THE TYPE 23 BOOST PROPULSION GEARBOX

BY

B. C. Cooper, B.Sc. (Eng.), R.C.N.C.
(*Sea Systems Controllerate*)

Introduction

The development of the overall engineering design of the Type 23 frigate has been covered in an earlier issue of the *Journal*[1], the salient points of the main propulsion system being as follows:

system designation:	CODLAG (combined diesel electric and gas turbine)
number of shafts:	two
cruise drive:	main motor on propeller shaft
boost drive:	Spey gas turbine connected to gearbox
reversing:	by main motors only

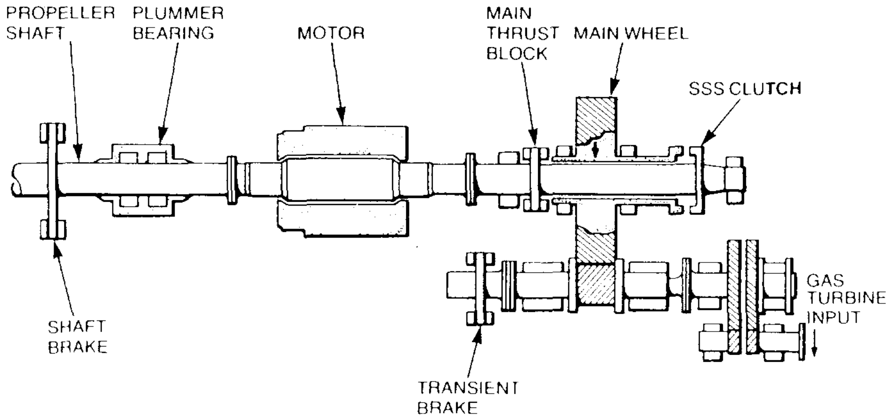

Fig. 1—Arrangement of motor and gearbox (port shaft)

The propulsion arrangement is shown in Fig. 1. Normal cruise operation and astern operation are provided by electric motors which are powered from the ship's generators. In this condition the gearbox and gas turbine are disconnected from the shaft line by the SSS clutch—in its 'locked out' position—and all gear elements are stationary. To minimize noise generation the forced lubrication system is not used in motor only drive, the thrust block, SSS clutch bearing and shaft bearings being 'splash' lubricated.

For higher ship speeds the Spey is connected to the propeller shaft via the gearbox and SSS clutch and can provide power either on its own or in combination with the electric motor.

General Description of Gearbox

The gearbox configuration is shown in Fig. 2. The Spey power turbine drives directly into the primary pinion via a Metastream type high speed coupling. The pinion meshes with one primary wheel (i.e. it is a single train not a locked train gearbox) and this is in turn connected through a quill shaft and flexible coupling to the secondary pinion. Drive then passes via the main wheel to the input half of the SSS clutch at the forward end of the gearbox. The clutch output is connected to the output shaft, part of the propeller shaft line, which is quilled through the main wheel and, at its aft end, carries the main thrust collar within the main thrust block. The aft flange of the shaft is connected to the main motor armature, the support for which is provided by journals in the thrust block and first plummer bearing.

