TRANSMISSION DESIGN FOR FUTURE FRIGATES AND DESTROYERS

BY

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Introduction

When the topic for the above presentation was proposed it was expected that, by the date fixed for its delivery, there would be a fully documented case history of a new warship transmission design to discuss. Despite several postponements, this expectation has not been fulfilled. It is therefore not possible to base this article on technical argument of the design features of a new transmission. As an alternative approach it has been decided to describe the constraints imposed on a typical Ship Department design section when facing a new project and show how these are met and scheduled during the design development phase of a new warship. Since there is also considerable topical interest in new proposals for reversing transmissions, some of these are also discussed.

It should be noted that the procedures described are only applicable to Ship Department 'in-house' warship designs. Designs prepared by shipbuilders under a 'design-and-build' type of contract appropriate for small warships and auxiliary vessels are handled by a different routine.

The Industrial Base

Before discussing the design process in any detail, it is worth surveying the industrial environment in which the work is carried out. Most readers will have heard of an organization called the Navy and Vickers Gear Research Association—NAVGRA for short. This unique organization was set up in the aftermath of the second World War with the objective of creating a national capability for making better gears for warships. To start with, many of the members of a very diverse industry came together and were sponsored by the Navy to research and develop the various aspects of their enterprise. The essence of the success of the organization was that the results of most of this research were shared among the participants. Over the years a common baseline of design and manufacturing technology was accumulated. At the same time the gear manufacturing industry crystallized itself into major centres of expertise until, at the end of the NAVGRA programme in 1979, three industrial members were left—Vickers Shipbuilders, David Brown Gear Industries, and the Marine Gear Division of GEC. Each of these companies shares a common fund of knowledge acquired at the Ministry's expense. All of them are happy to sit together around the table and argue with us about the finer points of our design standards.

In the more recent years the main objective of the association was to acquire the skills necessary to make gearboxes smaller, lighter, and more heavily loaded. In this it was eminently successful; these skills were indeed acquired and are the envy of many of our international competitors. With hindsight it is something of a pity that funds did not permit parallel investigations into what makes gears quieter, cheaper, or more reliable in the long term.

To date, and certainly in the immediate future, our gas-turbine-driven warships will rely on controllable pitch propellers (CPP) as propulsors. Stone Manganese Marine Engineering Ltd, at Greenwich, are the only national manufacturers of this type of propeller in the power range of interest. They have developed and produced the XX hub which is identical in the three major classes of ships—Types 21, 42, and 22. When the requirements for this hub were prepared, there was very little experience to guide the designers. The margins built in were considerable. The design is therefore very conservative and has performed well at sea. Since this is the only example we have, there is a degree of uncertainty involved in scaling the design to meet a new requirement, particularly if the opportunity is taken to reduce the safety margins to apparently more reasonable levels. SMME are confident that they can produce an acceptable design but this remains an area of concern.

Experience, often bitter, has led to a major change in policy concerning the design of the hydraulic system to operate the hub. The aim was a powerful, accurate, and simple system. The former objectives were achieved but the latter was not. A 'simplified' system is now in being and at sea but a radical change to an open circuit system is a better solution and will be implemented in later versions of the Type 42 and 22. This design will be the basis of any future system. The manufacturer is only too glad to endorse this decision since it was his recommendation at the outset.

The Selection Process

So much for the background. What happens when a requirement for a new gearbox and transmission arises?

The first point to appreciate is that probably no previous transmission is likely to be suitable. In the recent past, classes such as the Type 42, 21, and 22 have been based on the Olympus and Tyne gas-turbine prime movers and the transmission design of these ships has been constrained to enable a standard gearbox to be used together with many common transmission parts. Future designs are very likely to feature the Spey derivative gas turbine (SM1A) and may also have diesel or electrical cruise drives. The next point is that a new transmission system would probably require a shore test facility to demonstrate the performance of many other features of the plant besides those of the gearbox and its associated auxiliaries. The shore test facility must be built and operating well in advance of the first of class setting off on sea trials for the experience gained from the trials to be useful. Gearboxes, by their very nature, take a relatively long time to design and make.

Taking these points together they result in the need for a very prompt start on the part of the transmission designer as soon as a likely ship programme has begun. In practice this means that the Power Transmission Section (D151) and the Forward Design Group of the Ship Department must work together to establish feasibility and the special requirements of a particular design at the earliest possible stage. The gearbox will have to be tailor-made to meet these requirements. What is more, the design will have to be frozen at a relatively early stage in the ship design process and the ship subsequently altered round it if changes are necessary and delays to be avoided.

To illustrate the point, FIG. 1 shows a typical ship design programme. The obvious key date is the ship order date. From this it is possible to work forward to the Dockside Date for the delivery of the gearbox to the shipbuilder. Working in the opposite direction the design-and-build programme for the shore test facility is established.

