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Mr. Joseph

Mr. Rogers

HYDRAULIC OPERATION AND POSITION INDICATION OF CARGO AND BALLAST VALVES ON LIQUID CARGO SHIPS

R. A. Joseph, B.Sc., F.I.E.E., F.I.Mar.E.*, and
J. A. Rogers, M.I.Mar.E.†

It is important to consider the equipment used in controlling liquid cargoes as a *system* and not as a series of components. The essential characteristics of such a system are reliability, simplicity of operation and minimum overall cost. The latter must include first cost, maintenance costs and loss of ship operating time caused by failures.

After detailing the types of cargo and ballast valves used, the hydraulic actuators suited to each are described under the broad headings of piston and hydraulic-motor types. The effective coupling of actuator to valve is studied, including the matching of their characteristics and protection from overstressing of valve components. A full description is given of a device which programmes the speed and the torque output of a motor driven actuator in response to a feed-back of information concerning the cargo valve position.

Indication of valve position can be given locally and remotely. In the latter case the signals can be electrical, hydraulic or pneumatic. Facilities must generally be available for emergency operation. Energy storage, handwheels and hand pumps are used for this purpose. The paper concludes with notes on hydraulic power units and control equipment employed in the hydraulic circuits.

INTRODUCTION

Cargo ships are intended to carry cargo from port to port. Time spent loading and discharging is time and money wasted. The pump, pipeline and valve systems of liquid cargo ships must be designed to permit the fastest possible turn-round with the minimum use of labour.

These systems are also used during ballasting and tank cleaning when ease and convenience of operation and control is equally important. Power operation is employed for an increasing proportion of valves on crude oil carriers and refined product carriers for one or both of two reasons:

a because the sizes of the valves, and the hydraulic pressure drop across them during pumping, are such that the power required to open and close them is beyond manual capabilities, and

b because control from a remote central console (or consoles) is required for convenience and saving of manpower.

As an example of the first of these reasons, on a typical 356 000 dwt crude carrier having a pumping pressure of 17 bars, it would take at least four hours to operate one of the 700 mm tank valves by hand, the reason being that the operating thrust is such that a very high gear ratio would be needed to reduce the handwheel effort to an acceptable figure.

ESSENTIAL CHARACTERISTICS OF THE SYSTEM

Reliability

No system can ever be 100 per cent reliable for its whole life, but it is most important that the operator can have complete confidence that the order he gives to the actuator will be obeyed and that all automatic operating and protective features will function. The designer must work with this fact uppermost in his mind.

Simplicity of operation

The sequence of human operations must be simple and well within the mental and physical capabilities of the men concerned. The number of men required to work the system must not be greater than the number readily available from the ship's staff at loading and unloading times.

Minimum overall cost

The important word is "overall". The cost components include:

initial cost of equipment;
its installation, including pipework and labour;
maintenance costs; and
ship "down time" caused by cargo system failure.

The first and possibly the second, can be accurately forecast. Any skimping of quality to reduce initial cost can have very substantial effects on the third and fourth. An enforced stay in port, or worse still, in dry dock, can run up vast costs when interest on idle capital and staff salaries are included.

It is obvious that a liquid cargo carrier which cannot be loaded or discharged is a singularly useless piece of equipment, and this must be remembered when the cargo system components are chosen.

Co-ordination of system sections

It is not sufficient to have the best possible cargo valves, actuators, hydraulic power unit and control components. These components must match together to make a co-ordinated system. Failure to ensure this co-ordination can result *inter alia* in inadequate remote control, shortage of hydraulic power and,

* Managing Director, Hydraulics and Pneumatics Ltd.
† Chief Engineer, Hydraulics and Pneumatics Ltd.

worst of all, damage to valves or to actuators due to failure to match their operating characteristics.

CARGO AND BALLAST VALVES

The types principally employed and the actuators used to operate them on modern tankers are:

Wedge gate	:	Linear and rotary
Bulkhead	:	Linear and rotary
Butterfly	:	Semi-rotary and rotary
Ball	:	Semi-rotary and rotary
Non-return	:	Linear and rotary

In general a valve requires a greater effort to open than to close. This is particularly true of the wedge gate where the wedge is driven into its seat until it is sealed. As the sides of the seat grip the gate an excess of force is required to break it out again.

Butterfly, ball and bulkhead valves show this characteristic to a lesser extent. In the closing operation it is essential that the moving element is able to travel far enough to ensure tight sealing. The wedge gate travels until the valve resists the closing thrust exerted by the actuator. In the butterfly valve there is normally some positive mechanical limit to travel set so that the paddle is then in its fully sealed position. Before reaching this point, the seat resistance may have been allowed to stall the actuator.

The ball and bulkhead valves rely primarily on accurate positioning of the moving element to seal tightly and this is achieved with mechanical stops.