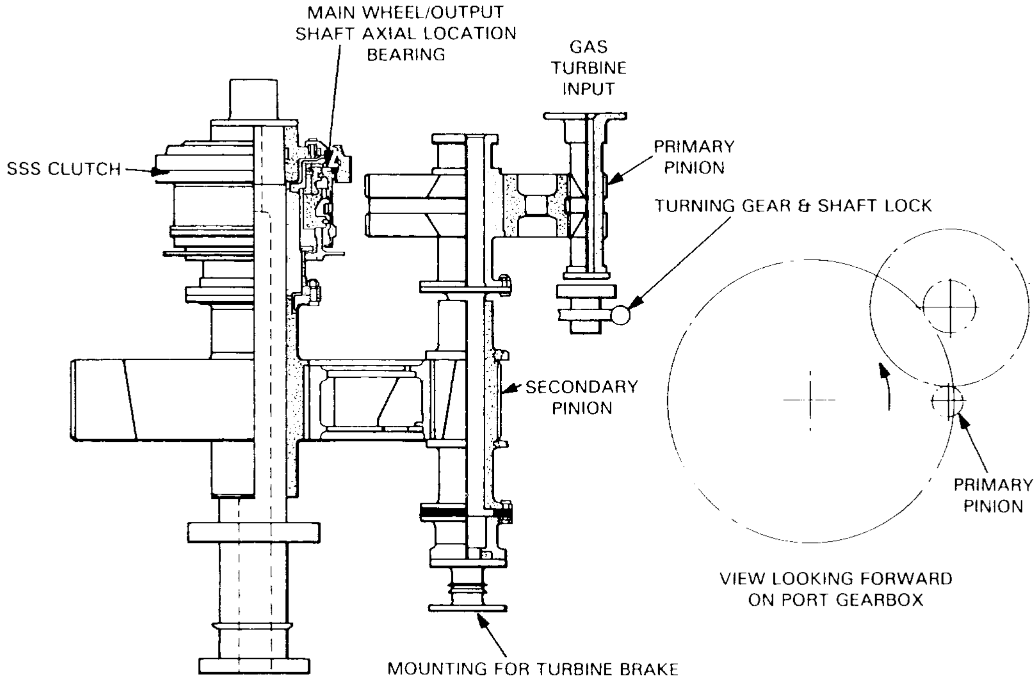

Fig. 2—Type 23 gearbox configuration

Gearbox Construction

Gear Elements

The primary gears are double helical and the secondaries single helical. Both primary and secondary pinions are forged integral with their shafts, the gear teeth being carburized. The primary wheel is also forged, with carburized teeth, but is shrunk on to a carbon steel shaft. The main gearwheel comprises a rim in which the teeth are cut, these being induction hardened, bolted to a fabricated centre which is in turn shrunk and keyed on to the main wheel shaft. It is the use of an induction hardened main wheel that has permitted the adoption of a single gear train, by virtue of the higher permissible tooth loading.

All gear teeth are finish ground to achieve the accuracy of toothform necessary for both good load distribution and minimum noise generation.

Journal Bearings

All journal bearings are of standard 'medium wall' construction, liners being provided where necessary to facilitate adjustment of bearing alignment during build.

Thrust Bearings

The primary gears, being double helical, generate no external axial forces and therefore require no thrust bearing. A location bearing on the primary wheel forward journal maintains the position of the gear pair within the gearcase.

The single helical secondary gears generate a considerable axial force and this must be balanced if equilibrium is to be maintained. In previous gearbox designs (e.g. COUNTY Class and SSNs) the axial load resulting from a single helical gear mesh has been absorbed by a conventional tilting pad thrust bearing, an arrangement which works very well; it does however have some disadvantages, notably:

(*a*) Increased maintenance load.

(*b*) Possible source of failure.

(*c*) Small reduction in gearbox efficiency.

