FRANCO TOSI REVERSIBLE CONVERTER-COUPLING TRIALS

BY

B. C. COOPER, B.SC. (ENG.), R.C.N.C.
(*Sea Systems Controllerate*)

ABSTRACT

R.N.-sponsored trials on a Franco Tosi Coupling are described, in both steady state and transient conditions. With the '79' design, the manufacturer's predicted performance figures were validated and no weaknesses of any significance were found.

Introduction

The operation, development and use of the Franco Tosi reversible converter-coupling (RCC) were described in the last issue of the *Journal*[1]. The R.N. test programme, though mentioned in the introduction, was not covered there. The present article deals with this—the R.N. test rig, the tests, and their results.

Firstly, as a reminder of the principle of the RCC, FIG.1 illustrates the main components—pump, turbine and stator vanes. In ahead operation, with vanes withdrawn, operation is as with a conventional fluid coupling, oil circulating between pump or turbine as shown in FIG.2. However with vanes inserted the direction of oil flow entering the turbine is reversed, causing the output to rotate in the opposite direction to the input—as FIG.3. Hopefully this brief description is sufficient to make sense of the remainder of this article.

Background

The Royal Navy has been interested in the potential of the Franco Tosi coupling from the early days of its development (in the latter part of the 1970s) and kept a close eye on the trials programme conducted by Franco Tosi at their works in Italy. However the first application of the unit might well have been in the Royal New Zealand Navy when it was decided that the steam propulsion plant in one of their LEANDER Class frigates (H.M.N.Z.S. *Taranaki*) should be replaced by Tyne gas turbines driving fixed pitch propellers through a reversing gearbox. Design and manufacture of the new gearbox was contracted to GEC Marine and Industrial Gears Ltd and the Franco Tosi coupling was selected as the most suitable reversing mechanism. Manufacture of both gearboxes was complete when the conversion programme was abandoned.

At roughly the same time Franco Tosi was itself producing the gearboxes for the Italian Navy helicopter carrier *Guiseppe Garibaldi*.

Returning to the R.N. interest—from the outset there was a desire to conduct our own trials programme, partly to verify the performance claimed by Franco Tosi but principally to establish any characteristics important to the integration of the coupling into a transmission system, i.e. any peculiarities in behaviour of which the gearbox designer should be aware.

First thoughts were to construct a test facility that could exercise the coupling over its full range of performance. Unfortunately it soon became evident that, even in the late 1970s, this would cost several million pounds. Here the matter rested until 1983 when it was decided to investigate the

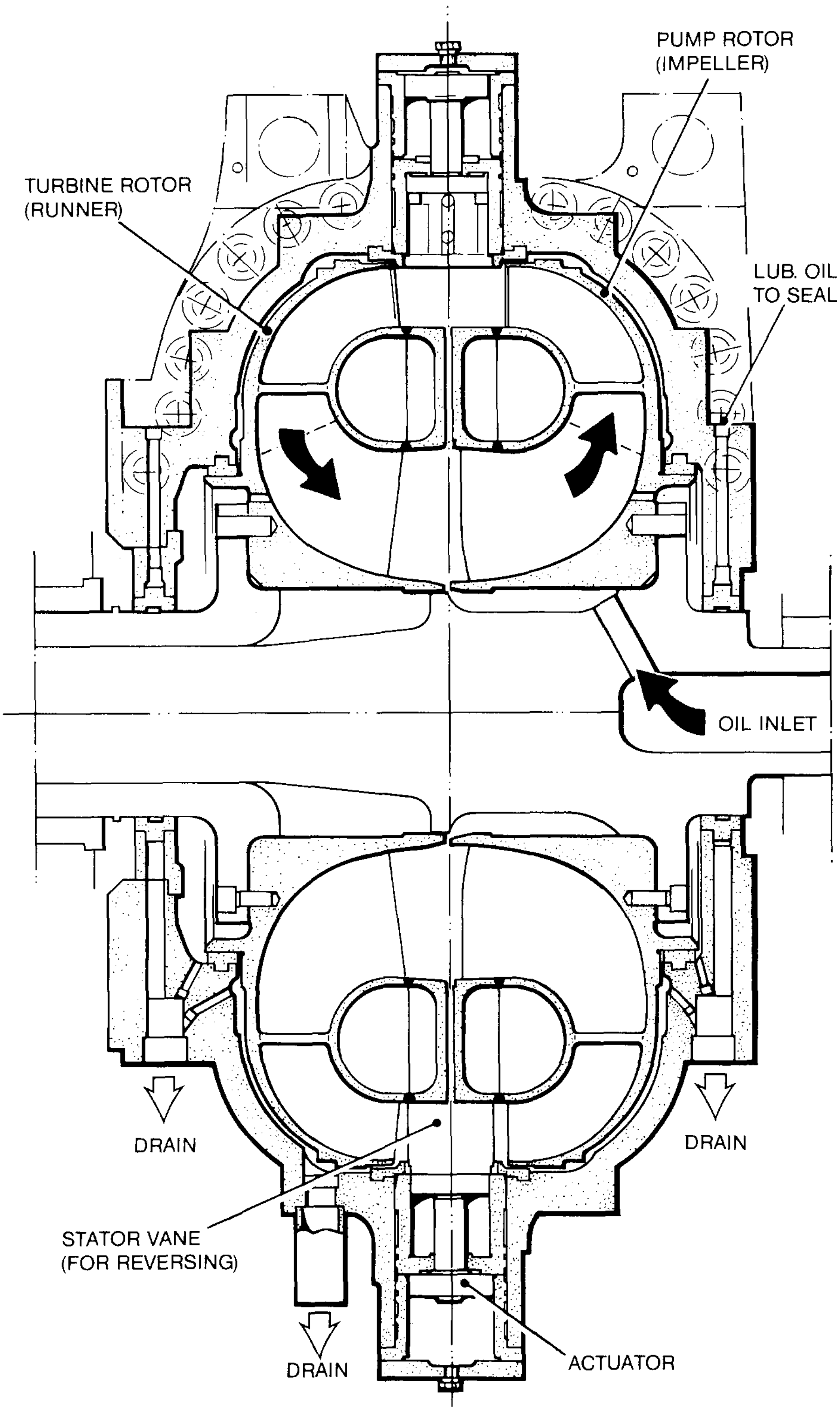

FIG. 1—SECTIONAL ARRANGEMENT OF REVERSIBLE CONVERTER COUPLING

possibility of a 'cheap' arrangement which, whilst not covering the full capability of the coupling, would provide 'value for money'. Accordingly rather than going out to contract with a specific technical requirement the approach adopted was to see what hardware and services could be made available by industry and assess:

(*a*) What test rig could be built from these?

(*b*) What range of test could be carried out?

(*c*) What additional equipment (and cost) would be involved?