To stop an actuator in the fully open position of the valve, either a positive mechanical stop is used or the hydraulic power supply is cut off at a predetermined position in the valve travel. The mechanical stops may be incorporated in the valve itself, in a gearbox mounted on the valve, or in the actuator. Some degree of adjustment during commissioning is generally required.

HYDRAULIC ACTUATORS: CONSTRUCTION AND CHARACTERISTICS

This paper deals with power operation by hydraulic means. Most of the features and characteristics of the actuators and their associated control equipment apply equally to pneumatic equipment. The main factor influencing the choice of hydraulic, rather than pneumatic, systems is the increased effort available for a given size of actuator. Hydraulic actuators are normally used at pressures between 69 bars and 138 bars whereas the pneumatic units are seldom worked above 7 bars.

Linear actuators

These are simple piston and cylinder units. Design points to be considered include:

The seals: The seal materials chosen must be compatible with the hydraulic fluid used. For this reason the type of fluid should not be changed without reference to the recommendations of the makers of the actuators and other components. The piston will normally be fitted with a double acting hydraulic seal or, alternatively, a P.T.F.E. composite seal for reduced friction.

Sometimes the actuator is mounted directly on to a valve which does not have a top seal, i.e. the actuator bottom seal is the valve top seal. This seal must then be compatible with whatever cargo or ballast may be carried. In refined product carriers this can present a problem. Such vessels should certainly have an operating life of 20 years and it is not easy, at the time of build, to specify every cargo that may be carried during that period. Fortunately many modern seal materials have a very wide range of resistance. However, it is always wise to mount, in this instance, an actuator having a twin seal arrangement and a vent between seals. This will ensure that should one or other seal fail at any time there is no fear of contamination of:

a) Product or
b) Hydraulic fluid.

The piston rod: This will come into contact with the hydraulic fluid and whatever may be below the lower end cap. At that point there may be a spacer or mounting stool between the actuator and valve or the two may be close coupled.

If there is a spacer (stool), the piston rod may be exposed to local atmosphere unless the spacer is enclosed and oil filled. Long spacers between valve and actuator are usually used on Product Carriers to ensure that there is a vent between products carried and the system hydraulic fluid to avoid contamination should an oil seal fail.

The corrosive effects of sea water are obvious enough. Carbon steel is eaten away almost visibly. Chrome plating, which gives a hard surface resistant to seal wear, is porous and permits corrosion of the base metal and resultant blistering and flaking of the chrome. This in turn causes damage to the seals as the blisters foul the seal lips and hence allows oil leakage from the actuator. The porosity of chrome plating can be reduced by the use of a suitable intermediate plating treatment before the chrome is applied. Unfortunately the best of these intermediate materials are rapidly attacked by hydrogen sulphide which is present in many crude oils. An obvious material to resist corrosion by sea water, crude oil and petroleum products, is stainless steel. A number of different types is available. None is completely corrosion resistant but the most resistant is EN 58J. Unfortunately this has a relatively low ultimate tensile strength—approximately 49 kgf/mm^2 (31 tons/in.^{2}). Both EN56 and EN57 have higher UT strengths of the order of 63-71 kgf/mm^2 (40-45 tons/in^2) but their corrosion resistance is much lower.

High nickel alloys are available which have corrosion resistance substantially greater than that of EN 58J. The cost ratios are even greater than for stainless steel. In these alloys both high corrosion resistance and high tensile strength can be combined in one material. This is the most expensive of all!

There is no one answer to this problem of combining strength and corrosion resistance.

If the rod diameter is high in proportion to the maximum load, the material strength specification can be relaxed. If the piston rod, for design reasons, must be highly stressed, then the problem becomes more acute. The choice of material in any one case must depend on the fluids which will contact the rod surface, the strength requirements and the ease with which the rod can be replaced. Each decision must be made in the light of these considerations and of the cost factors involved. Recent experiments with materials previously considered completely unconventional for such applications, have proved most encouraging. A substantial number of piston rods of this new type have now been in use at sea for periods of up to eighteen months and more are being supplied. There is good reason to believe that an economic answer to this difficult problem has at last been obtained.

Operating thrusts

Gate valve actuators are normally designed to close the valve with rod extended. Bulkhead valve actuators are normally inverted and operate through a bridge piece so that the rod is retracted in the normal closed position thus reducing the amount of time during which it is exposed to the tank contents and corrosive elements. Piston actuators may be operated by various hydraulic systems.

