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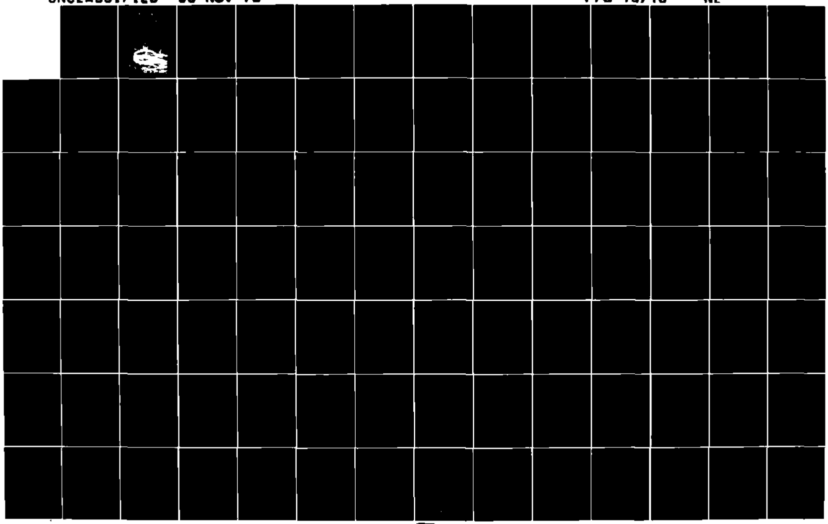

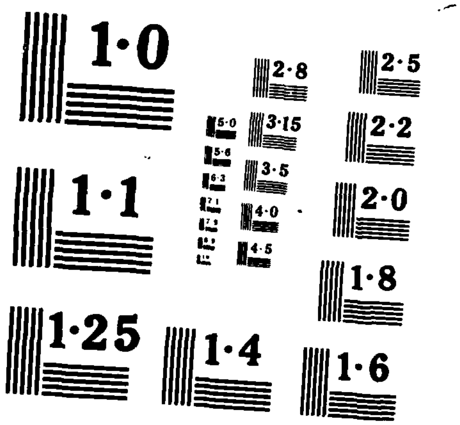
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FIFTH SHIP CONTROL SYSTEMS SYMPOSIUM

OCTOBER 30 – NOVEMBER 3, 1978

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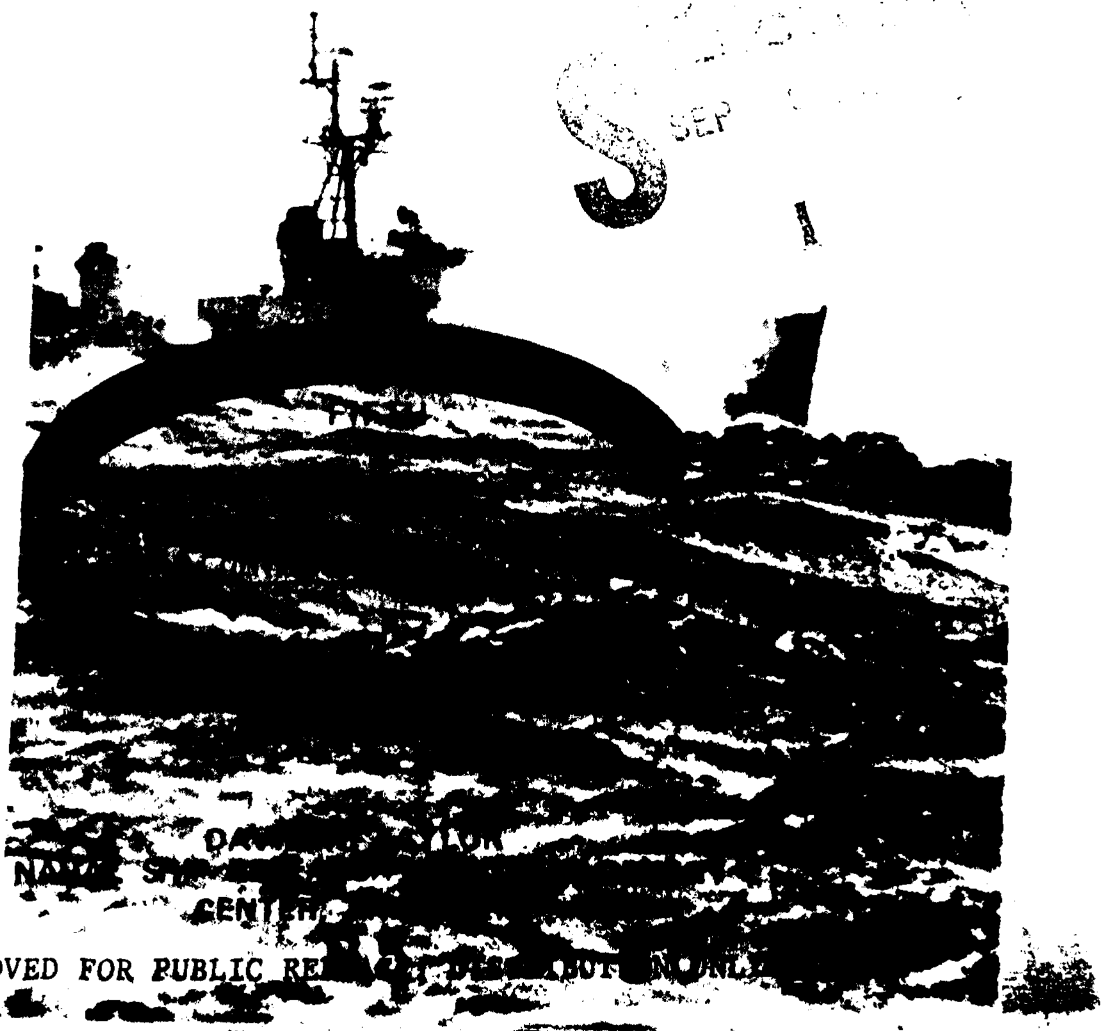

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CONTENTS

AO177 SIMULATION OF SHIP PROPULSION AND BOILER SYSTEMS

by Donald E. Labbe
General Electric Company

ABSTRACT

A computer model of the AO177 Navy Oiler propulsion system was utilized to predict the ship, propeller, and boiler performance during maneuvers. These studies determined (1) the turbine valve throttling rate for maximum maneuvering capability without violating boiler pressure and drum level constraints and (2) the ahead steam flow required to initiate a crash-back maneuver such that the boiler drum level would not exceed its specified limits. These values have been specified for the corresponding marine turbine control ITC * settings.

To predict boiler drum level and pressure during rapid steam load transients, in particular crash-back, a detailed boiler model including the following was utilized: (1) boiler mass and volume parameters (2) boiler transient fluid transport and temperature characteristics, (3) feedwater and fuel flow parameters, (4) desuperheated steam requirements and (5) header accumulator effects.

The ship and propeller speeds were simulated by modelling the following: (1) ITC system including the Malfunction Proportional Control system and throttle valve control utilizing speed feedback, (2) turbine throttle valve flow characteristics, (3) turbine, gear, and shaft inertia and torque, (4) propeller torque, thrust, and efficiency and (5) ship's weight, resistance, and hull efficiency.

The stopping maneuver was most sensitive to turbine valve throttling rate. For example the ahead reach from full ahead (21.5 knots) was 8450, 6000, and 4100 feet for turbine valve throttling rates of 0.75, 1.67, and 50 (crash-back) % per second, respectively. The boiler drum level limits are not exceeded by a steam flow change consistent with a throttling rate of 1.67% per second or less for both upward and downward power maneuvers; however, the crash-back rate of 50% per second can only be used when the initial ahead flow is high to maintain the drum level below the specified normal limits.

INTRODUCTION

Improved ship maneuvering capability could prevent a collision. The ITC, an integrated turbine control developed by General Electric Company, features an automatic crash-back mode. Crash-back decreases the time and distance required to stop the ship over that required by the normal maneuvering throttle rate. Crash-back is defined as closing the ahead valve fully then opening the astern valve fully at the fastest rate allowed by the valve hydraulics (approximately 4 seconds total). The normal maneuvering throttle rate corresponds to the throttle valve movement rate used in maneuvering, usually determined by boiler constraints (typically 1% valve movement per second).

* Trademark, General Electric Company

The objectives of the AO177 Navy Oiler propulsion system computer simulation are to determine the following characteristics of normal maneuvering and crash-back:

1. The optimum normal maneuvering throttle rate which maximizes maneuvering capability without violating normal drum level specifications (± 4 inches) or superheater outlet pressure specifications (540 psig to 618 psig).

2. The gain in stopping capability offered by crash-back from initial ship speeds of 20% to 100%.

3. The effect of crash-back maneuvering on boiler drum level and superheater outlet pressures for initial ahead turbine flows of 0% to 100%.

4. The minimum ahead turbine flow required to initiate a crash-back maneuver such that the normal drum level specifications are not exceeded (± 4 inches).

5. The maneuvering gains of utilizing the Malfunction Proportional Control system which is part of the ITC system to maintain drum level below a specified set point during a crash-back maneuver from any initial propulsion turbine flow.

Large drum level swells in the boiler are usually associated with crash-back maneuvers from low initial ahead turbine flows. There is a very rapid increase in steam demand by the turbine as the astern valve is opened fully at its fastest rate. This rapid change in steam demand causes a drop in boiler pressure and an increase in the firing rate which produces more steam in the boiler evaporator tubes and other components. This steam essentially displaces liquid which appears as an increase in water level in the steam drum.

One way to limit the swell associated with crash-back is to prevent the initiation of crash-back at low initial ahead turbine flows. A throttle valve position sensor has been provided to perform this function.

Establishing limits on the crash-back operational mode may prevent large drum level changes in the boiler, but it limits the potential maneuvering gain of crash-back. For example, if 50% or more ahead turbine flow is required to initiate a crash-back maneuver, the crash-back maneuver cannot be utilized for less than about 80% ship speed in the steady state. Even if a collision situation developed, with the ahead valve at less than 50% open, the crash-back mode could not be initiated.

The ITC includes a Malfunction Proportion Control System (MPC) which provides throttle valve position override control under abnormal conditions. The MPC system senses drum level, superheater outlet pressure, turbine speed, and turbine vibration. In the current commerical design, the MPC drum level control is disabled in a crash-back maneuver. However, my moving the throttle lever from the crash-back position forward to approximately the 100% astern speed position, the drum level MPC is reactivated. In the AO177 ITC system design, the MPC drum level control is currently active in a crash-back maneuver, as specified by the Navy.

The MPC drum level control has two set points. The ahead or astern throttle valve is signalled to begin throttling down below 100% when the measured drum level exceeds the lower set point and to close proportionately between the lower and upper set points. The throttle valve slue rate limit to an MPC signal is about 50% per second. The drum level swell should be maintained below the upper set point when the drum level MPC is active.

Ship maneuvering predictions were generated with a computer model including the following: (1) The General Electric ITC including the Malfunction Proportional Control System (MPC), (2) the General Electric ahead and astern throttling valve flow characteristics, (3) the General Electric turbine and gear torque and inertia, (4) the propeller torque, thrust and efficiency and (5) the ship weight, resis tance, and hull efficiency.

Two Combustion Engineering V2M8WW boilers each rated at 92,600 lb/hr steam production have been selected for the AO177 Navy Oiler.

The predictions of drum pressure, superheater outlet pressure and drum level for the boilers were generated with a model including the following features: (1) two characteristic fluid transport velocities for steam produced by flashing and by combustion, (2) boiler metal and water mass and volume parameters, (3) boiler hot gas, and steam temperature characteristics, (4) feedwater and fuel controls, (5) steam requirements of the propulsion turbine, generator turbine, and desuperheater load, and (6) steam header storage characteristics.

The boiler and ship models were coupled in one computer program and the effects of varying certain parameters on ship and boiler performance were observed jointly. This is consistent with the basic approach of comparing maneuvering gains with boiler performance.

SHIP PROPULSION MODEL

To predict the motion of the ship and its sensitivity to turbine valve movement rates, computer simulation models of the main components were integrated. Modeling relationships from a previous ship motion study were utilized as a starting point (Reference 1) along with the General Electric ADA74 program for solving differential equations (Reference 2). The turbine valve flow characteristics and the turbine and shaft torque, thrust, and inertia characteristics were supplied by the GE Company. The ship and propeller characteristics were provided by Avondale Shipyards, Inc. which were obtained from a model test (Reference 3). Other propeller data for regions not covered by the model test report were extracted from a propeller report (Reference 4) with information on similar propellers.

A block diagram of the ship propulsion model is presented as Figure 1. The essential features of the model are illustrated. A more detailed description of the component modeling is presented below.

ITC

The ITC system links the throttle input to the turbine valve position with propeller speed as the governing parameter. The essential features of the ITC network are the following:

1. Throttle input is converted to valve lift signal.

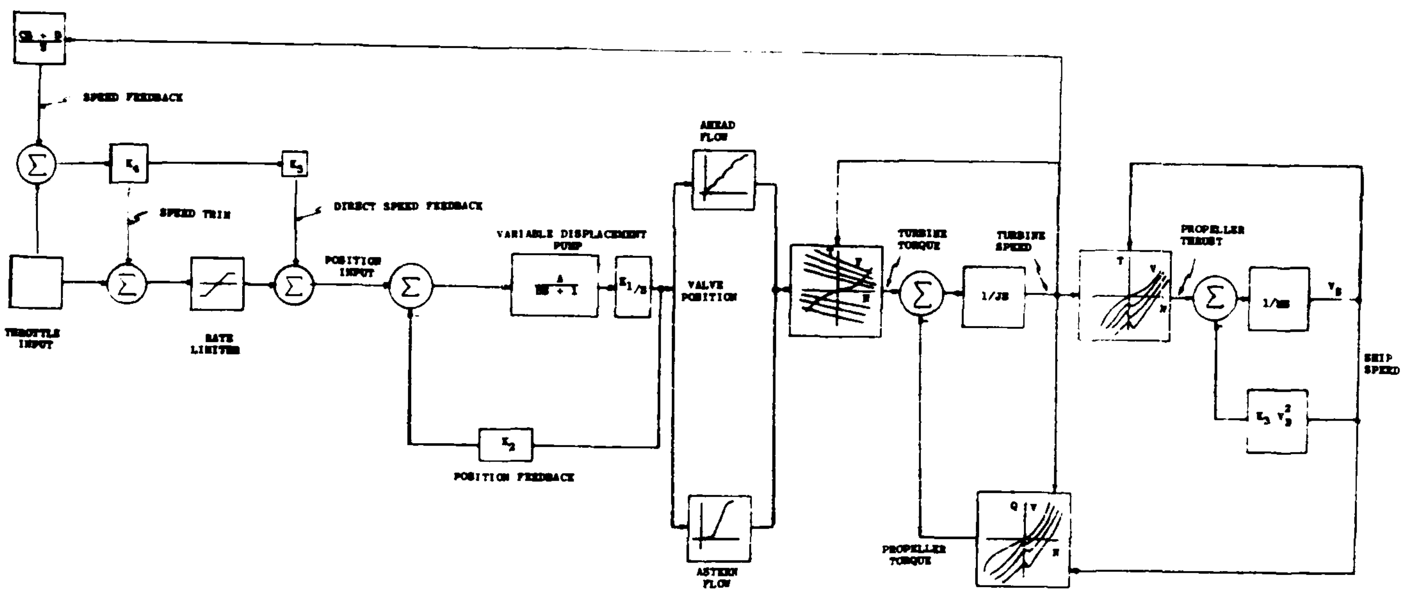


FIGURE 1 TURBINE - SHIP CONTROL SIMULATION

2. Speed feedback interacts with the valve lift signal.

a. Speed trim signal is added to the valve lift signal and the total is input to the valve movement rate limiter.

b. A direct signal from the speed feedback network is added to the desired valve position with a value of up to 6%. The signal bypasses the rate limiter and is, therefore, a step input to the desired valve position input. This circuit is activated for propeller speeds of less than 40% of rated speed.

3. Rate limiter sets the rate of valve opening or closing in % per second.

4. Hydraulic system represents the response of the variable displacement pump to the desired valve position signal.

The gains and lag coefficients used in the model were determined from the electronic circuitry used in the ITC network.

Turbine Valves Steam Flow

The flow characteristics of the ahead and astern valves are given in Figures 2 and 3. These data are used to convert the valve signal from the ITC model to steam flow. A constant pressure steam source at 590 psig astern and 585 psig ahead is assumed. Pressure variations within the specified limits for the boiler are assumed to have minor effects on turbine inlet steam flow.

Turbine and Shaft

The torque characteristics of the turbine are represented by the following equations over the operating range:

$$\text{Ahead, } Q/Q_r = 2.2 - 1.2\ V/V_r$$

$$\text{Astern, } Q/Q_r = 1.0 + 0.4\ V/V_r$$

where:

Q = torque
V = propeller speed
r rated conditions

The total inertia of the turbine and shaft components is 1.316 X 10^6 ft-lb-sec^2. Rated conditions of 100 RPM propeller speed and 24,000 HP are used in the inertia and torque evaluations.

Propeller

The thrust and torque characteristics of the propeller for positive advance velocity and positive propeller rotational speed are given in the David Taylor Model Basin Test Report (Reference 3). These data were put in terms of a modified advanced coefficient, σ, and are plotted in Figures 4 and 5. No data for this propeller were available for negative advance velocity and/or negative rotational speed. Since the simulated ship maneuvers would include operation within these regimes, these data were assumed to be the same as Wageningen Type B propeller with a similar pitch to diameter ratio, a similar blade area,

Figure 2 Ahead Flow Vs. Valve Lift

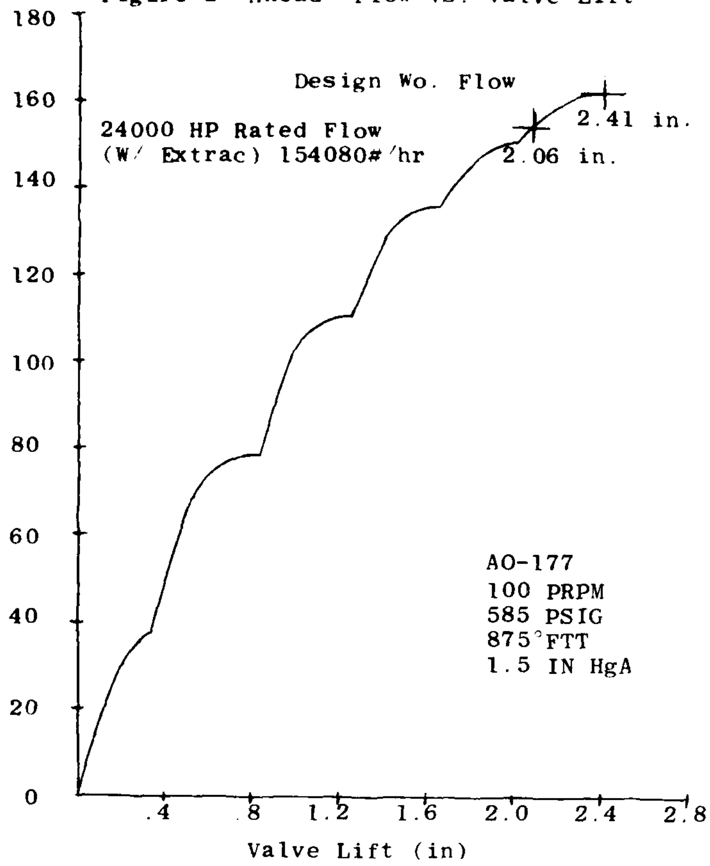


Figure 3. Astern Turbine Flow Vs. Valve Lift

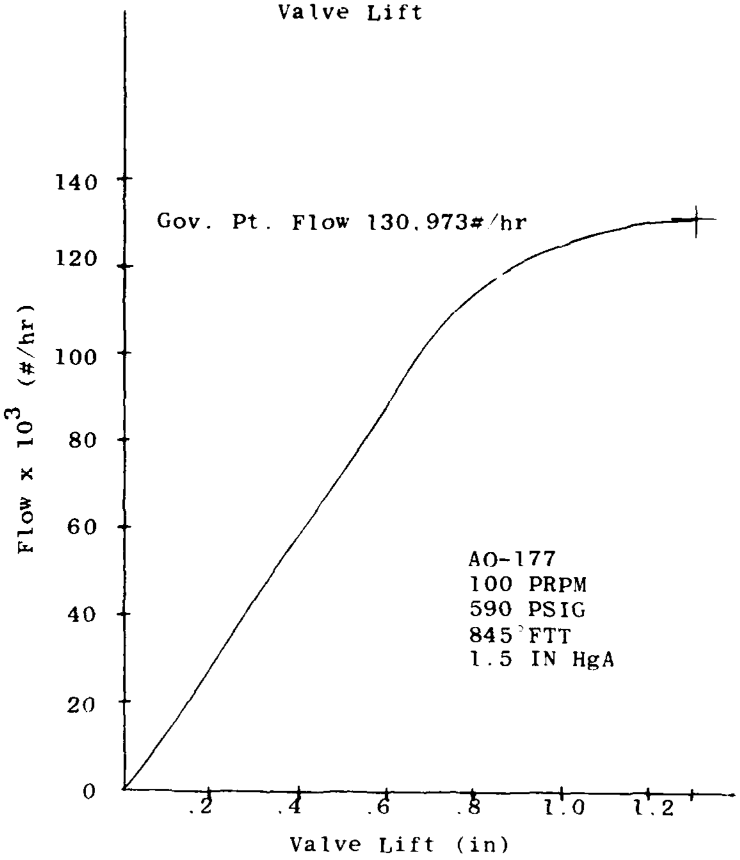


Figure 4. Propeller Torque Coefficient

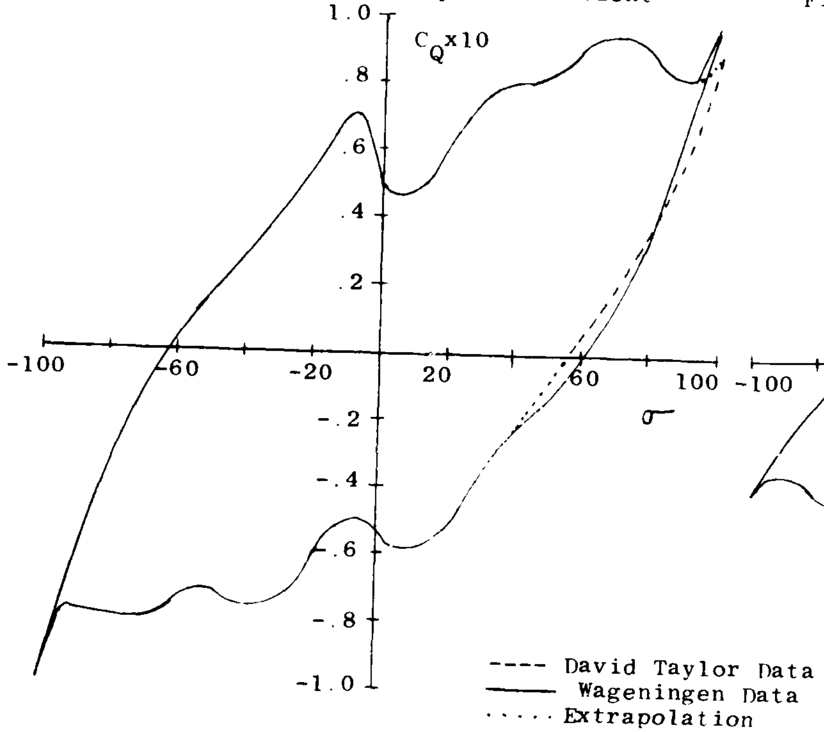
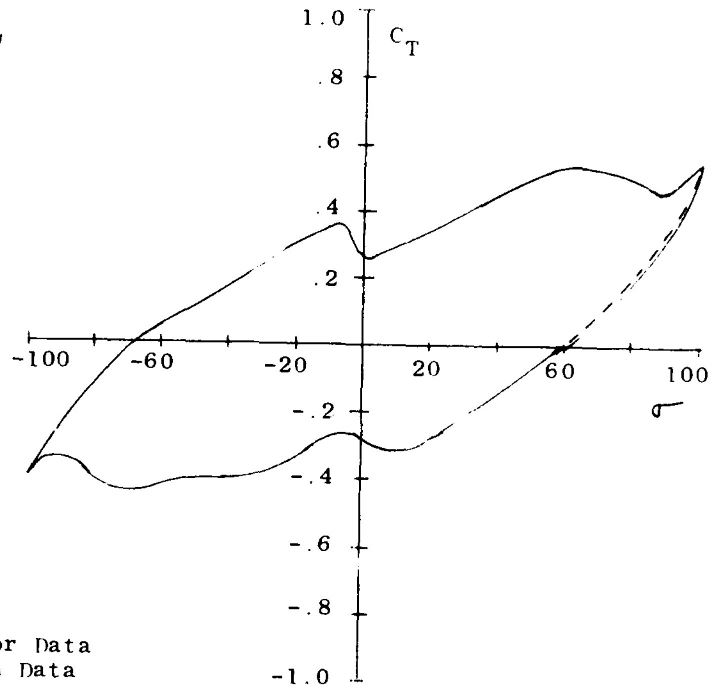


Figure 5. Propeller Thrust Coefficient



To maintain the drum level swell below the normal maneuvering limit of +4 inches (Reference 8), the crash-back mode would not be allowed for initial boiler loads of less than 60% of rated. This corresponds to an ahead turbine flow of about 53% and an initial ship speed of about 82%. This constraint appears overly restrictive, particularly in light of the ahead reach gain offered by crash-back even at low initial speeds as shown in the previous section.

The drum level limited crash-back maintains the steam drum level below the MPC upper level set point. The MPC control circuit limits the drum level by closing the astern valve as the level approaches the upper level MPC set point. Predictions of drum level for the drum level limited crash-back maneuver for initial conditions of 0% to 100% rated speed are also shown in Figure 10. The ahead reach predictions of drum level limited crash-back are included in Figure 9. There is only a small difference between the ahead reach characteristics for crash-back and drum level limited crash-back due to the high MPC upper level set point of 9 inches. If a lower set point value had been selected, causing more closure of the astern valve, the difference in ahead reach characteristics between the two crash-back modes would be larger.

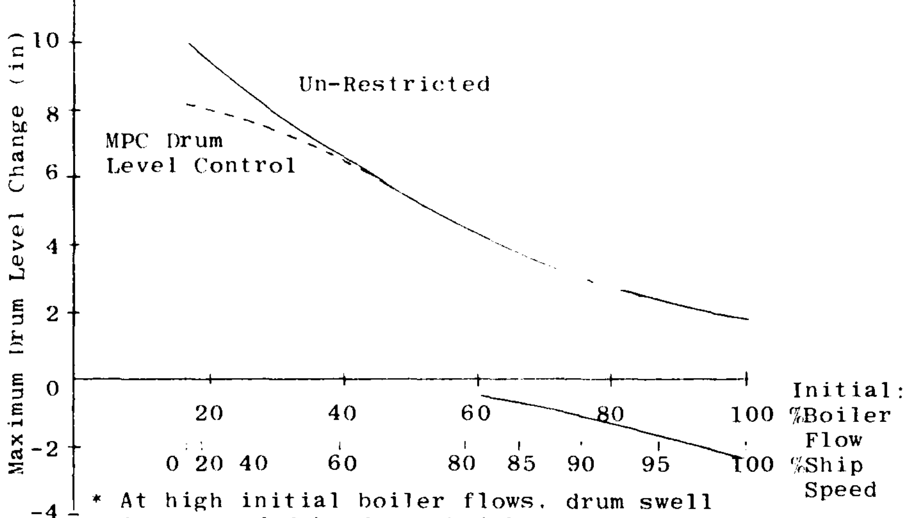
Sensitivity studies of drum level predictions to fuel flow parameters and initial steam load assumptions were conducted for the initial condition of 20% rated ship speed. The effect of varying the fuel flow slue rate limit from 6% per second to 3% and 12% per second produced a change of less than 10% on drum level. The faster slue results in a lower drum swell as would be expected from a faster responding boiler. However,the decrease in swell is not very appreciable (9.7" to 9.3").

These results justify the simplified approach to the fuel flow system. That is, reasonable variations to the assumptions on the fuel flow system model result in small effects on the drum level predictions for crash-back from low initial boiler loads.

The sensitivity of drum level predictions to initial boiler load is also small. In the main study the generator/turbine flow from each boiler was assumed to be constant at 8860 lb/hr each. To obtain a drum level sensitivity to initial load, the 20% initial speed crash-back case was rerun assuming generator/turbine steam loads of zero and 17720 lb/hr each. The predicted drum level rose from 9.7 inches to 10.1 inches for the zero generator/turbine load, and dropped to 9.3 inches for the higher generator/turbine loads. The reason for the change in drum swell is that the evaporator has slower fulid transport properties at lower loadings resulting in longer steam residence time and consequentially larger swells.

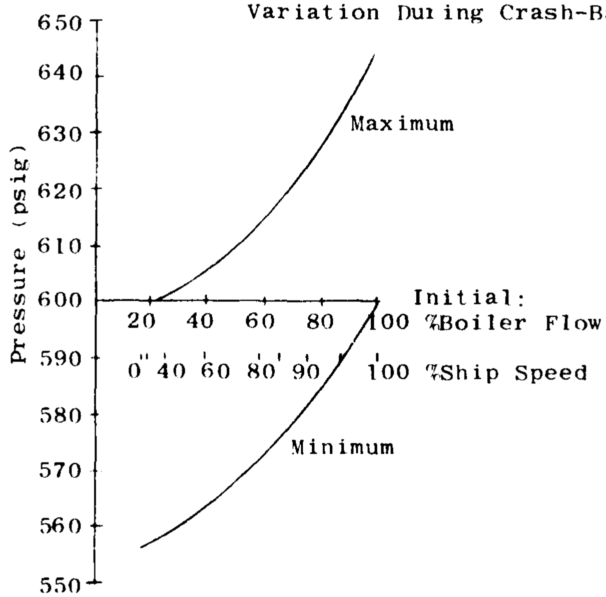
For high initial boiler loads the superheater outlet pressure violates the upper limit specification of 618 psig during the first portion of the crash-back maneuver. For the initial condition of 100% ahead steam flow, the steam drum pressure is about 647 psig with a superheater outlet pressure of 600 psig. In a crash-back maneuver the ahead throttle valve is fully closed before the astern valve is open. Due to the decrease in superheater flow, the superheater pressure reaches approximately 644 psig. As the astern valve is opened, the superheater pressure quickly falls below the upper limit specification. Since there are MPC controls on superheater outlet pressure, the pressure set points or instrumentation should be selected to allow the crash-back maneuver to occur without interferring with the normal

Figure 10. Maximum Drum Level Change For Crash-Back Maneuvers



* At high initial boiler flows, drum swell is preceeded by drum shrink.

Figure 11. Superheater Outlet Pressure Variation During Crash-Back



Crash-Back Vs. Normal

Crash-back offers an ahead reach advantage of 14% to 31% over the normal maneuvering throttle rate (1.67%/sec) for initial ship speeds of 20% to 100%. Comparisons of ahead reach for initial ship speeds of 20, 40, 60, 80, and 100% at throttle valve movement rates of 50% (crash-back), 1 2/3%, and 1% per second are illustrated in Figure 9. The ahead reach advantage of crash-back is even more attractive (22% to 44%), if the throttling rate is 1%/second.

Ahead reach predictions were also generated for the drum level limited crash-back, a mode of operation in which MPC drum level control is activiated. The following assumptions were made:

1) The MPC lower and upper drum level setting were +5 inches and +9 inches,

2) The astern valve would be signalled to close proportionally between +5 inches and +9 inches, that is, fully closed at +9 inches,

3) The drum level signal to the MPC was represented by a first order lag (a time constant of 0.25 seconds) of the predicted drum level.

These lower and upper drum level limits are consistent with the alarm points specified in Reference 8. The lag is representative of instrumentation response time.

The predicted results show that the drum level limited crash-back has nearly the same ahead reach advantage as the crash-back mode. The results are also presented in Figure 9. Obviously, these results are highly dependent on drum level. If the actual drum level change is more severe than predicted, there would be a larger difference between the ahead reach values for the crash-back and drum level limited crash-back. Likewise, for smaller drum level changes the difference is less.

As the initial ahead ship speed is increased, the drum level swell becomes smaller and the difference between ahead reaches for the crash-back mode and drum level limited crash-back mode become smaller.

Steam Drum Level and Superheater Outlet Pressure During Crash-Back

Crash-back maneuvers from low initial boiler loads will result in large steam drum level excursions. The predicted drum level swells for crash-back maneuvers from initial ship speeds of 0%, 20%, 40%, 60%, 80%, and 100% of rated are illustrated in Figure 10. The largest predicted swell is 10.1 inches and corresponds to the 0% initial ship speed crash-back maneuver. This swell barely exceeds the guage glass limit of 9 inches. The consequence of such a large swell would be to decrease the effectiveness of the steam separator and allow more water droplets to pass into the superheater. If water droplets come in contact with the superheater tube walls, evaporation occurs leaving mineral deposits on the tube interiors. Significant deposits could reduce the heat transfer properties of the superheater. However, the duration of the swell is so short (less than 30 seconds) that little effect on the superheater would be expected.

Figure 8. Zero To Full Ahead; Ship Speed Vs. Time

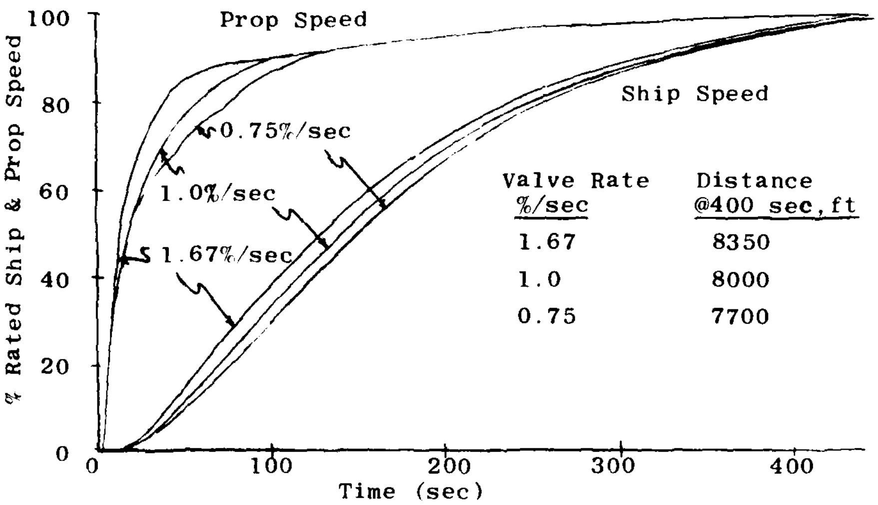
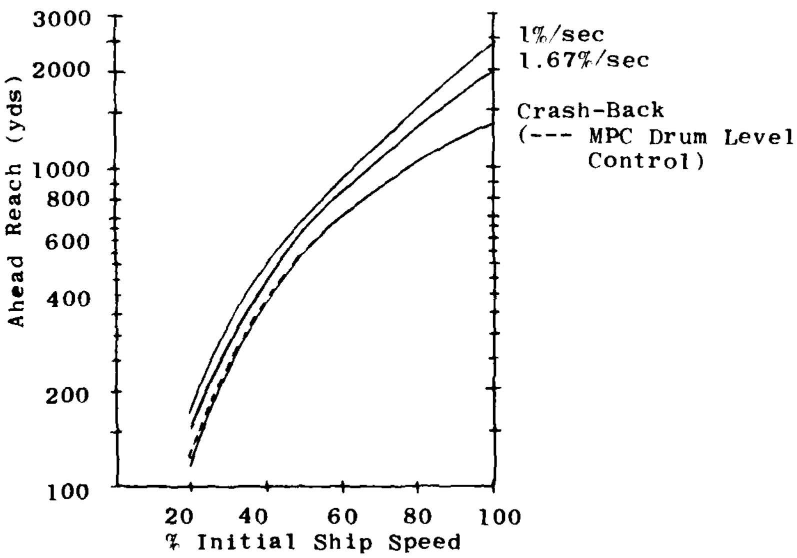


Figure 9. Ahead Reach For Crash-Back And Normal
Throttle Valve Maneuvering Rates



the feedwater was automatically controlled by a Copes-Vulcan system. Fuel oil, air, and atomizing steam were on hand control. The drum pressure (not superheater outlet pressure) was maintained within -10% and +4% of the reference pressure.

The only reported results of the test are the maximum or minimum water level change and the maximum superheater outlet temperature. No details are available on the actual steam, feedwater, and fuel flow changes during the transient. The following assumptions were made in the model:

1) Linear steam flow change Vs. time,
2) High pressure gain on the fuel flow control.

It is assumed that the controls were adjusted to minimize drum level change for the given transient. That is, the test results are assumed to represent the minimum possible drum level changes for the given flow changes. Also, the drum pressure is less sensitive to load changes than the superheater outlet pressure. This justifies the use of high pressure gains on fuel control which tend to minimize predicted drum level changes.

The predicted drum level changes are listed in Table 2 for comparison with the test data. The assumed fuel flow gain of the drum pressure control is 8% psi. The model predictions on drum level swell are within 10% of the test data.

Table 2. Boiler Drum Level Data and Boiler Model Predictions

Steam Rate From	Steam Rate To	Time Seconds	Drum Level Change Inches	Predicted Gain = 8%/psi
10	80	45	+3	+2.9
80	10	45	-3	-3.2
10	80	28	+4	+4.4
80	10	28	-2 1/2*	-5.3

* This data seems suspect, since this transient resulted in less shrink than the slower one above.

SHIP MANEUVERING PREDICTIONS

Normal Maneuvering

The maneuver selected for analysis was "zero to full ahead". Valve movement rates of 1.67, 1.0, and 0.75% per second were simulated. The resultant steam flow and ship maneuvering characteristics are illustrated in Figure 8.

The time required to reach 90% of ship's rated velocity from zero initial velocity decreases from 337 to 318 seconds as the turbine valve movement rates are increased from 0.75% to 1.67% respectively. The relatively large change in valve movement rate (greater than 2:1), therefore, has little effect on the overall ahead performance of the ship.

5) Auxiliaries including evaporators, losses, steam atomizers, etc.

The flow to the propulsion turbine is directly related to the ahead and astern valve positions assuming steam at the rated conditions (Figures 2 and 3). The effects of changes of pressure and temperature on turbine flow were neglected. The error in turbine flow at 10% boiler flow is about 10% due to the low superheater outlet temperature at low loads (approximately 675°F at 10% flow and 865°F at 100% flow). However, the superheater outlet temperature rises rapidly with boiler flow and the error rapidly becomes smaller (50% boiler load, 825°F and 2% flow error).

The steam flow to the generator turbine is assumed to be constant at 8860 lb/hr for each boiler during maneuvers. Also, the steam flow to the group designated auxiliaries is assumed to be constant at 4750 lb/hr for each boiler. The values are based on Avondale heat balances provided for the A0177 system.

The desuperheated steam requirements for deaerator makeup and feedwater pump turbine drive are lumped together and dependent on turbine extraction flow and feedwater flow. In the model the crossover extraction supplying the deaerator and the low pressure extraction feedwater heating are automatically activated at 50% turbine ahead flow. At lower flows all the steam required by the deaerator is supplied by the desuperheaters and feed pump turbine exhaust. For a crash-back maneuver the desuperheated steam requirements for deaerator makeup and feed pump turbine represent as much as 15% of the total boiler flow. Consideration of the transient behavior of the desuperheater flow is therefore important.

The primary desuperheater is in the steam drum. The heat removed from the superheated steam serves to evaporate water in the steam drum. This feature has been included in the model. However, the rate of steam production by the desuperheater is very small compared to the rest of the system and its effects are minimal.

In summary the calculated boiler flow is the sum of the base load (generator turbine and auxiliaries), propulsion turbine flow and desuperheated steam flow (including deaerator makeup steam).

Steam Header Effects

The long steam header running from the boiler to the propulsion turbine acts as an accumulator. A model was developed to predict the boiler flow at one end of the pipe as a function of the turbine flow at the other end. The results indicate that the header has very little storage capacity and has virtually no effect on the pressure and level calculations for the boiler system.

RESULTS

Boiler Model Qualification

The qualification of the boiler model is based on the good comparisons with test data from a V2M8 test boiler (Reference 7). Four maneuvers were analyzed; two upward power maneuvers and two downward. The boiler steam load change for the upward maneuvers was 10% to 80% in 45 and 28 seconds. Likewise for the downward maneuvers the steam load change was 80% to 10% in 45 and 28 seconds. During the test

1) +4 inches of drum level from set point yields full scale signal (proportional),
2) 100% flow error between steam flow and feedwater flow yields full scale signal (proportional),
3) Reset rate, 0.5 reset/minute.

The following modelling assumptions were made relative to the feedwater control valve:

1) Maximum flow limit of 140% of rated,
2) Linear flow characteristic,
3) Response of feedwater flow to signal represented by a first order lag with a 2.0 second time constant.

The pressure and level predictions were not overly sensitive to reasonable feedwater control valve assumptions, particularly for rapid maneuvers like crash-back.

The fuel control is designated by the specification as a proportional plus reset control based on superheater outlet pressure. The control settings were not specified, and the following values were selected to establish reasonable boiler responses for this analysis:

1) Proportional signal; 3.5% change in fuel flow per psi superheater outlet pressure error,
2) Reset rate; 0.5 reset/minute.

These settings maintain the superheater outlet pressure within 2 psi of the specification for the most severe downward maneuver at the normal throttle rate (specification = 618 psig, calculated = 620 psig). The following modelling assumptions were made relative to the fuel system:

1) Maximum fuel flow limit of 120% of rated flow and minimum firing limit of 5% of rated flow,
2) Linear flow characteristic,
3) Slue rate limit of 6% per second,
4) Response of fuel valve to signal represented by a first order lag with a 1.0 second time constant.

The maximum and minimum firing limits are consistent with the load requirements of the boiler specifications. The slue rate limit is consistent with the value used by Combustion Engineering in their drum level analysis (Reference 6) and is representative of the rate at which the fuel system can change load.

The pressure and level predictions of crash-back maneuvers are not overly sensitive to the fuel system modelling assumptions. Faster fuel responses result in slightly lower drum level swells. Results of sensitivity studies are presented in a later section.

Boiler Steam Flow During Maneuvers

The steam load is assumed to be equally divided between the two boilers. The steam flow is divided among the following components:

1) Propulsion turbine
2) Generator turbine
3) Deaerator for makeup
4) Feedwater pump (turbine drive)

ported out of the steam drum. The increase of volume occupied by vapor in the evaporator, water drum and downcomers due to flashing and combustion heating is limited by the discharge rate of steam being exhausted from the evaporator tubes. The rate of steam discharge into the steam drum is governed by the fluid transport velocity in conjunction with the vapor content within the evaporator.

For example, in a normal upward maneuver, the boiler steam flow increases slowly with time. In the first portion of the transient the boiler pressure drops and flashing in the steam drum and evaporator occurs. The rate of steam discharge from the evaporator tubes is less than the steam production rate in the tubes. The increased vapor content within the tubes results in a larger driving head gradually increasing the transport velocity. As the vapor content in the tubes increases, the steam discharge rate from the evaporator tubes increases and finally surpasses the rate of steam production in the evaporator. The volume of the evaporator occupied by vapor gradually decreases bringing the drum water level back down until a steady-state condition is reached.

The maximum increase in drum swell is therefore related to the rate of change in fuel flow (gain of the fuel control circuit) and the rate of change of steam flow.

For the crash-back maneuver the increase in steam flow can be as much as 65% in 2 seconds. The drum pressure drops rapidly initially, and flashed steam with a long residence time is produced at a high rate in the boiler. The rate of flashing is reduced as the combustion evaporation increases. When the combustion heat input maintains the drum pressure, the flashing ceases and the drum level rise reaches its maximum and begins to decline toward the set point level.

When only one characteristic fluid transport velocity is used, the predicted drum level swells of either normal maneuvers or rapid maneuvers are reasonable, but not both. That is, a faster fluid transport velocity must be used with normal load change rates to predict the low drum level swells recorded in the data. But to predict the high drum level swells associated with very rapid maneuvers, a much slower fluid transport velocity must be used. The method of using two characteristics fluid transport velocities within the boiler model does however, reasonably predict drum level swells for both normal and rapid maneuvers.

The values of the two characteristic fluid transport velocities are based on Combustion Engineering steady state circulation data (Reference 6), provided for this study. The velocities are functions of the volume of the evaporator occupied by vapor.

Any effect of the economizer on drum level changes has been neglected. A constant economizer outlet temperature of 345°F was assumed.

Feedwater and Fuel Controls

A three element feedwater control system has been specified for this boiler. No settings for the controls were designated, since these are usually tuned in the field during initial boiler startup. Therefore, values were selected for fast stable response for purposes of this analysis. No implication is made that these are the optimum settings, in lieu of final tuning of the boiler in the field. The control settings used for the crash-back analysis are as follows:

pressure falls or rises according to the sign and magnitude of the energy imbalance. For example, in the first portion of an upward maneuver the steam produced by combustion heating is not sufficient to provide the required steam flow. This difference is supplied by flashing saturated water in the steam drum and evaporator tubes. If the pressure drops at a high rate, water in the water drum and downcomers may flash as well. When the net steam produced by combustion heating equals the steam flow requirement, the pressure stabilizes.

The boiler metal acts as both a heat source and a heat sink. If the boiler pressure drops, the saturation temperature also decreases and the heat stored in the metal is transferred to the water according to the temperature difference. If the boiler pressure increases, the saturation temperature rises and some of the combustion heating is absorbed to warm the metal.

Separate consideration is given to the superheater which has a range of outlet temperatures from about 675°F at 10% load to about 865°F at 100% load. In an upward power maneuver the superheater metal warms up thereby reducing the heat available to the main evaporator bank area. For a downward power maneuver the superheater metal cools off allowing extra heat to reach the main evaporator bank. Since the superheater metal tends to slow down the evaporator response to combustion heating, the effect was included in the model.

Steam Drum Level Dynamics

The major differences between the steam drum level model for normal maneuver analysis as in References 1 and 5 and this model is the increased degree of detail included to predict drum level excursions during very rapid steam load changes that occur during a crash-back maneuver. Two characteristic fluid transport velocities were used instead of one average value. The first of these transport velocities was associated with the steam produced by combustion. The second was associated with the steam produced by flashing. Due to the geometric arrangement in the boiler, most of the steam produced by flashing travels through the main evaporator bank.

The main evaporator bank represents more than 60% of the evaporator volume in the boiler. The water drum is directly below the main evaporator bank. Flashed steam in the water drum would tend to rise and flow through the main evaporator bank.

Since the tubes in the main evaporator bank are of smaller diameter than the furnace area tubes and have a lower rate of vapor production (less driving head), the fluid transport velocity is substantially lower than the fluid velocity within the furnace area. A slow transport velocity results in a long residence time. Since most of the steam produced by combustion particularly at lower steam loads travels through the furnace area and has a much higher velocity, the residence time is much shorter. Thus, the steam produced by flashing has a much longer residence time in the evaporator than the steam produced by combustion.

