HIGH-POWER REVERSING GEARBOXES

THE CAH AND BEYOND

BY

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Introduction

The adoption of the gas turbine as the main prime mover for surface warships has introduced a new problem for the transmission system designer. For more than forty years the astern steam turbine has provided a convenient and simple solution to the problem of thrust reversal but the gas turbine is inherently unidirectional, and astern power has to be provided in some other way. The difficult choice between reversing the shaft itself or changing the pitch of the propeller is outside the scope of this article which presents a description of the gearbox for the Royal Navy's new anti-submarine cruiser, and uses it as the basis for a discussion on the different options available in the design of future high-power reversing gearboxes.

It is important to stress from the outset that the views expressed in this paper are those of the author and are not in any way official policy. Although the calculations involved in the design of marine gears are both extensive and complex, design decisions are still crucially dependent on the relative importance attached by an individual to the various features of the design. Having arrived in the Gearing Section of the Ship Department directly from the 'Dagger Course' this discovery came as something of a surprise. Precise mathematical reasoning is fundamental to the design process but at almost every stage subjective risk assessments and value judgements are equally necessary. It is also at times irritating not to have the freedom to select and combine the best features offered from each design but to be constrained by commercial factors to accept a single complete package. This article presents an opportunity to escape from such constraints and to put together a design which, in the author's view at least, best meets the real needs for a future naval reversing gearbox.

The CAH Gearbox

The cruiser gearbox, manufactured by David Brown of Huddersfield, is the most powerful reversing gearbox ever to go to sea. It provides two independent gear trains, one on each side of a large main wheel, to transmit the power from two Olympus engines onto a common propeller shaft. The machinery layout with two gas turbines placed forward of each gearbox is shown in FIG. 1, and two pictures of the gearbox itself, giving some idea of its size, are shown as FIGS. 2 and 3. The gearbox is symmetrical about the main shaft line, as can be seen from the diagrammatic layout (FIG. 4), and the following description considers one half only.

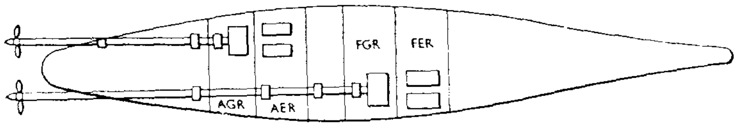


FIG. 1—LAYOUT OF CAH MACHINERY

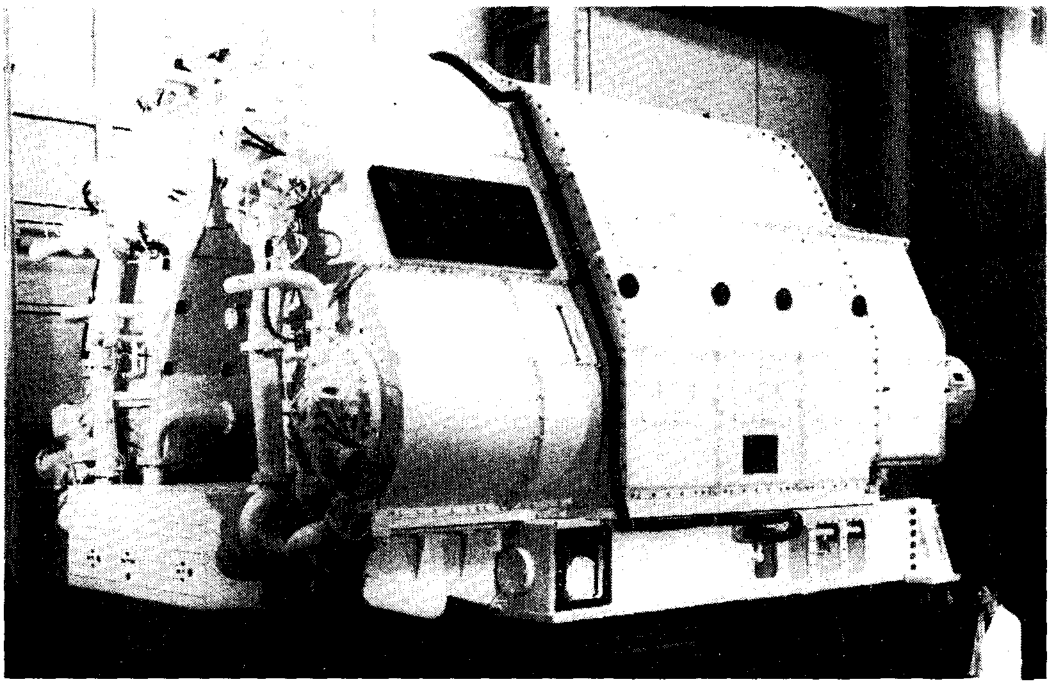


FIG. 2—CAH GEARBOX VIEWED FROM AFT
(Photograph by courtesy of D.B.G.I. Ltd.)

The drive enters the gearbox from an M4000 flexible coupling (identical to that used in the Type 21 frigate) which caters for the relative movements between the solidly-mounted gearbox and the raft-mounted turbines. The speed reduction is accomplished in three stages of single tandem, double helical, articulated gears, which is a departure from the more normal naval practice of using only two stages of reduction, with dual tandem intermediate gears forming a locked train. The choice of triple reduction was made partly to ease manufacture—all the gear elements, and especially the secondary wheels, are thereby smaller—and also because, in this particular case, additional gearbox length was more acceptable than additional width.

The primary gears are situated at the forward end of the gearbox and the output from the primary wheel (Y) is taken aft, over the main wheel (W) by a second flexible spacer coupling to a secondary pinion (K), which in turn transmits the drive through one of three alternative parallel paths, back onto

