SHORE TRIALS OF A MARINE PROPULSION SYSTEM

COMPUTER-CONTROLLED MANOEUVRING TRIALS

BY

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This article is based on a paper read by the authors before the Institute of Marine Engineers and discusses a means of performing simulated ship manoeuvres on the CAH shore test facility.

Introduction

This article is concerned with a series of computer controlled manoeuvring trials conducted on a marine propulsion system on a shore test facility (STF). These trials featured the use of a novel dynamic loading system, controlled to simulate the loading conditions which would be experienced at sea during manoeuvring, particularly crash stop manoeuvres from high ahead ship speeds.

Background to the STF

A comprehensive justification of the shore trials concept is not within the scope of this article; however, it is interesting to note that the cruiser propulsion plant represents the highest power per shaft used in any R.N. ship. A risk assessment of the problems likely to be experienced in the ship (Ref. 1) showed that the potential areas of concern which could be evaluated at a shore test facility were:

(*a*) gearbox;
(*b*) uptakes and downtakes;
(*c*) machinery controls.

Although none of the techniques and components used in the propulsion system are new, the combined use of clutches and fluid couplings at the high powers required was outside current R.N. experience. The potential problems associated with the design of uptakes and downtakes described in Ref. 2 are compounded in the cruiser because of the long ducting necessary due to the island superstructure. In order to reduce machinery manning requirements, the degree of automation used in the machinery control system is high. Consequently, extensive testing and evaluation was required prior to sea trials.

The principal advantages of the shore test facility were considered to be that:

(*a*) during sea trials, there would be many trials other than those associated with machinery and hence time for machinery trials will be at a premium;
(*b*) if extended machinery trials on the ship were necessary, substantial costs would be inevitable;
(*c*) there would be less scope on the ship for stopping and starting machinery and for making adjustments to control settings, etc.

Previous experience on sea trials of first-of-class warships indicates that the more difficult problems arise during transient manoeuvres rather than during steady state running. Before the cruiser shore trials, the manoeuvring performance was examined in more detail than previously using computer simulation. These studies highlighted some machinery dynamic problems, particularly during crash astern manoeuvres, which were considered sufficiently serious to justify full scale trials on the STF. The simulation studies indicated that the most severe machinery conditions occurred during a single engine per shaft crash astern manoeuvre and the problem was caused by the high power induced by the forward momentum of the ship acting on the transmission system via the fixed-pitch propeller. The machinery torque/speed conditions produced by this injected power could not be provided on a shore test facility by a totally passive loading system (i.e. dynamometer), and therefore there was a need to inject power into the propulsion system to simulate the momentum induced torque. The dynamic loading system adopted to impose the transient loads has become known as the Power Injection System.

The main advantages of performing simulated 'ship' manoeuvres on the STF are that:

(*a*) the machinery will have been demonstrated to have undergone a series of manoeuvres consistent with expected shipboard behaviour;

(*b*) any propulsion system limitations should at least be known and eliminated or countered by operating procedures before sea trials;

(*c*) extensive data will be obtained on gas turbines, hydraulic couplings and general transmission system behaviour which will allow design reappraisal and further studies to proceed with more confidence in machinery characteristics.

Propulsion System

Machinery Arrangement

Each shaft set of the propulsion system has two gas turbines which can drive together or singly into a reversing gearbox, the output of which is connected via the main shafting to a fixed-pitch propeller. Manoeuvring drive ahead and astern is transmitted via scoop-trimming double-circuit hydraulic couplings, and direct drive ahead via self-synchronizing clutches is used for economical cruise and high powers (see article p. 279). The shore test installation is a complete shaft set of main machinery, including full-scale uptakes and downtakes, gas turbines, gearbox, ancillaries, mainshaft thrust block and a short main shaft directly coupled to a water dynamometer. The machinery layout is shown diagrammatically in FIG. 1. The propulsion controls are fully represented and local, ship control centre and bridge levers are fitted.

Propulsion Control System

In normal operation, overall control of the engine power and the machinery is obtained using a single lever (Power Demand Lever—PDL) for each shaft set from the bridge or the ship control centre.

Individual manual control of the propulsion plant is available at the local control centres in the machinery spaces. In normal operation, these will be unmanned.

A detailed explanation of the functioning of the propulsion control system is outside the scope of this article. However, it is pertinent to later discussions to outline the control system functions and sequencing, and the machinery behaviour during a typical crash stop manoeuvre:

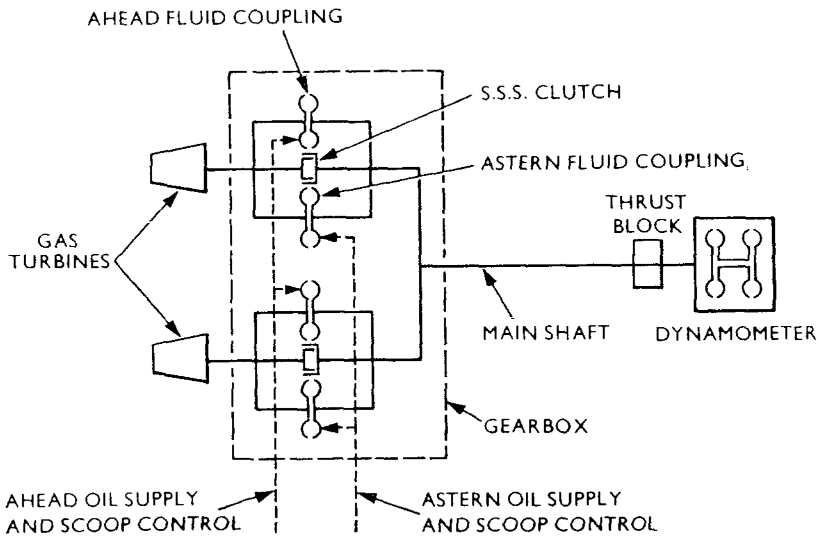

FIG. 1—ARRANGEMENT OF SHORE TRIALS MACHINERY

(*a*) PDL moved to required astern position;
(*b*) gas turbine reduces to idling;
(*c*) (if in direct drive) clutch actuator moves to the 'ready to disengage' position;
(*d*) (if in coupling drive) ahead coupling begins to empty;
(*e*) astern coupling begins to fill;
(*f*) (if in direct drive) reverse torque disengages the clutch (with transient brake assist if necessary);
(*g*) power reapplied to gas turbine;
(*h*) main shaft decelerates, reverses, and ship subsequently stops.

Potential Problem Areas

The foregoing procedure gives the control system requirements in principle; in detail the control sequencing was investigated using computer simulation methods, and the following problem areas were highlighted:

(*a*) *Clutch Disengagement*

The gear ratios of the gearbox, particularly between the clutch and hydraulic coupling lines, were designed to facilitate the engagement and disengagement of the clutch. To disengage the clutch during decelerations or crash stops, the reverse torque required to create a speed differential can, in most cases, be provided naturally by the reduction in power of the gas turbine combined with the ship momentum effects. Additional torque of the required sense is available from the filling coupling (ahead or astern) and, as a last resort, from a transient brake fitted on the intermediate speed gear line upstream of the clutch.

(*b*) *Power Turbine Speed*

As the ahead power is taken off, and the transmission changes from ahead to astern drive, the point of reapplication of power must be chosen with care to achieve a compromise between high power dissipation in the filling astern coupling and low power turbine speed. Power reapplied too soon leads to a high rate of increase of oil temperature in the oil coupling; power reapplied too late leads to low power turbine speed and possible reversal which were expected to have undesirable effects on the power turbine bearings.

