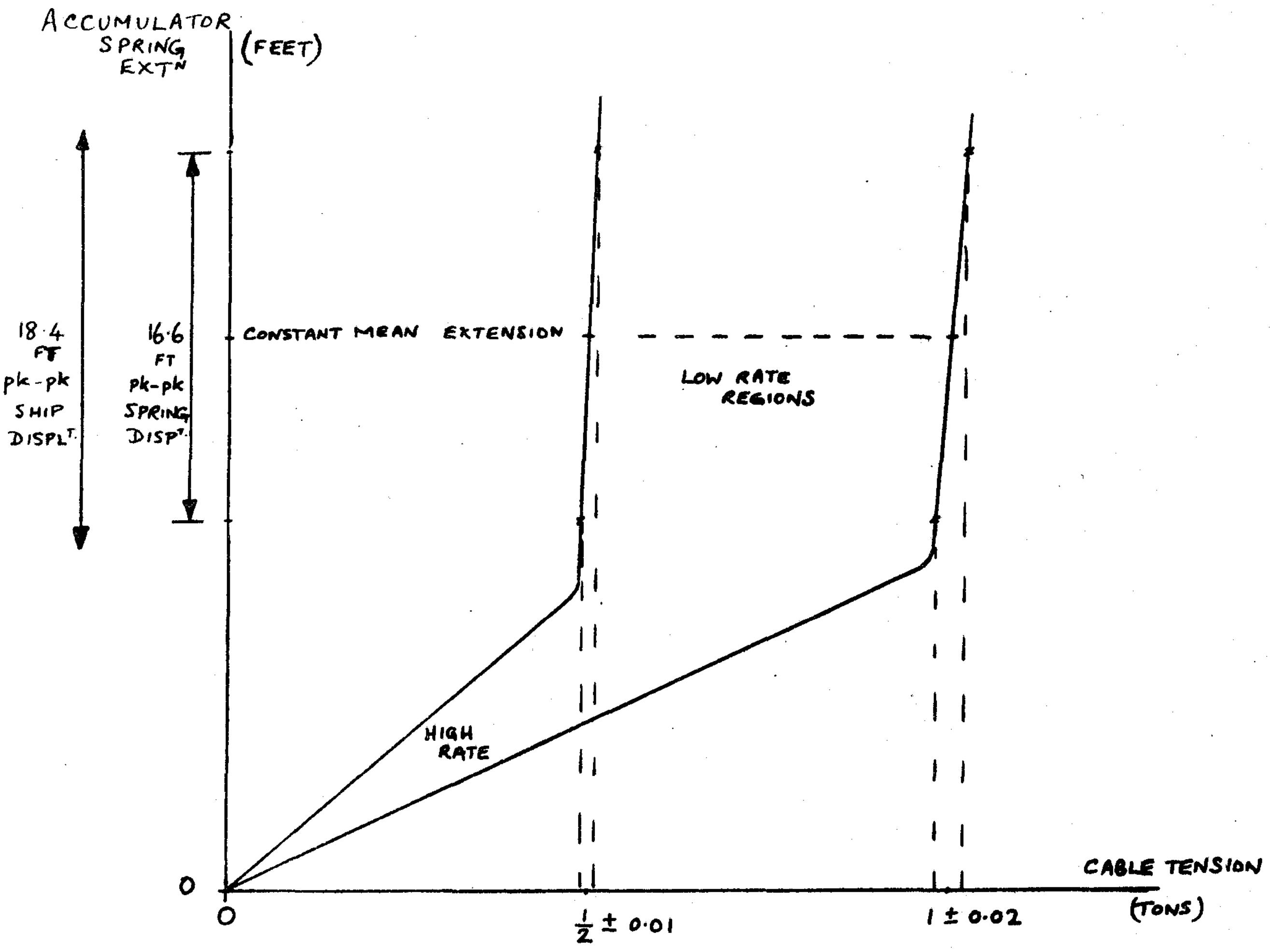
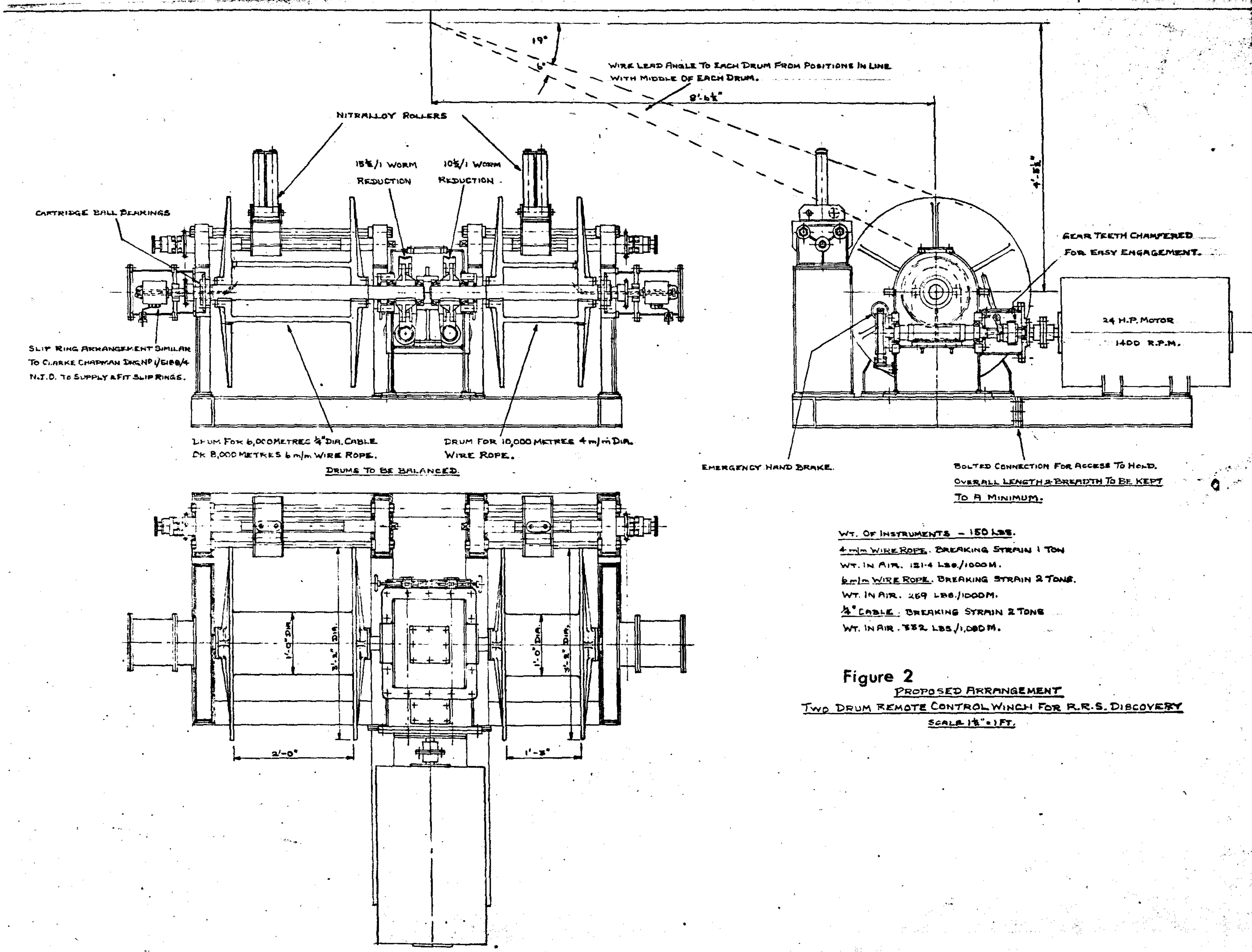