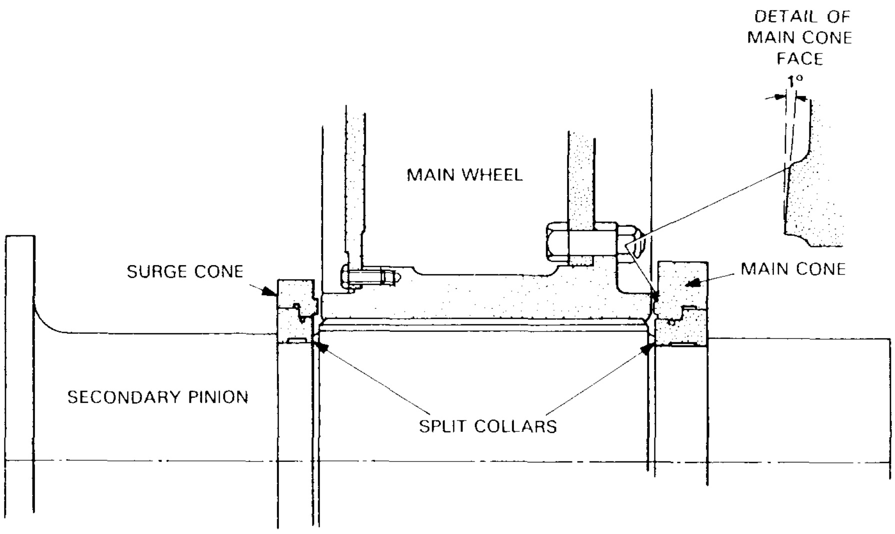

FIG. 3—ARRANGEMENT OF SECONDARY GEAR THRUST CONES

In the Type 23 design a different approach has been adopted, involving the use of 'thrust cones' (FIG. 3). Essentially the thrust cone comprises a flat surface machined on the side of the main wheel rim which contacts a similar face on a collar shrunk on to the pinion shaft. Since the axial load is now effectively generated and absorbed at the same radius, there is no longer a significant overturning moment and the gears behave in a very similar manner to a double helical pair. The somewhat complex arrangement of fitting the cones on to the pinion shaft, involving a collar shrunk on to a split ring, is necessary because it would be impossible to grind the gear teeth if they were forged integral with the shaft.

The actual method adopted for fitting the cones to the pinion shaft is in fact one of the most important aspects of the design, and in the case of Type 23 it was demonstrated in a test rig for approximately 500 hours at full load and one hour at 150% load.

From FIG. 3 it will be seen that the term 'thrust cone' is something of a misnomer, the cone angle being only 1°. However this is sufficient to create the convergent/divergent passage between the cone surfaces which in turn generates the hydrodynamic oil film and prevents metal to metal contact.

Thrust cones are provided at both ends of the gears although only one, the main cone, is subject to heavy load. The other, the surge cone, provides axial location and accommodates small reverse loads seen during braking and resulting from ship movement when the gears are stationary.

In the case of Type 23 the cone surfaces on the main wheel are unhardened whilst the pinion thrust surfaces are nitrided. This 'hard on soft' combination has been adopted to minimize the risk of interaction if one or other surface is damaged in any way.

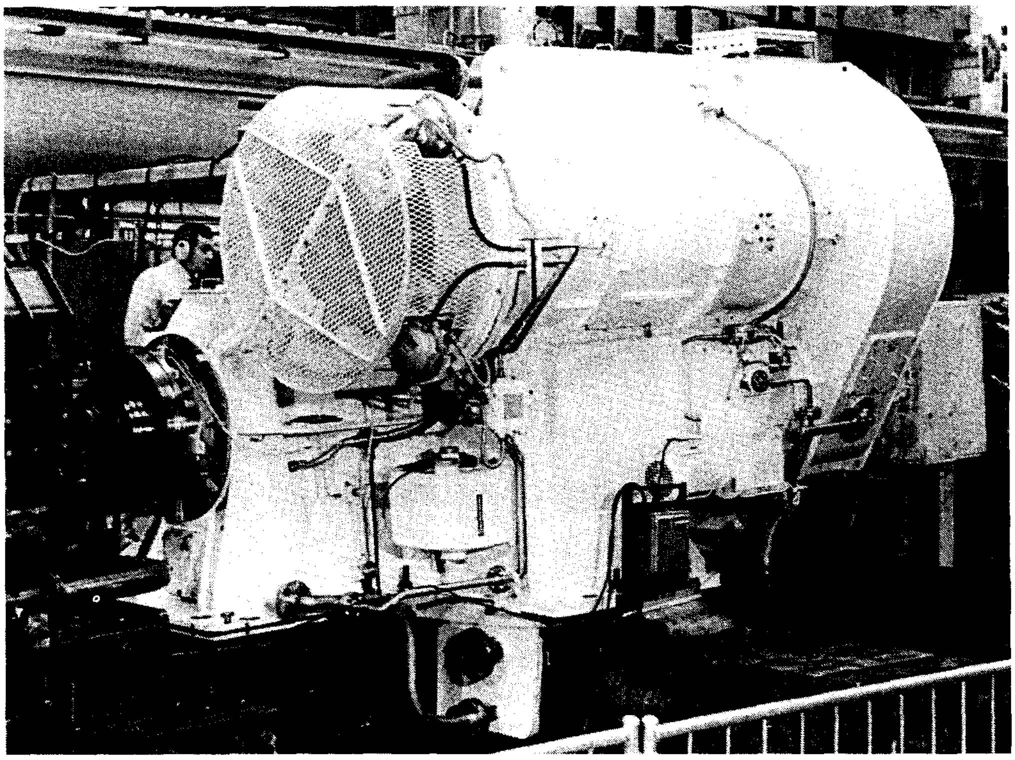

FIG. 4—TYPE 23.01 PORT GEARBOX

Axial location of the secondary gear pair is provided by the main thrust block via an internal thrust collar within the SSS clutch. To avoid the stationary main wheel being dragged forward and aft during ahead/astern manoeuvres, the clearance of the SSS internal thrust is greater than that of the thrust block.