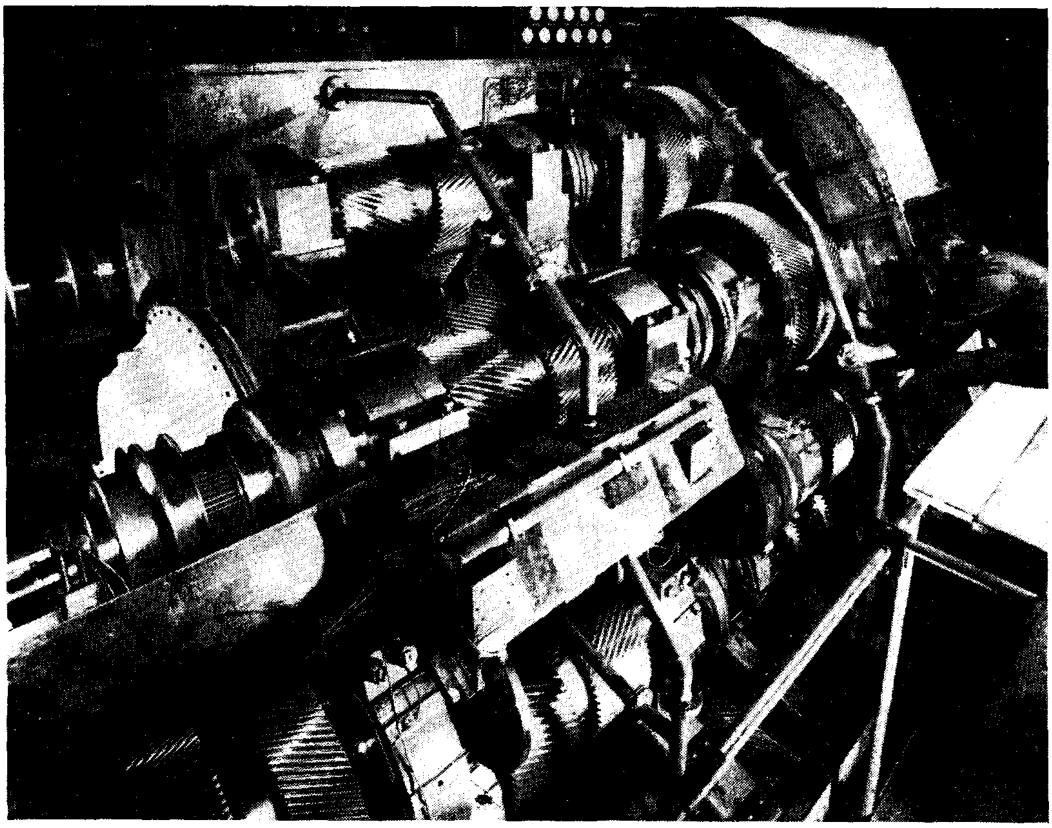


FIG. 3—CAH GEARS VIEWED FROM FORWARD, SHOWING PRIMARY WHEEL LOWER LEFT, FINAL DRIVE PINIONS CENTRE, AND INTERMEDIATE GEARS RIGHT
(Photograph by courtesy of D.B.G.I. Ltd.)

