REPLENISHMENT AT SEA

BY

K. J. MITCHELL, (*Technical Director G.E.C., Process Engineering, Ltd.*) and
D. J. STRONG, B.SC., C.ENG., M.I.E.E., R.N.S.S. (*Ship Department, M.O.D.(N).*)

The paper reviews the present state of development of new form of replenishment-at-sea system, for use in Sea State 6 conditions, being developed jointly by G.E.C. (Process Engineering) Ltd. and the Ministry of Defence (Naval). The theoretical principles behind the new system are described in some detail and the essential control relationships are presented. A variety of analogue computer outputs contrasting the behaviour of the new system as compared to constant tension systems, are discussed and illustrated.

A fundamental objective in evolving the control philosophy for the new system has been to eliminate the oscillatory tendency present in a constant-tension system. This tendency is discussed and the reasons for it determined. It is shown that a system in which control is based only on tension measurement is fundamentally unsatisfactory in the sense that it cannot meet in an adequate manner the various requirements of an R.A.S. system designed to work in rough sea conditions. The reasons for this state of affairs are described. A table comparing performance of existing Royal Navy systems with the new system is presented. The prototype hardware being built for sea trials in the late summer of 1970 is described and illustrated.

INTRODUCTION

Initial studies into the problem of replenishing naval ships underway at sea were undertaken by G.E.C. for the Ministry of Defence (Naval) in 1963 and were then concerned with the design of an improved constant-tension winch, which at that time was considered an essential basis for a naval underway replenishment system.

Existing systems based on the constant tension principle have been highly developed in both the British and U.S. Navies and have operated satisfactorily for some years up to certain sea state conditions. However, during the course of the early study sufficient was learned of the behaviour of a system based on this principle to throw doubts on its suitability for use in severe sea conditions.

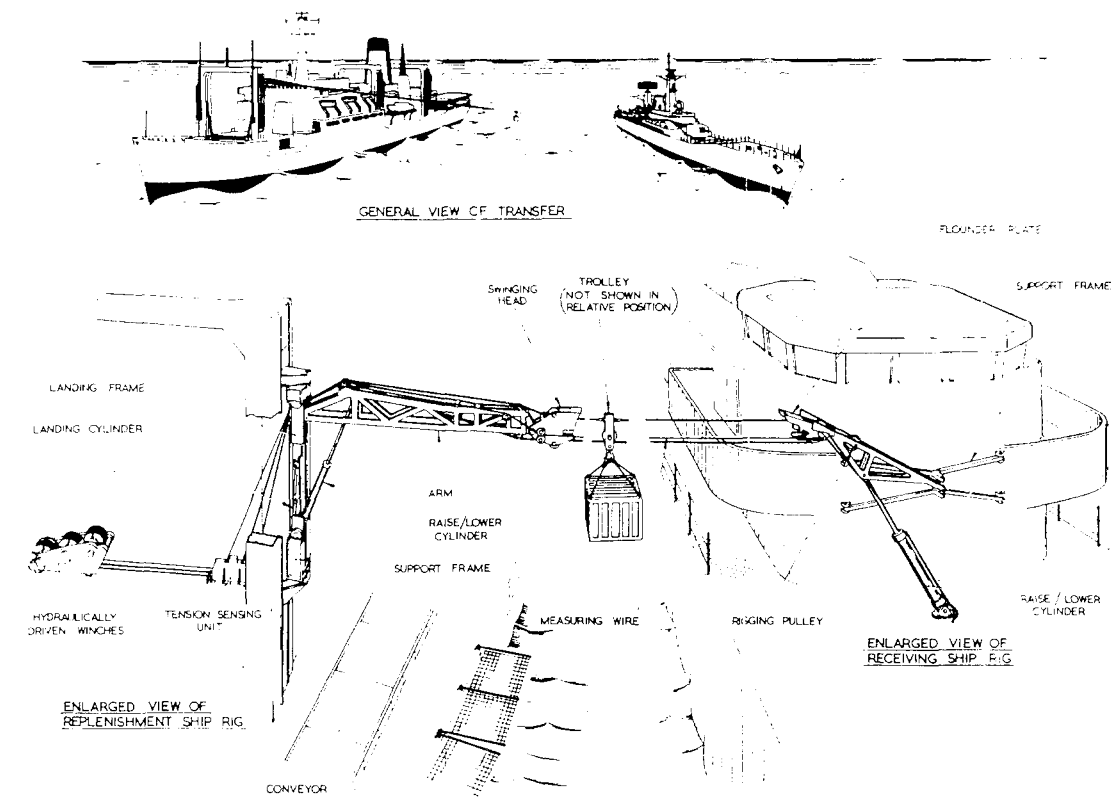

FIG. 1

The objective, therefore, following these preliminary studies has been the further development of a replenishment system by means of which the transfer of packaged or fluid materials could safely take place under more severe sea state conditions than has hitherto been possible and eventually to increase the range of application for such equipment to suit present day requirements for ocean drilling rigs and super tanker-to-tanker underway fluid transfer operations.

Upon examination it was found that a constant tension replenishment system could be inherently oscillatory and when applied to practical dimensions tended to give rise to systems having a resonance with a period close to the roll period of some of the smaller vessels. This situation it was felt could constitute an explanation of difficulties experienced with existing equipment in rough seas. An analogue computer simulation clearly demonstrated the natural tendency of such a system to go into uncontrollable oscillation under certain conditions and subsequently a variety of modified control techniques were examined with a view to damping down this oscillatory tendency while at the same time preserving the simplest possible arrangement of equipment and departing as little as possible from well proved existing techniques.

In practice a system has been evolved which appears to offer a dramatic improvement in the behaviour of the load in conditions where massive vertical oscillation would normally be expected and a prototype naval underway replenishment system employing the new principle of control is now under construction and is illustrated in FIG. 1.

Various attempts have been made to overcome the violent oscillation sometimes experienced with existing R.A.S. equipment, including the use of relatively high line tension and rapid load transfer velocities. Both of these trends are broadly undesirable, the first because it leads to unnecessarily heavy equipment in relation to the actual load carried, and the second because it leads to additional complication at the terminals, particularly the receiving ship terminal. This, in the case of a warship, tends to give rise to an undesirable maintenance problem

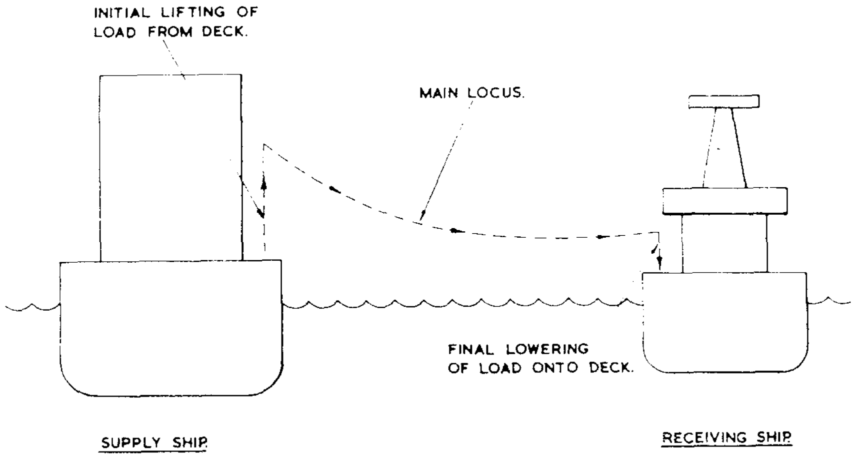

FIG. 2—IDEALIZED LOAD PATH FROM SUPPLY TO RECEIVING SHIP

and to take up space which would better be devoted to weapons or other equipment for which the ship was originally designed.