(*c*) *Coupling Oil Temperature*

As the main shaft decelerates at a rate dictated by the applied astern power and the opposing (ship way generated) 'propeller' power, the power dissipation in the astern coupling builds up to a maximum (typically) just before shaft reversal, producing a high rate of increase in oil temperature. A balance between stopping performance and constraints on the allowable oil temperature (due to possible oil degradation) has to be obtained.

(*d*) *Main Shaft Stall*

Gearing efficiency and friction studies indicate that (in common with other propulsion systems of a similar type) the main shaft will stall, i.e. remain at zero rev/min, for considerable periods during manoeuvres from high ahead speeds. The stalled propeller shaft will produce a lower astern thrust (thus resulting in a reduction in stopping performance) and could lead to mechanical damage to the transmission system. In addition, all the engine power is dissipated in the astern hydraulic coupling for the duration of the stall, thereby maintaining at a high level, or increasing, the coupling oil temperature.

The common factors in the above problems are the high power fed into the system via the propeller and the characteristic behaviour of the hydraulic couplings, particularly at high slip conditions.

The simulation studies showed that the worst case for these problems arose during single engine/shaft crash stop manoeuvres. This factor is of significance in the design of the power injection system, as will be seen later.

Manoeuvring Studies

Having established that problems existed, more detailed studies using computer modelling techniques were pursued in attempts to assess whether the problems could be designed out of the system. Manoeuvring procedures and control system parameters were determined in order to maintain the machinery within the known design constraints. Sensitivity studies were performed to assess the effects of uncertainty in the defined characteristics of the machinery system.

However, it was concluded that due to doubt about the reliability of some performance data used in these computer simulations it was necessary to obtain additional data before continuing simulation work.

An extensive work programme was initiated to procure performance data of relevance to the propulsion system. The programme included work in the following areas:

Hydraulic Couplings	— manufacturers tests on half-size coupling
	— manoeuvring trials in County Class destroyer which has similar type of propulsion system (half-size couplings)
Shaft Stall	— manoeuvring trials in County Class destroyer
Propeller and hull	— tank tests on scaled models in steady and dynamic conditions.

Whilst these trials provided valuable data, the data obtained suffered from the disadvantage of requiring extrapolation to full size. Nothing more could be done on propeller characteristics until sea trials, but the STF was available for experimental purposes and it was decided to make maximum use of this facility to provide the full-scale data required on coupling performance and shaft stall conditions and thereafter to examine the manoeuvring problems outlined above.

Power Injection System

Basic Concept of Power Injection

During a crash stop manoeuvre from a high ahead speed, propeller torque derived from the ship's way is imposed on the machinery system. With the fixed-pitch propeller, this torque combines with the inertia torque of the machinery tending to maintain the ahead rotation of the mainshaft. After the shaft reverses, and whilst the ship is still moving ahead, the effect of continuing propeller feedback is felt as an additional resistance to acceleration of the main shaft in the astern direction. Thus, for astern rotation the system will be operating at higher torque levels than those given by the astern propeller law. This high torque loading will continue until the ship reaches steady astern condition.

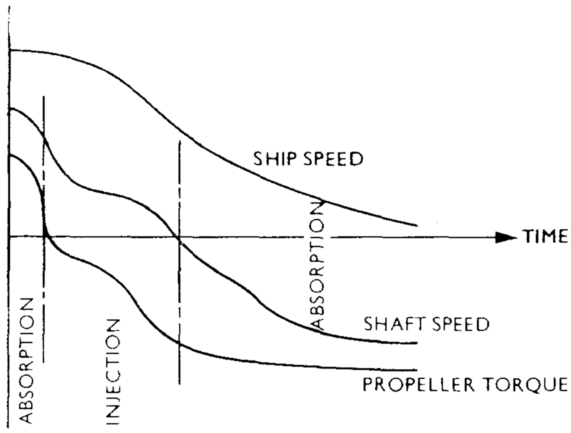

FIG. 2—TYPICAL STOPPING TRANSIENT

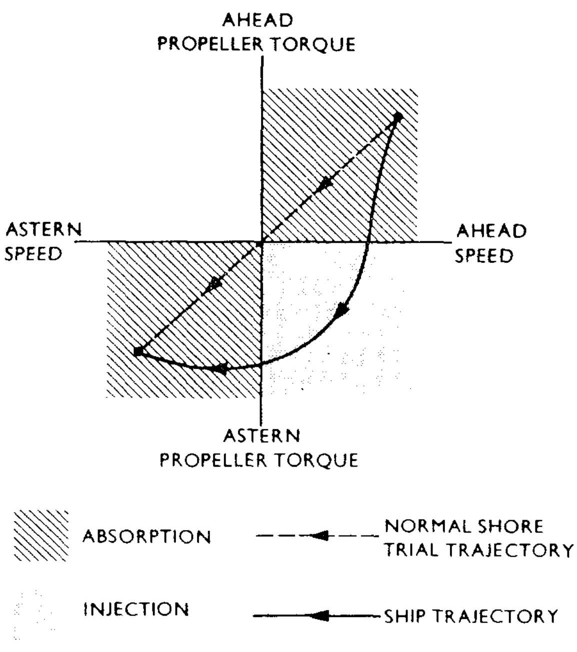

FIG. 3—TYPICAL TORQUE/SPEED TRAJECTORIES

Simulation studies of the ship manoeuvring showed that the most arduous condition for the propulsion machinery occurred during single engine/shaft crash stop manoeuvres.

FIG. 2 shows the speed, propeller torque, and shaft speed transients expected during a typical crash stop manoeuvre. In terms of propeller loading the manoeuvre can be divided into two periods of power absorption separated by a period of power injection. This can be more clearly observed from FIG. 3 which shows the propeller torque/shaft speed trajectory throughout the manoeuvre. Superimposed on FIG. 3 is a typical torque/speed trajectory which would be available using the STF machinery arrangement in normal test conditions. Of significant importance is the requirement to impose high torque loads on the system at very low and zero main shaft speeds.

Machinery Requirements

To reproduce these loading conditions on the STF for a wide range of crash stop manoeuvres requires the provision of controllable load-absorption and power-injection devices.

The double-circuit water-impeller dynamometer is bi-directional and in principle can provide the load absorption for ahead and astern operation. The natural torque/speed characteristics (obtained with preset control valves) of the dynamometer are not compatible with the requirement. Dynamic load control of both ahead and astern compartments is therefore essential.