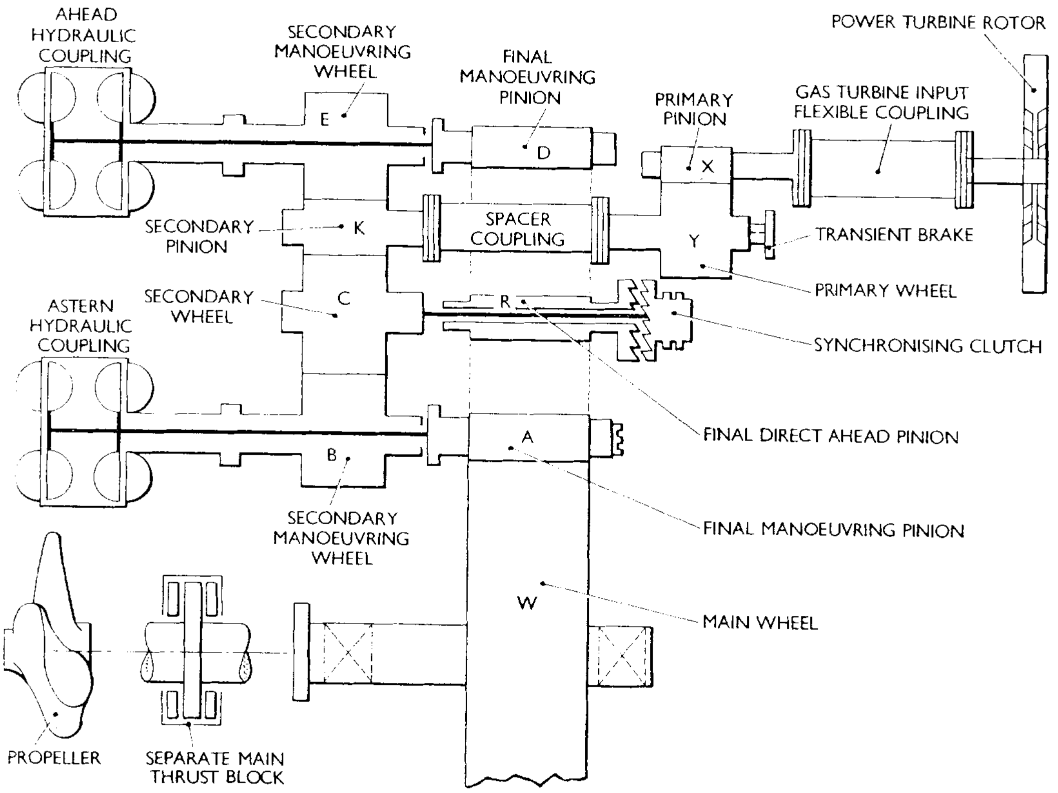


FIG. 4—DIAGRAMMATIC ARRANGEMENT OF CAH GEARS

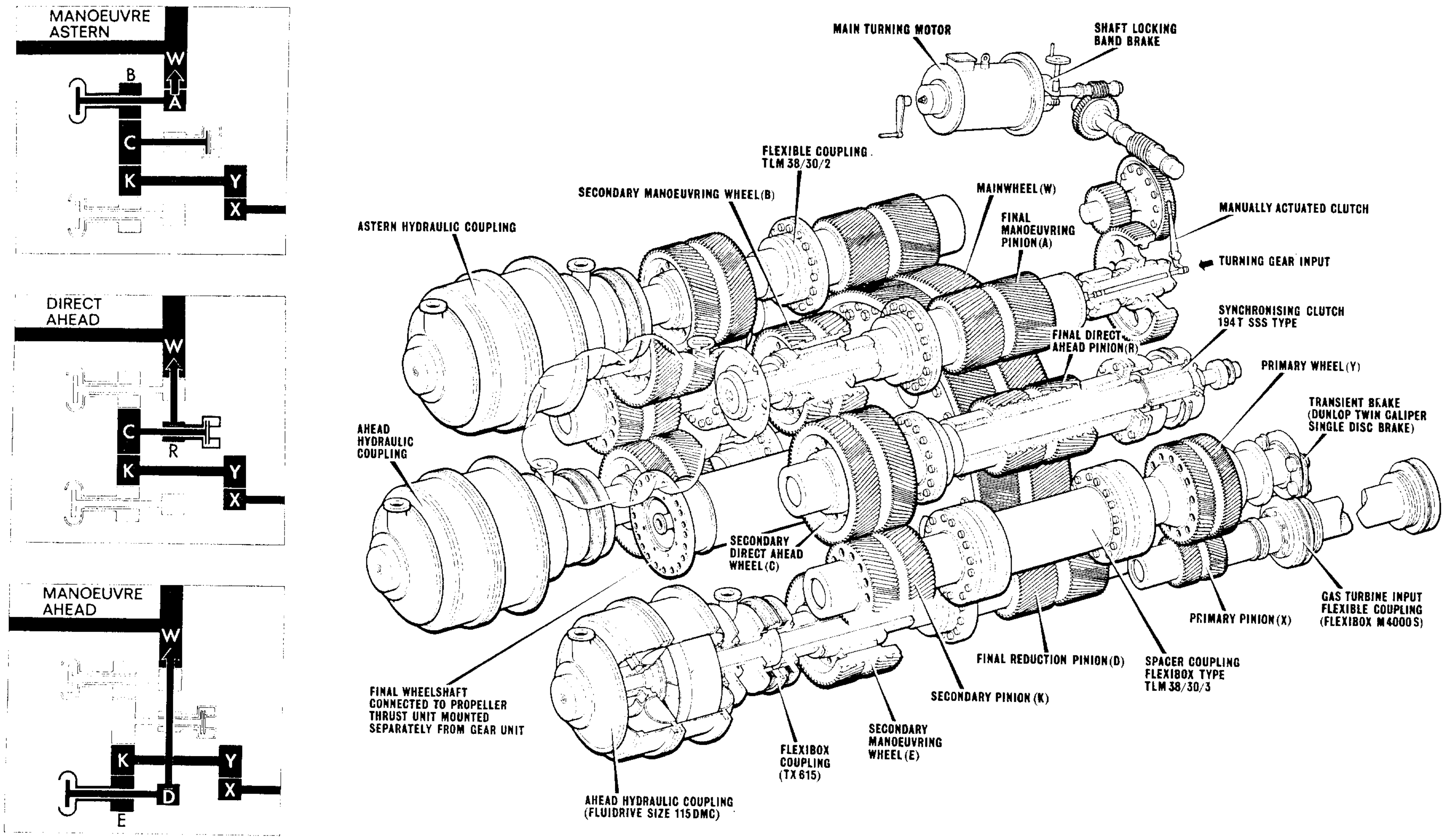


FIG. 5—PICTORIAL ARRANGEMENT OF CAH GEARS

the main wheel. The lower path is formed by the secondary manoeuvring wheel (E), the ahead fluid coupling (situated aft of the gears) and the final manoeuvring pinion (D). This path provides ahead manoeuvring drive. The middle path provides ahead direct drive through a second wheel (C), also in mesh with pinion (K), through an SSS clutch, and onto pinion (R). Astern drive is taken through the upper fluid coupling by a third manoeuvring wheel (B) in mesh with wheel (C) and forward again to final manoeuvring pinion (A) and so to the main wheel. A good impression of the overall layout of these components, and the three modes of operation, can be gained from Fig. 5.

This arrangement provides the facility for shaft reversal by filling and emptying the relevant fluid coupling and it also allows the Olympus to drive for long periods at high powers through the SSS clutch, so avoiding the power losses which would result from extensive use of the fluid couplings.

An attractive feature of this design is the freedom to accelerate the ship smoothly from coupling drive to clutch drive without the need to stop, or even slow, the shaft. This is made possible by selecting the gear ratios of the intermediate gears (wheels E and C) so that, when the drive is taken through the ahead fluid coupling, the output of the SSS clutch rotates faster than its input. The relative motion across the clutch is therefore in the correct sense to permit safe engagement, and the transfer of drive is easily accomplished by bringing the clutch to the 'ready to engage' state and then emptying the fluid coupling.

The transfer back to coupling drive requires a torque reversal across the clutch to disengage the two halves. This torque reversal occurs naturally if astern drive is selected and the controls have only to wait until the shaft speed has fallen to a limiting value before filling the astern coupling. The transfer to ahead coupling drive can be more difficult as, under certain conditions especially at lower speeds, the gas turbine does not slow down quickly enough to reverse the torque and to pull the clutch out of engagement. If the ahead coupling were to be filled before the clutch disengaged it would be constrained to run with an unusually high speed differential between input and output leading to heavy churning losses and overheating. To avoid this unpleasant situation, a small twin-calliper disc brake is fitted to the primary wheel shaft, and this can be applied whenever necessary to further slow the turbine and so disengage the clutch.

Mechanical aspects

In terms of the gear elements themselves the design is conservative: the details are given in Table I. The total power transmitted is, however, very high and the large size of the gearbox introduces its own problems, one of which is the provision of adequate flexibility between the various components. This problem is aggravated by the use of double helical gears.

Involute gear pairs will tolerate considerable variations in the gear centre distances without distress—indeed it is this feature which has led to their widespread use. Their tolerance to angular displacements is however far less, as such displacements give rise to the heavy concentration of load at one end of the teeth. By careful design of the gearcase structure, the four bearings required to support any meshing gear pair can be maintained in the correct relative positions, but maintaining this order of accuracy across the length of a large gearbox is quite impossible; some flexibility has, therefore, to be introduced in the shafts connecting these gears to prevent the inevitable distortions from creating unacceptable end loadings. Flexibility can be provided either by using thin drive shafts, made as long as possible

	Primary		*Secondary*			*Final*	
	Pinion	*Wheel*	*Pinion*	*Manoeuvre Wheels*	*Direct Wheels*	*Pinions*	*Wheel*
Reference Diam	11·9898	29·7428	22·4530	36·4450	41·3264	20·0314	121·7006
No. of Teeth	52	129	69	112	127	53	322
Active facewidth	12·9921		18·1102			27·953	
Helix Angle	29° 50′ 15″		28° 35′ 38″			28° 4′ 23″	
Normal DP	5		3·5			3	
Material	EN36(C&H)		EN36 (C&H)	EN 30b		EN36 (C&H)	EN 30b

by passing them back through the centres of the gears they connect (quill shafts), or by using some kind of flexible coupling. Both techniques are used in the cruiser with quill shafts taking the drive from the fluid couplings through the secondary wheels B and E to the final pinions, and flexible couplings isolating the primary and secondary gears.