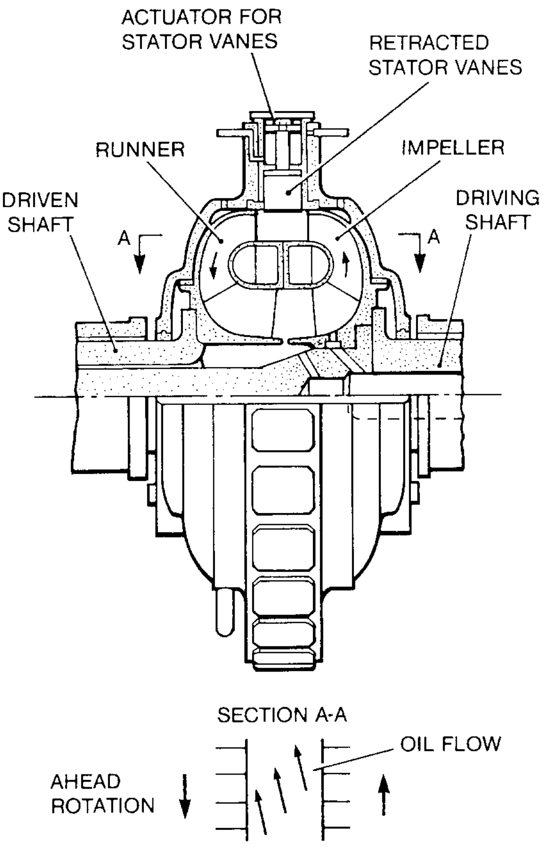

FIG. 2—AHEAD OPERATION

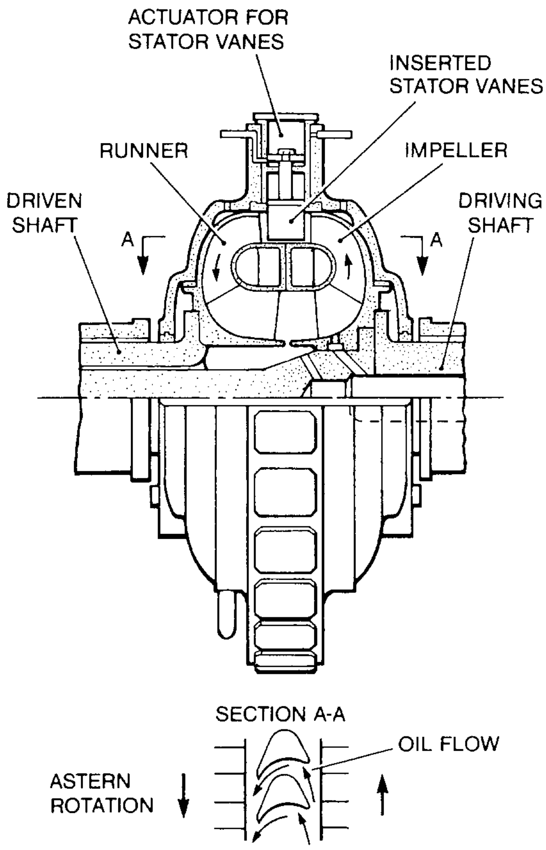

FIG. 3—ASTERN OPERATION

David Brown Gear Industries made the most favourable response, being able to provide an electric drive motor, dynamometer and other items together with a test bed and adjacent area for control room and services. Whilst the available motor and brake placed some major limitations on the range of trials that could be conducted (as will be discussed later) the overall capability was adequate. DBGI were therefore given the go-ahead to complete the design and construction of the facility.

Fortunately at this time the Franco Tosi couplings from the cancelled *Taranaki* project were available for sale and the opportunity was taken to purchase one for the R.N. programme (at a bargain price). The other unit was bought by the U.S. Navy to be incorporated into its FFG7 shore test facility at NAVSSES, Philadelphia.

Design of the Test Rig

A schematic layout of the main mechanical components is shown in FIG.4. The rig itself is seen in FIG.5. The main components are:

Item 1 4000HP Synchronous Motor

Capable of producing 3000kW at a constant 1000 rev/min.

Item 2 Fluid Coupling

To provide variable input speed to the RCC from the fixed speed motor.

Item 3 Flexible Coupling

Actually a Type 42 'Olympus' drive unit, used here not so much for its flexibility but mainly as a spacer piece between fluid coupling and gearbox to allow room for the fitting of an input flywheel required for a later phase of testing.

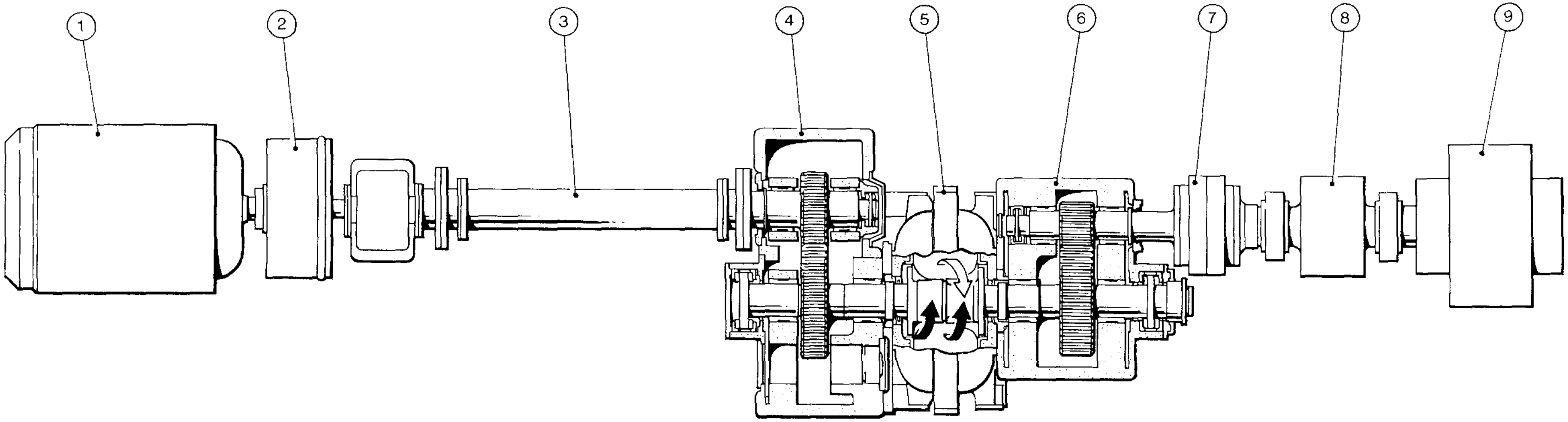

Fig. 4—Layout of main mechanical components for Phase 1 testing

1: 4000 HP synchronous induction motor
1000 rev/min fixed speed
2: Fluidrive 41 SCR Mk.4 scoop control fluid coupling
3: M4000 coupling (to provide distance)
4: DBGI purpose built 1.66:1 reduction gear unit
5: Franco Tosi 1000mm RCC coupling
6: DBGI purpose-built 2.5:1 speed increasing gear unit
7: Flexibox TLP 6000 safety element coupling
8: downstream flywheel
9: Froude DPY 7D hydraulic dynamometer