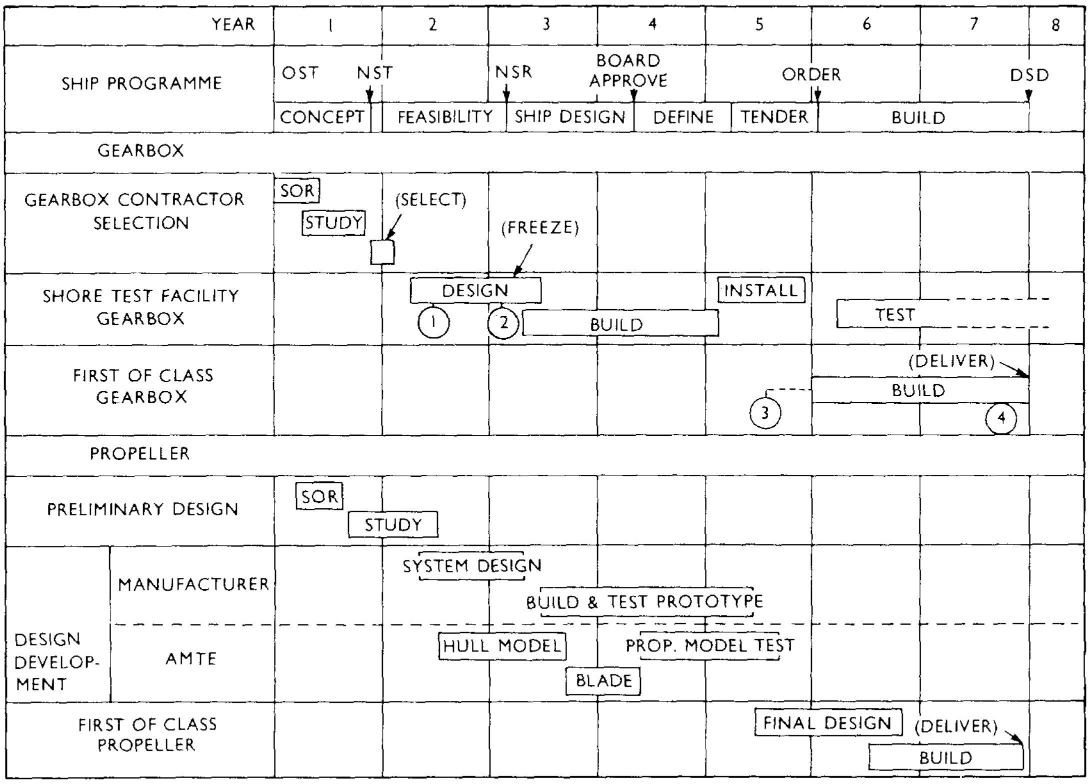

FIG. 1—TYPICAL DESIGN PROGRAMME FOR GEARBOX AND PROPELLER

Concentrating on the very earliest stages of the gearbox design project, the process that leads to the selection of a gear manufacturer is considered first.

In 1967, when the Type 42 destroyer was designed, this selection was made at the latest possible date so that the ship requirements could be fully defined. A comprehensive statement of requirements was drawn up and a design competition held. The gear makers all sent in complete design proposals which were judged by the Ship Department. On this occasion, David Brown's design was selected and that went into the ship with only minor changes. For this ship, a shore test facility was considered unnecessary and was, indeed, barely feasible in the time available.

In 1978, when the requirement for a future destroyer began to emerge, a different approach was adopted. Here, the intention was to involve the gear manufacturer at a much earlier stage in the development of the ship design before even a tentative specification could be written. This was made clear to each potential contractor at the start of a rather different competition. A less precise Statement of Requirements than formerly was compiled, containing only provisional information on engine centres and reduction ratio. With this went a list of priorities for special requirements which would be given particular weight when submissions were eventually judged. The weighting factors for the major attributes were:

Reliability	5
Low Noise	4
Maintainability	4
Ease of installation	2

Note that nothing was said about the need for minimum weight or space or even cost. Costs were, however, required to be assessed together with the usual range of other design and production features.

The three major manufacturers duly forwarded their proposals and so D151, together with the Directorate of Naval Ship Production, began the assessment of the relative merits of each. The adjudication process involved a numerical method of evaluating the various attributes together with a considerable amount of engineering judgement. At the end of the day, the design features of each submission came out to be very nearly equal. There were greater differences, however, among the commercial features of the proposals, particularly those associated with production costs and schedules. The outcome of this closely contested issue was that the submission by GEC Marine and Industrial Gears was finally selected. Further development of this particular design has, however, been suspended pending a review of the future programme. The subsequent steps in the design and production programme are set out in the following paragraphs.

Gearing Design Development

The starting point for the purchase of any new equipment must be the Procurement Specification itself. During the period leading up to the order for the gearbox, an extensive series of feasibility studies are undertaken by the Ministry and selected manufacturers to establish the best possible configuration of propulsion machinery. The output from these studies—shaft power and speed, the positions of the engines, the rake of the shafts, location of the drain tank, and so on—forms one major set of inputs to the Procurement Specification. These items are specific to the ship and will be different for each new design.

In addition to this design definition, however, the Procurement Specification must also contain, or at least make reference to, three other sets of information: it must specify the technical standards to which the design must conform, it must define what other information tools and support equipment the Ministry requires, and, perhaps most important of all, it must make clear to the contractor how the Ship Department will be involved in the approval of the design as it progresses. As much of this information is common to most gearbox designs, it can conveniently be published as a Naval Engineering Standard (NES).

During the past two years, D151 has devoted considerable effort to producing a comprehensive gearing NES. This has raised some important questions related to technical details and to our methods of doing business. As far as the technical matters are concerned, chief consideration has been given to those areas which have been the source of most trouble in the recent past: fasteners and the means of locking them, instrumentation, cleanliness and preservation, and non-destructive testing are five significant areas examined. On the administrative side, the writing of the NES has provided an opportunity to define thoroughly the MOD's requirements for design assurance, for reliability and maintainability, for support, and for the control of design changes.

The starting point for the definition of the support aspects of a new project is *Ship Department Publication 10, the MANDUS manual (Management of Design for Upkeep and Support)*. The principal tasks which are the direct responsibility of the Equipment Design Sponsor are listed in TABLE I. Even this long list is by no means all embracing and many further items could be added. Most of these tasks require some input from the gear manufacturer and the NES defines in detail what he is expected to provide.

As part of the selection process, the competing manufacturers are required to produce a Design Submission Report. The first step following placement of the order is to update this report in the light of the final procurement specification and to extend it to include detailed programmes for design and manufacturing tasks. This report after presentation to D151 will be critically discussed with the firm, a process which is formalized as a First Design

TABLE I—*Responsibility of equipment design section*

Upkeep codes
Maintenance schedule to agree with codes
Adequate spares and tools called up
Establishing training needs
Initiating maintenance evaluation
Adequate information in datum pack
Establishing removal routes and maintenance envelopes
Establishing requirements for A.S.Es.
Ordering drawings in time for codification action
Checking that master record is accurate

Review. This review will take place some eight to ten weeks after the order is placed. During the review, the Ministry will approve the report and the programmes it contains as the basis for detail design.