It can be seen from Fig. 5, however, that the quill shafts themselves have been fitted with flexible couplings, not to provide for angular distortion, but to isolate the fluid couplings from the axial movement of the main wheel. Such axial movement occurs when the main thrust collar moves through the clearance of the main thrust bearing—more than 2mm in the CAH. The provision of flexibility between the various sets of gears gives rise to the term 'articulated' in the description of the gearbox configuration. Although flexible couplings provide both articulation and axial isolation they do take up a great deal more space than directly-coupled quill shafts and their use can often be avoided, or at least minimized, by using single helical gears.

Each set of meshing gears must be positively located, radially by journal bearings and axially by thrust bearings. With spur and single helical gears every gear element has to be individually located against axial movement but the use of the double helix provides relative axial location for all the gears in a common mesh, so that only one thrust bearing is required for the whole train of gears. At first sight this appears to be entirely advantageous but in a marine gearbox the main wheel will usually move with the main thrust collar during thrust reversal, and the isolation from this movement provided by the sliding of the single helical gears may well justify the cost of additional thrust bearings.

Fluid Couplings

The fluid couplings fitted in the CAH are very similar to those used in the GMDs and GP frigates. In this application, however, they provide the only means of shaft reversal and the necessary increase in capacity is provided by increasing the working circuit diameter and by using a double fluid circuit. The back-to-back configuration adopted in this design balances the internal thrusts generated by the oil rotating within the coupling, although a small thrust bearing is still required to locate the coupling against shock and gravitational forces (Fig. 6). The fluid level within the coupling is controlled by a hydraulically-actuated scoop which can be inserted to drain the rapidly circulating oil from the chamber around the coupling.

With four couplings, two clutches, and two brakes in each gearbox, the control problem is significant and an extensive series of interlocks is required to prevent maloperation such as that which has occurred in the GMDs. The main constraint is that of excessive oil temperature which can arise through a variety of circumstances and is therefore difficult to prevent entirely. Rather less likely, though certainly more dangerous, is the risk of overstressing the fluid couplings by filling them when they are rotating at more than the allowable speed. The general design of the gearbox is robust and tolerant to error, but the control system is very complicated and is likely to provide the most significant threat to the system reliability.

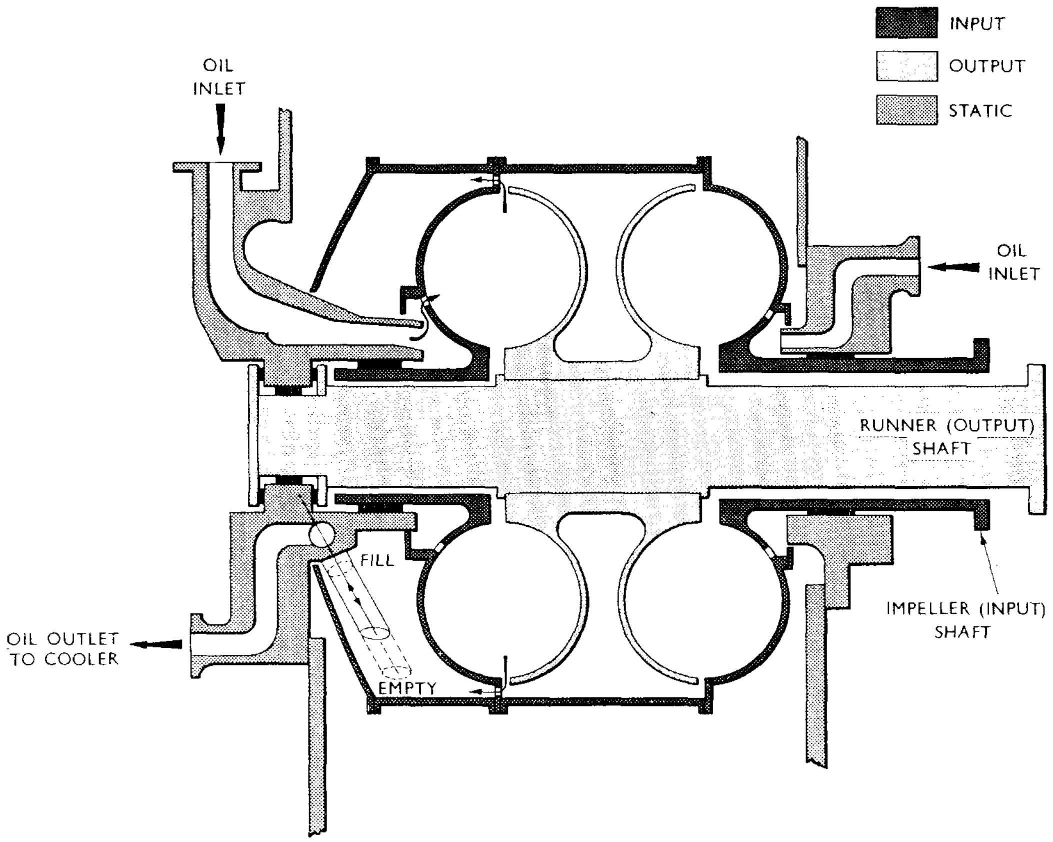


FIG. 6—DOUBLE-CIRCUIT, SCOOP-CONTROLLED, FLUID COUPLINGS FITTED IN CAH

The Choice of the CAH Gears

Having examined in some detail the design which was eventually selected, a comparison can be drawn with the alternatives offered by GEC (then AEI) Ltd. and by Vickers Ltd. The reasons behind the selection can also be considered.

Vickers Design

By far the most radical of the three designs was that proposed by Vickers Ltd. This utilized the reversing ability of a suitably connected pair of epicyclic gear trains. As the concept of reversing epicyclic gears has not been discussed in the *Journal* before, a brief diversion is warranted.

The basic epicyclic gear train contains a central sun gear (C), surrounded by a number of planet gears (B) whose axes are joined together by a common planet carrier (D) (FIG. 7). These planets mesh in turn with the internal teeth on the annulus (A). The main advantage of epicyclic gears lies in the