Figure 1. Required accumulator spring characteristics.



Wt. Of Instruments – 150 Lbs.
4 m/m Wire Rope. Breaking Strain 1 Ton
Wt. In Air. 121·4 Lbs./1000 M.
6 m/m Wire Rope. Breaking Strain 2 Tons.
Wt. In Air. 269 Lbs./1000 M.
¼" Cable. Breaking Strain 2 Tons
Wt. In Air. 332 Lbs./1,000 M.

Figure 2
Proposed Arrangement
Two Drum Remote Control Winch For R.R.S. Discovery
Scale 1⅛" = 1 Ft.

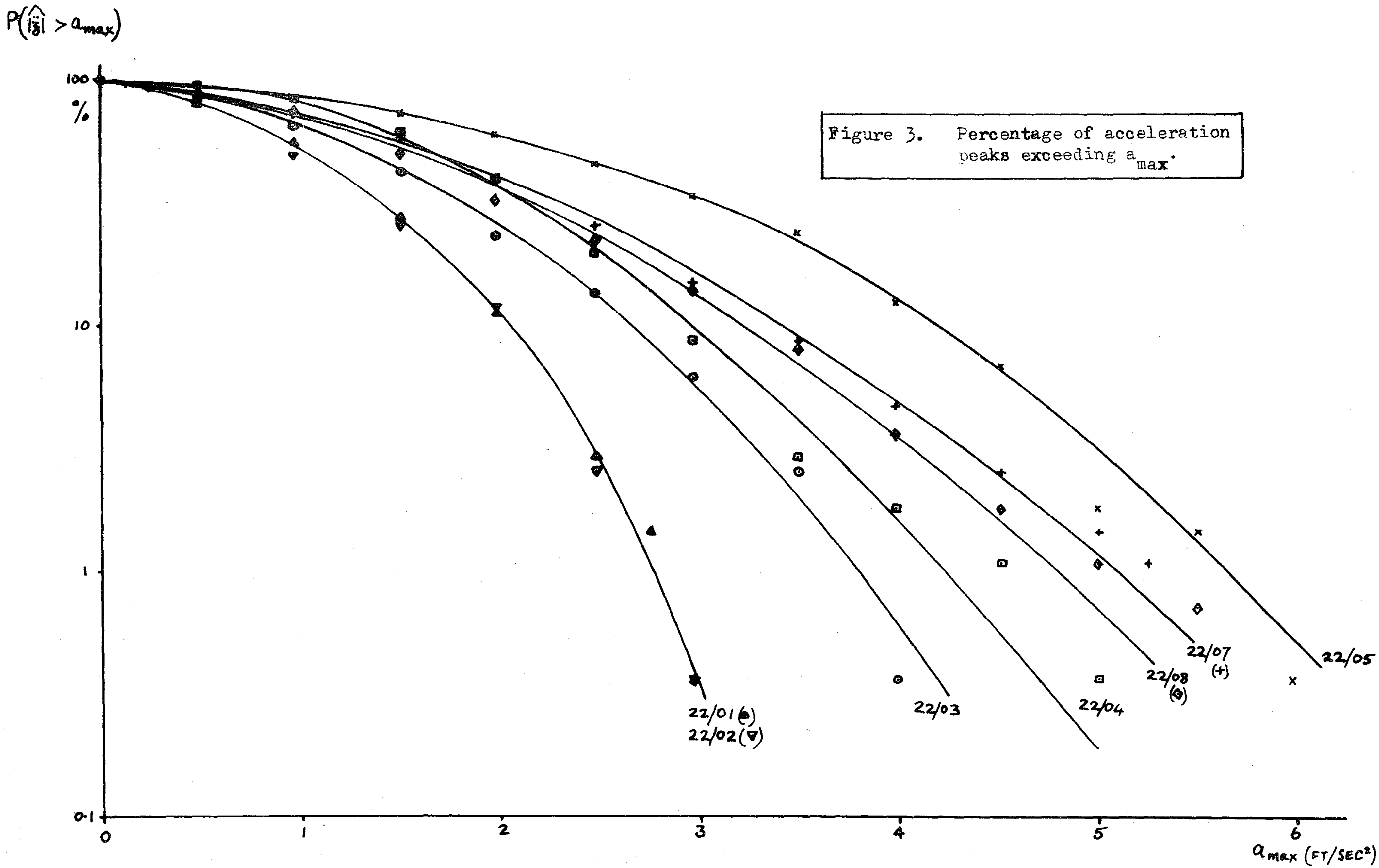


Figure 3. Percentage of acceleration peaks exceeding a_{max}.

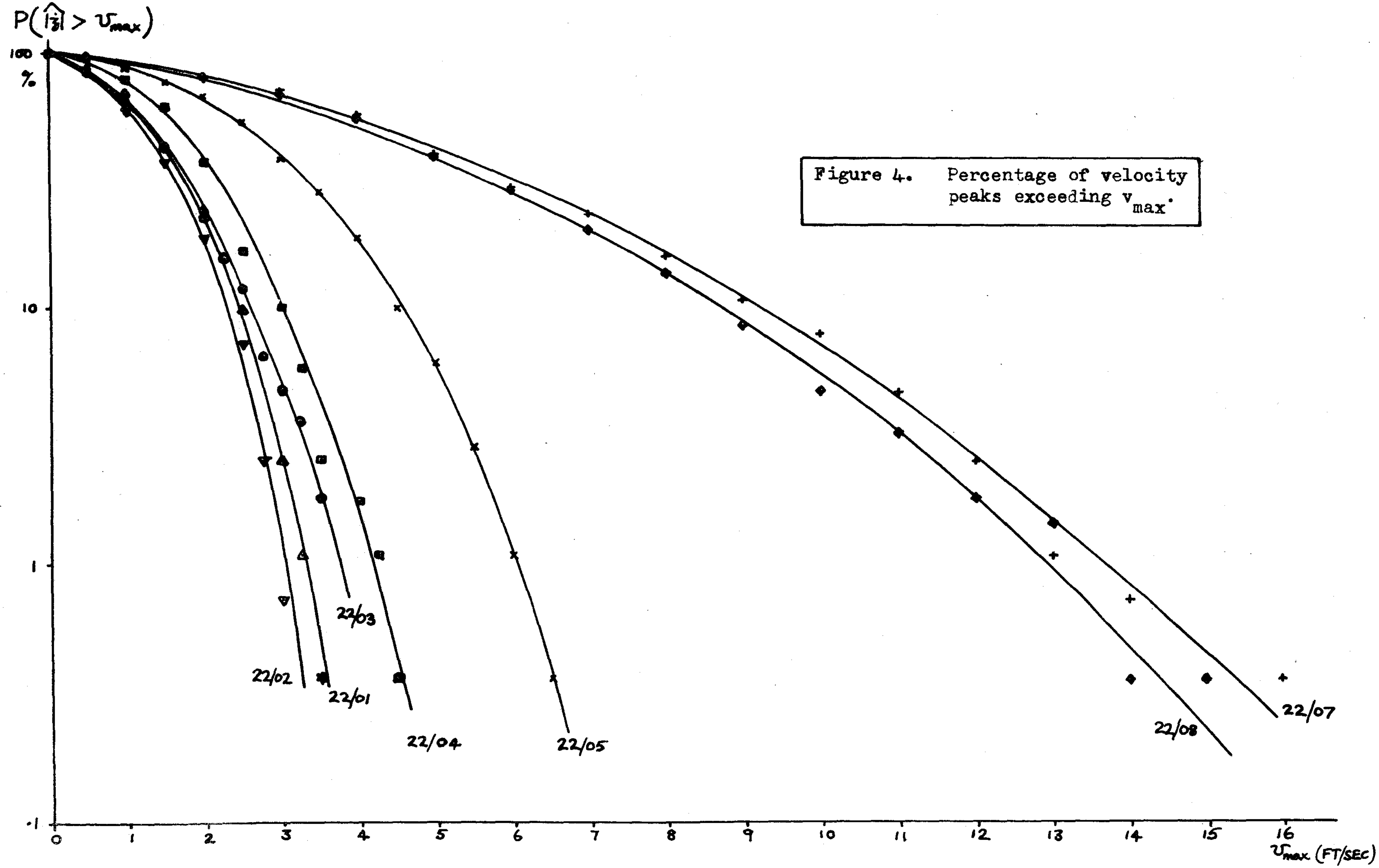


Figure 4. Percentage of velocity peaks exceeding v_{max}.