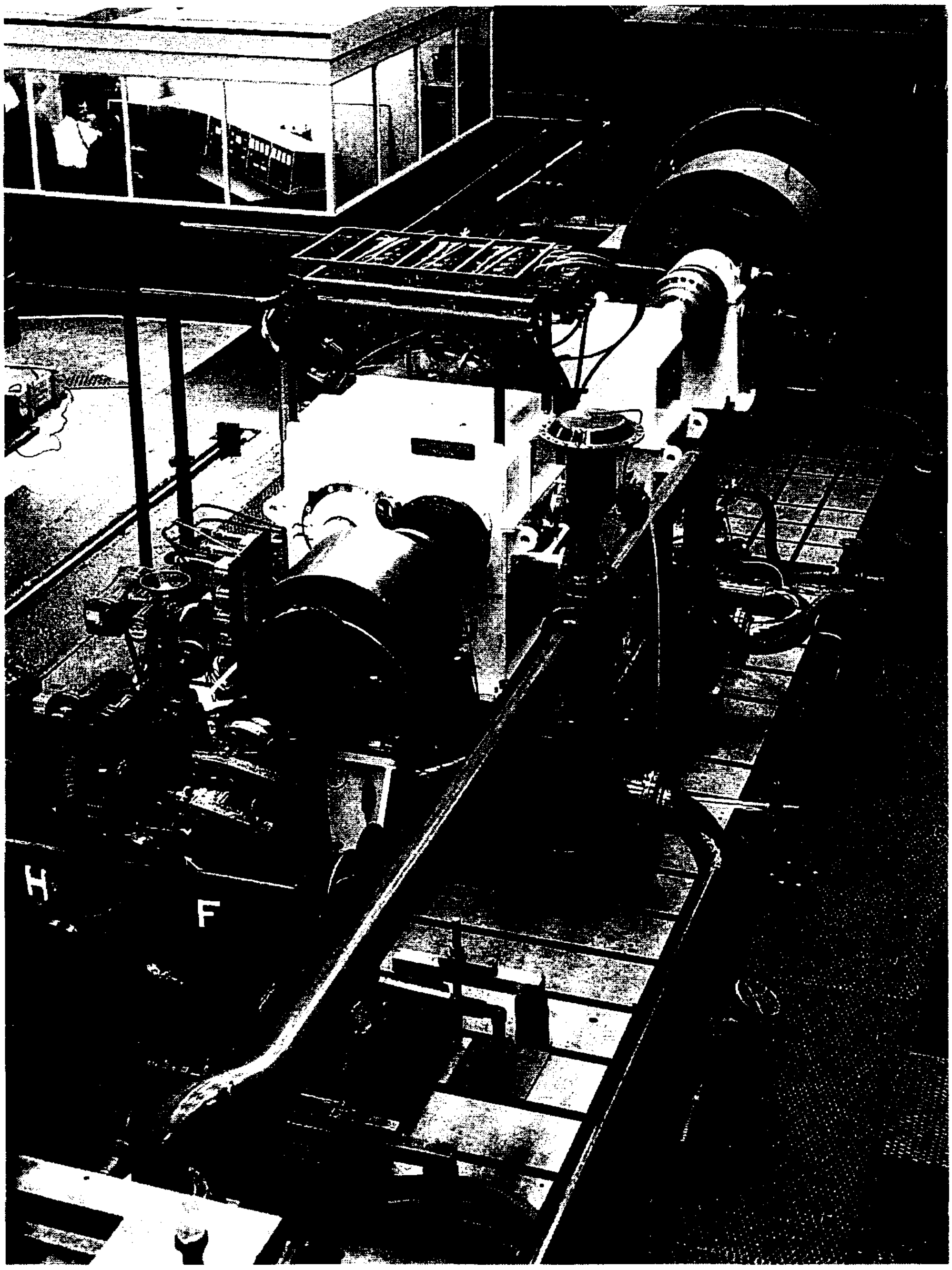

FIG. 5—THE TEST RIG AND CONTROL ROOM, VIEWED FROM THE DYNAMOMETER END AND LOOKING TOWARDS THE MOTOR

Item 4 Gearbox No 1

This is a purpose-built gearbox with 1.66:1 ratio single helical gears providing a maximum input speed to the RCC rotor of 600 rev/min. The wheel line of the unit provides bearing support for the RCC's 'pump end'. The limiting steady state speed of 600 rev/min is the maximum to allow all steady state testing to be carried out without overloading the drive motor (i.e. operation at speeds greater than 600 rev/min with 100% slip requires more than 3MW). However the gearcase has been designed to allow the gear ratio to be changed to 1:1 and for inclusion of an additional gear train with ratio 2.7:1—the reasons for which will be described later.

Item 5 Franco Tosi Coupling (Fig.6.).

Item 6 Gearbox No 2

This purpose built gearbox contains a single helical gear pair with a ratio of 2.5:1. The unit provides bearing support for the RCC's 'turbine end' and the main oil supply route for the coupling.

Item 7 Torque Limiting Coupling

Fitted to disengage the flywheel from the output gearbox in an emergency.

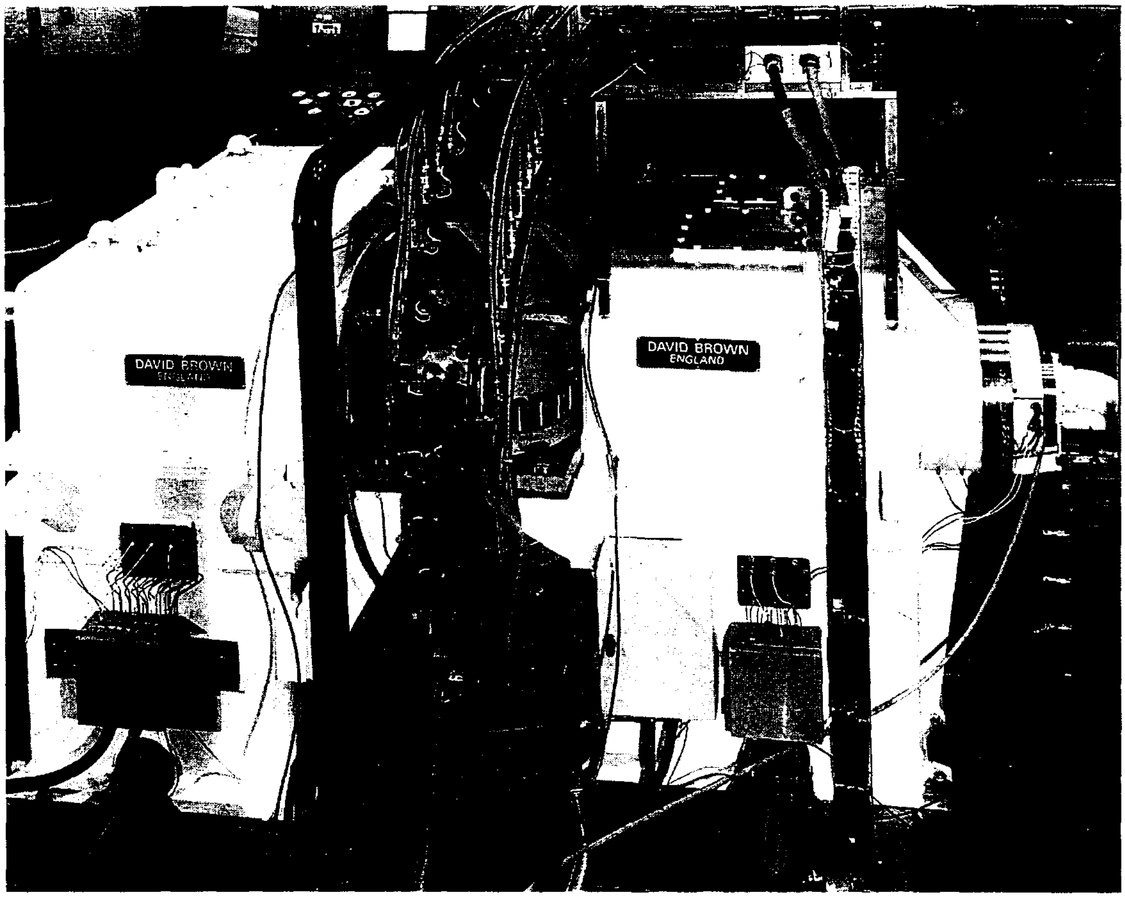

Fig. 6—The Franco Tosi coupling between the input and output gearboxes. There is one hydraulic control line for each of the 26 stator vanes

Item 8 Output flywheel

Designed to run safely at 2500 rev/min when the RCC is operated at 1000 rev/min. The referred inertia of the flywheel is such that at a speed of 1750 rev/min it represents the inertia of a Type 42 propeller shaft, propeller and entrained water at maximum speed.

Item 9 Dynamometer

Used for steady state testing. Being a unidirectional unit which can provide torque for one direction of rotation only, it is disconnected during ahead/astern reversals. It can however be fitted 'either way round' to support testing of ahead and astern steady state performance.