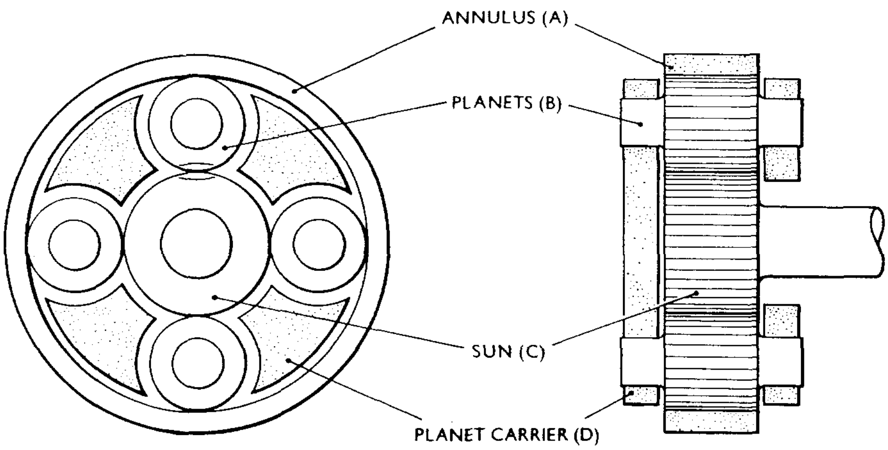


FIG. 7—BASIC EPICYCLIC GEAR CONFIGURATION

number of parallel power paths provided by the planet wheels. An epicyclic gear with say four planets would ideally transmit only one quarter of the total power through each mesh point, giving an equivalent reduction in gear face width over a more conventional pinion and wheel arrangement. The problem facing the designer who wishes to utilize this feature is how to ensure that the load is shared equally between the various planets, not only in the steady state but during transient manoeuvres and indeed during torsional vibrations. This is accomplished, to a greater or lesser degree by providing flexibility, either in the annulus, in the sun gear, or in the planet bearing supports.

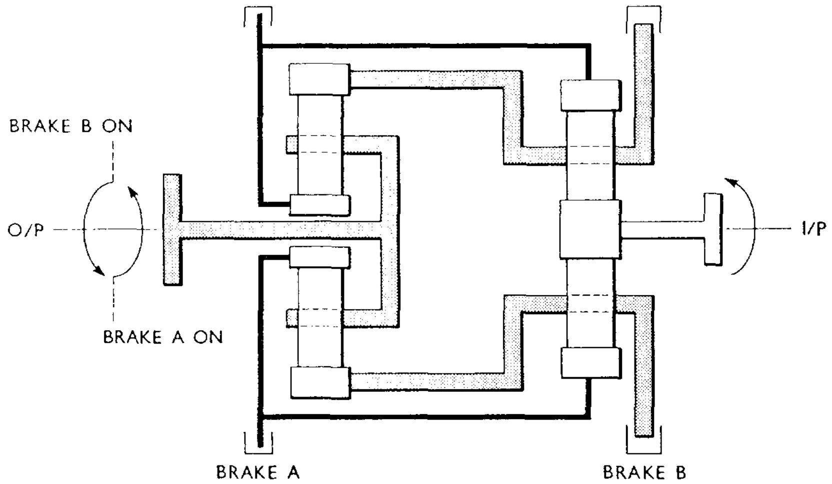
An epicyclic gear train can be operated in a number of ways depending on which of the three coaxial elements are held fixed—the annulus, the planet carrier, or the sun. It can be seen from TABLE II that when the planet carrier is held fixed the sun and the annulus rotate in opposite directions, whereas if the annulus is held the planet carrier will follow the rotation of the sun. By suitably combining two epicyclic gear trains, and by attaching brakes to two of the elements, this reversing property can be harnessed to change the direction of the gear output by holding first one brake and then the other. A number of such arrangements are possible, three of which are shown in FIG. 8.

TABLE II—*Behaviour of epicyclic gear trains*

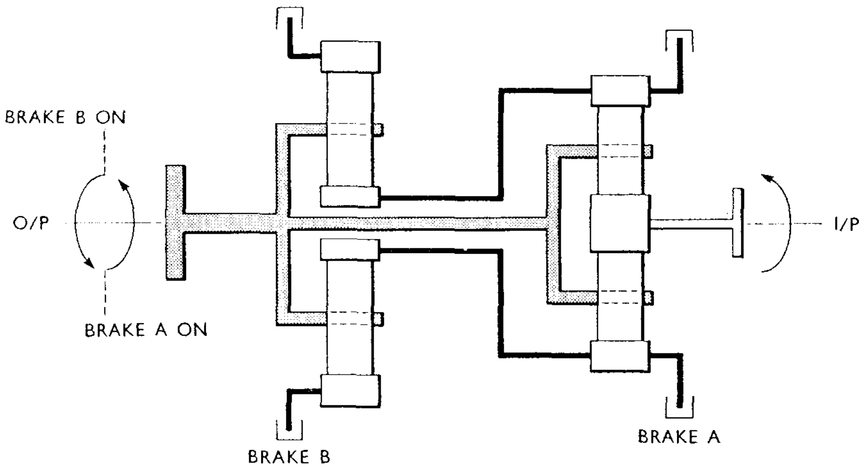
	Component			*Reduction Ratio*	
	Sun	*Planet Carrier*	*Annulus*	*Formula**	*Range Available*
Planetary Gear	Input	Output	Fixed	$\frac{A}{S} + 1$	3:1 to 12:1
Star Gear	Input	Fixed	Output (Reversed)	$\frac{A}{S}$	2:1 to 11:1
Solar Gear	Fixed	Output	Input	$\frac{A + S}{A}$	1·1:1 to 1·7:1

* A = Number of teeth on Annulus, S = Number of teeth on Sun

SCHEME 1 BRAKE A ON GIVES PLANETARY/SOLAR BRAKE B ON GIVES STAR/PLANETARY



SCHEME 2 BRAKE A ON GIVES PLANETARY/SOLAR BRAKE B ON GIVES FREE/PLANETARY



SCHEME 3 BRAKE A ON GIVES PLANETARY/FREE BRAKE B ON GIVES FREE/SOLAR

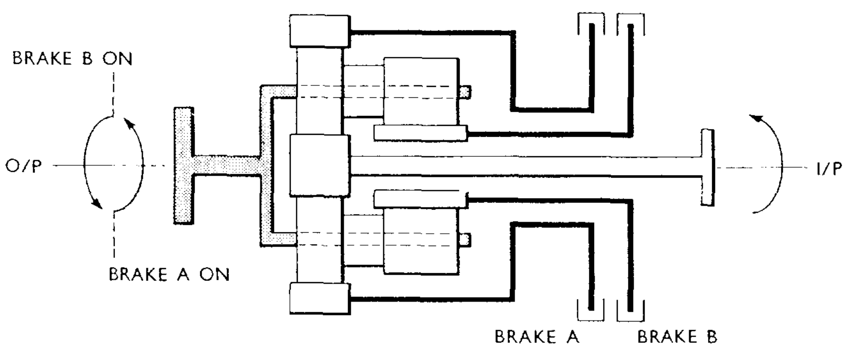


FIG. 8—REVERSING EPICYCLIC GEARBOXES—ALTERNATIVE ARRANGEMENTS

The Vickers proposal for the CAH gearbox used two primary reversing epicyclic gearboxes based on scheme 1 of FIG. 8, one on the output of each Olympus. The drive from these reversing gears was combined by a conventional parallel shaft gear onto the propeller shaft. The arrangement is shown in FIG. 9.