Double acting: Pressure is applied to one side only of the piston and the other side is allowed to exhaust to tank. The ratio of thrust when the rod is extending to that when it is retracting, is that of the full piston area to the area of the annulus (i.e. cylinder area minus piston rod area). If the actuator is operating a wedge gate valve, the closing thrust will be greater than that of opening if the same pressure is applied in each case. This is unacceptable and so a higher pressure must be used for opening.

$$\text{Opening thrust } To = Po \times (Ap - Ar)$$
$$\text{Closing thrust } Tc = Pc \times Ap$$

Where Po = Pressure applied to lower side of piston:
Pc = Pressure applied to upper side of piston
Ap = Area of piston
Ar = Area of piston rod

When the actuator design is finalized (i.e. Ap and Ar are settled) then Po and Pc must be chosen so that To/Tc is within acceptable limits. For most valves this will be between 1·25 and 1·5. The two pressures may be obtained either:

by using a power unit with two output pressures and running twin pressure mains to the actuator controls, or:

by using a pressure reducer for each actuator, or for banks of adjacent actuators. The use of pressure reducers gives a more flexible system with different ratios available for different situations, but generally it is more expensive than a twin pressure power unit system.

In any double acting system two control pipes must be taken from the control valve to the actuator. The extent of the pipework involved in double acting systems is illustrated in Fig. 1.

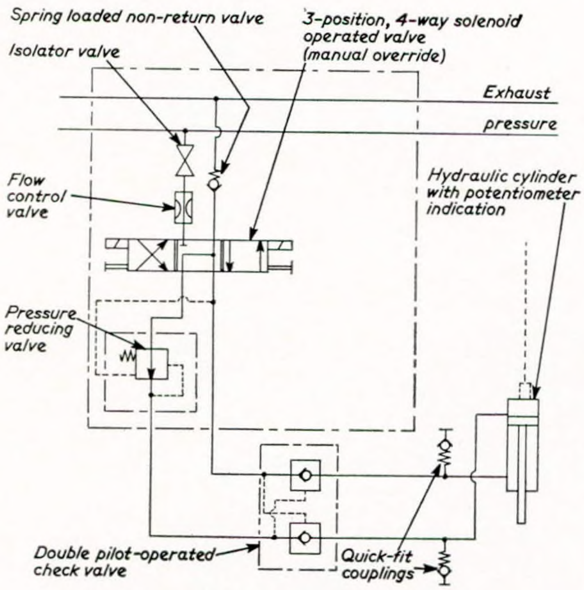

FIG. 1—*Double acting system*

Differential area: The system pressure (P) is continuously applied to the annulus (Fig. 2). A continuous main pipe can be employed with actuator feeds tee'd from it. The full area, or top side of the piston, is then either connected to the system

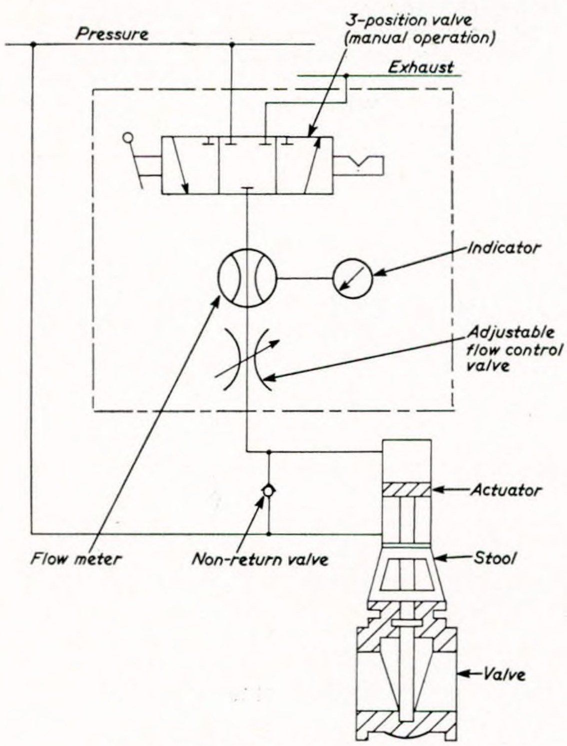

FIG. 2—*Differential area system*

pressure or to tank via a control valve. Only a single pipe is required from control valve to actuator, so piping costs are lower than for double acting.

Opening takes place when the top side is connected to tank, and so:

$$To = Po \times (Ap - Ar)$$

Closing takes place when both sides of the piston are connected to system pressure and the closing thrust depends on the difference in area between the two sides of the piston:

$$Tc = Pc \times (Ap - (Ap - Ar)) = Pc \times Ar$$

So, as $Po = Pc$ in this system

$$\frac{To}{Tc} = \frac{Ap - Ar}{Ar} \text{ i.e. } \frac{\text{Annulus Area}}{\text{Rod Area}}$$

It is a simple matter to select the rod size so that this ratio is in the desired range of 1·25 to 1·5.

