HYDROSTATIC TRANSMISSIONS

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PART III

APPLICATIONS

The final part of this article describes some marine applications of hydrostatic transmissions.

Cranes

One of the best applications of hydrostatic transmissions is in cranes. They give the following advantages compared with electrical drives for the hoist motion:

(i) Infinitely variable speed from zero to maximum hoist and lower
(ii) Smaller size, reduced inertia
(iii) Protection against abuse and overloading due to force limit from the pressure relief valve
(iv) Lower first cost
(v) Reduced maintenance.

Messrs. Stothert and Pitt Ltd. have incorporated electrically driven displacement controlled hydrostatic transmission in their 'Stevedore' range of deck cranes for the hoist motion and throttled controlled hydraulic actuation for the

FIG. 1—STEVEDORE CRANE AT WORK ON M.V. 'SALERNO'
(*Courtesy of Stothert and Pitt Ltd.*)

luffing and slewing motions. FIG. 1 shows a battery of these cranes at work on the M.V. *Salerno*.

FIG. 2 shows the circuit diagram for the hoist hydrostatic transmission. It will be seen that this is a boosted circuit, allowing the system to regenerate. This is obviously necessary in the case of a crane since it is a classic example of positive and negative power cycles.

FIG. 3 shows the layout of the hydraulics. A Plessey 'Hydrostabil' tilting head unit is used for the hoist motion. It can be seen that the whole system makes a neat package and is readily accessible for maintenance.

The variable displacement pump is submerged in the fluid tank and is connected by the piping (which can be seen in FIG. 3) to a fixed displacement motor which drives the hoist drum through reduction gearing.

Safety features need to be incorporated to cover the possibility of hydraulic failure. The circuit diagram shows a brake ram which is urged on mechanically and is held off hydraulically. The brake can be seen in FIG. 3 on the opposite side of the gearbox from the hoist motor.

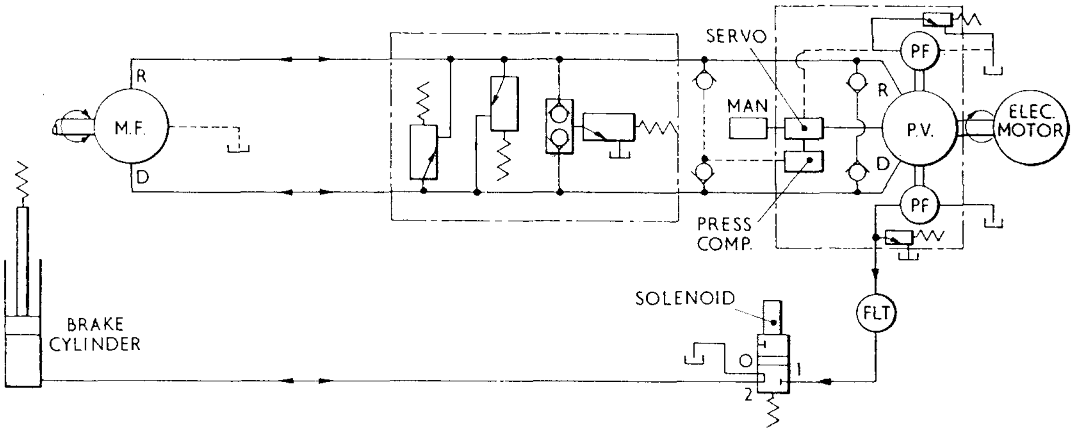


FIG. 2—HYDRAULIC CIRCUIT FOR CRANE HOIST MOTION
(*Courtesy of Stothert and Pitt Ltd.*)

The hoist control is linked to a servo cam mounted on the side of the hydraulic reservoir. This cam moves a pilot spool which unbalances a follow-up piston in the servo motor, causing it to duplicate the movements of the pilot spool. The servo motor is linked to the pump by a rack and pinion drive.

FIG. 4 shows the power envelope for a five ton 'Stevedore' crane. The limit on lowering is the corner horsepower of the transmission. On hoisting, the limit is the power available from the prime mover. This is shown as a hyperbola on the hoisting power envelope.

To ensure that the hoist system doesn't attempt to operate outside the specified power limit, an overriding power limiting control is fitted. An integral