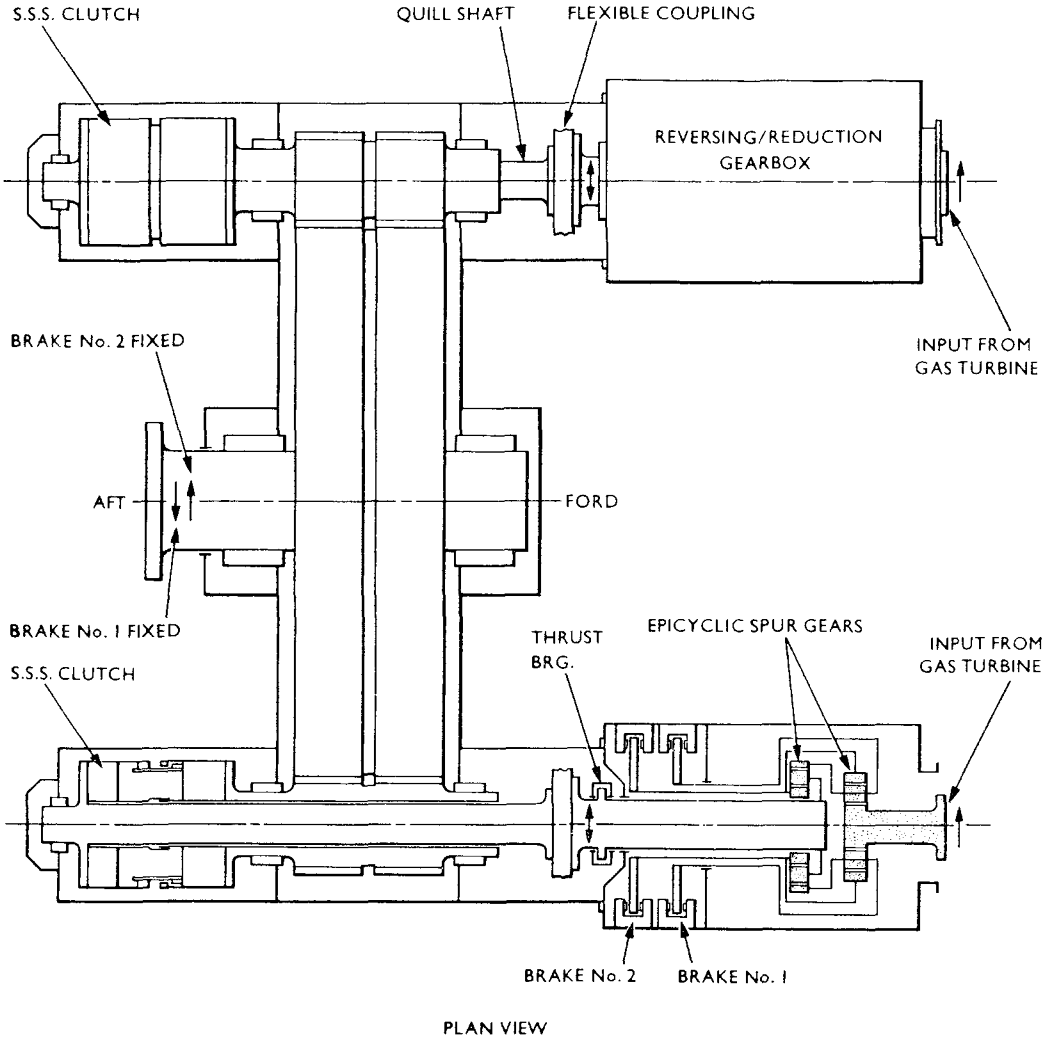


FIG. 9—VICKERS LTD. PROPOSAL FOR CAH GEARS USING TWO PRIMARY REVERSING EPICYCLIC GEARBOXES

GEC Design

The third alternative, that proposed by GEC, utilized conventional gearing with fluid couplings to affect reversal, as in the DBGI design. There were however several important differences as can be seen from FIG. 10. Perhaps the most noticeable change is that the direct-drive SSS clutch is mounted in line with the ahead fluid coupling, rather than on a separate shaft of its own. This scheme has the advantage of reducing the number of rotating elements but, as the clutch is on the same line as the ahead fluid coupling which must slip to transmit power, the clutch output must rotate more slowly than its input. If the clutch were of conventional design it could not be safely engaged. To overcome this difficulty an 'inverted' clutch is used which engages when the output shaft overtakes the input rather than the other way around. This is a safe and effective system but in order to transfer to direct drive a temporary speed reduction is necessary. The change of drive is accomplished by

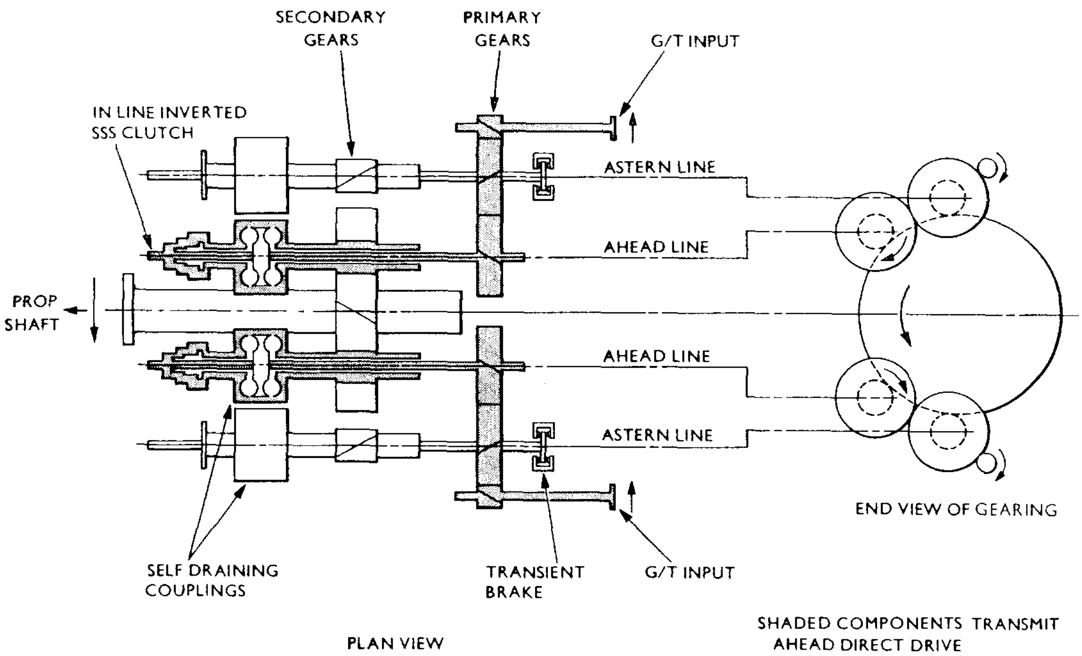


FIG. 10—DIAGRAMMATIC VIEW OF AEI LTD. PROPOSAL FOR CAH GEARS

reducing power on the gas turbine and applying a turbine brake to reduce the speed of the input below that of the output. The clutch then engages and locks, so removing a control limitation and allowing full power to be reapplied.