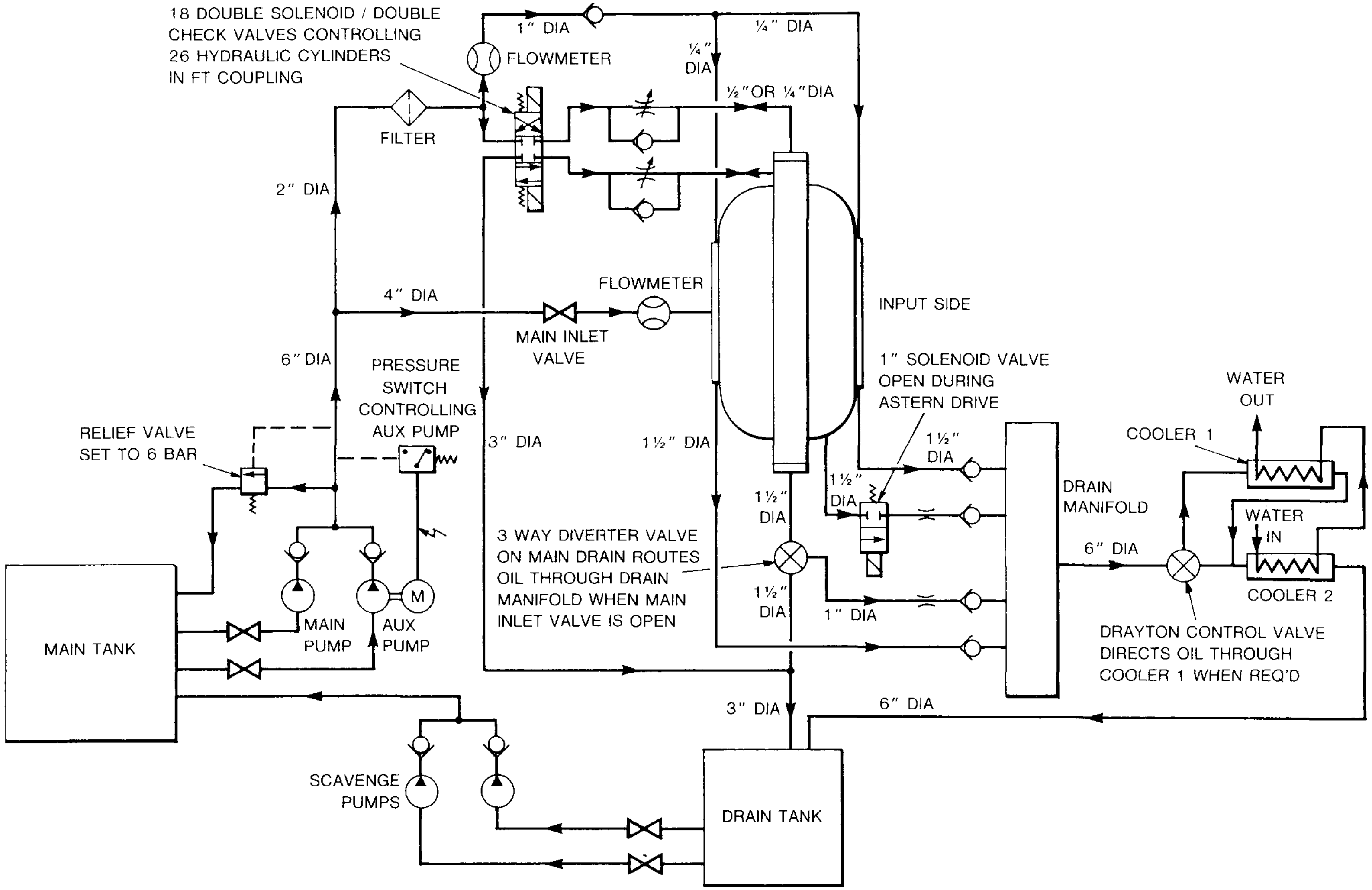

FIG. 7—FRANCO TOSI LUBRICATION AND HYDRAULICS SYSTEM

Lubrication and Cooling System

Oil is the medium by which power is transmitted and energy losses through inefficiency rejected (as heat); even in this limited power facility it demands a sizeable system, as shown in FIG.7.

(*a*) Main Lub. Oil Tank
Containing 9000 litres of OM-100.

(*b*) Main Lub. Oil Pumps
Two 75HP pumps (one running, one standby) each with a capacity of 450 U.S. gall/min at 6 bar pressure. Normal operating pressure is 6 bar.
The main supply (and drain) pipework is 6 inch bore.

(*c*) Main Oil Flow Control Valve
A 4 inch motorized valve fitted in the supply line to the coupling controls the oil throughout. The flow rate is measured using a turbine type flowmeter.

(*d*) Coupling Oil Seal/Hydraulic Actuator Supply Lines
The oil supplies to the coupling input and output oil seals and the vane actuators are taken from the main supply system through 8 micron filters.

(*e*) Lub. Oil Drains
Oil from the coupling discharges through four drain lines—main, astern and inlet/outlet seals. The principal flow path is through the seal drain lines, the main drain being used to empty the coupling.

(*f*) Main Lub. Oil Coolers
The drains from the coupling are collected into one 6 inch manifold, the output from which is directed through either one or both of the two lub. oil coolers (in series) as required using a temperature controlled valve. The total heat dissipation capacity is 2.5MW.

(*g*) Scavenge Tank
Positioned below floor level to assist drainage. Automatic pump out to the main lub. oil tank.

Instrumentation and Data Collection

Instrumentation and control is centralized in a purpose-built console (FIG.8) which consists of three elements, the data collection unit, the visual display unit, and the control for vane insertion and withdrawal.

Data Collection Unit

Based on a microprocessor. Used for data logging, rig control and results analysis, the software being produced in-house by DBGI. The main features of the unit are:

(*a*) *Alarm Monitoring*. Monitors a total of 160 channels of data at a scanning rate of 3 scans per second to ensure that each channel is within a predefined range. Any channel outside this range results in either a bell alarm or rig shutdown depending on the reading.

(*b*) *Data Storage*. There are two storage rates. Under normal steady state conditions data from all 160 channels is stored every 2 minutes or every 50 rev/min change in either input or output speed.
During a reversal, the time from maximum ahead to maximum astern speed is between 7 and 60 secs (depending on initial input

FIG. 8—THE CONTROL CONSOLE

speed). For this the storage rate is increased such that the 160 channels are recorded three times per second for 1 minute. The changeover to fast storage rate is automatic.

(c) *Visual Display Unit.* Up to 20 instrument channels can be continually displayed on the VDU, values being updated every 20 secs.

Test Rig Instrumentation and Control Panels

Panel No.1—Coupling parameters, input side e.g. input torque and speed, oil flow rates and pressures

Panel No.2—Coupling parameters, output side e.g. output torque and speed, oil seal and hydraulic pressures and flows

Panel No.3—Lub. Oil pump start/stop and related instrumentation. Fluidrive coupling scoop control

Panel No.4—Gearbox and flywheel bearing pressures. Rig services e.g. Electrical supplies, cooling tower

Vane Insertion/Withdrawal Control

Within control panels 1 and 2 are also fitted the control switches and position indicators for the coupling stator vanes. Each of the vanes has 'in' and 'out' indicating lights.

The 26 vanes are operated by 18 solenoid valves each controlled by its own switch on the control panel. 15 switches/solenoids operate just one vane each, one operates three vanes and two operate four vanes. Notwithstanding the available permutations of vanes inserted/withdrawn in normal operation it is a case of 'all in' or 'all out' since large fluctuations in stress are generated in the rotating elements when individual or groups of vanes are inserted while power is being transmitted. It is however permissible to insert half the vanes at low power in order to hold the output shaft stopped.

Phase 1 Test Programme

The major elements of the test programme were:

(*a*) *Steady State Running Ahead Against Brake Load.* All stator vanes *out.* The coupling inlet oil temperature and pressure were maintained at 55-65°C and 6 bar respectively. The input speed was held constant at a range of speeds from 300—600 rev/min and brake loads applied to generate slip across the coupling of between 10% and 80%.

As a special trial, to establish characteristics at 100% slip, the output shaft was locked stationary and the coupling operated under the same conditions.

(*Note.* Coupling slip is defined as:

$$\text{Slip} = \frac{100 \times (\text{Input Speed} - \text{Output Speed})}{\text{Input speed}}$$

Input speed is always positive

Output speed is positive when the output rotates in the same direction as the input, negative if it is contra rotating

Hence 0% slip: input and output rotate in same direction same speed
100% slip: output shaft stopped, input rotates
200% slip: input and output rotate in opposite directions at the same speed)

(*b*) *Steady State Running Astern Against Brake Load.* All stator vanes *in.* The input conditions were the same as in (*a*). Brake loads were applied to generate slip across the coupling of 190% to 120%. Additionally, as in (*a*), the output shaft was locked and the coupling operated with all vanes inserted—giving 100% slip.