The drum level swell is directly related to the increase in the volume occupied by vapor in the boiler components (i.e. evaporation tubes, water drum, and downcomers). The vapor displaces liquid and this displaced liquid volume is absorbed by the steam drum. In normal upward maneuvers the drum pressure drops gradually until the combustion evaporation heat input produces more net steam than is trans-

Figure 7. Boiler Model Components

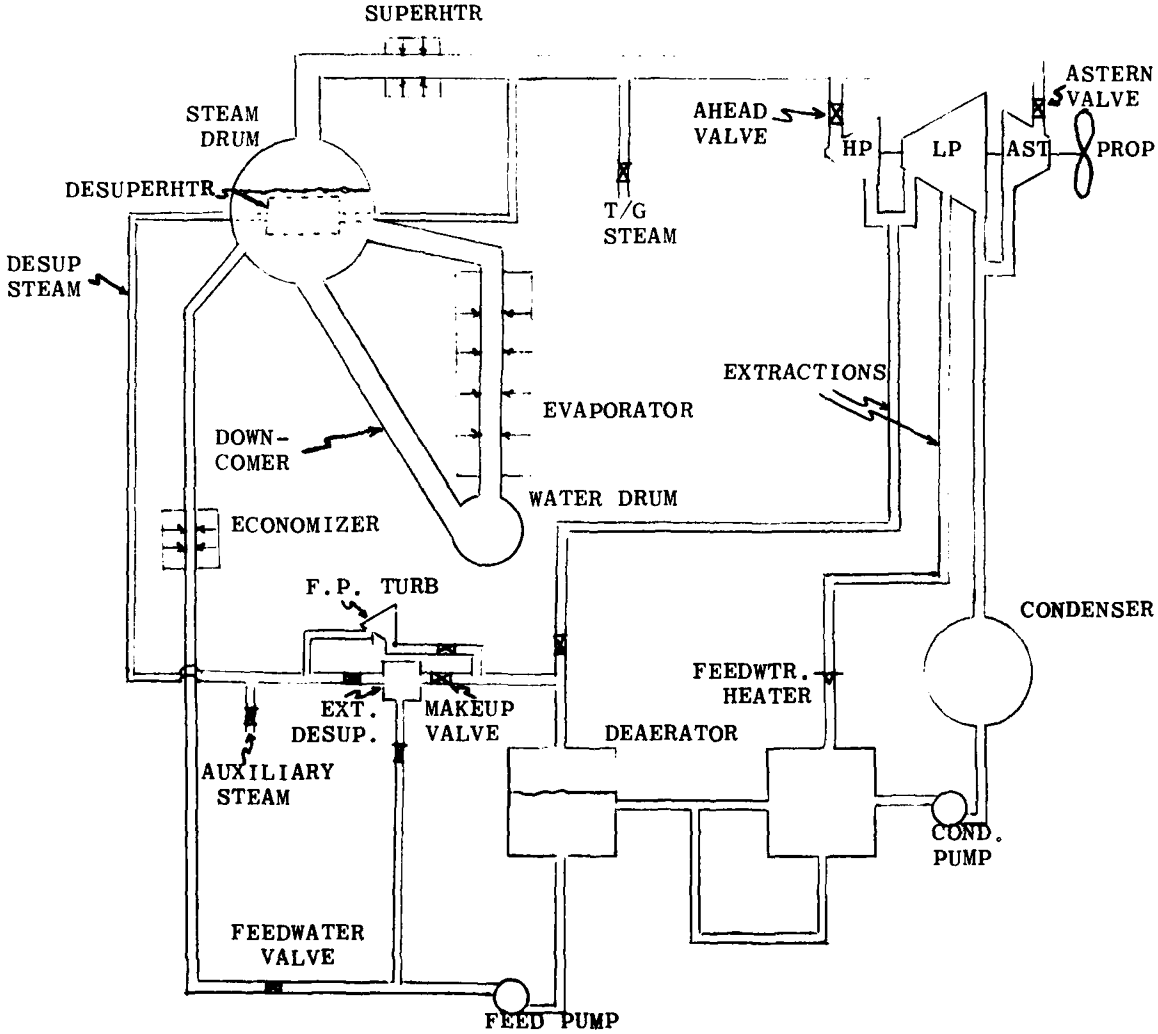
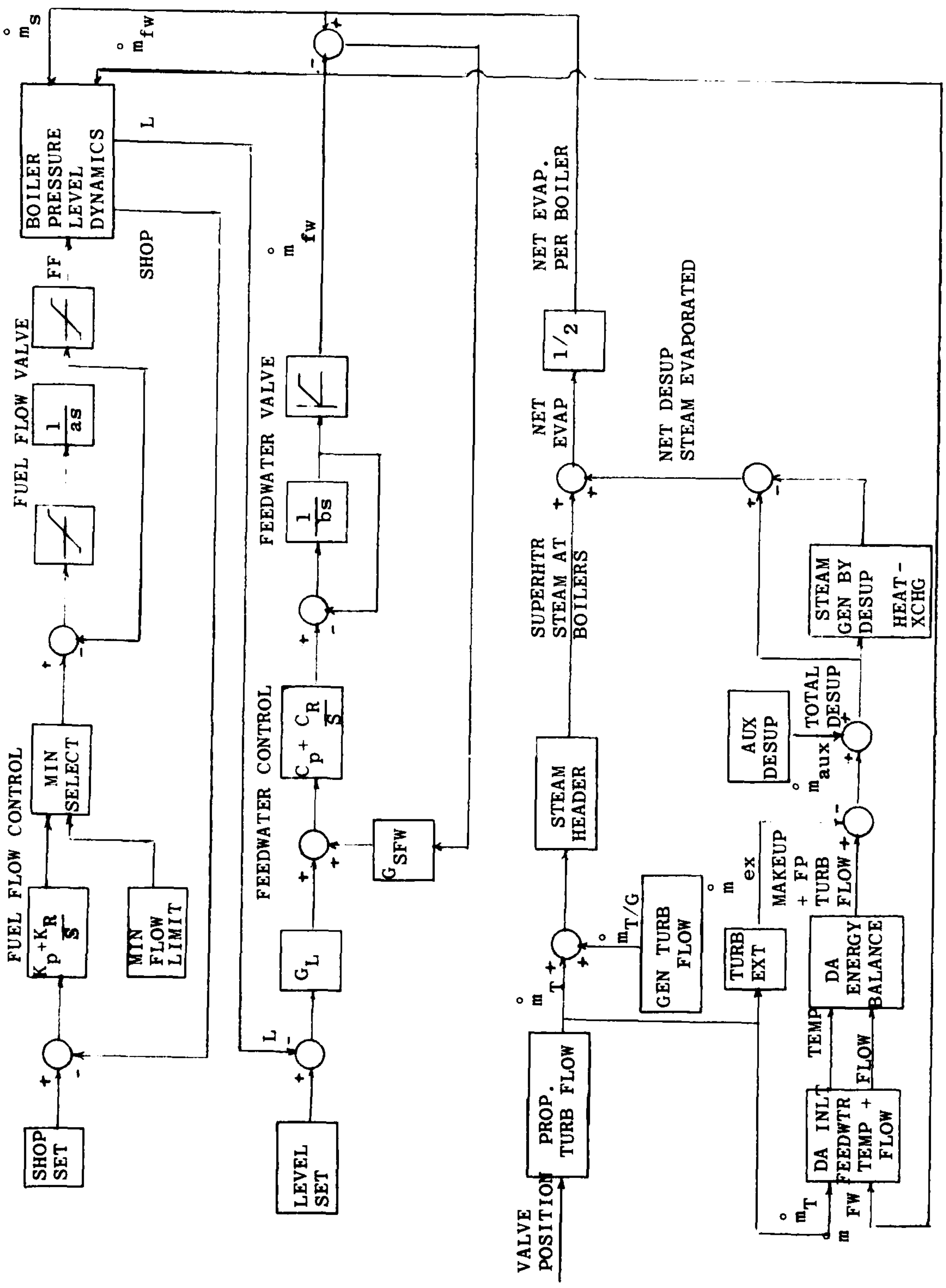


Figure 6. Block Diagram of Boiler Model



BOILER SIMULATION MODEL

Model Components

The boiler simulation model was utilized to predict the key boiler operational parameters; drum level, drum pressure, and superheater outlet pressure. A block diagram of the boiler model is presented in Figure 6 and a schematic of the features included in the model is presented as Figure 7. The model uses equations and modeling assumptions similar to those used in a previous Navy study for the DE1052 (Reference 1). The following discussion presents the interrelationship between the boiler model components.

Fuel Flow. (1) The steam flow, feedwater flow, and fuel flow (heat input) are entered into the boiler pressure and drum level dynamics block. (2) The error between the reference pressure and the calculated superheater outlet pressure is entered to the fuel control block. (3) The fuel control is a proportional plus reset and signals the desired valve position to the fuel valve block. (4) The fuel valve dynamics is simulated by the following (a) slue rate limit, (b) minimum flow limit, (c) first order lag valve response and (d) a linear flow travel curve.

Boiler Steam Generation. (1) The propulsion turbine flow is determined by the ahead/astern throttle valve position block. The steam is assumed to be at the rated temperature and pressure. (2) The turbine/generator flow block assumes a constant flow of 17,726 lb/hr in accordance with the Avondale heat balances. (3) The steam header block simulates the accumulator effects of the steam header on boiler superheater steam flow. (4) The turbine extractions block simulates the extractions for feedwater heating and deaerator make-up as a function of propulsion turbine flow. (5) The deaerator make-up steam flow and the boiler feed pump drive turbine flow block calculates the desuperheated steam flow required for the feedwater pump turbine and the deaerator make-up as a function of feedwater flow and turbine extraction flow. (6) The auxiliary block assumes the auxiliary desuperheated steam load is 9500 lb/hr including evaporators, atomizers, losses, etc. (from Avondale heat balances). (7) The boiler steam generation block determines the total boiler evaporation load from both the superheated and desuperheated steam loads.

Feedwater Flow. (1) The drum level and the difference between steam flow and feedwater flow are input to the feedwater control block. (2) The feedwater control is a proportional plus reset and provides the desired feedwater valve position signal to the feedwater valve block. (3) The feedwater valve dynamics is simulated with a first order lag valve response and a linear flow travel curve. (4) The feedwater flow is input to the boiler pressure and level-dynamics block, desuperheated steam flow block and feedwater control block.

Pressure Dynamics

The simulated drum pressure and superheater outlet pressure dynamics are essentially the same as used in the DE1052 study and the MST-14 Marine Reheat study (References 1 and 5). Pressure calculations are based on a control volume approach to the boiler including the steam drum, evaporator, water drum, and downcomers. The pressure change is related to the system energy imbalance, that is, the difference in energy between the outgoing steam and the combination of incoming feedwater and combustion evaporation heating. The system

Ship movement,

$$(1 + \gamma)\, m \frac{dV_s}{dt} = (1 - \nu)\, T - R$$

Drive Train Rotation,

$$2 \pi I \frac{dn}{dt} = Q_T - Q_R$$

Ship Resistance,

$$R \propto V^n_s$$

where:

γ is the virtual mass coefficient (entrained water) = 0.08

m is the ship mass

ν is the ship propeller interaction coefficient

T is the propeller thrust

R is the propeller resistance

I is the drive train inertia referred to propeller speed

n is the propeller speed in rps

Q_T is the turbine torque

Q_p is the propeller torque

V_s is the ship velocity

The equations for the propeller and turbine characteristics are given in previous sections.

Table 1. Ship Data

Displacement	27380 Tons
Ship Velocity, *V_s	21.5 Knots
Propeller Speed *	97.8 RPM
Ship Efficiency	0.74
Wake Coefficient *	0.23
Thrust Coefficient *	0.13
Propeller Efficiency *	0.63
Hull Efficiency *	1.12

Ship Resistance, R

$R \propto V_s^n$

n	V_s (Knots)
2	19
3	19.5
4	20.375
5	21.125

* At rated shaft HP of 24,000 and friction coefficient, C_A, of 0.0005 (Reference 3).

and the same number of blades (Reference 4). To judge the similarity between the A0177 and Wageningen propellers, the torque and thrust data were presented on the same graphs (Figures 4 and 5). The differences between the Wageningen and Taylor torque characteristics are shown to be relatively small and the thrust characteristics are an exact match. This favorable comparison established the credibility for using the Wageningen data for the A0177 propeller in the range where no Taylor data were available. This does not imply that the two propellers will behave exactly alike, but rather for the purposes of this study, use of the Wageningen data leads to realistic predictions of propeller and ship motion.

Cavitation was assumed to have no effect on the ship's maneuvering in the full ahead to full astern range. The available cavitation data in both the Taylor and Wageningen reports is shown to be at very high propeller speeds, and somewhat above the predicted full power propeller speed for this application. Therefore, for lack of specific cavitation data no representation for cavitation was included in this study.

The following equations were used in the propeller torque and thrust analysis:

$$Q = C_Q P D^3 \left(V_A^2 + n^2 D^2\right) \text{ , torque}$$

$$T = C_T P D^2 \left(V_A^2 + n^2 D^2\right) \text{ , thrust}$$

$$C_Q = f_1 (\sigma) \qquad \text{(Figure 4), torque coefficient}$$

$$C_T = f_2 (\sigma) \qquad \text{(Figure 5), thrust coefficient}$$

$$\sigma = 100\ nD/\sqrt{V_A^2 + n^2 D^2} \text{ , modified advance coefficient}$$

where:

V_A is advance velocity, ft/sec, speed of approach of water to propeller

$V_A = V_S\ (1-W)$

V_s = Velocity of the ship, ft/sec

W = Wake coefficient

n is propeller rotational speed in rps
D is propeller diameter in ft.
P is water density, lb/ft^3

Using the modified advance coefficient, σ , all four quadrants of torque and thrust data can be represented. The four quadrants correspond to positive and negative advance velocity and propeller rotational speed.

Ship

The model of the ship includes the data presented in the David Taylor Model Basin Report (Reference 3), as listed in Table 1, and the equations of ship's motion as presented below:

valve movement. No instances of falling below the low pressure specification were predicted. However, a very slow fuel response could allow the superheater outlet pressure to drop below the lower specified limit during a crash-back maneuver. The predicted pressure transients are illustrated in Figure 11.

Predictions of drum level and superheater outlet pressure during normal maneuvers are presented in Table 3. For the most severe upward maneuver the predicted swell just reaches the +4 inch drum level specification.

Table 3. Drum Level and Superheater Outlet Pressure Variations During Normal Maneuvers

Throttle * Command		Maximum Drum Level Change	Superheater Outlet Pressure (PSIG)	
From	To	Inches	Maximum	Minimum
0%	100%	+4.0	600	582
0%	-100%	+3.2	600	582
-100%	100%	-0.5 to +3.5	619	591
100%	0	-0.5 to +1.9	620	600
0	-100 **	+1.8	600	583

* Normal throttle valve movement rate of 1.67%/second used for this study.

** Normal throttle valve movement rate of 1.0%/second.

CONCLUSIONS

1. Crash-back reduces the predicted ahead reach by 14% to 31% for initial ship speeds of up to 100% when compared to the normal maneuvering throttle rate of 1.67% per second. An even larger ahead reach advantage is realized, if a slower maneuvering throttling rate is selected.

2. Predicted boiler drum level changes range from 10.1 inches to -2.3 inches for initial ship speeds of 0% to 100%, respectively. The ship is assumed to be operating in a steady-state condition initially. That is, the boiler steam flow corresponds to a specific ship speed (100% speed corresponds to 100% turbine flow or 24,000 HP). For crash-back maneuvers the maximum drum level swell increases with decreasing initial boiler steam flow.

3. Limiting the allowable drum level swell for a crash-back maneuver to the normal maneuvering drum level range of +4 inches, as specified, restricts the use of crash-back to initial boiler flows of at least 60% of rated flow which corresponds to about 82% of rated ship speed in the steady state.

4. The allowable crash-back maneuvering range could be extended by relaxing the ± 4 inch drum level specification. The potential gain in maneuvering capability as compared to allowable emergency drum

level limits may justify this change.

5. The Malfunction Proportional Control System (MPC) is available and may be used by the operator to limit drum level swells during a crash-back maneuver. If the MPC drum level limits are set above the ± 4 inch restraint envelope (e.g. lower set point 5" and upper set point 9"), nearly all of the predicted ahead reach advantage of crash-back is preserved. Not only could the crash-back mode be utilized in the entire ahead flow range, but the maximum drum level excursion would be limited to the value of the MPC upper limit set point.

6. In the initial portion of a crash-back maneuver with high initial boiler flows (i.e. high ship speeds in the steady state), the superheater outlet pressure exceeds the upper specification for a very short time (less than 2 seconds). This is a result of closing the ahead valve fully before opening the astern valve, causing a drop in the superheater flow and a rise in the superheater outlet pressure approaching the drum pressure. Any override controls on boiler operation or turbine throttle valve position sensing superheater outlet pressure should be designed to accept a pressure spike resulting from a crash-back maneuver without interferring with the boiler or turbine throttle valve operation.

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DIGITAL CONTROL OF GAS TURBINE-DRIVEN SHIPS
IN THE MANEUVERING REGION

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ABSTRACT

This paper presents an approach for the digital control of gas turbines used as prime movers on ships. Beginning with the derivation of the mathematical model of the system that comprises the ship, the gas turbine and the variable pitch propeller, this paper then moves on to the linearization of the model and the controller design. By Taylor series expansion, a number of linear models, each valid for a particular operating condition, are obtained. In these linearized state-space models, the two control inputs are throttle demand and propellor pitch rate and the three states are shaft rpm, free turbine torque and power lever angle. A set of control laws, one for each model, are derived and stored in the computer which then selects the correct controller for a given operating condition. The control laws are derived via the minimization of a performance function and implemented simply by an appropriate set of feedback and feedforward gains.

Simulation studies, using the DDH 280 Class destroyers of the Canadian Navy as an example, indicate that the present specifications are easily met. Namely, the required ship speed changes can be achieved with shaft rpm variations and maximum shaft torque well below the allowable levels.

I. INTRODUCTION

In the DDH 280 Class destroyers of the Canadian Armed Forces, propulsion is provided by a twin propeller system with a block diagram as shown in Figure 1. There are two gas turbines on each shaft, one for operation in the maneuvering region (low speed) and the other in the cruising region (high speed). In the maneuvering region and in the steady-state, each shaft rotates at a constant speed for all ship speeds so that only the pitch controls the ship speed. In the cruising region, the ship is at full pitch and the shaft speed is changed to give the desired ship speed. For the DDH 280, it is required, in the maneuvering region, that the shaft speed be maintained within ± 15 rpm of the nominal value of 80 rpm during the pitch changing transient and ± 5 rpm of the same value in steady-state. At no time is the shaft rotational speed to drop below 45 rpm else damage to the gas turbine can result due to improper lubrication. The control problem, for the power plant and propeller pitch in the maneuvering region, is: given a ship speed command, produce the pitch and throttle controls (see Figure 1) such that the pitch can reach its final value (corresponding to the ship speed command) as rapidly as possible without exceeding safe torques, etc., and with the shaft speed maintained within the specified tolerance.

The "Programmed Control" [1] technique is used presently on the DDH 280. In this method, a block diagram of which is given in Figure 2,

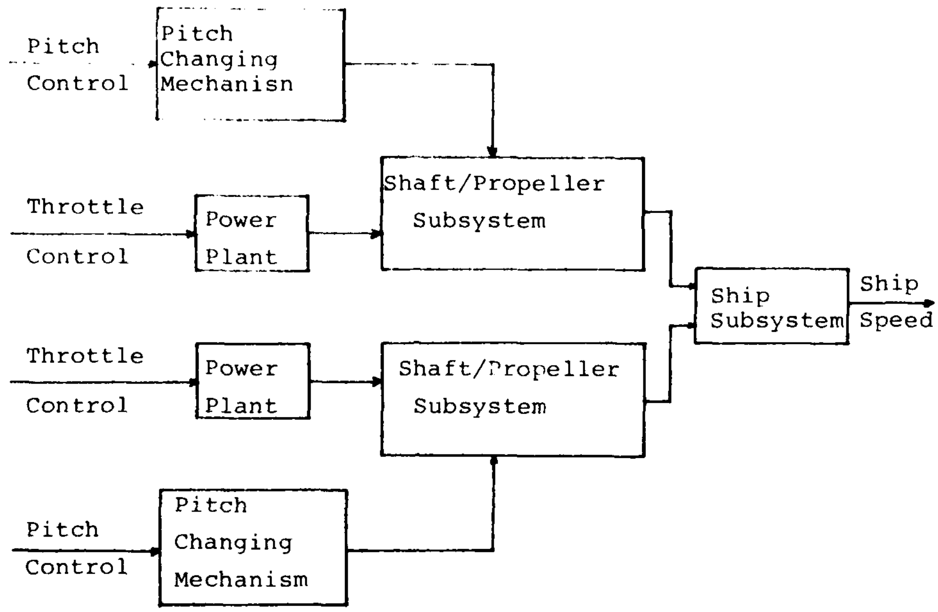

Figure 1. Propulsion Plant Block Diagram.

a ship speed order is translated into pitch, throttle and shaft speed orders through their respective schedules which relate the steady-state values of each order to ship speed. The logic block ensures synchronization between the pitch movement and throttle commands while the protection circuits limit the commands so that the propulsion plant will always operate in the "safe" region. This "Programmed Control" is implemented by analogue means. Although its performance is satisfactory, one should explore, for the new ships, the possibilities of digital control because of the latter's recognized benefits of versatility, compacity, reliability and its ability to perform complex functions and hence produce, potentially, a better control system.

The purpose of this paper is to present an alternative to the "Programmed Control" of marine gas turbine/controllable pitch propeller installations. The controller is digital and the design is obtained via state-space techniques. Section II describes the modelling of the propulsion plant and III, the development of the control strategy. The simulation results and conclusions are contained in Sections IV and V respectively.

II. MODEL DEVELOPMENT

II.1 Introduction

The propulsion plant block diagram is given in Figure 1. For modelling purposes, it is divided into the four subsystems of (i) pitch changing mechanism, (ii) gas turbine (the power plant), (iii) propeller and shaft subsystem and (iv) the ship. In each case, a nonlinear model is first developed followed by a Taylor series expansion of the model about an operating point. A linear model is then obtained by retaining only the first order terms of the expansion. Except for the gas

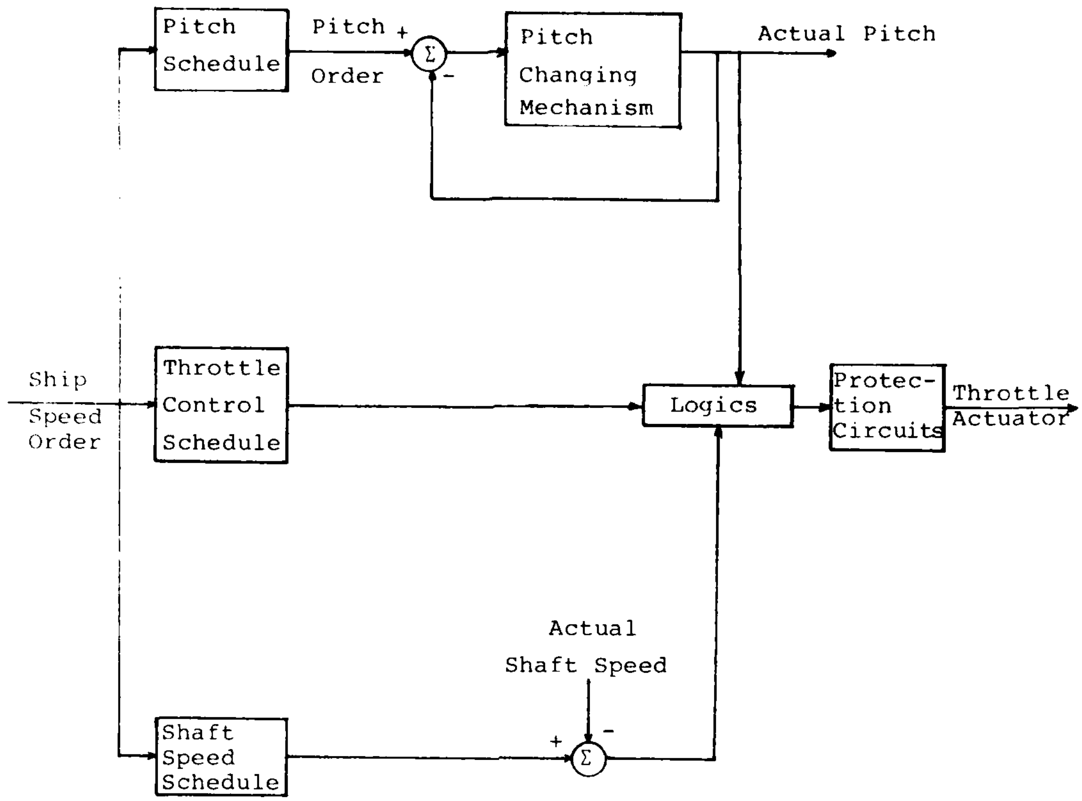

Figure 2. "Programmed Control" Structure.

turbine, where the Olympus TM3B is modelled instead of the Pratt and Whitney FT4A-2 actually installed on the DDH 280 because the latter's characteristics were not available, the data used and the equipment modelled are those of the DDH 280. A description of the complete modelling process is rather lengthy. This paper will only outline the important steps and the reader is referred to [2] for details.

II.2 Propeller Pitch Changing Mechanism

Figure 3 is a block diagram of the mechanism relating u_p, the pitch piston pump control (pitch control) and R_p, the pitch ratio.

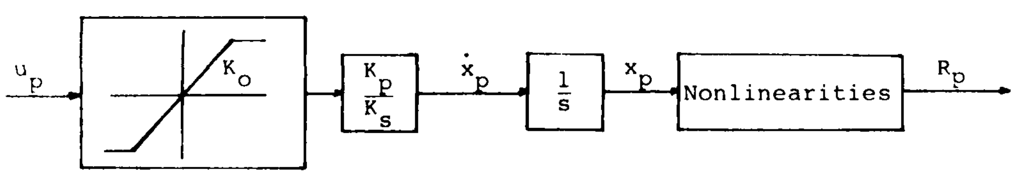

Figure 3. The Pitch Changing Mechanism.

The nonlinear block relating x_p, the pitch piston displacement, and R_p, is given by

$$R_p = \frac{x_p}{K_1(a-bx_p)} \tag{1}$$

where $K_o, K_p, K_1 \triangleq$ proportionality constants

$a \triangleq R_a(1 + \tan^2\theta_o)$

$R_a \triangleq$ moment arm

$b \triangleq \tan\theta_o$

$\theta_o \triangleq$ off-set angle

From Figure 3, for small changes about the nominal values, one can write, assuming no saturation,

$$\Delta(\dot{x}_p) = K_p \, \Delta(u_p) \tag{2}$$

where $\dot{x}_p = dx_p/dt$ and $\Delta(\cdot)$ denotes an incremental change of the bracketed quantity and for convenience, the brackets are dropped in the sequel. Similarly, linearization of (1) about a nominal pitch displacement x_{p_o} gives

$$\Delta R_p \approx \left.\frac{\partial R_p}{\partial x_p}\right|_o \Delta x_p \tag{3}$$

where the notation $|_o$ denotes the evaluation of the partial derivative at the nominal values. From (1), (2) and (3), we obtain the relationship

$$\Delta\dot{R}_p \approx \frac{K_p}{K_1(a-bx_{p_o})^2}\,\Delta u_p \tag{4}$$

which is a linear differential equation having the rate of change of the pitch ratio as a function of the pitch control input.

II.3 The Gas Turbine

As mentioned in Section II.1, the gas turbine modelled is the Olympus TM3B [3],[4] which has the nonlinear relationship

$$\dot{Q}_e = f(Q_e, F_e, N_e) \tag{5}$$

where $Q_e \triangleq$ free turbine torque

$f(\cdot) \triangleq$ a nonlinear function of the bracketed variables

$F_e \triangleq$ engine fuel flow

$N_e \triangleq$ free turbine rotational speed $= K_g N$

$K_g \triangleq$ gear ratio

and $N \triangleq$ shaft rotational speed.

The linearization of (5), valid for small perturbations around nominal values, is given by

$$\Delta \dot{Q}_e \approx \left.\frac{\partial f}{\partial Q_e}\right|_o \Delta Q_e + \left.\frac{\partial f}{\partial F_e}\right|_o \Delta F_e + \left.\frac{\partial f}{\partial N_e}\right|_o \Delta N_e \tag{6}$$

Letting $\left.\frac{\partial f}{\partial Q_e}\right|_o \triangleq$ EPOLE, then (6) becomes

$$\Delta \dot{Q}_e \approx -\text{EPOLE}(\Delta Q_e) + \text{EPOLE}\left[\left.\frac{\partial Q_e}{\partial F_e}\right|_o \Delta F_e + \left.\frac{\partial Q_e}{\partial N_e}\right|_o \Delta N_e\right] \tag{7}$$

Now F_e is a nonlinear function of PLA, the power lever angle and for small variations around the nominal values, and assuming no saturation, we have

$$\Delta F_e \approx \left.\frac{\partial F_e}{\partial \text{PLA}}\right|_o \Delta \text{PLA} \tag{8}$$

The PLA is positioned by the throttle control and the equation of motion is

$$\dot{\text{PLA}} = -\text{APOLE(PLA)} + \text{APOLE(Th)} \tag{9}$$

where 1/APOLE is the time constant of the PLA actuator and Th is the throttle control. Substituting (8) into (7) and together with the small signal version of (7) gives the linear differential equations of the turbine

$$\begin{aligned}\Delta \dot{Q}_e &\approx -\text{EPOLE}(\Delta Q_e) + \text{EPOLE}\left[\left.\frac{\partial Q_e}{\partial F_e}\right|_o \left.\frac{\partial F_e}{\partial \text{PLA}}\right|_o \Delta\text{PLA} + \left.\frac{\partial Q_e}{\partial N_e}\right|_o K_g N\right] \\ \Delta \dot{\text{PLA}} &\approx -\text{APOLE}(\Delta\text{PLA}) + \text{APOLE}(\Delta\text{Th})\end{aligned} \tag{10}$$

II.4 Propeller And Shaft Subsystem

The propeller thrust T, available to accelerate the ship, and the propeller load torque Q, are given by

$$T = T(R_p, N, V) \tag{11}$$

$$Q = Q(R_p, N, V) \tag{12}$$

where $R_p \triangleq$ pitch ratio

$N \triangleq$ shaft rotational speed

$V \triangleq$ ship speed

The relationships in (11) and (12) include the effects of wake, thrust deduction and relative rotative efficiency. Again, linearization of (1) and (2) about nominal values gives

$$\Delta T \approx \left.\frac{\partial T}{\partial R_p}\right|_o \Delta R_p + \left.\frac{\partial T}{\partial N}\right|_o \Delta N + \left.\frac{\partial T}{\partial V}\right|_o \Delta V \tag{13}$$

$$\Delta Q \approx \left.\frac{\partial Q}{\partial R_p}\right|_o \Delta R_p + \left.\frac{\partial Q}{\partial N}\right|_o \Delta N + \left.\frac{\partial Q}{\partial V}\right|_o \Delta V \tag{14}$$

The equation of motion of the shaft is

$$2\pi I\dot{N} = K_g Q_e - Q - Q_\ell \tag{15}$$

where
- $I \triangleq$ total system inertia as seen by shaft and is assumed constant
- $K_g \triangleq$ gear ratio
- $Q_e \triangleq$ free turbine torque
- $Q_\ell \triangleq$ shaft friction and gear train losses and is a function of N.

From (15), we have

$$\Delta\dot{N} \approx \frac{1}{2\pi I}\left[K_g \Delta Q_e - \Delta Q - \Delta Q_\ell\right] \tag{16}$$

Using $\Delta Q_\ell \approx \left.\frac{\partial Q_\ell}{\partial N}\right|_o \Delta N$ and substituting (14) into (16) give

$$\Delta\dot{N} \approx \frac{1}{2\pi I}\left[K_g \Delta Q_e - \left.\frac{\partial Q}{\partial R_p}\right|_o \Delta R_p - \left.\frac{\partial Q}{\partial V}\right|_o \Delta V - \left(\left.\frac{\partial Q}{\partial N}\right|_o + \left.\frac{\partial Q_\ell}{\partial N}\right|_o\right)\Delta N\right] \tag{17}$$

II.5 Ship Subsystem

Assuming constant mass, the motion of the ship can be described by the first order nonlinear differential equation

$$(1+\alpha)M\frac{dv}{dt} = T_1 + T_2 - R_T \tag{18}$$

where
- $V \triangleq$ ship speed
- $M \triangleq$ ship mass
- $\alpha \triangleq$ entrained water fraction } nonlinear functions of V
- $R_T \triangleq$ hydrodynamic drag } nonlinear functions of V
- $T_1, T_2 \triangleq$ propeller thrust of the first and second propeller, respectively.

By assuming α to be a constant and retaining only first order terms in a Taylor series expansion of (18), one obtains

$$\Delta\dot{V} \approx \frac{\Delta T_1 + \Delta T_2 - \left.\frac{\partial R_T}{\partial V}\right|_o \Delta V}{(1+\alpha)M} \tag{19}$$

II.6 The Linear Model

We choose the incremental changes in ship speed, ΔV, shaft speed, ΔN, free turbine torque, ΔQ_e, power lever angle, ΔPLA, and pitch piston displacement Δx_p as the states and pitch piston pump control, Δu_p, and the throttle control, ΔTh, as the inputs. We then represent the propulsion plant, for small deviations from nominal values, by the linear state-space equation

$$
\begin{bmatrix} \Delta\dot{V} \\ \Delta\dot{N}_1 \\ \Delta\dot{Q}_{e_1} \\ \Delta\dot{PLA}_1 \\ \Delta\dot{XP}_1 \\ \Delta\dot{N}_2 \\ \Delta\dot{Q}_{e_2} \\ \Delta\dot{PLA}_2 \\ \Delta\dot{XP}_2 \end{bmatrix}
=
\begin{bmatrix}
a_{11} & a_{12} & 0 & 0 & a_{15} & a_{16} & 0 & 0 & a_{19} \\
a_{21} & a_{22} & a_{23} & 0 & a_{25} & 0 & 0 & 0 & 0 \\
0 & a_{32} & a_{33} & a_{34} & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & a_{44} & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
a_{61} & 0 & 0 & 0 & 0 & a_{66} & a_{67} & 0 & a_{69} \\
0 & 0 & 0 & 0 & 0 & a_{76} & a_{77} & a_{78} & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & a_{88} & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0
\end{bmatrix}
\begin{bmatrix} \Delta V \\ \Delta N_1 \\ \Delta Q_{e_1} \\ \Delta PLA_1 \\ \Delta XP_1 \\ \Delta N_2 \\ \Delta Q_{e_2} \\ \Delta PLA_2 \\ \Delta XP_2 \end{bmatrix}
$$

$$
+ \begin{bmatrix}
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & b_{42} & 0 & 0 \\
b_{51} & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & b_{84} \\
0 & 0 & b_{93} & 0
\end{bmatrix}
\begin{bmatrix} \Delta u_{p_1} \\ \Delta Th_1 \\ \Delta u_{p_2} \\ \Delta Th_2 \end{bmatrix}
\tag{20}
$$

In the state and input vectors, the subscripts 1 and 2 serve to distinguish between the first and second propulsion unit[1]. The equations used to form (20) are (3),(4),(7),(13),(17) and (19) and the elements of the matrices can be identified from those equations as, for example,

$$a_{11} = \frac{1}{(1+\alpha)M}\left(\left.\frac{\partial T_1}{\partial V}\right|_o + \left.\frac{\partial T_2}{\partial V}\right|_o - \left.\frac{\partial R_T}{\partial V}\right|_o\right)$$

$$a_{12} = \frac{1}{(1+\alpha)M}\left(\left.\frac{\partial T_1}{\partial N_1}\right|_o\right)$$

$$a_{21} = -\frac{1}{2\pi I}\left(\left.\frac{\partial Q_1}{\partial N_1}\right|_o + \left.\frac{\partial Q_{e_1}}{\partial N_1}\right|_o\right)$$

1. A propulsion unit comprises a power plant, a shaft and propeller and a pitch-changing mechanism.

$a_{33} = -\text{EPOLE}$

$a_{44} = -\text{APOLE}$

Not all of the elements of the matrices are expressed here since it is a relatively simple matter to obtain the rest. It should be noted that (20) is valid only for incremental changes around the nominal values and the values of the elements of the matrices will change with operating conditions. Both the nonlinear and linear models of the propulsion plant were simulated on a digital computer. The nonlinear model was verified against results from [5]. The linear model was then verified against the nonlinear model for several sets of nominal values. The necessary partial derivatives were evaluated graphically. In both cases, we obtained good agreement between the different responses.

Since each propulsion unit in (20) receives separate commands, one can separate (20) into two fourth-order state-space equations. Their mutual coupling through ΔV can be accounted for by treating ΔV as a disturbance term in each equation. Thus, by partitioning (20) along the first and fifth rows and columns respectively, we obtain for the first propulsion unit

$$\begin{bmatrix} \Delta\dot{N}_1 \\ \Delta\dot{Q}_{e_1} \\ \Delta\dot{PLA}_1 \\ \Delta\dot{x}_{p_1} \end{bmatrix} = \begin{bmatrix} a_{22} & a_{23} & 0 & a_{25} \\ a_{32} & a_{33} & a_{34} & 0 \\ 0 & 0 & a_{44} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} \Delta N_1 \\ \Delta Q_{e_1} \\ \Delta PLA_1 \\ \Delta x_{p_1} \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & b_{42} \\ b_{51} & 0 \end{bmatrix} \begin{bmatrix} \Delta u_{p_1} \\ \Delta T_{h_1} \end{bmatrix} + \begin{bmatrix} a_{21} \\ 0 \\ 0 \\ 0 \end{bmatrix} \Delta V \qquad (21)$$

and a similar equation for the second propulsion unit.

Past results [6] have shown that the shaft line[1] can be controlled independently of the propeller pitch. It is easy to decouple the shaft line system from the pitch system in (21) since the bottom row of the system matrix has all zeros. If we eliminate this row then and include the Δx_{p_1} term in the disturbance ΔZ, we have

$$\begin{bmatrix} \Delta\dot{N}_1 \\ \Delta\dot{Q}_{e_1} \\ \Delta\dot{PLA}_1 \end{bmatrix} = \begin{bmatrix} a_{22} & a_{23} & 0 \\ a_{32} & a_{33} & a_{34} \\ 0 & 0 & a_{44} \end{bmatrix} \begin{bmatrix} \Delta N_1 \\ \Delta Q_{e_1} \\ \Delta PLA_1 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ b_{42} \end{bmatrix} \Delta Th_1 + \begin{bmatrix} -\frac{1}{2\pi I} \\ 0 \\ 0 \end{bmatrix} \Delta Z \qquad (22)$$

for the shaft line and

$$\Delta\dot{x}_{p_1} = b_{51} \Delta u_{p_1} \qquad (23)$$

[1] A shaft line comprises a power plant, a shaft and a propeller.

for the pitch control where

$$a_{1} = -\frac{1}{2\pi I}\left.\frac{\partial Q_{e1}}{\partial N_1}\right|_o$$

and

$$\Delta Z = \left.\frac{\partial Q_1}{\partial N_1}\right|_o \Delta N_1 + \left.\frac{\partial Q_1}{\partial V}\right|_o \Delta V + \left.\frac{\partial Q_1}{\partial x_{p1}}\right|_o \Delta x_{p1} \qquad (24)$$

The disturbance ΔZ is purposely manipulated into the form of (24) in order that it represents the incremental propeller load torque. Equation (24) is equivalent to (14) if we let $\Delta R_p = \left.\frac{\partial R_p}{\partial x_{p1}}\right|_o \Delta x_{p1}$ in (14).

To recapitulate, for the control problem, (20) is first separated into two fourth order systems and then each system is further decomposed into a third order shaft line equation and a first order pitch control equation. A disturbance term, representing the propeller load torque, acts on the shaft rotational speed directly and its role is to take care of the coupling effects between the decomposed systems.

III. THE CONTROLLER

III.1 Introduction

The procedures in Section II.6 that separate a propulsion unit into a shaft line model and a pitch control model are necessary to simplify the controller. The pitch system, being of first order, is relatively easy to control. The pitch is simply moved until it reaches the desired position. The shaft line controller is designed via the linear quadratic technique [7]. A nominal, or desired, trajectory is first computed. In this trajectory, the shaft speed state is constant at 80 rpm while the other two states (free engine torque and power lever angle) and the control (throttle) vary such that the 80 rpm is maintained. During operation, whenever the actual trajectory deviates from the nominal, a control correction is added to the nominal control to minimize the deviations.

III.2 The Nominal Trajectory

Consider the minimization of the cost function

$$J = \frac{1}{2}(x_{k+1} - \hat{x}_{k+1})^T W(x_{k+1} - \hat{x}_{k+1}) \qquad (25)$$

by choosing the input u_k in the nonlinear, discrete state-space equation

$$x_{k+1} = f(x_k, u_k) \qquad (26)$$

where x_{k+1} and $\hat{x}_{k+1}$ are the actual and desired state vectors, respectively, at time (k+1)T, T the sampling interval, W a positive semi-definite weighting matrix whose elements reflect the relative costs for deviating from the nominal value of each state variable and f represents the nonlinear function. By expanding (26) in a Taylor series around x_k^o, i.e. let $x_{k+1} = x_k^o + \delta x_{k+1}$ and $u_k = u_{k+1} + \delta u_k$, we have,

$$x_{k+1} - x_k^o = \delta x_{k+1} = \left.\frac{\partial f}{\partial x_k}\right|_o \delta x_k + \left.\frac{\partial f}{\partial u_k}\right|_o \delta u_k \qquad (27)$$

where if we define

$$A_k = \frac{\partial f}{\partial x_k}\bigg|_o , \quad B_k = \frac{\partial f}{\partial u_k}\bigg|_o \tag{28}$$

then

$$\delta x_{k+1} = A_k \delta x_k + B_k \delta u_k , \quad \delta x_o = 0 \tag{29}$$

Then substituting (27) into (25) for x_{k+1} and carrying out the minimization by setting $\frac{\partial J}{\partial \delta u_k} = 0$ gives

$$\delta u_k = (B_k^T Q B_k)^{-1} B_K^T W(\hat{x}_{k+1} - x_k^\circ - A_k \delta x_k) \tag{30}$$

so that

$$u_k^\circ = u_{k-1}^\circ + \delta u_k \tag{31}$$

and

$$x_{k+1}^\circ = x_k^\circ + A_k \delta x_k + B_k \delta u_k \tag{32}$$

The computation of u_k in (30) requires A_k and B_k which, because of the need for accuracy, have to be computed at each x_k°. To reduce storage and real time requirement, (22) was further reduced to the second order equation

$$\begin{bmatrix} \delta\dot{Q}_e \\ \delta\dot{PLA} \end{bmatrix} = \begin{bmatrix} a_{33} & a_{34} \\ 0 & a_{44} \end{bmatrix} \begin{bmatrix} \delta Q_e \\ \delta PLA \end{bmatrix} + \begin{bmatrix} a_{32} \\ 0 \end{bmatrix} \delta N + \begin{bmatrix} 0 \\ b_{42} \end{bmatrix} \delta Th \tag{33}$$

where δN now becomes a disturbance term and the subscript 1 is dropped for convenience. The discretization of (33) then gives

$$\begin{bmatrix} \delta Q_{e_{k+1}} \\ \delta PLA_{k+1} \end{bmatrix} = A_k \begin{bmatrix} \delta Q_{e_k} \\ \delta PLA_k \end{bmatrix} + B_k \delta Th_k \tag{34}$$

where

$$A_k = \exp \begin{bmatrix} a_{33} & a_{34} \\ 0 & a_{44} \end{bmatrix} T$$

$$B_k = \int_0^T \left\{ \exp \begin{bmatrix} a_{33} & a_{34} \\ 0 & a_{44} \end{bmatrix} \tau \right\} \begin{bmatrix} 0 \\ b_{42} \end{bmatrix} d\tau \tag{35}$$

and T is the sampling interval. In going from (33) to (34), which is the model for computing the nominal trajectory, δN is taken to be zero since the shaft speed is to be constant at 80 rpm.

To satisfy the constant shaft speed requirement, there must be no net torque available to accelerate the shaft so that Q_e must equal Q/K_g where Q is the propeller load torque and K_g the gear ratio. This latter condition dictates that the first desired state in x_{k+1}, $\hat{x}_{1k+1}$, be equal to Q_{k+1}/K_g. With $W = \begin{bmatrix} 1 & 0 \\ 0 & 0 \end{bmatrix}$ in (25), i.e., no weight on the PLA state, δTh_k will ensure that

$$Q_{e_{k+1}}^{\circ} = Q_{k+1}/K_q \tag{36}$$

and it does not matter what x_{k+1} is, the resultant PLA°_{k+1} is that value compatible with (36). In fact, x_{k+1} is always taken to be zero for convenience. The next task is the prediction of Q_{k+1} at time kT. Recall from (12) that $Q = Q(R_p, N, V)$ and this nonlinear relationship is stored in the computer. It will be seen in Section III.4 that at kT, the value of R_p at (k+1)T can be predicted. By assuming that N and V remain constant from kT to (k+1)T, Q_{k+1} can then be computed by evaluating (12) using the values of R_p, N and V at (k+1)T.

To obtain A_k and B_k in (35), several sets of values, each set corresponding to a particular x°_k, of each of the elements in (33), i.e., a_{33}, a_{34}, etc., are stored. Then linear interpolation is used to produce the other values of a_{33}, a_{34}, etc., if the actual x°_k is not among one of those stored. Having obtained the values for a_{33}, a_{34}, etc., the matrices A_k and B_k can be computed from (35). This interpolation method helps to reduce the large storage requirement which otherwise would have been needed because large linearization errors will arise from a coarse grid of models. However, we discovered from simulation studies that small linearization errors in one interval will propagate to subsequent intervals causing a gradual divergence of the nominal trajectory from the ideal. To overcome this problem, a periodic reinitialization using a nonlinear model is performed. This procedure entails calculating a nominal trajectory from a stored nonlinear steady-state model of the shaft line. The procedure is accurate if it occurs at T_c seconds from the instant the pitch has stopped changing. The interval T_c is equal to the time which the power plant transient takes to decay after a propeller pitch change.

In summary then, the nominal trajectory consists of a constant shaft speed state, a throttle control and two other states of Q_e and PLA which take on values, as pitch position and ship speed vary, consistent with the constant shaft speed requirement. The computation at each time kT involves predicting the propeller load torque for the next interval, obtaining the linearized parameters necessary to calculate A_k and B_k in (35), and finally the generation of the nominal control and states through equations (30) to (32).

III.3 The Control Correction

Let the nonlinear, discrete shaft line equation be

$$x_{k+1} = F(x_k, u_k) \tag{37}$$

so that linearization of (37) around the nominal values give

$$\Delta x_{k+1} = \left.\frac{\partial F}{\partial x_k}\right|_o \Delta x_k + \left.\frac{\partial F}{\partial u_k}\right|_o \Delta u_k$$

$$= \phi_k \Delta x_k + H_k \Delta u_k \tag{38}$$

where

$$\phi_k = \left.\frac{\partial F}{\partial x_k}\right|_o , \quad H_k = \left.\frac{\partial F}{\partial u_k}\right|_o$$

$$x_{k+1} = x^{\circ}_{k+1} + \Delta x_{k+1}, \quad x_k = x^{\circ}_k + \Delta x_k$$

and $\quad u_k = u^{\circ}_k + \Delta u_k \qquad (39)$

Suppose we apply the nominal control u_k^o at kT but because of modelling errors or disturbances, the actual state vector at (k+1)T is x_{k+1} and not x_{k+1}^o. The deviations from the nominal values, if small, can be adequately modelled by (38) so that the control connection Δu_k can be obtained by minimizing

$$J_1 = \frac{1}{2} \Delta x_{k+1}^T M \Delta x_{k+1} + \frac{1}{2} \Delta u_k^T R \Delta u_k \tag{40}$$

subject to the constraint (38). The weighting matrix M is positive semi-definite while R is positive definite.

Substituting (38) into (40) and setting $\frac{\partial J_1}{\partial \Delta u_k} = 0$ yields the optimal control correction

$$\Delta u_k^* = -(R + H_k^T M H_k)^{-1} H_k^T M \emptyset_k \Delta x_k \tag{41}$$

In implementing (41), Δx_k is taken to be the difference between the measured and nominal state vectors and $\emptyset_k$ and H_k are obtained from discretization of (22) following the same steps that give (34) and (35) from (33). For the control correction calculation, ΔZ in (22) is assumed to be zero since it is not possible to measure the disturbances.