The main thrust bearing is of conventional type having ahead and astern thrust rings. It incorporates thrust metering equipment and the facility for fitting a resonance changer if required.

Gearcase

For simplicity the gearcase (FIG. 4) comprises a single main fabricated unit into which all the gear elements are fitted. For manufacturing reasons the main thrust block is a separate item secured to the gearcase by a bolted vertical joint. Support from the hull is by three 'feet', one forward of the main wheel, one aft of the main wheel and the third offset from the line of these two in order to resist torque reaction forces.

Lubrication System

Gas Turbine Drive

The main gearbox and power turbine lubrication is conventional, being supplied from a forced lubrication system. One manifold supplies gear sprayers, and a second the gearbox bearings. Oil for the SSS clutch baulk ring and pawls is provided from the adjacent main wheel bearing feed via radial and axial drillings in the shaft.

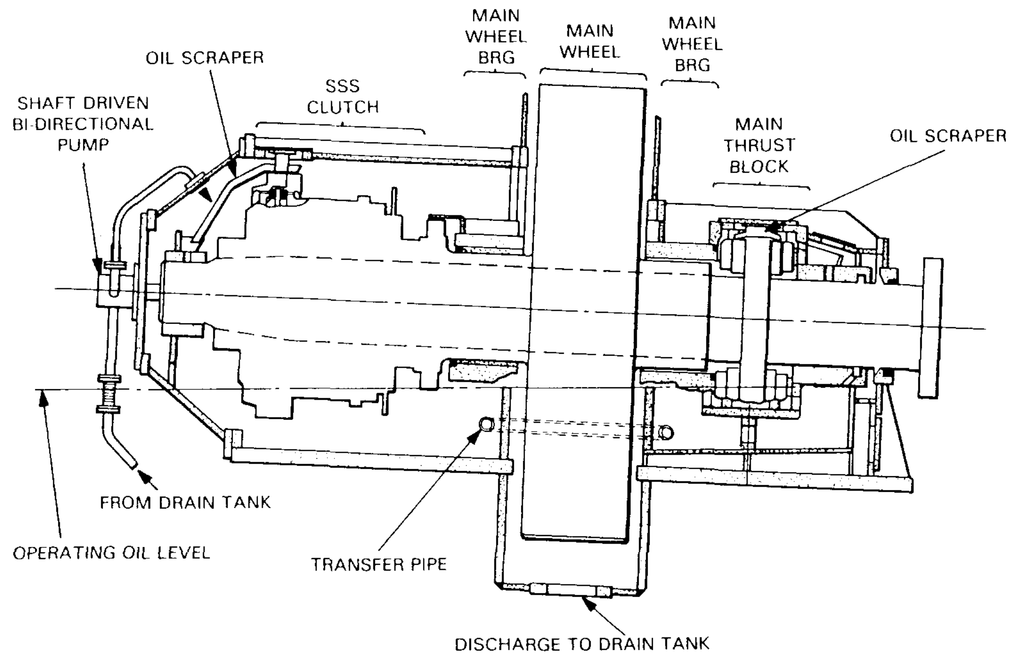

FIG. 5—LUBRICATION SYSTEM FOR MOTOR DRIVE

Motor Drive

The forced lubrication system is not required when in motor drive, the output shaft, SSS clutch thrust bearing, main thrust pads and shaft journal bearings being lubricated by oil picked up from oil baths as shown in FIG.5. The forward and aft compartments are connected to provide a self-levelling system. Because some oil will be lost due to splashing and ship movement causing spillage over the weirs, a small make-up pump, of 3 m^3/hr capacity, is provided, driven by a toothed belt from the end of the output shaft. The pump has the secondary function of maintaining an oil flow through the baths, removing heat and avoiding the need for direct cooling.

The motor drive lubricating oil system continues to operate in boost drive and is designed to meet full power lubrication requirements.