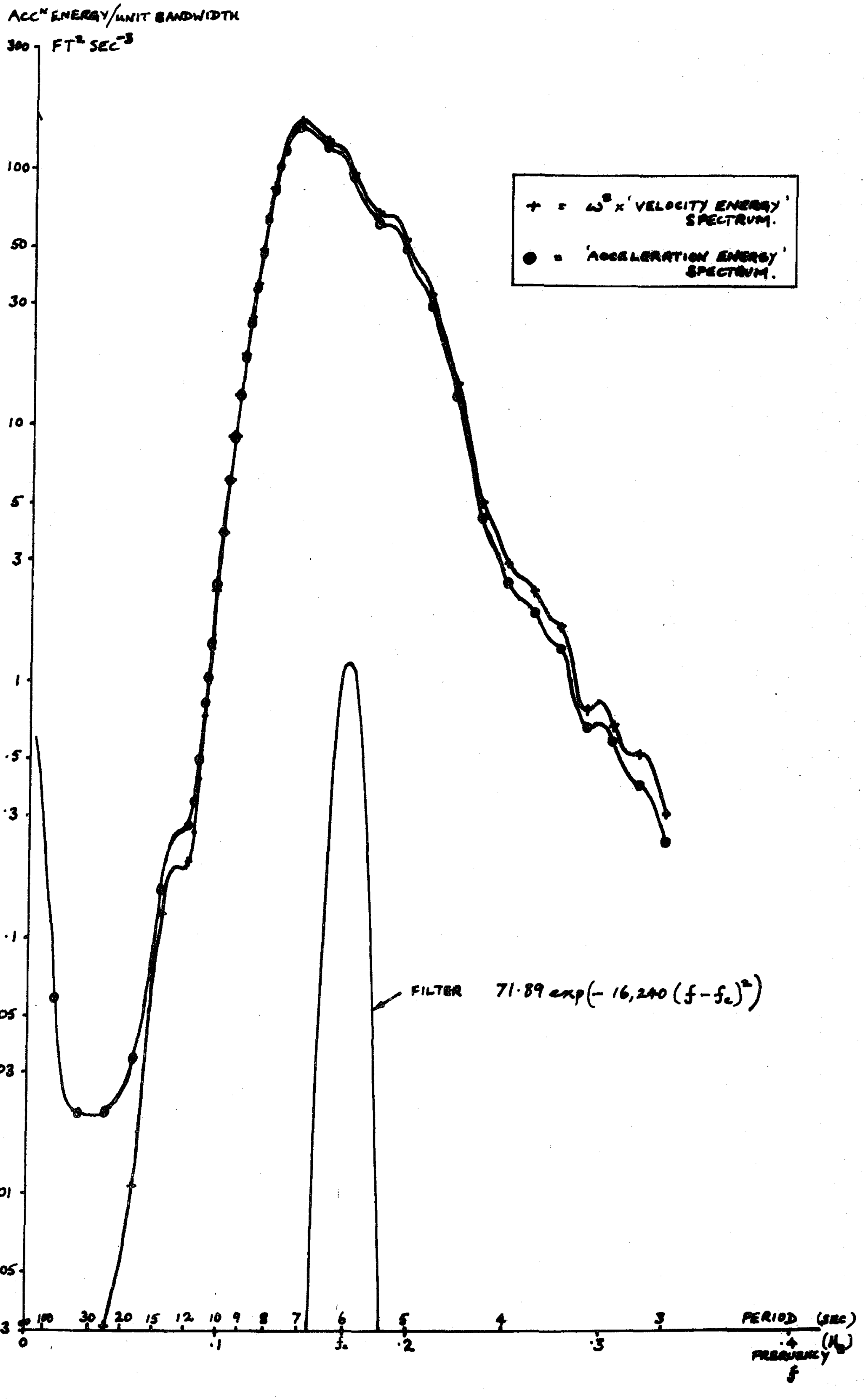


Figure 5. Comparison of energy spectra derived from acceleration and velocity records (22/05), smoothed by gaussian filter shown.