Acceptable performance of the control correction relies on a suitable choice of the weighting matrix M. This is because the relative magnitudes of the elements in M affect the amount of correction applied to each state. Normally, M is a diagonal matrix so that weights are assigned only to the individual states and not to products of states. However, we have found from simulation studies that a diagonal M did not give satisfactory control corrections. A fuller, non-diagonal M must therefore be considered. It can be a long trial and error process to pick the correct weights for a non-diagonal M. To reduce the number of computer runs, an unconventional but novel approach was used. The procedure considers the minimization of

$$J_2 = \frac{1}{2} \Delta x_N^T P \Delta x_N + \frac{1}{2} \sum_{j=1}^{N-1} (\Delta x_j^T P \Delta x_j + \Delta u_j^T R \Delta u_j) \tag{42}$$

subject to the constraint (38). Again P is a positive semi-definite weight matrix. The difference between using J_1 and J_2 is that the former minimizes the deviations in the next interval while J_2 minimizes the deviations over the next N intervals. Thus J_2 possesses a "look-ahead" property. The control correction that minimizes J_2 has the form

$$\Delta u_j^* = -(R + H_j^T K_{j+1} H_j)^{-1} H_j^T K_{j+1} \emptyset_j \Delta x_j \tag{43}$$

where K_{j+1} is the solution of the discrete matrix Riccati equation [8]

$$K_j = \emptyset_j^T K_{j+1} (I + H_j R^{-1} H_j^T K_{j+1})^{-1} \emptyset_j + P$$
$$K_N = P \tag{44}$$

Thus the computation of (43) requires advance knowledge of the nominal trajectory for N periods in the future which, clearly, is not possible. However, it is well-known that for large N, and constant and controllable $\emptyset_j$ and H_j, $\lim_{N \to \infty} K_j = K$, a constant. Noting that M and K_{j+1} play the same role in (41) and (43) respectively, it was decided to first

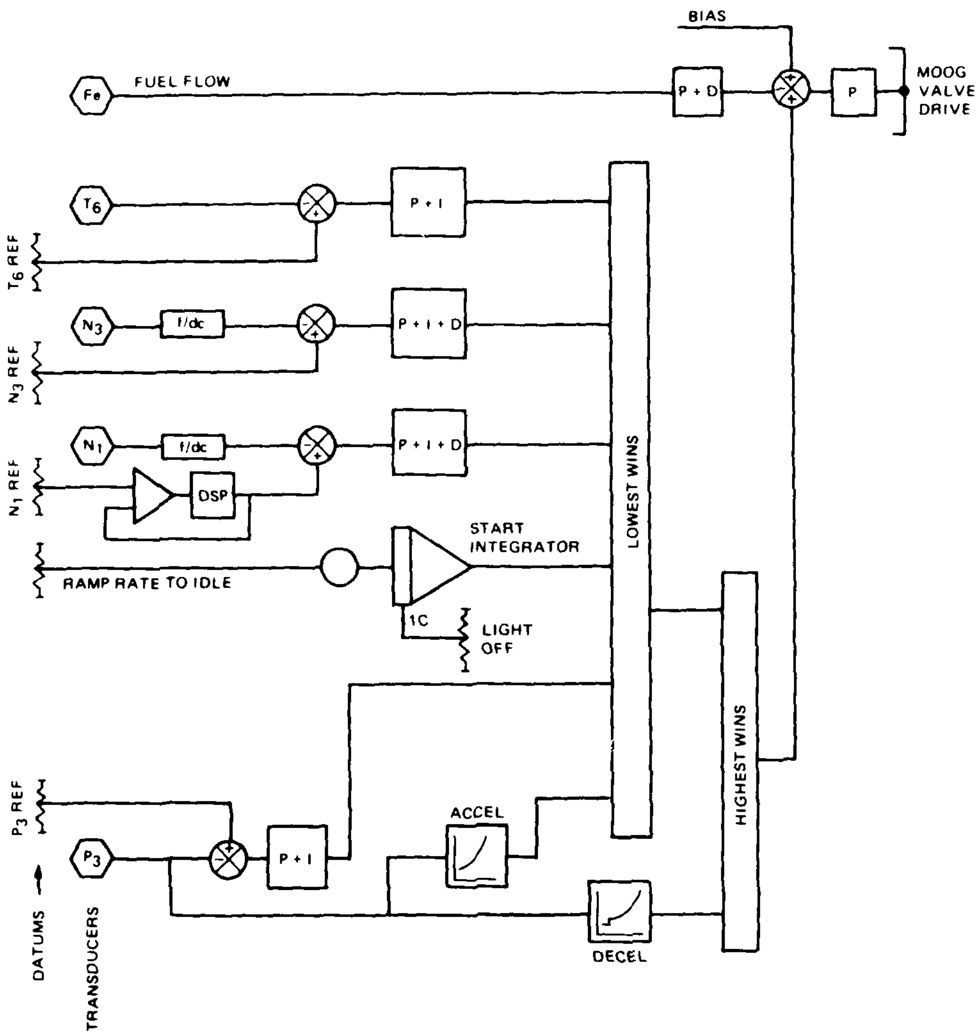

FIGURE 3 PROGRAMMED CONTROL FUNCTIONS

- Governor Optimisation. If we consider the curves of Figure 4 it is seen that the basic engine inertia time constant varies by a factor in excess of 10:1 and the slope of the steady state relationship between speed and fuel flow also varies by 5:1 over the engine operating range. Fixed settings for the gains of the 3-term control cannot provide an optimum governor over the whole speed datum range. The programmable nature of the control made it easy for the proportional and derivative gains to be related to engine running conditions as a function of LP compressor speed.

Sequencer

The programmable sequence system is a one bit machine which operates switch outputs according to a sequence flow programme held in a R.O.M. store. The sequence is established according to the states of switch inputs and their logical interconnection and this is incorporated in the sequence programme in the form of logic and timer functions.

The programme may be held in up to 2048 lines of programme, each line consisting of a 16 bit word divided into an Instruction code zone (4-Bits), address zone (11 Bits) and a parity zone (1 Bit).

The instruction code set includes logic (AND, OR and Exclusive OR), read (LOAD and LOAD INVERSE), write (STORE accumulator, SET to '1', SET to '0'), jump (JUMP IF '1', JUMP IF '0' and JUMP UNCONDITIONAL) and Timer (TIMER CLEAR and TIMER).

Address zones are for inputs, outputs and scratchpads, the latter containing 512, one bit storage locations for immediate results. The address zone is used in Timer instructions to define the clock rate (in multiples of 40 mSecs and final count value which enables timers from 80 mSecs to 2½ hours to be programmed. Up to 64 timers can be provided.

General

Both the P.A.C. and Sequencer use a high level software language readily understood by engineers with a minimum of training. Unlike microprocessor based systems they do not require any software support packages such as compilers and interpreters; programmes are entered in decimal form via a keyboard system which provides automatic conversion to binary for storage in P.R.O.M.S.

Such is the computing power of the P.A.C. for example that the complete control programme for the trials installation described occupied less than 200 lines of programme.

4. P.A.C. CONTROL DESCRIPTION

Figure 3 is a block diagram of the control functions programmed on the P.A.C. They are typical of functions which are necessary for industrial and marine controls.

- T6 control was provided as a topping governor with proportional and integral error control.

- Power turbine speed was governed as a topping function with a 3 term error controller.

- LP compressor speed was programmed as a full range 3-term control and the datum of the controller was used as the throttle input.

  This loop had the following interesting features which, on previous purely analogue controls, were very difficult to implement.

  - <u>Datum ramp rates</u> - Marine applications generally require controlled speed raising and lowering rates. This was easily achieved with the P.A.C. by using the digital set point (DSP) module. This is an Up/Down counter, the rate of which can be controlled from software.

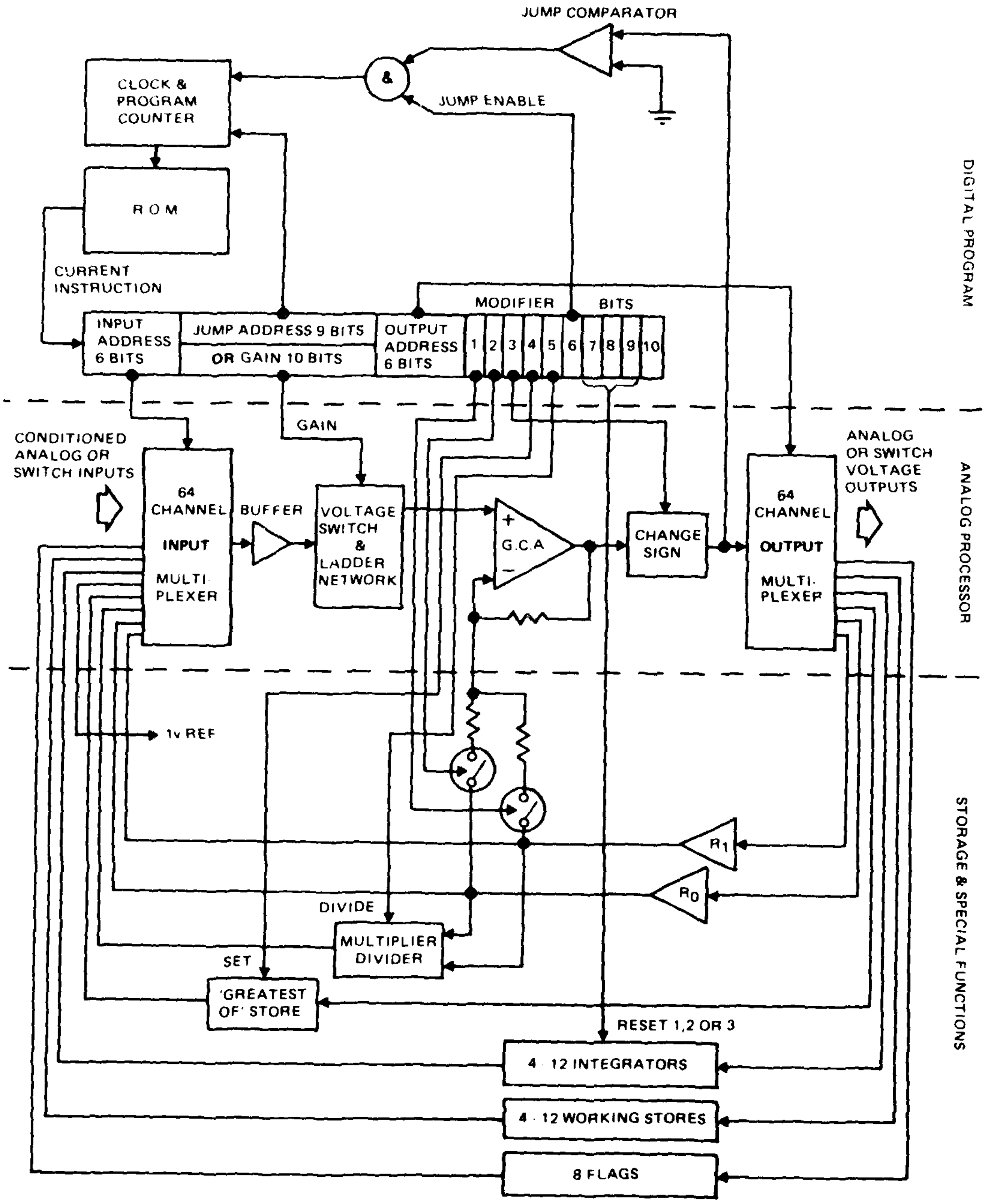

FIGURE 2 BLOCK DIAGRAM OF P.A.C. STRUCTURE

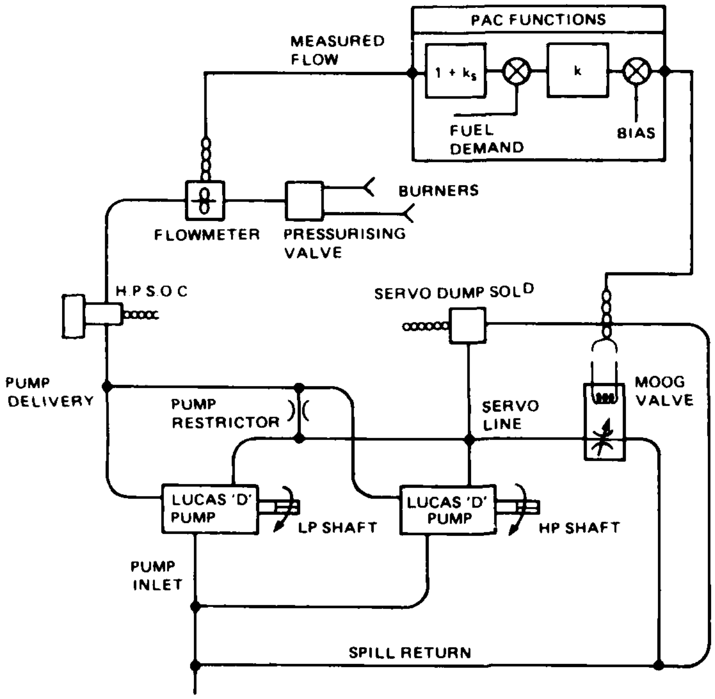

FIGURE 1 HYDRAULIC INTERFACE FUEL SYSTEM

3. REVIEW OF THE P.A.C. AND SEQUENCER OPERATION

P.A.C.

The Programmable Analogue Control is an analogue machine, ie. inputs and outputs are dealt with in analogue form, wherein control algorithms are formed via the use of a programme held in a R.O.M. store.

All lines of programme are available, each line consisting of a 32 bit word divided into zones for input, gain, output and modifiers.

6 bit input and output zones are decoded as each line is accessed to select an input and output analogue multiplexer. The input so selected is simultaneously gain modified by the contents of a 10 bit gain zone providing gain variation from 0 to 10.23 in increments of 0.01.

The analogue value (input x gain) may be further modified at each programme line by the action of modifiers selected by programming a '1' in the appropriate bit positions of a 10 bit modifier zone in this case (-RO), (-ve). These, if selected, will subtract the analogue values held in working stores RO and R1 and negate the final result to give a potential computation at each line of $(A.G. \pm B \pm C) \pm$.

A further modifier will convert the line to a 'jump' instructions, which will cause a jump to the line number specified in binary form in the gain zone if the input called for is positive. Others call for a divide (instead of multiply when the analogue multiplier is specified as an input), reset 'greatest of store', reset integrators etc. A diagram showing the basic structure of the P.A.C. is shown in Figure 2.

Objects

The objectives of the trials were:-

- to assess the ability of the equipment to perform control tasks in a Shore Trials environment

- to become familiar with programming techniques

- to determine advantages of the programmable equipment

- to establish the usefulness of the equipment as a development tool

As a means of achieving these objectives, the equipment was programmed to control an Olympus TM3B module. The interface hydraulic equipment was not considered to be representative of a production marine system. It was chosen to interface easily with the standard TM3B equipment and was a means of testing the electronic content of the fuel management system. However, as will be described later, the hydraulic equipment was very simple and behaved well during the tests. It is being considered as a basis for future systems.

The controller was programmed to carry out typical control functions. Attempts were not made to carry out programming for a particular marine specification for this was not the intention.

2. DESCRIPTION OF EQUIPMENT

Shore Trials Facility

The Shore Trials facility housed a TM3B module (gas generator and power turbine). The intake and uptake are typical of a ship installation. Generated power is absorbed by two water brake dynamometers in series. Load absorbed is controlled by the adjustment of valves which control the mass flow of water entering and leaving the carcase of the dynamometer. These valves are manually controlled in order to run the power turbine at propellor law conditions.

Gas Generator

The Olympus engine which is aero derived is fitted with two engine driven swash plate fuel pumps and the flow is delivered to the duplex burner system via a pressurising valve.

Hydraulic Interface Hardware

Figure 1 shows a schematic of the installed hydraulic interface fuel system. The electronic control interfaces with the engine pumps via a modified electrohydraulic servo valve. The delivered flow is measured by a rotating vane flow meter.

Thus the hydraulic equipment content of the system was very small and easily installed in the marine module. However, it is a fail safe system and not fail set with manual reversion, as would normally be required by a system intended for ship operation.

The standard marine system shut down facilities were retained to ensure full protection of the engine and test bed using the fuel shut off cock and a high speed shut down by means of a servo solenoid dump valve. Trips based on turbine overspeed and overtemperature were fully operative.

EVALUATION OF A PROGRAMMABLE CONTROLLER AND SEQUENCER
IN A SHORE TRIALS ENVIRONMENT

by Norman P. Lines - Rolls Royce Limited, Industrial and Marine Division.
and David E. Mann - Ultra Electronic Controls Limited.

SUMMARY

The evaluation of a UEL programmable engine controller and programmable sequencer was carried out on a MOD(PE) shore trials facility at Rolls Royce Industrial and Marine Division headquarters, Ansty.

The UEL equipment was on loan to Rolls Royce as part of a Department of Industry 'Pre Production Order' scheme. The evaluation became the initial phase of an exercise to determine the most suitable control equipment for Rolls Royce Industrial and Marine engines in the 1980's. The paper describes the form of the evaluation from simulation trials through to engine testing. A review of the controller and sequencer philosophy is made and special programming techniques are high lighted.

The suitability of the equipment for industrial and marine applications is considered and the users impressions on handling are discussed.

Finally reliability and accuracy aspects are reviewed.

1. INTRODUCTION

Background

Rolls Royce use control equipment which utilise a variety of technologies; hydromechanical, pneumatic and electronic analogue. None of these are ideal for reasons of reliability, accuracy and the complexity of incorporating modifications to systems. The latter reason is particularly important because the control system can never be fully defined for all types of application. Changes to combustion hardware to improve efficiency may require modifications to the control system which are not easily incorporated by a change in gain or a set point. Programmable equipment can assist greatly in that area. The hydromechanical content of a system can be kept general purpose and particular requirements accommodated in software.

It was with some of these factors in mind that in 1976 Rolls Royce embarked on an exercise to determine the most suitable control equipment for industrial and marine gas turbines to be commissioned in the 1980's.

The initial phase of this exercise was to look at UEL programmable controls. This was achieved by tests on equipment initially loaned to Rolls Royce by the Department of Industry under a 'Pre Production Order' scheme. This is a system whereby potential customers for equipment can conduct evaluations without the initial capital expenditure. At the end of the loan period the equipment can be purchased if desired.

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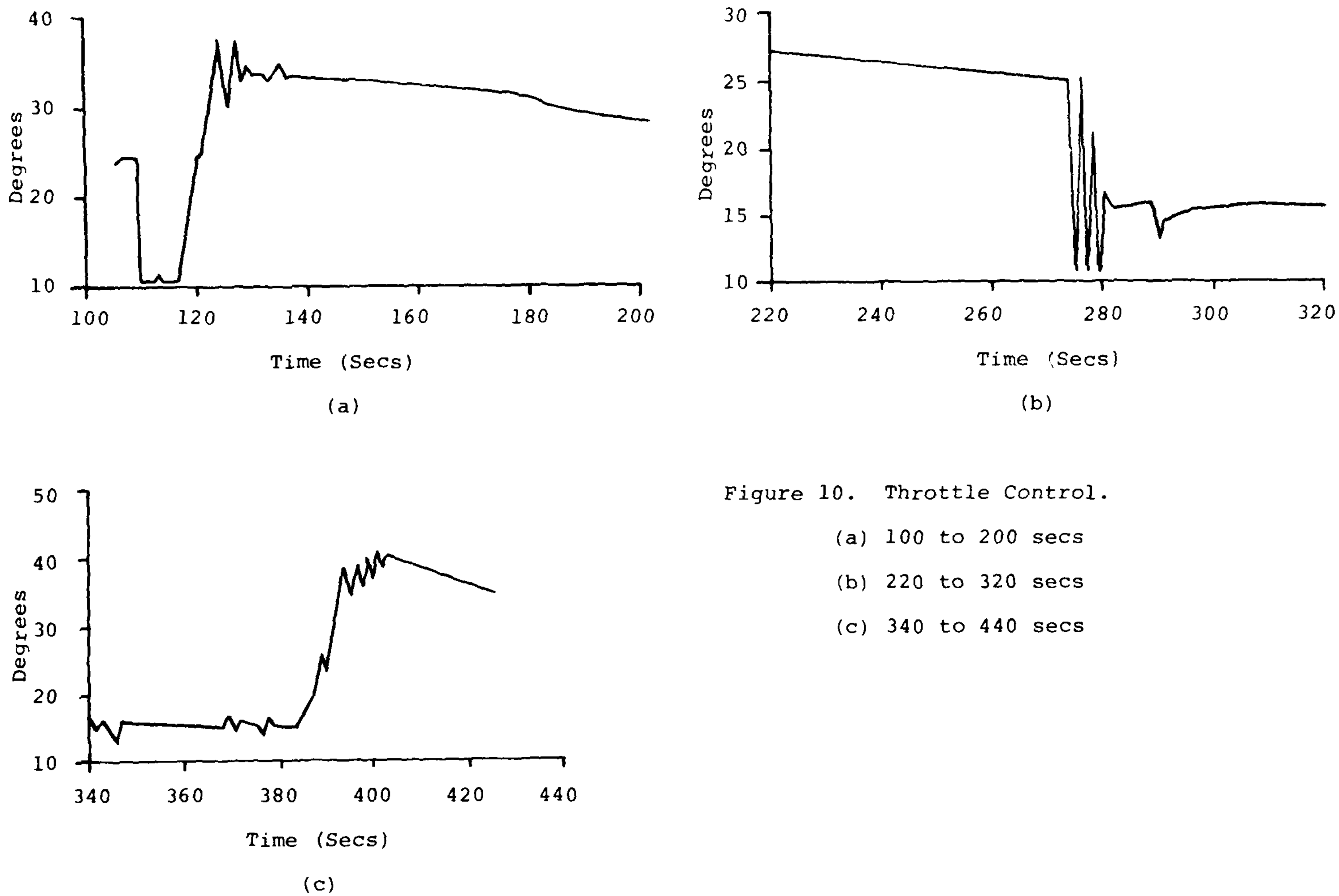
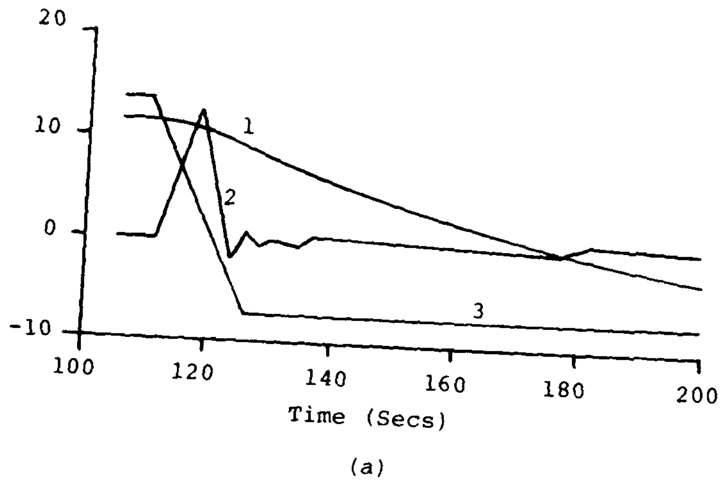


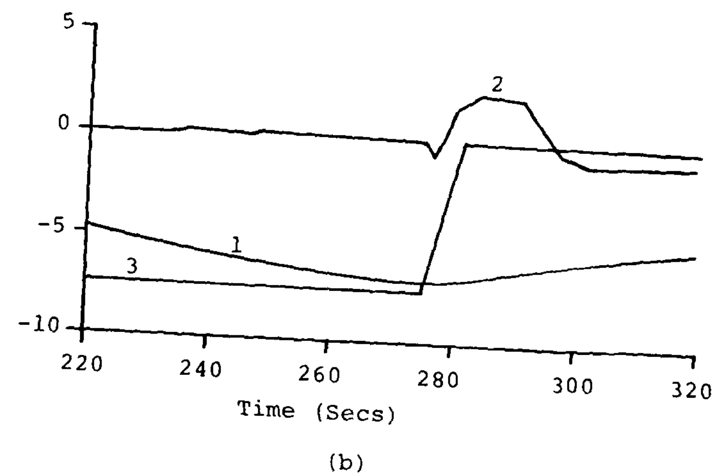
Figure 10. Throttle Control.

(a) 100 to 200 secs

(b) 220 to 320 secs

(c) 340 to 440 secs





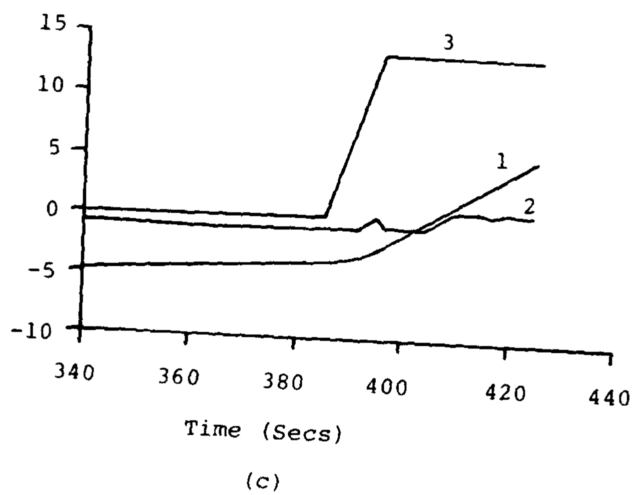


Figure 9. Plant Responses.

(a) 100 to 200 secs

(b) 220 to 320 secs

(c) 340 to 440 secs

1 - Ship Speed, knots

2 - rpm Errors

3 - Pitch Ratio x 10

Table 2 lists the noise added and their magnitudes. Figures (9) and (10) display the plant and throttle control responses, respectively. Responses in the first hundred seconds are not shown as they are similar to those in Figures (6) and (7). In Figure (9), an overspeed condition in the shaft occurs after the input order changes from

Variable	Noise Standard Deviation
Shaft Speed	0.4 rpm
Ship Speed	0.1 knot
Free Turbine Torque	1.5%
Power Lever Angle	0.15 degree
Piston Displacement	0.1 inch

Table 2. Noise Characteristics.

full ahead to full astern pitch. This condition is due to the propeller load torque going negative. The shaft speed error is still within the specified ± 15 rpm limit and the overall response throughout the command sequence is satisfactory.

V. CONCLUSIONS

The aim of this paper is to present a digital control scheme for gas turbine driven ships. The design procedure begins with the modeling process and the subsequent linearizations that yield a linear state-space e quation of the plant. A nominal trajectory, in which the shaft speed is constant at 80 rpm, serves as a reference trajectory wihch the plant must follow. Deviations from the nominal are sensed by a control law which also provides the control correction necessary to minimize the deviations. Simulation results, under different ship speed commands, have demonstrated the effectiveness of the new control scheme. The throttle control was smooth and the shaft speed did not vary more than ± 15 rpm from the nominal value of 80.

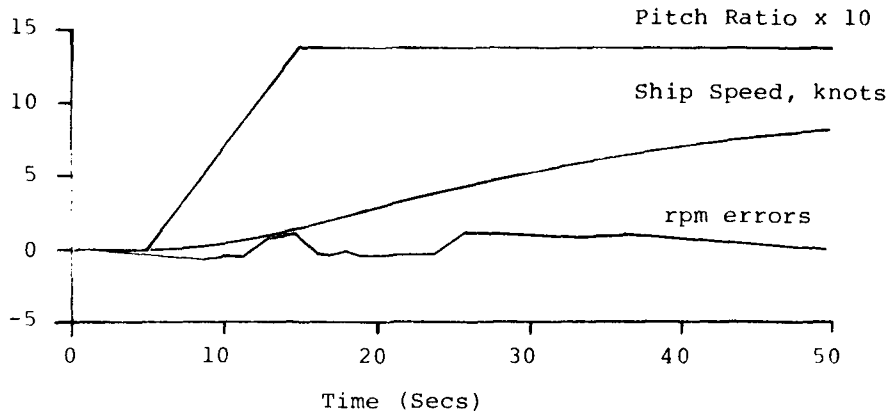


Figure 6. Plant Response, Weights (ii).

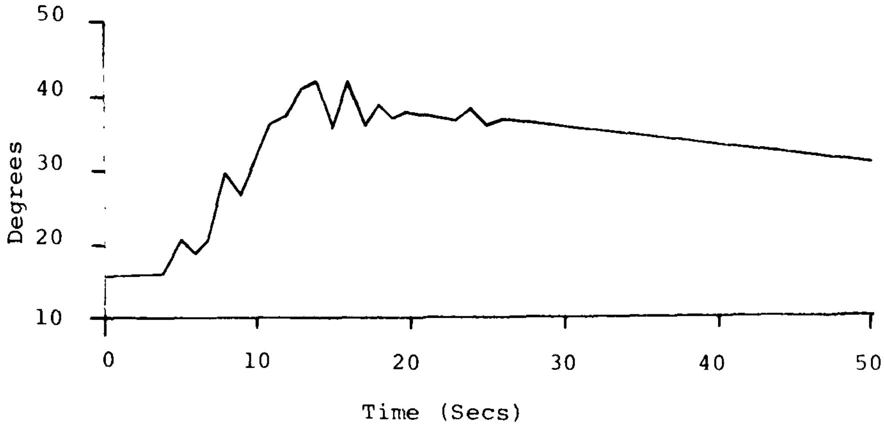


Figure 7. Throttle Control, Weights (ii).

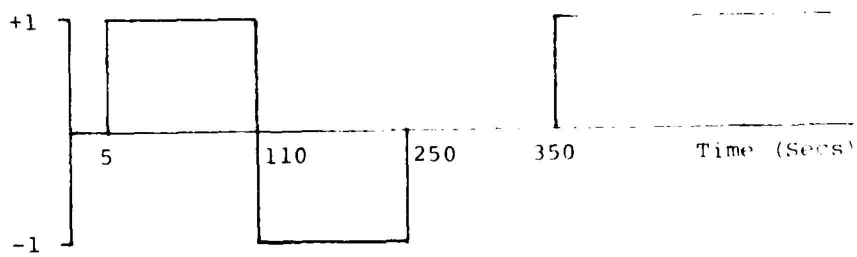


Figure 8. Input Command.

speed error are reduced.

The second experiment examines the plant behavior under a multiple command sequence. Figure (8) is the specific sequence applied and +1 and -1 on the vertical scale corresponds to full ahead and stern pitch respectively. The weights (ii) in Table 1 are used. Measurement noises of zero mean and Gaussian distribution are also added.

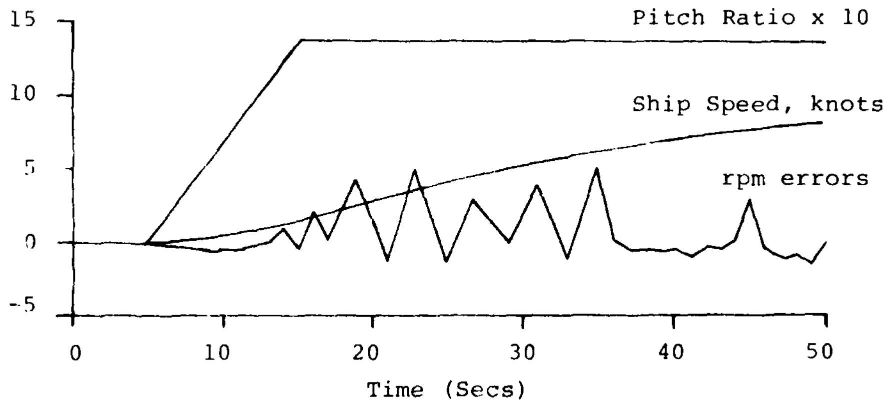


Figure 4. Plant Response, Weights (i).

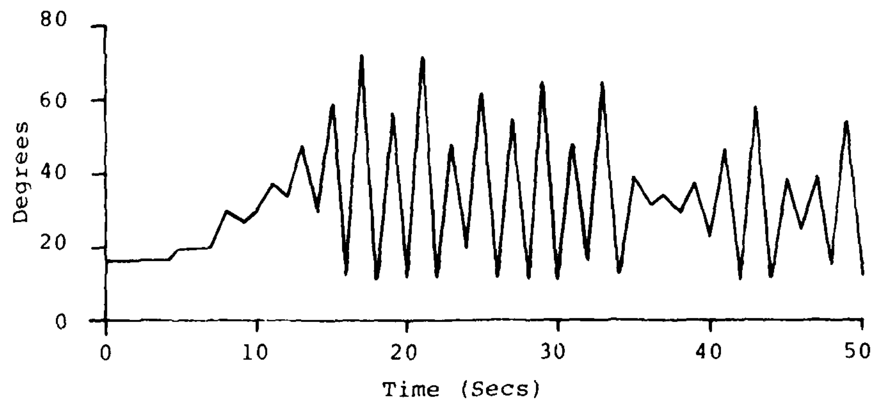


Figure 5. Throttle Control, Weights (i).

$$u_{p_k} = \text{dez}\left\{\frac{1}{K_p}\ \text{sat}\left(\frac{\hat{x}_{p_k} - x_{p_k}}{T}\right)\right\} \tag{49}$$

where

$$\text{dez}(\ ) \triangleq \begin{cases} (\ ), & \text{if abs}(\ ) > Dz \\ 0, & \text{if abs}(\ ) \leq Dz \end{cases}$$

and Dz is the width of the dead zone. In a noisy environment, Dz can be chosen based on the statistics of the noise to guarantee that the probability of falsely applying a control due to noise is less than some figure chosen by the designer.

It should be noted that while the control in (48) assumes that the piston displacement can move at any rate less than $Rate_M$, only two rates are available in the DDH 280. However, we consider the former requirement to be also practical.

In Section III.2, prediction of the pitch ratio R_p is necessary for the computation of the nominal propeller load torque. It is easy to obtain $x_{p_{k+1}}$ once u_{p_k} is known by using (46). Substitution of $x_{p_{k+1}}$ into (1) then gives $R_{p_{k+1}}$.

IV. SIMULATION RESULTS

The main simulation package consists of programs that simulate the whole nonlinear propulsion plant (i.e., both propulsion units), routines to generate the nominal trajectory and control corrections and various output subroutines. The sampling interval is one second.

The first experiment studies the effects of weights on the behavior of the states. Only one propulsion unit is considered. The input command is a step that occurs at five seconds into the simulation and its magnitude is the ship speed that corresponds to full ahead pitch in the maneuvering region. Figures (4) and (5) are the state and control responses for weights (i) in Table 1 and Figures (6) and (7) correspond to those using weights (ii). In the case of weights (i), the control correction is very sensitive to the shaft speed errors. The

Variable	Weights	
	(i)	(ii)
Shaft Speed	1	1
Free Engine Torque	0	.05
Power Lever Angle	0	.05
Throttle Control	0	0.1

Table 1. Weight Assignment.

erratic throttle response in Figure 5 is a result of this high sensitivity and the plant performance is not satisfactory. A significant improvement in performance results when the weights are changed to that of (ii). The throttle control is smoother and the variations in shaft compute $\bar{K}$ and use it to replace M in (40). This gave good results for diagonal P and it is much simpler (requiring a lesser number of computer runs) to pick weights for a diagonal matrix. The computation of $\bar{K}$, performed off-line, is possible by assuming constant ϕ_j and H_j and a constant nominal trajectory. Although an obviously erroneous assumption, this is not important. The matrix $\bar{K}$ serves only as a weighting matrix to replace M in (40).

To implement (41), we divide the nonlinear plant into seventy-seven linearized models valid for different regions of ship speed and propeller pitch ratio. Discretization of these models follow and the matrices that multiply Δx in (41), i.e., the control laws, are then computed and stored in a two-dimensional array. At each kT, a two-dimensional pointer, based on the measurement of ship speed and propeller pitch ratio, retrieves the appropriate control law from the array.

III.4 Propeller Pitch Control

In the pitch control system of Figure 3, by assuming no saturation in the first block at the moment, the equation relating the pitch piston displacement x_p, and the piston control u_p is

$$\dot{x}_p = K_p u_p \tag{45}$$

The discrete version of (45) is

$$x_{p_{k+1}} = x_{p_k} + K_p T u_{p_k} \tag{46}$$

where, as in Section III.2, T is the sampling interval. Let the ordered (or desired) piston displacement at time kT be $\hat{x}_{p_k}$. Then the control that will move the piston displacement to the desired position is

$$u_{p_k} = \frac{\hat{x}_{p_k} - x_{p_k}}{K_p T} \tag{47}$$

To accommodate the saturation block in Figure (3), (47) is modified to

$$u_{p_k} = \frac{1}{K_p} \operatorname{sat}\left(\frac{\hat{x}_{p_k} - x_{p_k}}{T}\right) \tag{48}$$

where

$$\operatorname{sat}(\ ) \triangleq \begin{cases} (\ )\ , \text{ if } \operatorname{abs}(\ ) < \text{Rate}_M \\ \\ \text{Rate}_M \times \operatorname{sign}(\ ), \text{ if } \operatorname{abs}(\ ) > \text{Rate}_M \end{cases}$$

and where abs( ) denotes the absolute value of the bracketed quantity and Rate_M is the maximum rate at which the piston can move.

In the absence of measurement noise, (48) will position the piston displacement exactly. However, in practice, the presence of piston displacement measurement noise will cause the pitch piston to oscillate. To prevent this cycling, a dead zone is incorporated in the control so that the control is applied only if it is outside the dead zone. Equation (48) then becomes

- Full range compressor delivery pressure governing was programmed as a proportional and integral control. This is not a requirement of marine systems, however it was a useful alternative control loop whilst software modifications were being carried out on primary controls.

- The P3 signal is also used to provide acceleration and deceleration control functions.

  Simply, schedules of maximum and minimum permissible fuel flows against P3 are used during rapid accelerations and decelerations.

- The start schedule was a programmable integrator. Following a start command the initial condition of the integrator was held at a value corresponding to light-off fuel demand. Subsequent to the engine lighting and reaching a preset running condition, the integrator was set to operate and the start fuel demand was ramped up at a controlled rate until control of the engine was transferred to the LP speed governor at idling (throttle).

- The computed control channel outputs are compared at a system of lowest and highest wins logic to determine the demand to the fast response fuel control loop sketched in Figure 1. A fuel flow is demanded from the pumps via an electrohydraulic servo valve. The delivered flow is measured by a flowmeter and compared with the demand within the P.A.C. Gain and stability of the loop are under control of the P.A.C. programme.

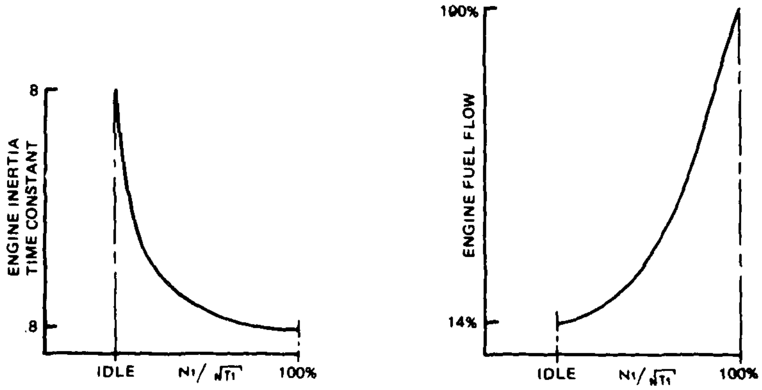


FIGURE 4 ENGINE STEADY STATE AND DYNAMIC CHARACTERISTICS

5. SPECIAL FEATURES OF IMPLEMENTATION OF CONTROL SCHEME

The implementation of the control scheme contained some features which,because of the programmability, overcame some of the problems which occur with dedicated analogue equipment.

Saturating Integrators

Dedicated analogue controls which consist of several 3 term controllers which are compared at a lowest wins gate to determine which loop has control authority, suffer from an integrator saturation problem. A control parameter which is below datum demands a control output which attempts to achieve the datum running conditions. The below datum error signal causes the integrator of the loop to travel to its saturation level. The integrator cannot be limited to a preset level because it is necessary that it can demand fuel flow outputs over the full operating range of the engine. Remember that when the loop is controlling the total fuel demand is held on the output of the integrator as the proportional and derivative, errors will be nominally zero. Thus the dedicated analogue control must allow the integrator to run freely. This causes a reduced performance of the control loop during transients which can only be satisfactorily overcome by special provision of additional analogue control equipment. If we considered the case of a transient reduction in load which causes control to be transferred to the power turbine speed governor, then the datum must be exceeded for an appreciable time in order to reduce the integrator output from its saturation level to a new steady state control level.

The integrator gain is usually quite low and thus the derivative and proportional terms have to compensate.

This discussion leads to the requirement that the output of a non controlling integrator should be held at a value marginally in excess of the fuel demand of the controlling loop. The excess is necessary to ensure that interaction between loops does not occur. Now in the transient load change considered above the integrator is in a position to assume control almost immediately. This reset function is difficult to implement in hardware items, but is easily achieved using software.

Controller Gain Adjustment

The requirement for controller gains to be adjusted to suit engine running conditions has been discussed. This can be satisfied in hardware but a software solution is more flexible and can cater more easily for variations within types of engines.

The following areas of an engine development nature were used as means of gaining experience of handling the PAC programme on the test bed as a development tool.

Starting

During the programming phase of the exercise it was known that the engine to be used for engine tests would be fitted with a development standard of combustion hardware. It was an ideal opportunity therefore to review the exact requirements of a starting system and develop a programme which would allow experimentation during engine tests. The major requirements are:-

- The start temperature should be less than 450°C
- The engine must achieve idle within 45 seconds

These criteria tend to conflict which can be seen if we consider current start procedures.

The engine is cranked over by an air start motor and a fixed level of fuel flow demanded to start the engine. The engine lights and exhaust gas temperature rises rapidly. When the light is detected, (normally a preset level of exhaust gas temperature), additional fuel is admitted to the engine, either at a controlled rate or as a schedule of compressor delivery pressure.

Thus in the attempt to reach idling quickly, additional fuel is admitted whilst the temperature is still rising following the initial light off. This only causes the temperature to rise further and is not desirable for the engine. If the engine had been allowed to stabilise without the addition of fuel above the light off level, the exhaust gas temperature would have peaked and then reduced to its steady state level. It is clear that the most suitable time for fuel to be increased during the starting cycle is subsequent to the peak of temperature. Thus a mechanism was programmed as an alternative to the standard system to achieve this. Programmable timers were also provided in the sequencer outputs to 'open shut off cock' and 'start ignitors', to assist in start optimisation for the engine.

Deceleration Control Parameter

Compressor delivery pressure has traditionally been used as the base for acceleration and deceleration schedules. It was suggested that LP spool speed would be a more suitable parameter for decelerations than the conventional P3 parameter. P3 has the disadvantage that an accidental partial flame out causes immediate further reduction in P3 which escalates the fuel flow reduction into a complete flameout situation.

This 'change of base' exercise was reprogrammed on the test bed and trial runs carried out within a few minutes - an experiment certainly not practicable on a conventional system.

6. PROGRAMMING TECHNIQUES

P.A.C.

The programmable nature of the P.A.C. enables the use of software techniques to overcome difficulties experienced with hardwired analogue systems. Such a technique was employed to apply integrator reset functions to overcome the saturating amplifier problem discussed in paragraph 5 above.

Consider the typical 2-loop controller shown in Figure 5. The programme compared the low wins inputs and output to establish the non controlling loop, by subtracting the output from each input and looking for a positive value. A jump routine 'initial conditioned' the non controlling integrator to the low wins (fuel demand) value. In effect therefore, since the proportional output from the controlling loop is nominally zero, the non controlling loop takes over as soon as its proportional output goes negative. This technique gave very acceptable results with bumpless transfer and zero overshoot.

It became apparent that the use of 'jump' routines could eliminate the need for more than one integrator and the control scheme indicated in Figure 6 was therefore programmed. Here the proportional terms of the control loops concerned were compared as before to establish the controlling loop. A jump routine applied the appropriate control error and gain input to the integrator which was than added to the 'proportional' low wins output to provide a two or three term control loop as before. Up to five control loops have been programmed in this way using a single integrator with very satisfactory results.

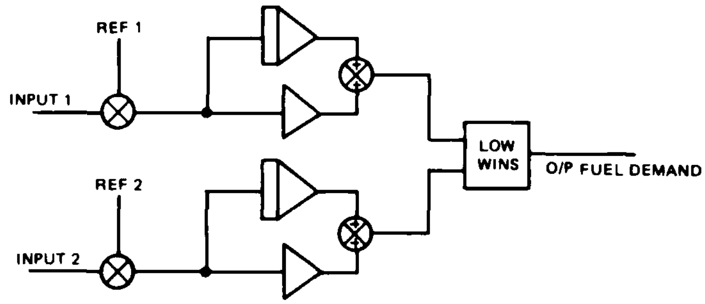


FIGURE 5 TYPICAL 2-LOOP CONTROLLER

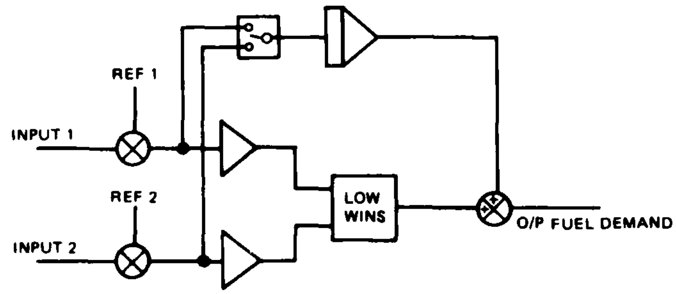


FIGURE 6 SINGLE INTEGRATOR CONTROLLER

Sequencer

The relatively straight forward logic for a sequence system does not require any special techniques to implement the sequence programme. However, system integrity can be enhanced by appropriate use of scratchpads and two examples of this are described.

The first combines an input with an allocated scratchpad to provide a software noise filter. Any input from an electro-mechanical switch will be subject to contact bounce, whilst electro-magnetic and electro-static interference will introduce stray noise; both may give rise to a false state if sampled at the appropriate time. Hardware noise filters will alleviate the effect but a more satisfactory solution is to use a software filter. The logic arrangement for this is shown in Figure 7. In principle the input concerned is stored in a scratchpad immediately after the logic programme; immediately before the logic programme the input is 'exclusive ORED' with the scratchpad. The logic programme is jumped around if the input and scratchpad states are not the same. In this way a filter nearly equal to the sequence recursion time of 40 mSecs is introduced. In the second example an output is set to the required state along with a scratchpad allocated to the output; at any point in the programme therefore the output and scratchpad states should be the same. If these are 'exclusive ORED' together normal operation will be indicated by a '1' output and a failure, either in the output driver or scratchpad, by a '0'. This can be used in a jump routine to give an alarm indication as required.

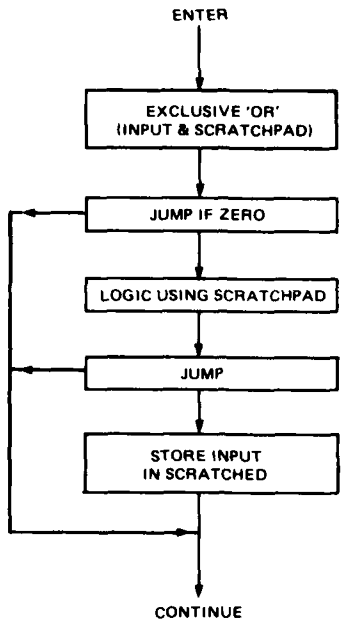


FIGURE 7 SCRATCHPAD FILTER LOGIC

7. PREPARATION TRIALS

Simulation Trials

A full range simulation of a TM3B was set up on an HY 48 analogue computer. The PAC control system was interfaced with the simulation and handling and optimisation trials carried out. This exercise proved invaluable as it provided a means of programme checking and familiarisation of operating the system prior to engine running. Some general optimisation of control loops was also carried out.

Rig Trials

As explained earlier the programmable control output was a fuel flow demand to a flow control loop comprising servo valve, fuel pumps and flowmeter. It was possible to arrange this equipment on a hydraulic rig and thus study the performance of this type of control loop and optimise it prior to engine running.

The response requirements of this loop vary between industrial and marine applications. The most demanding is the industrial system where, in order to contain power turbine overspeed within limits when load is rejected by the grid system, fuel flow must be reduced from 100% to a controlled deceleration line in approximately 150 mS without undershoot.

The system under test achieved this requirement with proportional feedback only. However, derivative feedback was programmed as an option in order that the proportional gain could be increased and thus reduce the steady state error of the loop. This is desirable for the setting up of acceleration, deceleration and start schedule functions.

8. DISCUSSION OF EXERCISE

Programming

The programming codes are very quickly learned by a control system engineer. Subsequent to the initial programming phase and simulation trials the Rolls Royce engineer was confident in all operational aspects of the equipment. Control schemes are generally drawn in analogue form and it was found possible to programme directly from the block diagram although it is advisable to insert a flow chart stage for documentation purposes. Thus programming is simple and still allows the user control engineer to implement the control scheme and does not rely upon the equipment manufacturer for support software.