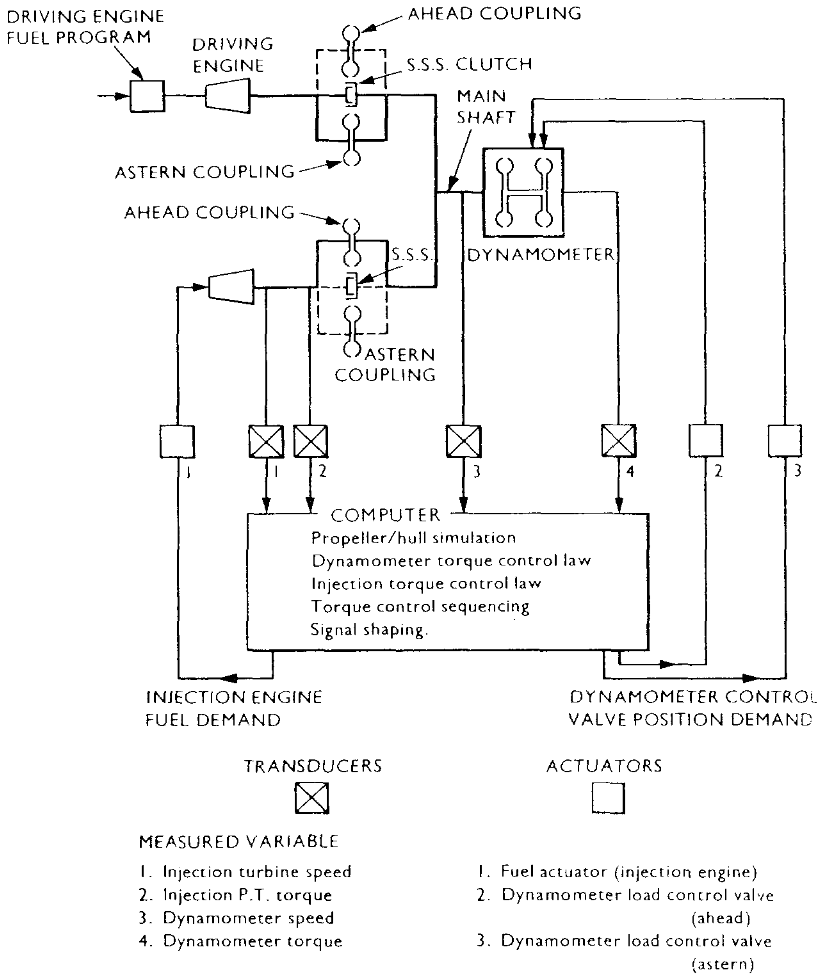

FIG. 4

The power injection source adopted for the system was one of the installed gas turbines driving through its ahead fluid coupling, the other gas turbine providing the manoeuvring power. This arrangement was compatible with the intended usage of power injection—which is the simulation of single engine/shaft manoeuvring conditions. It had the advantage over other proposed devices and arrangements of requiring neither additional capital expenditure or major machinery modifications.

Power Injection Control System

The power injection control system is required to compute the load torque to simulate ship conditions and to control the load absorption and power output of the dynamometer and injection engine respectively so that this load torque is imposed on the manoeuvring engine and transmission during the manoeuvre.

The basic requirements of the control system were defined in broad terms by the knowledge of the required manoeuvring loads and of the capabilities and operating regimes of the dynamometer and injection engine. A mathematical model of the system was used to examine the feasibility and

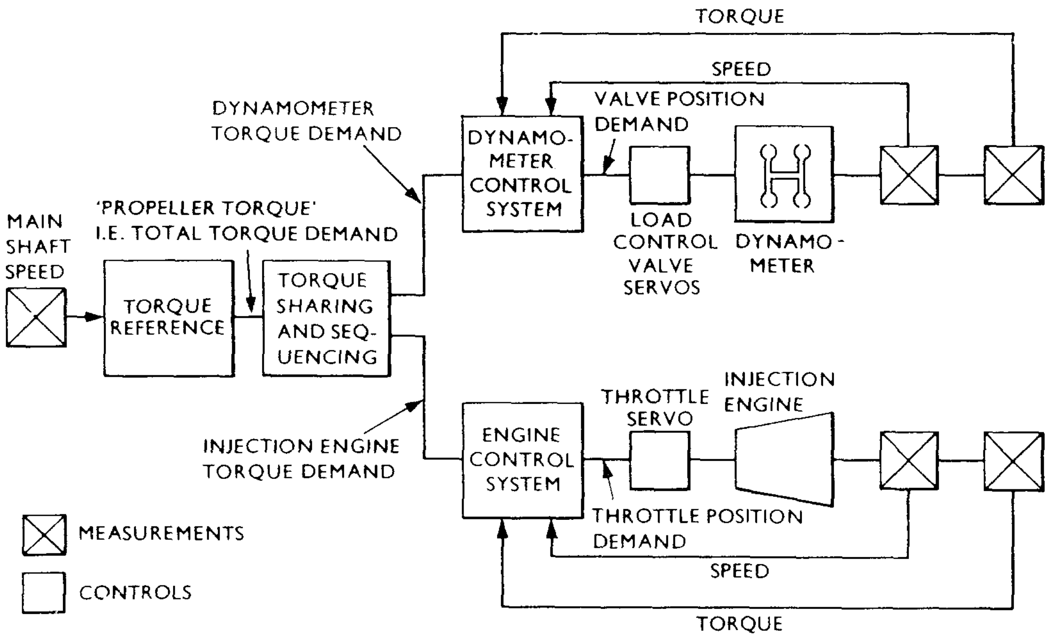

FIG. 5

controlability of the proposed system. The model was formulated such that it could represent both the ship and STF configurations, providing the feature of almost immediate comparisons during the development of the control system. The computer-based control system for power injection, indicating the main control function and the measurement and control signals involved, is shown in FIG. 4. Further details of the power injection control system can be found in Ref. 3.

The control system comprises four basic sub-systems as shown in FIG. 5 and which are described below.

Torque Reference System

Since the power injection system is required to reproduce loading conditions on the manoeuvring engine and transmission consistent with those imposed by the propeller under equivalent ship manoeuvring conditions, the load torque imposed by the power injection control system must be made equal to the propeller torque experienced by the ship under these conditions.

The torque reference signal for the control system is obtained from the solution of the single degree of freedom ship motion equation, the propeller torque and thrust equations, and the hull/propeller interaction factors. The solution of the equations inherently formed part of the power injection control system and pointed the way towards a computer-based control system.

Dynamometer Control System

This system computes the dynamometer load control valve (LCV) position required to produce a given load torque at a given shaft speed. The demand LCV position signal is passed to the existing LCV position servo systems. Proportional and integral torque error signals are used to close the control loop.

Injection Engine Control System

This system computes the required injection engine throttle position from an input torque demand and the signal is passed to the existing on-engine fuel control system. A proportional torque error term closes the control loop, and phase lead compensation is used in the feed forward path.