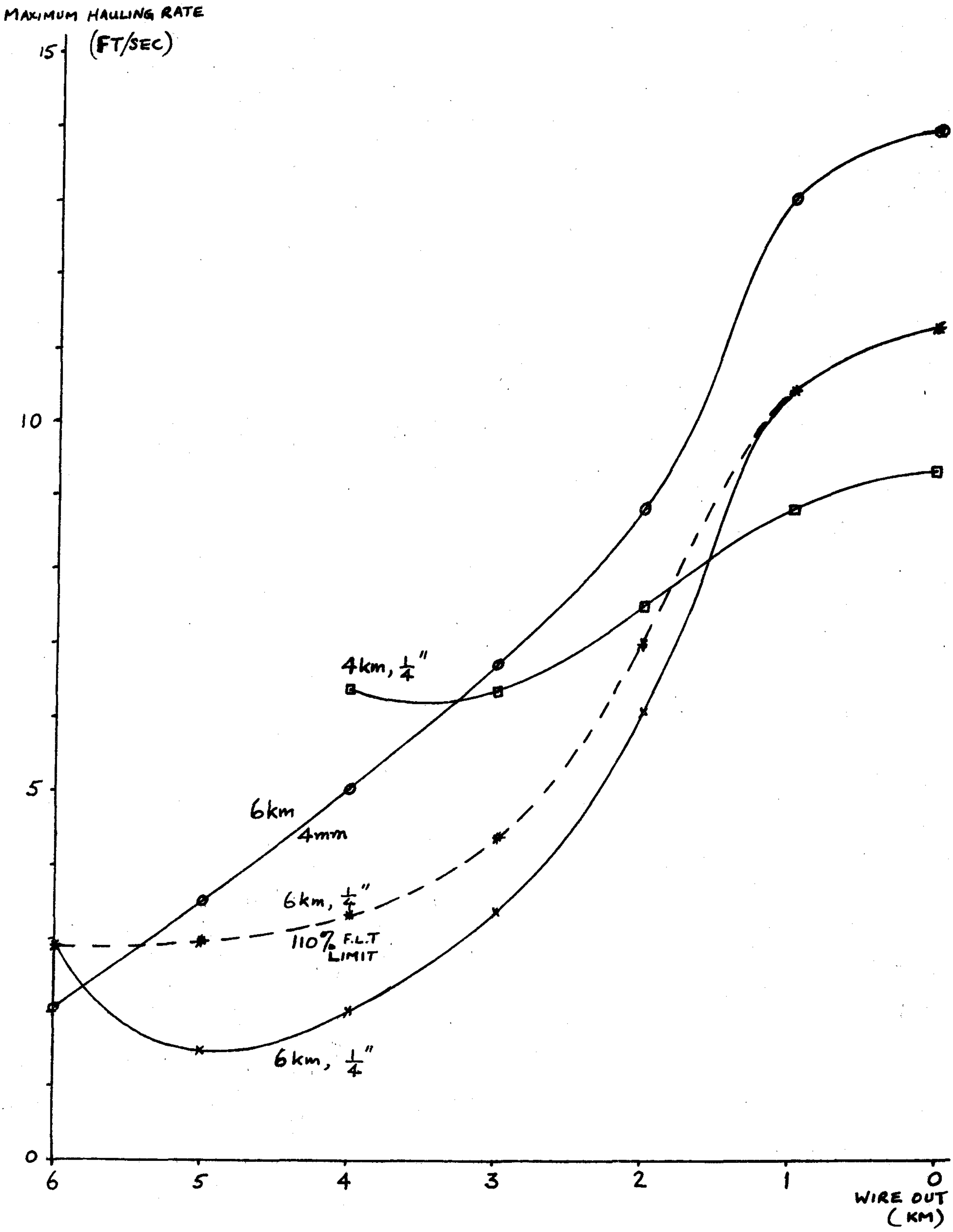


Figure 6. Maximum hauling rate against wire out.