After this Review, the manufacturer will start to order the heavy forgings, the plating, and some sub-contract components which form the long-lead items. During the detail design phase (which is likely to last about eight months) progress will be monitored by a series of monthly meetings at which outstanding queries are resolved. When the manufacturing drawings are nearing completion a Second Design Review will be held at which the issue status of the drawings to be used for manufacture is confirmed. After this review any further changes to design must be subject to formal approval by the Ministry using the Design Alteration Procedure based on the DNSP 22 Form.

Although the design is nominally complete by the time of this Second Review, experience has shown that a considerable percentage of the detail drawings will still be outstanding as will much of the other design and support information required by the NES. Design activity is therefore expected to continue well into the manufacturing period.

As far as manufacture is concerned, the fabrication of the gearcase itself is likely to form the critical path with the fabrication and machining of the main gear wheel running closely behind. Manufacturing times for main gearing are fairly long—two years being a typical figure—but the advent of new machine tools may well offer useful reductions.

The programme shown in FIG. 1 indicates a nine months installation period of work at the Shore Test Facility before the ship gearbox is ordered. Numerous modifications will arise during build and setting to work and a Third Design Review to agree the precise alteration status that will be used will be held one month before the ship box is ordered. A Final Design Review will be held one month before this box is delivered; at this, the handover package to the shipbuilder is agreed and the Modification State Zero defined. After this date the only mechanism for changing the design is the Ship Department Equipment Modification Procedure.

Gear Testing

Before leaving the subject of the gearing it is perhaps pertinent to discuss the role of testing. Because of the powers involved, main gearing suppliers do not normally possess the facilities required to subject a main gearbox to a realistic set of load conditions. Any customer must therefore choose from three courses of action which provide increasing levels of assurance but carry increasing cost penalties: a simple spin test at the manufacturers' works; a back-to-back test also at the manufacturers' works in which a torque is locked into two gearboxes coupled together by suitable devices; or a full shore test facility.

As far as naval main propulsion gearing is concerned our knowledge is such that given an accurate description of the load we can design most types of components with a high degree of confidence that they will not fail. Notwithstanding this we continue to experience embarrassing failures when new designs enter service. There are several reasons for this:

Omission of particular load cases during design.
Underestimate of level of load.
Manufacturing errors.
Detail design errors.

The prime aim of any gearbox test programme must be to search out such faults. Testing may also be required to build confidence in those critical areas of design where analytic techniques are less well founded; perhaps the most obvious of these being high-speed line vibrations.

The simple spin test provides a useful demonstration that the gearbox has been completely and correctly assembled. As such it is normally applied to every gearbox before delivery. Unfortunately it provides no information on the ability of the design to carry load.

The usefulness of back-to-back testing varies with the configuration adopted. It can subject at least part of the gearing to representative load conditions and, as such, is likely to reveal some of any defects present.

At first sight, full shore testing of the main gearing may seem to be using a sledgehammer to crack a nut. The likely number of defects in a new gearing design will not be great and, even if it could reveal them all, it would be difficult to justify a shore test programme purely on the ground that it improved gearbox reliability. However, the introduction of a shore test programme brings many other benefits in its wake. It freezes the propulsion system design at an early stage in the project and so maximizes the time available for the development of the support package. Drawings, BRs, and maintenance instructions are all usefully advanced. Spare gear can be more fully defined and better organized. Installation and maintenance practices can be refined and demonstrated. Numerous trials can be performed which form a solid data base that is of inestimable value when analysing problems later in life.

The cost and time savings to be made from not having a shore test are very tempting and it is obviously necessary to examine each case on its merits. Nonetheless, having observed the benefits that have flowed from the CAH shore test facility, it is believed that shore testing will pay for itself quite handsomely on any significant Class of ships with newly designed propulsion machinery.

Controllable Pitch Propeller Design and Development

Looking now at the propeller end of the design, the procedure is different for a number of reasons. Firstly, as already said, there is only one national manufacturer. Secondly, the design of the propeller is divided between two agencies—the manufacturer and AMTE Haslar. Through D151, the manufacturer is responsible for the mechanical design of the propeller and operating system, and AMTE is responsible for the hydrodynamic design of the blades and hub. The two meet at an inter-face somewhere near the blade root and here the manufacturer specifies the stress levels for the blade material and AMTE provides blade design to sustain the stresses.

During the concept stage the main parameters of the propeller will be established—principally the diameter and rotational speed. These, together with loading data from manoeuvring simulations, will provide enough information for the manufacturer to produce a first estimate of the propeller design making some assumptions about the blade design. This will provide sufficient detail to identify whether there are weak areas of sufficient divergencies from present practice to justify further development work.

AMTE cannot start work in earnest on the hydrodynamic design until the hull lines have been established and full model testing carried out. Using information provided by the manufacturer on the hub size necessary to house the blade servo motor and hold the blade roots, the blading can be refined and a model propeller built and tested to give a final estimate of the blade loadings for the ship. These will be fed back to the manufacturer and, hopefully, they will not be too far from the first estimates he made at the start of the process. The manufacturer can then complete his design and produce manufacturing drawings for all parts including the blades. Production for the first of class is then possible.

During the AMTE design development phase the manufacturer will not have been idle since design of the CPP hydraulic system and its controls will be progressed. Future designs will be based on the open circuit hydraulic system development for later Type 42s and 22s. These arrangements are constrained by the need for retrofit into an existing design of ship. There will be differences when a design is prepared specifically for a new project. It is intended that any new system will be set up ashore and proved on a test rig which will include a dummy propeller and the intended servo motor. Together with its associated plant control unit, this makes a convenient and useful shore test package and eliminates the need for duplication in the whole ship/shore test facility.

Shafting Design

To connect the gearbox to the propeller a length of shafting is necessary together with intermediate bearings and the hydraulic tubing running in its bore. This again is the propeller manufacturer's design and supply responsibility. Design is straightforward to well established rules. There may be room for debate over the disposition of the intermediate bearings. From experience, it is preferable to avoid more than one outboard bearing and oil-lubricated stern seals which tend to turn themselves into water-lubricated ones. Plummer blocks are also preferred to roller bearings. Two major suppliers are available for both plummer bearings and face-type stern seals. Healthy competition can therefore be encouraged.