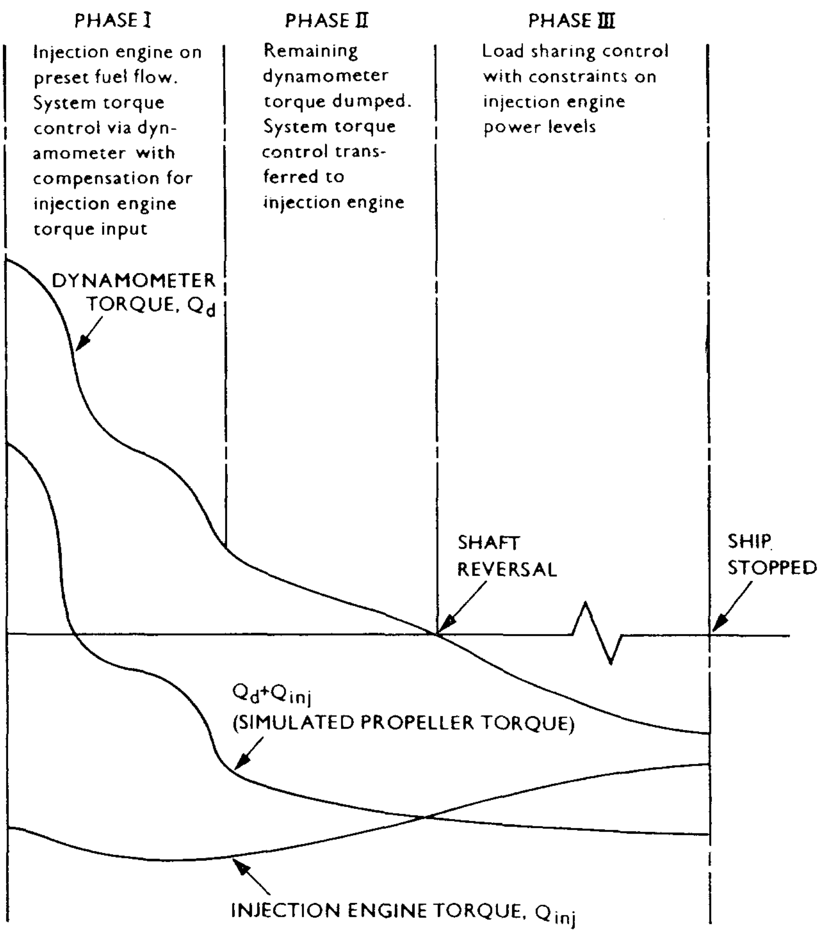

FIG. 6

Torque Sharing and Sequencing System

This control system provides the necessary interface between the torque reference system and the individual torque control loops (described above) to ensure that the overall torque requirements are satisfied and to allocate control responsibility and torque sharing between the dynamometer and the injection engine during the course of a manoeuvre. The main problem encountered in the development of the control system was concerned with the arrangements for timing the 'entry' and 'exit' points of the injection engine. From FIGS. 2 and 3, the power injection requirement occurs in the middle phase of the manoeuvre, indicating that the injection engine is required only during this phase. However, because of the time factors involved in getting the injection engine and coupling into the system and capable of supplying the required injection power at the time required, there was no alternative to arranging that the injection engine and coupling be connected and supplying power to the system before the start of the crash stop manoeuvre. This required the dynamometer to absorb much higher power in order to dissipate that produced by the injection engine, always resulting in the net torque experienced by the driving engine being equal to the ahead propeller torque.

In addition, since the dynamometer is not capable of absorbing the required load at low astern speeds, the period of power injection had to be extended to compensate. At higher astern speeds, the dynamometer can absorb the total required astern load but it was considered that to allow it to do so would pose the difficulty of deselecting (i.e. taking out of circuit) the injection engine and coupling whilst maintaining overall torque control on the system. The most practical solution was to leave the injection engine in the system until the manoeuvre was complete, and to adopt a load sharing control policy for this phase of the manoeuvre.

Because of this, an overall control and sequencing system was determined with the injection engine and coupling in circuit and supplying power to the machinery system throughout the manoeuvre. The system adopted, split the control requirements into three phases (denoted Phase I, II and III). The phase boundaries and typical machinery transients and control modes are shown in FIG. 6.

Application of System to Other Manoeuvres

Although primarily designed to simulate single engine/shaft ship crash stop manoeuvring conditions, the control system can be easily adapted to cater for acceleration and deceleration manoeuvres, which generally require only the control of the dynamometer torque. For these cases, single and twin engine operation can be covered, limited only to the maximum loading capability of the dynamometer.

Computer Controls System

The real time solution of the ship motion and propeller/hull equations required for power injection, and the general complexity of the control system, led to the adoption of a computer-based control system. Consideration was given to both analogue and digital computers since both types could satisfy the primary requirement of torque control. Other secondary considerations, however, led to the selection of a general-purpose digital computer; these considerations included:

(*a*) data logging availability;

(*b*) reduced initial setting-up and commissioning period;

(*c*) reduced setting-up and checkout before each series of manoeuvres;

(*d*) reliability and program security;

(*e*) application to other tasks on the STF and other projects.

Computer Software

Although the primary function of the control computer was to control the machinery during power injection manoeuvres, it was realized that other machinery control modes and system monitoring requirements would almost certainly arise during the course of the project. To allow considerable flexibility of control policies and to facilitate the incorporation of such additional requirements, the computer control system was implemented using a general-purpose control and simulation package with subroutines tailored for each particular function. (In retrospect, this decision has been validated on innumerable occasions, and it allowed the shore trials programme initially conceived to be considerably extended and modified with relative ease.)

The computer/operator interface operates via the teletype and an extensive range of control commands, information requests and system monitoring functions are provided. All control actions and sequencing performed or required by the computer are checked and violations are flagged to the operator. The computer continuously monitors the state of the plant using a set of prescribed limit and rate values for particular machinery and control variables. Again, violations of these allowed values are flagged to the operator via the teletype and, in particular cases, to the plant operator via indicator lights as hazard warnings. The computer operator can request information on any or all system variables at the teletype. The computer software includes a data logging facility by which sixty channels of information can be recorded on a magnetic tape. This facility is intended primarily for detailed trials performance analysis, but can also be used in a more restricted manner for on-site data replay on a U.V. recorder as an aid in trials diagnostics.

Instrumentation Systems

The installation of the computer control and monitoring system was achieved using, with one exception, existing actuators and transducers. The input requirements of the control actuators (dynamometer load-control valves and injection engine throttle) were compatible with the outputs of the computer digital-to-analogue conversion system and direct connections could be made via computer-control/normal-control switching arrangements. The only items of special purpose instrumentation required were power turbine torquemeters.

In addition to a wide range of machinery performance parameters, all significant control system variables (e.g. ship speed, propeller torque, demand torques, etc.) used within the computer were data logged for subsequent analysis.

Safety Considerations

In normal test-house running, human operators have total control over the plant, aided by the normal system protection/warning devices. For power injection manoeuvring using the system described, the operator has control only over the manoeuvring engine, and the computer exercises control over the dynamometer and the injection engine.

It was realized that, during power injection, malfunctioning of machinery control functions (including the computer control channels) could produce unusual failure effects (because of the non-normal operating conditions) which could render immediate diagnosis and operator intervention extremely difficult. For these reasons, a safety assessment study was conducted using simulation techniques.

The study showed that there was little benefit in attempting to recover the manoeuvre in failure conditions and a single action 'terminate manoeuvre' function was provided. This function effectively reduces the fuel flows on each engine to below idling in 100ms and deselects the engines from the transmission system.

The study also recommended the use of protection and warning devices in addition to those already in existence for normal running.

Control System Testing

To fully evaluate and test the computer control system before power injection manoeuvring on the shore trials, the system was subjected to two series of comprehensive functional and operational checks.

The first series of tests were performed using the control computer in conjunction with a hybrid computer model of the machinery system and control consoles to exercise the control system in conditions as near realistic as possible, and thereby to establish the control system parameters for the various trials requirements.

The second series of tests were performed on the shore trials machinery. These examined the performance of the individual torque control loops and the torque reference system in normal machinery conditions (i.e. acceleration, deceleration, steady state). Since the effectiveness of the control system depends somewhat on the operating characteristics of the machinery system, the control system tests were preceded by a fairly extensive series of machinery trials designed to establish those aspects of machinery performance relevant to power injection; the information obtained was used to update the control software as appropriate.

The particular functions of each series of tests are summarized as follows:

(*a*) *Simulated Machinery Tests*

(*i*) Specification of control system parameters;

(*ii*) Sensitivity of control system performance to fixed parameters;

(*iii*) Specification of trials requirements and operational procedures;

(*iv*) Trials data base for comparison with real trials.

(*b*) *Real Machinery Tests*

(*i*) Performance characteristics of dynamometer, engines and fluid couplings;

(*ii*) Evaluation of control system in real conditions:

(*a*) Dynamometer control,

(*b*) Engine control;

(*iii*) Evaluation of instrumentation systems.