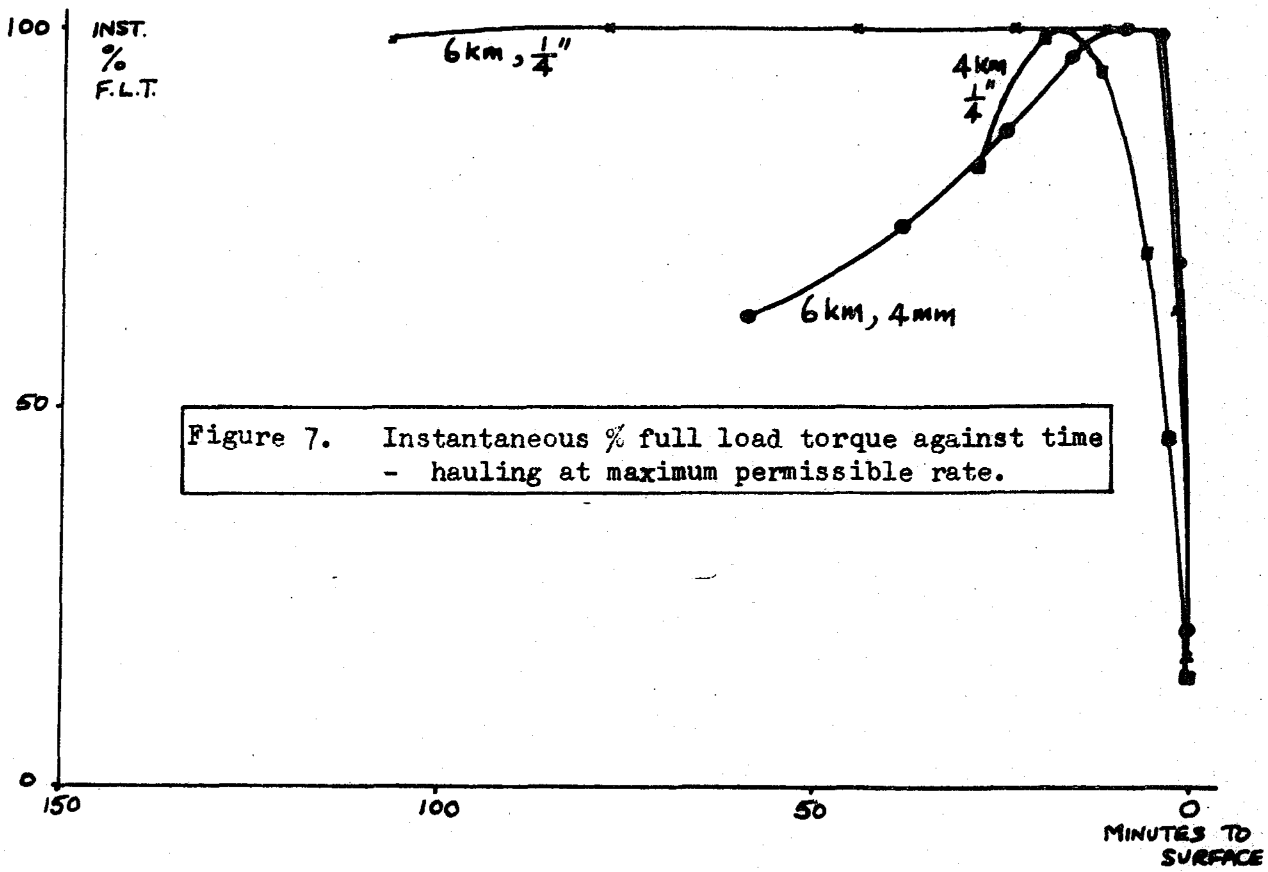


Figure 7. Instantaneous % full load torque against time - hauling at maximum permissible rate.