An ideal replenishment-at-sea system will provide safe passage for the cargo from the hold of the supply vessel to the hold or stowage in the receiving vessel (see FIG. 2). Thus, the entire 'hold-to-hold' handling complex should ideally be designed as a single integrated system although we are concerned here with part of such a system capable of semi-automatic 'deck-to-deck' transfers and suitable, with relatively minor variations, for use in a wide range of ships and rigs. Although the equipment has been evolved with the problem of packaged cargo primarily in mind, it is capable of ready adaptation for the transfer of fluids. The design requirements for a typical naval system could well be as follows:—

(*i*) A naval replenishment-at-sea system must be of the highest reliability and designed in conformity with the underwater shock resisting specifications appropriate to the two ships.

(*ii*) It should be possible to operate the system safely at night, if needs be, and in weather conditions where perfect visibility may not be available.

(*iii*) Rigging and operation of the system should both be as rapid as possible, the trolley travel speed being such as to make full use of the installed prime mover horsepower.

(*iv*) The system must be capable of operating in the worst specified set of circumstances with minimum loss of and damage to cargo transferred and with maximum safety to personnel operating the equipment.

(*v*) The system should be so designed as to enable it readily to form part of a fully automated 'hold-to-hold' transfer system. This latter requirement is of particular importance when the transfer of guided missiles or other costly items of equipment has to be undertaken.

The following brief specification is quoted as an example but the principles described in this paper may be applied to equipment designed for other loads and speeds:—

Maximum normal payload	— 2 tons
Nominal rope tension T_H	— 3 tons
Maximum nominal ship separation	— 200 ft
Maximum horizontal highpoint acceleration accommodated (approx.)	— 12 ft/sec^2

Maximum vertical highpoint acceleration accommodated (approx.)	— 10 ft/sec^2
Maximum highpoint velocity accommodated	— 20 ft/sec

SHIP MOTION AND SEA MOTION

The principle of transferring packaged or fluid cargoes from a supply ship to a receiving ship or ships, or from super tanker to tanker, steaming on parallel courses, is distinct from that of transferring stores to a stationary rig from a supply vessel and involves different approaches to the problem of station keeping. The naval or ship-to-ship problem is essentially one of seamanship and assumes that both ships are under way and able to maintain station with sufficient accuracy using the normal propulsion and steering systems of the ships. The second case which is applicable in principle to any situation involving a moving ship in relation to a stationary body (for example the replenishment of a lightship), involves an entirely different supply ship positioning problem. In this case it is clear that in certain states of wind and tide the normal propulsion and steering systems of the supply vessel are incapable of maintaining the vessel in fixed relationship to the stationary drilling rig.

The five-motion response of a ship to a given combination of sea and wind conditions is a matter of considerable complexity and for analytical treatment a variety of simplifying assumptions become necessary. It is clear that with two ships steaming on parallel courses and comparatively close together (in the range 80 to 200 ft) the wave formations in the water between the two vessels will be influenced not only by the generally prevailing sea state and wind conditions but also by the wave formation of the two vessels.

These are likely to have varying characteristic periods and will coincide at intervals. When this occurs a relatively high wave is produced in the water between the vessels and this probably constitutes the limiting condition for wave height against which the replenishment system must be designed. That is to say, immersion of the load under these circumstances must be avoided by appropriately designing the geometry of the system. This point is of great importance since the coincidence of bow waves is a phenomenon which is likely to occur at intervals of 5 to 25 seconds, depending on the mass of each of the two vessels, their distance apart and the prevailing natural wave amplitude. It is therefore inevitable that sooner or later coincidence of bow waves will occur at the point and time when the load is at the lowest position in its path between supply and receiving ships.

It is fortunate that wave stability considerations (Ref. 1) place a natural limit on wave slope of 1 in 7. Therefore it is safe to assume that the form of the waves between the two ships will be contained within an isosceles triangle with a base angle not exceeding 10 degrees. This enables some approximation to be made of the limiting wave height (apart from spray) which can occur. It is also fortunate that the reduction in pressure in the water between two ships, due to interaction, produces some additional clearance between the maximum wave and the underside of the load.

The yardstick taken for detailed design of system dynamics is that the system shall be fully operational for over 98 per cent of its life at sea, when replenishing between two given vessels, with a risk of less than one in a hundred thousand of immersing the load.

Ocean Wave Statistics (Ref. 2), indicates that in the North Atlantic 98·5 per cent of all waves will be lower than 21 ft. Therefore the system is designed to be fully operational in Sea State 6 (waves 13-20 ft, see TABLE II).

To simplify design procedures, it is convenient to express the motions of the two ships in a form readily reduceable to common engineering terms. The

TABLE I—*Ship motions expressed as simple harmonic motions*

	Supply Ship		*Receiving Ship*	
Component	*Amplitude*	*Period (Secs)*	*Amplitude*	*Period (Secs)*
Roll	$\pm 8°$	15	$\pm 10°$	9
Pitch	$\pm 2°$	8	$\pm 2°$	6
Heave	$\pm$ 10 ft	9	$\pm$ 10 ft	7
Sheer	$\pm$ 5 ft	14	$\pm$ 5 ft	10
Yaw	$\pm 1\frac{1}{2}°$	16	$\pm 1\frac{1}{2}°$	8

TABLE II—*Sea-state code*

Code Figure	*Description*	*Mean Max. Height of Waves —ft.*
0	Calm (Glassy)	0
1	Calm (Rippled)	0–1
2	Smooth (Wavelets)	1–2
3	Slight	2–4
4	Moderate	4–8
5	Rough	8–13
6	Very Rough	13–20
7	High	20–30
8	Very High	30–45
9	Phenomenal—as might exist at the centre of a hurricane	Over 45

five-motion behaviour of the selected ships is so expressed in the form of simple harmonic motions in Table I. Surge—the sixth motion—has little significance in the context of replenishment at sea and is not included. Ship motion statistics indicate that the chance of one ship exceeding the amplitude in TABLE 1 is less than one in a thousand in Sea State 6. It follows that the risk of immersing the load due to a dynamic overload on the circuit system is less than one in a hundred thousand, for the following reasons:—

(*a*) Additional margins are incorporated in the design.

(*b*) Extreme adverse conditions are unlikely to occur simultaneously in both ships. Unfortunately lack of statistical data on the relative motions between ships steaming in close company on parallel courses, precludes making reliable calculations in this area.

(*c*) Replenishment will usually take place in conditions lower than Sea State 6.

Having arrived at an understandable expression of basic ship motion, it is clearly a straightforward exercise in geometry to translate the significant components of the various motions into actual movements of the high points on the two vessels to which the replenishment system is attached.

In practice, of course, it is the horizontal and vertical components of motion in a plane at right angles to the main axes of the ships which are of significance to the design of the replenishment system. These components may be represented by approximately rectangular 'envelopes of motion' which define the maximum excursions in the one plane produced by the combination of resolved components with the ships at fixed nominal separation distance. Needless to say such motions must be considered to have completely random phase relationship. In practice roll, heave and sheer are the most significant components of motion when considering the planes at right angles to the axes of the vessels.

A further matter of importance is that the sea-keeping characteristics of ships vary widely. While the brief specification mentioned earlier will ensure satisfactory performance between the two ships for which the system was designed, other pairs of ships will have different limiting operational conditions. Nevertheless, the fact that the torque/inertia ratio is fifty times better than current practice means that in any situation far greater opportunities will be made for safe replenishment at sea.

Furthermore, in cases where the 'hold-to-hold' system is not fully automated, there exists the problem of handling the cargo on the decks of the two ships so that a practical consideration arises automatically to limit the objective of the designer. It is in general considered that difficulty in handling cargo on deck arises for rolls greater then $\pm$ 10 degrees, depending very largely on the size of the vessels involved and the dimensions of the packages being handled.