Spring return: In this system a spring located on one side of the piston is opposed by system pressure on the other. The stored spring energy can be used to operate the valve if hydraulic pressure fails and the design can be fail-safe to open or fail-safe to close. Spring thrust increases as the spring is compressed and falls as the piston moves away. The design must be such that there is still an acceptable spring thrust when it is in the extended state. The system is efficient in that only one pipe is taken to each actuator, but system pressure must overcome the strength of the spring in its fully compressed state and still leave the necessary thrust to operate the valve. Normally the spring is used to close the valve and so $P \times (Ap - Ar)$ must exceed the strength of the extended spring by a sufficient amount to get the wedge out of its seat. This extended spring strength must be adequate to force the valve into its seat hard enough to seal it.

Utilization of actuator materials: In both double acting and differential area systems, the maximum thrust required in a wedge gate valve actuator is To. In both cases this equals the maximum acceptable system pressure times $(Ap - Ar)$, i.e. the annulus area, but with differential area, the closing thrust

$$Tc = P \times Ar$$

so the rod area must be large enough to ensure that the valve seals. A large rod area means a small annulus area and so the cylinder bore must be larger to give adequate $(Ap - Ar)$:

With double acting, $Tc = Pc \times Ap$.

As the full cylinder area is used for closing, there is no need to have a large area piston rod and consequently there is no need to have a large diameter cylinder.

The result is that a double acting system, with small diameter rods (adequate for mechanical strength requirements) will result in smaller and cheaper actuators than a differential area system requiring large diameter piston rods and consequently large diameter cylinders. The double acting system has smaller (and therefore cheaper) actuators but the resultant cost saving may be offset by the extra cost of the additional control pipe required for each actuator.

It is now generally found that in Pump Rooms where pipe runs are relatively short, a double acting system is cheaper, but that on deck, if long pipe runs are involved, a differential area system may be more economical.

If the corrosive nature of the cargo demands the use of expensive piston rod materials, there will be more attraction in the double acting system where rod diameters are smaller.

The spring return system is also inefficient in the use of actuator materials, because hydraulic effort is absorbed in overcoming the spring, and actuator lengths have to be increased to accommodate the spring discs or a helical spring. These factors are offset by pipe saving and the fail safe feature is attractive in certain circumstances. Generally these actuators are only used on relatively small valves.

Piston operated semi-rotary actuators

These (see Fig.3) are employed to turn butterfly or ball valves, normally through 90°. A piston or pair of opposing pistons causes a toothed rack to move tangentially to a pinion thus turning the pinion shaft which is coupled to, or fits over, the valve spindle. (The use of a hollow actuator shaft allows the fitting of the pinion directly on the valve spindle, reducing the height of the assembly subs tantially. In alternative designs the piston rod moves a link or series of links which turn the output shaft, or a crescent-shaped cylinder and piston with a circular axis provide the necessary turning motion for the output shaft.

If a differential area system is used a single piston can be employed, or pistons and cylinders of different diameters can be fitted on opposite ends of the piston rod. With double acting systems, it is usual to employ two equal diameter pistons and cylinders opposing each other. Butterfly valves do not require

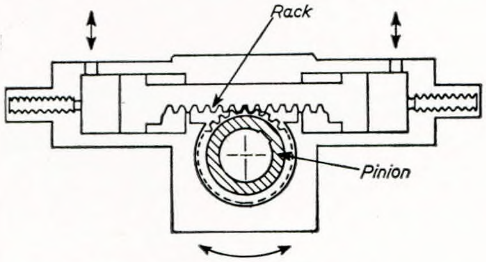

FIG. 3—*Semi-rotary actuator*

the same high ratio of opening to closing effort that is needed by a gate valve, so double acting systems are normally used unless very long pipe runs are involved on deck or unless most of the actuators in the system are of the differential area linear type. In this case the semi-rotary actuators can be double acting and still "fit-in" to the same system without any problem.

Seal problems are similar to those on linear actuators but piston rod problems are easier as the rod is not in contact with cargo or ballast. With rack and pinion actuators it is possible to fit two drive racks, one on each side of the pinion. Each rack has its own cylinder s). A double acting double rack actuator with four equal sized cylinders is a very compact unit giving high output torques. In all designs it is usual to incorporate an adjustable end stop in at least one end of the travel and often in both.

Motor driven screw link actuators

Butterfly valves under flow conditions exhibit a tendency, when partly open, to override the actuator and shut. This tendency is surmountable by an additional valve in the system layout when using rack and pinion type actuators, and can be avoided with certain piston link operated designs but these can give rise to other problems, such as non-linear torque characteristic, and component wear problems on link joints.

Butterfly valves are sometimes used for throttling duties and in such cases it is essential that the actuator will hold the valve paddle absolutely fixed in any partly open position. This latter feature, and resistance to "slam", are inherent in the units incorporating a hydraulic motor driving a lead screw on which travels a nut which imparts rotation to the valve spindle by a suitable link system. This design is particularly suited to the higher ranges of output torque.