(*c*) *Reversal Tests.* Starting from a number of input speeds between 300 and 600 rev/min (ahead and astern) the stator vanes were inserted (or withdrawn) to cause reversal—simulating ship ahead/astern and astern/ahead manoeuvres.

Phase 1 Results

Typical results are shown in FIGS. 9 to 13, with the expected (theoretical) results as a comparison:

(*a*) *Steady State.* Whilst the results are averages of a number of runs under similar conditions the coupling did exhibit good repeatability of performance. From the figures it can be seen that, for a given input speed and slip, the input and output torques/powers are significantly higher than expected (the explanation for which must be in the detailed design of the pump and turbine units) but the efficiency curves align very well.

In order to provide a feel for how these results would extrapolate to an actual ship installation: if it is assumed that:

(*i*) the normal slip range will be 14-28% ahead and 180-160% astern;

(*ii*) that torque and speed are related by a square law ($T \propto N^2$)—which is theoretically the case for a fluid coupling;

then the predicted torque, power and slip at 1000 rev/min are as shown in TABLE I.

(*b*) *Reversals.* FIGS. 12 and 13 show the transient performance of the RCC during rehearsals. All vanes moved within 4 secs of initiation, reversal of output rotation being completed in about 20 secs.

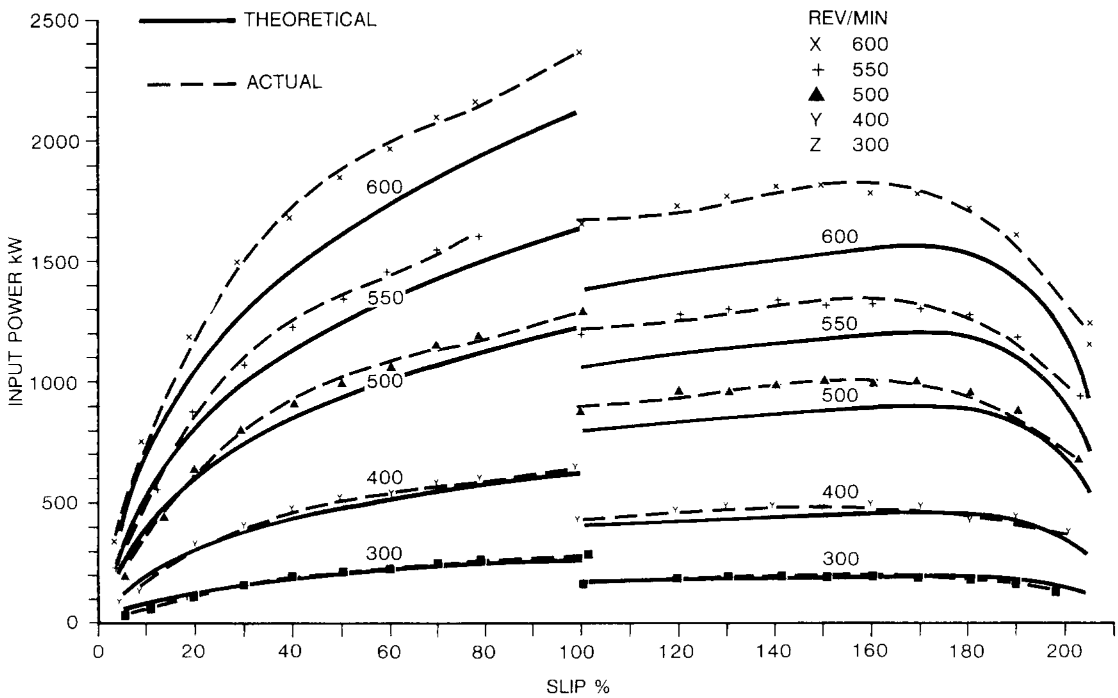

FIG. 9—INPUT POWER V. SLIP AT DIFFERENT SPEEDS

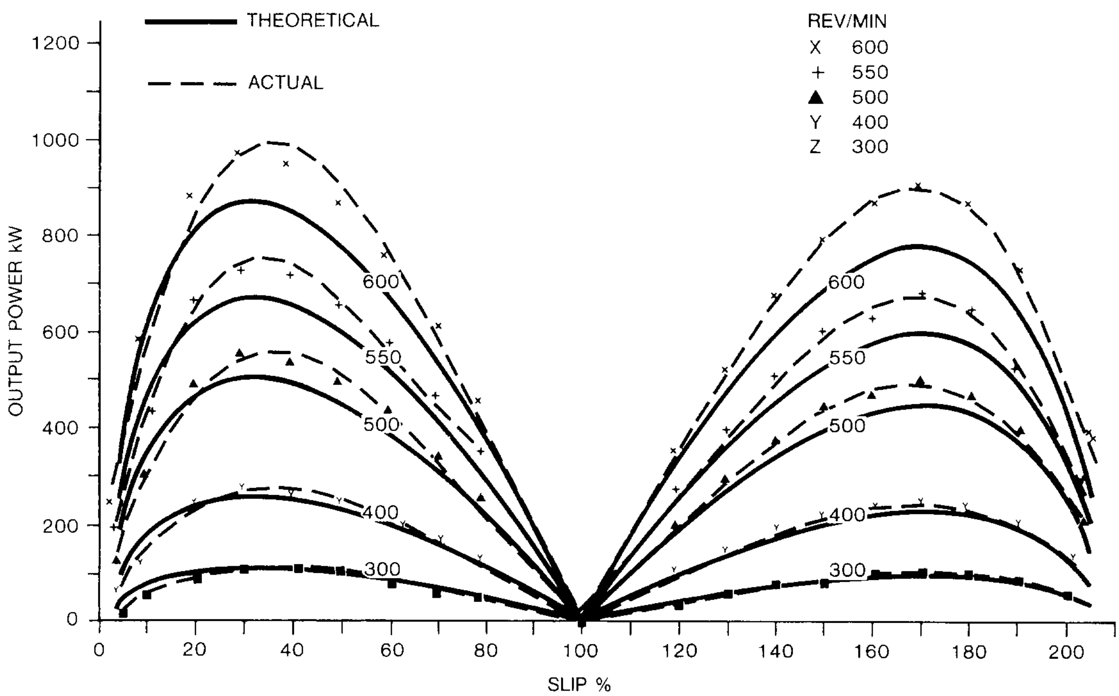

FIG. 10—OUTPUT POWER V. SLIP AT DIFFERENT SPEEDS

TABLE I—*Predicted torque, power and slip for a ship installation at 1000 rev/min*

	Ahead			*Astern*		
	Torque kN.m	*Power* kW	*Efficiency* %	*Torque* kN.m	*Power* kW	*Efficiency* %
Input	38–61	3978–6371	69–80	72–76	7573–7951	48–50
Output	36–58	3065–4388		45–60	3780–4015	