Reversing Gears

There is a need to determine a clear development policy for the immediate future. Before discussing the ways and means of reversing, it is worth considering the need in order to establish the potential benefits. The disadvantages of following an exclusive CPP policy are summarized below:

(*a*) *Noise:* most of our current experience shows that there is little to choose between the underwater noise levels of fixed-pitch propellers and CPPs. The use of agouti improves both types but a pump jet propulsor is best of all in this respect. At the extreme ends of the frequency spectrum, the CPP tends to be noisier. If operation occurs away from designed pitch, the noise performance is likely to get substantially worse. This tends to suggest that its use in conjunction with diesel prime movers might lead to a significant noise penalty.

(*b*) *Efficiency:* the CPP is less efficient than the equivalent fixed-pitch propeller (FPP) because of the diameter of the hub, the thickness of the blade roots, and the restriction of the Blade Area Ratio by the need for the blades to pass each other at zero pitch. Estimates of the difference vary but a typical figure is 5 per cent. Reversing transmissions associated with FPPs might reduce this advantage by virtue of built-in mechanical losses, showing a net penalty against the CPP of 4 per cent. In the days of highly-priced and scarce fuel, this loss might be significant in ship design terms.

(*c*) *Availability:* if anything goes wrong with the underwater or shaft-mounted parts of a CPP system, there is little alternative to docking the ship for repairs. An all-inboard reversing system has a distinct advantage in this respect. Our early CPP systems now at sea have been found to have a number of unreliable underwater features that have given rise to several unprogrammed dockings. These have brought CPP systems into a certain amount of disrepute. Modifications are being introduced to solve these problems but the final solution cannot be cheap and will take time to implement.

(*d*) *Single Screw Reliability:* with the above problems in mind reliability estimates for a future single-screw ship will be coloured by the CPP system's somewhat chequered history to date. Until present unreliable features can be shown to have been eliminated, a single-screw CPP warship seems risky in reliability terms. Strangely enough, the Navy's first single-screw CPP warship, H.M.S. *Exmouth,* was in no way troubled by propeller problems. She had a standard 'commercial' CPP system.

(*e*) *One National Source of Supply for CPPs:* the implications of this situation are obvious. Naval requirements must be tailored to reflect what industry can provide. It should also be noted that there is only one national source of supply for fixed-pitch propellers.

(*f*) *Minimum First Cost:* CPP propellers and their supporting systems are by no means cheap. Reversing gear systems might be cheaper. The development work that would be necessary to achieve and prove the reversing capability must, however, be taken into account. This might cost as such as £4M on present estimates. That sum is difficult to justify in our current climate of financial stringency. Taking all these points together, they do not make an overwhelming general case for more reversing gear development.

Considering now the ways of achieving a reversing transmission, the methods used in TRIBAL Class frigates, COUNTY Class destroyers, and the CAH have been well publicized[1,2]. All these designs use fluid couplings which act as clutches to engage selected gear trains to give either ahead or astern output rotation. The power that a fluid coupling can transmit is limited by its physical size so that a compromise has to be made between the power required in manoeuvring drive and the size of gearbox that can be fitted in the ship. The manoeuvring power having been so determined, the next problem is to provide an alternative direct-drive transmission line which bypasses the power-limited fluid couplings when manoeuvring drive is not required.

In the TRIBAL and COUNTY Classes, the direct-drive lines can only be selected by changing the arrangement of a number of manually-operated clutches. This is quite acceptable in a COSAG installation where the primary drive is a reversible steam turbine. In the CAH, an automatic facility to change from one drive mode to another is provided. The same feature will be necessary in future gas-turbine ships of similar configuration. While the size and weight penalty entailed by such an arrangement is tolerable in the CAH,