The hydraulic motor in some designs is controlled by a cam driven valve which responds to the actuator movement. As the actuator approaches the fully open or fully closed positions, the valve causes the motor to reduce speed and to reduce its output torque. In the open position the motor is stalled by an adjustable stop in the valve mechanism and in the closed position the motor is stalled by the paddle entering its seat. When the motor is reversed to move the paddle out of its seat, or from the "open" stop, maximum operating torque is available. The method of operation of such a valve is described in the following section.

Motor driven rotary actuators

Application: These actuators (see Fig. 4) are widely used on wedge gate, bulkhead, globe, screw down non-return, ball and butterfly valves. The actuator output is in the form of a rotating drive shaft. This rotation must be converted into the form of operation acceptable to the valve mechanism.

Valves which incorporate linear motion of the moving element, i.e. the first four mentioned, either employ a long nut

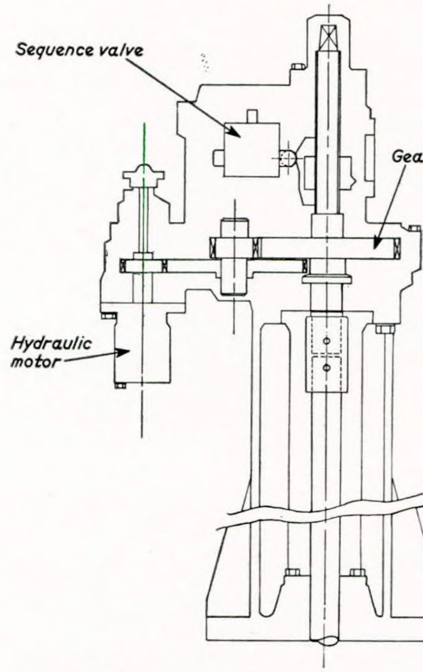

FIG. 4—*Rotary actuator*

which is rotated in stationary bearings thus absorbing the rising motion of the valve spindle, or the non-rising valve spindle itself is rotated and an internally threaded section in the moving element of the valve rides up and down on the rotating spindle, thus imparting the rise and fall motion to the element.

Butterfly and ball valves requiring 90° of rotation are fitted with reduction gear boxes designed to convert the full rotation of the actuator—say 30 turns—to one quarter turn. In this case the gear box ratio would be 120:1.

In all cases the output of the rotary actuator is effectively geared down so that the output torque of the actuator need only be proportionately lower than the valve requirement.

Construction and operation: Fundamentally a rotary actuator consists of a hydraulic motor, a reduction gear box and a vertical output shaft. If the actuator is mounted on the deck and is driving a valve at the bottom of a tank, the two will be coupled by a long connecting rod—or "reach rod"—which passes through the deck (Fig. 5). The actuator is mounted

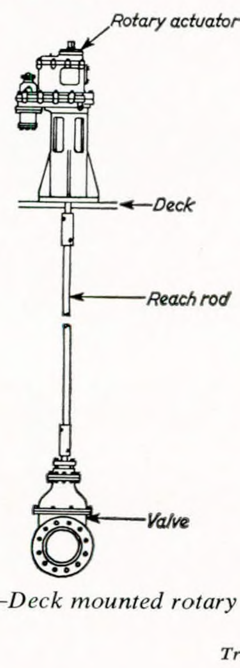

FIG. 5—*Deck mounted rotary actuator*

on a stand which incorporates a gas seal through which the reach rod passes. As the actuator output shaft should be truly vertical, to line up with the reach rod, some means must be provided to offset the deck camber. Normally a tapered pad is welded to the deck and the stand is bolted to it. The design of the stand must provide easy access to the coupling which connects the actuator shaft to the reach rod.

The most important feature of a rotary actuator is the in-built automatic control. The full output torque must be available to break the valve out of its seat. It is essential that a wedge gate valve is driven into its seat with a lower torque. Part of the energy available to drive the valve home is the kinetic energy of the rotational system consisting of actuator components, reach rod and valve components. Large valves require heavy reach rods and relatively high actuator speeds. The inertia to be absorbed when stopping the system can be substantial and could easily force the valve so hard into its seat as to make re-opening by the actuator impossible. To reduce this tendency the control valve should:

a Reduce the output torque of the actuator during the closing stages, and

b Reduce the speed of the actuator by approximately 50%.

Torque reduction can be obtained by reducing the effective hydraulic pressure across the driving motor to a figure suitably below system pressure. Speed reduction can be effected by throttling the flow of hydraulic fluid.