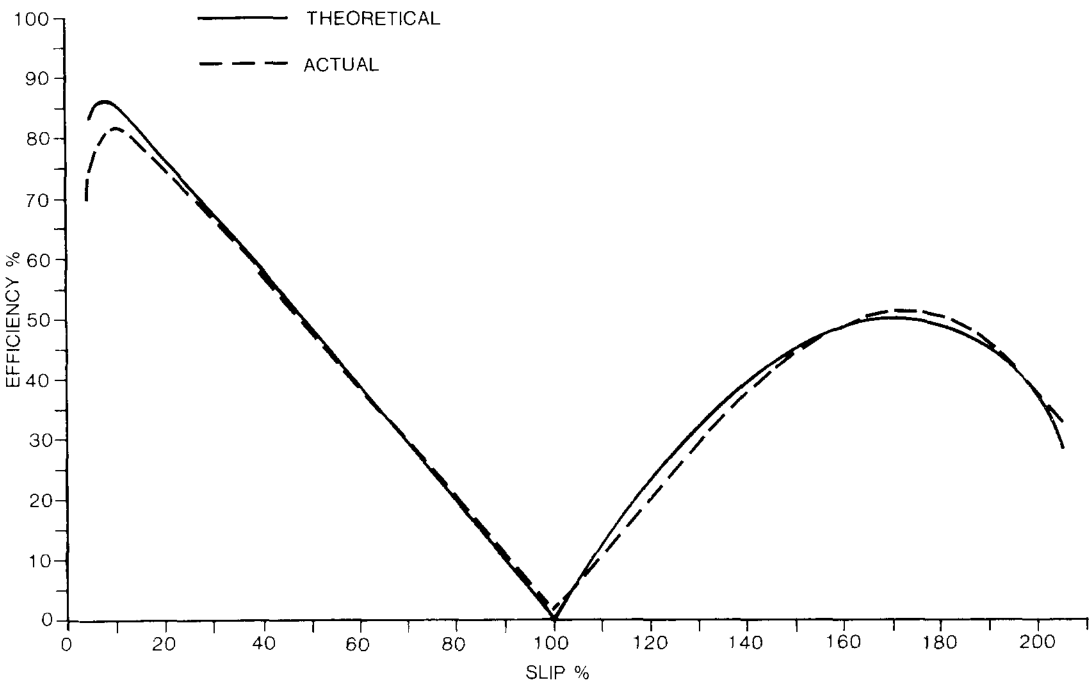

FIG. 11—EFFICIENCY V. SLIP

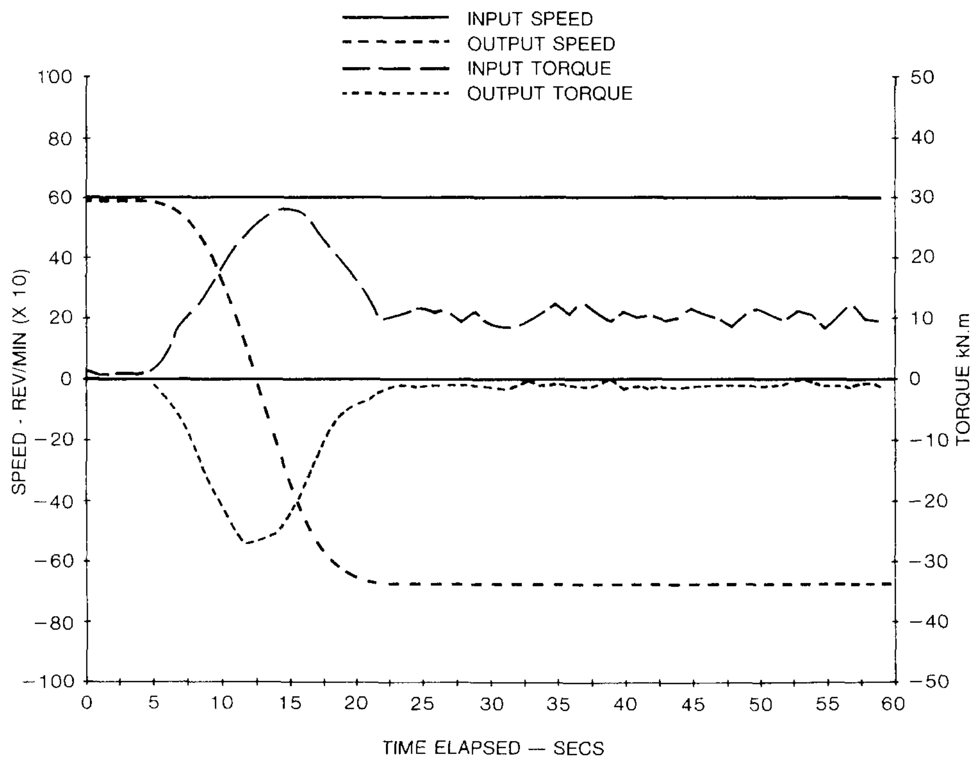

FIG. 12—REVERSAL FROM AHEAD TO ASTERN AT 600 REV/MIN INPUT SPEED

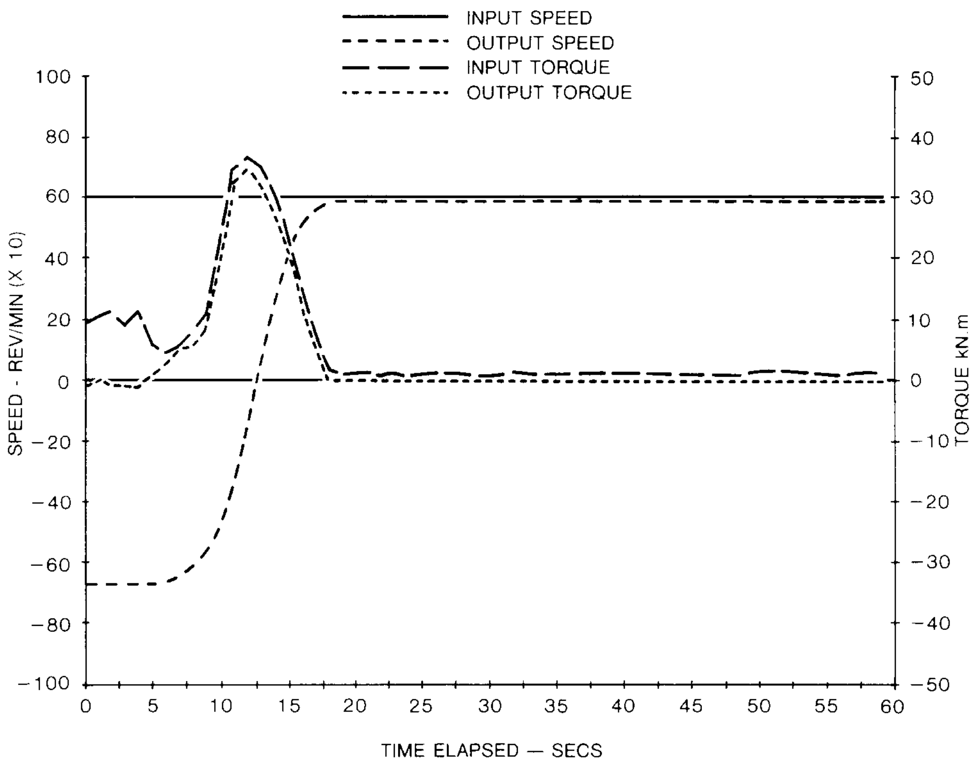

FIG. 13—REVERSAL FROM ASTERN TO AHEAD AT 600 REV/MIN INPUT SPEED

Conclusion

From this first stage of testing at input speeds up to 600 rev/min it was demonstrated that the RCC performed as well as, or better than, predicted by Franco Tosi. Results of other trials (not described here) investigating the effect of varying lub. oil pressure and temperature showed operation to be relatively insensitive to such variables.

Phase 2 Test Programme

Whilst it is not possible to operate the coupling over a wide slip range at any speed above 600 rev/min due to the limited power of the motor (as indicated by the above predictions for 1000 rev/min) it was useful to operate at 1000 rev/min so that one or two points at low slip could be checked and confirmed against theoretical values. Additionally there was considerable interest in reversing performance at higher speeds and, with some rig modifications (FIG.14), this could be investigated adequately. The principal changes necessary were:

(*a*) Change of gear ratio in the input gearbox from 1.6:1 to 1:1.

(*b*) Addition to the input gearbox of a power take off (ratio 2.7:1) to drive an input flywheel. The purpose of the flywheel was to act as an energy store to supplement the inertia of the motor during reversals—without this the motor would stall. (Vital statistics of the flywheel are: weight 6 ton; length 1.2m; diameter 0.88m; maximum speed 2700 rev/min. To stand alongside this during a full speed reversal was a considerable test of character, leading to mental questioning as to the ability of the NDE examination to pick up a critical size crack in such a large lump of metal.)