Machinery Control During Trials

No particular problems were experienced with the engine control loop, the system response being generally in accordance with computer model predictions. However, considerable difficulty was founded in establishing stable conditions on the dynamometer during the transient tests, particularly during slam accelerations from low shaft speeds. Since these machinery conditions are similar to the conditions that would exist during phase III of the power injection manoeuvres, the observed instability was of great concern. An acceptable response was obtained by increasing the inlet water flow rate, reducing the integral error gain and incorporating a phase lead compensator. The response of the dynamometer in decelerating conditions was satisfactory and consistent with model predictions, indicating that satisfactory torque control in phase I of the power injection manoeuvres would be achieved. The preliminary series of the power injection manoeuvres reflected the above findings. Control of the dynamometer during phase I (basically a load-shedding operation) and engine control during phase II were satisfactory. No difficulties were obtained in the phase I to phase II changeover where control was transferred from the dynamometer to the injection engine and smooth transfers were obtained for all manoeuvres. Phase III (i.e. astern mainshaft) operation posed some initial difficulties in maintaining system stability. This astern instability was not always present, and poor repeatability was evident, suggesting that the load absorption characteristics of the astern dynamometer compartment were inconsistent. It is believed that the problem occurs because the astern dynamometer compartment becomes ‘choked’ due to the relatively long ahead running period with high inlet water flows. This produces maximum increase of load astern initially (which is a desirable feature) but, as the astern speed increases, the dynamometer ‘unchokes’ and the water content reduces rapidly. The resultant reduction in dynamometer torque creates a transient unbalance in the astern load sharing which the engine cannot cope with because of its rate limited (throttle) response. The resultant effect in the system is to set up oscillations and continual load transfer between the dynamometer and the injection engine.

Experiments with various water inlet flows proved unsuccessful, and the problem was overcome by eliminating the astern load sharing controls and allowing the dynamometer to follow a natural astern load line by presetting the astern control valve and forcing the injection engine control system to control the overall torque requirements. The modification was effective for all manoeuvring conditions but required additional care to ensure that the proportion of astern load absorbed by each device was maintained at levels consistent with machinery safety.

This problem arises because the injection coupling will be operating at high slips since each side of the coupling will be rotating in different directions. The power dissipated in the coupling is the power output of the injection engine plus that proportion of the manoeuvring engine power not absorbed by the dynamometer. Thus, to avoid high power dissipation (and hence high oil temperatures) on the injection coupling, the proportion of the astern load absorbed by the injection coupling must be maintained as low as possible. For each manoeuvre, however, there exists a lower limit to the usable injection power because, at low injection-engine power levels, the ahead coupling slip will tend to approach 200 per cent. (each side of the coupling running at the same speed but in opposite directions). Any reduction in injection-engine power (to meet a lower torque demand) can only result in speed reduction on the injection engine, thereby increasing the coupling slip above 200 per cent. The coupling will now be operating in an unstable mode, and torque control becomes extremely difficult (see FIG. 12). The minimum torque that the injection engine and coupling can transmit to the system whilst maintaining the coupling slip below 200 per cent. can be calculated for each manoeuvre. Using the dynamometer torque/speed characteristic, the required L.C.V. position could be found: a setting of 60 to 70 per cent. closed gave satisfactory astern load distribution.

The preliminary manoeuvres were consistent with the predicted behaviour and provided sufficient operational information of the system to proceed with high-power manoeuvring and shaft-stall trials.

Machinery Performance

Manoeuvring Trials

A total of 78 power-injection manoeuvres were performed covering a wide range of single engine/shaft crash stop manoeuvres.

The first series of 20 manoeuvres was concerned with general aspects of machinery manoeuvring performance (e.g. clutch disengagement, power turbine behaviour, ship control sequencing) and to provide data on the transient performance of the hydraulic couplings particularly in high power, high slip conditions.

The second series of 58 manoeuvres was designed to further the investigation of the effects of main-shaft stall on the propulsion system and to examine the effectiveness of proposed stall breaking and stall avoidance procedures.

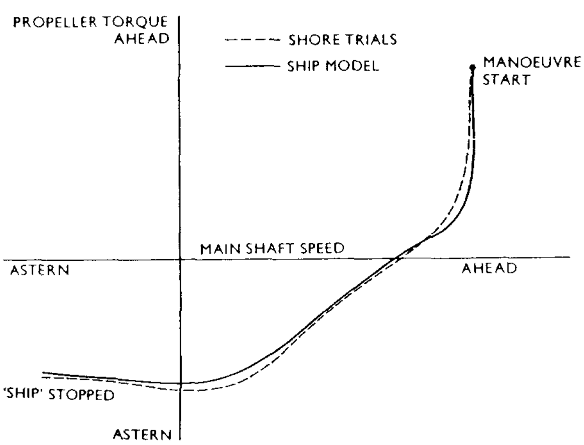

FIG. 7—SHORE TRIALS/MODEL MAIN SHAFT TRAJECTORIES

The automatic transmission mode changeover (clutch to coupling drive and vice versa) were examined in controlled slam acceleration and deceleration conditions.

Dynamic Loading

The trials conducted have demonstrated that the power injection system has adequately represented the expected shipboard loading conditions on the propulsion system.

Comparative torque/speed trajectories are shown in

FIG. 7. The manoeuvre shown is a non-stall manoeuvre from maximum-ahead to maximum-astern power levels. (Further trials/model correlations are shown for the shaft stall trials in FIGS. 8 to 11 inclusive.)

General Machinery Performance

(*a*) *Clutch Disengagement*

For all manoeuvres from the direct-drive condition, the clutch disengaged satisfactorily without assistance from the transient brake fitted on the upstream side of the clutch.

(*b*) *Power Turbine Speed*

The power turbine speed droop during the ahead to astern transmission changeover was not as pronounced as predicted, and is acceptable.

(*c*) *Coupling Oil Temperature*

The peak values of astern coupling oil outlet temperature were signicantly lower than that predicted from simulation studies, the difference being in the range 10–20°C. Some uncertainty in the accuracy of the measured temperatures exists because of the location of the temperature transducers in a down pipe (forced by restricted access) where total oil immersion cannot be guaranteed. However, based on the measurements made, the problem of high coupling oil temperature is not sufficiently pronounced to cause any oil degradation problems.

(*d*) *Ship Control System*

The ship and machinery control system performed satisfactorily during the manoeuvres and all torques, speeds, etc. were maintained within the design criteria.

(*e*) *Shaft Stall Trials*

The various types of manoeuvres performed in connection with main shaft stall fall into the following categories:

(*i*) Stall producing manoeuvres—to determine thrust block loads required to produce consistent stall conditions;

(*ii*) Stall breaking manoeuvres—to investigate three specific types of stall breaking procedures:
  (*a*) self break astern,
  (*b*) stall break astern,
  (*c*) stall break ahead;

(*iii*) Stall avoidance manoeuvres—to investigate a specific type of stall avoidance procedure in conditions which would otherwise produce a stall.

Thrust Block Loading

In order to relate the results of these procedures to the ship case, it was necessary to ensure that the static friction (stiction) loads in the two systems are compatible. There are several reasons why this is not the case: no main-shaft plummer bearings on the shore test system, negligible thrust at the thrust block, and the machinery bearing load reactions different due to the action of the power injection machinery. In order to overcome this problem, provision was made to increase the static friction loads on the shore test system by applying pressure to the ahead and astern thrust block pads, thereby squeezing the thrust collar. Estimates of the required thrust block pressure were made using results of earlier main-shaft breakaway trials—pressures in the range 10 to 20 bar (145 to 290 lbf/in^2) depending on manoeuvring conditions were calculated.

In practice, it was found that these pressure levels were insufficient to produce stall and consistent stalls could only be produced using pressures in the range 75 to 100 bar (1088 to 1450 lbf/in^2). It is thought that this situation arises because the dynamic build up of static friction in the STF is less rapid than on the ship system. The factors relevant to this argument are:

(*a*) On the STF, the stiction loads in total emanate from oil lubricated bearings. On the ship system, contributions to the stiction loads come from both oil and water lubricated bearings.