Equipment

The equipment behaved well during the installation and commissioning phase of the programme. The equipment has been described in detail in previous papers given by UEL personnel and reviewed here. An attractive feature for the user is the limited amount of peripheral equipment which is required for commissioning purposes. The engine control comprises one rack unit and a pin programme unit (which is replaced following commission by erasable proms). The only other useful item of equipment necessary is an oscilloscope. The engine was run using the pin programme unit and changes to controller gains and other basic programme changes were carried out with the engine running. Thus control loops can be optimised as simple as analogue controls but adjustments are not available for unauthorised personnel to tamper with, as is sometimes a problem with analogue equipment.

It was also possible to make modifications in software very easily to produce changes in control philosophy. New hardware would have been necessary with a dedicated analogue or hydromechanical system.

Fault Finding

Trouble shooting is another area where the equipment has advantages. The mechanism of the controller is multiplexed analogue and computations are carried out at the gain controlled amplifier. An oscilloscope can be connected to the amplifier and the output value displayed at each programme instruction. Thus the equipment has all the advantages of programmability but problems within individual control loops can be diagnosed as with a conventional dedicated analogue system.

Performance

Some interesting statistics are listed below which demonstrate the reliability of the equipment although information was gathered over a short period of time.

Motoring hours	0	43 minutes
Running hours	33	58 minutes
Number of starts (subsequent to sequencer commissioning)		39 (all successful)
Average time to light		8 seconds
Average maximum TET (for soak temperature below 150°C)		430°C
Average time to 2000 rpm		45 seconds

Problem Areas

The installation was not totally free of problems although those that occurred were relatively minor commissioning incidents.

- A sequencer output was faulty. The output when addressed failed to operate. It was replaced and commissioning continued normally.

- The power supply incorporates an inverter which generated a high frequency noise. Initially at low signal levels some of the control parameters were affected. Suitable filters overcame this problem on site but the manufacturers were aware of the problem for future designs.

- Another problem was connected with a mechanical aspect of the equipment. The circuit board edge connectors were not full production standard. This caused some problems with board insertion into the racking system. Again the manufacturers recognised the problem and subsequent units were equiped with a satisfactory system.

Programmable Controllers

A gas turbine operator requires an overall installation which has high reliability and integrity, and is not directly concerned whether the equipment is dedicated analogue or programmable. The G.T. manufacturer, however, has many problems in order to provide this reliable system. The gas turbine is undergoing continuous modification to improve life and to satisfy new requirements especially in the combustion system. These modifications necessitate control system changes which are not always easily implemented on dedicated equipment. It may require a change in fuel valve profile or the addition of circuit boards. The turn round time in development alone is considerable and the implementation of production modifications can be lengthy and costly. Thus the control specification is not as well established as is imagined and therefore the gas turbine manufacturer has a need for programmable equipment both for development and production purposes. The UEL PAC proved very useful for Rolls Royce requirements because it appeared within the limited testing to have the reliability of dedicated analogue systems and complete system programming could be carried out by a control engineer. Extensive fault finding equipment was not necessary.

As mentioned previously this exercise has formed the initial phase of a Rolls Royce study into future control systems. The alternatives available with programmable equipment are microprocessor systems and programmable analogue. The study is not yet complete and the final choice of system will be dependent upon many factors both commercial and technical. However, from Rolls Royce experience to date, some indications of the relative merits of the alternatives is made here.

It is suggested that with the present standard of microprocessor control simple programming and fault diagnosis will not be possible. It will be possible to develop a high level language for ease of programming but this requires purchase of both hardware and expensive software packages from the control equipment supplier. Fault finding may be relatively more difficult and would require extensive equipment.

The move towards microprocessor control will have some cost advantages but much of the hardware required for gas turbine control will be for signal conditioning and this will be a high percentage of the total, irrespective of the type of equipment chosen.

The processor will perform satisfactory control but possibly only via the services of specialist programmers. In addition, considerable training will be required for service personnel. Microprocessor systems may well be considered strongly for future controls because of their ability to perform other tasks than control. Engine health monitoring is a function that can be performed by microprocessors. Thus the commonality of controller, sequence and health monitor will make the microprocessor a very competitive system.

Hydromechanics

As previously mentioned, testing of the hydromechanical system used was not a part of the exercise but it is worth noting that the system behaved exceptionally well. The simplicity of the system is attractive and work is being carried out to select suitable loop components and to determine a means of fail set operation which would be required for marine applications. A fail safe loop of this type has recently been used to control a gas turbine in an industrial application.

Further PAC Applications

As a footnote, the UEL PAC has been used ior further development testing of fuel system requirements. It has also been used in an entirely different role wherein it was programmed to control the dynamometer in order to run the shore trials simulation of the CAH through deck cruiser with various ahead and astern propellor laws. To achieve this, mainshaft torque and speed were measured, and torque demands made to the dynamometer and load control valves. A switch selected the required running law.

This particular unit is now retained by Rolls Royce IMD for continued use for internal development activities of the type mentioned above where simple reprogramming makes it very attractive for this type of development engine testing. Its wider use in operational areas is still subject to a continued study in comparison with the more normal type of digital microprocessor system and the final outcome of this study is still awaited.

ACKNOWLEDGEMENTS

The authors wish to acknowledge that this evaluation was partly supported by the Ministry of Defence. However, where opinions are expressed they are those of the authors.

MANSIM - A COMBINED SIMULATOR AND SIMULATION SYSTEM FOR SHIP STEERING

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ABSTRACT

A computer-based system for simulation of ship steering and ship traffic in restricted waters is presented. The system is making use of a general purpose, interactive computer facility for simulation of dynamic processes (GENSIM), comprising both a hardware system, including four colour TV-screens, trackerball, keyboards and a console for process control and supervision, and an associated software package (SUPERSIM).

The Main Modules of MANSIM

- The SHIP module describes the dynamics of the ships using differential-, logic-, and algebraic equations.

- The ENVIRONMENT module describes the environmental factors and comprises topological description of the area under consideration.

- The OPERATE module allows the operator to interact with the system. By use of predefined push buttons, the operator controls the SHIP module.

Particulars of the Simulator System

- Ships may be manoeuvred/steered on the TV-screens under the influence of the environment (current, wind, bottom, etc.).

- Ship traffic operation/transportation analysis may be performed in both a deterministic and random way.

The Main Objectives for such Simulations

- To analyse ship manoeuvrability.

- To analyse a ship's ability to navigate a given fairway and to determine the risk by that navigation.

- To analyse ship accidents with respect to manoeuvring possibilities and limitations.

- To determine procedures which will minimise the risk of accidents during navigation in a given area.

INTRODUCTION

Det norske Veritas is engaged in a large research project in cooperation with Norwegian maritime authorities, the Norwegian

Technical University, shipowners and underwriters to cover the various cause relationships of collisions and groundings.

The working tasks of the project include case studies of casualties and the study of manoeuvring qualities. The latter directed to finding out time margins and manoeuvring limitations in critical situations due to ship characteristics. Finally, results will be evaluated through simulation studies. In these studies the need for a facility able to perform manoeuvring experiments by computer simulation has become evident.

Existing ship handling simulators are usually not well suited for such purposes because of the often laborious and costly task of changing operation area and also system limitations like the impossibility of showing nearby coast lines or including other ship traffic. Other kinds of computer manoeuvring simulation programs do not usually have the possibility of interactive steering of ships or simultaneous presentation of the dynamics.

Based on existing experience with interactive simulation of dynamic processes in DnV, the development of a ship steering simulator to fulfil the following requirements was decided:

- Simultaneously visualization of the ship's positions and headings with the scene of simulation (fairway, harbour, etc.).
- Interactive steering of the simulated ships.
- Realistic description of ship manoeuvring properties.
- Manoeuvring of two or more ships simultaneously.
- Operation of two controllers simultaneously and independently for simulation of collision and traffic situations.
- Easy definition and change of scenario and environmental conditions.
- Inclusion of realistic wind and sea state conditions.
- Inclusion of random and probabilistic effects.
- Optional time scaling (real time, faster or slower).
- Documentation of the simulated events including comprehensive statistics.

A computer facility like this would of course be of value for a wide range of applications. Computer simulation of ship manoeuvres is applicable not only to ship handling training, but also to ship design and harbour design /1/, /2/.

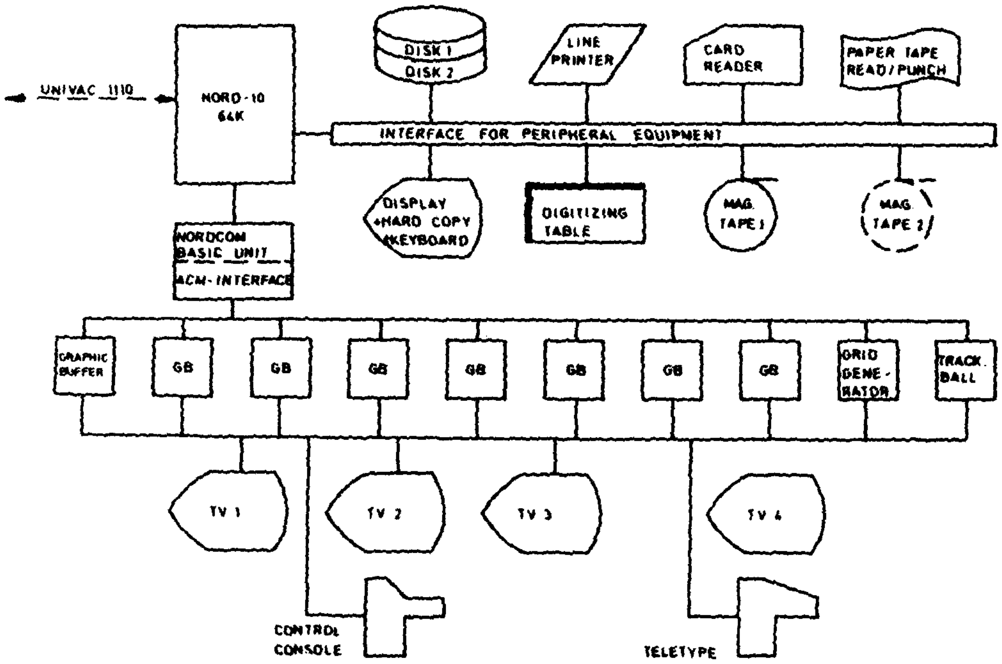

FIG. 1 The hardware configuration of GENSIM

THE SIMULATION SYSTEM

Prior to developing the manoeuvring simulation system, DnV has already developed a general purpose, interactive computer facility for simulation control and supervision of dynamic processes (GENSIM) /3/, comprising both a hardware system and an associated software package (SUPERSIM).

The versatility and flexibility of this system made it suitable for implementation of the ship steering simulation model, MANSIM, as another application program.

The Hardware System

The total hardware system of GENSIM is shown schematically in figure 1; the following units are of particular interest:

- Process computer (NORD-10, 64K, 16 bits)
- Disk and magnetic tape stations
- Alphanumeric keyboard with display
- Line printer
- Colour TV-screens
- Hard copy unit
- Trackerball
- Modular process console with lighted push buttons

The Software System

The general software system, SUPERSIM /4/, comprises several administrative and mathematical modules and is based on FORTRAN. The subroutines may then be linked together in different ways to simulate configurations of one type of system or to simulate different types of systems.

This approach, as for other high-level simulation languages, allows the user with only a working knowledge of FORTRAN, to concentrate on the phenomenon to be simulated rather than the mechanism for implementation and execution of the simulation.

This is also true for other application programs implemented, such as OILSIM /5/ and GASP 4 /6/. Particulary, GASP 4 is a general simulation program well suited for simulation of combined systems (both discrete and continuous). The MANSIM system is constructed in the same way as these programs and uses modules from all of them.

By combining these modules and still conserving their best properties, one will have a system which will be able to simulate manoeuvring/steering of ships and ship traffic both in micro- and macrocosmos and in both a deterministic and random way.

DESCRIPTION OF MANSIM SOFTWARE MODULES

The program system is modular in such a way that new modules may easily be introduced without the necessity of rewriting other parts of the system. The main modules are shown in the figure 2:

Module INITIATE

The module describes the scenario to be used for a paritcular simulation study and includes:

- Initiation of the module ENVIRONMENT; i.e. topographic description of the waters where the simulation is to take place, including definitions of necessary weather and sea state conditions (wind, waves and current).
- Initiation of the module SHIP; i.e. definition of own ship (SHIP 0) and other ships (SHIP n, n = 1,...) taking part in the simulation study; including main ship data, manoeuvring properties, initial position, heading and speed.
- Selection of steering mode for each SHIP n; i.e. mode HUMAN or mode AUTO.

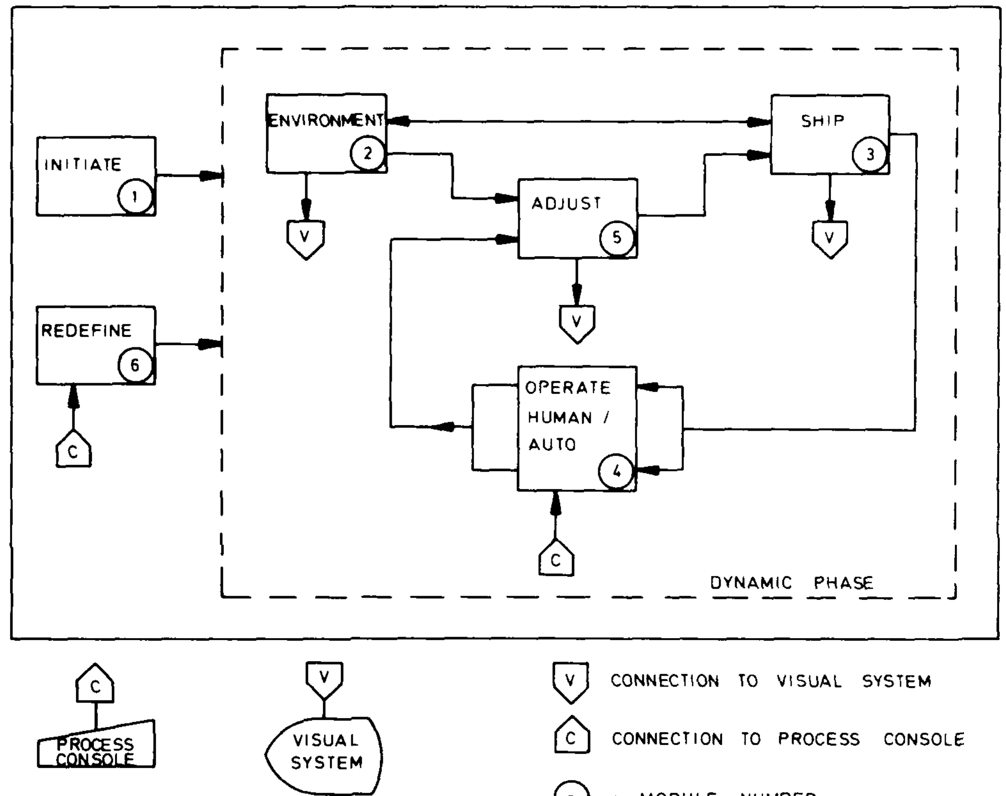

FIG. 2 A Block Diagram of MANSIM

Module ENVIRONMENT

This module describes the environmental conditions of the simulation scenario. Some of the conditions will impose disturbances on the SHIP module.

Fixed data. These are a description of the area under consideration including:

- Land contours defining the extent of navigable waters. The land contours will be generated from charts by computer graphic methods.

- Depths of waters (mean water depths).

- Positions and main characteristics of buoys, marks, lights etc.

Variable data. These consist of those environmental effects which vary in time both in regular or random ways:

- Tidal depth variations.

- Water currents (velocity and direction) including both tidal and residual currents.

- Wind speed and directions.

- Wave direction and significant heights and/or period.

Module SHIP

This module contains the dynamics of the ships using algebraic-, logic- and differential equations. Equations for different types of ships will be used if necessary. The models should be as realistic as possible so that the operator will have the feeling of steering a real ship. The program system ensures that the dynamic models may easily be changed according to the state of the art about the dynamic of ship manoeuvring, and the aim and accuracy requirements of the study to be performed. The dynamic equations include limitations of the control devices (power, steering) and realistic time delays for engines and steering gear. Realistic reactions of maloperation of control devices can be included.

The choice of dynamic model is to a certain extent dependent on the manoeuvring coefficients available /7/, /8/, /9/. Most models today depend on manoeuvring coefficients obtained by a planar motion mechanism of a ship model tank. But it is possible to obtain manoeuvring coefficients from full scale sea trials and the model must then be adjusted accordingly /10/.

The ship data including the manoeuvring coefficients and the control device parameters are generated by the INITIATE module. Most of the manoeuvring coefficients will change with changes of environmental data (defined in module ENVIRONMENT), through the module ADJUST.

The output of module SHIP is the resulting position and heading of the ship and all state variables of the dynamics. All this information is presented to the operator by the visual system and is also stored for later analysis.

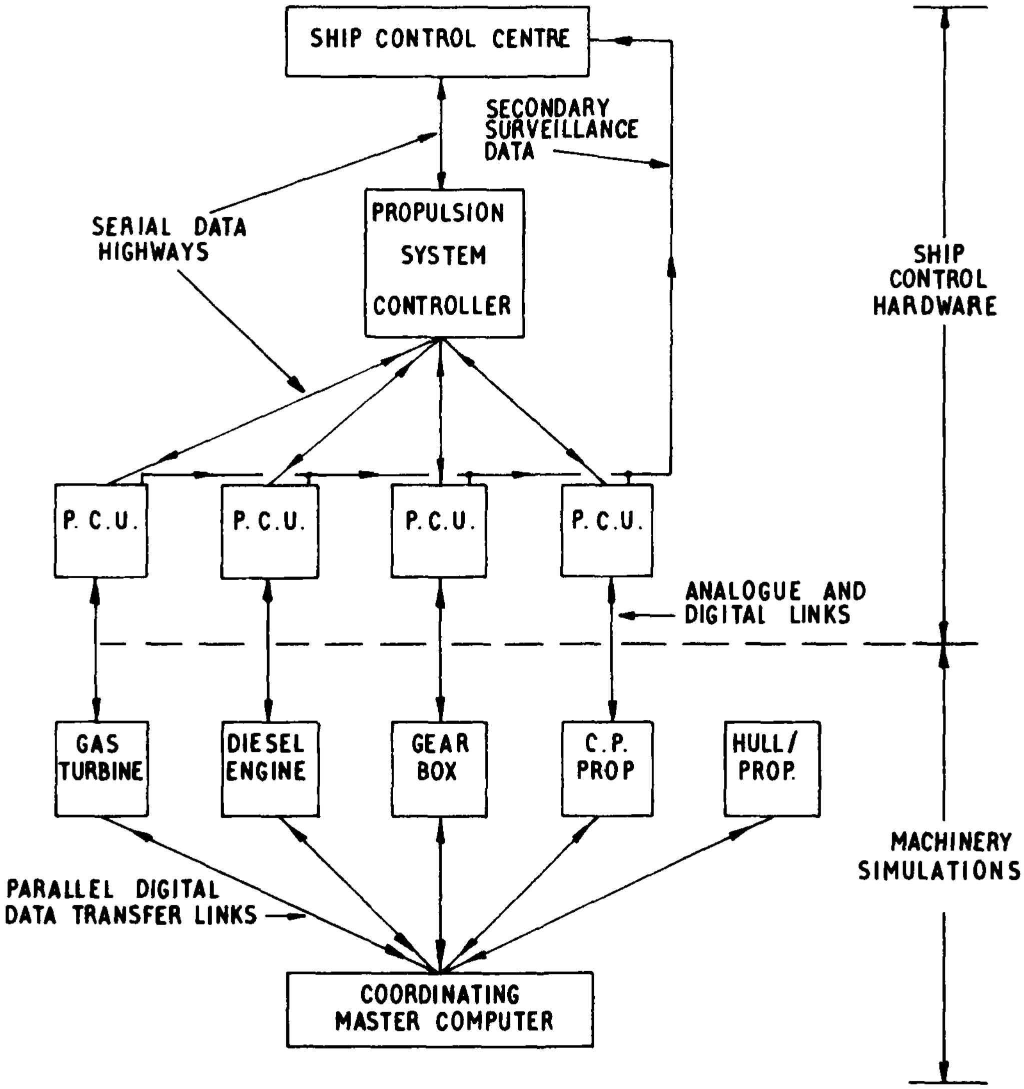

Figure 2. Illustrates communication between simulated propulsion machinery and control system

To enable software to be developed in a methodical manner and to a consistent standard, it is intended to design application programmes and operating systems under the guidance of a software methodology known as MASCOT (4). This regulated approach to software development imposes a style of programming under a well defined set of rules which should simplify the task of writing programmes to a consistent standard.

MACHINERY SYSTEM REQUIREMENTS FOR AN EVALUATION CENTRE

Having identified the need for a Machinery Control Evaluation Centre, consider now the facilities to be provided. The options are:

(1) Full scale machinery systems.

(2) Real-time dynamic simulations.

(3) A mixture of machinery systems and real-time simulations.

Examining first the full scale machinery option, it is immediately apparent that extensive test facilities would be required to house and service main propulsion plant, electrical power generation equipment, chilled water systems etc. Furthermore, if the machinery is to be operated over its normal working range, then massive power absorption capability would be necessary. If, in addition, the influence of the propeller on propulsion system behaviour was incorporated, then sophisticated power injection equipment would also be required. Such a system would be complex and expensive to implement.

The preferred option is to evaluate future control concepts against real-time dynamic simulations. This has advantages with respect to cost and flexibility and, provided the limitations of the mathematical modelling techniques are understood, then this approach will form a sound basis for the analytical/theoretical appreciation of both machinery system performance and its associated control scheme. It has the further advantage that evaluation of controls can be carried out ahead of availability of the actual machinery.

The third option would only be employed when simulation was not practicable for reasons of system complexity. For example, a subsequent section of this paper explains that a number of small diesel generators will be used to represent the electrical power distribution system because real-time simulation is not possible without over simplification.

COMPUTING REQUIREMENTS FOR MACHINERY SIMULATIONS

The trend towards the use of digital computers for simulation purposes has increased in recent years as the cost of digital systems has fallen, while that of analogue systems continues to rise. It is not intended to debate the relative merits of analogue versus digital, but for the purpose of this particular application, the digital approach was adopted because of its superiority in terms of cost, flexibility and maintainability. Another factor which influenced the choice was the availability of advanced digital equipment combined with fast numerical computing methods which made possible the real-time simulation of complex machinery systems.

A number of computer configurations were considered before selecting the machinery simulation approach illustrated in Fig 2. This shows a number of dedicated computers, each representing a specific item of machinery, with interactions and data transfer between machines being coordinated by a more powerful master computer. The chosen computer configuration is not the most economic in hardware terms, but carries reduced technical risk by virtue of minimal communication problems, and is considered to have a high degree of flexibility with respect to changes in machinery simulation configurations.

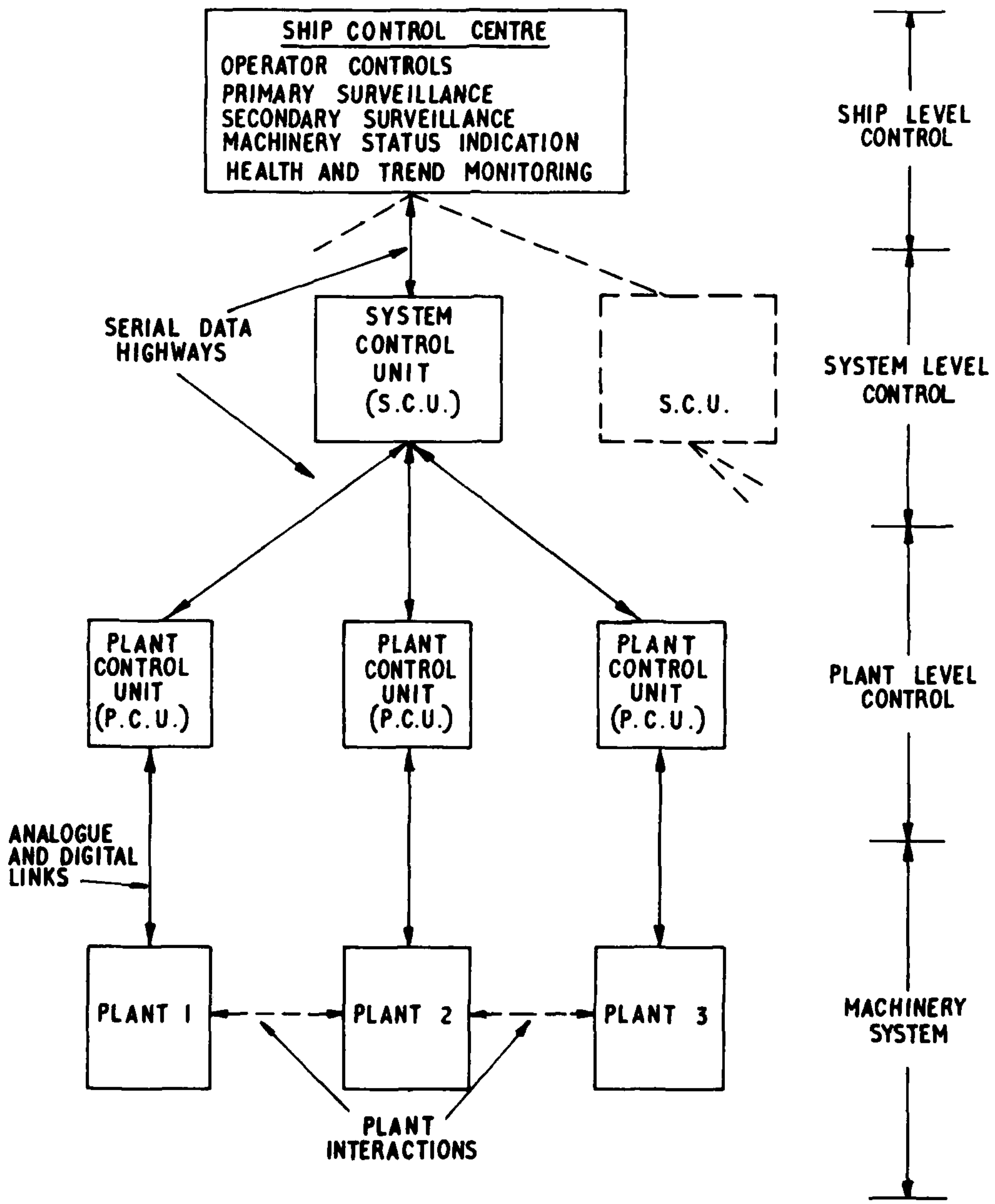

Figure 1. Hierarchical structure of control system

Ships in the current R.N. building programme use mainly analogue electronics for control, data transmission and data display. The general philosophy has been for centralized control from a Ship Control Centre with analogue data and command links to and from the machinery spaces (1). Reversionary control requires individual items of machinery to be manned and overall control coordinated from a central point.

Results of current research indicate that future generations of ships are likely to exploit recent advances in digital technology by the use of mini and micro computers for the control and surveillance of all machinery systems. The general philosophy which is presently preferred is the hierarchical structure, shown in Fig 1, with three distinct levels of control, namely:

(1) Plant level.

(2) System level.

(3) Ship level.

It is foreseen that at plant level, common micro processor based hardware will be used to control all items of machinery with appropriate changes in software to suit different applications. The scheme will have distributed computing power with micro computers at plant, system and ship levels and, apart from a few analogue control loops at Plant Control Unit (P.C.U.) level, digital technology will be used throughout. Primary control and surveillance of all major items of plant will be exercised through the P.C.U. which will accept plant input/output information in either analogue or digital form and will communicate to a higher level of control, a System Control Unit (S.C.U.) via a serial data highway. The third and highest level of control, the Ship Control Centre (S.C.C.), will have two-way command and data communication with all machinery control systems. Secondary surveillance data (used for health and trend monitoring) will be transmitted directly from the P.C.U. to the S.C.C. where it will be manipulated and displayed as an indicator of machinery health. Alternatively, the secondary surveillance data can be stored for subsequent analysis.

COMPUTER HARDWARE AND SOFTWARE REQUIREMENTS FOR MACHINERY CONTROL

Since the primary purpose of the Evaluation Centre is to assess the performance and suitability of processor based control and surveillance concepts, it is essential that both hardware and software aspects of the chosen processors and their associated peripherals, are fully appreciated. A secondary role of the Centre is foreseen as providing long term hardware and software support of processor based systems throughout their service life and this requires detailed knowledge of how the chosen control strategy is implemented.

Research studies have indicated that to cover the range of tasks envisaged for machinery control and surveillance, both mini and micro computers will be required, each having a 16 bit word length (2). Ideally, the processor should be based on a commercial range but will be required to meet the full military environmental specification. In addition, a comprehensive and fully supported software development capability is needed to simplify the programming task. The selected processor should be adaptable, particularly with regard to input/output interfacing, to meet the variety of control and surveillance needs of propulsion and auxiliary machinery.

To ensure uniformity throughout the field of machinery control, it is intended that all ship application programmes be written in CORAL 66, (3), a high level language, used widely by M.O.D. in real-time control applications. The language, which is simple to learn, produces an efficient code with consequent fast execution time (2). Compilers have been developed for many of the current range of mini and micro computers.

CONSIDERATIONS LEADING TO THE DEVELOPMENT
OF A MACHINERY CONTROL EVALUATION CENTRE

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ABSTRACT

Ships in the current Royal Naval building programme use mainly electronic analogue technology to exercise control of machinery systems but research indicates that future designs are likely to exploit recent advances in mini and micro computer technology. To minimise risk during the development stages of new machinery control systems, there is a need to carry out comprehensive evaluation in advance of availability of the machinery systems. It is proposed to evaluate future control concepts against real-time simulations of the machinery system. This paper examines the requirements of an Evaluation Centre to provide these facilities.

INTRODUCTION

In providing automation for warship machinery, previous control system design practice has been to consider each plant or group of plants as a separate requirement. This arose largely from constraints imposed by the type of control technology employed e.g. analogue electronics, pneumatics, fluidics etc. With the advent of cheap and reliable mini and micro computers it is now considered that a totally integrated supervisory scheme for the whole ship is technically feasible. These future schemes are likely to be based on advanced digital technology with extensive use of mini and micro computer for machinery control, data transmission and data display. However, there is increased risk of difficulties being experienced in introducing equipment into service which exploits new techniques. The desire to reduce such risks has led to the formulation of a requirement for an Evaluation Centre capable of assessing the performance of future machinery control and surveillance systems.

If the adequacy of these systems is to be fully explored in advance of the availability of the first ship and also in advance of the availability of the machinery then the most convenient and flexible means of assessing total system performance is to test against real-time simulations of the machinery systems. The following text considers the facilities needed to evaluate future control systems and reports progress made towards providing them.

CURRENT AND FUTURE CONTROL SYSTEM PHILOSOPHIES

The degree of automation applied to warship machinery has progressively increased with successive ship designs for two main reasons:

(1) Increased machinery performance and interactions between sub-systems makes uncoordinated (manual) control hazardous from a machinery standpoint.

(2) Operational and manning policies have led to un-manned machinery spaces and a consequential increase in automation and comprehensive information display systems.

/10/ K.J. Åstrøm and C.G. Källstrøm: "Identification of Ship Steering Dynamics", Automatica, Vol. 12, 1976.

/11/ C.L. Crane: "State of the Art on How to Include Human Control into the Method of Investigation", Symposium on aspects of navigability of constraint waterways, including harbour entrances, Delft, April 24-27, 1978.

/12/ K.H. Drager et.al.: "A Study of Relationships Between Different Causes of Collisions and Groundings", 3rd Int. Symposium on Marine Traffic Service, Liverpool, 1978.

/13/ L.I. Jacobsen et.al.: "Analysis of the Collision Between....." DnV report No. 78-128 (Confidential Commission report).

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Implementation and testing of MANSIM is just started. Although lacking on-site experience with this system, experience with similar systems on the same hardware configuration is excellent (OILSIM, STEAMSIM).

As for the MANSIM system, some of the main objectives are:

- To analyse ship manoeuvrability.
- To analyse a ship's ability to navigate a given fairway and to determine the risk by that navigation.
- To analyse ship accidents with respect to manoeuvring possibilities and limitations.
- To determine procedures which will minimise the risks of accidents during navigation in given area.

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/3/ F. Krogh: "GENSIM: A General Purpose Simulator and simulation System - Applied to Steam Power Plants (STEAMSIM)", Conference on Simulator Training for Seagoing Engineers, Gosport (UK), November 4, 1976.

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/6/ A.A.B. Pritsker: "The GASP IV Simulation Language", John Wiley & Sons, New York, 1974.

/7/ H. Eda and C.L. Crane: "Steering Characteristics of Ships in Calm Waters and Waves", SNAME 1965.

/8/ J. Strøm-Tejsen: "A Digital Computer Technique for Prediction of Standard Manoeuvres of Surface Ships", NSRDC Report No. 2130, Dec. 1965.

/9/ N.H. Norrbin: "Theory and Observations on the Use of a Mathematical Model for Ship Manoeuvring in Deep and Confined Waters", SSPA report No. 68, 1971.

collision /13/.

A simulation study includes the two ships, properly modelled, and two operators control the ships according to the log books. A number of questions may be answered:

- To what extent are positions and time events reasonably correctly noted in the log books, charts and testimonies?

- What influence has wind and current?

- What influence has shallow water or interaction with banks or ships?

- What unintended manoeuvres could actually have taken place immediately before the collision which caused the sudden turn?

- Knowing the relative positions of the two ships at the moment of collision and the manoeuvres immediately before - what was the positions of the ships at the moment where the situation become critical?

- Could the collision have been avoided by alternative emergency manoeuvres?

Risk Evaluation

A ship is supposed to travel along a straight track towards an offshore terminal when a failure occurs. The ship may hit offshore installations surrounding the terminal if the control of the ship is lost. By repeated runs an assessment of the probability of collision between a ship and an offshore installation can be made /14/.

CONCLUSIONS

The paper describes the proposed layout of a general purpose manoeuvring simulator and simulation system (MANSIM) where the man/machine communication is based on colour screens, process consoles with push buttons, tracker balls and alphanumeric keyboards.

Trends in the construction of new supervisory and control systems show an attitude towards the use of similar hardware systems. Further, in some of the new manoeuvring training simulators the visual presentation is digitally generated.

Some of the main advantages by using this type of visual system are:

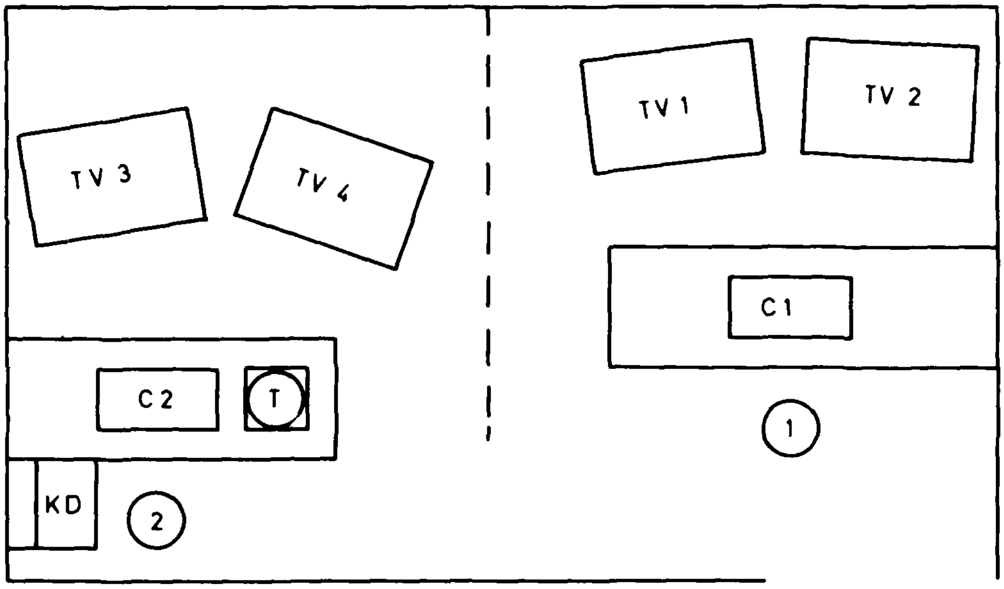
- A number of "other ships" may be generated and steered individually.

- Large number of lights, buoys, etc. can be introduced.

- Easy to change scenario for a different exercise.

- Different types of realistic weather conditions may easily be superimposed.

- Computer generated scenes can be composed in a quick and economic way.



1	OPERATOR	C 2	INSTRUCTOR CONSOLE
2	INSTRUCTOR	K D	KEYBOARD WITH DISPLAY
C 1	OPERATOR CONSOLE	T	TRACKERBALL

FIG. 8 Layout of system for two operators

EXAMPLES OF USE

Grounding Analysis

A ship has grounded on the Norwegian coast. The voyage before the accident is reconstructed from the official inquiry and the ship's log book. A thorough analysis has detected a great number of possible causes among which is the influence of wind and current on the ship's course made good /12/. In the simulation study the ship is steered according to the log book and the effects of drift forces are evaluated. Also alternative avoidance manoeuvres in such a situation can be tried out.

Collision Analysis

Two ships meet in a harbour entrance channel. One makes a sudden turn towards the other and collision is unavoidable. From documents and testimonies the two ship's manoeuvres up to the unexpected turn are known and their relative position just after the

If the simulation is performed with two or more manually steered ships, it is recommended that two operators take part in the simulation (figure 7). Operator 1 will control SHIP 0 with the predefined set of buttons on the process console and receive information from the simulation by screens 1 and 2.

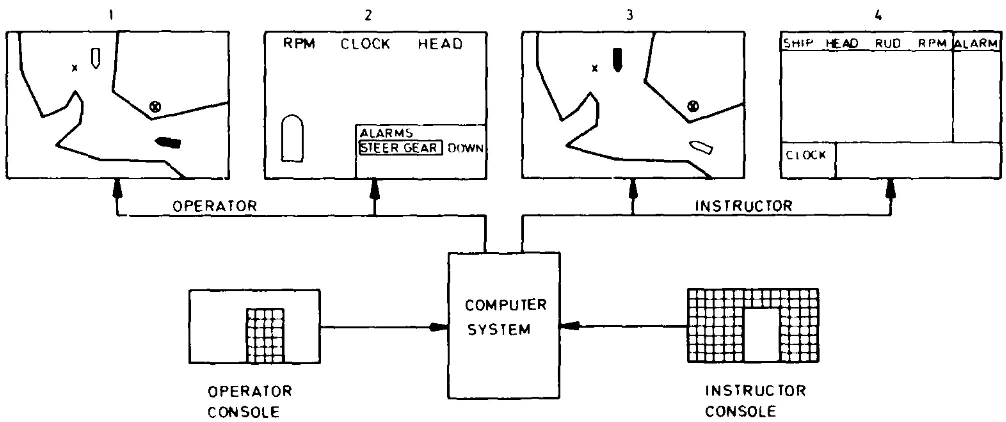


FIG. 7 Example of information on the four TV-screens in a collision study

Operator 2 ("Instructor") will have an identical process console, but with the SHIP 0 buttons covered, to control the other ships and to supervise and control the simulation and the documentation, as previously explained. The instructor will also have at his disposal an alphanumeric keyboard and a trackerball (to position a marker in the plane of a TV-screen) for easy communication with the system and the simulation process (figure 8). The instructor will use screens 3 and 4 for information. One screen will usally present main state variables of the ships, SHIP n, and warning and failure messages concerning these ships; the other may show the operational area and the simulated ships (as on screen 1) or other selected information.

A special unit allows hard copies of the TV-pictures to be made at any time during the simulation. All data concerning the simulation run will be stored on files.

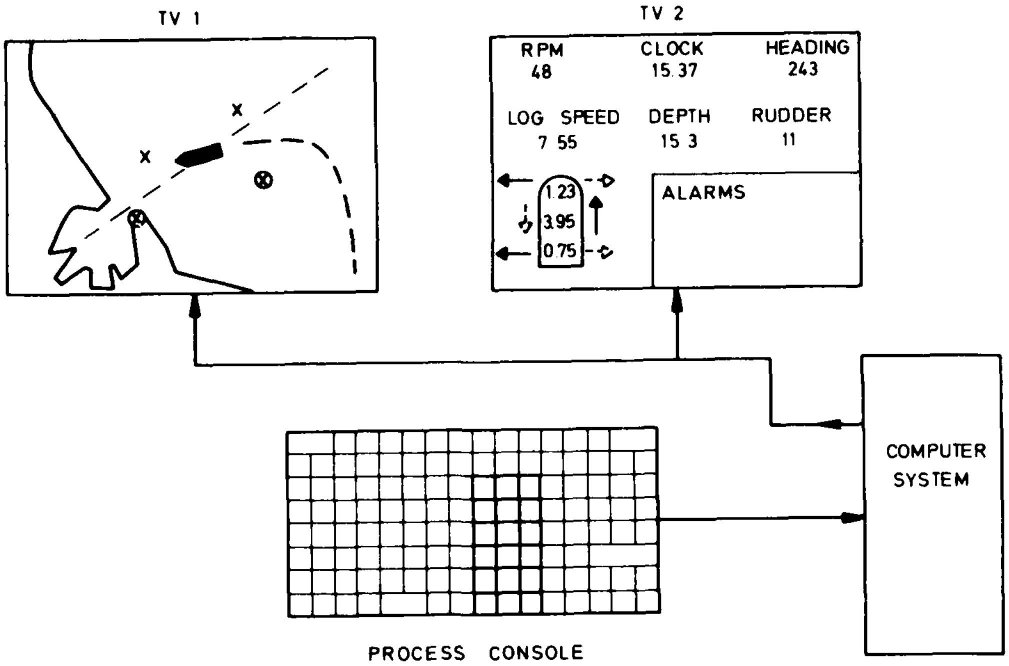


FIG. 5 The two main screens for SHIP 0

When simulation is performed with only one manually controlled ship, two more screens are available for additional information. These may be for instance current or wind maps or graphs of the process variables (figure 6).

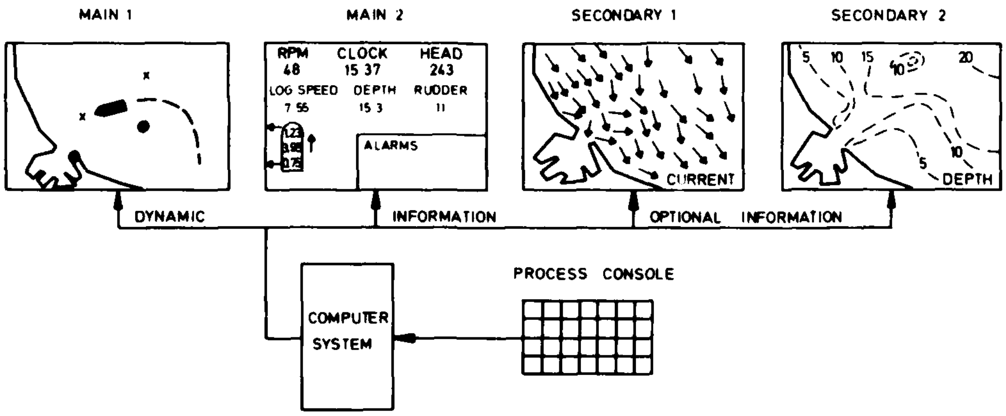


FIG. 6 Example of information available for single ship control

Module ADJUST

This module collects actual environmental data and actual control orders for each ship. The module then performs calculations and updating of all manoeuvring coefficients and other state variables for the module SHIP before another dynamic step is performed.

Module REDEFINE

By predefined buttons on the process console the system may in a limited way be redefined during the run. This may for instance be scaling of operation area or change of steering mode of ships (figure 4).

The system may also be stopped at any time by merely pushing a specific push button on the alphanumeric keyboard. The alphanumeric display investigates the type of action to be taken by answering: "DO YOU WANT THE MENU (Y OR N):". By answering Y (yes), the menu is displayed and the operator is asked to select the operation that is wanted. Each selected operation may consist of a number of (sub)-items to be specified. In this way may the operator be able to redefine the system before continuing.

This module also includes a failure module, where failures on own ship (SHIP 0) may be initiated through a random generator. Failures may for instance be loss of power or steering or other functional failures.

After a simulation study is ended, all operations and results are stored on disk files. These data may then be investigated either in tabular form on the line printer or presented as curves on the screens. By repeating runs with failures, statistics for accidents may be gathered.

VISUAL PRESENTATION

The results during a simulation run will be presented dynamically on colour TV-screens. Four screens are available for the system. The information presented on the screens may be chosen according to the problem studied.

Screen No. 1 should always show an overhead view of the area under consideration and the dynamic motion of the ships included in the simulation study (figure 5). The ship to be manually steered (e.g. SHIP 0) may have a colour distinguishing it from the others. The overhead view may be scaled up or down on screen No. 1 or on another screen.

The map-like overhead view is chosen because of its simplicity and ease of change. The picture will also contain much of the information used in map navigation or radar navigation. Whether such a picture gives sufficient information for realistic real-time steering of ships is an open question.

Screen No. 2 will usually be reserved as instrument panel for the manually controlled ship (SHIP 0, figure 5). All navigational and functional information which is normally available on the ship bridge will be displayed on the screen.

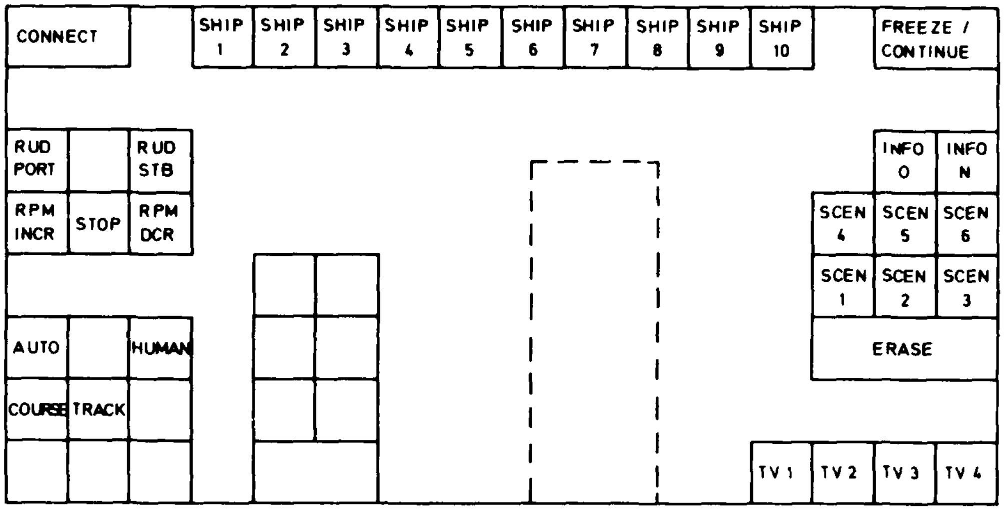


FIG. 4 Proposed Mansim Process console for "Instructor"

Part of the process console is reserved for operation of SHIP 0, which is always operated in mode HUMAN (figure 3). A number of push buttons define the engine orders and a pair of buttons control the rudder angle. By for instance pushing the button "SLOW AHEAD" the RPM value will change according to the characteristics of the engine until the desired value is reached. The button will light up and the light remain until a new engine order is given. By pushing the "PORT RUDDER" button the rudder will turn to port at a prefixed rate (according to the characteristics of the steering gear) until the same button is pushed again.

The remaining process console is reserved for operation of the other ships introduced in the simulation (figure 4). A number of push buttons defines the particular SHIP n which should receive orders. The CONNECT button then couples SHIP n to the desired rudder order or speed (RPM) increase or decrease.

Other buttons can be defined for direct and predefined change of operation area or scale of area, presentation of state variable curves, change of operation mode and error initiation.