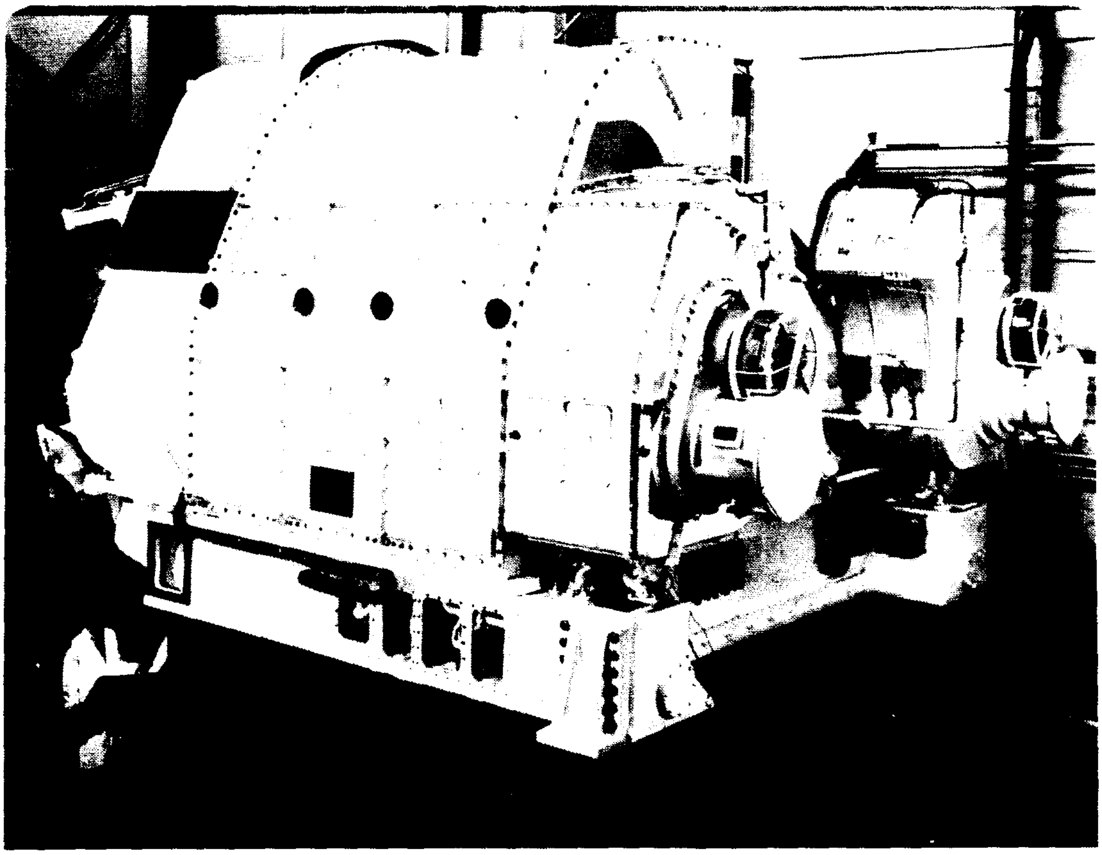

FIG. 2—THE CAH GEARBOX

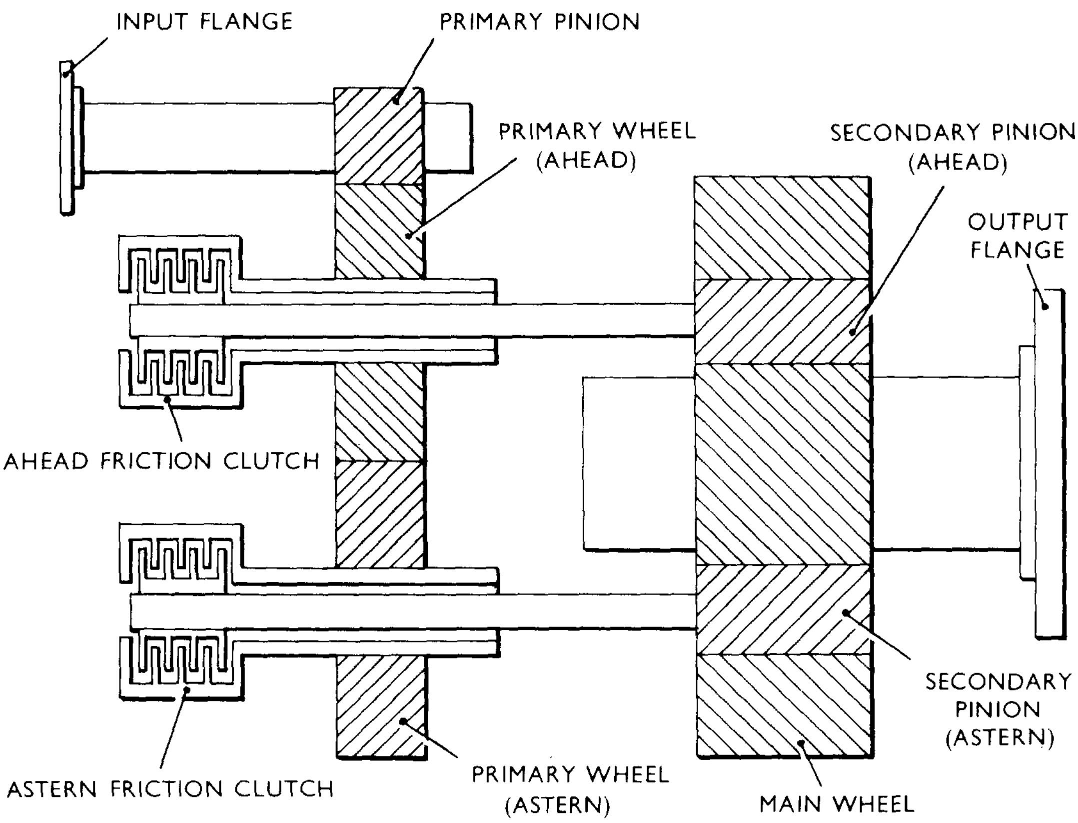

FIG. 3—SIMPLE GEARBOX USING FRICTION CLUTCHES

the size of the gearbox can be judged from FIG. 2; this same is not true in ships of frigate and destroyer size.

What, therefore, are the alternatives for a small ship? There are, of course, some indirect means such as hydraulic or electric drives which are possibly feasible now or might be in the future. These avenues are being explored elsewhere in the Ship Department.

Of the directly mechanical drives available, epicyclic gears have considerable merit since, by braking or releasing elements of an epicyclic gear, the output may be reversed. This avenue was thoroughly explored by the Ship Department in the mid 1970s and has not been pursued further, principally because the brake technology necessary would be expensive and time consuming to develop. Epicyclic gears also lose much of their attractiveness where two or more engines have to be combined to drive one shaft.

Friction clutches of the size necessary to handle the power of current gas turbines are now on the market in the U.S.A. The U.S. Navy is already testing them and one, at least, shows promise. Friction clutches might replace fluid couplings in our present arrangements and eliminate the need for direct drives to bypass them. A simple gearbox using friction clutches is shown in FIG. 3. Its operation is straightforward but, in ahead drive, for instance, the astern gear train still rotates, an unhappy situation for the unloaded gear meshes, and the astern clutch elements are rotating counter to each other at up to full speed—a condition known as 200 per cent. slip. Trials have shown that many friction clutches do not behave well in this condition, apart from being intolerant of repeated engagement and disengagement. Remembering that to get a ship up to a buoy in adverse conditions might demand a hundred engine movements, the prospects for high-power seagoing friction clutches are not good.