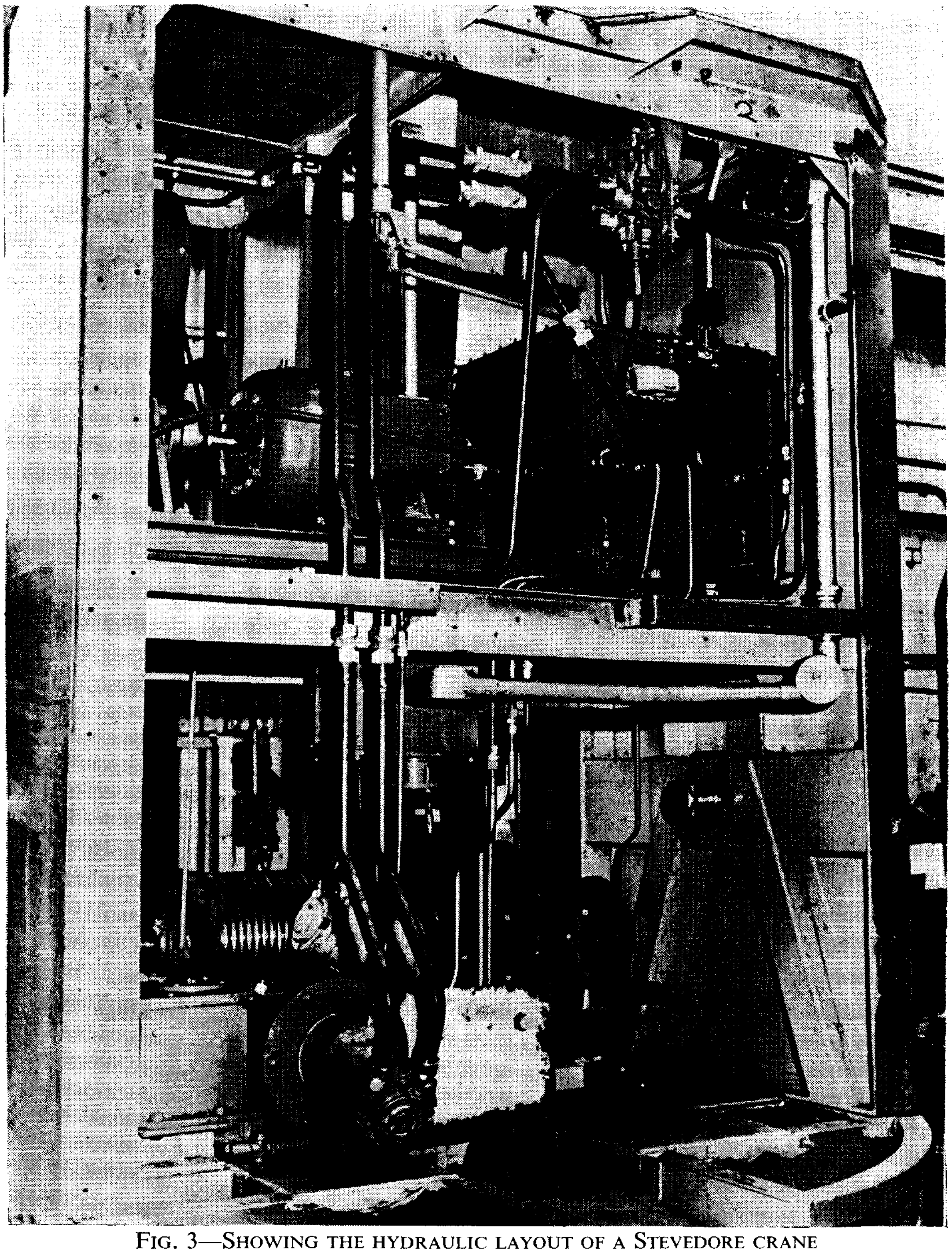


FIG. 3—SHOWING THE HYDRAULIC LAYOUT OF A STEVEDORE CRANE
(*Courtesy of Stothert and Pitt Ltd.*)

power controller overrides the servo cam and modifies the servo motor movement so as to give reduced pump output as increased delivery pressure is felt by the power controller. The power controller has a nest of three springs which balance delivery pressure acting on a piston. The characteristic deflections of the three springs when plotted as pressure against delivery give three intersecting straight lines which approximate very closely to the constant horsepower hoisting performance curve (FIG. 4).

A pressure cut-off or stall valve is integral with the power limiter. This can be set to a pressure equivalent to a nominal percentage above safe working load. The valve admits main pressure to a second area of the power controller piston

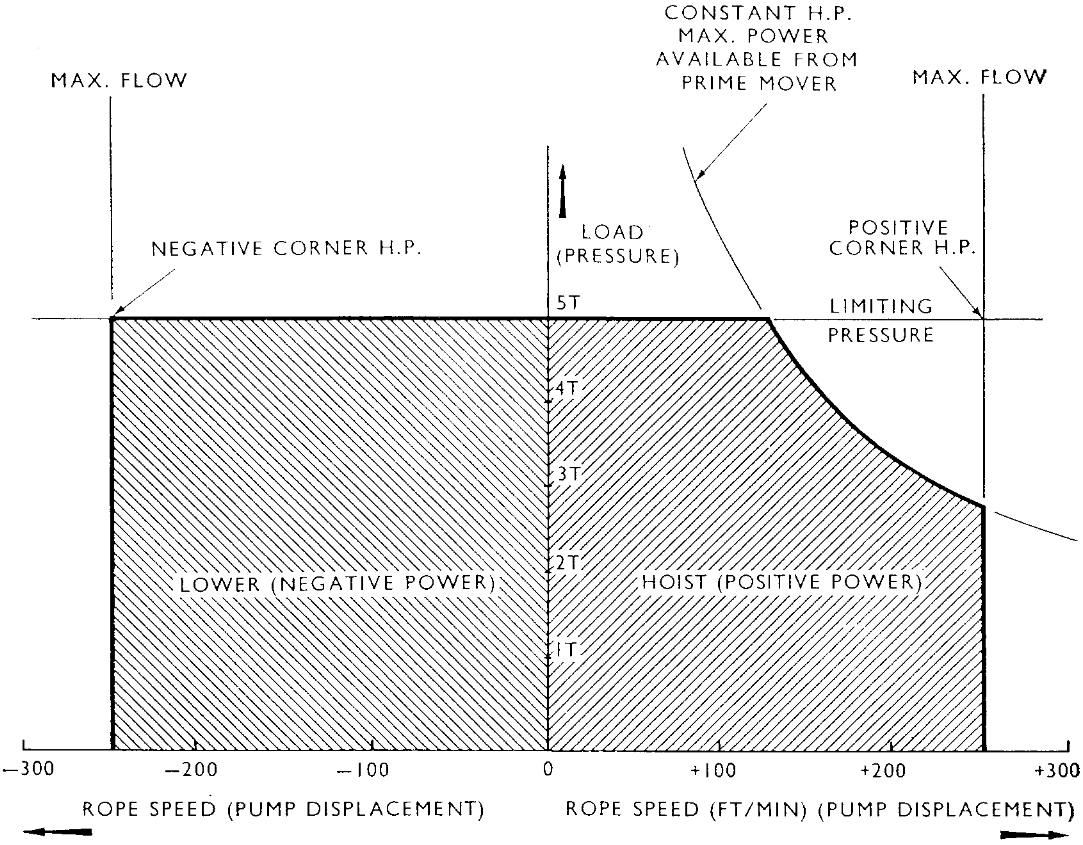


FIG. 4—PERFORMANCE CURVE FOR 5-TON DECK CRANE

so as to cause it to complete its full stroke and hold the servo follow-up piston in the position corresponding to zero pump delivery.