Mode AUTO. By generation, any of the SHIP n may be given the operation mode AUTO. The AUTO function may be either a simple course controlled conventional autopilot algorithm, a "follow close to predefined track" algorithm or a decision algorithm which simulates the adaptive behaviour of a human controller. Many proposals for criterions used by a human controller in adaptive situations exists /11/. The simulation system provides a facility where such models can be tested and compared, as different human models are easily implemented in the system.

At present the system is designed to handle a maximum of 11 ships simultaneously (SHIP 0 + SHIP n, n = 1,..., 10).

Module OPERATE

By this module the simulation process is controlled either interactively by the process console or by a predefined algorithm representing an autopilot or a human controller. For each SHIP n the operation mode is given in the generating module INITIATE, and may be changed through module REDEFINE.

Mode HUMAN. For interactive man-process operation a process console with predefined push buttons is connected to the system (figures 3 and 4). For ease of use, and to improve the view of the console, the buttons are grouped and coloured according to function. Illumination of a button indicates that a push is acknowledged by the system, the light remains until the requested function is completed.

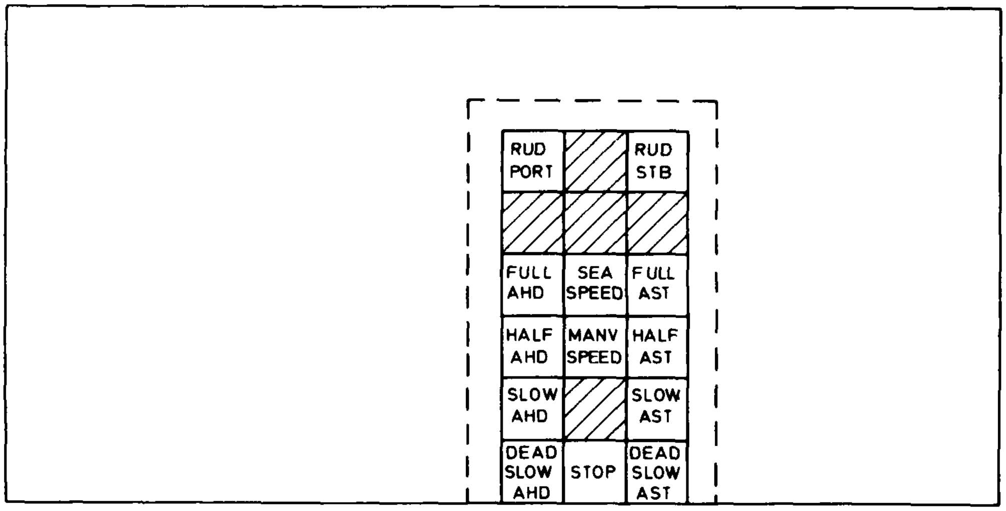


FIG. 3 Proposed Mansim Process console for Ship 0

The developed form of the Evaluation Centre can be seen from Fig 2. which shows the communications between a simulated propulsion system and a hierarchical control system. It is anticipated that, at a later date, additional items will be added e.g. chilled water system, diesel generator system etc until eventually all major machinery systems are incorporated.

It is intended to base the simulation computing complex on Digital Equipment Co. PDP11 series computers. This choice was influenced mainly by the desire for compatibility with existing NGTE (Naval Mechanical Engineering Dept) computing facilities and its associated support software. Major items of machinery will be programmed to run on PDP11/03s and 11/04s depending on the complexity of the simulation. Data transfers and interactions between the smaller computers will be coordinated by a PDP11/60 master computer.

MACHINERY SIMULATION TECHNIQUES

Simulations of engineering systems are created for a variety of reasons and their purpose normally dictates the simulation strategy employed. In this instance, the objectives are twofold; to predict machinery system performance while operating under various control system constraints and secondly to assess control system designs during the early stages of development. Simulation techniques for these purposes can be divided into two main categories, namely transfer function models and component based parametric models.

Transfer function models define input/output relationships by means of linear differential equations with constant coefficients. Most engineering systems, however, have non linear characteristics and therefore, this simulation approach is restricted to linear systems. Furthermore, if several output parameters are required from the model then transfer functions must be developed for each parameter with a consequent increase in complexity. This approach to simulation relies heavily on experimental data to formulate the models and frequently test data is not available, particularly for new machinery designs. Because of these shortcomings transfer function models are considered to be of only limited use in this application.

Parametric simulations represent engineering systems by establishing the physical relationship that exists in each element of a process or machine. Hence, given the inputs of an element and the physical laws that govern its behaviour, then a mathematical model can be derived which can generate most of the system variables as outputs. Because of the general nature and flexibility of the parametric model, this approach was chosen as the preferred simulation method.

With the development of digital simulation techniques and the application of fast numerical methods to simulation, it is now possible to devise real-time parametric models which represent complex, non-linear machinery systems over their entire operating range. Experience has shown that it is advantageous to consider these complex systems as a combination of interactive sub-systems (5). For example, models of a self synchronising clutch, a fluid coupling and a marine brake would combine to form part of a transmission system gearbox simulation. This building block approach enables validated models to be selected from a sub-system library and linked into the required system configuration for design or evaluation purposes.

It is intended, at the Evaluation Centre, that each major item of plant, a gas turbine, a diesel engine, a gearbox etc., is simulated to run real-time in a dedicated micro or mini computer while interactions with the remainder of the system are handled by data transfers through the master computer. The approach adopted has numerous advantages, some of which are:

(1) Each major item of plant can be modelled independent of the main simulation.

(2) Plant Control Units (P.C.Us.) can be interfaced directly with that part of the overall simulation which they are intended to control.

(3) For purposes of conceptual studies the machinery system can be rapidly reconfigured.

(4) Because models are written in a generalised form, most system variables are available as outputs.

Simulations have been developed for many items of plant which make up alternative arrangements of main propulsion systems. In addition, some auxiliary machines have been simulated while others are in the process of being modelled. Items for which simulations have been completed are:

Gas turbines: Olympus, Tyne and S.M.1A.
Propulsion diesel engine, turbo-blown.
Shaft and propeller inertia and friction effects.
Propulsion system gearboxes, including epicyclic.
Fluid couplings.
Self synchronising clutches.
Marine brakes.
C.P. propellers.
Propeller torque and thrust characteristics.
Hull resistance characteristics.
Chilled water plant.

The one major item not available as a real-time simulation is the diesel generator system. A diesel engine simulation is available but the complexity of the electrical distribution system is such as to prevent real-time simulation without over simplification. As an alternative, it is intended to install a mini-electrical generation system, typically 3 diesel generators (15-20 kVA) each driving into a representative electrical system load. While this compromise solution will not be truly representative of a full scale electrical distribution system, it will enable the overall control philosophies to be examined in the context of total ship machinery control.

PURPOSE OF AN EVALUATION CENTRE

The primary purpose in creating an Evaluation Centre is to provide the means to assess the performance of future control, surveillance, data transmission and information display options early in the design stage. The variety of tasks that can be undertaken, divide into three broad headings with numerous sub-divisions in development activities. The tasks are:

Development of Control, Surveillance and Display Systems.

This is considered the main function of the Centre. It is anticipated that during the development phase of a machinery system design cycle contributions will be made to the following areas:

(1) Examination of control and surveillance philosophies and operating strategies.

(2) Evaluation of processor hardware, software proving and processor/machinery interfaces.

(3) Procedure for local control of machinery (PCU control).

(4) Assessment of information display systems and Ship Control Centre lay-out.

(5) Evaluation of data highway configurations.

(6) Assessment of secondary surveillance and machinery health monitoring.

(7) Procedures for maintenance, servicing and fault diagnosis of processor based control and surveillance systems.

(8) Examination of operator reaction and training requirements.

Feasibility Studies of Machinery Systems

During post commissioning stage of the Evaluation Centre, feasibility studies will form only a small proportion of the work load. However, the facility will be ideally equipped to undertake studies of this nature once its primary development role has been established. The intended library of sub-systems will provide the flexibility required to examine various machinery configurations from both operational and control strategy standpoints.

Post Design Modifications and Proving Trials

It is foreseen that the Evaluation Centre will be used to develop trials procedures for 'first of class' proving trials and subsequent machinery acceptance trials. Use of simulations will enable procedures to be developed which optimize the use of trials time available. The facility should also prove to be a valuable aid to investigating in service machinery problems.

CONCLUSIONS

The need for a Machinery Control Evaluation Centre arose mainly from the planned change from analogue to digital technology and the intention to consider totally integrated control and surveillance schemes for the whole ship. Risks associated with this change in technology and control philosophy will be substantially reduced by an evaluation capability such as described. In addition, benefits will stem from a better understanding of the physical processes within the machines as a result of the mathematical modelling programme.

Most of the mathematical modelling programme has been completed, and implementation as real-time simulations on individual computers is in progress and scheduled for completion in mid 79. Evaluation of the first processor based control scheme is due to commence at the beginning of 1980.

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Any views expressed within the paper are those of the author and do not necessarily represent the final concerted opinion of the Establishment or the Director General Ships.

TRAINING SIMULATORS FOR SHIP PROPULSION PLANTS

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1. SUMMARY

On the basis of the computerized marine automation system DataChief, a training simulator for ship propulsion plants is developed. The simulator is designed for marine training colleges.

A complete propulsion plant training simulator consists of an "engine control room" with realistic modern plant monitoring and control equipment, the "machinery space" with individual panels representing the main machinery components, and a "bridge" with main engine remote control and communication facilities for the instructor. The simulators are designed to give the trainee better understanding of a modern propulsion plant, and to provide training of correct actions under normal and abnormal/emergency situations.

The simulators are based on medium-size digital computers. The dynamic models describing the process under consideration can be interconnected in many different ways and can be adjusted to specific plant requirements.

The main features of the simulator are:

- remote control of propulsion plant,
- realistic overall plant alarm monitoring including logging of alarms and recording of process variables,

- remote/local control of all main pumps,
- manual or automatic synchronizing and load-sharing of generators,
- flexible system for changing/resetting of process state and introduction of malfunctions,
- simulation of component overhaul by fault resetting at local panels.

The simulator is developed in two versions, diesel- and steam turbine plant simulator. Due consideration to the differences between such plants are taken, both in simulator arrangement, simulation methods and functions.

The first installation took place in spring 1978.

2. INTRODUCTION

The simulator has been designed to meet the following functional requirements in particular:

(A) Operator training

Help giving the trainee better understanding of a modern propulsion plant, especially when malfunctions occur.

Training of correct actions under normal and unnormal/emergency situations.

(B) Investigation of system performance

Basic control systems studies.
Basic plant performance studies.
Training in use of modern instrumentation.
Training in process fault finding.
Simulating system performance when different system components are introduced.

In order to achieve this, the control room is equipped with a complete modern system for UMS-operation, including alarm system, remote control of propulsion plant, alarm logging, condition monitoring system, remote control of all main pumps and automation of electric power generations.

This system is controlled by a digital computer, which is also used for process modelling. Thereby it also controls the local panels in the engine room and the bridge control system with instructor facilities.

But the main elements in the training simulator is a set of dynamic models representing the process under consideration. The models can be interconnected in many different ways and be adjusted to specific plant requirements.

3. GENERAL ARRANGEMENT OF THE SYSTEM

The general arrangement of the system is shown on fig. 1. Basically it consists of three separate rooms.

- Control room.
- "Bridge"/instructor's room.
- Engine rooms.

In the control room, both the main switch board and the centralized control system are located.

The main switchboard is represented by the generator synchronizing equipment, allowing realistic electric power system simulation.

The control system is divided into two sections:

- alarm section,
- remote control system (AutoChief II).

The system's digital computer and I/O electronics, paper tape reader punch and floppy disc are located in the alarm section.

Separated from the control room by a "window wall" and removable door section, is the instructor's room. Here the bridge control console is placed together with the main communication equipment:

1. Teletype.
2. Acoustic sound equipment.

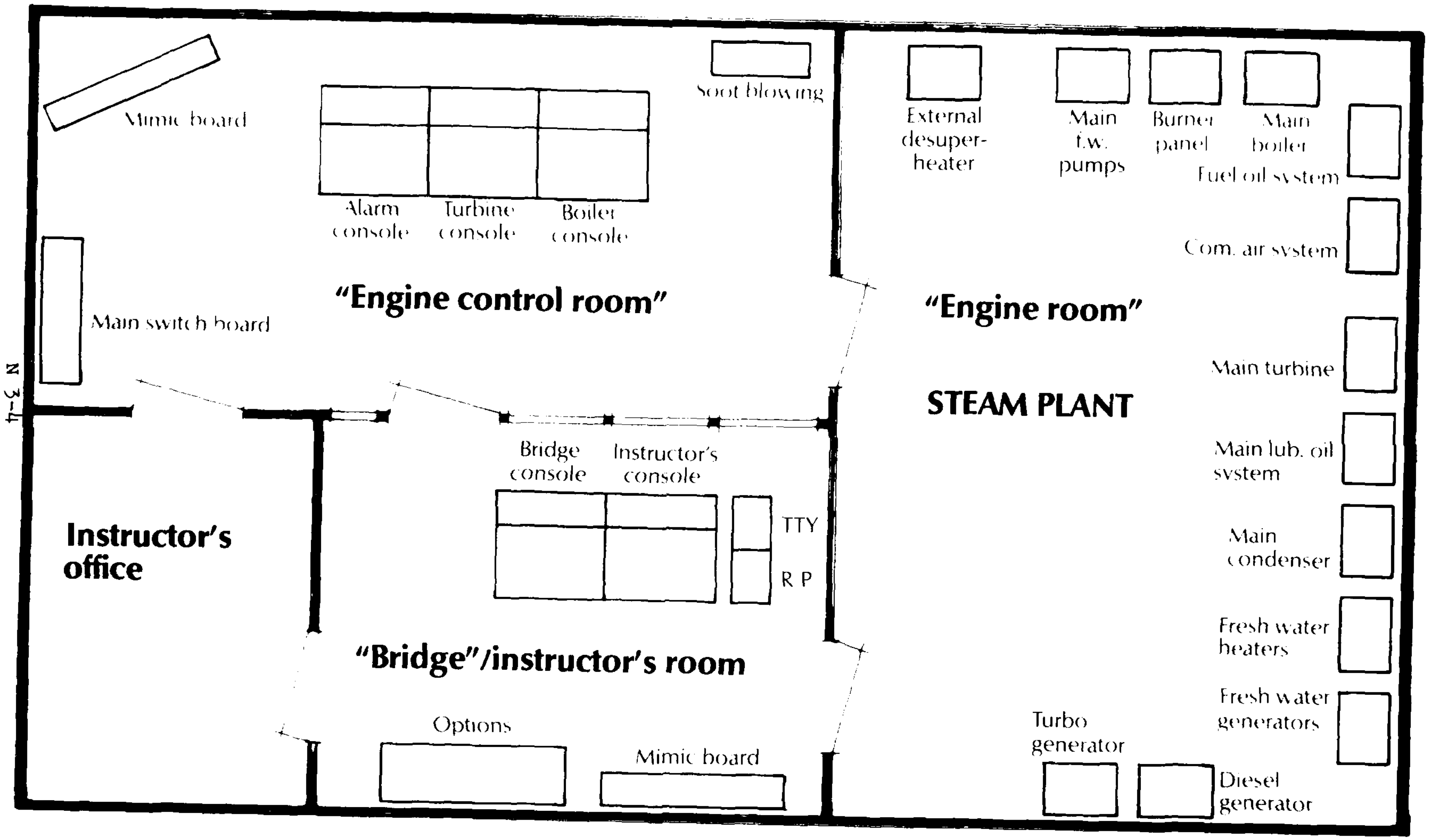

Fig. 1 SIMULATOR ARRANGEMENT

In a room corresponding to the local engine room, all simulated components are represented by "local boxes" where indicators and control functions normally placed locally on a real ship are found.

Equipment for resetting of trip and for simulating repair of malfunctioning components are also available on the local panels.

The trainee(s) are normally confined to the control room, but will take frequent visits to the engine room to check the state of the controlled component, reset malfunctions, practice in manual operation of control valves etc.

The instructor can watch the trainee's action through the glas wall or the open door. He will frequently be present in the control room and the engine room for guidance and direct observation.

Mimic panels are placed in the control room and the engine room to give new instructors and trainees fast introduction to process details.

4. MATHEMATICAL MODELS

Mathematical models of both steam and diesel engine plants are developed. These models are based on physical conseptions of the behaviour and characteristics of the plant, and is modular so that future modifications and/or extensions can be made.

4.1 Diesel engine plant

A block scheme of a software simulated diesel plant is shown on fig. 2.

The main features are:

- Slow speed, single acting, large bore diesel engine with turbochargers.
- High/low temperature fresh water cooling system with central cooler.

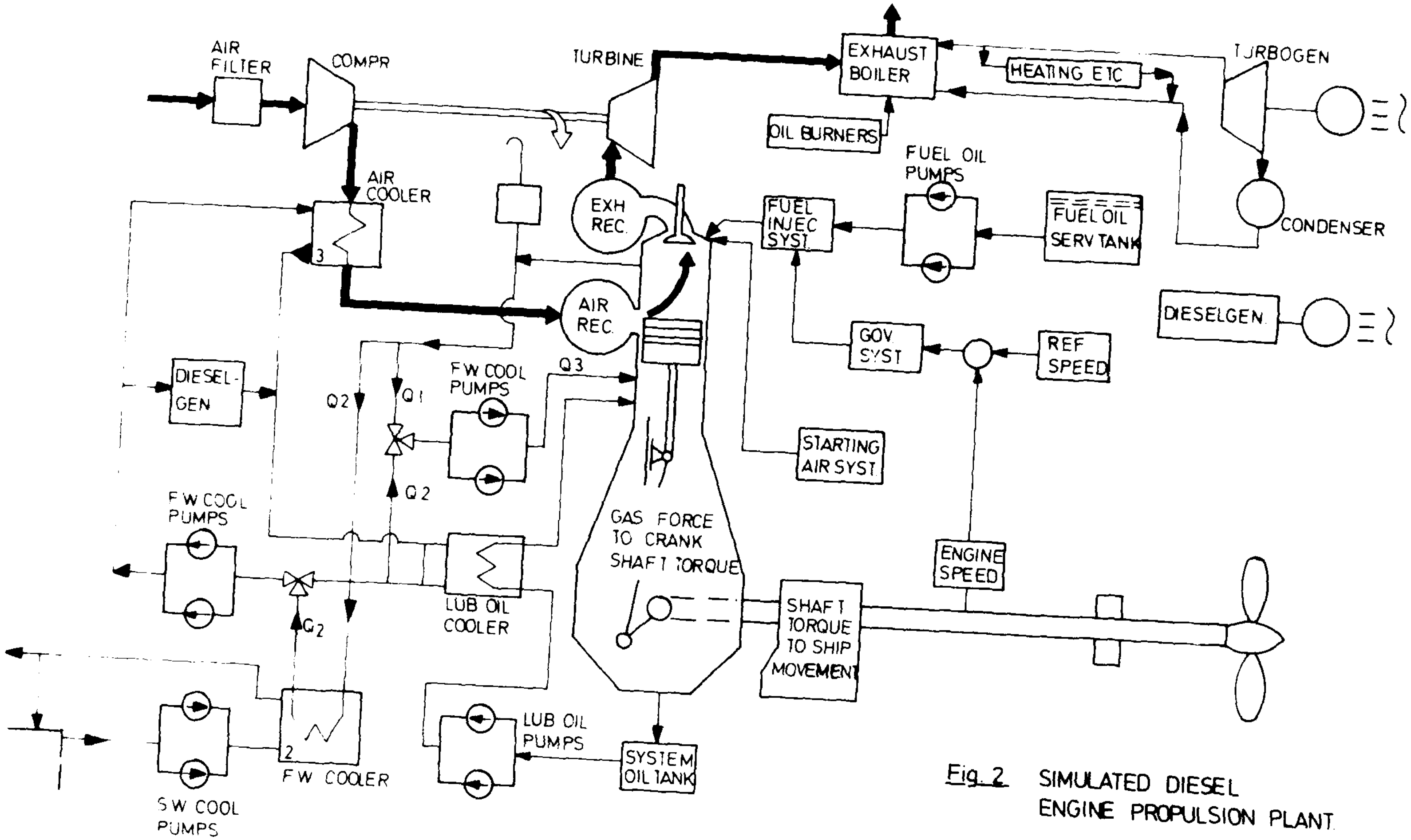

Fig. 2 SIMULATED DIESEL ENGINE PROPULSION PLANT

- Sea water as coolant to 2 fresh water coolers.
- 1 turbogenerator driven from exhaust fired boiler and/or oil fired boiler.
- One diesel engine driven generator.
- C.P.-propeller.
- Type of m.e. fuel; diesel or heavy oil.

4.1.1 Engine Model

This model is schematically shown on fig. 3. As one can see, this model is working together with the electronic bridge control system AutoChief II and a program calculting the shaft moment.

In the model, the RPM is calculated due to fuel pump setting and propeller moment. This RPM is used by the rest of the model, which generates a new moment and pump setting. In this way, adjustment will continously be done until equality is reached.

Propeller pitch controller position is input to a servo model, which generates the actual pitch.

The system is equipped with a noise generator, simulating the effect from sea and wind.

4.1.2 Cylinder Model

The cylinder model consists of a general cylinder model working together with the model of air/exhaust system. The model is taking individual input from each cylinder, and errors may be set by the instructor. All errors must be addressed to a special cylinder (example leak exhaust valve cyl. 5).

Input to the model is generated according to the actual running condition, and all the parameters will have a dynamic response when running condition changes.

If an error is initiated in the cylinder, this will influence on other parameters as realistic as possible.

The instructor may initiate max. 15 errors for each cylinder, and these have to be reset by the students. Reset is done by push-buttons in the engine room. There is one push-button for each error.

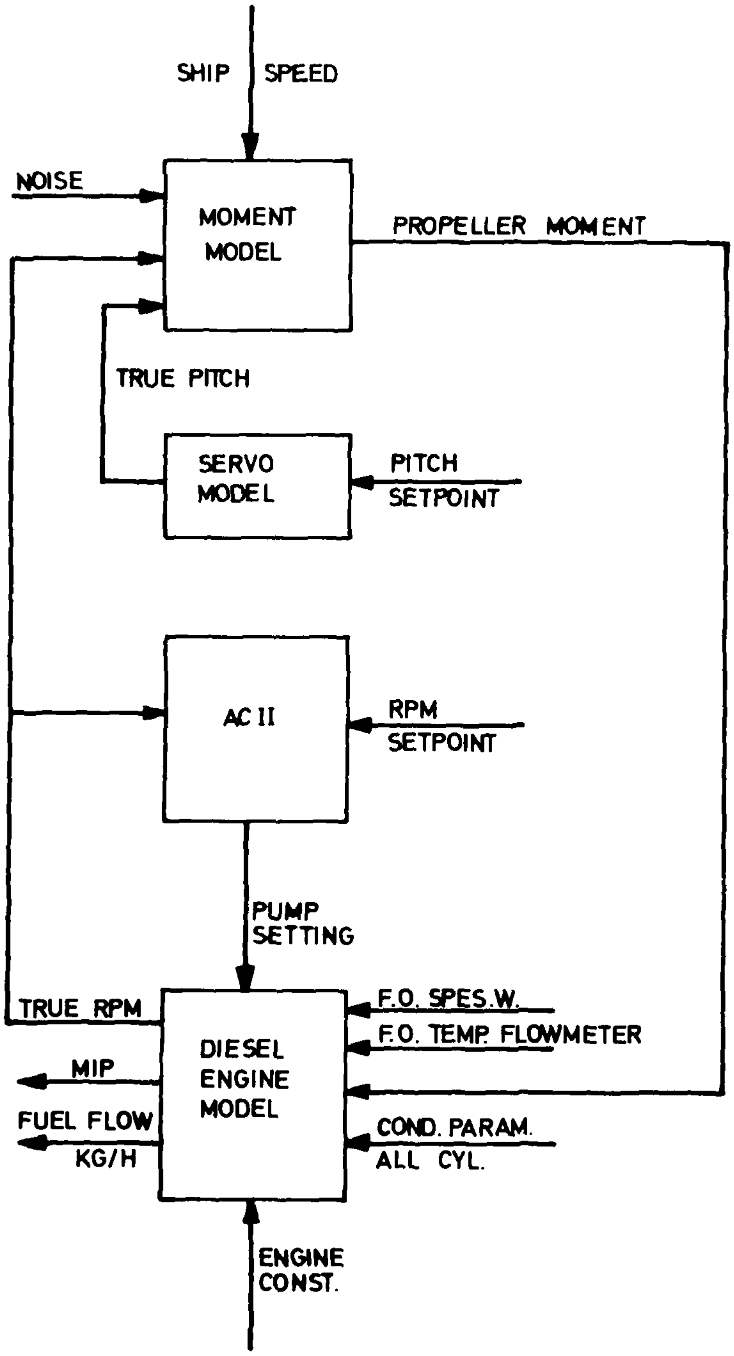

Fig. 3
SIMPLIFIED MODEL OF THE MAIN ENGINE MODEL.

4.1.3 Boiler System

The boiler system model comprizes models of:

(1) Exhaust boiler.

(2) Auxiliary boiler.

The system employs steam separation in a common steam drum and circulation pumps for forced water circulation between the auxiliary boiler's water system and the exhaust boiler. A superheater tank is placed in the gas duct after the exhaust boiler.

The water level in the steam drum is controlled by a common feed water flow control valve. This control is always active, also when only the exhaust boiler is in operation.

The auxiliary boiler is equipped with two oilfired burners. A simple burner management system is included.

The combustion control system for control of air flow and oil flow is modelled.

The boiler safety system is included, giving trip of auxiliary boiler at loss of furnace flame and emergency low water level and shut off of feed water supply at emergency high water level.

4.1.4 Steam System

The steam system is composed of the following sub-components:

(1) Turbo generator turbine.

(2) Vacuum condenser.

(3) Steam dump system.

(4) External steam load.

The model of the turbine driver includes speed controller and will, together with the generator model represent the turbo generator with sufficient realism.

The model of the vacuum condenser is very simplifed, but includes the main effects of the vacuum pump operation and the influence of the cooling sea water flow and sea water temperature.

The steam dump system is represented by a pressure controlled steam dump valve, dumping steam to the condenser at high steam pressure.

External steam loads are:

(1) Accommodation steam.

(2) Steam for fuel oil heating.

(3) Steam to deck machinery.

(4) Steam to tank cleaning etc.

These steam loads are to be set by the instructor.

4.1.5 Pump Model

The pump model is designed to calculate flow and pressure in the fluid after the pumps.

The calculation is based upon separate pumps characteristic for the different pumps and the back pressure in the actual flow line. The back pressure (pressure drop) is calculated in the actual system model.

Power consumption, depending on running point, is also calculated. This is used by the electric power model.

The instructor may introduce faults on start signal, stop signal and main contractor, and these faults have to be reset in the engine room.

4.1.6 Cooler Model

This is a dynamic model, calculating the K-value and the output temperature on the two fluids.

It is taken care of changes in heat transfer due to changes in fluid velocity.

Fluid velocity will change due to many factors like:

- By pass.
- Changed resistance in cooler.
- Variations in viscosity.

The instructor may introduce reduced efficiency on the cooler, and when detected, this may be reset by pushing the reset push-button in the engine room.

AD-A159 084 PROCEEDINGS OF THE SHIP CONTROL SYSTEMS SYMPOSIUM (5TH) HELD AT U S NAVAL..(U) DAVID W TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER ANN.. D E LABBE ET AL. 2 4

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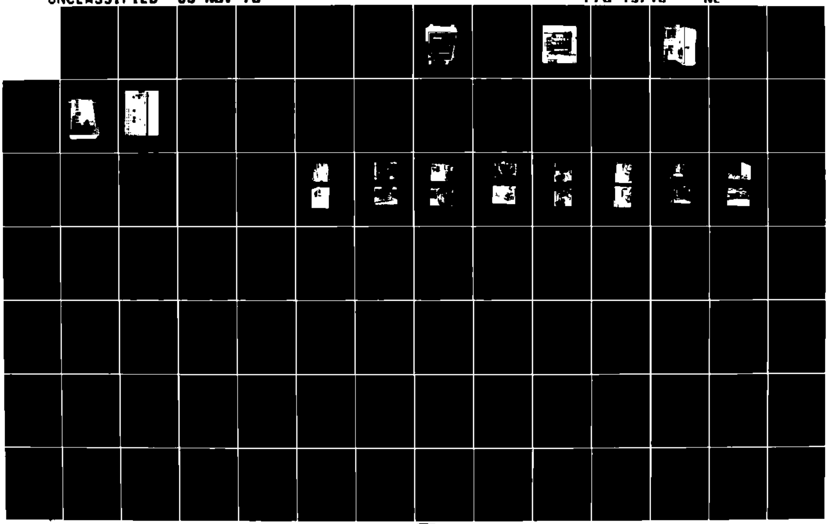

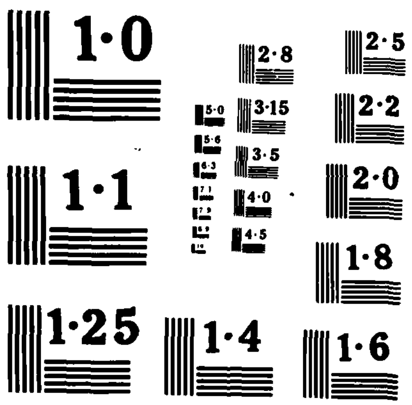
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4.1.7 Temp Control Model

This model is designed to control the fluid temperatures in the system. There are three controllers in the system which are controlling temperature:

- Oil temperature inlet main engine.
- Fresh water temperature outlet main engine (HTFWS).
- Fresh water temperature inlet air coolers and dieselgenerator (LFTWS).

Common to these controllers is that if fault is introduced, the control must be manual. This is done by switching to local on the panel, and use the potensiometer and the temperature instrument.

4.1.3 Process Noise Simulation System

To give the simulation a realistic appearance it is important that certain of the variables are "animated" by process noise.

The simulator software includes a white noise generator and a set of digital band pass filters for generation of coloured noise spectra.

It will suffice for most purposes to add noise to one key variable, namely propeller torque. The noise on the propeller torque represents the disturbances introduced by the wave movements.

The propeller torque noise will disturbe the diesel remote control and thus spred to the rest of the models.

The instructor does not have to care about noise spectra and effective amplitudes, but can by a special "sea state" command and a number from 0 to 10 specify the wind and sea state according to Beauforts scale.

Noise can be added directly to most process variables by the instructors from the teletype if special effects are to be demonstrated. Malfunction of a transmitter resulting in excessive noisy signal is one example.

If the purpose is to study a control system, it is convenient to be able to exaggregate the noise on the measured and controlled variables and thus more clearly demonstrate the effects of the parameter setting of the control system under study.

4.1.9 Engine Room Sound Simulation System

A major problem in engine room simulation is that of creating realistic sound enviromental conditions.

A sound generation system produce a reasonable realistic audible sound picture in the control room and the engine room.

The system is computer controlled and will mix characteristic sound-pictures from the various types of equipment.

The sound-system has been delivered with very good results and is highly recommended.

4.2 Steam Turbine Plant

The software simulated steam turbine plant comprises of:

- Main and aux. boiler.
- Main fuel oil pumps.
- Main turbine.
- Main condenser.
- Feed water heaters.
- Main boiler feed water pumps.
- Turbo generator/diesel generator.

4.2.1 Main Boiler

The boiler is modelled as an accurate unlinear dynamic model of approximately 30th order.

The rotary air preheater is modelled, including influence of bypass damper position on air and gas side and rotor stop. The operation of the dampers may be automatic or manual.

The fuel oil heating unit and control system is included.

The burner actuators are modelled in great detail included flame scanners.

4.2.2 Main Turbine

The main turbine model gives an accurate description of the HP, LP and astern turbine.

Shaft torque, shaft power and RPM are computed. The steam pressure/temperature is computed at the various extraction points. The state of the steam leaving the LP turbine (enthalpy, water content) is given.

The extraction valves may be operated automatically or manually. The hydraulics of the turbine control actuator is simulated to give a realistic response.

The propeller dynamics is naturally included. So is also the translatory movement of the ship.

4.2.3 Main Condenser

The condenser is simulated so that proper vacuum, hot well level, condensate temperature, sea water flow, sea water diff. pressure and sea water outlet temperature is represented for all loads.

Models of the scoop, sea water circulation pump and condensate pumps are present. The effect of air leakage and the vacuum pump operation is included.

Simulation of condenser cleaning and plugging of condenser tubes is possible.

The condensate pump is modelled to the extent of showing the effect of caviation due to low hot well level.

The scoop operation is automatic or manual.

4.2.4 LP Heaters and Deaerator

Two LP heaters are included in the software simulator system. This makes it possible to obtain a realistic simulation of the condensate temperature entering the deaerator and an accurate computation of the LP turbine extraction flows.

The deaerator is simulated so that proper feed water temperature and suction pressure for the feed water pumps are achieved througout the load range. It is possible to show plant operation in an emergency situation without the deaerator in operation.

4.2.5 HP Feed Water Heaters

The HP feed water heaters are simulated to give, in conjunction with the main turbine model, accurate computations of the extraction steam flow from the HP turbine and to give the correct feed water outlet temperature at various loads.

The water levels are controlled by simulated proportional level controllers. Variation in level and heat transfer coefficients may be introduced and its effect demonstrated.

4.2.6 Main Feed Water Pumps

Both feed pumps are simulated so that actual pump performance is represented for the entire load range. Automatic recirculation to deaerator at low load is provided for.

The pump turbines are simulated to give the correct dynamic relation between throttle valve position/ inlet steam condition and pump RPM, f.w. pressure and water flow.

The steam flow to the pump turbines and the exhaust steam state is modelled accurately.

5. ENGINE ROOM EQUIPMENT

The situation for the "man onboard" is having the different engine room equipment located all over the engine room. To simulate this situation, different local engine room control panels are supplied, one for each main component, instead of one common "Local Engine Control Console".

For the plant, the local panels for each of the following engine room systems/components are provided:

Diesel engine plant:

1. Main engine, including air coolers and turbochargers.
2. Diesel generator.

3. Turbo generator.
4. Boiler system.
5. Fuel oil treatment.
6. Temperature control system.
7. Sea water pumps.
8. Fresh water pumps, -LTFWS.
9. Fresh water pumps, -HTFWS.
10. Lubricating oil pumps.
11. Fuel oil booster pumps.
12. Servo oil pumps. (Option).
13. Compressors, - including start air compressors and instrument air compressor.
14. Indication panel.

Steam turbine plant:

1. Main and aux. boiler.
2. Burner managment.
3. Main fuel oil pumps.
4. Main turbine.
5. Main condenser.
6. Feed water heaters.
7. Main boiler feed water pumps.
8. Turbo generator.

As examples of these local panels the main engine panel (fig. 4) and the boiler system panel (fig. 5) relevant to the diesel engine plant are described.

5.1 Main Engine Panel

The panel have 15 reset push-buttons, each representing one type of fault. A push on one of these buttons will simulate that an overhaul/repair have been made.

In addition, the panel have provision for adjustment of the fuel rack position for each cylinder.

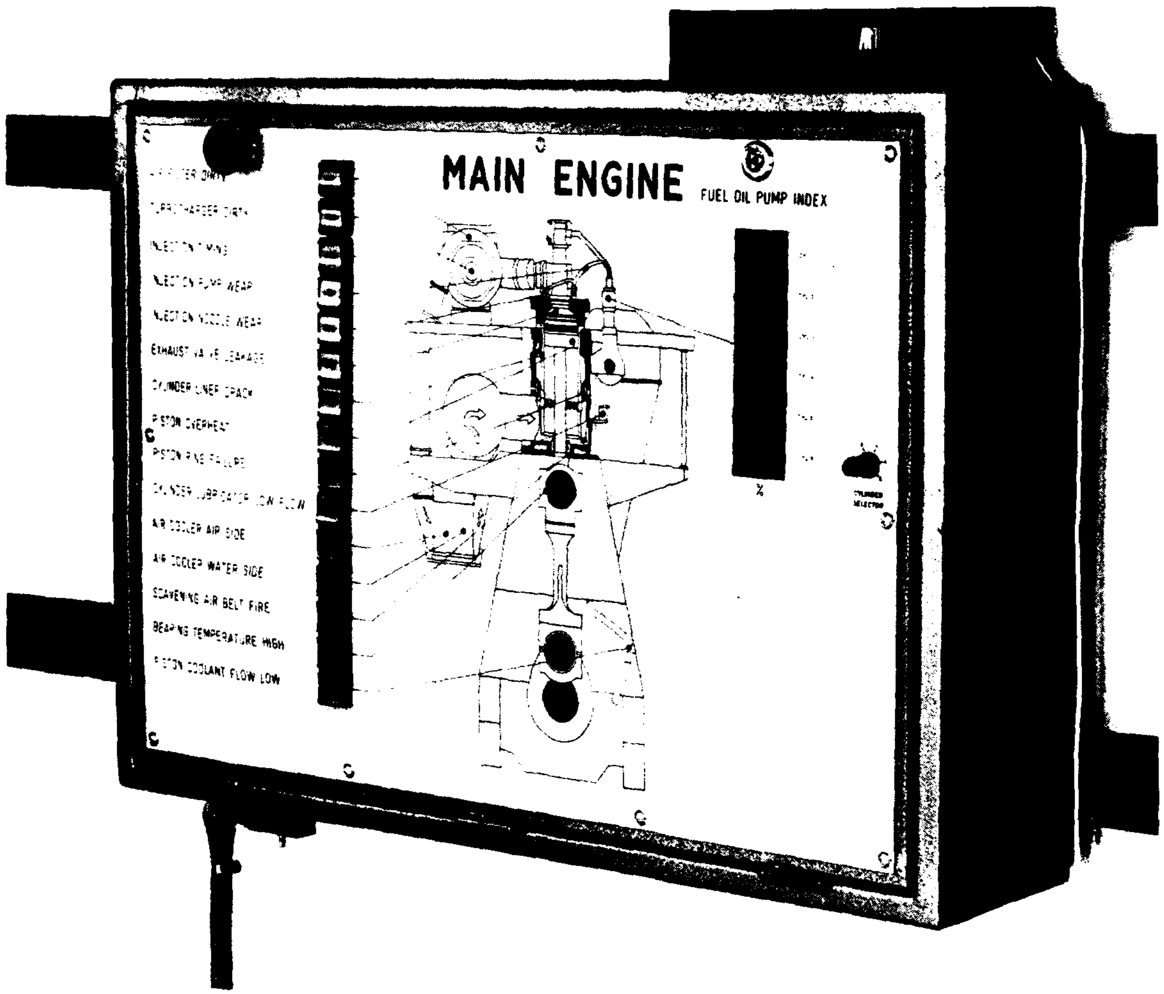


Fig. 4 LOCAL PANEL MAIN ENGINE MIMIC.

5.2 Boiler System Panel

The auxiliary boiler and the exhaust boiler are represented on a common panel.

The boiler's control valves can be operated manually from the local panel. The "remote - local" switches represent three-way valves mounted in the control air line after the I/P converters. In position "remote" the control signal is taken from the I/P converter (signal from controller model.). In position "local" the control signal is taken from the manually operated air control valve, represented by the potensiometer on the panel.

By means of the drum pressure and drum level indicator, and the control potensiometer, it is possible to simulate emergency operation of the boiler from local boiler stand.

The auxiliary boiler can be tripped from the panel.

Start/stop push-button/lights for feed water pump, combustion air fan, fuel oil pump and water circulation pump are provided.

The two burners of the oil fired auxiliary boiler can be started/stopped from the panel.

Flame out results in trip of auxiliary boiler. Repair of faulty control valves (open/closed/sticking) is simulated by transferring to local manual control and keeping local control for a minimum period of time.

Press on the "reset controller" push-button simulates repair of steam pressure master controller.

Press on the "reset exhaust boiler" push-button simulates major cleaning of exhaust boiler. (Soot blowing and washing.)

Faults in any of the pumps/fan will be reset when the pump/fan is restarted from the local panel.

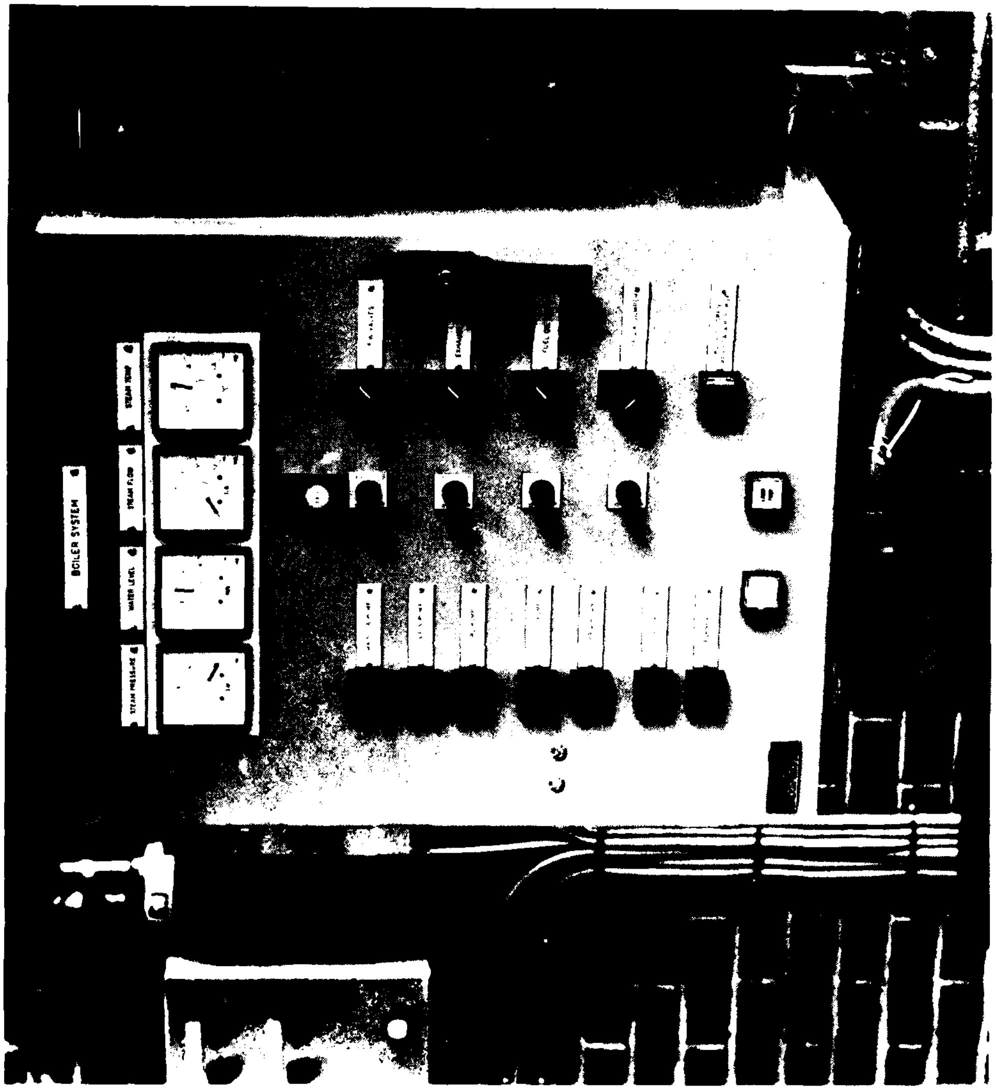


Fig. 5 LOCAL PANEL - BOILER CONTROL.

6. CONTROL ROOM EQUIPMENT

The control room comprises of the control console, a teletype and the electric switchboard.

6.1 Control Console

The control console for the steam turbine plant consist of sections for:

- Alarm system.
- Monitoring system.
- Pump system.
- Turbine remote control.
- Boiler control.
- Burner management.

This system is closely similar to the computerized control system (DATACHIEF/TURBINE) delivered to more ships.

The control console for the diesel engine plant is shown on fig. 6. This system is based upon the computerized control system DataChief III and incorporate the following features:

- Engine room unit of the bridge control system.
- Group alarm panel with display and back-up alarm unit.
- Condition monitoring.
- Pump system.
- Automation of electric power generation.
- Alarm logging.

Both the steam turbine plant control system and the diesel plant control system are standard products and are therefore not discussed in detail here. (See references 1, 2, 3 and 4.)

Apart from modern compact design of both the lay-out and functioning of the control systems a main feature of the systems is condition monitoring. This implies that a number of measurements, which is beyond the demands of the classification societies, UMS rules is made available.

FIG. 6 CONTROL ROOM CONSOLE WITH ALARM-, INSTRUMENTATION AND REMOTE CONTROL PANELS FOR MAIN ENGINE AND S.E. MACHINERY.

For the diesel plant system, the condition monitoring system employ

- indication of both cylinder and fuel oil injection pressures. (See fig. 7.)
- piston ring monitoring.
- thermal load of cylinders.
- monitoring of turbochargers and air coolers.
- Trend monitoring.

The important aspects of this particular system are to give the students a better understanding of fundamental operational problems and to give to them a first impression of a kind of system which will be more commonplace on future ships.

Particularily the trend monitoring is an unusual feature. By this method, the effect of gradual fouling of components can be studied. This is a help for prediction of overhaul. On the simulator this is achieved by special traning programs which contain the history of the plant parameters. Corrective actions (overhauls/adjustments) can then be trained.

6.2 Electric Switchboard

By use of this control panel, see fig. 8, the trainee can stop and start the stand-by generator, syncronize and connect (disconnect) the generator to the "line".

Syncronizing of the stand-by generator is done by increasing/decreasing (RAISE/LOWER) the engine (turbo) speed. Speed variation is observed on the frequency-indicator (Hz). The phase difference is indicated by rotating light-emitting diodes (syncronoscope). The rotation direction and speed will change depending on the speed difference between the two generators.

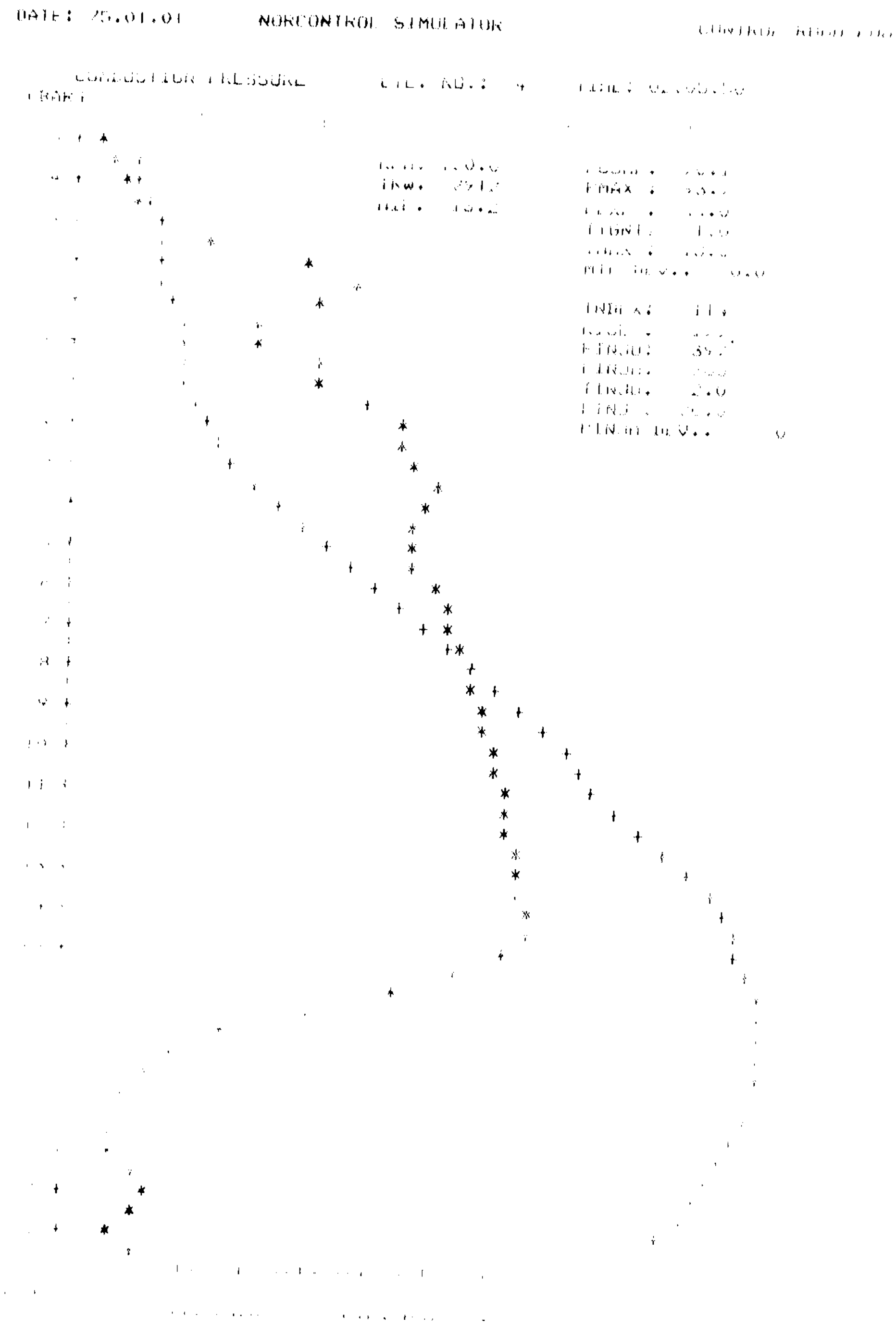


Fig. 7

Cylinder and injection pressure diagrams.