An advantage of this arrangement is that the losses in direct drive are minimized by positively locking one fluid coupling and also, surprisingly, by causing the astern coupling to operate with exactly 200 per cent. slip, i.e. with both halves rotating in opposite directions at the same speed. Under this condition, equal and opposite flows are generated in each half of the coupling. This stalls the circular flow which normally provides the linkage between input and output and losses are therefore lower than at slightly higher or lower slips. This feature is of significance even when the coupling is nominally empty as some small oil flow is always provided to remove the heat generated by windage.

The fluid couplings themselves are also unusual in that they do not employ emptying scoops. In most applications fluid couplings are required to transmit power for long periods and the efficiency of the coupling is therefore of considerable importance. In the reversing gearbox application, fluid couplings are only used for manoeuvring, when efficiency is of little importance. As the torque on both sides of the coupling must be equal—there is no torque reaction member—the power loss in the coupling is directly proportional to the speed difference between input and output. The percentage difference in speed is known as slip. It is normal practice to limit slip to below 4 per cent. to minimize losses and it is this requirement, when combined with the input speed, which determines coupling diameter. If efficiency is of less importance, a higher slip can be tolerated, say up to 8 per cent., and the coupling size correspondingly reduced.

The oil flow path within the coupling is so arranged that the coupling is self-draining. Quite how this is achieved is not certain although it would seem likely that the back of the impeller is so shaped as to act as a centrifugal pump, providing the incoming oil with a sufficient head to overcome

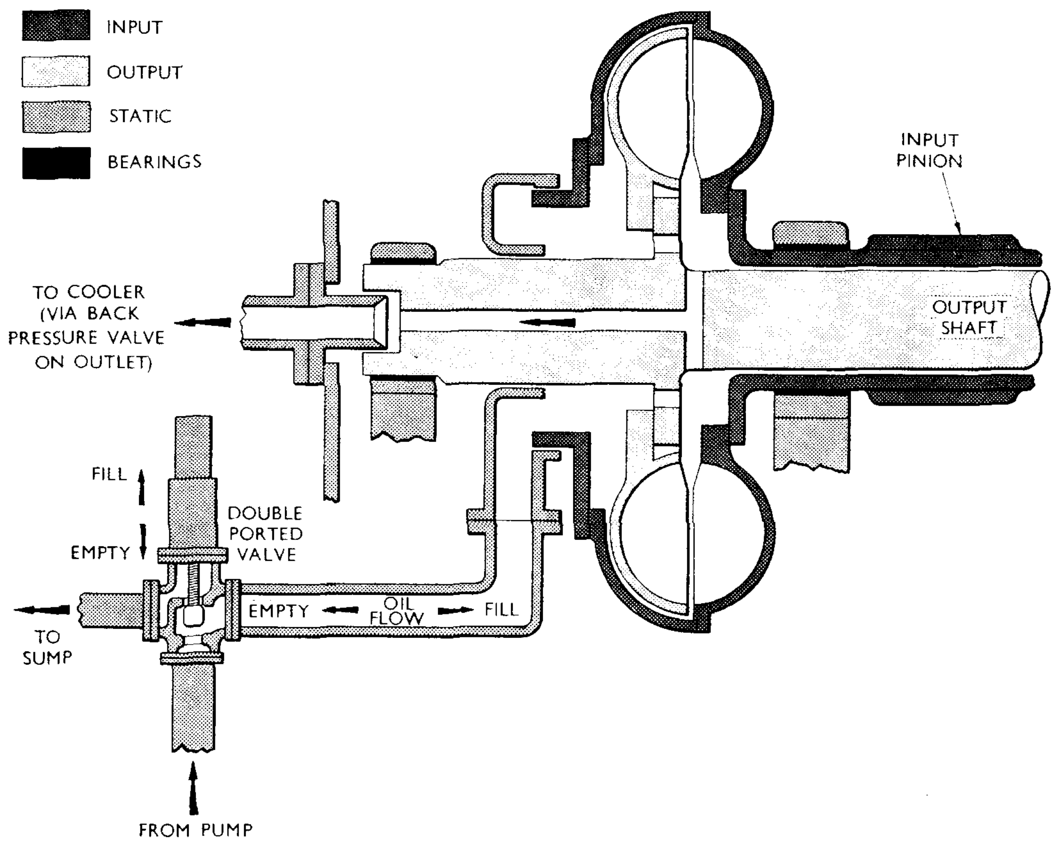


FIG. 11—SELF-DRAINING, HIGH-SLIP, SINGLE-CIRCUIT COUPLING (SYNCHRO DRIVE LTD.)

the centrifugal forces generated within the working circuit. This reverses the normal oil flow path through the coupling, and the ports provided towards the inner radius of a traditional coupling, through which it would fill, are reversed to evacuate the oil past a back pressure valve to an oil cooler. The rate at which the coupling empties therefore determines the oil throughput, and if the coupling is supplied with a flow from a pump of greater magnitude than the draining flow, it will fill; if the supply is shut off, it will empty. The coupling can be controlled simply by a single double-ported valve (FIG. 11) which, when it is set to empty, shuts off the pump and connects the filling line to drain.

This design is very attractive as not only are these self-draining couplings smaller and lighter than the more traditional coupling but they are also likely to be even more reliable as most of the in-service failures of fluid couplings have been associated with the scoop mechanism. There are, however, two problems: concern as to the controllability of the oil level under all ship-manoeuvring conditions and fear of excessive oil temperatures. Information on the performance of this device is very limited although some satisfactory results have been obtained from a small model. Full-scale development trials would, however, be needed before the device could be fitted in a warship.

Comparison of CAH alternatives

When the three designs are compared, a fairly clear pattern emerges with the Vickers design significantly smaller than the others but involving a step change in gear technology and a significant degree of risk, especially in the development of the high-power disc brakes. The GEC design was attractively simple with the least number of rotating elements but its effectiveness rested heavily on the performance of a novel and as yet untried design of fluid coupling. The David Brown design was large but simple and made up entirely of components which had already been proven in naval service.