SSS Clutch

The SSS clutch is identical in its basic operation to any other in service, but is obviously much larger in order to accept main shaft torque. It is 36 inches in diameter at the main drive teeth. The input half of the clutch is bolted directly to the forward end of the main wheel shaft and the output half to the forward end of the main shaft, via an oil-injected taper hub.

The clutch is of the servo-controlled type, actuated by an externally mounted control cylinder which selects the 'locked out' and 'active' (i.e. pawls engaged or ratchetting) states; lub. oil is used as the servo operating fluid. No provision is made for 'lock in' under normal operation, but there is a mechanical lock which can be engaged for maintenance/emergency routines, e.g. to turn the shaft astern on turning gear.

Operational clutch states are:

(*a*) Locked out.

(*b*) Baulked (mechanical baulk to prevent pawl engagement when clutch input is rotating faster than clutch output).

(*c*) Pawls engaged/ratchetting.

(*d*) Engaged.

The size and position of the clutch means that removal and replacement is a major task. For this reason all likely wear points can be refitted *in situ*, e.g. pawls, ratchet ring, and thrust pads.

Brake

The gearbox brake plays an important role in propulsion plant control—see below—and accordingly must be highly reliable. The brake selected for the duty is a standard marine/industrial double disc, four caliper unit manufactured by Twiflex and operated by LP air. It is fitted directly on the end of the primary wheel quill shaft.

Surveillance and Control Instrumentation

A standard instrumentation fit is provided with the gearbox, e.g.:

(*a*) manifold lub. oil pressure;

(*b*) bearing temperatures;

(*c*) clutch position;

(*d*) brake on/off indicator;

(*e*) input bearing vibration;

(*f*) tachometers (output shaft speed measurement).

Additionally the first of class is fitted with vibration transducers on each bearing cap; follow-on ships will be fitted 'for but not with'.

A local control panel provides the facility for direct operation of both clutch and brake.

Operational Characteristics

Operation of the propulsion plant is basically straightforward, simply requiring the 'flick of a switch' to obtain motor drive or the pressing of a button to start the gas turbine. Unfortunately one small factor does complicate things somewhat, namely that the SSS clutch can only be engaged or disengaged when the input is running slower than the output (as with any SSS clutch). This means that, for example if the Spey is at idle, deselected, with the clutch input at say 80 r.p.m., whilst the main shaft speed is only 70 r.p.m., then the clutch cannot be engaged; it will 'baulk'. The control system is therefore programmed to monitor input and output speeds and, if