Independent Mode - provides separate and independent operation of all rooms where the data generated in one room has no impact on the other rooms. This mode can provide simultaneous part-task training for all rooms.

Accelerated Mode - enables the instructor to move from one training situation to another without being inhibited by actual plant transition times.

Initial Conditions

Cold Iron - dockside with shore steam and shore electrical power.

Auxiliary Steaming - dockside, one boiler, no engine, two ship service turbogenerators in service.

Underway 1 - one boiler under automatic control, 15 knots, two ship service turbogenerators in parallel service, diesel generator lined up for automatic start.

Underway 2 - two boilers under automatic control, 25 knots, two ship service turbogenerators in parallel service, diesel generator lined up for automatic start.

Device 19E22 has capability for expansion to provide:

- Split-plant training for 4 and 8 boiler systems
- 600 PSI high-value systems
- Enlisted personnel training.

The Math Model

To provide these features, the 1200 PSI Propulsion Plant system's mechanical and electrical components must be modeled in mathematical form, the math model.

A math model is simply a representation of a specific item in mathematical notation. For Device 19E22, the model, whether defined by algebraic or differential equations, must be in a form suitable for solution via digital computation techniques in both real-time and faster than real-time situations. In addition, the overall integrated model must be separable into three independent engineering space models as defined by the shipboard bulkheads and isolation valves.

The accomplishment of these and other specified requirements, necessitates a comprehensive modeling approach wherein the identification of the physical parameters upon which the model is based are consistently maintained within and between models. The basic parameters of interest are those concerned with fluid flow, mass and heat transfer, kinetics, and dynamic response and control characteristics. These include pressures, temperatures, densities, volumetric and mass flow rates, heat transfer characteristics, inertia effects, and appropriate open and closed loop transfer functions. Additionally, the modeling task is complicated by the many input parameters, both continuous and discrete, which control and otherwise affect system performance. Consequently, the many output parameters displayed must be computed to yield realistic values over the trainer operating range. Therefore, the availability of valid data defining component and system operating characteristics is of prime importance. In lieu of complete definition electrical central. Four initial conditions are available for immediate recall by the instructor. All instrumentation and certain preselected valves automatically assume the proper setting for the desired initial condition upon command by the instructor.

A 24-bit general-purpose computer handles computation and execution routines necessary for all real-time simulations. A basic iteration rate of two per second has been determined to be adequate based upon observation of propulsion plant dynamic characteristics and instrument variations. Various peripherals such as a magnetic tape unit, disc memory, and teletypewriter with page printer, papertape reader, and papertape punch provide input/output capabilities. A FORTRAN compiler is included to provide efficient program modification and maintenance.

Device 19E22 is capable of operating in three modes: Integrated, independent, or accelerated. It is possible for any training problem to be halted, or frozen, to permit discussion of procedures. Release of the freeze condition permits the training exercise to continue in the normal manner. During the freeze condition all indications will remain in the positions or conditions that existed at the moment of freeze initiation.

Casualty simulation controls are grouped at the instructor station as related malfunctions. The math model detects the positions of these controls and causes certain effects in various parts of the simulator. Some casualty simulations are discrete; e.g., fully operational or total failure, and some are variable, such as partial loss of vacuum to any of a very large number of valves. With this feature, instruction can be provided for any number of casualty combinations and degrees of failure. Several causes can initiate a casualty. Under normal trainer operation, where all compartments operate in an integrated manner, casualties result automatically in accordance with the propulsion plant math model. The casualty controls at the instructor station serve to initiate problems in a convenient manner. For example, loss of fuel oil pressure can be initiated immediately instead of waiting for the math model to determine when the fuel oil service tank becomes low or empty. These casualty controls are also useful to provide training when the trainer is operated in the independent mode where the fire room, engine room, and power-distribution room operate separately.

By manipulating the instructor's station controls, the instructor is able to select training situations from the simplest to the most complex. This capability allows trainees to respond to increasingly difficult training problems as their proficiency increases. Not only can different situations of varying difficulty be created, but the complexity of some of these problems is controllable; e.g., the effects of water in the fuel oil could be controlled to create subtle or catastrophic casualties. Sufficient disc storage capacity is provided to permit real or time storage of parameters required for real-time playback. This includes a 30-minute moving window of real-time data, of which the recording and playback is under the control of the instructor.

Trainer Modes

Integrated Mode - provides simultaneous and interactive operations of the engineering spaces on a common training problem where all data generated in one room affects responses in other rooms. This mode provides total propulsion plant training.

sits in the cockpit; for radar operators, the interface is at his console; and for team systems such as combat systems, each operator is situated at a console or plotting board. So, how do you train a team to operate a system that is located in five spaces where about 800 valves are located overhead, under deck plates, and on several levels? In several cases, where muscles are needed to move them! Machinery noises, approaching the threshold of pain, are important clues to system operations. How do you create this environment? Must appearance fidelity be 100 percent or can cost-saving trade-offs be made? How can instructors control the conditions of the simulated machinery located in almost 200,000 cubic feet from a central location? Can a way be developed to change the positions of 800 valves remotely (by computer control) to save problem setup time? How do you simulate two boilers which are each about the same size as a small two-story house? How can the maze of piping be simulated? Mechanical problems were the rule in conceiving this trainer. The real-time computer techniques for trainers were established so this provided no new anticipated problems other than deriving and coding a math model for the 1200 PSI Propulsion Plant.

Several fact-finding trips were made to Newport to learn enough about the 1200 PSI system in order to prepare a specification for Device 19E22. The result was a trainer having the following salient features:

The 1200 PSI Propulsion Plant Trainer provides realistic real-time simulation of the fire room, engine room, auxiliary machinery rooms and electrical central of the DE-1078, a destroyer having a two-boiler 1200 PSI, single-screw propulsion system. Equipment and systems are functionally simulated to train personnel in start-up, normal operation, shutdown, and casualty control of the propulsion plant.

Device 19E22 generates all visual and aural responses necessary to completely simulate operational conditions to provide personnel with a hands-on environment without the necessity for utilizing operational equipment. To accomplish this, Device 19E22 consists of full-scale simulated versions of equipments found in the fire room, engine room, auxiliary machine rooms, and electrical central of a DE-1078 class ship. Those equipments directly related to steam generation, propulsion, and the electrical generation and distribution system, are simulated and operable. All equipments are arranged to duplicate actual physical layout aboard a DE-1078 class ship. Included are all controls, indicators, simulated equipments, pipes, and casings necessary to represent the operational system. The capabilities and characteristics of the real system, as manifested by changes in gauge readings of water level, steam temperature and pressure, RPM, voltage, and current are simulated by use of digital computer, electronics, and electromechanical means. All sound and visual effects experienced during actual plant operation are also simulated.

An instructor's console is provided to set up initial conditions; introduce malfunctions; and monitor system parameters, student actions, and responses. Plant operation can proceed in real time from any of four initial conditions, or the problem may move rapidly in an accelerated mode to any desired plant condition. At any time during real-time operation the instructors may induce equipment failures, and procedural casualties will result from any operator errors. The trainer may operate as an integrated system or as three independent problems - one each in the fire room, engine room, and auxiliary rooms/

Trainer, Device 19E22, was installed at the Surface Warfare Officers School in Newport, Rhode Island. For the first time, hands-on training is being provided for the engineering spaces of a 1200 PSI ship - training consisting of normal and casualty procedures. Not only can this training be provided on an around-the-clock basis to increase the number of much-needed qualified personnel, but it can be done without consuming fuel and in complete safety.

Device 19E22 consists of life-size mockups of the fire room, engine room, auxiliary rooms 1 and 2, and the electrical central of the DE-1078 destroyer escort. The machinery within these rooms is, in most instances, working replications of the corresponding machinery aboard ship. Gauges and smoke indicators are simulated, machinery sounds are duplicated with operational intensity and fidelity, shafts turn, and valves function. Added to these normal features, casualty indications such as boiler flareback, tube rupture and high-water carryover extend training to a degree never considered before in the steam propulsion community. Training, similar to that which has been available to naval air and submarine personnel, is finally available to those providing the surface engineering skills.

This training capability does not come too soon. Almost 200 ships, including destroyers and carriers utilize 1200 PSI steam systems. The Navy's need for converting from 600 PSI to 1200 PSI systems stemmed from a requirement for more room aboard ship to accommodate more sophisticated weapons systems without increasing ship size. The 1200 PSI systems produce higher temperatures, pressures and steam generation rates with smaller water volumes. These plants are more complex, more demanding, and less forgiving than the 600 PSI systems. Consequently, more qualified people are needed to operate them to minimize, if not avoid, casualties similar to those that have killed naval personnel and disabled ships in the past. A need to qualify personnel in the total propulsion system was essential.

A first attempt at this training without dedicating ships was installation of a 1200 PSI Hot Plant at Great Lakes. This trainer, using operational equipment, is housed within a large building and approximates the layout within the DE-1078 class engineering spaces. It can provide normal and maintenance training but not without mutual interference. Casualty training, in the hands-on approach, cannot be provided - who would deliverately induce a high-water casualty and destroy the turbine if the trainees were not quick enough to correct the situation - or low-water that would destroy a boiler? Not only would personnel be injured (or killed) but necessary training is curtailed for an extended period. Furthermore, the extremely high initial cost for the Hot Plant, which is estimated to be nearly $40 million, and the high cost of fuel limits the Hot Plant approach as an effective training system.

Consequently, the naval training community approached the Naval Training Equipment Center, in May 1973, for a more effective training system. The solution evolved into Device 19E22, a fully simulated trainer.

PROPULSION PLANT TRAINING - THE SIMULATION APPROACH

Conceptual details for Device 19E22 piqued the imagination. Trainers are used to develop competence for man in the man-machine interface. For a pilot, the man-machine interface surrounds him as he

REAL-TIME SIMULATION OF A STEAM PROPULSION PLANT

by Cecil N. Goff
Naval Training Equipment Center

ABSTRACT

A ship propulsion training device has been developed which combines full-scale physical replication with math modeling and digital computer simulation. This trainer, Device 19E22, 1200 PSI Propulsion Plant Trainer, is now operating at the Surface Warfare Officers School, Newport, Rhode Island. The development of this large trainer resulted in several first-time achievements: An integral math model of a complete ship's steam propulsion plant and auxiliaries with real-time digital computation; electronically generated and computer controlled machinery sounds including normal and casualty operating conditions; and full-scale mockups of fire room, engine room, auxiliary spaces and electrical central with functioning man-machine interface instrumentation and controls. With these features this trainer is able to simulate plant conditions where casualty training can be provided to a degree never before available in the Navy's surface propulsion plant training.

Generic math models were developed for pumps, turbines, heat exchangers, etc., and design data for each component was then incorporated with the generic model to develop specific models. The automatic combustion control system was modeled as well as the ship's electrical generators and power distribution system. Programming of the math models required approximately 50,000 instructions. The basic iteration rate is two per second by a 24-bit machine with 64K words of core memory and 750K words of disc memory.

In addition to providing a new dimension in training, this device has potential as an engineering tool to evaluate changes in plant components and operating procedures.

INTRODUCTION

As part of an overall goal of accommodating larger and more sophisticated electronics and weapons systems aboard ship without increasing ship's size, the 1200 PSI Steam Propulsion Plant was developed in the early 1950's. Designed to be a smaller, more efficient, and lighter machinery plant, the 1200 PSI made its first appearance in the fleet in 1954 with the commissioning of the USS Norfolk (DG-1). A revolutionary change in the philosophy of engineering plant design, operation, and the arrangement of the equipments pushed design parameters to the limits of the known state of the art. For the first time, U. S. Navy vessels were equipped with automatic feedwater and combustion control systems. A new concept of operation was brought to the engine room and fire room community, and subsequently the requirement for development of an extensive training program in the areas of operation, maintenance, and management for the new plant became apparent.

A major milestone was attained for the United States Navy's surface training program in January 1978, when the 1200 PSI Propulsion Plant

The key factors behind this trend is

- safety
- economics of operation

It is expected that the simulators described are able to improve the training situation very much, due to the real life character of the systems.

11. REFERENCES

1. S. Espestøyl: "Computerized Automation and Condition Monitoring" IFAC/IFIP second international symposium, Washington 1976.

2. S. Espestøyl, O.M. Sivertsen: "General Hardware and Software for Engine Room Monitoring and Control Systems", IFAC/IFIP symposium, Oslo 1973.

3. A. Andreassen, S.K. Johnsen: "Computerized Systems in Diesel Engine Rooms, Installation Testing and Operation", IFAC/IFIP symposium, Oslo 1973.

4. E. Haaland: "Experience with Condition Monitoring of Ship Propulsion Diesel Engines". ICMES Conference, Paris 1977.

9.3 Test of main engine cyl. 1.

Run the engine on steady state condition and indicate cyl. 1 to get a normal indication diagram for comparison.
Introduce F.O. injection nozzle wear, fault no. 10162.

To avoid the main engine RPM controller to compensate for the poorer efficiency on cyl. 1 by increasing the fuel link position this should be locked.

This is done only to demonstrate the isolated effect of the fault intoduced for educational purposes.

Set absolutely calm weather.

Observe the following effects:

- The exhaust temperature will have an increase.
- The power/MIP of this cylinder will have a decrease.
- The temperatures of the liner, the cylinder cover and the exhaust valve will have a small increase.
- The fresh water and the oil outlet temp. will have a very small increase.
- The total power of the main engine will be reduced.

Indicate cylinder no. 1 and observe the effect of the weared nozzle on the diagram of the injection and the combustion pressure.

Reset the fault by pushing the correct button on the main engine mimic panel.

10. CONCLUSION

The diesel plant simulator now installed was initiated by Norwegian marine training authorities and the Norwegian Ship Research Institute. It reflects a trend in modern process control training, where traning of the handling of complicated processes via its control systems is focused.

9. EXAMPLES OF OPERATION

The diesel plant simulator are used for regular courses from the autoumn of 1978. It is therefore a bit early to describe its impact on the training beyond the big expectations of everybody involved. Here, therefore, examples from the acceptance tests are described so that some idea of the capabilities of the simulator can be had.

9.1 Test of Dieselgenerator

Introduce a partially close FW cooling valve, fault no 1406, and increase the load on the diesel generator to above 600 KW. The flow will be reduced to about half its normal value, and the cooling water temperature will increase.
The control system will try to maintain the temperature within limits by max. opening.

As this is not enough, alarm is given at 80°C and shut down of the diesel generator will be performed at 90°C.

When tripped, the FW temperature will decrease steadily.

Reset fault no. 1406.

9.2 Test of Fresh Water Low Temp. System

Introduce heavy fouling on the sea water side of the fresh water cooler no. 1. (Fault no. 2004.)

Observe the following responses.

- The pressure drop on the sea water side of the cooler will increase while the pressure drop on the fresh water side will maintain its normal value.

- The sea water flow will be reduced while the flow on the fresh water side will stay normal.

- The outlet temperature on the sea water side will decrease while the outlet temperature on the fresh water side will increase.

Reset the fault by pushing on the reset fault push-button on the local pump panel.

8.1 Hardware

The real-time computer system has an efficient input/output system, both for analog and digital signals and an advanced interrupt handling system.

The basic computer system comprizes:

(1) Central processing unit including necessary control logic: Nord-42.

(2) 64K 16-bit ram semiconductor memory.

(3) Input/output electronics.

(4) Computer bus extender.

The system is delivered with the following I/O capacities:

Analog input :	16 signals
Analog output :	42 signals
Digital input :	256 signals
Digital output:	224 signals

The central processing unit, fast memory and pheripheral interface electronics is placed in two card frames.

The I/O electronics are placed in two additional card frames. There is room for 5 more cards in the top frame, allowing for future extensions of:

(A) 80 digital output signals, or
(B) 160 digital input signals, or
(C) 40 analog output signals, or
(D) 64 analog input signals.

The three card frames are mounted in two easily accessable racks in the alarm console.
Two extra card frames can be installed if more I/O spare capacity is required.

8.2 Software

All the programs of the system are written in assembly code. Apart from plant model programs, alarm-, control- and monitoring program, the system is delivered with comprehensive software for man/machine communication and self-check.

14 DIESELGENERATOR SYSTEM FAULTS

1401 % LOW GAIN - DIESEL RPM CONTROLLER
1402 % HIGH GAIN
1403 % UNSTABLE OUTPUT
1404 % CONSTANT OUTPUT

1405 % HEAVY MECH. LO PUMP WEAR

1406 % DG FW VALVE PARTIALLY CLOSED
1407 % DG FW VALVE CLOSED
1410 % DG START AIR VALVE CLOSED

15 EL. POWER SUPPLY SYSTEM FAULTS

1507 % HEAVY CONSUMER CONNECTED (DECK MACHINERY)
1510 % MAJOR SHORT CIRCUIT (BLACK-OUT)

1505 % TG OVERLOAD RELAY FAILURE
1506 % DG OVERLOAD RELAY FAILURE

1501 % LOSS OF TG MAGNETIZATION
1502 % LOSS OF DG MAGNETIZATION

1503 % HIGH VOLTAGE REGULATOR GAIN - TG
1504 % HIGH VOLTAGE REGULATOR GAIN - DG

11111 % CENTRAL SYSTEM MONITOR FAILURE

100 CYLINDER SYSTEM FAULTS

%
% CYLINDER 1
%

10611 % CYL. 1 - HP PUMP WEAR
10612 % - INJECTION NOZZLE WEAR
10613 % - INJECTION TIME EARLY
10614 % - INJECTION TIME LATE
10711 % CYL. 1 - PISTON RING FAILURE
10712 % - PISTON BLOW BY
10713 % - CYL. LINER CRACK
10714 % - REDUCED CYL. LUBR. FLOW
10715 % - LOW FLOW
10716 % - EXHAUST VALVE LEAKAGE

521 % MEDIUM AIR FLOW RESIST. INCREASE
432 % SCAV. AIR BELT FIRE

Fig. 10.
Types of faults which can be introduced.

- Transfer of eng. room simulation model - from disc to computer's memory.
- Stopping of eng. room simulation model.
- Transfer of eng. room simulation model - from computer's memory to disc.

7.2.2 Communication

The communication system enables the instructor to operate on the simulator model.

By this system he is able to change model parameters or to set/reset faults. The number of faults which can be set is more than 400. Some few of the faults are described on fig. 10.

The communication system enables

a) direct operation (on-line)
b) indirect operation (off-line)

Direct Operation (on-line):

"Direct operation" means an operation that allows the operator to get directly into the simulated process, or fetch results directly from the process.

Indirect Operation (off-line):

The instructor communication system may introduce various faults in programmed sequence. Consequently, the instructor must be able to program the sequence in advance.

To do this, the instructor has to put the Instructor Communication System in an indirect (off-line) operation mode.

8. COMPUTER SYSTEM

The DataChief systems delivered both to steam and diesel propelled ships, employ a medium sized computer and an I/O-system built into the central control consoles shown on fig. 6. The same hardware/software system have been employed for the simulator system.

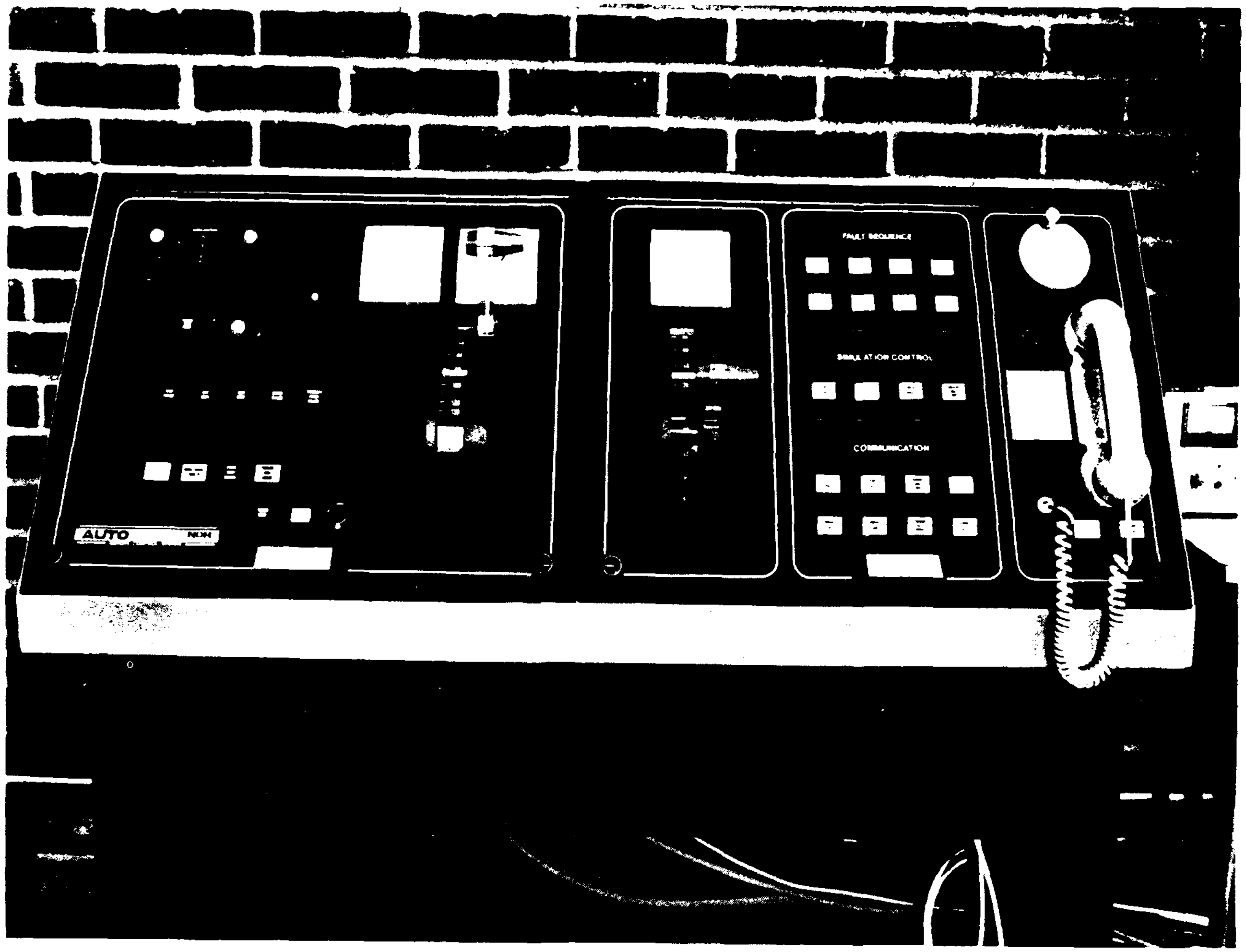


Fig. 9 INSTRUCTOR'S STATION - "BRIDGE CONTROL".
COMMUNICATION AND OPERATION PANEL

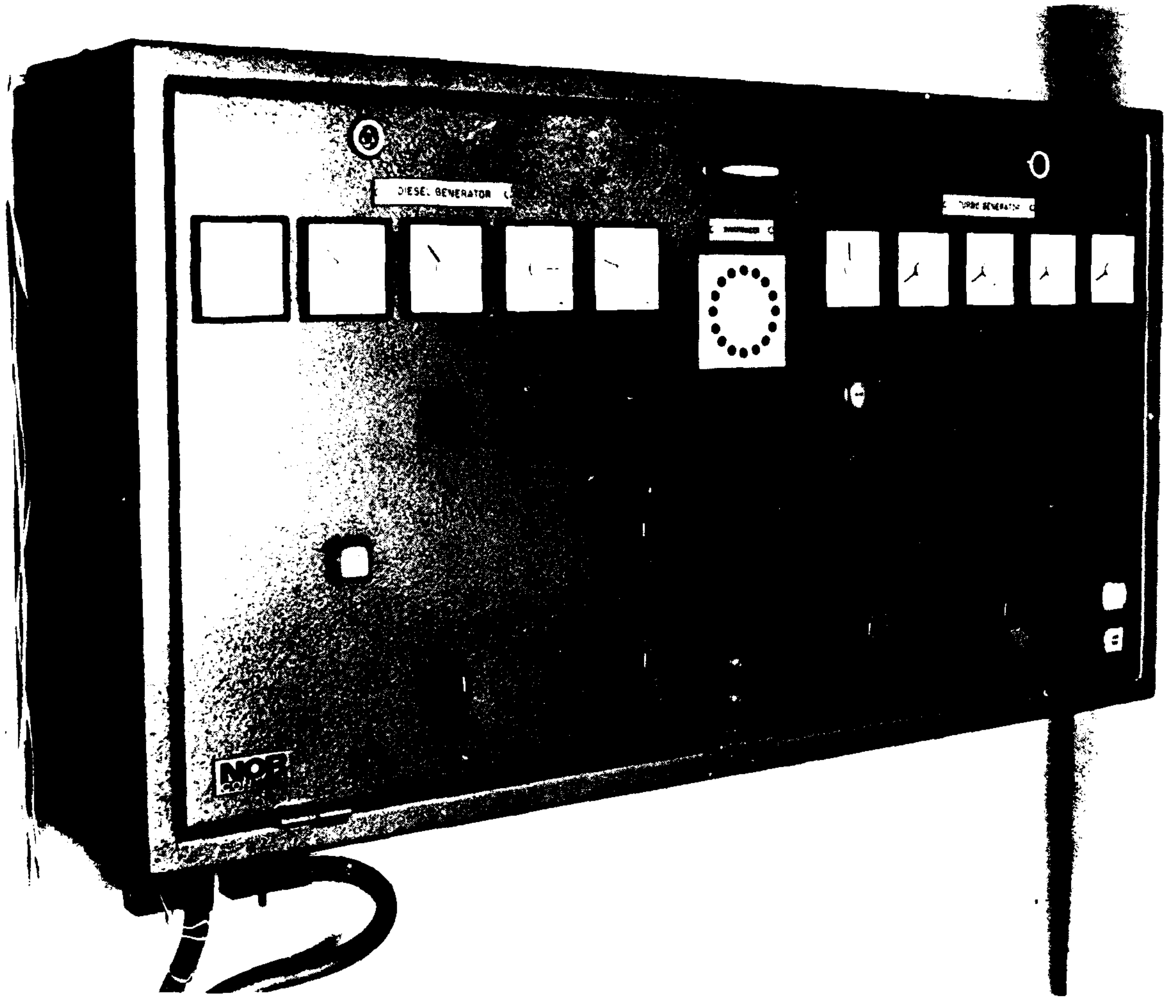


Fig. 8 SIMULATED MAIN SWITCHBOARD IN CONTROL ROOM.

7. BRIDGE/INSTRUCTOR CONSOLE

This console is located in a separate room (see fig. 1) and comprises

- bridge control unit
- instructor communication

The console is shown on fig. 9.

7.1 Bridge Control Unit

As can be seen from fig. 9 this unit comprises:

- RPM controller.
- RPM instrument.
- Engine order telegraph.
- Push-buttons for change between bridge control and engine room control.
- Push-buttons for different messages.
- Equipment for emergency run/stop.

This is a standard unit delivered to many ships.

In addition to this unit there is a propeller pitch controller of same design as the RPM controller. This unit is mounted together with an instrument showing the actual pitch, which is the output from the propeller servo model.

7.2 Instructor Communication System

The instructor communication system comprises a printer and a panel with push-buttons, arranged with respect to:

1) Simulation control.
2) Communication.

7.2.1 Simulation Control

The simulation control system will control the simulation in regards to:

- Starting of eng. room simulation model.

in this area, certain internal model parameters must be capable of adjustment to improve or otherwise modify system performance at a later date.

The math modeling task, therefore, can be broken down into smaller development tasks which must be attacked and solved in sequential fashion. First, the scope of the simulation task requires definition; that is, the list of equipments to be simulated must be tabulated and the specific operating characteristics of this equipment defined. Second, the signal inputs and outputs required to interface the software model to the trainer hardware must be defined. Third, a model development concept is required whereby complex models will evolve from the combining of small, less complex generic models. Fourth, the independent room models must be developed by integrating various component, subsystem, and system models. Finally, the integrated mode model must be developed by combining the three independent mode models.

The approach followed in the Device 19E22 math model development and implementation task is based upon a building block concept. The building blocks consist of basic generic math models for various process and equipment categories such as boilers/condensers, heat exchangers, turbines, pumps, generators, and pneumatic control elements. After the various basic blocks are developed, they are converted into class specific models, representing specific DE-1078 Class component equipments as described in the appropriate Type I Manuals, by the addition of pertinent input/output signals associated with controls, indicators, valves, casualties, etc. In general, class specific models are derived directly as flow diagrams. These class specific models are then integrated with other related models to form interim subsystems and systems. For example, feedwater or condensate header models integrate specific pump and valve models, electrical power distribution bus models integrate specific generator and bus tie logic models, steam header models integrate boilers with specific turbines, valves, etc. This process of building up to more complex integrated models is continued until a complete independent mode model is developed for the fire room, engine room, and auxiliary rooms. By then integrating the three independent mode models, the complete integrated mode trainer math model is developed.

Throughout the above generic model development and integration process interaction between models, subsystems and systems are tested against available data contained in Type I Manuals, Ships Information Books and other sources. Where data is not readily available, attempts are made to obtain both qualitative and quantitative data through discussions with qualified operating personnel as well as observation of actual power plant performance on DE-1078 Class ships.

During the early stages of the mathematical model effort, technical investigations were made into the transient as well as steady state behavior of appropriate thermodynamic processes. These investigations included literature searches and discussions with qualified steam propulsion plant design and operating personnel, as well as others knowledgeable in the process control and simulation field. The results of these investigations led to the formulation of the building block concept described earlier.

Because many of the processes inherent in a steam propulsion plant are repeated in one form or another in various components and systems, the development of basic generic math models for the DE-1078 Class plant was of utmost importance. The availability of such models ensures continuity from model to model and avoids unnecessary duplication of effort and potential errors in technical approach due to misinterpretation or misunderstanding of data and system performance requirements. Development and derivation of pertinent basic generic math models utilized throughout the trainer such as the boiling/condensing process, heat exchangers, turbines, pneumatic control systems, and electrical power generation was necessary.

Typical examples of the systems modeled are the major fluid systems which fall into one of the following categories:

Main Steam System (Superheated)
1200 PSI and 150 PSI Desuperheated Steam System
Condensate and Feed Systems
Main and Auxiliary Condenser Air Removal System
Seawater Circulating System
Main Turbine Lube Oil Service System
Lube Oil Service System
Fuel Oil Service System
Automatic Boiler Control System
Auxiliary Exhaust System

Main Steam System. The main steam system functions to transport superheated high-pressure and temperature steam from the boilers to various steam driven components. The model includes all active valves, controls and instruments as well as appropriate piping losses. The modeling includes establishing total main steam requirements for boiler loading according to component flow rates, valving, lineup and plant status. Steam thermodynamic parameters are computed at component inlets and provide temperature and pressure inputs to all local and remote instruments that are to be functionally simulated. When the trainer is operating in the integrated mode, the steam flow is dynamically computed according to plant status and equipment alignment; when the trainer is operating in the independent mode, the room isolation valves are logically closed, and each room has its own independent steam parameters under instructor control.

1200 PSI and 150 PSI Desuperheated Steam Systems. The auxiliary steam system supplies lower temperature and lower pressure steam to support equipments in the propulsion plant not designed for main steam temperature and/or pressure conditions. All piping, valving, and instrumentation that are functional are modeled. The modeling provides the steam properties at the desuperheater exit and at the inlets to supplied components.

Condensate and Feed Systems. The condensate and feed systems function primarily to return condensed steam from main and auxiliary condensers to the boiler regulated by the system dynamics. Piping, tanks, valves and instrumentation are modeled. The modeling establishes the flows out of the main and auxiliary condensers and flow into and out of the reserve fresh water tank and fresh water drain tank consistent with condensate pump and fresh water drain tank pump operation, valve alignments, and deaerating feed tank (DFT) status including level control action, and also establishes water properties at the DFT inlet, and proper feed properties to the boiler consistent with DFT pressure, feed water cooler and pump operations, and regulating valve position.

Main and Auxiliary Condenser Air Removal Systems. This system removes air from the condensers. To satisfy the simulation, all necessary air ejectors, piping, and valving are modeled.

Seawater Circulating System. The seawater circulating system provides cooling seawater to the main condensers, lube oil coolers, and Ships Service Turbo Generator (SSTG) coolers. The simulation model establishes seawater flows and temperatures according to the ship operating conditions.

Main Turbine Lube Oil Service System. This system functions to supply lube oil to the bearings of the main engine and reduction gears to provide proper lubricating and cooling. To satisfy the simulation, all necessary piping, valving, and pumps are modeled.

Lube Oil Service System. This system supports the lube oil service system by supplying new and/or cleaned oil for main and auxiliary machinery use. The only functional valving and instrumentation modeled is that associated with the lube oil purifier.

Fuel Oil Service Suction and Discharge System. This system provides fuel oil to the boiler burners at a pressure consistent with combustion requirements. All required pumps, valves and instrumentations are modeled to establish the systems normal operation, start-up and shutdown.

Automatic Boiler Control (ABC) Systems. There are three ABC subsystems modeled:

Automatic Combustion Control (ACC) - The ACC subsystem is modeled and incorporated in the associated subsystems.

Feedwater Control (FWC) - The FWC subsystem is modeled.

Feed Pump Control (FPC) - This subsystem functions to maintain a constant differential pressure across the feed water regulating valve and a minimum flow rate through the main feed pumps. This subsystem is modeled.

Auxiliary Exhaust System. This system functions to maintain pressure of the auxiliary exhaust system by regulating valves which either dump excess steam to the condensers or admit augmenting steam from the auxiliary steam system. This system is modeled.

Casualties

The single most valuable advantage that simulators have over other training methods is the capacity to provide casualty training without fear of harm to personnel or damage to equipment. Following is the casualty training capability of Device 19E22.

Fire Room Casualties.

Booster Pumps - Total failure of pumps, pumps turned off.
Main Feed Pumps - Total failure of pumps, pumps turned off.
DFT - Shell pressure decays to zero.
Ruptured Internal Boiler Pressure Part - Catastrophic failure.
Fuel Oil Service Pumps - Total failure. Pumps turned off.
Water in Fuel Oil - Simulated by fluctuating fuel oil service pump output pressure, flame flicker and flameout.
Fire in Boiler Air Casing - Visual effect.

Surface Blow - Casualties related to faulty piping and improper system alignment sequencing.
Auxiliary Exhaust Augmenting Valve - Fail closed.
Auto Combustion Control Steam Demand - Fail high.
Auto Combustion Control Steam Demand - Fail low.
Air Flow Transmitter - Decay to zero.
Forced Draft Blower A/M Station - Decay to zero.
Four-Way FDB A/M Station - Fail in present position.
Fuel Oil Supply Valve - Fail in closed position.
Fuel Oil Supply Valve - Fail in present position.
Recirc Valves - Fail closed.
Main Feed Pumps - Loss of control air.
Feedwater Flow Controller - Output signal goes to maximum.
Feedwater Flow Control Valves - Ruptured diaphragm.
Control Air - Decay air receiver pressure to zero.
Forced Draft Blower Air Damper - Jam shut.
FDB Oil Pumps - Fail.
FDB Governor - Fail.

Engine Room Casualties.

Lube Oil Service Pumps - Fail, pump off.
Engine Lube Oil Pump - Fail.
Lube Oil Unloader Valve - Fail open.
Lube Oil Cooler - Fail.
Lube Oil Unloader Valve - Fail closed, trainer freezes 10 minutes after initiating casualty if corrective action is not performed.
L. O. Strainer Clogged - Clog only one strainer at a time, fire alarm activated and trainer shall freeze if oil pressure not relieved prior to removing a strainer.
Turbine Rumble - Simulates a warped rotor caused by failure to jack the turbine when steam is applied.
Unusual Noise in Reduction Gear - Random metallic sound.
Hot Bearings - Output oil temperature exceeds $180^{\circ}F$ or cause a greater than $50^{\circ}F$ rise across a bearing.
(Norm Temp Rise - $35 \pm 5^{\circ}F$, Norm Out Temp - $160 \pm 10^{\circ}F$)
H. P. Turbine Thrust Bearing - Failure, Temperature rise greater than $50^{\circ}F$ or oil temperature greater than $180^{\circ}F$. Rotor shift of 0.040 ± 0.01 inches. Alarm after 5 minutes.
Ahead Throttle Jammed - Ahead throttle wheel locked in present position and freezing the computed throttle position.
Ahead Throttle Deactivated - Throttle wheel turns, no response.
Main Circulating Pump - Fail.
Scoop Flapper Valve Jammed - Fail open at ship speed less than 12 knots.
Scoop Flapper Valve Jammed - Fail close at ship speed more than 12 knots.
Main Condenser First Stage Air Ejectors - Fail.
Main Condenser Second Stage Air Ejectors - Fail.
Loss of Gland Seal Steam.
Thermostatic Recirculation Valve - Fail closed.
Main Condensate Pumps - Fail so that water will rise in hot well and start to cover tubes.
Auxiliary Exhaust Unloading Valve - Fail closed.

Auxiliary Space Casualties.

SSTG Lube Oil Strainer - Clogged, fire alarm activated if oil pressure not relieved prior to unscrewing T-handle assembly.

SSTG Excessive Lube Oil Pressure.
SSTG Turbine Rumble - Warped rotor.
Unusual Noise in Reduction Gears - Random metallic sound.
Hot Bearings - SSTG Low Pressure End Journal Bearing - Oil temperature to exceed 180°F.
SSTG Auxiliary Circulating Pump - Fail.
SSTG First Stage Air Ejectors - Fail.
SSTG Second State Air Ejectors - Fail.
SSTG Loss of Gland Seal Steam.
SSTG Auxiliary Condensate Pump - Fail off so water will rise in hotwell and over tubes.
SSTG Governor - Fail High.
SSTG Clogged Saltwater Strainer - Loss of coolant.
Diesel Solenoid Start Valve - Fail to operate.
System Induced Overload.
SSTG Line Frequency - Fluctuate more than five percent.
SSTG Voltage - Fluctuate more than one percent
Shock Trip - SSTG and Diesel generator near miss explosion simulated - Ability to reset tripped generator shall be provided.
Phase Grounding - SSTG short to ground for only one phase of each SSTG.
SSTG Reverse Power Relay - Fail to operate.
SSTG Loss of Excitation Voltage.
SSTG Loss of Residual Magnetism - Provided when SSTG is idle.
Diesel Fuel Starvation - Fuel flow down 50 percent.
Diesel Coolant Loss.

Accompanying photographs (figures 1 through 9) show various trainer viewsand comparisons between real and simulated equipments.

SUMMARY

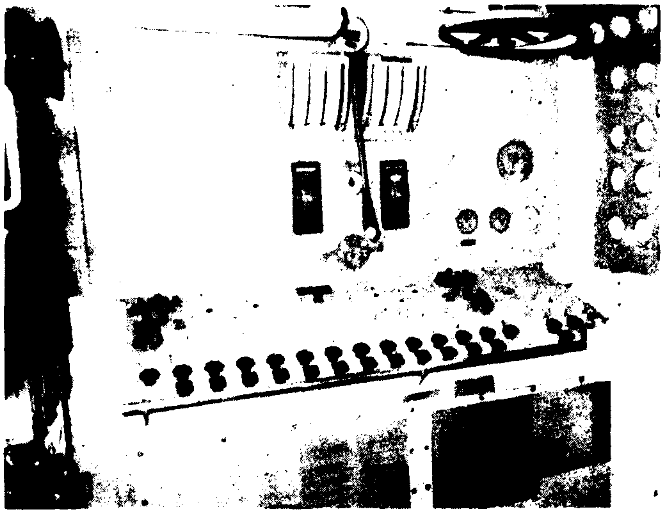
The training problem in the fleet today has been compounded by the greatly reduced tempo of operational at-sea training exercises and the impact of budgetary cutbacks and fuel conservation efforts. In response to fleet requirements and to help fulfill the need for training in an operational environment at greatly reduced cost, the Naval Training Equipment Center has procured a 1200 PSI Steam Propulsion Plant Trainer from Hydrosystems, Inc., Farmingdale, New York.

The Propulsion Plant Trainer, a single unit procurement, has been developed to supplement and support three existing engineering officer and engineering supervisor courses at the Surface Warfare Officers School, Newport, Rhode Island. Working with entering students of varied backgrounds and constrained by established course lengths and the limited availability of school ships, the terminal goal of the school is to turn out students who, upon reporting aboard their assigned ships, can perform the routine duties of their billets and within several months be competent in virtually all aspects of their jobs. Device 19E22 will have the capability of replacing some of the time now devoted by the school to training at sea and will bring a new concept in hands-on training capability to the surface engineering community.

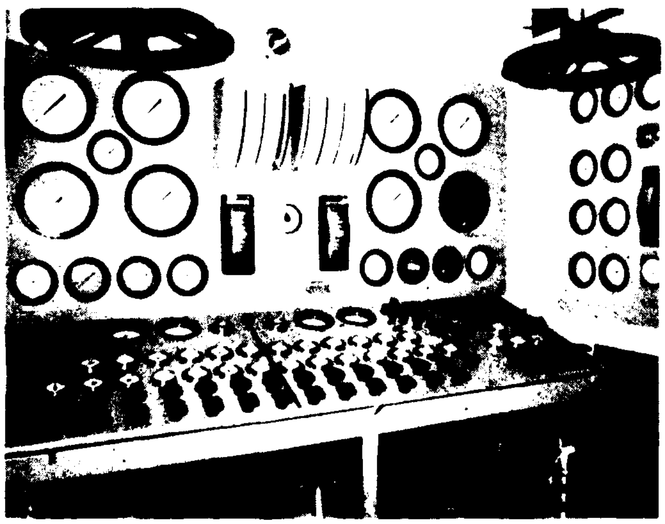
A. Real

B. Simulated

Figure 1. Light-Off Port

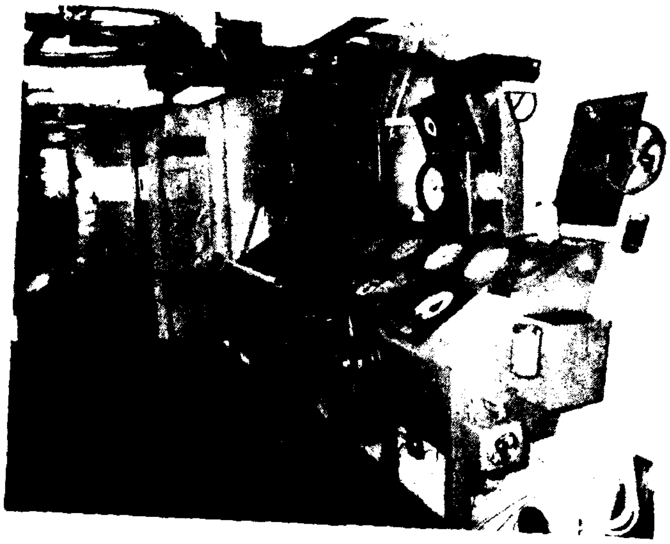


A. Real

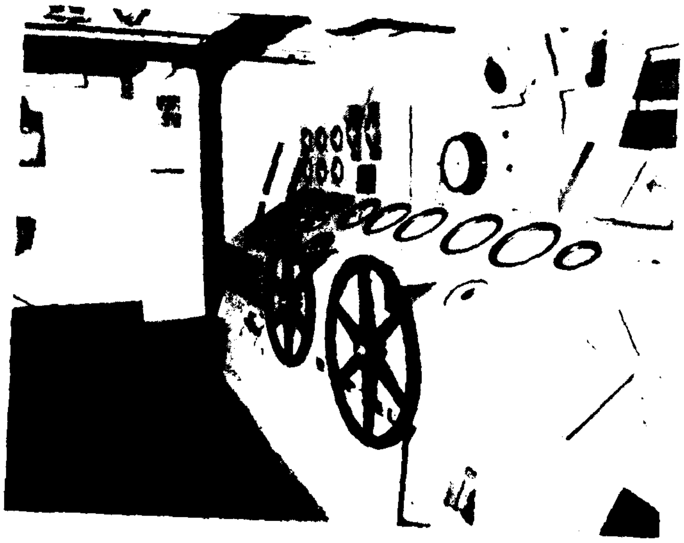


B. Simulated

Figure 2. Automatic Boiler Control Console



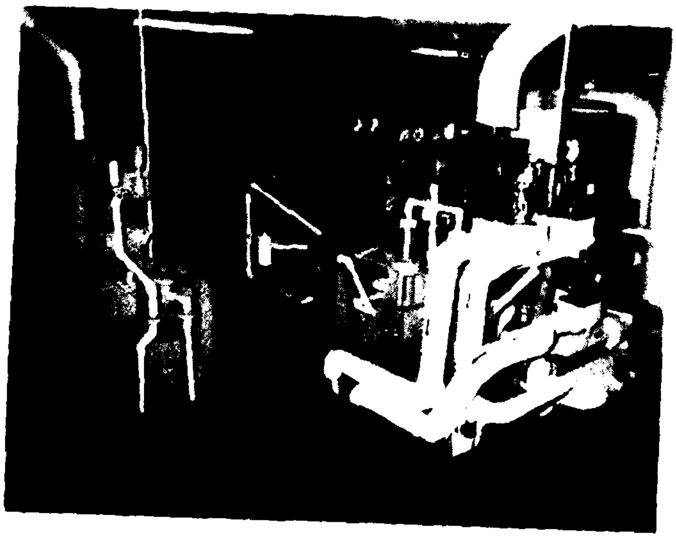
A. Real



B. Simulated

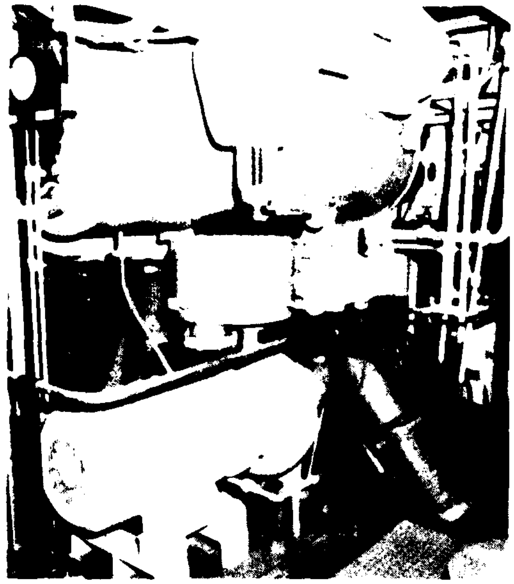
Figure 3. Throttle Room

A. Real

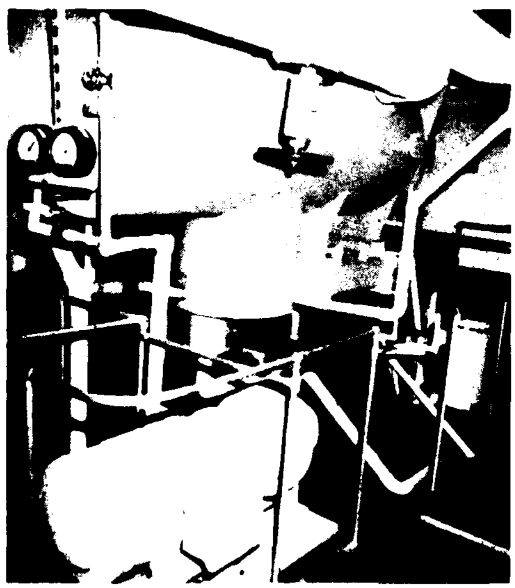


B. Simulated

Figure 4. [illegible] Air Compressor and Turbogenerator



A. Real



B. Simulated

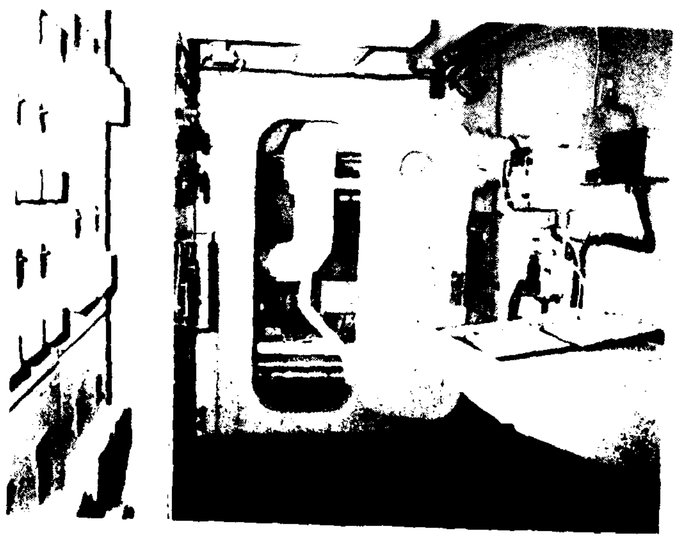
Figure 5. Turbogenerator - Lower Level

A. Real

B. Simulated

Figure 6. Diesel Generator

A. Real



B. Simulated

Figure 7. Emergency Switchboard Room

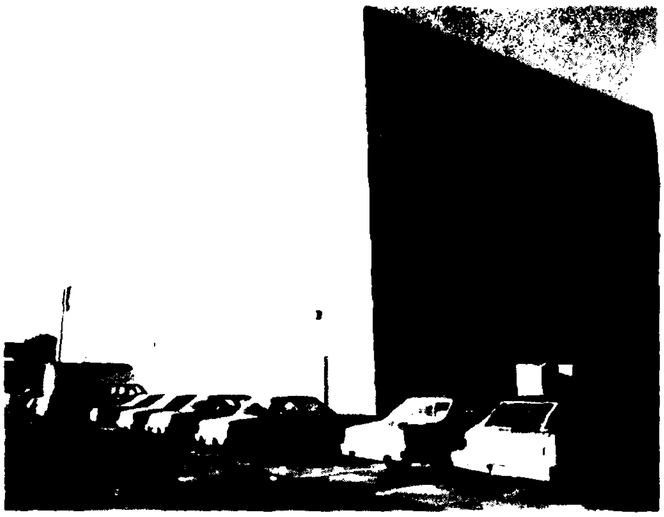


Figure 8. Robinson Hall, Device [illegible] Building

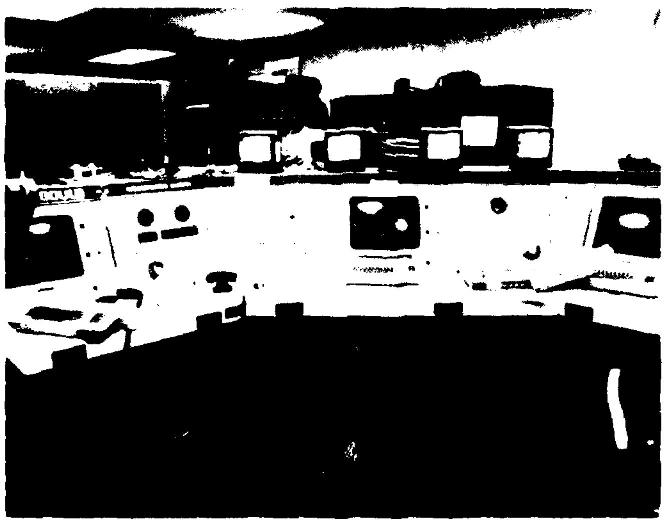


Figure 9. Instructor Console

REAL-TIME SIMULATION OF A STEAM
PROPULSION PLANT
ANNEX A

The fire room control air math model follows as an example of the math models required to simulate the 1200 PSI Propulsion Plant. [illegible] 1A through 7A provide the inputs, outputs, and constants for the control air system model. Flow charts for this model are provided [illegible] Figure 1A.