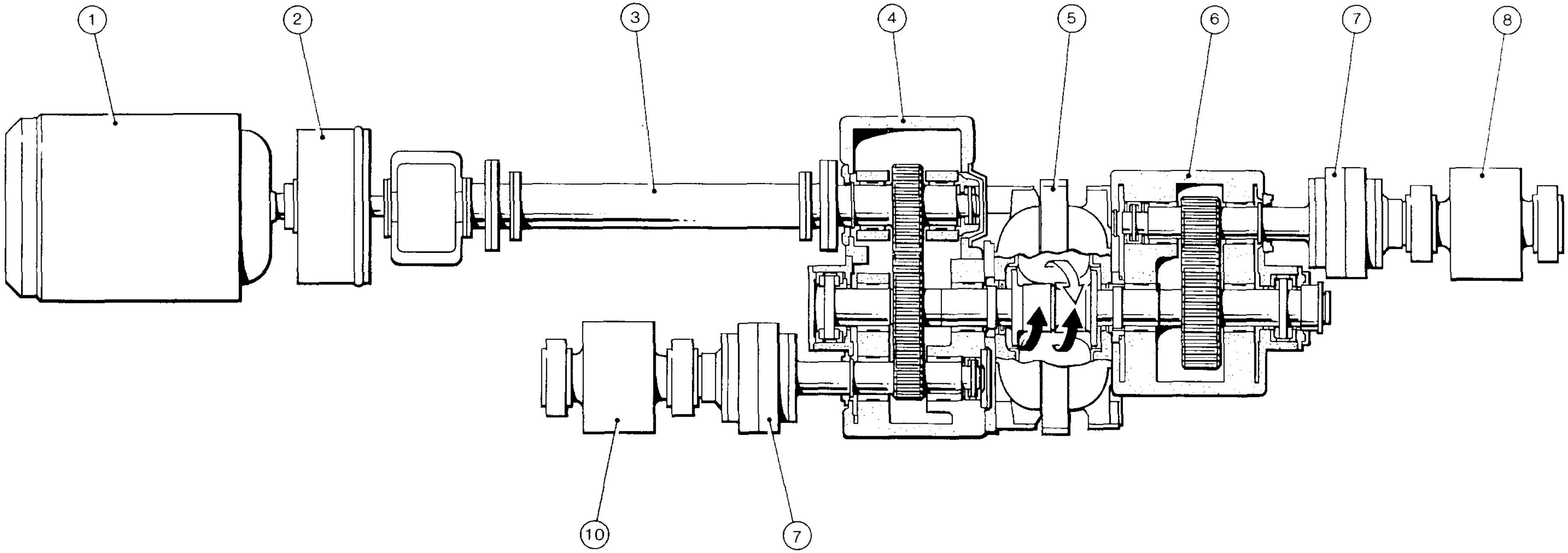

Fig. 14—Layout of main mechanical components for Phase 2 testing

1: 4000 HP synchronous induction motor
1000 rev/min fixed speed
2: Fluidrive 41 SCR Mk.4 scoop control fluid coupling
3: M4000 coupling (to provide distance)
4: DBGI purpose-built 2.7:1.1 speed increasing gear unit
5: Franco Tosi 1000mm RCC coupling
6: DBGI purpose-built 2.5:1 speed increasing gear unit
7: Flexibox TLP 6000 safety element coupling
8: downstream flywheel
10: upstream flywheel to supplement motor during reversals

TABLE II—*Slip range limitation in Phase 2 trials*

Input Speed rev/min	*Slip Range* %	*Limiting Factor*
600	4-20	Fluidrive coupling cooling capacity
700	4-20	Motor power
800	3-15	Motor power
950	3-5	Max speed achievable due to slip in input fluid coupling

In addition to the limitations on testing imposed by maximum motor power, further restrictions were created by the input fluid coupling—with the result that steady state running was limited to certain ranges of slip (see TABLE II).

No steady state astern running was possible with the vanes *in*, as even at 600 rev/min the slip created by the drag load of the brake would have required nearly full power from the motor, causing problems for the input fluid coupling. In fact this was not too important as a full set of results at 600 rev/min had been obtained in the Phase 1 trials in which, due to the different input gear ratio, full motor power of 3MW was available at this speed rather than the 1.8MW obtainable in the Phase 2 'high speed' configuration.

Phase 2 Results

(*a*) *Steady State.* The general trend of results was similar to that found in Phase 1, i.e. the input and output torques at any given condition were higher than predicted but efficiencies were lower.

(*b*) *Ahead-Astern Reversals.* Results of ahead-astern reversals at 950 rev/min are shown in FIG. 15. It can be seen that, even with the energy stored by the input flywheel, the coupling input speed dropped considerably when the vanes were inserted.

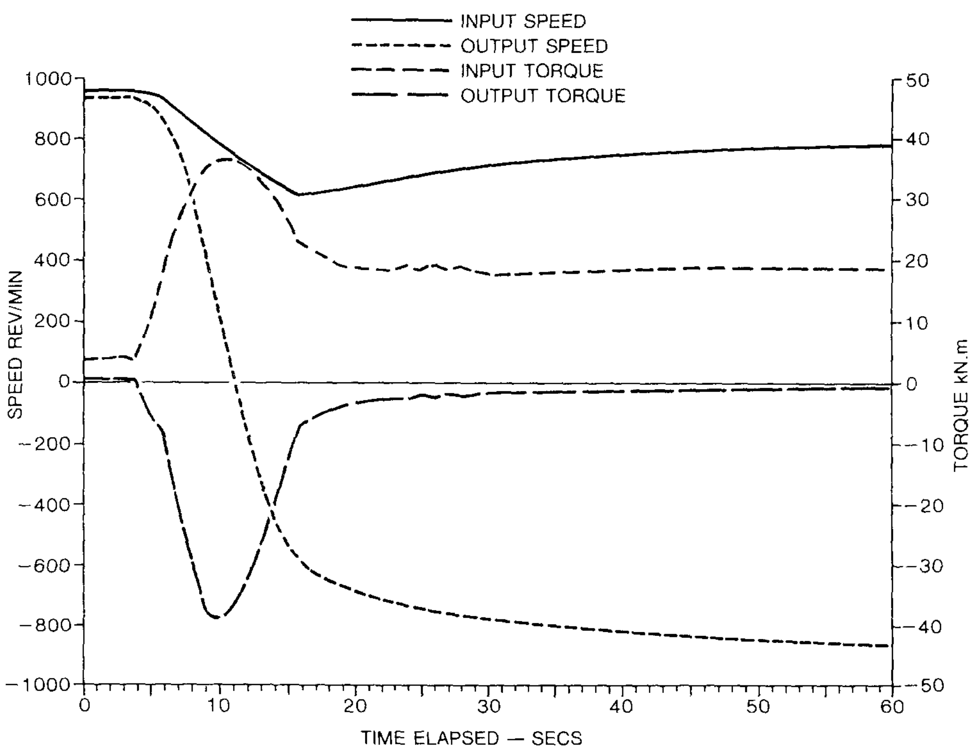

FIG. 15—REVERSAL FROM AHEAD TO ASTERN AT 950 REV/MIN INPUT SPEED (BLADES IN)

Vane movement begins 5 secs into the scan and it can be seen that the reversals were effectively complete within a further 15 secs (albeit with a lower input speed after this time). The time taken from initial vane movement to the last vane being fully in was typically in the range 5–7 secs.

(*c*) *Astern-Ahead Reversals*. The response of the coupling during astern to ahead reversals at 800 rev/min is shown in FIG.16. As with the ahead-astern tests, the reversals were effectively complete within 15 secs of initiation of valve movement.

Vane withdrawal times were longer than the insertion times, being typically 11 secs at 600 rev/min and 20 secs at 700 rev/min. These lengthy times were the result of two particular vanes (at the ends of the hydraulic control lines) being slow to move. At 800 rev/min one valve frequently failed to withdraw at all which is considered to be a failure mode. One conclusion of the test programme is therefore that, with the current hydraulic system design, vanes should not be withdrawn at input speeds above 700 rev/min. This is unlikely to be of practical significance to any ship installation.