(*b*) Typical viscosities of oil and water are 40 and 1cSt respectively and therefore the flow from the water lubricated bearings will be greater than that from the oil lubricated bearings, resulting in a more rapid rise in friction coefficient in the water lubricated bearings.

This difference in coefficient rise time is significant as the stiction torque associated with the water lubricated bearings is approximately 25 per cent. of the total ship stiction and, in order to produce the fast initial rate of increase of stiction, thrust block pressures in excess of those calculated from purely static considerations are required. In the stalled condition, however, the high thrust block pressures will impose very much higher stiction levels than estimated for the ship, consequently reducing the correlation to ship conditions in terms of stall duration. The thrust block pressure was dumped to the lower values after stall had occurred in an attempt to maintain 'static equivalence' to the ship case.

Stall Producing Manoeuvres

These manoeuvres were performed to determine the thrust block pressures required in practice to produce consistent shaft stall condition for the various manoeuvring conditions required. An example of such a manoeuvre is shown

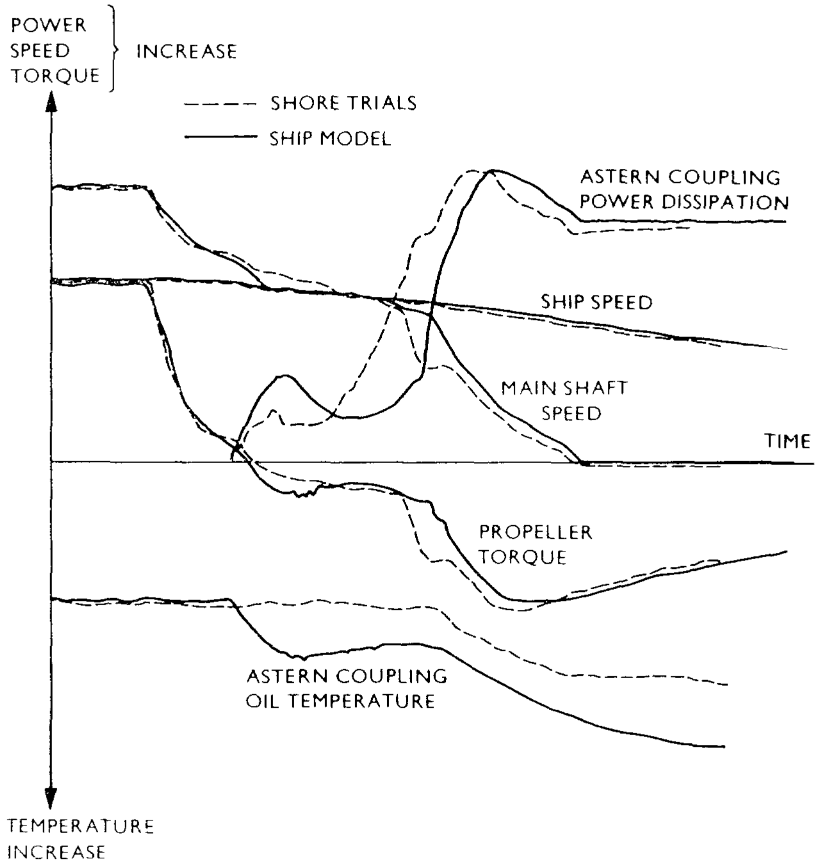

FIG. 8—STALL MANOEUVRE

in FIG. 8, which was performed with a thrust block pressure of 75 bar. The equivalent ship model is superimposed for comparison. The trials/model comparisons shown in FIG. 8 (and FIGS. 9, 10 and 11) were time synchronized at the start of the manoeuvre. The subsequent loss of synchronization is due to the differences between trials and model control system sequencing parameters (i.e. coupling filling rate, engine fuel application time, etc.). This difference could be resolved by running the computer model with the actual trial control system timings. A further point to note particularly in FIGS. 9 and 10 is that the stall break procedures shown are manually initiated, resulting in additional timing differences. The significant feature is not, however, exact time synchronization but the general equivalence of corresponding model and trial variables in terms of magnitude, form, and relationship to each other. In these terms the results shown exhibit very close correlation with the exception of coupling oil drain temperature which has already been noted.

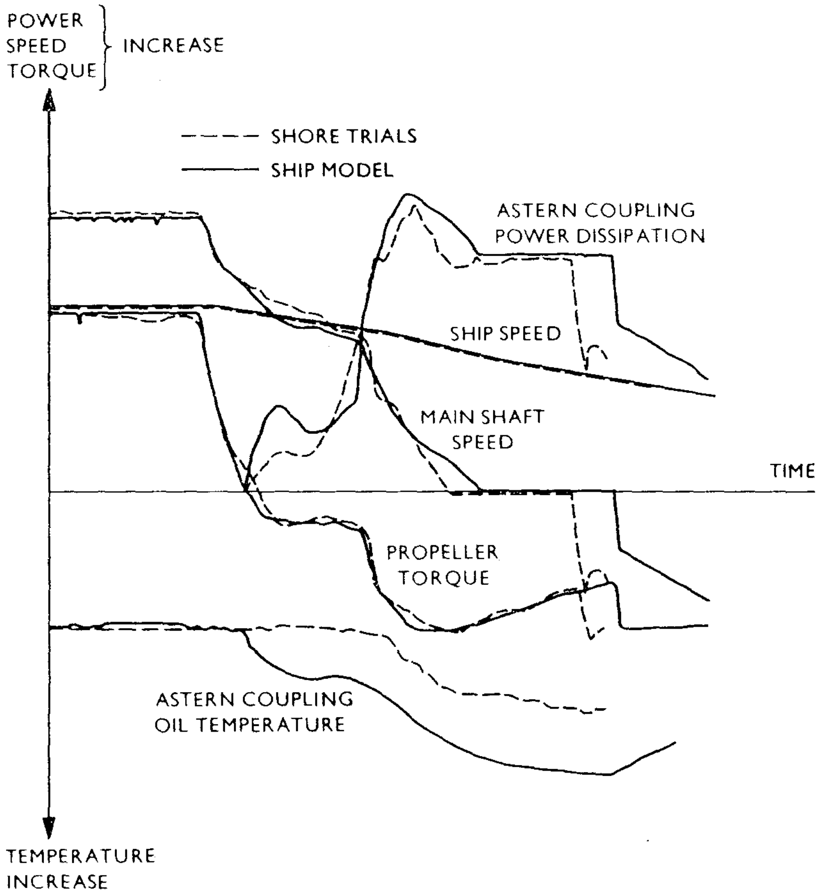

FIG. 9—STALL BREAK ASTERN MANOEUVRE

Stall Breaking Manoeuvres

Three variations of stall breaking procedure were examined:

(*a*) Self-break astern, where the main shaft is left in stall to breakaway astern of its own accord due to the reducing 'propeller' torque.

(*b*) Stall break astern, where the power level on the manoeuvring engine is increased after stall to produce a net increase in 'astern driving' torque on the system.

(*c*) Stall break ahead where the power level on the manoeuvring engine is reduced (to idling) after stall to allow the 'propeller' torque to drive the main shaft ahead with minimal opposition from the manoeuvring engine; and thereafter to re-apply manoeuvring power to achieve astern rotation.

Examples of the stall break astern and ahead manoeuvres are shown in FIGS. 9 and 10 respectively. The manoeuvring conditions and thrust-block pressure are the same in all cases and the equivalent ship model responses are shown for comparison.