Deck Machinery

Hydrostatic transmissions are already established for the actuation of many kinds of deck machinery and we can expect further expansion in this field.

Messrs. Boulton-Paul Aircraft Ltd. have developed a series of power packs for a variety of deck machinery drives. One such power pack is shown in FIG. 5(*a*) and the circuit diagram is shown in FIG. 5(*b*). This power pack supplies up to the hydraulic motors which drive the slewing, hoisting and topping (luffing) motions of a derrick for a coastal vessel. Part of the system can be switched to operate the anchor winch.

The hydraulics are powered by a 100 h.p. Diesel. There are three variable displacement pumps: one $7\frac{1}{2}$ cu in./rev and two $3\frac{1}{4}$ cu in./rev. It can be appreciated from FIG. 5(*a*) what a neat package this hydraulic system presents.

Bow Thrusters

Messrs. Stones Manganese Marine Ltd. have recently added a hydrostatic bow thruster to their range. These bow thrusters give an infinitely variable lateral thrust of up to five tons either to port or starboard. A photograph of one of these units appears in FIG. 6.

A constant speed unidirectional prime mover drives a variable displacement pump. The pump is a Lucas IP3000 axial piston swash plate unit. The pump supplies fluid to a Ruston ten-cylinder high torque slow speed motor, the normal pump input speed being 1800 r.p.m. and the propeller speed from zero to plus or minus 450 r.p.m. The maximum system pressure is 2500 lb/sq in. The fluid connection is through a vertical strut to the motor which is located in the pod. The transmission has a boosted closed circuit and is cross-connected through the relief valve.

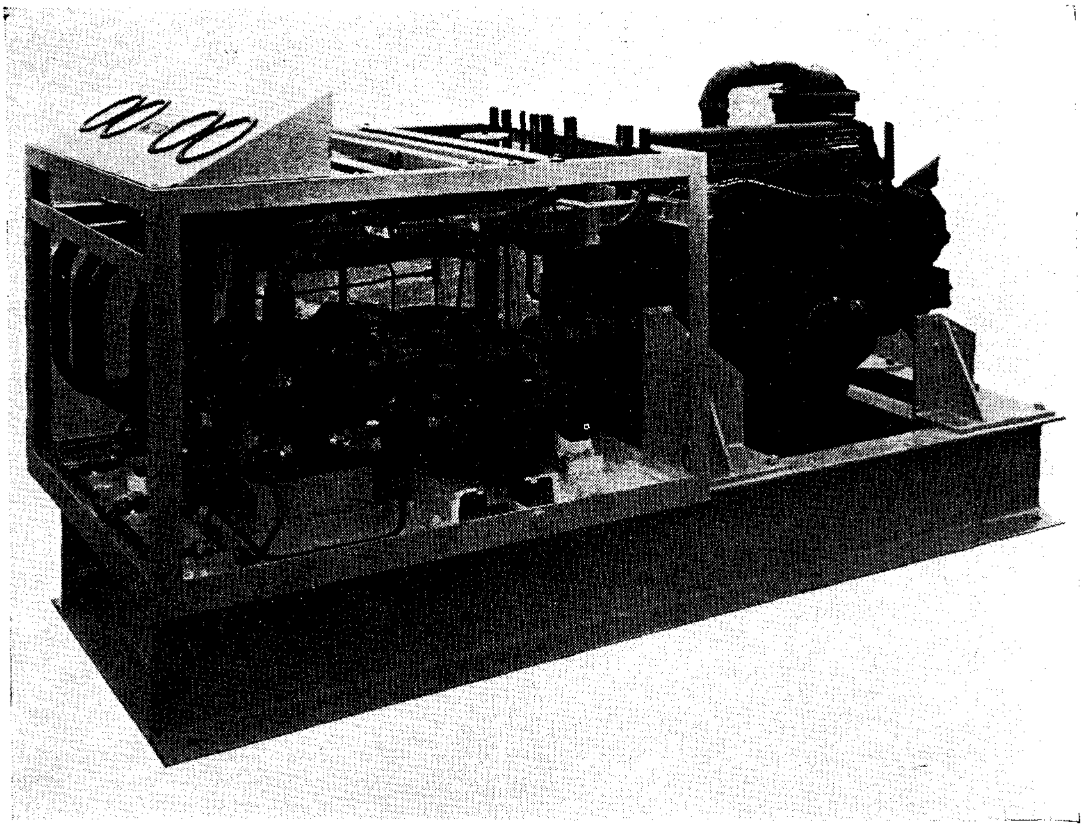


FIG. 5(*a*)

The propeller is fixed pitch and runs in a steel tunnel. The pod is suspended in the tunnel which is machined in way of the blade tips to give small clearances for maximum efficiencies. Protection of the propeller from the ingress of foreign objects is inherent in the hydraulic relief valve.

Control of the propeller speed is facilitated by means of a follow-up servo actuating the pump swash. The control lever can be arranged to be local and/or bridge-operated.

FIG. 7 shows a cutaway diagram of the system.