GENERAL TECHNICAL DESCRIPTION

In the past it has been general practice to make use of a jackstay or highline for the support of the load and a separate inhaul/outhaul system to traverse the load between the two ships (see FIG. 3). The jackstay is held at a predetermined constant tension which may be achieved in a variety of ways and systems have been employed making use of gravity counterweight arrangements, air/hydraulic ram tensioners and constant tension winches of various designs. Except for the gravity counterweight system, it is necessary to employ control gear of some kind in order to keep the tension constant. As the objective is to keep the tension perfectly constant, it will be seen that immediate limitaations occur in the achievable response of the system because of the time constant of the tension measuring devices and because of the inertia associated with the tensioning device and the rope itself.

However, it is by no means clear that a practical constant tension system must have a worse performance than a 'perfect' one would have.

The essential requirements which any jackstay system must meet are:—

(*a*) The jackstay must pay in and out as necessary to accommodate the horizontal fluctuations in highpoint separation.

(*b*) The load must follow vertical movements of the ships to a sufficient extent—for example, if the two ships were heading into sea their heave relative to the sea would be small but the absolute heave in Sea State 6 would be $\pm$ 10 ft. If the load remained stationary in space, its movement relative to the sea surface would be $\pm$ 10 ft.

(*c*) Any oscillatory movement of the load must be sufficiently damped.

If the paying in and out of the jackstay is to be controlled by rope tension alone, these requirements are in conflict since (*a*) requires that a small change

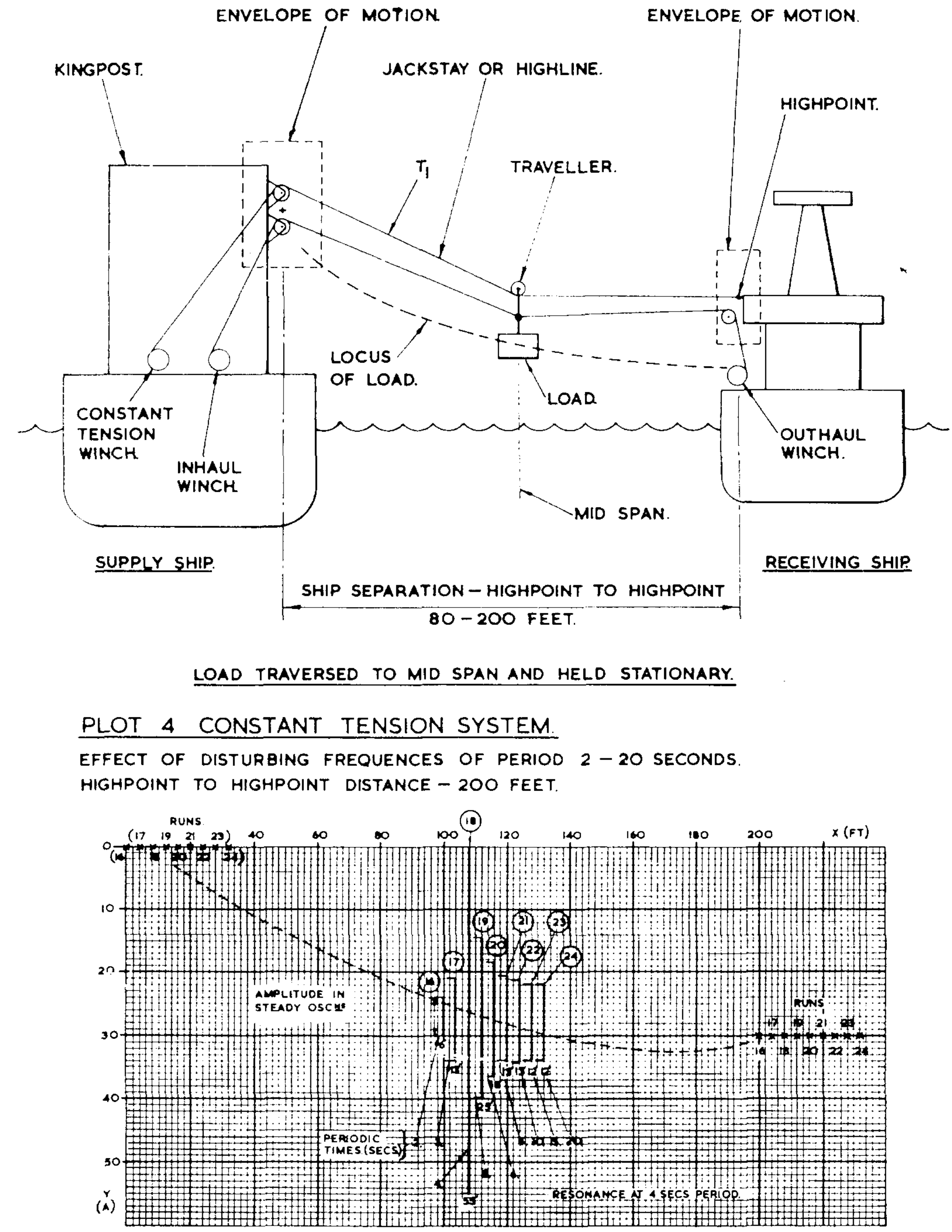

FIG. 3—JACKSTAY OR HIGHLINE SYSTEM WITH COMPUTER OUTPUT SHOWING TENDENCY TO RESONANCE

in tension shall produce a large change in jackstay velocity, (*b*) requires that a large change in tension shall produce only a small change in velocity and (*c*) requires that a small change in velocity shall produce a significant change in tension.

A 'perfect' constant tension system would meet requirement (*a*) completely, but would not meet requirements (*b*) and (*c*) at all. A practical constant tension system meets requirement (*a*) imperfectly, owing to the inertia of the components, and the resulting tension fluctuations cause displacements of the load from the equilibrium position, which becomes large as ship motions increase. At the same time, the change of tension with velocity is not nearly great enough to satisfy requirements (*b*) and (*c*). A closer approach to the 'perfect' constant tension characteristic would therefore result in an improvement in the perform-

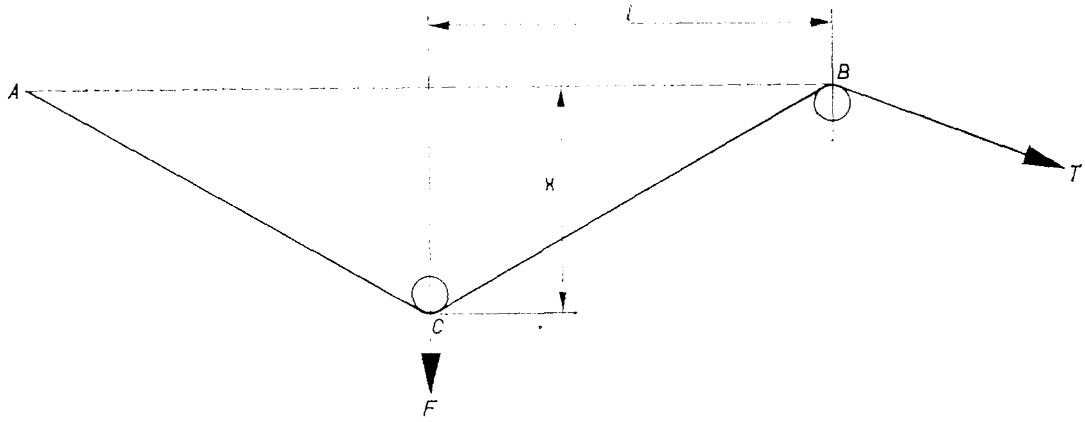

FIG. 4

ance against requirement (*a*) and a further worsening of the performance against requirements (*b*) and (c). It is possible therefore that a practical, imperfect constant tension system could have a performance rather better than a 'perfect' constant tension system.

Further, the 'perfect' constant tension system is inherently oscillatory because although deflection of the load does not cause a change in rope tension, the change in rope geometry causes a restoring force. This gives a low natural frequency of oscillation which is more likely to be excited by the ship motions.

In this connection and with reference to FIG. 4, consider a horizontal wire with one end fixed at A, while the other end passes over a frictionless pulley at B to a device which maintains a constant tension in the wire.