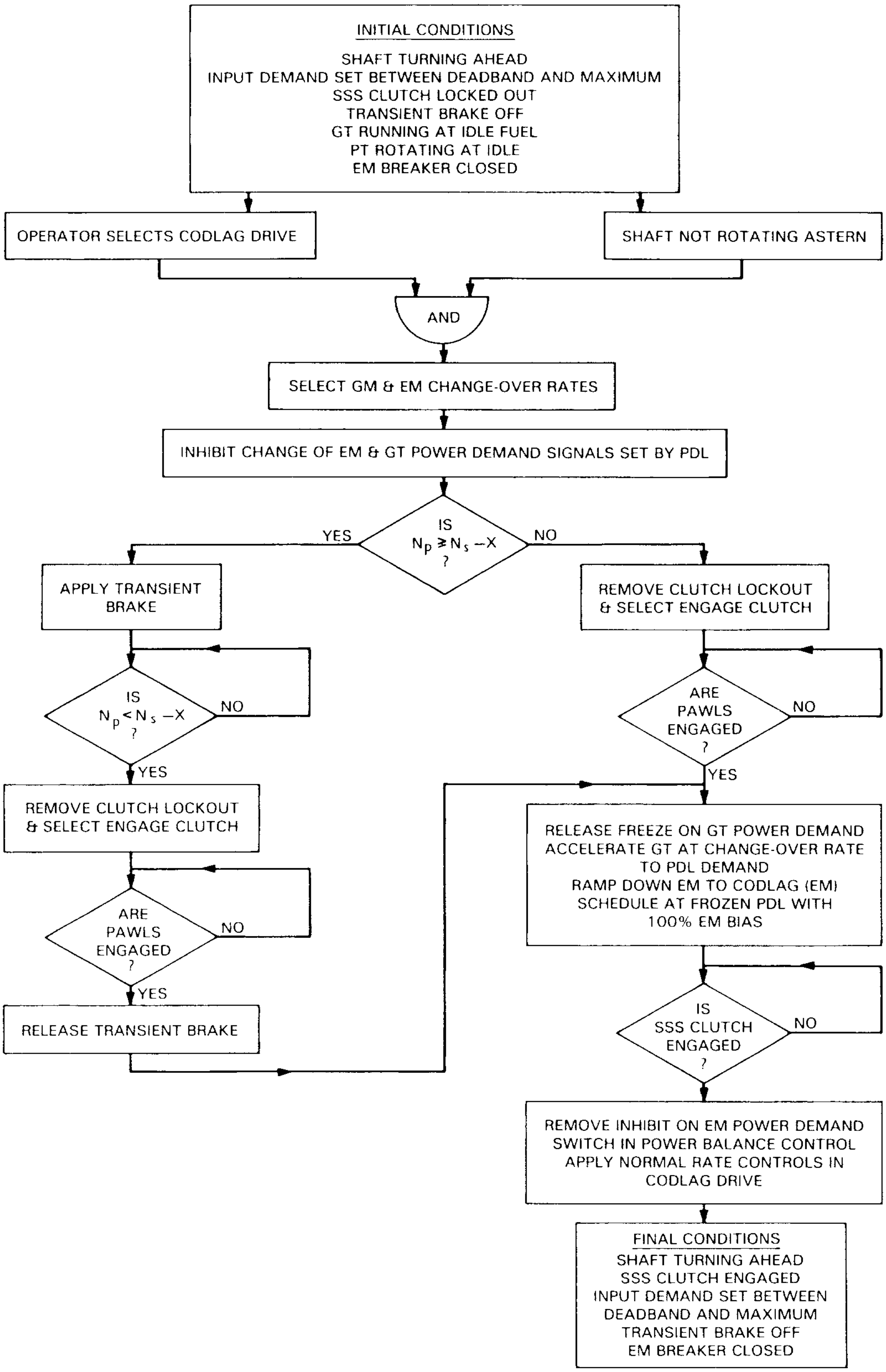

FIG. 6—PROPULSION PLANT CHANGE-OVER: ELECTRIC MOTOR TO CODLAG DRIVE

EM: electric motor
GT: gas turbine
N_p: clutch input speed
N_s: clutch output speed
PDL: power demand lever
PT: power turbine
X: deadband width