The latest reversing device available is the reversing hydraulic coupling now being tested by the Italian firm of Franco Tosi. In essence, the design concept is very simple and is shown diagrammatically in FIG. 4. It consists of a conventional hydraulic coupling made up of driving and driven elements in a casing filled with oil. When the coupling is in use, there is a continuous flow of oil through the casing from the centre, through the elements, to the outlet ports round the circumference. The drive is disconnected by shutting off the oil supply so that the coupling empties itself. Round the periphery are arranged a number of radial vanes which can be inserted into the oil path between the two elements. The vanes are shaped so that they change the direction of the oil circulation, thus reversing the direction of rotation of the driven element. While it is not as efficient as a good fluid coupling, it transmits about the same power as one coupling from a CAH gearbox but it is considerably smaller and lighter. Testing of a full-size coupling (1 m diameter) is well underway in Italy and developments are being monitored carefully. FIG. 5 shows the latest test rig under construction with the coupling casing top removed. The two elements and one blade in the periphery can be seen. FIG. 6 shows the rig being assembled and its associated steam turbine, gearboxes, and brake. The tops of some of the vane actuators (26 in all) can be seen.

The coupling's power transmission capacity is limited by its size. The familiar compromise in choosing the balance between size and manoeuvring power is necessary. A bypass drive line is therefore still necessary to achieve full-ahead engine power in typical applications.

Two warship designs featuring this coupling, are already in hand. The first is by the Italians themselves intended for their version of H.M.S. *Invincible,* called the *Guiseppi Garibaldi.* The prime movers are two GE LM 2500 gas turbines on each of two shafts. Each engine has a reduction gear train with a coupling and clutch in the intermediate speed line as shown in FIG. 7. This is

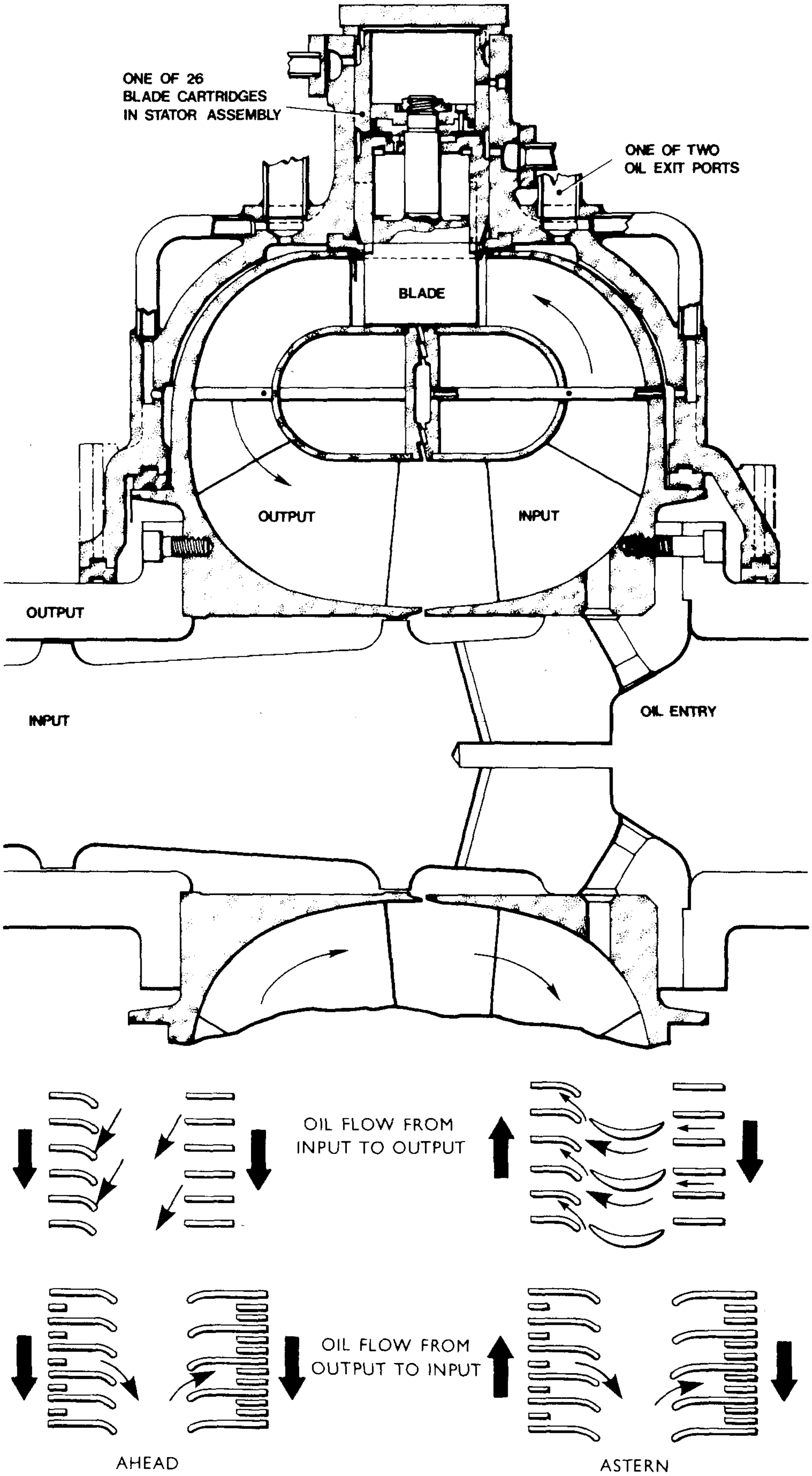

FIG. 4—FRANCO TOSI HYDRAULIC REVERSING COUPLING (SHOWN IN ASTERN RUNNING POSITION) —SECTION THROUGH OIL CIRCUIT AND DIAGRAMS SHOWING OIL FLOW