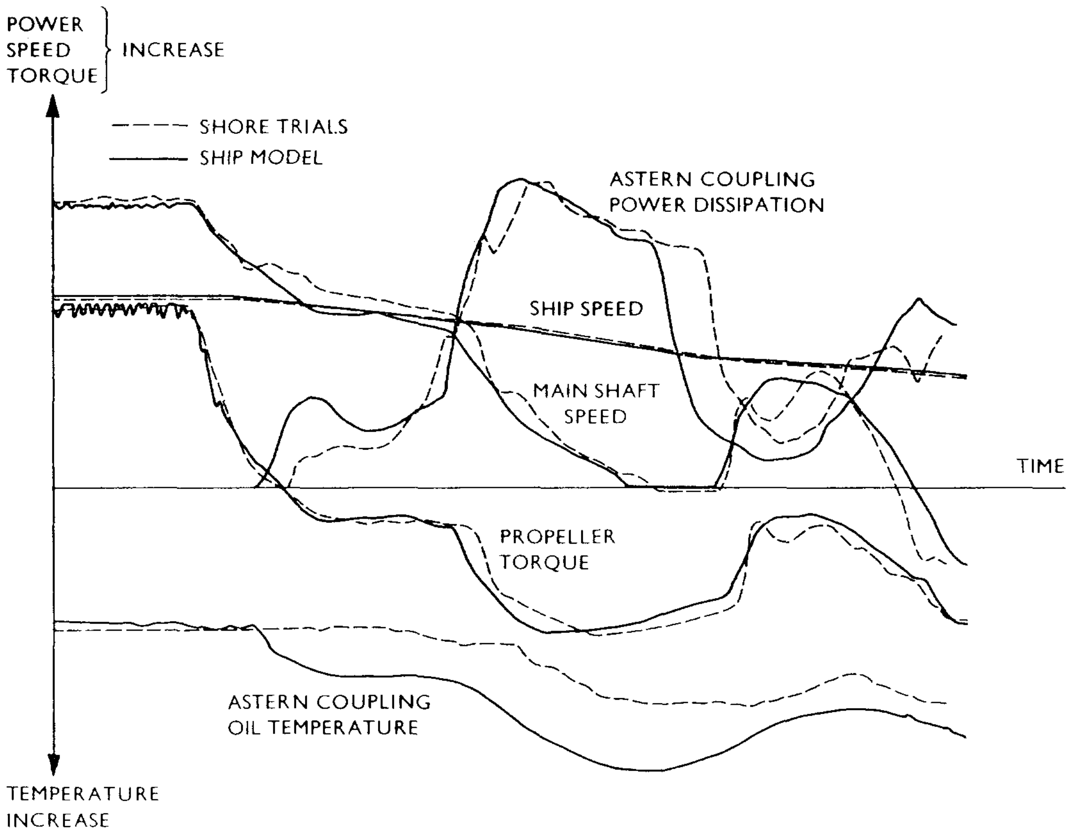

FIG. 10—STALL BREAK AHEAD MANOEUVRE

Stall Avoidance Manoeuvres

Computer simulation studies showed that main shaft stall can be prevented by an additional net decelerating torque applied to the system just before the main shaft reaches zero speed. The procedure adopted for stall avoidance is inherently simple, involving the application of additional power on the manoeuvring engine (by increasing the throttle demand) at a specified low ahead shaft speed. It is known, however, that the additional power applied and the time of application are fairly critical to the success of the procedure and, to ensure repeatability on the shore test, the procedure was implemented as an automatic feature of the system.

An example of a successful avoiding manoeuvre is shown in FIG. 11.

Effectiveness of Stall Break and Avoidance Procedure

It was not always possible to predict with accuracy when stall would occur and, on occasions, stalls happened when no thrust-block loading was applied. This occurs for several reasons, thought to be based mainly in the inaccuracies associated with the control of the manoeuvres. The difference between the 'propeller' torque and the engine driving torque is small, and this small torque less the stiction torque causes the shaft reversal. Hence small errors in

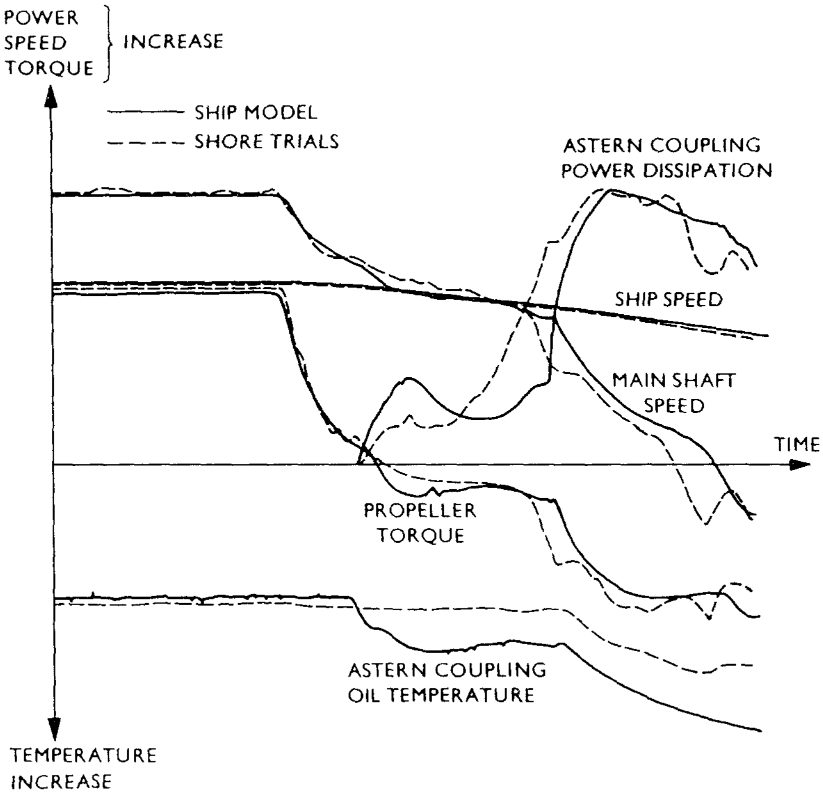

FIG. 11—STALL AVOIDANCE MANOEUVRE

control of the injection engine torque, driving engine torque, and dynamometer torque have large effects on the torque causing shaft reversal and also on bearing loadings and, consequently, the rate of oil loss from the bearing and stiction torque. Some manoeuvres, seemingly identical, were thus found sometimes to stall and sometimes not to stall.

However, consistent shaft stall conditions could be achieved and were sufficient to establish the effectiveness of the breaking and avoidance procedure. In terms of stall-breaking capability, the ahead and astern breaking procedures were equally effective, the stall time being approximately the same for both methods. The astern break procedure is less complicated operationally but produces the more severe machinery conditions (i.e. higher torque levels, higher coupling power dissipation). On the other hand, if the ahead procedure is used, the ship stopping performance will be reduced (since astern thrust will be lost during the period of ahead rotation). On balance, the ahead break procedure would be recommended for the ship case.

The stall avoidance trials confirmed that there exists a fairly restricted envelope of additional throttle/application speed parameters within which the avoidance procedure is successful, but that a common set of parameters could be obtained to cover all manoeuvring conditions examined. The procedure is incorporated as an automatic feature of the ship control system, and is to be extended to cater for possible stalls coming from astern.

Hydraulic Coupling Characteristics

Performance data relevant to the operating characteristics of the hydraulic couplings were obtained from steady state and dynamic trials.