Let the horizontal distance AB be $2l$.

Let the sag of the centre C of the wire be x, when a vertical force F is applied at C.

Let the tension in the wire be T.

Equilibrium of forces at C gives

$$F = 2T \cdot \frac{x}{\sqrt{l^2 + x^2}} \quad (1)$$

The system behaves as a non-linear spring with the force-deflection characteristic given by equation (1).

If a mass M is hung on to a pulley at point C, an oscillatory system will be created whose natural frequency will vary with the deflection. If the deflection is small the period will be

$$P = 2\pi \sqrt{\frac{Ml}{2gT}} \quad (2)$$

For larger deflections, the period will be increased by the factor $\left(1 + \left(\frac{x}{l}\right)^2\right)^{\frac{3}{4}}$.

If, for example, the sag was half the span, this factor would be $2^{\frac{3}{4}} = 1{\cdot}68$, i.e., the resonant frequency of the system is amplitude dependent. It follows that oscillation can be excited by quite a wide range of frequencies, most of which exist in the spectrum of ship motion.

For example, a 2-ton load on a 6-ton cable spanning 150 ft would have resonance periods in the band 3·9—6·6 seconds per cycle. It is pertinent that wave periods for the North Atlantic are predominantly in the 5-8 second band,

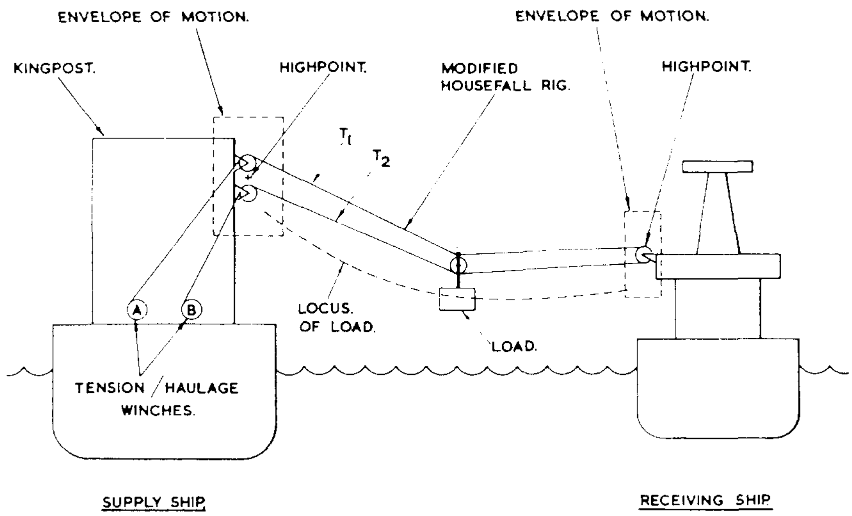

FIG. 5—SIMPLE MODIFIED 'HOUSEFALL' OR LOOP RIG WITH TENSIONING SYSTEM

according to Hogben and Lumb, in their book *Ocean Wave Statistics*, published by H.M. Stationery Office.

In order to damp oscillation of the load, it is necessary that there shall be a nett extraction of energy from the rope system in each cycle of operation. With a 'perfect' constant tension system this does not occur, since there is no change in tension during the cycle. At the other extreme from the constant tension system, if the winches were held stationary, there would also be no damping, since although there was a large change in tension, there would be no change in winch velocity. Between the extremes, there is an optimum relationship between tension change and winch velocity which gives maximum damping.

The simple jackstay with inhaul/outhaul system involves the use of an outhaul winch on the receiving ship. Such an arrangement has the disadvantage of taking up space in the receiving ship but has the advantage that the traversing of the load into the zone of the receiving ship is under the control of the receiving ship itself. If the position of the load is to be well controlled, it is necessary that both inhaul and outhaul ropes be taut. This necessitates the inhaul and outhaul winches (or at the very least the inhaul winch), being under constant tension control, in the same way as the jackstay. The load vertical position is then dependent on the inhaul and outhaul rope tensions, as well as the jackstay tension—this is likely to result in further worsening of dynamic performance.

FIG. 3 includes an analogue computer output of a constant tension system in which the load has been traversed to mid span and stopped. After this the load has been subjected to a series of disturbing frequencies in the range 2—20 seconds period. It will be seen that the marked resonance occurred at 4 seconds period showing the need for further control elements in the system if violent oscillation of the load is to be avoided under all circumstances.

A great variety of other layouts have been proposed or tried out in the past, one of which is the so-called 'Modified Housefall' rig (see FIG. 5). In this system no separate jackstay is employed. Instead the load is shared between two ropes forming a continuous loop and both tensioning of the system and traversing of the load are achieved by means of a pair of winches placed in the supply ship. A number of variants of this scheme have been suggested involving separate arrangements for tensioning and traversing but the basic principle remains the same. This rig has the advantage of requiring the minimum of equipment in the receiving ship and this is one reason for its adoption by G.E.C. The scheme has a number of other advantages, one of which is the use of smaller diameter rope for a given load and therefore smaller mechanical components throughout the rope handling parts of the system.