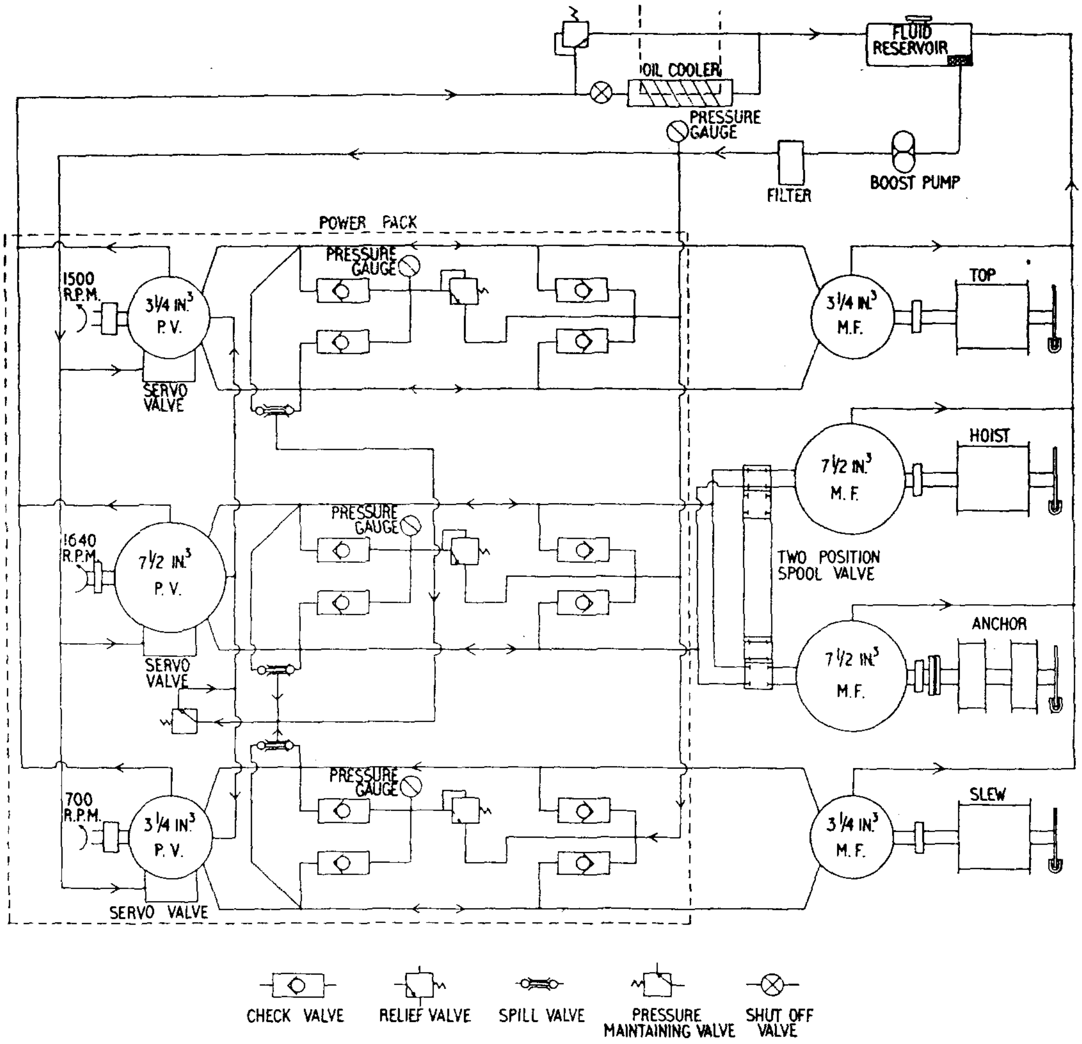
Stones M. M. Ltd. produce mechanically-driven bow thrusters with controllable-pitch propellers in the larger thrust ranges.

Active Rudders

In 1961 the Admiralty asked the National Engineering Laboratory if their hydrostatic transmission could be installed to drive the active rudders of a coastal minesweeper. N.E.L. proposed an arrangement using variable speed (ahead and astern) hydrostatic transmissions with controls to match pump displacements to engine speeds, and, interconnection of the two systems to enable one pump to drive both motors for low power requirements.

The Admiralty, however, against the advice of N.E.L., decided to fit a pair of transmissions which used fixed displacement pumps and motors with cross-over valves to control direction, and with speed controlled by varying engine speed. The reason for this decision was to give the same control method as that for H.M.S. *Kirkliston* which had been fitted with a pair of mechanical gear-driven active rudders, and comparisons between the handling of these two vessels were to be made. The vessel chosen for this test was H.M.S. *Weston*.

This was certainly not the ideal form of transmission and, in fact, embodies only one of the advantages of hydrostatics (actuation remote from the prime mover) with all the disadvantages plus one extra one with the introduction of a cross-over valve.



HYDRAULIC CIRCUIT

FIG. 5(*b*)

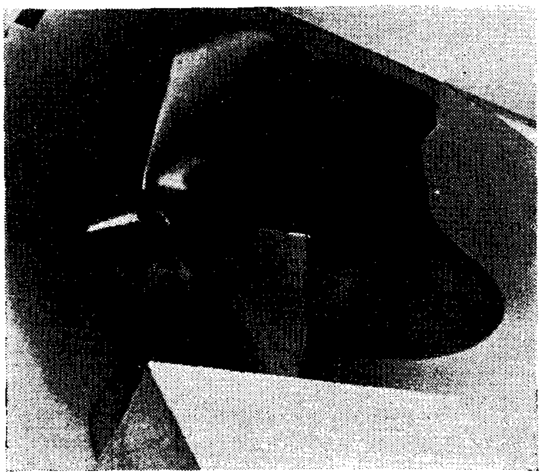


FIG. 6—S.M.M. HYDRAULIC BOW THRUSTER

The layout eventually adopted was that shown in FIG. 8 and a photograph of the unit is shown in FIG. 9.