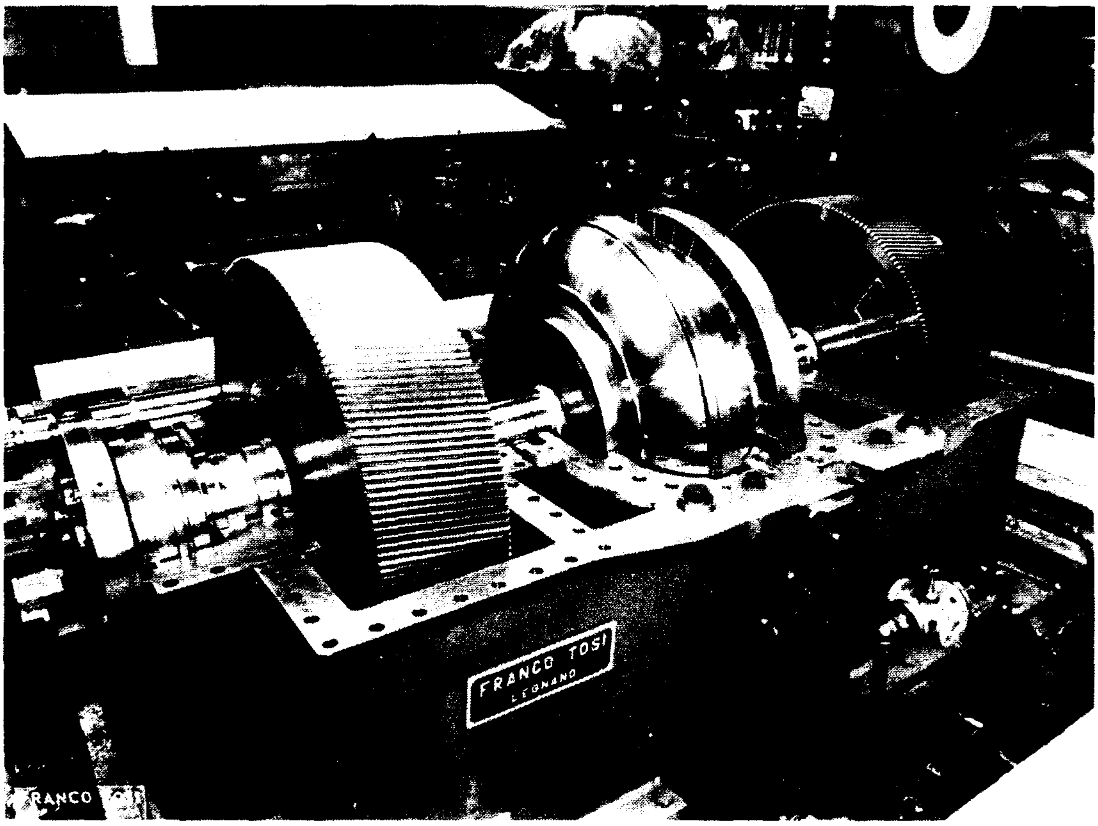

FIG. 5—1 METRE FRANCO TOSI COUPLING TEST RIG UNDER CONSTRUCTION

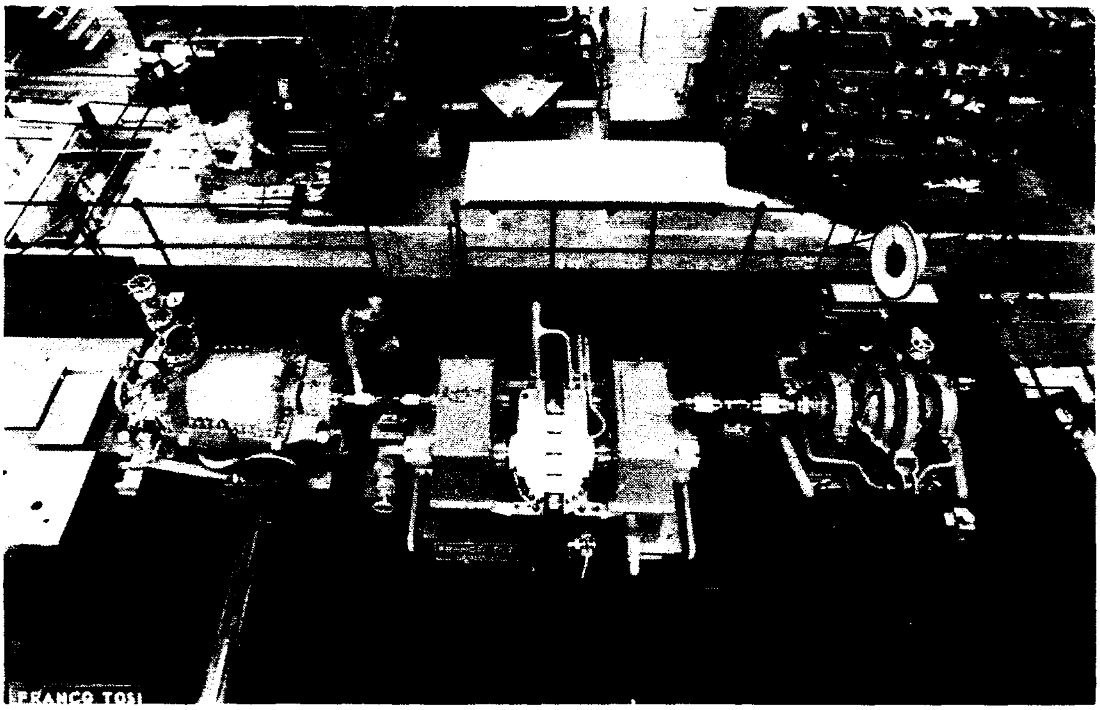

FIG. 6—1 METRE FRANCO TOSI COUPLING TEST RIG BEING ASSEMBLED WITH ITS ASSOCIATED STEAM TURBINE, GEAR BOXES, AND BRAKE

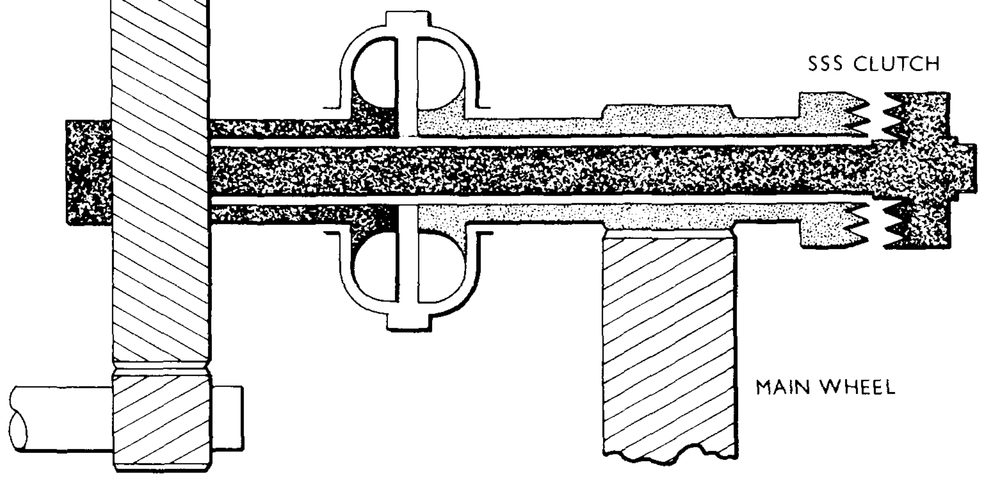

FIG. 7—ARRANGEMENT OF FRANCO TOSI COUPLING IN THE 'GARIBALDI' GEARBOX

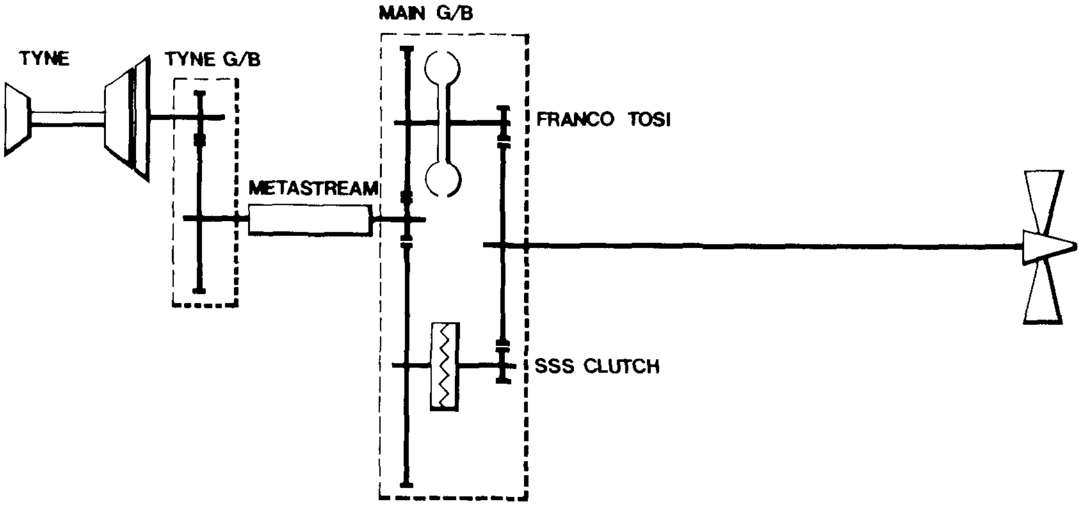

FIG. 8—ARRANGEMENT OF FRANCO TOSI COUPLING IN THE 'TARANAKI' GEARBOX

the simplest possible arrangement but it involves some complexity of the control system to achieve bumpless changes between manoeuvring and direct drive. When changing from manoeuvring to direct drive, for example, the output element of the coupling initially runs slower than the input, as a result of hydraulic slip. The relative rotational speeds of the synchronizing clutch are therefore in the wrong sense for it to engage. This is overcome by momentarily reducing the power of the engine and applying an input line brake until synchronism is achieved and the clutch engages. This can be disconcerting to a captain who rings on 'Full Ahead' only to see his engines reduce to idle power, if only for a few seconds. The arrangement does have the saving grace that coupling filling and emptying operations are only carried out when the two rotating elements are synchronized and stability problems do not exist.

The second design in hand has been devised by Vospers for a replacement gearbox for the conversion of a Royal New Zealand Navy Y100 frigate

Taranaki from steam to gas-turbine drive. A description of the initial proposals has already been published [3]. A contract for the gearbox has been placed with GEC Marine and Industrial Gears. In this ship, the steam plant is to be removed and replaced by one Tyne RM1C engine on each shaft. The arrangement of the gearing is shown in FIG. 8. Here the input drive is split between the intermediate speed lines with the clutch in one line and the coupling in the other. The ratios are arranged so that, in direct drive, a slip of about 30 per cent. is imposed on the coupling. This arrangement