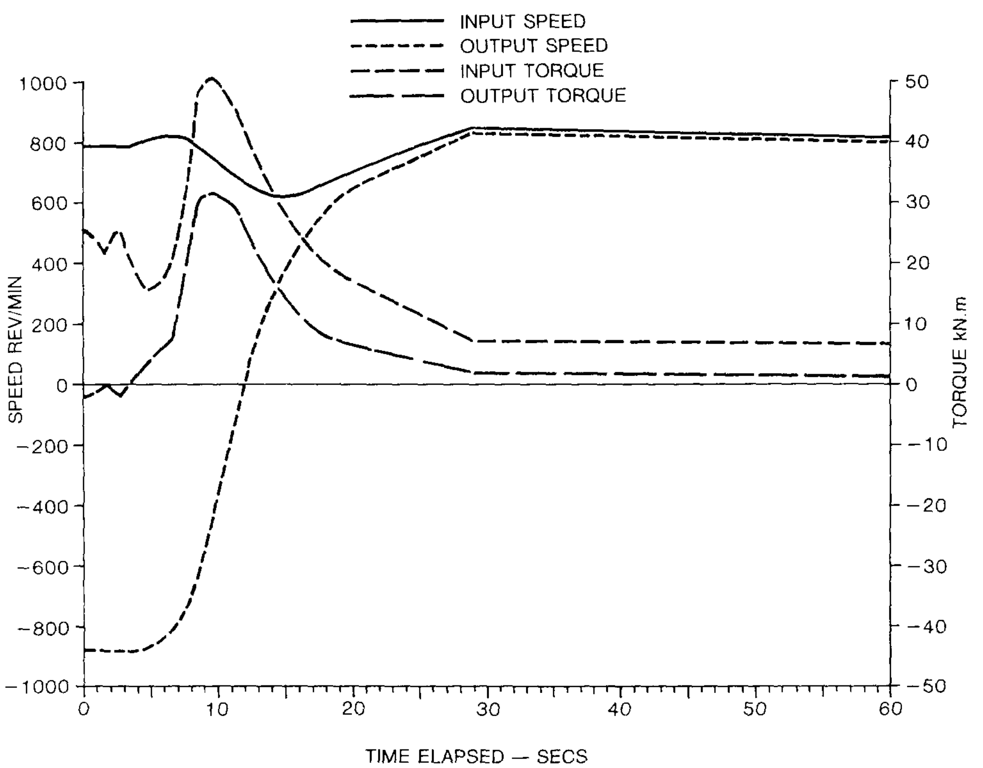

FIG. 16—REVERSAL FROM ASTERN TO AHEAD AT 800 REV/MIN INPUT SPEED (BLADES OUT)

(*d*) *Thrust Measurement*. The initial design of the test rig did not include any provision for the measurement of the axial thrust generated within the coupling between pump and turbine rotors—a parameter that might be significant to a gearbox designer. During Phase 2 a strain gauge thrust recording system was fitted to both input and output shafts but unfortunately results were unsatisfactory due to the low strain being measured (which necessitated high amplification and introduced serious problems of drift in the readings).

(*e*) *Cavitation*. Cavitation can be a problem in fluid couplings. However Franco Tosi calculations indicated that there should be no difficulty with blades out at the slips (0–20%) and lub. oil supply pressures (4–6 bar) tested. This was confirmed during trials.

With blades in, cavitation is predicted at speeds above 1000 rev/min input speed with an oil pressure of 6 bar and 200% slip. Astern operation of the test rig was confined to no load when a slip of 210% was typical. Cavitation was seen to occur at approximately 750 rev/min with a 6 bar feed pressure and, once set in, continued down to approximately 650 rev/min; however no specific cavitation trials could be carried out due to the limitations on 'blades in' operation.

Phase 3 Testing

The coupling tested in Phases 1 and 2 (ex *Taranaki*) was the original '79' design (stemming from 1979), however in 1984 Franco Tosi introduced a modified version (designated '84') which had thicker rotor vanes of improved shape and redesigned stator vanes—with the object of reducing internal stresses and increasing efficiency. The U.S.N., who had been testing the other *Taranaki* coupling (the sister to the R.N. unit but of opposite hand) purchased and tested the new variant but found that it was in fact less efficient than its predecessor. The turbine unit 'blading' was subsequently further modified by Franco Tosi but to no avail. At this point U.S.N. interest reverted to the '79' design and the '84' was set aside.

Franco Tosi continued to investigate the problem with the '84' unit and produced proposals for still further modifications, but with no opportunity for retesting in the U.S., were unable to validate predictions.

Having now completed Phase 2 of our own testing, thoughts ranged to possible trials of what should have been the latest design standard '84' unit. As there was no possibility of (nor justification for) obtaining the finance to purchase new rotor and vane assemblies, approaches were made to the U.S.N., to establish the availability of their unused coupling, and to SSS Gears (who market and manufacture the coupling outside Italy), to see whether it was possible to convert the casing of the R.N. coupling to accept rotors and vanes of opposite hand. The outcome of this activity was that:

(*a*) The U.S.N. unit was made available for testing in the U.K.

(*b*) SSS Gears Ltd offered to upgrade the '84' rotors to Franco Tosi's latest design and to make appropriate modifications to the R.N. casing to accommodate these (and the vanes). Most importantly of all, this was free of charge.

The R.N.'s contribution to this excellent example of collaborative development was to pay for the stripping and re-building of the test rig, followed by further testing.

It would be nice to report that the outcome of such co-operation was an unqualified success but unfortunately trials demonstrated that the hoped for improvements in performance had not been achieved and even a little further 'tweaking' did little to alter the situation.

Regrettably by this time (the end of 1988) funding for the Franco Tosi evaluation programme had been exhausted and it was necessary to decommission the test rig. In fact little more could be done without turning it into a development facility, which was something that it was not intended to be.

Overall conclusion

Testing of the '79' design RCC validated Franco Tosi's predicted performance figures and identified no weaknesses of any significance in its concept, design or manufacture. Useful lessons have been learned on detailed aspects of how the coupling could best be integrated into the design of a reversing gearbox but no particular difficulties are forseen.

Unfortunately the '84' design does not quite meet performance expectations although this is unlikely to be critical in a practical application. Such penalty as there is might be worth trading for what appears to be a physically stronger and more easily manufactured (cheaper?) rotating assembly. Franco Tosi continue to look into the problem and are confident that the discrepancy between trials performance and predictions from model testing is due to variations in construction rather than a fundamental deficiency in design.

Overall, MOD would be fully prepared to accept the use of Franco Tosi couplings within an R.N. gearbox if such an arrangement was proposed by a lead shipbuilder. As such the RCC stands alongside (but in competition with) conventional fluid couplings as used, successfully, in the COUNTY and CVSG Class gearboxes.

This article concentrates upon the R.N. trials programme. However, as noted, the U.S.N. were conducting trials on an actual ship shore test facility during the same period. Relations between the two groups were close throughout, with a full interchange of information and reports.

Acknowledgements

Firstly I must acknowledge that a large part of this article is based upon the trials reports prepared by DBGI Ltd. In this respect I would like to thank Mike Field who did sterling work in managing the design and construction of the test rig as well as running the early trials and also Nigel Hilling who then saw the project through to completion.

Secondly I must express my appreciation for the cooperation received from the U.S. Navy through both NAVSEA and NAVSSES, and particularly for the loan of their coupling—by the time this article is published it could well have arrived home.

Lastly, my thanks to Mr Clements of SSS Gears for his enthusiasm and commitment to the project.

Reference

1. Clements, H. A. and Fortunato, E.: Franco Tosi reversible converter-coupling with direct drive SSS clutch; *Journal of Naval Engineering*, vol. 31, no.2, Dec. 1988, pp.386–402.