The prime mover was a Thornycroft Diesel developing 150 h.p. at 1800 r.p.m. A fixed displacement N.E.L. designed pump of 20 cu in/rev (nom.) capacity feeds fluid to the pintle head of the swivelling stalk, the discharge and return fluid being passed co-axially down the stalk into the pod. Reversing is achieved by means of the cross-over valve which can be seen at the top of the stalk. Steering the unit is achieved by electrically driving the tiller, which can be seen below the cross-over valve. This rotates the whole stalk with the pintle head and valve fixed. An N.E.L. designed motor of 32 cu in/rev (nom.) capacity is anchored in the pod. This gives a reduction ratio of 1·74/1 on full load.

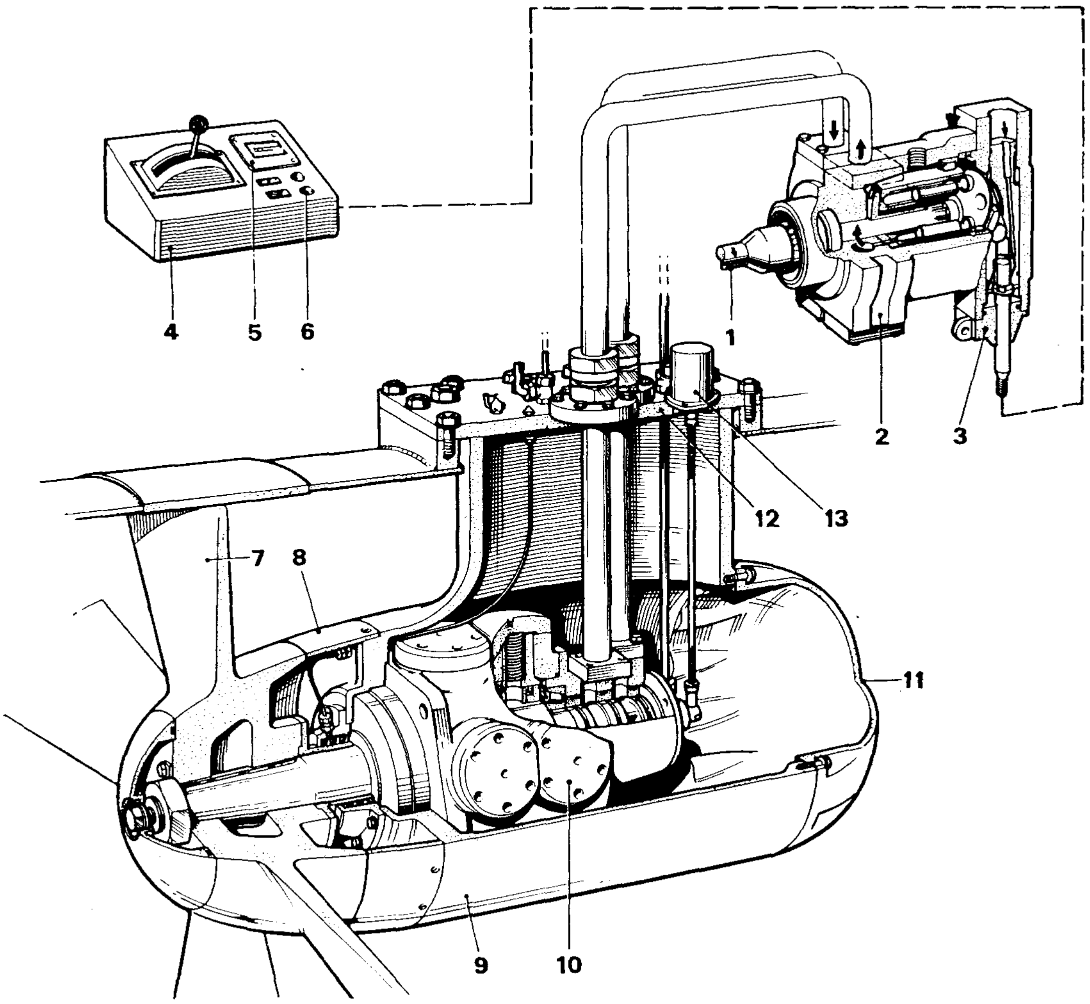


FIG. 7—DIAGRAMMATIC SKETCH OF S.M.M. BOW THRUSTER

1. Prime mover output shaft
2. Variable delivery pump
3. Servo valve assembly
4. Bridge control unit
5. Thrust indicator
6. Running lights
7. Propeller
8. Fairing cover
9. Main casing
10. Hydraulic motor
11. End cover
12. Mounting plate
13. Tacho generator

The propeller was a three-bladed shrouded design. The propeller seems to have been designed to absorb 150 h.p. at 920 r.p.m. Since the engine gave 140 h.p. at 1600 r.p.m., which was the engine speed to give 920 r.p.m. at the propeller, it is clear that the design point could not be reached.

Due to production delays, the shore testing of the units at N.E.L. was considerably protracted. The efficiency testing was done using an electrical dynamometer. Additional tests, with the rudder pod immersed below the water line in a large tank to simulate installed conditions as near as possible, were carried out. The propeller was not fitted for these tests; a paddle was fitted and consequently no thrust was generated.

Owing to limitations of the rig, the testing was limited to 130 h.p. The efficiencies measured could not be extrapolated with any certainty to deduce the efficiency at the design condition. It was, however, obvious that to meet the design condition, an input horsepower of around 200 would be needed.

N.E.L. recommended the use of Shell Tellus 27 oil or its equivalent as a result of their testing.