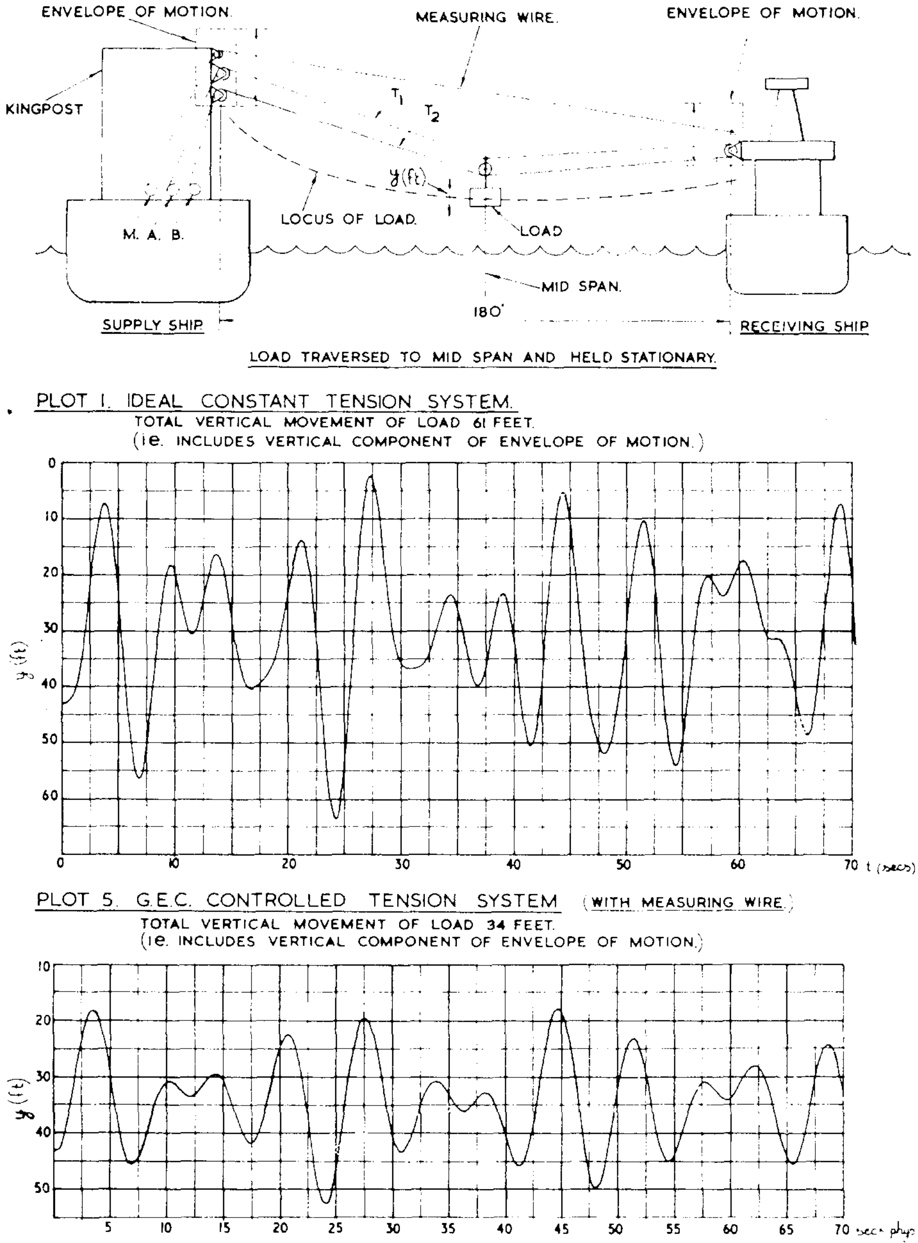

FIG. 6—ABSOLUTE VERTICAL MOVEMENT OF LOAD

The control solution adopted by G.E.C. following comprehensive analogue computer studies, is to cater for requirement (*a*) above by measuring the high-point separation velocity directly with a measuring wire and driving the winches from this. Tension error control is applied in addition and the degree of control can be adjusted to optimize the performance against requirements (*b*) and (*c*), which are not seriously in conflict.

FIG. 6 shows the general layout of the G.E.C. replenishment-at-sea system. It may be compared with FIG. 3 and FIG. 1 and will be seen to include the additional element in the form of the separate 'measuring wire' already mentioned. Also on FIG. 6 will be seen two analogue computer output plots contrasting the performance of the G.E.C. system with a constant-tension system. During these runs the load was traversed to mid span and the absolute vertical movement y ft measured over a period of time. In the case of these runs the vertical movement consists of two main elements. These are, firstly, the component due to the motions of the two ships which are in random phase relationship

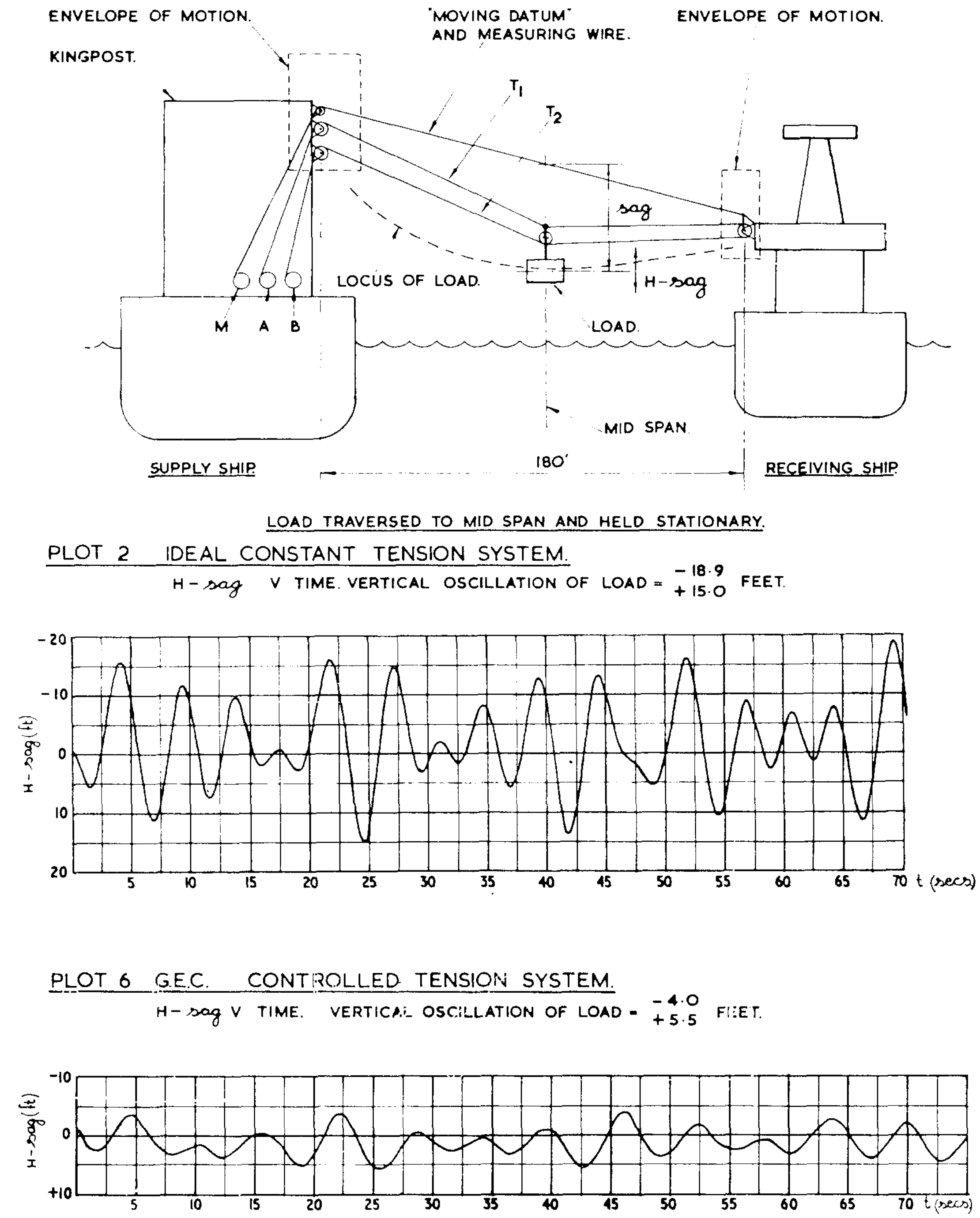

FIG. 7—VERTICAL MOVEMENT OF LOAD RELATIVE TO MOVING DATA

and as defined by TABLE I and secondly the component associated with the vertical oscillation of the load.

It will be noted that the maximum amplitude of movement with the constant-tension system is 61 ft. and with the G.E.C. controlled tension system 34 ft. As will be seen from the diagram, this latter dimension is mainly attributable to the vertical components arising from the 'envelopes of motion'. In view of the random phase relationship previously mentioned, it was necessary throughout the analogue computer work to make an initial assumption as to relative ship motion at the start of each run. The assumptions made in this case are noted on the diagram.

Referring now to FIG 7, it will be seen that here is introduced the concept of a 'moving datum', an imaginary straight line stretched between the highpoints of the two ships and occupying an identical position to the measuring wire previously introduced. If again the load be traversed to the mid span position and allowance made for the natural sag of the load carrying wires, then the variation of this figure 'H-sag' will be seen to be a means by which the effect of the vertical components of the 'envelopes of motion' can be eliminated and hence the remaining vertical oscillations are attributable to the control system. Plot 2, FIG. 7, shows this output for the constant-tension system over a period of some 70 seconds and shows excursions of minus 18·9 ft and plus 15 ft relative to the moving datum.

Plot 6, FIG. 7, shows a similar output of 'H-sag' for the G.E.C. system. As will be seen, the oscillations are in this case controlled to minus 4 ft plus 5·5 ft and represent a marked improvement in performance as compared to Plot 2. *This improvement is directly attributable to the use of the measuring wire.*

At this stage of the description, the effectiveness of the measuring wire system is fully apparent and it will be appreciated that this control concept can be applied with similar improvement in result to the conventional jackstay or highline system. However, in the case of the G.E.C. design the loop arrangement has been adopted because of the advantages previously mentioned.

It must be emphasised that the improvement in dynamic performance produced by the G.E.C. control system is in fact dependent on two main elements. These are:—

(*a*) The ability of the measuring wire system to respond virtually instantaneously to variations in ship separation.

(*b*) The adoption of hydrostatic drives for the main winch barrels A and B.