ensures that the relative rotational speeds of the synchronizing clutch are always in the correct sense for engagement or disengagement. Bumpless transfers between direct and manoeuvring drive are therefore guaranteed simply by filling or emptying the coupling. This is a simple operation to arrange in control terms with the added virtue of foolproof operation in manual control. There are some potential stability hazards in filling and emptying the coupling with an imposed slip and these still have to be explored.

At present (January 1981) testing is still not complete but the Ship Department is working closely with the British marketing licensee, SSS Gears Ltd, and the manufacturer to evaluate trials data as it is being generated. With the assistance of Y-ARD Ltd., a comparative study of various reversing schemes is in hand with the aim of making a full evaluation of the Franco Tosi coupling against its competitors in a given ship design. Future prospects for the use of this coupling in the Royal Navy will depend on the existence of a suitable application and the availability of funds to carry out any necessary further development in this country.

Concluding Remarks

The design of transmission systems for future warships remains an absorbing task. Whilst no radical changes are envisaged in the immediate future, many lessons have been learnt from our troubles of the recent past. These have been recorded in the new *Naval Engineering Standards for Gearing, Shafting, and Lubricating Oil Systems* that are shortly to be published. Section D151 and its contractors are confident that by working together the future fleet can be provided with simple, robust transmission systems which are properly documented and supported, and which require the minimum of effort to set to work and to maintain.

Acknowledgement

The photographs of the Franco Tosi test rig are reproduced with the kind permission of Franco Tosi, S.p.A. Legnano, Milan.

The views and opinions expressed are those of the authors and do not necessarily represent those of the Ministry of Defence.

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