The units left N.E.L. in January, 1963, but installation was not completed

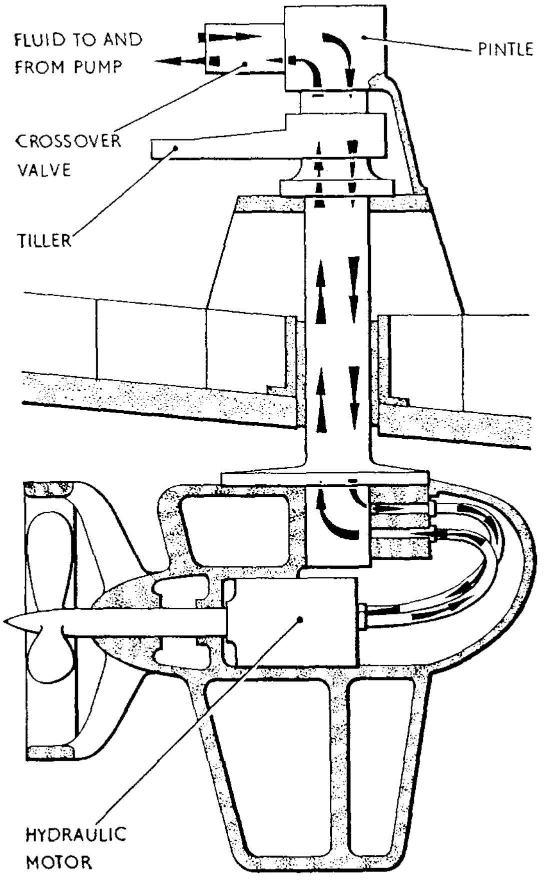


FIG. 8—SIMPLIFIED DIAGRAM OF ACTIVE RUDDER

until September, 1964. Clearly the units must have spent a lot of time lying around the dockyard. This time could well have been used for further development at N.E.L.

During basin trials at Devonport Dockyard, a run up through the speed range revealed that 1000 (engine) r.p.m. was the maximum obtainable. This condition was checked and found to be the same on both engines. Clearly the load was holding the engines down to this speed and this was the result of the propeller resistance and low transmission efficiency. A sea trial was called for after the basin trial since it is generally agreed that propellers require more power in basin trials than at sea.

The sea trial was carried out off Plymouth and little improvement was found over the basin trial. Over the measured mile, a maximum speed of 3 knots was attained with both rudders driving. After boosting with main drive and then changing to the active rudders, a speed of 5 knots was recorded. This was attributed to the fluid temperature rising to 137 degrees F.

For the second day's trial, the original oil (OM65) was changed to OM33 to give the viscosity conditions equivalent to Shell Tellus 27 (as recommended after shore trials). A mean speed of 5·8 knots was recorded with a maximum of 6·4 knots.

H.M.S. *Kirkliston*, which had been fitted with its gear-driven active rudders (manufactured by Transport Equipment Thornycrofts Ltd.), had attained a mean speed of 8 knots and apparently handled very well.

The difference here between the speeds of the two boats is indicative of the extremely low efficiency of the particular hydraulic system fitted to *Iveston*.

The main reasons for the exceptionally low efficiencies were thought to be due to the sharp pipe bends and the design of the cross-over value. The pipework installation on board was much more tortuous than that used in the shore trials.

In addition to the efficiency problems, there were sundry mechanical faults and the units were finally removed when sea water penetrated the pod cover seals and rendered the units inoperable.

The lessons to be learnt from *Iveston* are obvious and fundamental to the application of hydrostatic transmissions. These lessons have been heeded and embodied in a more recent minesweeper conversion.

Launch Tug

In contrast to the *Iveston* story, a 45 ft Launch Tug built by Messrs. W. R. Cunis Ltd. of London is a hydrostatic success story. This enterprising firm is the first to put a tug on the Thames with a modern hydrostatic transmission. FIG. 10 shows 'Hycu' during manoeuvring trials.