In a typical actuator both of these operations are initiated concurrently, within a single control unit built into the actuator casing, by a cam which moves with the operation of the actuator. Referring to the control valve and cam positions shown in Fig. 6, the sequence and method of operation are:

Starting with the cargo valve fully closed (cam position A and control valve position 2 , to command "cargo valve open", select position 1 on the control valve. Oil will flow through the non-return valve via a throttle and so the

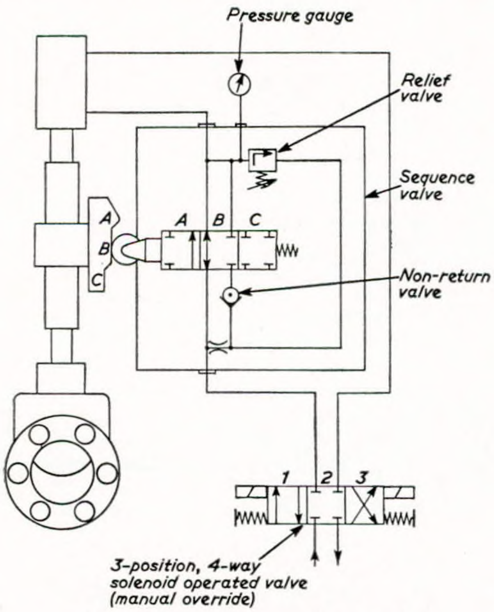

FIG. 6—*Sequence valve circuit*

actuator will develop maximum torque (dependent on system pressure) at reduced speed.

After approximately 3 turns of the actuator (approximately 10% of the c argo valve movement the cam reaches position B, thus cutting the throttle out of circuit and causing the actuator to increase to full speed at maximum torque.

Approximately 2 turns before the fully open position, the cam reaches position C, cutting off the hydraulic supply to the motor, and motion stops. The control valve can be returned to position 2.

To command "cargo valve close", select position 3 on the control valve. Oil will flow without restriction to the motor but can only return to tank through the back pressure valve and the throttle. The actuator will develop reduced torque (as dictated by the setting of the adjustable back pressure val ve at reduced speed. After approximately 2 turns the cam reaches position B and the return flow is unrestricted and so full torque is developed at full speed.

Approximately 3 turns before the fully closed position, the cam reaches position A when the return flow is again diverted through the back pressure valve and throttle, giving reduced speed and reduced torque.

As the wedge seals into its seat, the actuator is stalled by the valve resistance. Until the hydraulic supply is switched off by selecting position 2, the motor continues to develop the reduced torque under stalled conditions.

At any intermediate position of the cargo valve, the wedge movement can be stopped and the wedge held by selecting position 2.

Manual operation and local position indication: A normal requirement of power driven actuators is some means of manual operation if the power supply fails. With this rotary actuator it is a simple matter to fit a handwheel to the drive shaft to replace the motor power. The friction of the motor is removed from the system either by a motor declutching device or by a by-pass valve which ensures that pressure is not locked in the motor.

Local full scale valve position indication in the form of a pointer is an additional feature.

THE VALVE AND ACTUATOR ASSEMBLY

Coupling

Two distinct forms are employed.

Close coupling of valve and actuator: This is the system used for all linear (cylinder) and quarter-turn actuators. It is also used for rotary actuators driving valves in pump rooms and in deck lines and manifolds. The actuator is either bolted to the valve assembly direct or separated from it by a short spacer (see the section on the piston rod). The valve assembly may incorporate a reduction gear box (see motor driven rotary actuators). If the valve is submerged in a tank then the actuator will be submerged with it. Both must be so constructed as to withstand attack by all possible tank contents.

Reach rod coupling: When any form of actuator is submerged in a tank, it is not available for maintenance until the tank has been made fit for someone to work in it. To avoid this limitation, multi-turn rotary actuators mounted on the deck are commonly employed to drive all types of submerged tank valves. In such cases the valve and actuator are coupled by a reach rod (see the section on construction and operation of motor driven rotary actuators). Such rods are used in tanks up to 40m deep and there is no reason for any reasonable limitation to their length. The rod diameter is dictated by the torque to be transmitted and consideration of mechanical stability. Diameters are generally in the region of 35/50mm. A universal joint is employed at each end and at least one sliding expansion joint is incorporated. Where there is a change in direction of the rod greater than about 12° a bevel gear drive unit is generally used.

Matching valve and actuator

Whatever the type of valve and of actuator and however they may be coupled, there must be the closest possible matching of the output characteristics of the actuator (with allowance for losses in the reach rod, if used) and the input requirements of the valve. Amongst the items which must be matched are:

Speed of operation: The valve must not be closed too quickly or disturbance within the cargo or ballast pipework may cause damage. It must not close too slowly or tanks may be overfilled and the required degree of control may be lost.

Adequate thrust or torque: At all points in the opening/closing cycle the valve must receive adequate force to cause it to operate.

Avoidance of damage: The valve must *not* receive too much thrust or torque, otherwise valve stops, gear box components, etc., may be damaged by excess forces. In this context it must be remembered that damage can be caused by the inertia of a moving system as well as by static forces.

The complete analysis of all valve requirements and actuator outputs is too detailed a matter for inclusion in this paper.