Barrels A and B are in both cases driven by low-speed hydraulic motors with low inertia. In this way the inertia of the prime-mover electric motor is completely decoupled from the system via the swashplate controls. It will therefore be seen that the hydrostatic drives are a key feature of the G.E.C. system without which it would have a more limited capability for dealing with rough sea conditions. The measuring wire control system could in principle be applied to an electrically driven winch system making use of Ward-Leonard drives or some solid state equivalent. In this case, however, the drive motors coupled to the barrels A and B would be relatively high-speed electric machines having high inertias referred to the drums. This implies that at best a less satisfactory response would be achieved and the rough weather capability of the system would thereby be reduced.

It will be understood that the 'envelopes of motion' represent the horizontal and vertical components of motion occurring as a result of the combinations of ship motion but with constant nominal ship separation. On the other hand, FIG. 8 shows the relationship (approximately in scale) of the supply ship highpoint to those of a frigate and an aircraft carrier taking into account the variation in ship-to-ship separation which may in practice occur under operating conditions. As well as having an influence on system geometry, this diagram when considered in conjunction with the emergency breakaway condition leads to an appreciation of the required rope storage capacity of the drums.

FIG. 9 is a schematic diagram of the entire R.A.S. system, including the hydraulic power pack which is not visible in FIG. 1. The continuous variation in power demand imposed on the system because of ship motion is smoothed out by this use of a flywheel. The variable stroke hydraulic pumps forming part of the hydrostatic drives are 'up stream' of the prime mover electric motor from the overall response of the system.

As is clear from both FIGS. 1 and 8, hydraulic cylinder operated swinging arm terminals are used in both ships to raise and lower the loads relative to

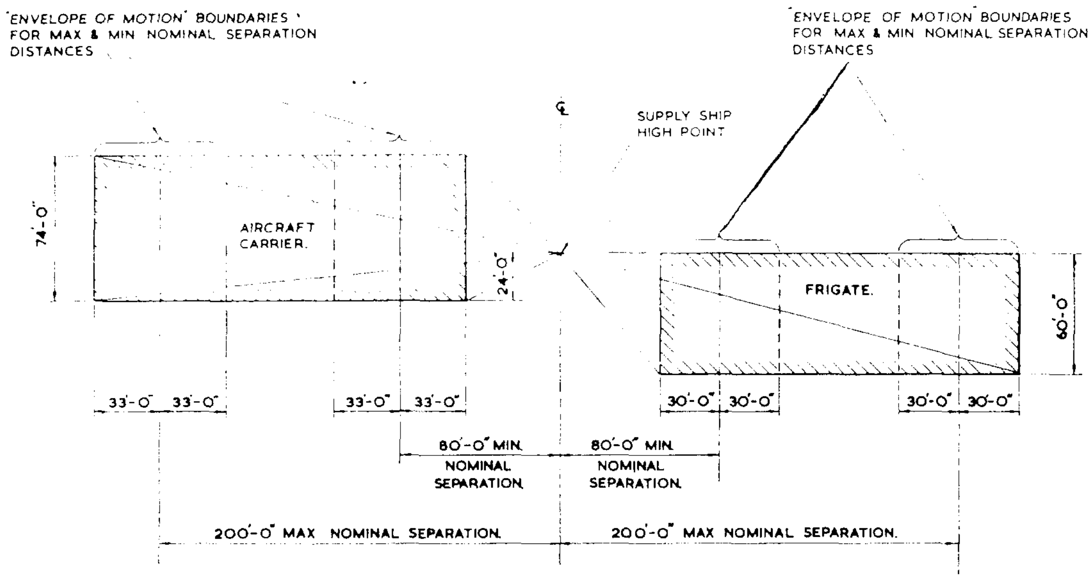

FIG. 8—DIAGRAM SHOWING TYPICAL WORKING RELATIONSHIP BETWEEN HIGH POINTS FOR 80–200 FT NOMINAL SEPARATION DISTANCES AND TABLE 1 SHIP MOTIONS

the respective decks. These parts of the load path are shown in FIG. 2. This terminal design has been adopted because it appears to be the most cost-effective arrangement of the many considered.

CONTROL AND TRANSFER VELOCITY REFERENCING

The approach to the solution of the control problem has been the examination on the analogue computer of a variety of feedback techniques, making use of realistic ship motions based on TABLE 1. Each approach was judged in terms of the reduction in load oscillation relative to the moving datum, as previously described. The best results were obtained when the control of swashplate angle of the pump groups powering the A and B barrels took the following form:—

$$\theta_A = -K_A \,.\, (T_1 - T_N) - K_{2A}\, \dot{l}_A$$

$$\theta_B = -K_B \,.\, (T_2 - T_N) - K_{2B}\, \dot{l}_B$$

Where θ_A and θ_B are the swashplate angles associated with winch barrels A and B respectively,

T_1 is tension in rope from Barrel A

T_2 is tension in rope from Barrel B

T_N is 'nominal' wire tension

$\dot{l}_A$ is haulage velocity demand for Barrel A

$\dot{l}_B$ is haulage velocity demand for Barrel B

K_A is tension error coefficient Circuit A

K_B is tension error coefficient Circuit B

K_{2A} is velocity coefficient Circuit A

K_{2B} is velocity coefficient Circuit B

K_A, K_B, K_{2A} and K_{2B} are constants.