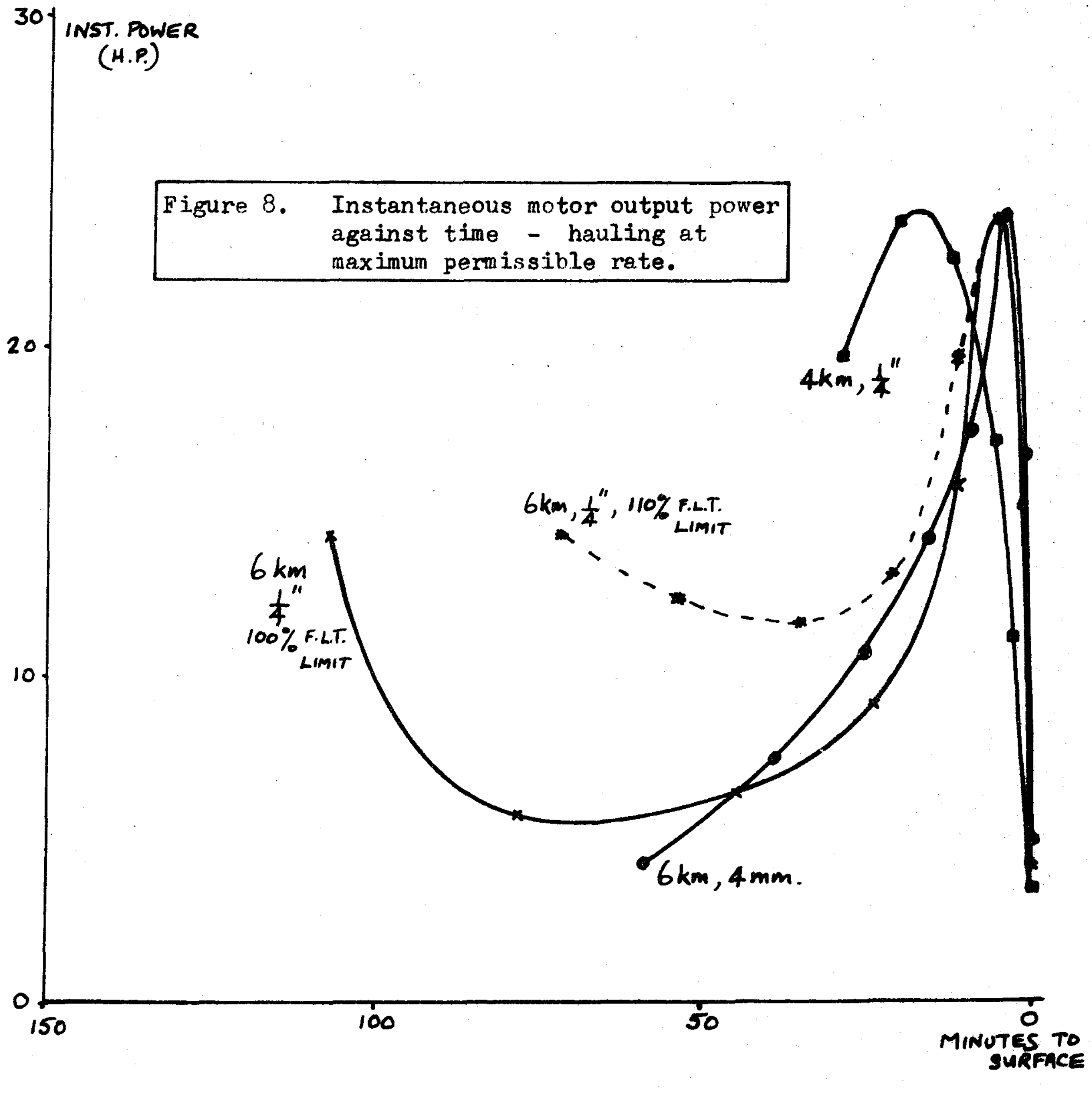


Figure 8. Instantaneous motor output power against time - hauling at maximum permissible rate.

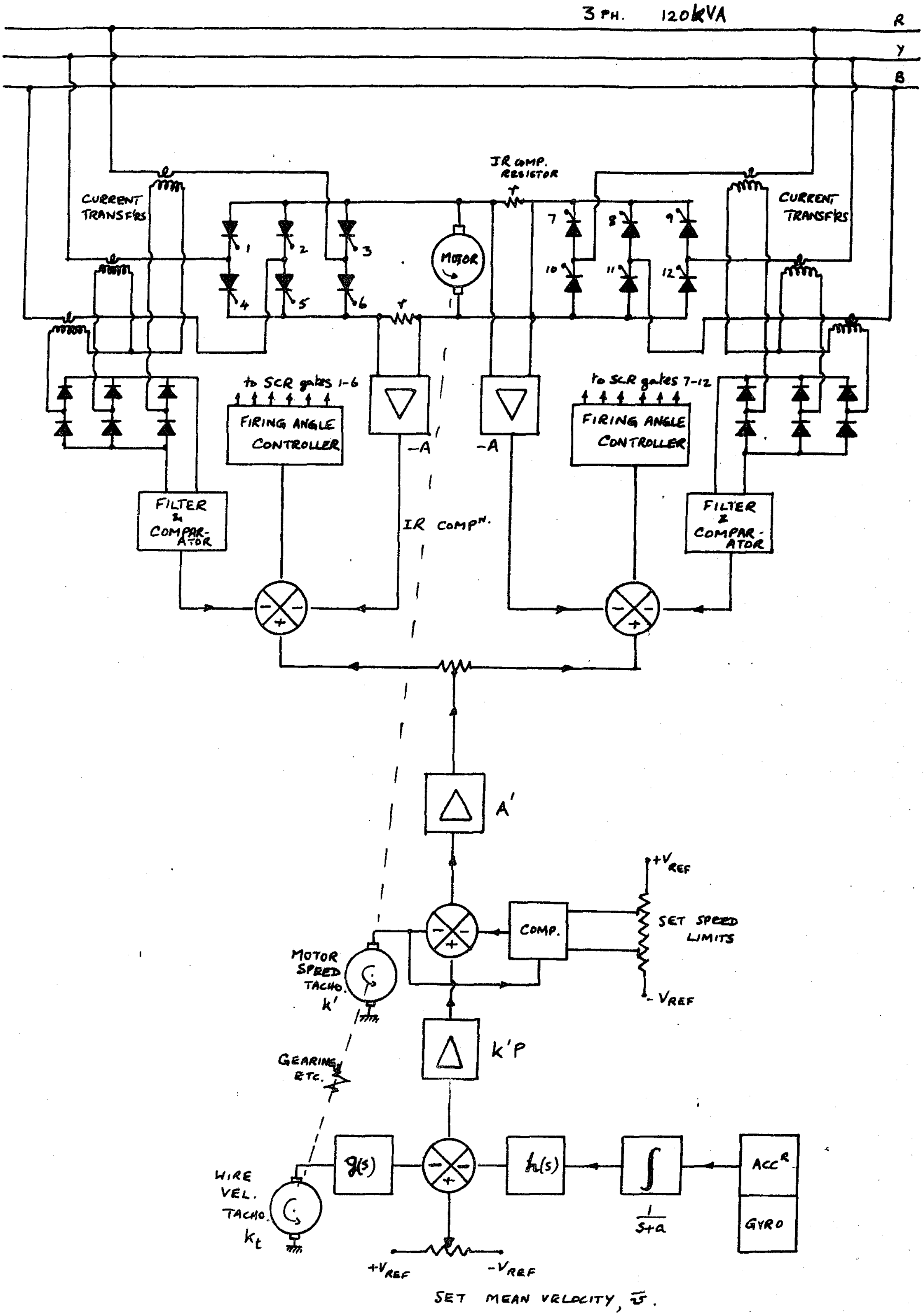


Figure 9. Rough schematic of winch motor control system.