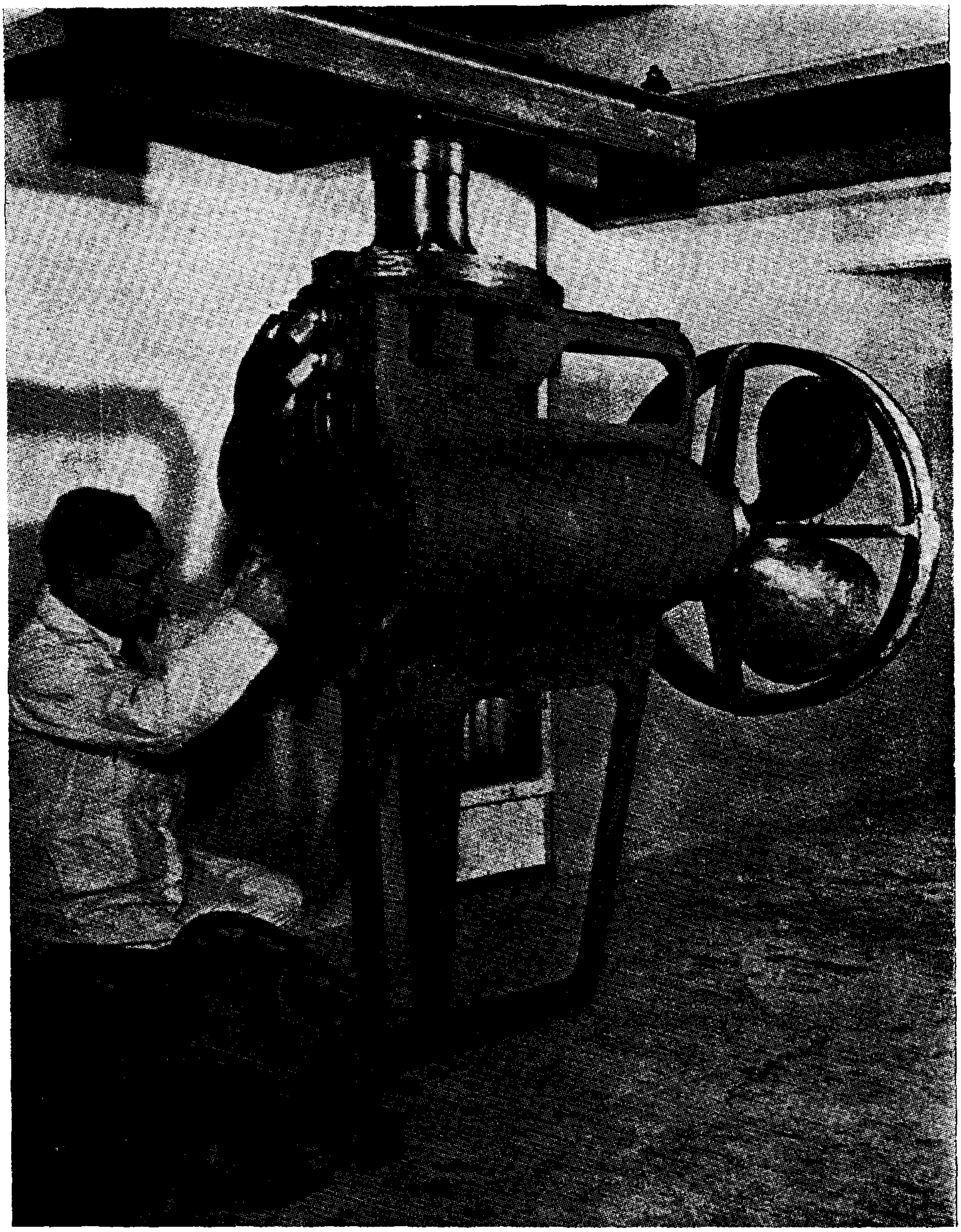


FIG. 9—ACTIVE RUDDER—MOTOR INSTALLATION IN RUDDER POD

A Thornycroft T400 Diesel drives a Dowty Dowmatic variable displacement pump at 1900 (max.) r.p.m. The pump feeds a Staffa radial slow-speed motor. The Staffa motor is coupled directly to a short intermediate shaft, carrying the ahead and astern thrust collar and this shaft is coupled directly to the tailshaft; the maximum shaft speed being 280 r.p.m. The propeller is 53 in. diameter and was specially designed by Bruntons (Sudbury 1919) Ltd.

Single lever bridge control operates both the pump displacement servo control and the speed control of the Diesel engine through a combined lever and cam system, which can be appropriately designed to give any engine speed related to propeller speed. In this tug, the design is arranged to raise the engine speed and propeller speed until a propeller speed of 220 r.p.m. is achieved at

FIG. 10—'HYCU' DURING TRIALS
(*Courtesy of W. R. Cunis Ltd.*)

full engine speed. The engine speed remains constant but by further operation of the pump servo control, as the bridge lever is moved, the propeller speed can be increased to 280 r.p.m.

A power control device (patent applied for) has been installed so that the propeller speed is reduced under heavy pull conditions overriding the bridge control lever position so that overload is prevented under heavy tow conditions.

The design was based on a maximum horsepower of 95 and at a propeller speed of 230 r.p.m., a bollard pull of 31 cwt was registered. When towing four barges the propeller speed was 250 r.p.m. and when running light, 280 r.p.m., giving a free speed of 8 knots.

The most efficient way of using power in a craft of this nature, which experiences a wide range of loads and speeds, often with rapidly changing load cycles, is a controllable-pitch propeller with the pitch programmed to match the prime-mover characteristics over the speed and load range. However, for small craft, this type of installation is, in proportion, extremely costly and complex. A good and cheap second best is to use a fixed-pitch propeller and drive it through a hydrostatic transmission. The inefficiencies of the hydrostatic system compared with a conventional drive are overcome by the ability to match engine power to shaft speeds and torques.

Other advantages for this type of craft are:

(*a*) Should a wire rope or other obstruction become entangled with the propeller, the relief valve will protect the system from damage.

(*b*) Shafting/engine alignment problems are eliminated. The engine and pump can be put in any desired position.

Other interesting features of this tug are:

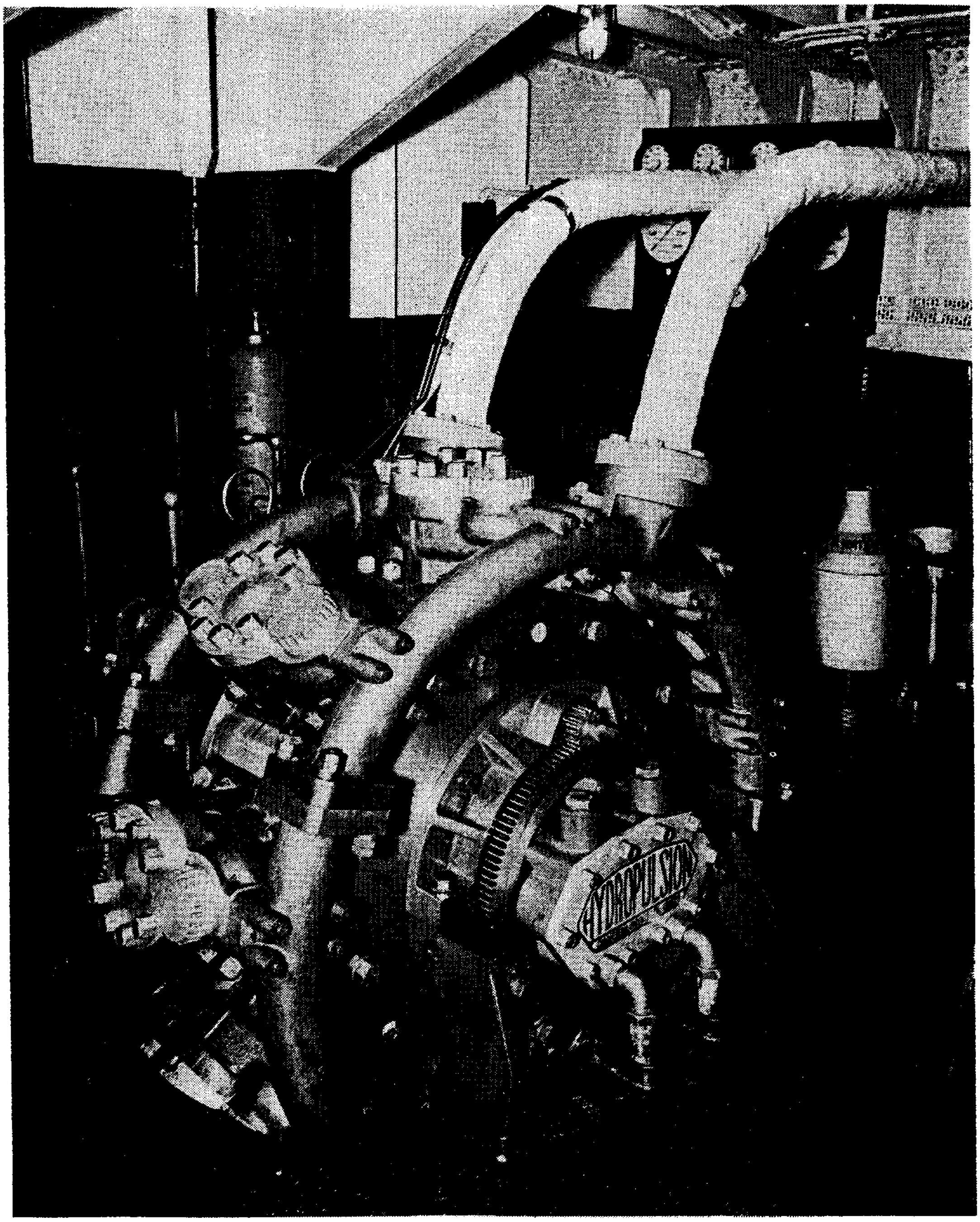


FIG. 11—HYDRAULIC MOTOR IN *Tom Jay*

(*a*) The wheelhouse can be raised and lowered by 3 ft 6 in. in half a minute hydraulically by a lever in the wheelhouse while the tug is under way. This is particularly convenient when towing light barges or operating in creeks.

(*b*) A specially designed tow hook is fitted which greatly reduces the possibility of the tug being rolled over if the tow rope lead is on the beam of the tug.

Prior to the development of the tug, Messrs. Cunis successfully converted a number of steam cranes to Diesel hydraulic drive.

This company now owns what is believed to be the first hydraulic tug, the *Tom Jay*. *Tom Jay* was built in the late 1940s for the Thames Steam Tug and

Lighterage Company, the hydraulic propulsion being designed by Messrs. G. F. Jones and M. G. Petty.

The system consisted of four Gardner 6L3 Diesels, each driving a radial pump directly and each pump feeding into a common main and thus into a radial hydraulic motor (FIG. 11). All units being of fixed displacement, speed control is effected by varying engine speed and direction changed by means of a cross-over valve. Although this would seem to us now to be a rather crude system, an important feature of this tug is that it was designed to operate on three only of the four engines, the fourth being available for maintenance. This has resulted in this tug having a remarkable record of availability.

Concluding Remarks

All the applications described above employ straight hydrostatic transmissions. There is, however, a growing interest in various types of hydro-mechanical transmissions. Although these transmissions are outside the scope of this article, it is worth digressing in order to describe briefly their main features.

A desire to combine the high efficiency of geared transmissions with the flexibility of hydrostatic systems has led to the development of a wide variety of hydro-mechanical transmissions. The simplest form of hydro-mechanical transmission is the single-stage hydro-differential transmission. This consists of a single stage epicyclic train with the input (sunwheel shaft) coupled hydrostatically to the annulus. By varying the hydrostatic ratio, the annulus can be rotated in one direction or the other or can be held stalled, thus providing some variation in the ratio of output to input speed.

It will be appreciated that to keep fairly high efficiencies, the proportion of the power transmitted hydraulically must be kept low. It will also be appreciated that the degree of speed variation is a function of the ratio of hydraulic to mechanical power transmitted. Thus, in this simple type of system, the transmission is generally arranged to give a small variation of output speed, but maintains efficiencies well over 90 per cent.

There are many variations on this theme, more complex than that described, which attempt to extend the speed range and yet maintain high efficiencies over at least parts of the speed range.

A simple hydro-differential transmission could be driven from a ship's main gearing to provide an efficient constant speed take-off over the range from (say) half full-power speed to full-power speed. This take-off would be useful for driving auxiliaries whose speed requirements do not fall off significantly with reduced power. If, for example, the main lubricating oil pump was driven in this way, the motor-driven pump would not have to cut in at speeds above half full power; thus reducing the electrical load over this power range.

Messrs. W. H. Allens, Sons & Co. Ltd. of Pershore have produced hydro-differential drives by combining hydrostatic transmission with their Stoeckicht epicyclics. In a 2400 h.p. application to drive a boiler feed pump at output speed of 4,050 to 4,500 r.p.m., efficiencies above 98 per cent are achieved, the hydraulic power transmitted being 107 to 133 h.p.

In the future, we can expect hydrostatics to take over a lot of the variable speed functions in ships which are now performed by electrical drives. Straight hydrostatic transmissions should grow in size, but for the higher powers, hydro-mechanical transmissions would be expected to play a more important role.

Finally, the authors would like to acknowledge the assistance and information given by the following: Mr. P. Cockell of Stothert and Pitt Ltd., Mr. D. Firth of the National Engineering Laboratory, Mr. H. Cedric Cunis of W. R. Cunis Ltd., Mr. Whiteman of Transport Equipment (Thornycroft) Ltd., Mr. J. Steer of Boulton-Paul Aircraft Ltd., and Mr. B. Black of Stones Manganese Marine Ltd.