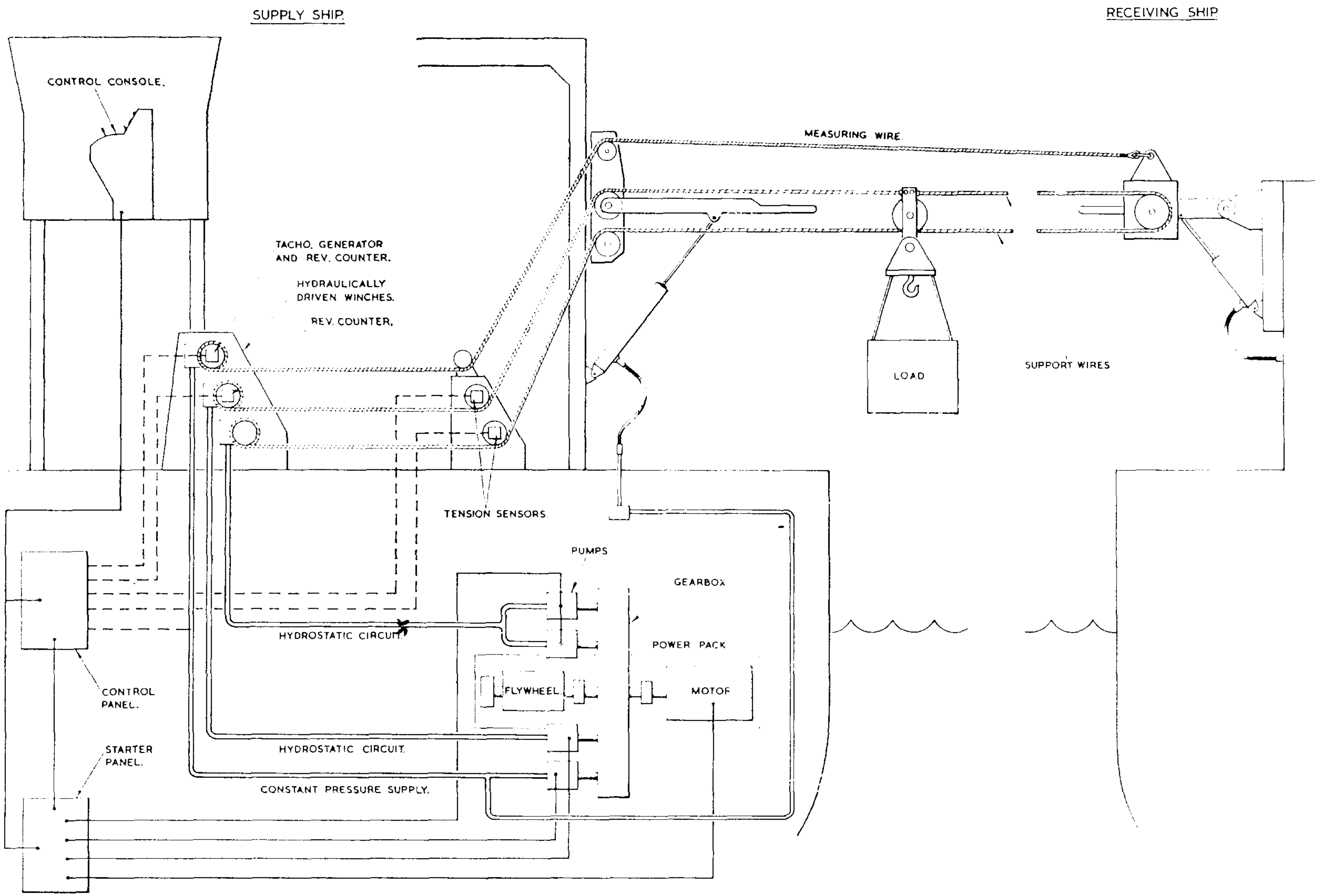

FIG. 9—DIAGRAMMATIC ARRANGEMENT OF TRANSFER SYSTEM

K_A and K_B are initially set up to give a critically damped response to a vertical step disturbance of the load at mid span; K_{2A} and K_{2B} being set to zero. K_{2A} and K_{2B} are then optimized to give the most suitable response under realistic ship motions as defined in TABLE I.

This control produced the improvements in load oscillation relative to the moving datum illustrated in FIG. 7.

A schematic diagram of the entire control system is shown in FIG. 10. It will be seen that both wires of the main loop pass over tension sensors which give rise to the two tension feedback signals T_1 and T_2.

The measuring wire system, which has its own constant tension hydraulic winch, is used to drive a revolution counter and a tachometer-generator. These give rise respectively to the l (distance) and $\dot{l}$ (rate of change of distance or velocity) signals.

It will also be noted that winch drum A is provided with a second revolution counter which gives rise to the feedback signal Z_{DA} and from a study of the diagrams this will be seen to represent the distance of the load from the supply ship. This output is used together with the l signal in the traverse control which is so arranged that within a predetermined distance of either ship the load velocity is controlled relative to that ship, changeover being achieved automatically at about mid span.

The total distance between highpoints on the two ships is divided into three zones referred to as the 'supply ship zone', 'central zone' and 'receiving ship zone'. Control is such that if the load traverse is stopped while in the supply ship zone, it will maintain station relative to the supply ship irrespective of any change of distance between the two ships and similarly in the receiving ship zone.

It will be readily understood that the determination of the zone occupied by the load at any given moment in the traverse may be achieved by comparing the output l from the revolution counter attached to the measuring wire winch drum M with the output Z_{DA} obtained from the revolution counter attached to barrel A, changeover occurring when $Z_{DA} = \frac{l}{2}$.

The Transfer co-efficient $$C_T = \frac{(l - K_{T1}) - (Z_{DA} - K_{T2})}{(l - K_{T1})}$$

and lies within the limits $1 \geqslant C_T \geqslant O$.

The velocity demand signals from the traverse unit to the swash control units A and B are l_A and l_B respectively.

Thus $$\dot{l}_A = (1 - C_T)\, \dot{l} + V$$
$$\dot{l}_B = (1 + C_T)\, \dot{l} - V$$

where $\dot{l}$ is the rate of separation of the highpoints and V is the desired traverse velocity.

Thus these equations defining the control of tensioning and traversing of the loop system and transfer velocity referencing to the two ships were directly evolved on the analogue computer and represent the fundamental basis for the design of the control circuitry of the G.E.C. replenishment-at-sea system. Upon further reference to FIG. 10 it will be seen that the control system includes a 'manual transfer velocity setting'. This permits the operator to select a given nominal transfer velocity and under normal circumstances would be left at the maximum setting. The functions of the traverse control unit may thus be said to be twofold:—

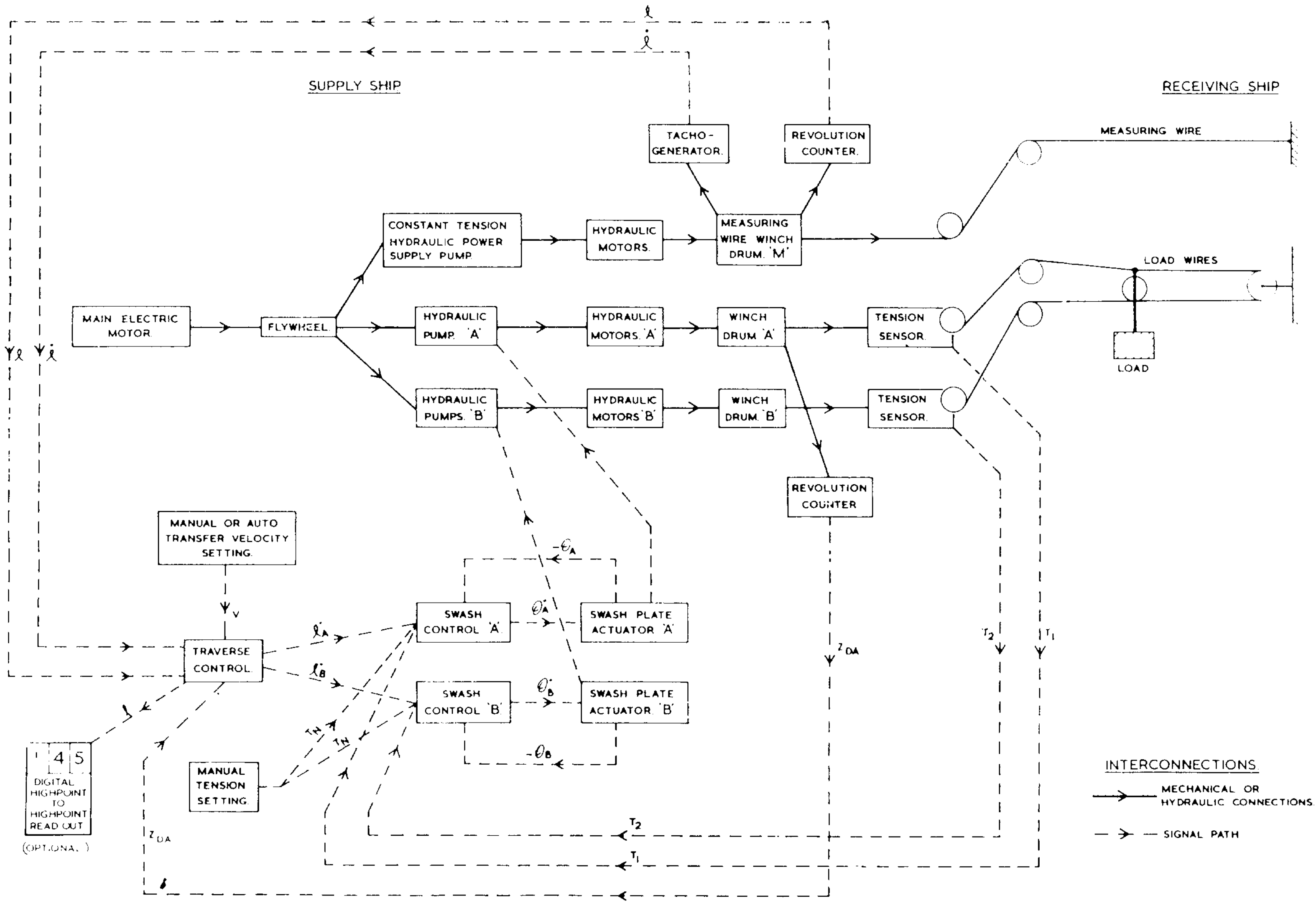

FIG. 10—CONTROL SYSTEM—SCHEMATIC

(*i*) To perform the automatic referencing of the load relative to the two ships, as previously described.