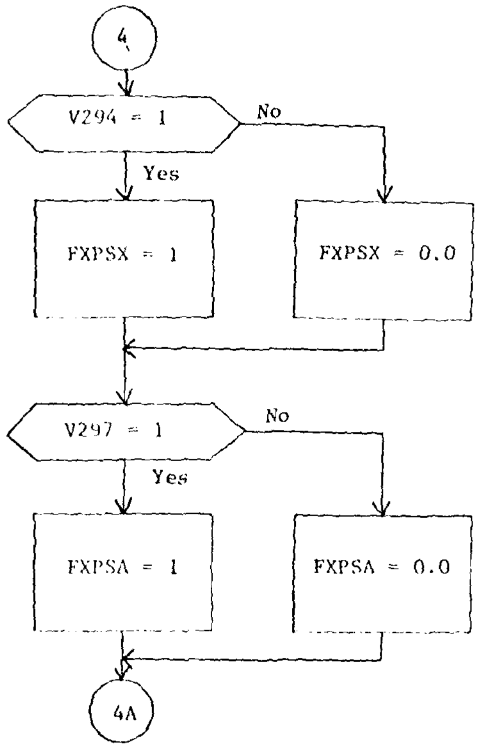


Figure 1A. Fire Room, Control Air (Sheet 5 of 12)

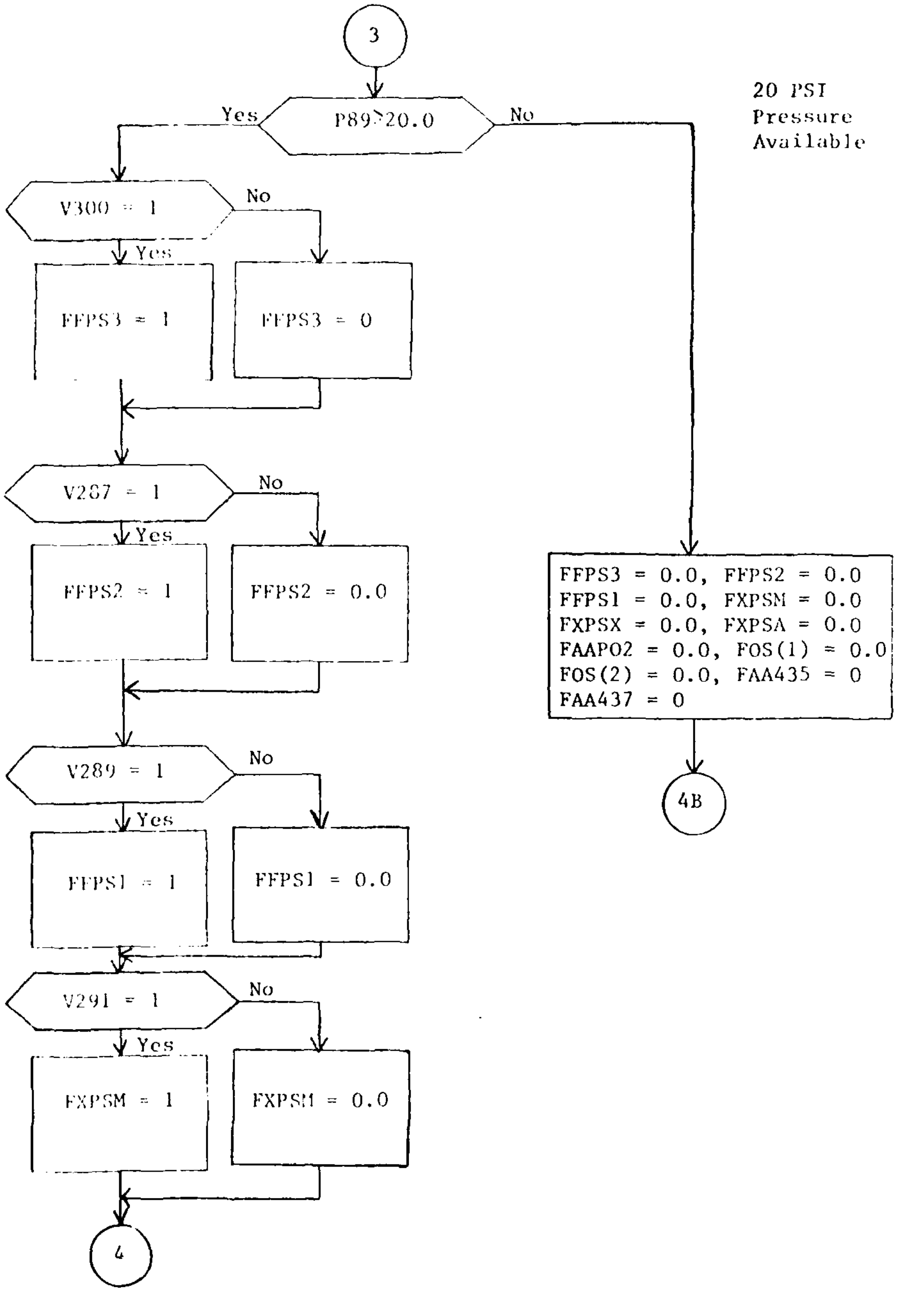


Figure 1A. Fire Room, Control Air (Sheet 4 of 12)

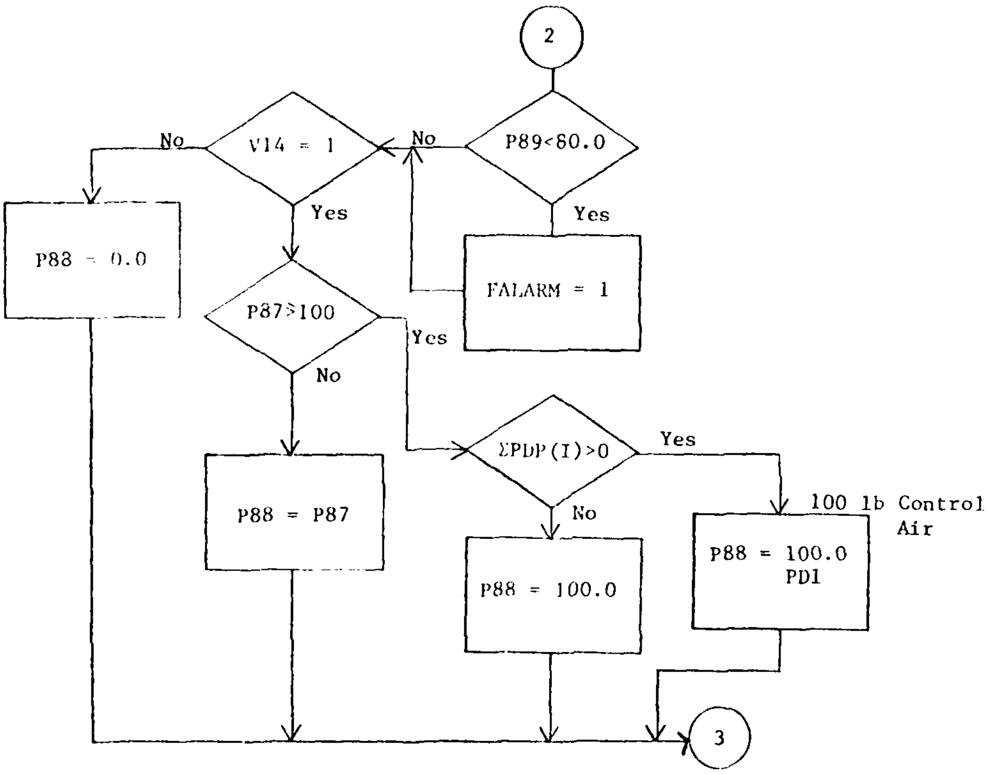


Figure 1A. Fire Room, Control Air (Sheet 3 of 12)

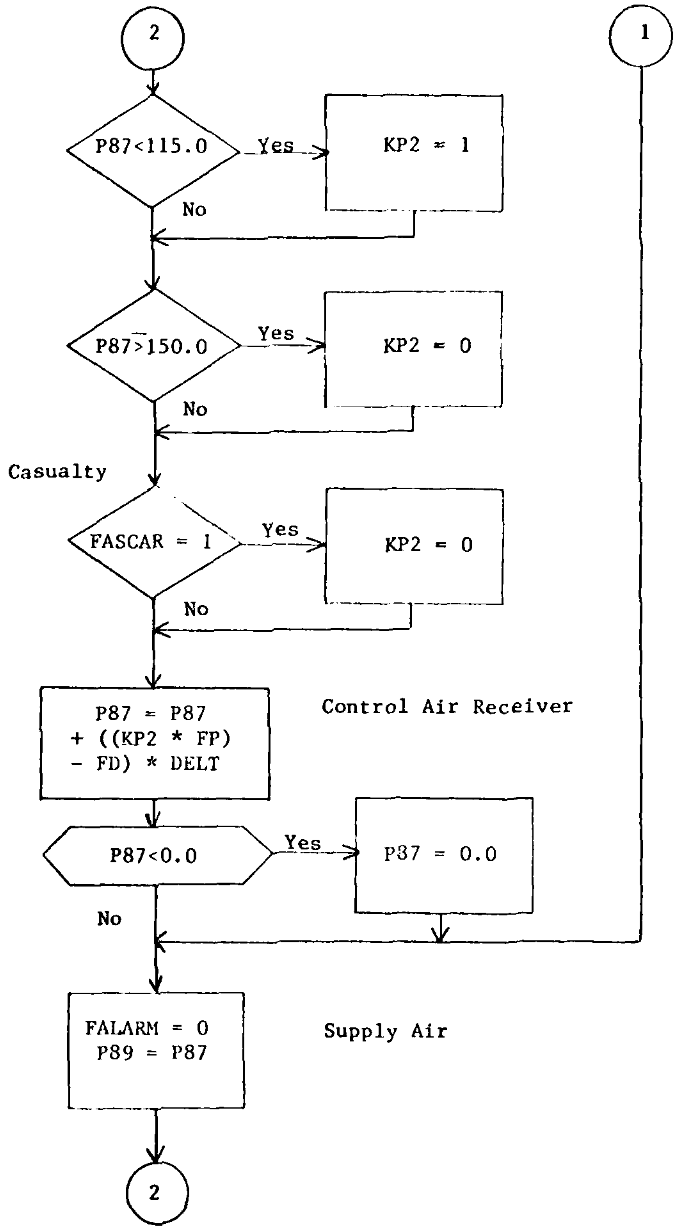


Figure 1A. Fire Room, Control Air (Sheet 2 of 12)

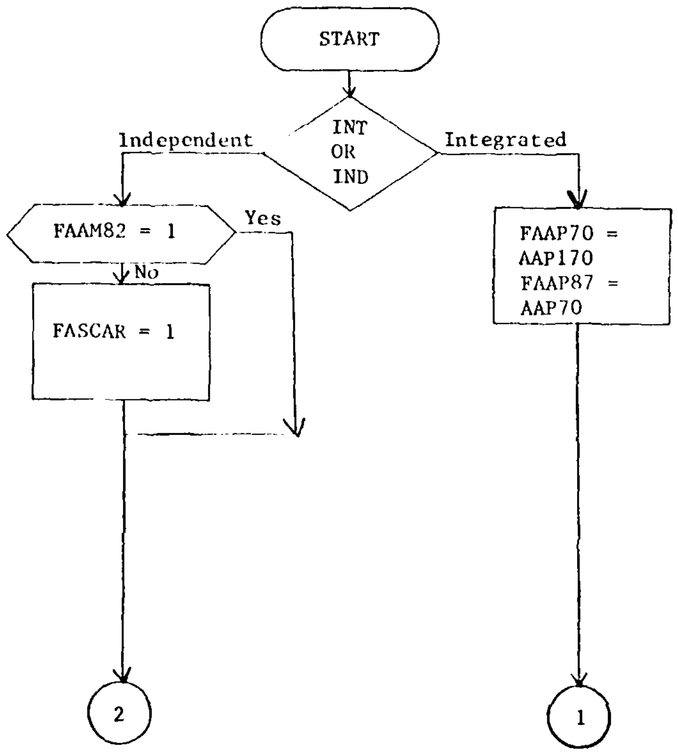


Figure 1A. Fire Room, Control Air (Sheet 1 of 12)

Table 6A. Control Air Continuous Internal Variables

Mnemonic	Description	Units
FAAP70	Low Pressure Air Receiver	psig
EAP186	Control Air Receiver	psig
EAP188	100 lb Control Air	psig

Table 7A. Control Air, Model Constants

Mnemonic	Description	Value	Units
FAAFP	Low Pressure Air Compressor Pumping Rate	0.7	psi/sec
FAAFD	Control Air System Leakage Rate	0.1	psi/sec
FAAPD1	Pressure Drop with Valves Open	10.0	psig
AADELT(1)	Iterative Time Increment	0.5	sec

Table 5A Control Air, Discrete Internal Variables

Mnemonic	Description	Units
FAAKP2	Low Pressure Air Compressor No. 2 - On/Off	1,0
FASCAR	Casualty - Causes L.P. Air Receiver to Decay to Zero	1,0
FAAS54	Select Switch, Low Pressure Start/Stop	0,1
FAAKP2	Low Pressure Air Compressor No. 2 - On/Off	1,0
FASCAR	Casualty - Causes Low Pressure Air Receiver to Decay to Zero	1,0

Table 4A. Control Air, Continuous Output Variables

Mnemonic	Definition	Units	Destination
FAAG25	Control Air Pressure	psig	G325
FAAG26	Air Lock Pressure	psig	G326
FAAG56	Control Air Pressure	psig	G456
FAP186	Control Air	psig	Engine Room G186
EAPE71	Lube Oil Service Unloading Valve	psig	Engine Room
EAPE73	Main Turbine Gland Seal Supply	psig	Engine Room
EAPE75	Main Turbine Gland Seal Supply	psig	Engine Room
EAPE78	Aux Exhaust Steam to Distiller Plant	psig	Engine Room
FAAP87	Control Air Receiver	psig	Fire Room P187
FAAP88	100 lb Control Air	psig	Fire Room P188
FAAP89	Supply Air	psig	Fire Room P189
FAAP81 (1)	Supply Air Pressure to Retr Soot Blo Controls	psig	Fire Room P195
FAAP81(2)	Supply Air Pressure to Retr Soot Blo Controls	psig	Fire Room P197

Table 3A. Control Air, Discrete Output Variables (cont)

Mnemonic	Definition	Units	Destination
FFPS1	Supply Air Enabled Excess Feed Valve	1-0	Fire Room P447
FFPS2	Make Up Feed Valve	1-0	Fire Room P449
FFPS3	Fresh Water Drain Collecting Tank Pump Discharge Valve	1-0	Fire Room P433
FOS(1)	Supply Pressure Enabled Fuel Oil Service Header Unloading V1A	1-0	Fire Room P435
FOS(2)	Supply Pressure Enabled Fuel Oil Service Hydraulics Unloading V1B	1-0	Fire Room P437
FXPSM	Supply Air Enabled Main Cond Unloading Valve	1-0	Fire Room P427
FXPSX	Supply Air Enabled Aux Cond Unloading Valve	1-0	Fire Room P429
FXPSG	Supply Air Enabled Steam Augmenting Valve	1-0	Fire Room P423
FXPSA	Supply Air Enabled Atmos Exhaust Unloading Valve	1-0	Fire Room P431
FAAP02	Supply Air Enabled to Feed-water Control Valve to Line Desuperheater (V557)	1-0	Fire Room Steam 150 lb
FAAP04	Supply Air Enabled for Steam Air Ejectors (V174)	1-0	Fire Room P421
FAA435	Control Air Supply to Fuel Oil Unloading 1A	1,0	G435
FAA437	Control Air Supply to Fuel Oil Unloading 1B	1,0	G437
EAV81	Air Motor - Ahead Guarding	1,0	Engine Room V81

Table 3A. Control Air, Discrete Output Variables

Mnemonic	Definition	Units	Destination
MALOP - C AIR P DRN	Drain Open - Air Leak	1,0	Fire Room Malop
FAARP3	3 PSI Control Air Available	1,0	Fire Room Fwd Pump Controller Model
FADAO	65 PSI Control Air Available	1,0	Fire Room
FASFAS	65 PSI Air for Field EQ (Fuel Air System)	1,0	Fire Room
FASCAS	65 PSI Air for Console (Combustion Air System)	1,0	Fire Room
FASALH	65 PSI Air to Air Lock Hdr (ALH)	1,0	Fire Room
FAE46(1)	Emerg. Steam Stop Shutdown - Motor Controller - Enabled	1	A065/M248
FAV46(1)	Air Motor: Superheater Steam Outlet Boiler 1 - Shut	1	M249
FAE46(2)	Emerg. Steam Stop Shutdown - Motor Controller - Enabled	1	A067/M250
FAV46(2)	Air Motor: Superheater Steam Outlet Boiler 1 - Shut	1	M251
FAE167(1)	Emerg. Steam Stop Shutdown - Motor Controller - Enabled	1	A064/M252
FAV167(1)	Air Motor: Desuperheater Steam Outlet Boiler 1 - Shut	1	M253
FAE167(2)	Emerg Steam Stop Shutdown - Motor Controller - Enabled	1	A066/M254
FAV167(2)	Air Motor: Desuperheater Steam Outlet Boiler 1 - Shut	1	M255
FALARM	Low Pressure Alarm (Control Air $<$ 80 psi)	1,0	E024

Table 2A. Control Air, Discrete Input Variables (cont)

Mnemonic	Definition	Units	Source
FAV653	100-150 Control Air Drain	1,0	V1653
FAV654		1,0	V1654
EAM118	Control Air On	1,0	Engine Room M118 Instructor
EAV271	L.O. Service Unloading	1,0	Engine Room V1271
EAV273	Main Turbine Gland Seal Supply	1,0	V1273
EAV275	Main Turbine Gland Seal Unloading	1,0	V1275
EAV278	Control Aux Exchanger to Steam Distiller Plant	1,0	V1278
EAV279	Steam Control to Ahead Guarding	1,0	V1279
FDV167(1)	Desuperheater Stop Valve 1A	1,0	RTI V167A
FDV167(2)	Desuperheater Stop Valve 1B	1,0	V167B
FMV46(1)	Superheater Stop Valve 1A	1,0	V46A
FMV46(2)	Superheater Stop Valve 1B	1,0	V46B

Table 2A . Control Air, Discrete Input Variables (cont)

Mnemonic	Definition	Units	Source
FAV311	To Fuel Oil Service Header Unloading Valve	1,0	V1311
FADV(I)[1,2]	Differential Control Panel Air	1,0	V1535A, V1535B
FAV536	Differential Control Panel Drain	1,0	V1536
FAV537	Differential Control Panel Drain	1,0	V1537
FAV538	Recirculating Panel Air	1,0	V1538
FAV539	Recirculating Panel Air	1,0	V1539
FAV540	Recirculating Panel Drain	1,0	V1540
FAV541	Recirculating Panel Drain	1,0	V1541
FAV634	Air Lock Release	1,0	V1634
FAV635(I)[1,6]	65 psi Air For Field Eq. (Fuel Air System)	1,0	V1635, V1636, V1637, V1638, V1639, V1640
FAV641(I)[1,8]	65 psi Air For Console (Combustion Air System)		V1641, V1642, V1643, V1644, V1645, V1646, V1647, V1648
FAV649	100-150 Control Air Supply	1,0	V1649
FAV651	100-150 Control Air Supply	1,0	V1651

Table 2A. Control Air, Discrete Input Variables

Mnemonic	Definition	Units	Source
FAAV14	100 lb Control Air Open/Close	1,0	V1314
FAAPDP(I)[1,8]	Steam Control Valves	1,0	V1306-V1309 and V1335-V1338 Open/Close
AASS54	LP Air Comp. No. 1 Select Sw., LP Start/Stop	0,1	Aux Room S. SW. 254
FAAM82	LP Air On	1,0	M82 Instructor
FAV281	To Retr Soot Blo Cont	1,0	V1281
FAV282	To Retr Soot Blo Cont	1,0	V1282
FAV287	To Makeup Feed	1,0	V1287
FAV289	To Excess Feed	1,0	V1289
FAV291	Unloading to Main Condenser	1,0	V1291
FAV294	Auxiliary Exhaust Unloading to SSTG Condenser	1,0	V1294
FAV296	Auxiliary Exhaust Augment Reducing	1,0	V1296
FAV297	Auxiliary Exhaust To ATM	1,0	V1297
FAV300	Feedwater Regulator Valve	1,0	V1300
FAV302	Feedwater to 150 psi Line Desuperheated	1,0	V1302
FAV304	To 1200/150 psig Red Valve For Steam To Air Ejectors	1,0	V1304
FAV310	To Fuel Serv Header Unloading Valve	1,0	V1310

Table 1A. Control Air, Continuous Input Variables

Mnemonic	Definition	Units	Source
AAP170	Low Pressure Air Receiver No. 1	psig	Aux Room G170
FAAP87	Control Air Receiver	psig	Fire Room G187
FAAP88	100 lb Control Air	psig	Fire Room G188

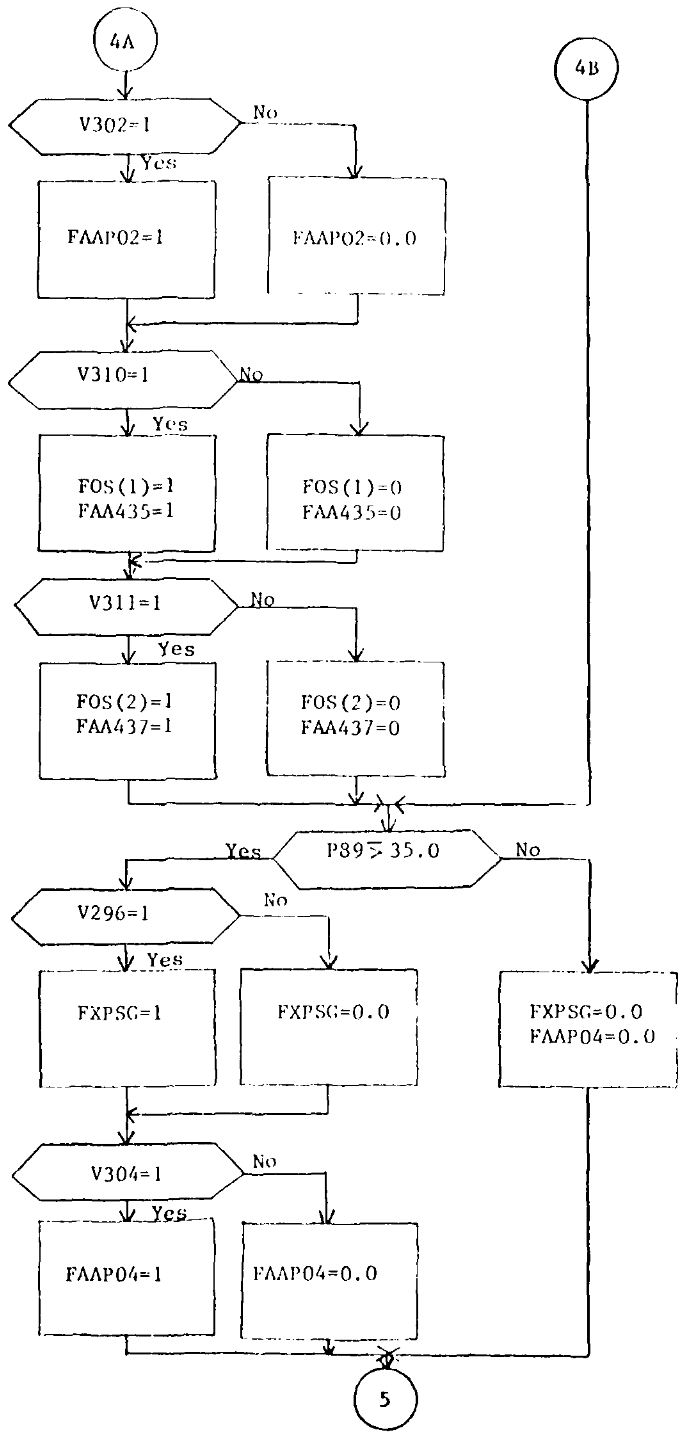


Figure 1A. Fire Room, Control Air (Sheet 6 of 12)

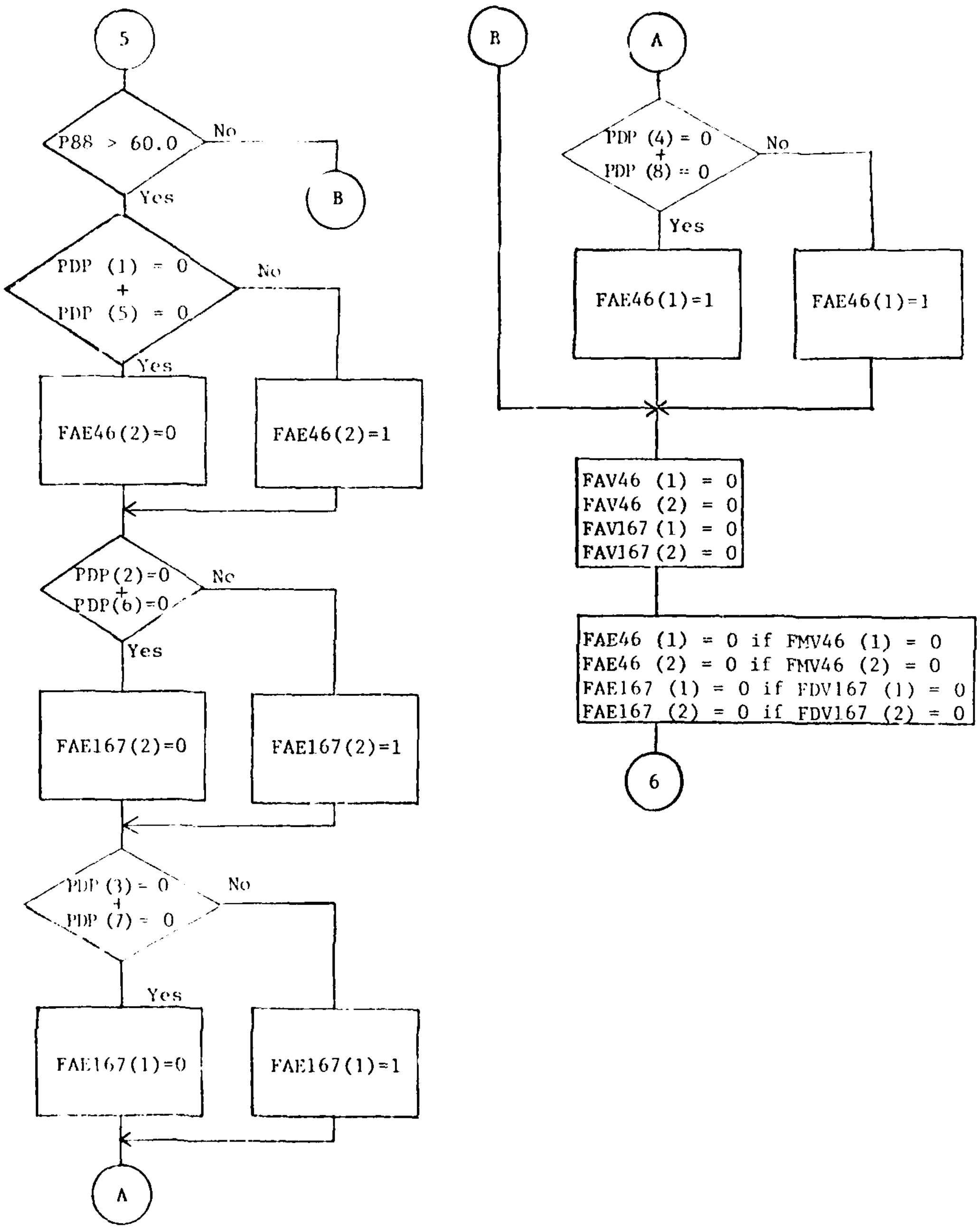


Figure 1A. Fire Room, Control Air (Sheet 7 of 12)

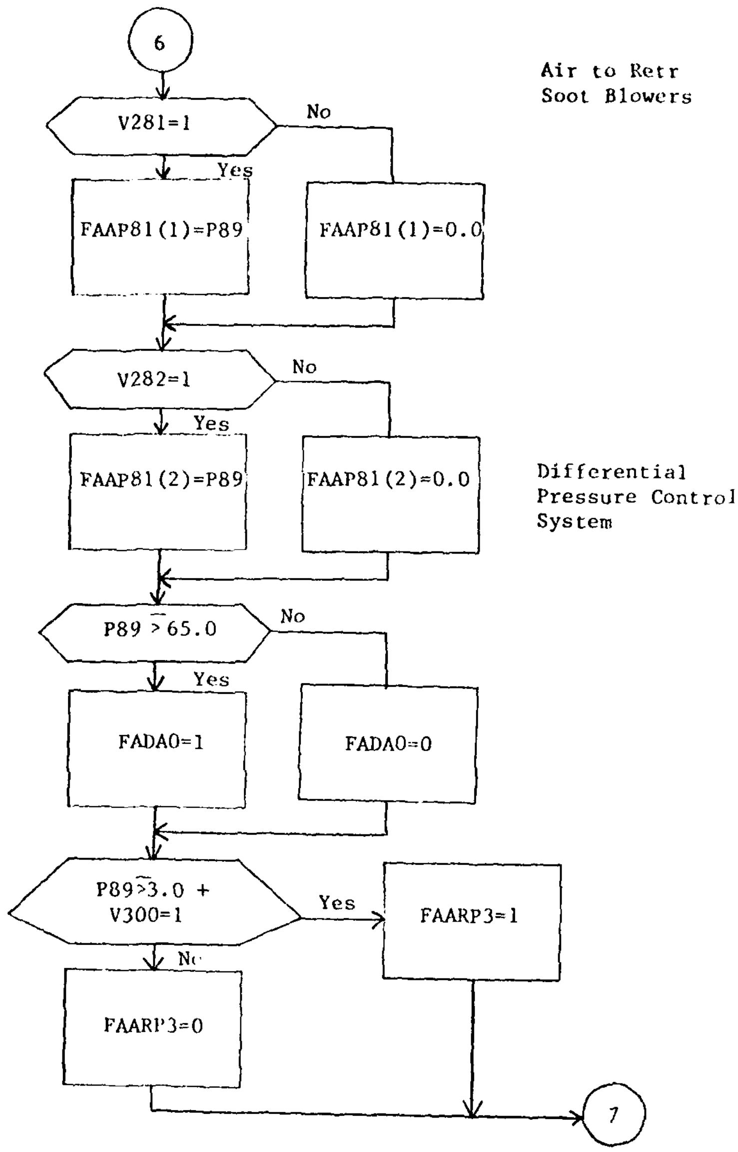


Figure 1A. Fire Room, Control Air (Sheet 8 of 12)

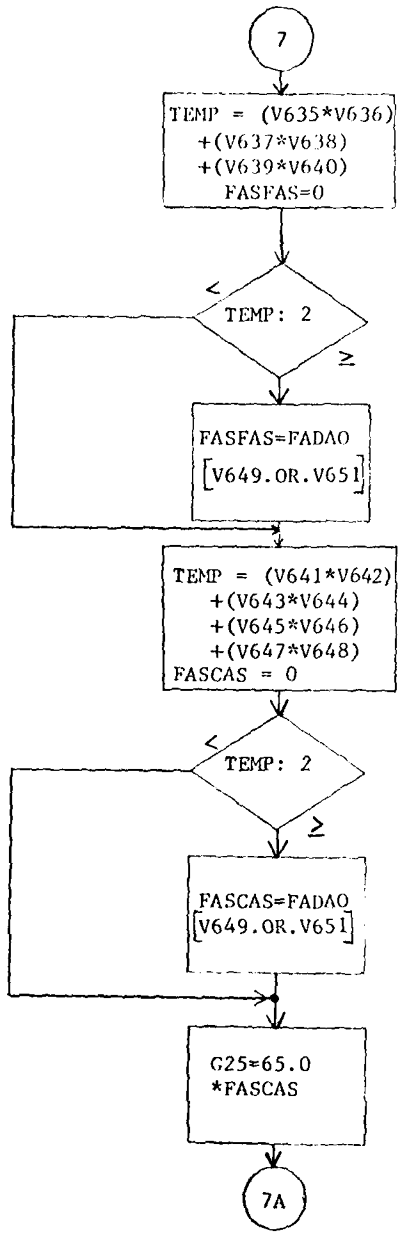


Figure 1A. Fire Room, Control Air (Sheet 9 of 12)

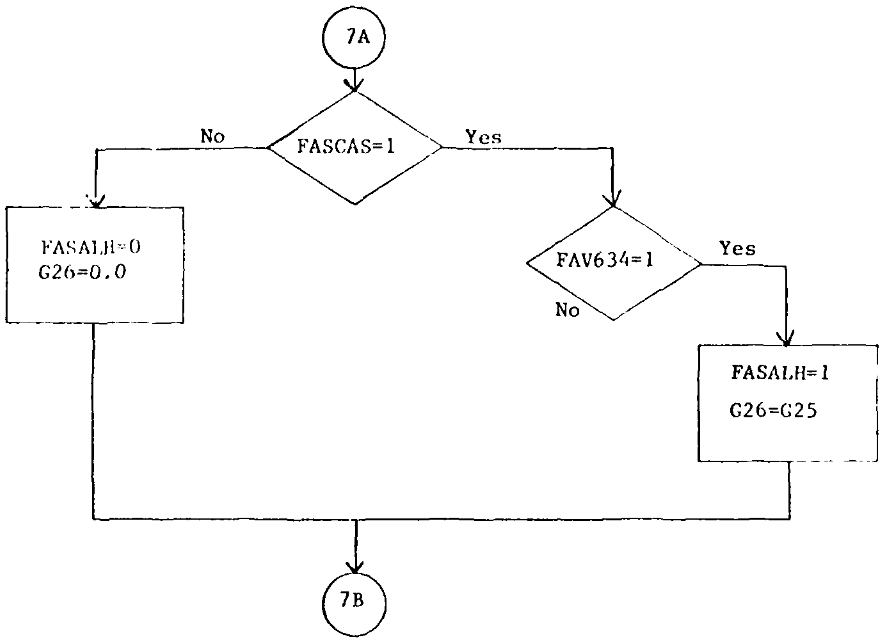


Figure 1A. Fire Room, Control Air (Sheet 10 of 12)

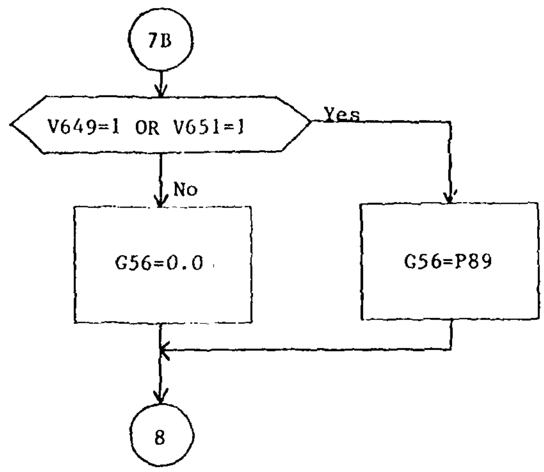


Figure 1A. Fire Room, Control Air (Sheet 11 of 12)

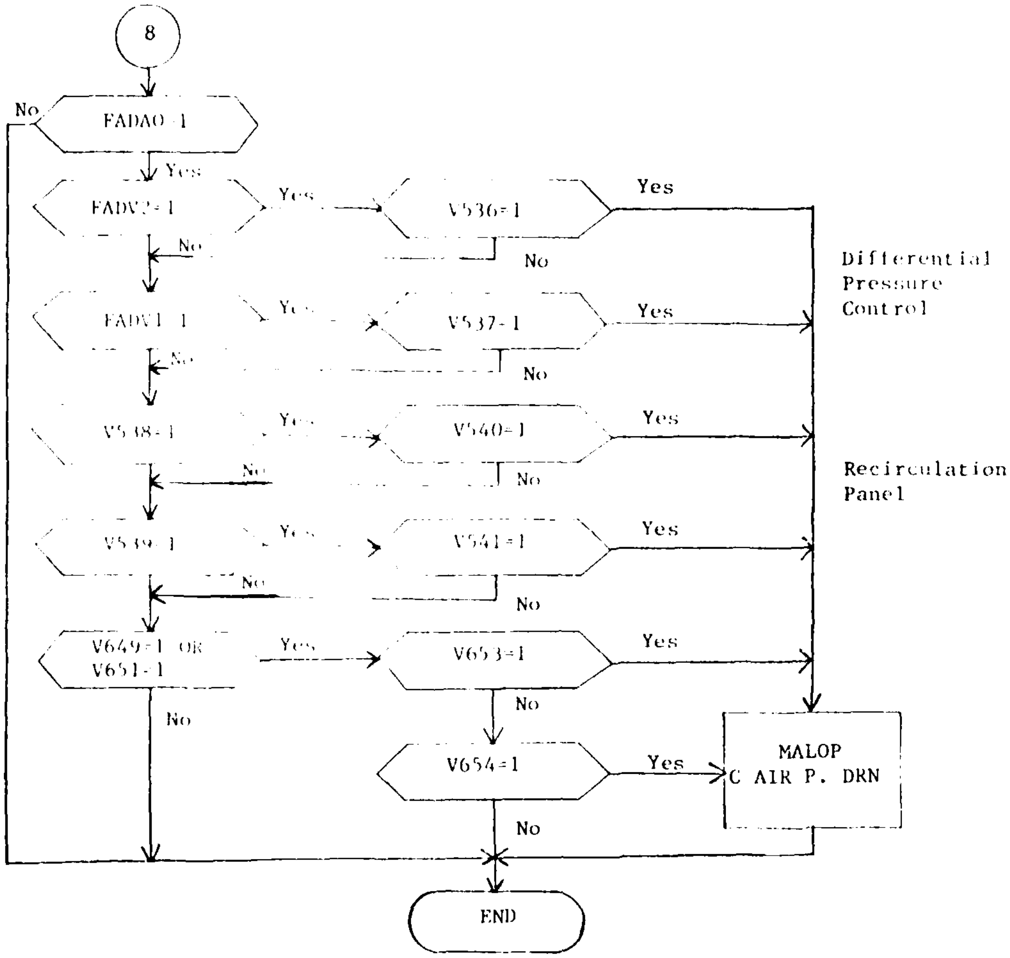


Figure 1A. Fire Room, Control Air (Sheet 12 of 12)

A STRUCTURED APPROACH TO MAN/MACHINE INTERFACE DESIGN FOR COMMAND AND CONTROL OF SHIPS' MACHINERY

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ABSTRACT

The complex modern warship which must operate in multiple roles and configuration in high risk situations, requires the design of the Man/Machine Interfaces to be concurrent with and given equal weighting to other aspects of its design. The system is so complex that the use of a structured approach to the design of the MMIs for the command and control system for its machinery is essential.

To meet the requirement for a structured approach to the process of MMI design, an algorithmic method is described. The algorithm used is not a true mathematical computational method but is a systematic approach whereby the MMI requirements are defined in a number of prearranged steps. A series of iterations through these steps is required as the MMI design is refined from the initial requirements to the detailed specifications, each step answering a specific MMI design problem.

Three basic design stages are identified between an initial design scenario and working up a design using mock-ups. These consider control spaces within the ship, control consoles within control spaces and panel designs for control consoles.

Part of the designer's task is to evaluate the cost and effectiveness of his design. A method of assessment is outlined which is compatible with the structured design approach.

Having designed a set of MMI options and carried out a cost and effectiveness analysis of them, the next stage in the design process is to evaluate them on a mock-up and resolve the remaining ergonomic problems by this means.

The algorithmic method proposed in this paper is a method of obtaining logically designed MMIs of high effectiveness.

INTRODUCTION TO THE MMI PROBLEM

MOD(PE) and EASAMS have been examining the requirements for design of the Man/Machine Interfaces (MMIs) for the Command and Control of Ships' Machinery in order to design the best interfaces in an age of increasing expense in manpower and increasingly competent automatic systems. The early stages of the work involved in defining the problem soon showed that the MMI design for warship machinery had much wider ramifications than straightforward control space and panel design. The interface and the man are system components in a hierarchy of systems interfacing with and reacting to command requirements in the normal state which will comprise a variety of operational modes. The systems must also be capable of control in a reversionary state when the best use must be made of the man's capability in this respect. The tasks for the man must be sufficient to keep him alert and interested yet he must be able to deal with a crisis situation at any time in his watch - for example, in the low period half an hour or so before his relief is due. An essential feature of the distributed machinery command and control system is good communications. The MMIs must allow for appropriate and reliable communications between systems, the command and the operator as well as for the control of men and machines which comprise the systems.

To ensure the most effective MMI design, the requirements must be considered from the earliest stage in the design. Only then can the aspects of control philosophy, requirements, methodology and manning be defined as the basic foundation on which the MMIs can be constructed. To this end an algorithmic approach has been adopted for use as a design method which ensures that the logic of the design encompasses all possibilities and that the manpower, plant, interface and ship designer's problems are given due examination and weighting.

A particular requirement for the MMIs in warships is the facility to provide an immediate reversionary capability after ship or system damage has taken place, and to strike the correct balance between men and automation, taking into account the naval need for an adequate number of men to fight the ship.

Command and Control of Ship Systems

Components. The command and control of ship systems is a hierarchical system involving two principle levels of activity. The highest command level is concerned with the overall policy and operating programmes. This involves balancing the requirements for the current tasks with the capability of the support systems for different situations. The lower control level is concerned with the operation of the ship's systems to meet the requirements set by the command decisions.

The higher level is defined by the overall command system and is embodied in the Captain of the ship and his command team. Operation of the ship systems is delegated to the system teams to control the system processes to meet the command requirements. In the machinery control context this involves operating the machinery and plant to meet the long and short term operational needs.

The Captain is in overall command, and embodies the main ship operating programme which he must operate in effectively a parallel process mode. He has a set of decisions based on his experience which are appropriate to many occurrences but when a situation arises that is not covered by an available decision he must be freed to concentrate on that situation by delegating other responsibilities to the members of his command team.

In addition to the task of maintaining the set of contingency decisions, the command structure has the function of routine operation of the ship systems, reconfiguring them from one mode to another as required by the command decision in order to obtain the results required. For each mode transition this involves planning and initiating the operation, controlling the transition, and terminating and checking the outcome.

This involves a number of features which must be incorporated in the MMI set, viz:

- Overall command value system
- Communications
- Command decisions
- Situation analysis and operational communications
- Planning
- Supervision and co-ordination
- Direct system operation
- System monitoring and switching
- Internal system functioning.

This command structure must respond to the two influences on the operation of the system, namely the behaviour of the environment and the behaviour of the system. Any decisions made must take account of both of these factors.

The problem of designing effective MMIs that take account of both human factors, and ship and ship system command and control factors is illustrated in Figure 1. This shows the overlap of human factors and ship factors which constitute the problem area. However, the design must also take account of a wider spectrum of human and ship factors in order to create effective MMIs.

The Approach to MMI Design

To meet the requirement for a structured approach to the process of MMI design an algorithmic method was adopted. The algorithm used is not a true mathematical computational method but is a systematic approach whereby the MMI requirements are defined in a number of pre-arranged steps. Each step in the design process can be identified and related to the preceding steps. It is not expected that each step would be capable of producing the correct solution immediately because required data could still be missing or ill defined. A series of iterations through the steps of the algorithm is required and as the process proceeds, the MMI design is refined from the initial requirements to the detailed MMI specifications.

The Design Algorithm. Three basic design stages have been identified. The design algorithm takes as its starting point a general description of the ship's role and proceeds to generate the proposed command structure. The first stage identifies the basic ship characteristics of size, shape and probable machinery based on the initial ship design scenario, projected ship roles and weapons. This stage of the algorithm involves the design work to a level of identifying the requirements for control spaces, their function, manning and configurations.