The choice between the three was not made easily. All three would probably have met the requirement, but the big difference between them was in the degree of risk that each involved. At the time the decision was made, the Ship Department was very busy with the Type 42 project at its height, the MCMV and the Type 22 taking considerable effort and the CAH itself, a major capital project, on a short timescale. Decisions of this type depend heavily on subjective value judgements and risk assessments—after all, how else can the choice between, say, a saving of four inches in length and a fall of, say, one per cent. in reliability be made even if firm figures are available?

The atmosphere prevailing at the time of the gearing design decision was one of caution. Recent experience with the complex reversing gearboxes in the GMDs and GP frigates had been far from satisfactory and the timescale of the CAH prevented any full development trials. (The gearbox supplied to Ansty was in no sense a prototype, as the manufacture of the gearbox for the first ship had to be underway well before results from the shore test facility became available.) On top of all the technical factors the various commercial differences between the three gearing firms also had to be considered.

The outcome of this process is, of course, already known—of the three firms David Brown were judged to have produced the best solution to the particular problem presented by the CAH gearing, but, as this analysis has shown, other options were available and the rest of this article will examine the various possible future solutions to the problem of reversing a high-power marine gas-turbine drive.

The Future Reversing Gearbox

Any reversing gearbox has to accomplish a number of separate functions: it has to be capable of transmitting the required power in both directions; it has to change from the ahead drive state to the astern; and most importantly it has to be capable of braking the shaft and the ship to give the necessary manoeuvring performance. The present state of gearing technology ensures that the first is no longer a problem and this is borne out by the exceptional reliability being achieved by gear teeth in service. Changing the direction of drive is rather less easy especially as the changeover has to be accomplished under high-torque loadings. Various clutch mechanisms are available based on a number of different principles, but only the SSS Clutch and the 'non-positive' fluid coupling have been proved in naval service.

The major problem area, however, is the dissipation of the large amounts of energy involved in braking the shaft against the way of the ship. The energy involved is directly related to the specified reversing performance, as the more the shaft speed is allowed to decay through the slowing of the ship by hull resistance, the less the energy fed back into the reversing mechanism. The traditional approach of specifying astern power as 30 per cent. of ahead power and leaving the dynamic performance to take care of itself is no longer adequate and a careful study of the ship's operational requirement is essential if the gearbox design margins are to be correctly set. There are significant penalties in asking for more than is necessary.

Before examining the specific options available, the particular requirements of gears for naval marine propulsion need to be defined. The prime requirement is undoubtedly for very high reliability. Failures within the main gearing immediately reduce the ship's capability and, more often than not, the corrective action is beyond the capacity of the ship's staff and turns out to be expensive and time consuming. Even one failure of the gearbox during the ship's life is probably unacceptable as a design target, which should be to aim for not more than say one full failure of the gearbox every fifty ship-years. An examination of the limited failure information available shows that most failures occur in precisely those components which multiply most rapidly in the complex reversing gearbox: flexible couplings and bearings. The other major weakness has been auxiliary drives. If adequate reliability is to be achieved, not only must these components be of careful and conservative design but the reliability of the clutch or coupling device must also be of a very high order.

Size, and especially length, is also of great significance and it is with the aim of reducing the percentage of the total ship volume occupied by the machinery that the specific tooth loads in R.N. gearboxes have been more than doubled in the last twenty years. A further reduction in gear-wheel size is entirely feasible; in fact, test gears have already been run up to a K factor of 1800 lbf/in^2. Unfortunately in typical naval installations the gears themselves occupy considerably less than one half of the gearbox length and the law of diminishing returns operates with some force. As loadings rise, the additional sensitivity of the more highly-rated gears to errors in material, manufacture, or maintenance, come increasingly to outweigh the successively smaller percentage reductions in total gearbox size. Probably the main gain from raising the allowable loading further would be the freedom this would give to use single gear-trains rather than parallel locked trains.

The final major factor, and one of increasing importance is cost. Obviously the details of a design will influence cost to a substantial degree but this is not a subject which the author has the knowledge to discuss. There is, however, a significant saving in both time and money if a single gearbox can be developed which is sufficiently flexible to accommodate a number of engine layouts and hence a number of different ships, without fundamental change. The concept of a universal design able to accommodate a number of layouts is particularly attractive at the present when there are pressures to start gearbox development work in advance of a clear ship definition.

There is little doubt that the next generation of R.N. ships will be powered by gas turbines, which, because of their limited life are inherently unsuited to single engine per shaft applications. It is therefore most likely that the future ship transmission system will involve one of the four layouts shown in FIG. 12. The concept does not change significantly if a diesel engine should replace one of the gas turbines. The requirement therefore is to produce at minimum cost, a reliable, compact, basic design which can easily be modified to accommodate these various input positions.

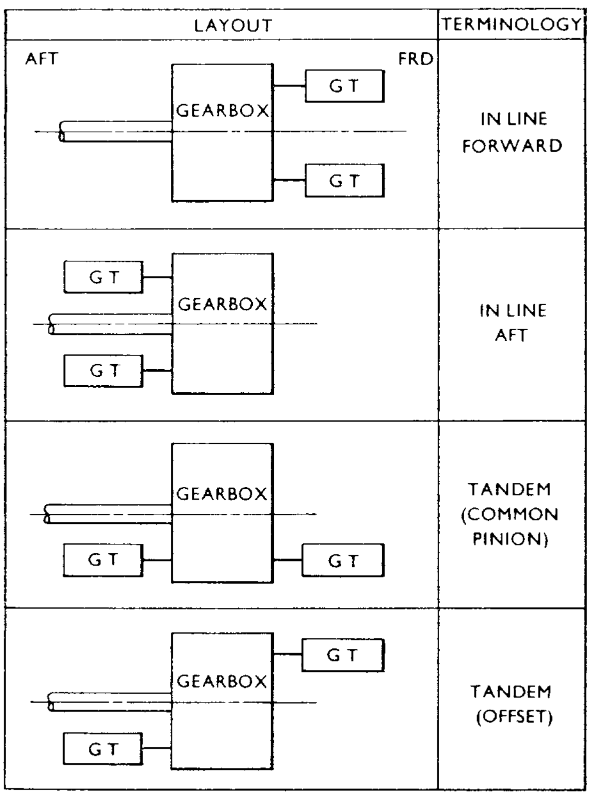


FIG. 12—TWIN ENGINE MACHINERY LAYOUTS