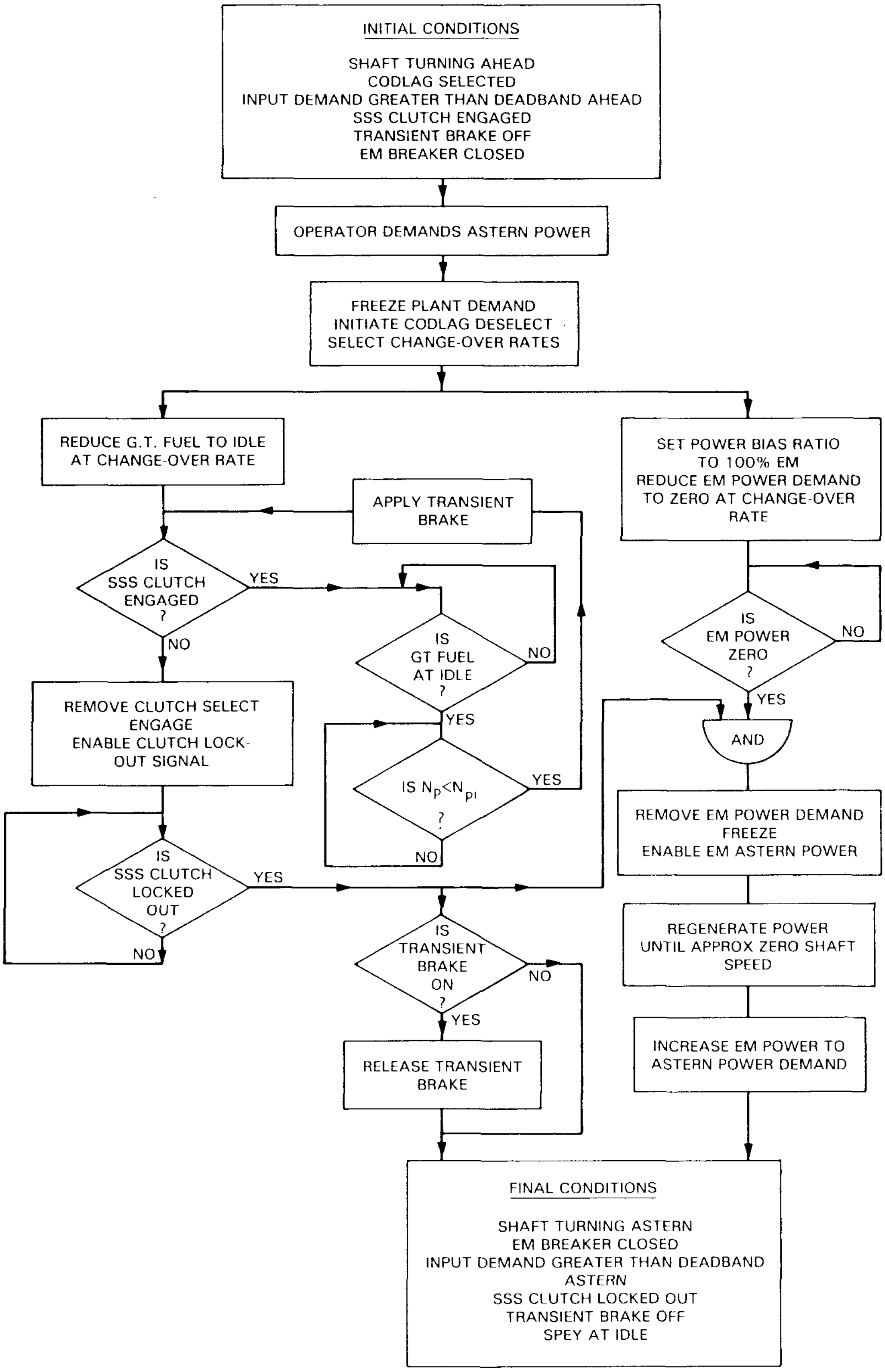

FIG. 7—AHEAD/ASTERN MANOEUVRE (INCLUDING CRASH STOP)
N_{pi}: max. gearbox speed for brake application

adverse, to apply the transient brake until input is running slower than output—see the logic sequence for changeover from electric motor (EM) drive to CODLAG in FIG. 6.

A similar circumstance arises if an astern demand is made when in CODLAG drive, i.e. the clutch must be disengaged before astern power is put on the motor. If gas turbine fuel is simply cut to idle then the propeller shaft might want to slow at a greater rate than the gearbox, and the clutch will remain engaged. The brake is therefore used to ensure that the gearbox decelerates faster than the shaft (see FIG. 7). (It should be noted that whilst neither the gearbox nor the Spey power turbine is designed for astern rotation as a normal operating condition, they can withstand limited periods in an emergency.)

Programme

The Type 23 frigate is the first R.N. warship to have been ordered under the policy of ' whole ship procurement' in which the contracted shipbuilder is responsible for the development of all 'specialized' equipment. In the case of the gearbox, a Statement of Requirements (prepared by MOD) was placed with Yarrow Shipbuilders Ltd. (YSL) in early 1982 as part of the contract for long lead items; under this contract YSL were given full responsibility for procurement of the gearbox, including design and necessary development work. Subsequently YSL invited tenders for design and manufacture of the first four ship sets of gearboxes from GEC Marine and Industrial Gears and David Brown Gearing Industries. Following a full appraisal of the two sets of proposals, GEC were awarded the contract in January 1983.

With detailed design commencing in January 1983, manufacture of the first gearbox took the programmed two years, production testing commencing in February 1985. The first ship set is now awaiting installation in Type 23-01 and manufacture of the next three ship sets is under way.

Acknowledgements

The author acknowledges with thanks the efforts of all those involved with design of the Type 23 gearbox—at YSL, GEC, and in the 23 Project Group.

Reference

1. Blackman, R. S.: Type 23 frigate—the engineering development; *Journal of Naval Engineering*, vol. 28, no. 1, Dec. 1983, pp. 5-15.