Consideration has already been given to some aspects of the coupling of linear actuators and wedge gate valves in the section on linear actuators operating thrusts. An interesting system which is typical of many of the valve-actuator combinations is that of a deck mounted multi-turn rotary actuator coupled by a reach rod to a wedge gate valve. When matching this combination the actuator designer has available the following variables:

The number of actuator output revolutions. This is determined by choice of the thread pitch of a shaft and mating nut which carries the cam driving the control valve. This thread pitch is determined from the number of turns required to operate the cargo valve fully. Fine adjustment is achieved by adjusting the cam position.

The reduced pressure drop across the motor, as applied by the control valve in the closing stages. This can be quickly and finely adjusted by rotation of a control head.

The reduced rate of flow of hydraulic fluid, as applied by the control valve in the closing stages. This is varied by the choice of one of a number of different diameter throttle units fitted in the control valve. This throttle is fitted during manufacture but can be changed in a few minutes on site.

The gear ratio by which the output shaft speed is related to that of the motor. This is chosen from an available range at design stage but the gear ratio can easily be changed. Change of the gear ratio proportionately changes the ratio of output torque to drive motor stalled torque.

Drive motor size. A range of interchangeable motors is available, having progressively increased output torques with correspondingly different speed/torque curves. Drive motor size is normally determined at design stage but a change of motor takes minutes only.

System pressure. Output torque varies in accordance with the motor characteristics, and system pressure.

By appropriate choice of the basic parameters, and fine adjustment of the readily variable features, a very wide range of actuator operating characteristics can be obtained. By this means it is possible to ensure that:

Adequate torque is available to open the valve from its sealed position. Suitably reduced torque and speed are applied at the right stage of closing to ensure that the valve goes far enough into its seat to seal *without* becoming jammed home. Cut off of actuator motion occurs at the required open position. The maximum torque applied, under any conditions, to the valve is such as to ensure that the valve and its associated gear box and stops (if any) are not damaged.

When considering this combination, note must be taken of the torque absorbed by the reach rod. This will have a significant value at starting (due to stiction) and at speed (due to running friction in couplings and bearings). Under stalled conditions torque can be "locked-in" the reach rod joints but this can be released by vibration.

The actuator must therefore be able to develop sufficient torque to overcome reach rod stiction and friction and still impart to the valve (if relevant through its own gear box), its own maximum torque requirement.

Under some conditions the whole of the maximum actuator torque can be applied to the cargo valve, and so the valve construction *must* be adequate to accept this torque without damage. This point must be considered at initial design stage. It will be seen that satisfactory operation of a valve and actuator system (particularly if linked by a reach rod) is *absolutely dependent* on design of the system as an integrated whole. There can be no question of just fitting an actuator "of sufficient size" to a valve. This simple approach can result in inadequate performance and possibly in damage to the valve.

These remarks have primarily been applied to wedge gate valves but the same fundamentals apply with other valve types although the required ratio of opening torque to closing torque will be nearer unity.

VALVE POSITION INDICATION

Local and remote indication

On most valve/actuator assemblies, provision is made for local indication of the valve position. This is generally a simple mechanical system by which a pointer gives this information. If control is from a position remote from the actuators there will probably be a need to indicate valve position at the control point. In many cases the indicators are built into a mimic diagram of the pipe layout on the central control console, thus giving an easily assimilated picture of the valve positions.

Remote systems

Hydraulic indication: A system particularly suited to linear actuators is that using a flow meter through which the oil required to operate the actuator passes. If, for example, an actuator cylinder requires the supply of one gallon of oil to move it through its full stroke, then valve position can be shown by the indicator head of a flow meter which is calibrated from "open" to "closed" and traverses this span when one gallon is fed through it. Flow meter indication can also be used in association with a rotary actuator. In this case it is necessary to calibrate the flow of oil required to operate the motor through sufficient revolutions to give full valve operation. The flow meter is then calibrated to show full valve movement when this quantity of oil passes.

A second form of hydraulic indication is the hydrostatic system in which the movement of a master cylinder attached to, or built integrally within, the actuator is exactly reproduced by a slave cylinder located where indication is required; e.g. in the console. The system requires provision for bleeding of air trapped in the hydraulic oil and provision for regular rephasing of the slave to line up with the master. The shorter the distance between the master and slave cylinders, the more accurate and speedy is the indication. The distances from control room to for'd actuators on a modern VLCC are generally considered to be too great for satisfactory hydrostatic indication. The system is also affected by substantial oil temperature changes.

Pneumatic indication: The travel of the actuator or valve can be used to operate pneumatic limit switches at each end and these can pass, or block, a pneumatic supply to an indicator device in the console, etc. This system normally gives "end position indication" only and does not show the full travel. Alternatively, valve travel can be used to operate a small pressure regulator in an auxiliary supply and the output of this regulator is fed to a pressure gauge. Appropriate calibration of this gauge causes it to indicate the position of the valve blade.