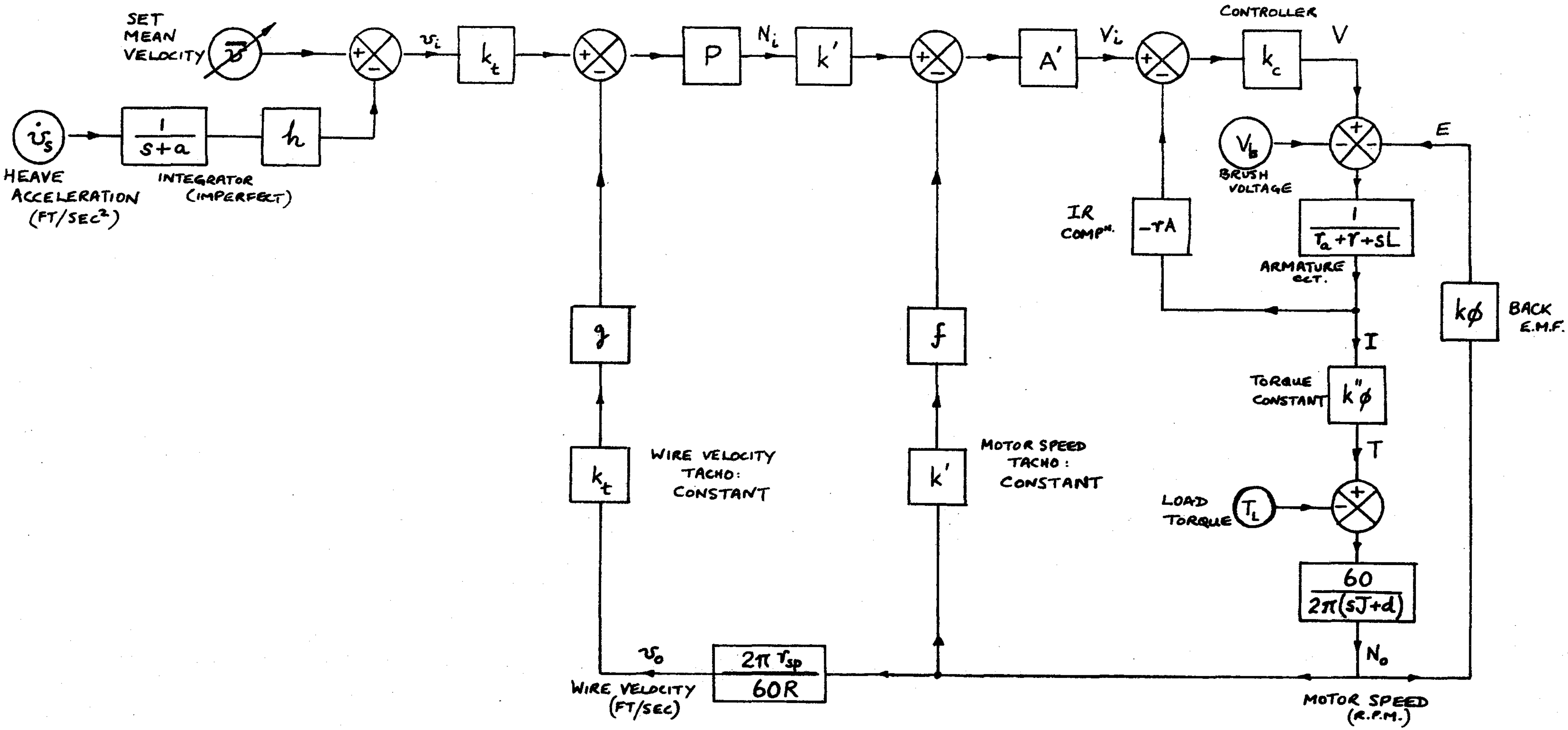


Figure 10. Servo system - Block Diagram.

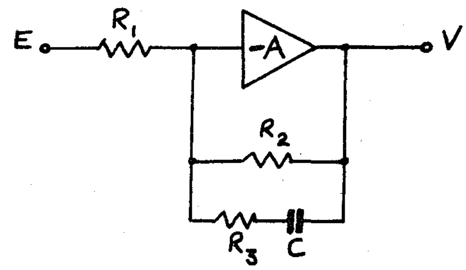


Figure 11. Phase advance network h(s).

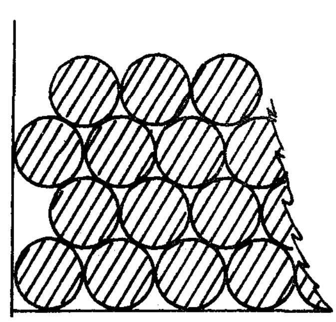
$$\frac{V(s)}{E(s)} = \frac{R_2 R_3 \left(1 + \frac{1}{SCR_3}\right)}{R_1(R_2 + R_3)\left(1 + \frac{1}{SC(R_2 + R_3)}\right)}$$

If $R_2 = (N + 1) R_1$

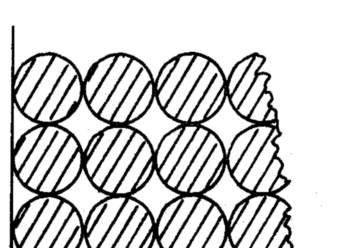
$R_3 = \frac{(N + 1)}{N} R_1$

$$\frac{V(s)}{E(s)} = \frac{1 + \frac{b}{S}}{1 + \frac{b}{(N + 1)S}}$$

$$b = \frac{1}{CR_3}$$



(a) Ideal spooling



(b) imperfect spooling

Figures A(a) and (b). Part sections through spooled wire.

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