The torque characteristics, defined in terms of a coupling stiffness coefficient, were obtained by driving the gas turbines through opposite sense couplings and, by controlling the power levels on each gas turbine independently,

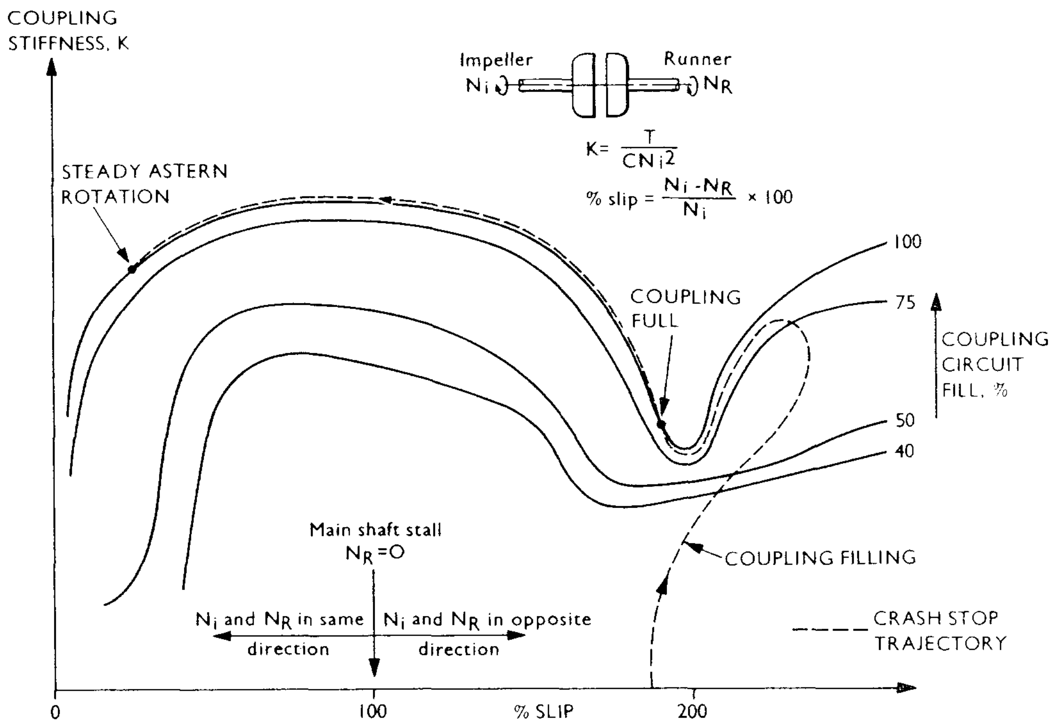

FIG. 12—HYDRAULIC COUPLING CHARACTERISTICS

steady conditions were produced on both couplings over the slip range 5 to 200 per cent. with various filling levels on the couplings.

The torque coefficient obtained is shown in FIG. 12. In general shape and numerically, the data correlated with the previous data obtained from manufacturers and extrapolation from half-size couplings tests for slips up to 200 per cent. The maximum stiffness obtained was approximately 10 per cent. less than that predicted. The characteristics for slips greater than 200 per cent. are defined experimentally for the first time. The trials results indicated that the stiffness of the coupling is also dependent on the coupling impeller speed and oil temperature, but these effects do not significantly affect the overall machinery performance.

The coupling torque coefficient was also calculated from measurements made during the power injection manoeuvres. The trajectory of the coefficient throughout a typical manoeuvre is superimposed in FIG. 12. The power dissipation capabilities of the coupling during crash stop manoeuvres is of importance in determining the manoeuvring power which can be applied by the engines. Previous tests on half-size couplings indicated that there occurred a significant reduction in oil through flow of the couplings, implying an increased coupling pressure, as the main shaft approaches zero shaft speed and, since this period coincides with the period of high power dissipation in the coupling, very high coupling oil temperatures would be obtained. There was no known way to extrapolate confidently to the full-size coupling. The manufacturers advised, however, that the full-size couplings would not exhibit this characteristic and this was in fact confirmed by the trials. From the results, the oil throughflow is independent of coupling speed and slip, remaining practically constant throughout the manoeuvre (except for the filling/emptying periods).

From the tests conducted, the hydraulic couplings have been shown to be adequately capable of performing crash stop manoeuvres and sufficient margins on coupling temperature exist. The comment made earlier on the accuracy of the oil temperature transducers must, however, be borne in mind.

Other Trials Using Power-Injection Systems

A number of other investigations that were carried out under simulated ship conditions were:

(*a*) Demonstration of correct coupling/direct-drive changeovers under slam acceleration conditions.

(*b*) Evaluation of the effectiveness of the coupling filling and the transient brake application in disengaging the direct-drive clutch.

The changeover from coupling to direct drive was shown to be satisfactory and the trials confirmed disengagement of the clutch could be achieved under all manoeuvring conditions. The transient brake is only required to assist clutch disengagement during manoeuvres from low ahead speeds in maximum displacement and fouled hull conditions.

Conclusions

These trials have shown that the relevance of ship manoeuvring performance data obtainable from a normal shore trials facility have been greatly enhanced by the inclusion of power injection to simulate the effects of the hull and propeller characteristics on the propulsion machinery.

Accepting that the hull and propeller characteristic data used in the power injection control system are extrapolated from model tests (and which cannot be confirmed until sea trials), the trial results showed that power injection was successfully applied and that adequate close control of the machinery was obtained. Comparisons of trial results and computer predictions using the full-sized data (i.e. engines and couplings) showed close agreement and will enable the significance of any variations to the ship performance equations to be assessed. The trials carried out have demonstrated the ability of the machinery to withstand transients and the control system sequencing to respond in the prescribed manner. The postulated problems of turbine underspeed and peak coupling temperature are not sufficiently pronounced to cause mechanical damage. Shaft stall during reversal has been thoroughly investigated; the likelihood is that it would have occurred during some manoeuvres in the ship. Shaft stall breaking and avoidance procedures originally derived from computer simulations were shown to work satisfactorily. The stall avoidance procedure has now been incorporated into the control system and will provide an automatic fuel addition near zero mainshaft speed. The procedures for breaking stall which can be evoked by the operators should the need arise have also been demonstrated. There are uncertainties and some facets of bearing stiction which cannot be fully explained; however, sufficient trials have been conducted to ensure that the ship case is adequately covered.

As a result of these trials a great deal of previously unconfirmed transient performance data relating to the propulsion machinery and ship control system have been obtained and may be used to assess any changes in ship configuration (e.g. displacement).

These trials have confirmed the effectiveness of the power injection technique and represent a significant advance in shore testing technique. The experience gained indicates that computer control of machinery could be used for a wider range of trials in any future shore test establishment and would reduce the time required for running manually operated trials.

However, in assessing the need to complete power-injection type manoeuvres on future shore trials facilities, it is necessary to thoroughly analyse the gains from such an exercise. The need to do computer-controlled manoeuvres and the resulting cost of their implementation depends heavily on the machinery configuration. For many configurations, sufficient information may be produced without recourse to complex, closely-controlled manoeuvring trials.

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References:

1. O'Hara, Capt. D. R.N., 'Shore Testing of Gas Turbine Ship Propulsion Machinery' —*Journal of Naval Engineering*, Vol. 22, No. 2, p. 188.
2. Standen. G., Bowes, J., Warsop, Capt. J. C., R.N., 'Machinery Installation in the Type 42 Destroyer'—*Journal of Naval Engineering*, Vol. 22, No. 2, p. 160.
3. Cleland, J. P., Griffin, A. W., Lines, N. P., 'The Control of Power Injection for Cruiser Shore Trials'—*Proc. 4th Ship Control Systems Symposium*, Oct. 1975.