(*ii*) Automatically to reduce the transfer velocity as the load nears either ship.

It is by the latter technique that the necessity for massive shock resisting terminal devices is avoided in the G.E.C. system.

During the course of the investigation, changes in the following variables were investigated:—

Load carried

Transfer speed

Nominal ship separation

Relative suspension height

Applied ship motion frequency

Relative phasing of realistic ship motions.

It is particularly satisfactory that none of these parameters appear too have a significant adverse effect on the stability of the system.

For naval systems the control desk would, of course, be placed in the supply ship and would incorporate the following features:—

(*a*) Remote start-up of the main electrical prime-mover motor and signal light indication that the system is running normally and ready for operation.

(*b*) Manual setting of the transfer velocity to the desired value (the approach and slow-down features are automatic, as previously described).

(*c*) Manual control of raising and lowering the load relative to the deck of the supply ship.

(*d*) Manual selection of the nominal tension T_N if it is desired to have the facility for transferring loads heavier than the design load in less severe weather conditions.

COMPATIBILITY WITH OTHER SYSTEMS

The practical problems which arise in designing an R.A.S. system for use in sea State 6 conditions can be resolved into three main areas. These are:—

(*a*) The effects of ship motion

(*b*) Raising and lowering the loads relative to the deck of the ship

(*c*) Rigging and unrigging of the system.

The effects of ship motion are firstly to produce a tendency for the load to oscillate and secondly to create geometrical problems as the load is being brought inboard over the gunwhale of the ship. In Sea State 6 conditions the raising and lowering of the load must be mechanized to a sufficient extent to avoid hazard due to the swinging of the load. The technique for rigging and unrigging of the system must also be devised with the problems associated with Sea State 6 ship motions in mind.

When all of these matters are considered together, the conclusion is reached that geometrical control of the load is necessary at both the supply and receiving ship ends of the system. This in turn implies that an arm design in accordance with these principles should be used in the receiving ship in all cases.

TABLE III—*Comparison between existing Royal Navy R.A.S. system and the new GEC/M.O.D.(N) system*

Parameter	*Existing R.N. System*	*G.E.C./M.O.D.(N)*
Load Handling	2 ton (2000 Kg)	2 ton (2000 Kg)
High Point Peak Relative Velocity	5 ft/sec (1·5 m/sec)	20 ft/sec (6·1 m/sec)
High Point Relative Acceleration	2·5 ft/sec^2 (0·77 m/sec^2)	12 ft/sec^2 (3·4 m/sec^2)
Inertia Coupled to Drum	55,000 lb/ft^2 (2,350 kg/m^2)	4,250 lb/ft^2 (181 kg/m^2)
Installed Horsepower	195	150
Peak Horsepower	195	1200
Limiting Sea State	4/5	6
Use of Terminal with Geometrical Control	No	Yes (Receiving Ship Optional)

On the other hand it is apparent that this ideal is in practice unachievable. This is because there are already in existence numerous ships designed round other types of R.A.S. rig. In designing the system, therefore, it has been necessary to keep under review the question of compatibility of the new equipment with existing receiving ships. With the roping arrangements chosen it is, of course, under all circumstances necessary to provide a flounder plate with pulley at the receiving ship end of the system. However, it is not absolutely essential that the flounder plate should be attached to a geometrically designed arm, desirable though this arrangement is in foul weather conditions. Instead it is possible to attach the flounder plate to the ship by simple shackle or alternatively via a short link member. With these arrangements it is necessary to lower the load by reducing the loop tension and a control for this purpose is provided. In this manner compatibility with existing systems can be achieved in a simple way with some sacrifice of foul weather handling capability. Equally it is possible to shackle the flounder plate to existing travelling highpoint units. In this case it is obviously not necessary to make use of the tension control as a means of lowering the load on to the deck. Indeed, this arrangement is almost as effective as a means of lowering the load on to the deck as a geometrically designed arm. The only disadvantage is that the lever arm effect cannot be entirely eliminated and a certain relative motion between load and deck, as a result of ship motion, is therefore inevitable.

OTHER APPLICATIONS

As already made clear the basic naval R.A.S. system outlined in this paper has potential for application to a number of similar marine handling problems. Several of these have been mentioned but they may be considered together as follows:—

(*a*) The replenishment of off-shore oil and gas drilling rigs.

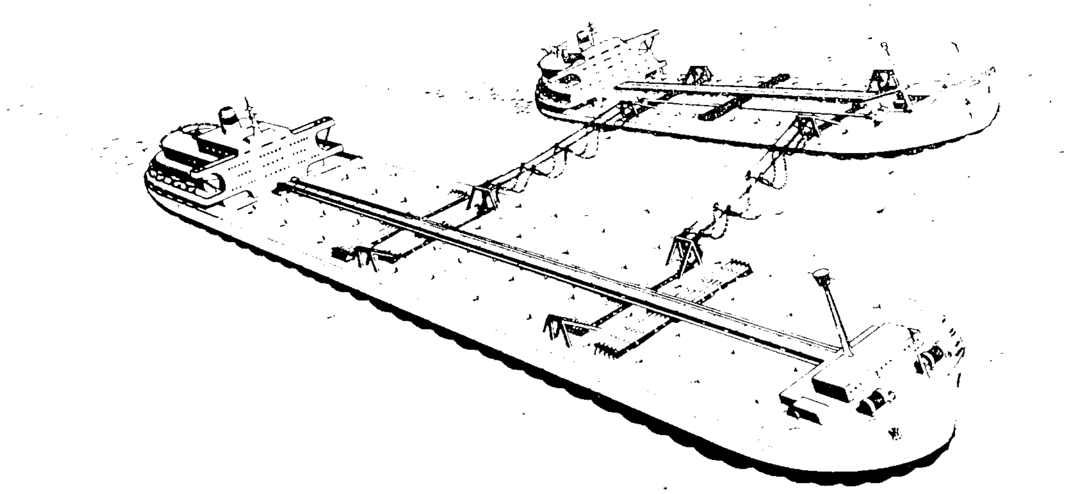

FIG. 11

(*b*) The replenishment of lighthouses and lightships.

(*c*) The transfer of crude oil from super tankers to smaller tankers while both are underway.

This last possibility is sketched in FIG. 11 and is particularly interesting because of its economic implications. The availability of the super tanker is appreciably increased and there is the possibility that major reconstruction of tanker terminals to take the large vessels now appearing could in certain cases be avoided. It will be seen that the super tanker is provided with a helicopter platform aft for crew change and, of course, is itself replenished using the same equipment employed to transfer its cargo. In this way it is only necessary for the super tanker to come into port when repairs and maintenance are necessary and the distribution of its massive cargo is conveniently undertaken by a fleet of smaller vessels serving a number of oil terminals.

CONCLUSION

Development of an improved design of naval R.A.S. system has now reached the point where the construction of prototype hardware of the type shown in FIG. 1 is under way. The system combines mechanical, hydraulic and electronic control elements to achieve the standards of performance demanded by the Royal Navy. It is expected that the prototype system will be at sea in the summer of 1970.

So far its development has presented a fascinating technical challenge and it is hoped that it will soon become possible to start on the development of the non-naval designs.

Acknowledgement

Development to date of the new G.E.C. naval replenishment-at-sea system has been undertaken as a joint effort with Ship Department M.O.D. (N), Bath. Acknowledgement is hereby made of the enthusiastic help and encouragement received at all levels in Ship Department and G.E.C. (Process Engineering) Ltd. without which progress to date would not have been possible.

References:

(1) 'Principles of Naval Architecture'—J. P. Comstock S.N.A.M.E. Publication 1967.

(2) *Ocean Wave Statistics*—Hogben and Lumb H.M. Stationery Office 1968.

(3) 'Replenishment-at-Sea'—Stephen Weintraub. *Naval Engineers Journal,* U.S.A., 1967.