If the distance between the actuator and the remove indicator is great, the response time may be somewhat protracted.

Electrical indication: As with the pneumatic system, electrical indication can be used to give either "end position" only, by limit switches operating lights or "full position" display. This latter system uses a milliammeter as indicator, this being fed by a circuit which includes either a potentiometer or a series resistance driven by the actuator movement. Such indicator systems will normally be operated by an intrinsically safe system (Fig. 7).

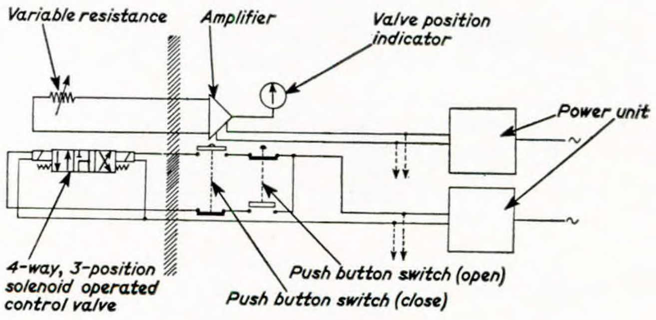

FIG. 7—*I.S. circuit*

EMERGENCY OPERATION

Provision must be made for operation of valves if hydraulic power ceases to be available due to such contingencies as failure of electrical power to the hydraulic power unit (duplicate supplies are normally employed to two separate motor pump units) or storm damage to hydraulic pipework or electrical control cabling. The last line of defence is manual operation of the cargo or ballast valves and provision is normally made for one or more of:

Local manual operation

Direct hand operation of the valve. This is possible by means of a handle or lever in the case of small butterfly, gate and other valves.

Accumulator storage

Sufficient energy can be stored in accumulators to operate piston type actuators. These accumulators generally use gas on one side of an elastic bladder and oil on the other side. The accumulator is precharged with nitrogen up to approximately two-thirds system pressures.

Accumulators are not suitable for operating motor driven actuators as the hydraulic flow required is usually too great.

Hand pump operation

All piston type actuators can be operated, albeit fairly slowly, by a simple hydraulic hand pump connected into the system. One or more such pumps can either be carried around to the actuators concerned or can be located centrally to groups of actuators and connected to them by fairly long flexible hoses.

These units are linked to the system either by screwed connexions or Tee fittings with appropriate isolator valves, or by the use of self sealing plug-in connectors.

Hand wheel operation

A hydraulic motor driven actuator cannot be operated by a hand pump as the flow required is too great. The presence of a motor indicates that gear reductions exist and by use of these, possibly with additional reductions, a handwheel drive can be used in an emergency. It is normal in these circumstances to arrange for the hydraulic motor to be declutched from the actuator, or hydraulically by-passed, to reduce the manual effort required as mentioned earlier. Even so, to operate some of the larger valves used on VLCCs by hand may take a matter of hours rather than minutes. After all, one or two men cannot easily provide the effort made by a 3OHP motor.

OTHER SYSTEM COMPONENTS

This paper primarily covers the actual valve operation and position indication. Other essential components include:

The hydraulic power unit

In which electrical power is converted to hydraulic power. Suitable stand-by and control features are incorporated. Pumps may be of gear or axial piston type. The latter generally have variable strokes to give high operating efficiency with resultant saving in electrical power consumption. Accumulators for emergency operation can be incorporated in the unit.

Control systems

This could be the subject of a full paper on its own. The hydraulic power to the actuators may be controlled manually at the actuator position or remotely at a control console.

In the latter case, the actual hydraulic flow may be controlled in the console. More often, there are control racks near each batch of actuators on deck and in the pump room, on which are mounted hydraulic directional control valves. These are remotely controlled by pilot signals. The pilot system may be electrical, pneumatic or hydraulic.

Hydraulic circuit components

The various types of actuator described require control of hydraulic flow (pressure or temperature compensation may be needed) locking in of hydraulic pressure, non-return valves, etc. A wide variety of components will be built into the system.

CONCLUSION

There can be no better way to conclude this paper than to revert to the section on "Co-ordination of system sections". However good the valves may be, however robust the actuators, however sophisticated the controls, absolute reliability will only result from the complete design of the system *as a system* and *not* as component parts. The materials employed must be suitable for the environment in which they will operate and this applies equally to corrosion and to storm damage. The components and the system must be designed for operation and maintenance under the extremes of temperature and wind and wave forces which will be encountered on ship-board. Last but not least, the system must be supported by adequate instruction manuals and backed up by an instantly available world wide service and spares organization.

Surely a demanding specification, but one which must be met if ships of modern value and complexity are to be kept operating at maximum efficiency.