The second stage of the algorithm continues the process of outline design of control spaces but probes deeper into inter-relationships between spaces. The basic size and position of workstations and their manning requirements are identified along with the necessary communications.

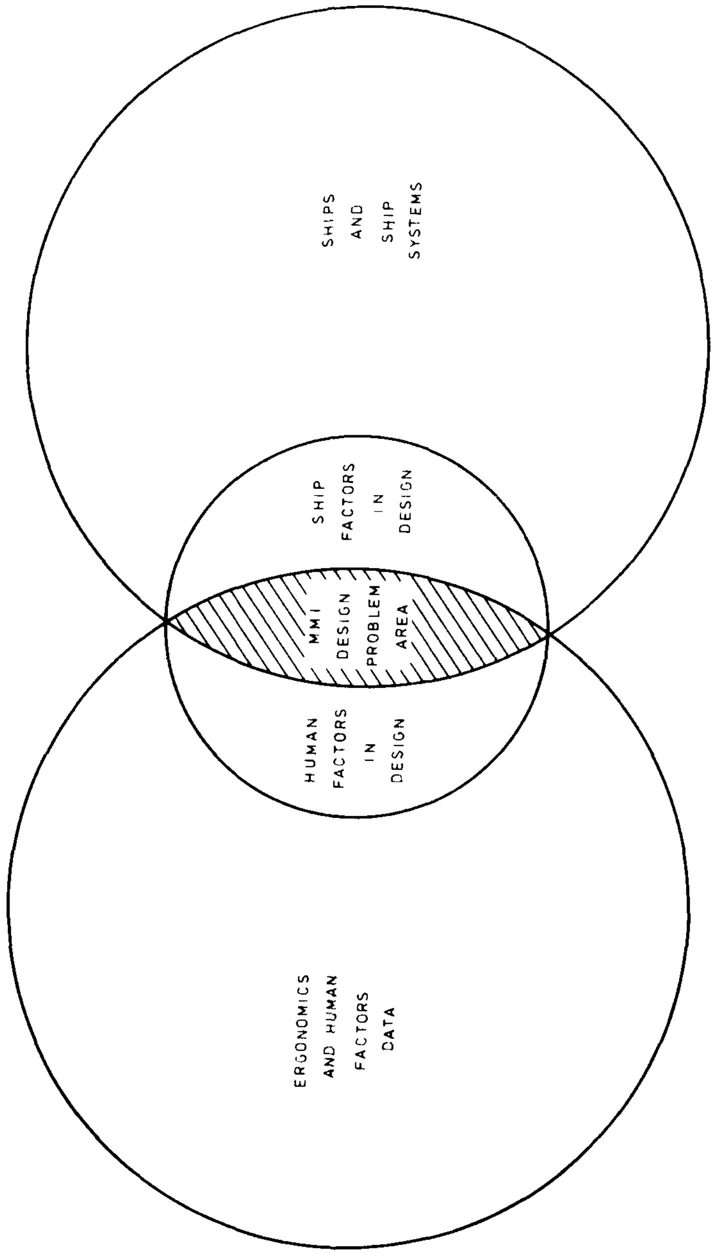

Figure 1. The MMI Design Problem

The final stage of the design algorithm is the detailed design of panels, taking into account ergonomic and engineering requirements and their integration into the overall command and control structure.

Cost and Effectiveness

Part of the designer's task is to evaluate the cost and effectiveness of his design and this requirement applies to the design of MMIs. However, the calculation of MMI effectiveness is seldom, if ever, rigorous since the designer must compare a number of differing aspects of dimensionally diverse natures, which may be ill defined or only partly perceived. This part of the MMI design process uses a method employing the logic of fuzzy sets. This accepts the numerically ill defined nature of the variables (i.e. their fuzziness) and produces a confidence factor for their linguistic grading. In the final analysis much must depend on the skill of the designer because proposed systems cannot be fully evaluated for cost and effectiveness in a formal sense, and the final arbitor must apply his own rules of sensitivity when making judgements. However, a structured approach to the method of comparing effectiveness of different solutions provides a valuable aid to the designer.

Mock-ups

Having designed a set of MMI options and carried out a cost and effectiveness analysis of them, the next stage in the design process is to evaluate them on a mock-up.

There will still be ergonomic problems in the final proposed design that are best resolved through the use of a mock-up and the amount of confidence the designer has in his selected option will dictate the type and standard of the mock-up.

MMI DESIGN METHODOLOGY

Whilst it is recognised that the actual design process will probably not proceed in a series of clearly defined steps each standing in its own right, the design methodology discussed here is intended to focus attention on the essential features that must be considered and the approximate order in which they occur. It is recognised that it will rarely be the case that a new system will be so innovative as to have no existing analogies. The methodology must therefore allow for the effect of the existence of knowledge regarding similar man/machine systems. Similarly it will often be the case that there are significant constraints imposed on the MMI by the rest of the design process. The methodology must also therefore be structured to provide the framework for collecting information about such constraints. The design algorithm involves an iterative approach, each stage drawing on initial options from the level below.

One of the objectives of this approach is to overcome the problem that the man presents to designers of control systems; that of allocating system functions to man or machine. As a purely functional component, replacing an item of hardware, man is noisy, intermittent, non linear and not very predictable. On the other hand, man is able to deal with unforeseen and undefined occurrences. Hence there is generally a requirement to incorporate man in the machinery system to be in command should anything unexpected happen.

The method of approach is to start at a level where no such problem of allocation of function between man and machine arises and the well defined man functions will be allocated to man and the machine functions to machines. The designer can then concentrate on the ill defined tasks, allocating them on the basis of his design experience or particular design requirements.

The Design Algorithm

The design algorithm takes as its starting point a general definition of the ship's requirements and using this information, proceeds through its three stages to generate the proposed command structure with emphasis on the control interface. Human factors and technological factors are combined in a series of iterations to produce design specifications from overall ship level to detailed functional panel level. Design variations result at the end of each of the stages and preferred solutions are selected from these.

These stages are shown in outline in Figure 2. Each stage draws on the options generated or known to be possible at the next stage and selects those that are suitable on the basis of the requirements of the preceding stage. The first stage consists of the steps which are shown in Figure 3. The principles of the stages are:

(a) To determine the purposes of the MMIs, define the machinery system, determine their operating pattern and derive a set of MMIs (control rooms, consoles, or panels) appropriate to the design stage.

(b) To analyse the effect of the design variables and assess their contribution to the MMI. Variables that are included cover manpower requirements, technology, sizing variables, siting variables, environmental and training factors.

(c) To assess the costs and effectiveness of each design option generated and to combine these assessments in order to select the best design options.

The power of the iterative procedure used by the design algorithm lies in the principle of requiring only sufficient detail of information at each step to be able to proceed to the next. The information obtained can then be developed in the light of knowledge gained from the later steps.

Each step of the algorithm has three parts: an input, a design process and an output. The input will be general design data and relevant data from the developing data base for the ship design from previous steps. The data are analysed in the light of the outputs of the previous steps of the algorithm to derive the desired solution. The output is the answer to the specific MMI problem and may be a single solution or several options. These then go forward to enlarge the data base and provide inputs for further algorithm steps.

The Design Stages

The MMI design algorithm takes as its starting point an outline description of the ship, its component systems and operational roles. Similarities to and departures from previous designs should be emphasised together with any general policy changes and special design criteria. Such specifications will indicate the general spectrum of

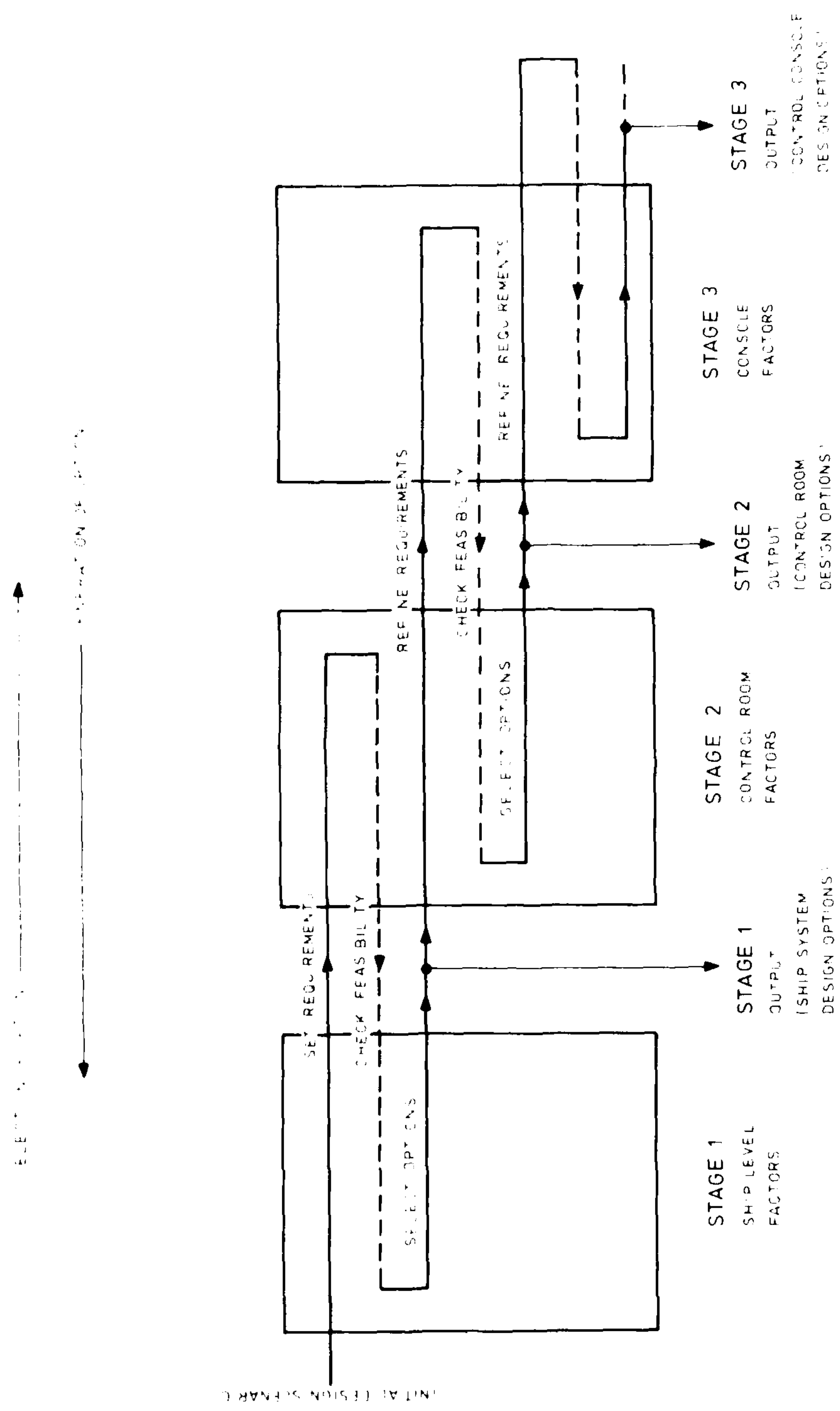

Figure 2 Algorithm Iteration Sequence

significant impetus for change to the design of machinery control consoles for naval ships. It is necessary, therefore, to briefly examine the nature of this impetus for change. The author believes that very often the specification of human engineering design criteria is a function of '.chnological and conceptual design limits and when either change. new human engineering criteria follow.

Existing propulsion control systems consist primarily of analogue electronic or pneumatic circuits for control of main propulsion machinery in order to prevent engine overspeed, shaft overspeed and underspeed, shaft overtorque, etc. Hardware logic is still used extensively for scan point alarm status surveillance. Control functions are physically located in the machinery control room and, in many cases, the control circuits themselves are designed into the machinery control console.

Future propulsion control systems, those that are being designed today, are likely to use mainly digital electronics and microprocessor technology in which control functions are dispersed throughout the ship in plant control units. Instructions from one or more primary control units would be received by plant control units via a highly redundant digital data highway operating at relatively high serial data rates. (14)

One of the many advantages in dispersed, as opposed to centralized, machinery control is that control units and machinery control consoles can be quite separate from each other and placed at any node along the data highway, including the bridge, operations room, MCR, engineering officer's cabin, etc. Furthermore, the separation of control circuitry and console hardware can result in significantly smaller and lighter consoles depending upon the display hardware chosen to implement the console design. It is interesting to note that the Italian LUPO-class frigate is the first naval ship to incorporate direct digital control of major components of the propulsion machinery system by using dual redundant centralized minicomputers and limited dispersed surveillance via microprocessor-based remote data acquisition terminals. (11) The console hardware, however, is conventional in that analogue meters and annunciator matrices are the primary sources of displayed information. Thus, although the control system has undergone a major change in design, the console itself has not.

ELECTRONIC DISPLAY SYSTEM TECHNOLOGY

Electronic display system technology has progressed to a state where major changes in machinery control console design are inevitable. Most, if not all, electronic display systems use microprocessors and semiconductor memory which have not only increased the intelligent capabilities of display terminals but have resulted in reduced system cost, reduced hardware size and weight, and increased reliability. Although a number of matrix display technologies are potentially available today, including plasma, electroluminescent, and liquid crystal panels, it is highly likely that CRT technology will continue to dominate the electronic display market in the near future by virtue of its relative low cost, high reliability, high degree of software support, availability of applications programs, and wealth of design pointed out in the ARL study (13), 70% of the instruments on some consoles are above the operator's eye height and, therefore, out of his primary field of view. The TROMP MCC design attempts to alleviate this problem by assigning frequently scanned instruments (engine and shaft parameters) to lower panels while placing infrequently scanned instruments in upper panels. In some console designs, however, viewing of annunciator windows on upper display panels is made difficult for the seated operator due to glare from luminaires.

Conversely, the watchsupervisor, standing or seated back from the display panel, can obtain an overall pattern of machinery status at the expense of not being able to read individual instruments unless he reduces his viewing distance or requests information from the watchkeeper.

Machinery Parameter Access

Other human engineering problems exist. In all MCC designs the console operator can select no more than one scan point for display on digital panel meters at any given time, thus placing an unnecesary memory load on the watchkeeper. For example, the DDH-280 MCC watchkeeper has available a single digital panel meter for display of scan point data. Although several frequently accessed scan points (engine parameters) have been assigned dedicated pushbuttons, the operator is required to remember individual scan point three-digit codes and to enter them via a thumbwheel switch. Many frequently accessed points are entered in this manner. Not only is this access method time consuming due to the manual operations involved, but it is prone to error, especially for infrequently accessed points which are nevertheless important under certain alarm conditions. No design is known in which the operator can select and display a small number of related scan points simultaneously.With their upper and lower limits, this feature would not only allow display of highly interacting paramters but it would allow some degree of operator control of displayed information. Different operators access different sets of scan points. At the present time, differences in operator preference for displayed information are not considered in the conceptual design stage of MCC development.

Integrated and Predictive Displays

Existing designs for machinery control consoles provide limited integration of information (with the sole exception of mimic displays), and provide no predictive information whatsoever. All machinery operating information displayed at the MCC is 'old' information and is, in itself, not formatted in such a manner to permit the console operator to predict future machinery states with much success.

It is the hoped in the remainder of this paper to demonstrate that the implementation of electronic displays, namely CRTs, in propulsion machinery control and surveillance has significant value in terms of ameliorating some of the problems just discussed.

THE IMPACT OF PROPULSION CONTROL TECHNOLOGY ON CONSOLE DESIGN

Developmental trends in both propulsion control design concepts and electronic display technology will, in the very near future, bring discussion is given of each of several general human engineering aspects of console design.

Grouping of Display Instruments

It is apparent that general human engineering principles related to grouping of display instruments by function, sequence, and priority have in the main been followed. In all cases, main propulsion system instrumentation (for engines, gearboxes, clutches, shafts, and propellors) appears to be separated from instrumentation for other machinery systems. In addition, instrumentation for each engine and shaft set is grouped separately. Thus, major functional groups of instruments are subdivided into minor groups by machinery subsystem.

Mimic Displays

Grouping of instrumentation in a literal physical format on a mimic display is generally thought to facilitate operator monitoring tasks; however, mimic displays have found limited use in naval ship machinery control console design (with the single exception of the Type 22 frigate). One author has gone so far as to comment with respect to the MCC of the TROMP-class ships:

"An attempt to position the operating and indicating instruments on their actual locations in a diagrammatic arrangement was tried at first, but this appeared to require too much panel space, and instructionally, was of limited value." (12)

From a human engineering perspective, however, mimic diagrams should be used where possible. This is because they reinforce internal models of the ship's machinery stored by the operator in his memory and thereby reduce memory workload. (See the section entitled Graphics Displays.) In addition, they are considered to facilitate operator training by helping the trainee to develop a model of machinery systems. (See section entitled Techniques for Reducing Operator Workload.) An experimental study by the Admiralty Research Laboratory in the United Kingdom of the effectiveness of mimic displays on submarine machinery control panels concluded that this type of display significantly reduces operator response times and error rates in identifying alarm conditions. (13) The major problem in the implementation of mimic displays with conventional process control instrumentation is that they do appear to require increased panel space. This is due in part to the size of commercially-available analogue meters and digital panel meters. It is also due to the fact that mimic displays by their very nature contain more information than that contained in conventional process control displays. As a consequence, conventionally implemented mimic displays impose extra visual workload on the operator because he is now required to scan a large area.

Patterns of Displayed Information

In all MCC designs reviewed, excepting that of the NITEROI, the watchkeeper views his displays from a distance of about one meter. Given that most display panels are no smaller than about 2m x 2m it is not unexpected that the watchkeeper is required to make excessive head and eye movements in order to scan display panels properly. It is difficult under these circumstances for the operator to obtain a clear pattern of displayed information and hence of machinery status. As achieve these objectives by considering operator information requirements in relation to system performance criteria during the process in which the display specification is developed.

EXISTING CONSOLE DESIGNS

A review of existing consoles for surveillance and control of propulsion and auxiliary ship machinery indicates that all design concepts are based upon the use of conventional process control instrumentation. (1) In particular, information displays consist of vertical and circular scale analogue meters, digital panel meters, annunciators matrices, and teletypewriters. Electronic display technology in the form of cathode ray tube (CRT) monitors or plasma panels, for example, has not been implemented in naval MCCs even though merchant ships have been using CRT displays for a number of years. (2,3,4)

Types of Machinery Control Consoles

Existing machinery control consoles can be divided into four different types.

MCCs of Canadian DDH-280, and US FFG-7 and DD-963-class ships are separate from the electrical console and are manned by a single operator. (5,6,7) This type of console is characterized by the fact that the watchsupervisor does not have a separate console, nor separate seating for that matter.

A second type of console design is that of the Brazilian NITEROI-class ship in which the watchkeeper is seated at a small console containing primary propulsion control and display instrumentation and monitors a number of small display panels located on a separate wall display at a viewing distance of perhaps two meters. (8) The watchsupervisory appears to be seated between and behind the electrical and machinery control console operators.

In the third type of console design, both the electrical and propulsion instrumentation panels are integrated into a single large console manned by two or more watchkeepers, depending on running conditions. Examples of this type of console are found in the machinery control rooms of the Royal Navy's Type 21 AMAZON, Type 42 SHEFFIELD, and Type 22 BROADSWORD-class ships and the Italian Navy's LUPO-class ships. (9,10,11) The display panels of the Type 22 frigate are particularly interesting from a human engineering viewpoint due to the extensive use of mimic diagrams.

The fourth type of MCC design is exemplified by that in the Royal Netherlands Navy's TROMP-class GM frigate. This design is characterized by its compactness and limited use of analogue instrumentation as a result of the application of the 'darkboard' principle and a decision not to use mimic diagrams. (12) The engineering OOW, together with the NBCD OOW and the electrical OOW, are seated on a raised platform directly behind the main console operators.

Human Engineering Aspects of Existing MCC Designs

No attempt has been made here to present an exhaustive coverage of human engineering aspects of existing MCC designs. Rather, a brief

HUMAN ENGINEERING CONSIDERATIONS IN DESIGN OF INFORMATION DISPLAYS FOR MACHINERY CONTROL CONSOLES OF FUTURE NAVAL SHIPS

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ABSTRACT

As the Canadian Patrol Frigate project is developed, studies are being undertaken by the Canadian Forces of a machinery control system which will be significantly different from control systems in existing naval ships. A conceptual design has been developed in which a fully digital control system will support the implementation of electronic information displays. Such systems offer unique capabilities for information display as compared to the capabilities of conventional process control instrumentation.

This paper discusses some of the major human engineering considerations relevent to the design of information displays for naval ship machinery control consoles. Existing machinery control console information display concepts are assessed from a human engineering perspective. Information requirements of the watchkeeper as supervisory controller and of the watchsupervisor as executive supervisory controller are discussed. In particular, emphasis is placed upon basic techniques for reducing operator workload in the context of CRT display systems. Lastly, a brief description is given of both the graphic and alphanumeric display pages which might be implemented in a machinery control console having only CRT displays.

INTRODUCTION

The Defence and Civil Institute of Environmental Medicine (DCIEM) has undertaken an investigation of propulsion machinery control console (MCC) design in support of the Canadian Patrol Frigate programme. In particular, we are interested in human enginering considerations which apply to the design of a complex man-machine interface such as an MCC. Such considerations derive not only from empirical studies of operator behaviour in control and surveillance systems but from trends in ship propulsion control and electronic display technology.

The purpose of this paper is to discuss some of the major human engineering considerations relevant to the design of information displays for naval machinery control consoles.

Information displays are fundamental components of all machinery control and surveillance systems. The function of an information display is to present process data in a coherent and meaningful format, both spatially and temporally, such that the information required for each operator task is readily available without inducing adverse levels of operator workload. The goal of human engineering is to

The time needed to build and set up a mock-up may be critical to its effective use. A mock-up, however elaborate, is useless if it cannot be produced in time.

The costs of mock-up work include not only the provision of the equipments but also the time of the people working on it. These costs must be assessed in terms of the benefits to be obtained and the overall project costs.

Mock-up work can be expensive, so it is essential to fully evaluate the use to which it will be put, bearing in mind that mock-ups are but one of a number of methods of representing and manipulating a design, albeit an essential one for complex control spaces.

APPLICATION

This method of defining the MMI requirements has been applied with considerable success, to an unconventional ship being developed for the Royal Navy. Although the concept was introduced late in the design process, part of the structured approach was used to define the machinery control MMIs for the bridge, operations room, machinery control centre and some local positions. The methodology is currently being used in the design of the MMI requirements for a warship in the mainstream of development. Application of the methods to these two very different ships has allowed the methods to be refined to the stage where they are ready for publication as a basis for design procedures.

Staff who have been associated with this project have included Marine Engineers, Human Factors Engineers, Naval Architects, Mathematicians and Physicists - all systems engineers applying a systems approach to the problem of the interface and balance between men and machines in ship machinery command and control - which can well be applied to many similar tasks.

CONCLUSIONS

The design of the MMIs of the complex modern warship required to operate in multiple roles and configurations in high risk situations must be concurrent with and given equal weighting to other aspects of its design. The system is so complex that the use of a structured approach to the design of the MMI for the command and control systems for its machinery is essential.

The algorithmic method proposed in this paper is one such method and leads to logically designed MMIs of high effectiveness.

Mock-ups exist as a means to an end and should not be regarded as an end in themselves - that is, they should be used as design tools and not as full scale models of design solutions. In the early stages of their use changes should be simply incorporated, the mock-up becoming a fixed representation only when all the difficulties of the design have been solved and the design itself is fixed.

During the experimental phase, the mock-up is used to evaluate general design parameters by, for example, measuring human performance at various representative tasks. It is necessary, however, to be aware of the danger of transferring the results of experimental studies in a laboratory to the real life situation, so care is necessary in designing the experiments and assessing their results.

A mock-up is an aid to evaluating a specific system layout in three dimensions and is an effective method of identifying any design problems which might arise purely from the physical size and shape of equipment items.

During the validation phase, the mock-up is used not only to check the various standard operating procedures, but also to verify that all operations can be carried out and that performance values do, in fact, correspond with those expected. Operator workloads can also be monitored to check that the number and skill levels of the operators are correct.

A mock-up representation of the actual system may be used to teach people how to operate the MMI. The emphasis on its use has transferred from mechanical aspects of design layout, through the experimental evaluation of operators' behaviour to use as a teaching aid for potential operators.

Once a MMI has been designed, the user must be persuaded to accept it and a mock-up is an effective means of explaining and demonstrating the advantages of a new design. When a mock-up has been used in the evaluation phases, it is probable that it will have far more impact than other methods in reconstructing rejected features in order to demonstrate the reasons for choosing the final design. In the final stages, it is desirable that the user's representative be fully involved in the detailed design work connected with the MMI so that he can readily accept the arrangement on completion.

The effectiveness of a mock-up as a design tool depends on a number of factors including:

- Fidelity
- Flexibility
- Speed of set-up
- Cost.

Fidelity covers the degree to which a mock-up can be regarded as representing the particular aspects of interest. The degree of truth in the representation affects the validity with which the experimental results can be transferred from the mock-up to the final system.

Flexibility covers the extent to which various configurations of MMI layout and design can be tried out.

has been found that in practice, costs are just as difficult to define exactly. The problems are multiplied when one tries to relate the two to provide a single measure on which a decision can be taken.

The assessment of effectiveness for all the options must be based on the same criteria but it is complicated by the ill defined nature of the criteria. Various methods of evaluating the effectiveness of complex systems are available, e.g. the single parameter method or the cardinal number ranking method. However, they are unsatisfactory for MMI purposes where the components of effectiveness are ill defined, only partially conceived, of different dimensions or non-dimensional and often derived by subjective assessment.

Quantitative measurement of MMI effectiveness is fraught with difficulty and there remains a need to manipulate qualitative linguistic variables using operations of logic to combine such effectiveness assessments as can be made, while allowing the use of quantitative data, and produce a measure that can be related to cost.

These problems resulted in the need to develop a new approach to the effectiveness problems which makes use of the concepts of fuzzy set theory. Fuzzy set theory provides a method of logical analysis whilst admitting the possibility of partial membership in a conventional set and so allows the possibility of building a decision process that can handle imprecise variables such as verbal descriptions. The components of effectiveness are individually assessed and these assessments are then combined together by a fuzzy algorithm to provide an overall assessment of effectiveness.

This assessment of effectiveness is then combined with the estimates of cost to produce the overall cost and effectiveness assessment of each stage of the design process to eliminate options that are unacceptable on grounds of cost or effectiveness. The reason for selection is incorporated in the specifications for design options that are carried forward for more detailed work.

USE OF MOCK-UPS AS A DESIGN AID

Mock-ups are a useful design aid which can be used at a number of levels. Consequently, it is necessary to evaluate the costs and benefits before deciding to proceed to a mock-up stage.

The function of the mock-up is to determine the mechanical and human factors engineering aspects of the system design, and to assess, in three dimensional terms, the compatibility of the design and operator interactions with the controls and displays. The mock-up can assist the designer and the user to fully appreciate that the design is complete, appropriate to the space into which the equipment is to be installed, and provides a reasonable work station for the operators. The mock-up can be used in validating the level of task automation in the context of operator workload and capability.

Mock-ups are used as design tools for:

- Experimentation
- Evaluation
- Validation
- Teaching
- Implementation.

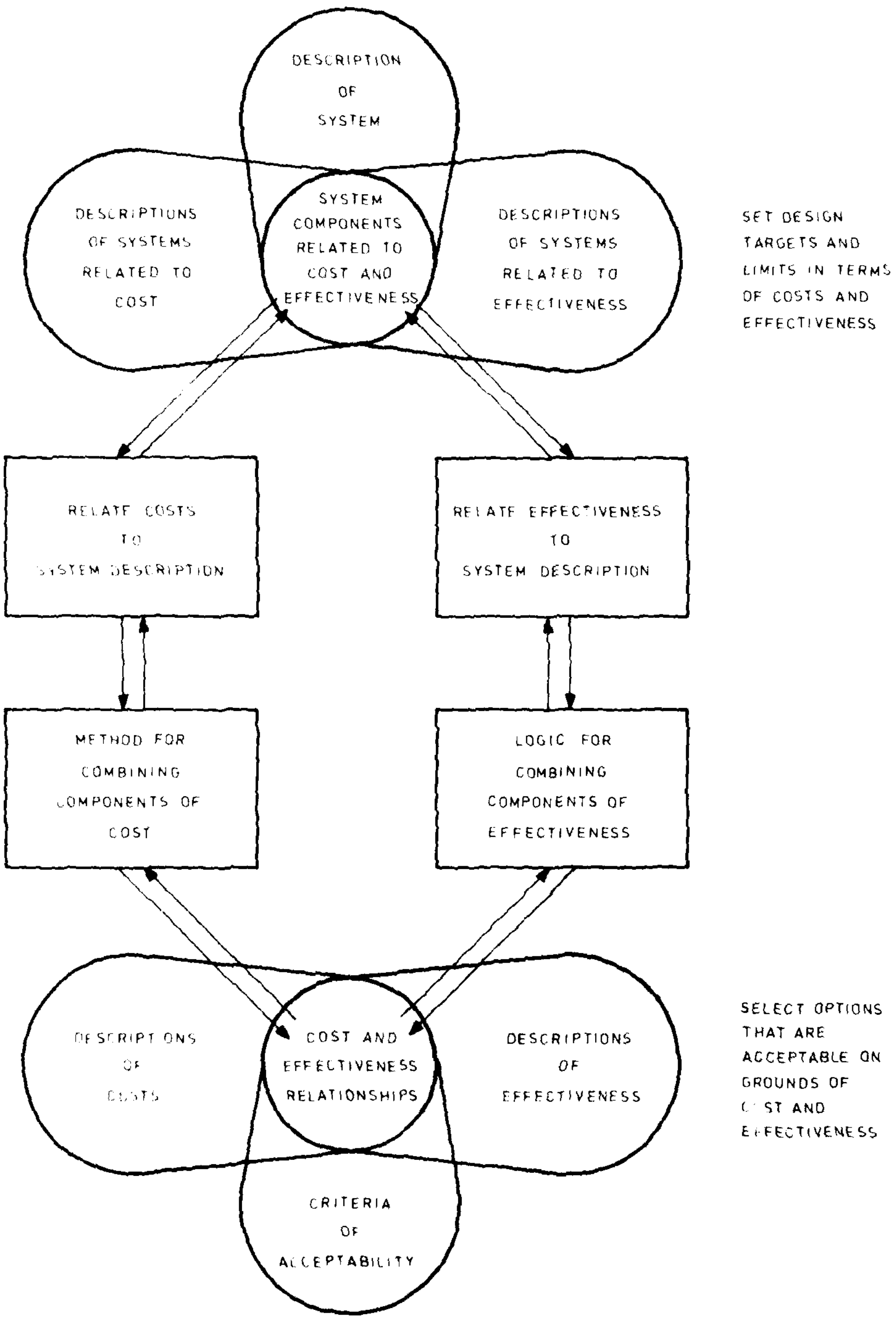

Figure 4. Assessment of Cost and Effectiveness

COST AND EFFECTIVENESS EVALUATION

This evaluation must be part of an on-going process, as the designs evolve using assessments of cost and effectiveness to reject possibilities which do not meet requirements in these respects.

When evaluating the cost and effectiveness of an option, acceptability is judged in terms of whether it will fulfil several criteria simultaneously. Relationships within the set of cost components and within the set of effectiveness components are assessed independently during each design stage and only combined to produce the cost and effectiveness assessments at the end of each stage, as shown in Figure 4.

A philosophy on cost applicable to this structured approach to MMI design is that between each stage a reasonable maximum cost limit should be set. The purpose of this is that considerations of cost are initially objective, i.e. independent of effectiveness criteria. Decisions about value for money are then made on the basis of cost savings in relation to decrements in effectiveness from the maximum effectiveness achievable at the cost ceiling.

Components of Cost

Components of cost which are considered include the following: procurement, maintenance and repair, training, running costs. These could be further sub-divided, e.g. training for maintenance and repair, and training for operation, or running costs for manpower and for fuel and other resources. A further sub-division might consider the costs associated with the trainees, instructors and training overheads. The design may be rejected if any of these components are unacceptable.

Components of Effectiveness

The actual choice of components of effectiveness depends on a number of factors. The design stage at which the assessment is being made will influence the choice; different criteria are relevant when dealing with ship level factors, control room level factors, work place or console level factors and panel component level factors.

Effectiveness assessments of an interface after each stage of the algorithm are aimed at determining the extent to which that interface will perform the operational requirements of the interface designed at the stage before. Components that have been considered at the control room/console level include:

- Safety from errors and omissions in operation
- Use of space in operation
- Ease of co-ordinating activity
- Independence of control/display groups
- Required skill of operator
- Adequate grace time for operator response to demand
- Control/display integration/compatibility.

In general the cost and effectiveness variables that are relevant to the design of MMIs defy precise measurement or accurate prediction. While this may be understandable for the effectiveness components it

(b) Secondary surveillance — those parameters which must be examined to provide information for plant health and trend monitoring or maintenance requirements.

(c) Maintenance — the parameters which must be logged to define maintenance tasks and those used in carrying out maintenance including setting up the system on completion of maintenance.

The parameters may be required for more than one of these purposes.

In addition to the categories above, allowance must be made for the need for manual or automatic data logging, alarm and warning systems and specialised safety devices.

Particular features to be considered at this stage are:

- Machinery operating routines and procedures
- Manpower requirements to work the interfaces
- Details of technology for control and display
- Overall work space and console sizing
- Control space layout, taking into account, communication links and other general relationships between work spaces and consoles, allowing access for servicing either the machinery or the interface
- Local lighting levels and protection from distractions
- Training requirements and personnel selection criteria.

Stage 3 of the Algorithm. This stage in the design algorithm is concerned with the detailed layout of the panels. Once the broad function of each console has been defined and the overall layout of the work space has been decided, the design of the panel layouts can be carried out, taking note of manual operational factors such as attentional transfer, detailed operating procedures, light sensitivity, individual control/display relationships, and the detailed instrumentation requirements for the plant.

This third stage, which leads to the detailed hardware design requirements is virtually a reiteration of the previous stages but going to a depth of detail necessary to finalize the design. In order to resolve any design difficulties and if it is warranted by the number of ships or the importance of the control system, this stage should be concluded by a mock-up of the control consoles when complete details of the control and display parameters and the uses to which the console will be put should be available.

The output of each stage of the algorithm may be a single design although it is more likely to be a set of options of comparable merit. To select a final design these options must be evaluated for cost and effectiveness.

which the ship can be operated in a number of modes. These modes, for the general warship case, were reduced for analysis to five representative cases:

- The action state
- The wartime passage state
- The manoeuvring state
- The peacetime passage state
- The harbour state.

These represent the most arduous state, states with medium levels of activity and the most relaxed situation for the prime task system. The MMIs must cope with all of these and any intermediate state.

Features to be considered at this stage are:

(i) System operating modes, e.g. critical operations, normal operations, secondary surveillance, maintenance.

(ii) Manning policy, e.g. naval regulations, action and damage control requirements, need to match the task to the man.

(iii) Control technology, e.g. automatic control level, pneumatic, hydraulic or electronic, reversionary and safety requirements.

(iv) Control space configurations, e.g. combined or separated control spaces, security requirements, career experience implications.

(v) Siting of control spaces, e.g. access to plant, nuclear protection, vulnerability considerations.

(vi) Environmental factors, e.g. noise and vibration, temperature and humidity, ventilation and other atmospheric protection, overall lighting.

Stage 2 of the Algorithm. This stage in the design algorithm is concerned with the overall layout of each control room or work space. The outline design derived from Stage 1 is examined, assessed and revised to a further level of detail to produce a refined control space layout, the basic display and control methods, and definition of such factors as the number of consoles or panels and the information to be displayed on each panel.

The design algorithm at this stage provides system descriptions at a level of detail sufficient to define which machinery systems are associated with each work place and whether the controls and displays are primarily for operation, surveillance, or for maintenance purposes. Each work place comprises some of the following displays and controls:

(a) Controls and primary surveillance : those controls and displays which represent system characteristics which are either directly under the operator's control or which cause major functional change of the plant in the event of a failure.

suitable machinery, manpower availability, and the capability required of the ship.

Stage 1 of the Algorithm. This stage in the design algorithm is concerned with the set of control spaces required to operate the ship, the distribution of command over these control spaces, definition of the systems controlled from each space and deriving design requirements for each space.

The ship's systems are assigned to the functional groups defined below and dependencies between the groups are established.

The functional groups are:

- Command
- Prime task
- Manoeuvring
- Common support
- Life support
- Environmental defence
- Secondary surveillance and maintenance.

Each of these systems comprises machinery with manual or automatic operation and surveillance together with inputs from other systems. There will be dependencies between systems in each of these areas which must be catered for by the MMI control space set. Each control space will contain some of the following command and control interfaces:

(a) Primary Interface : that interface containing all the centralised controls and displays relevant to the machinery system in question, e.g. Machinery Control Room (MCR).

(b) Secondary Interfaces : interfaces containing controls and displays that can be used as alternatives for command or control, e.g. secondary steering position.

(c) Reversionary Interfaces : interfaces with the minimum of control technology consistent with sound ergonomic practice for use when primary and secondary interfaces fail, e.g. engine local control position.

(d) Local Interfaces : basic machinery controls used normally for maintenance purposes, e.g. hand throttle control.

The purpose of this stage is to define those interfaces required for each ship system and assign them to control spaces. Purely mechanical considerations will dictate some of these assignments, particularly the local interfaces, and the remainder will be affected by the operational requirements of the ship's prime task. A warship can be considered to have a number of main roles or functions for

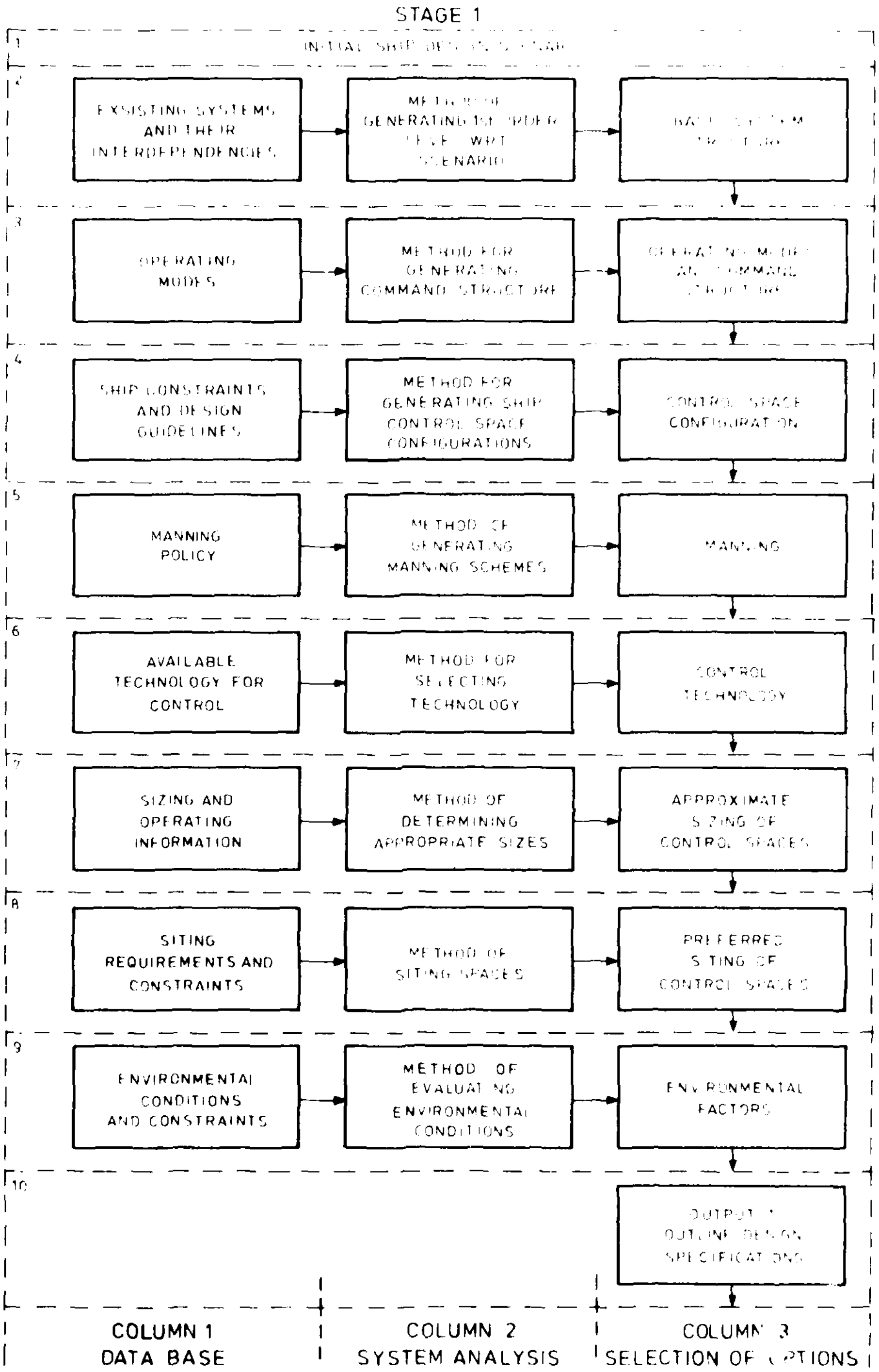

Figure 3 Design Algorithm

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experience. (15)

The question of whether or not electronic display systems, specifically CRT display systems, are feasible for control and surveillance of propulsion machinery is best answered by observing that conventional display instrumentation techniques have been supplemented and often replaced by CRT displays in many land-based process control industries such as mining, petroleum refining, and electricity generation and transmission. (16,17) Modern process control consoles are not only smaller, lighter, and more reliable than conventional consoles, but they permit the operator use of advanced interactive features that are simply not possible with conventional process control instrumentation. These interactive features require extensive human engineering research because they introduce design criteria which are not applicable to conventional system specifications.

CRT DISPLAYS IN MACHINERY CONTROL CONSOLE DESIGN

The implementation of a CRT display system consisting of CRT monitors, display generators having both alphanumeric and graphics capabilities, micro or minicomputer control, and disc display page memory, together with appropriate graphics software, constitutes a powerful display technology and permits the designer considerable latitude in developing a specification for machinery control console displays. In developing that specification, a number of potential system capabilities must be examined including:

- display page organization techniques;
- page selection techniques;
- data storage and retrieval;
- dynamic page information coding formats using colour and luminance highlighting;
- input command techniques;
- interactive man-computer dialogue providing visual feedback;
- enhanced alarm displays;
- integrated graphics displays;
- trend and predictive displays;
- mimic displays.

Each of these capabilities must be studied with reference to analyses of watchkeeper and watchsupervisor information requirements. Of equal importance in the development of a specification for a CRT display system is the human engineering consideration of operator workload.

The Nature of Operator Workload

If an information display system is to be used effectively by a supervisory control operator having a reasonable amount of related experience and job skills, then he should be able to monitor and access displayed information without incurring adversely high error rates under conditions of stress. Obviously the workload imposed by the display system must be optimized.

Operator workload is a factor often overlooked by display system designers because each display task, when examined in isolation, is relatively easy to learn and perform. Workload increases substantially, however, when a number of tasks must be performed interactively and virtually simultaneously to achieve a larger operator function.

Operator workload can be divided into at least three components; perceptual load, memory load, and cognitive load.

Perceptual load is determined by the amount of information which must be acquired by the operator during a single monitoring cycle or scan of the display surface.

Memory load is the amount of information which must be recalled by the operator from his memory system in order to decide whether or not the information available from the perceptual process satisfies machinery status operating criteria. The information available for recall and which specifies, amongst other things, the layout of machinery in the ship, the dynamics of machinery operation, and acceptable ranges levels of operating parameters, is called his 'internal' model of the machinery system. (18)

Cognitive load, on the other hand, is determined by the complexity of the algorithms, or sets of rules, which the operator uses in order to compare his internal model with information available from perception.

Techniques for Reducing Operator Workload

CRT display systems can significantly reduce operator workload through implementation of techniques such as:

- providing efficient methods for acessing individual CRT pages and locations on a page;

- providing a small amount of primary or high level information on a small display surface during routine machinery system surveillance;

- providing the operator with graphical displays of machinery subsystem structure;

- providing integrated displays;

- providing the operator with appropriate alternative responses under alarm conditions;

- providing limited operator control of information display configuration.

Accessibility of Displayed Information

Accessibility of information in the context of CRT display systems is defined to be

(a) the ease with which the console operator is able to select a desired display page from amongst the the many pages which comprise the page library, and (b) the ease with which the operator is able to select any location of a given CRT page using an electronic pointer.

The most appropriate technique for selection in both instances is dependent on the specific operator task. For a system having a number of display pages which significantly exceeds the number of display screens in the console the display designer must construct a hierarchical display page organization to which the operator can refer for page selecton purposes. The problem of availability does not exist in conventional process control display systems.

Primary Information Display

Since most perceptual input causes memory and cognitive processes to function as well, excessive workload is incurred by displaying large amounts of unnecessary information, especially when it is presented over a large display area. Thus, overall workload can be reduced substantially by restricting information input to that which must be continuously monitored during routine machinery operation. When abnormal operating conditions occur, the operator will selectively increase the amount of displayed information to a level appropriate to perform the required corrective action. The display system designer, therefore, must attempt to structure the presentation of information to permit the operator to accurately select only the information required for for each corrective action. CRT display systems are well suited to implement this workload reduction technique.

The application of the darkboard principle to the design of MCC display panels of TROMP-class ships is an analagous technique for achieving workload reduction. This principle states , in essence, that normal operating conditions are evidenced by null or 'dark' annunciator windows. In practice, however, the application of this principle has been limited to keeping the number of illuminated windows to a minimum. Many of the annunciators , in fact, indicate normal operation when illuminated. The evaluation of visual input is difficult under these circumstances. However, the darkboard principle is certainly a useful one. It does, however, require careful design in order to maintain operator confidence in the reliability of the display panel.

Graphical Displays

Graphical, as compared to alphanumeric, displays reduce both perceptual and memory workload. The functional and spatial relationships between various machinery systems and components are inherent in graphics displays. A large number of relationships can be shown on a single display surface. Furthermore, graphics displays employ different coding dimensions which are nonverbal in nature and, therefore, convey information very rapidly to the operator. With the exception of mimic displays, conventional machinery control console displays do not supply the watchkeeper with these kinds of relationships. Therefore, the operator is forced to recall them from an internal model of machinery systems which is held in his memory. Graphics displays, whether on conventional display panels or on electronic display screens, reduce memory load by reinforcing this internal model.

It is postulated that graphics displays not only reinforce the internal model but make it less abstract by reducing the apparent remoteness of the operator from machinery systems. This form of remoteness is a function of both geography (ie, the watchkeeper performs a supervisory control role within the confines of the machinery control room) and control (ie, the operator and the machinery he controls are separated by a 'layer' of automated control technology in which are imbedded many closed loops invisible to the operator).

Integrated Displays

An integrated display combines information from two or more sources on a single display surface such that the information is perceptually integrated by the operator into a single pattern. This type of display significantly reduces perceptual and memory workload.

"A good integrated display is one that is compatible with the operator's internal model of the control situation." (19)

Two types of integrated displays which could be useful in a machinery control console having CRT displays are a 'propulsion overview mimic' graphic and an 'engine status multiparametric' graphic. The former is simply a literal representation of engine, gearbox, clutch, shaft, bearings, and propellor appropriately drawn on the CRT and superimposed with static and dynamic alphanumeric data. The latter is a form of graphic in which a number of parameters, each represented by a pattern segment, are combined together to form a larger integrated pattern. (20) A parametric graphic permits the console operator to rapidly detect the onset of abnormal operating conditions simply by observing that the overall pattern is distorted.

Alternative Alarm Actions

Operator cognitive workload can be reduced if the 'intelligence' of the man-computer interface per se is utilized to assist the watchkeeper in problem solving and decision making. For example, it is possible to provide the watchkeeper with updates on maintenance requirements generated by a machinery health monitoring system or to present alternative courses of action upon the occurrence of an alarm condition. This design feature is not part of existing MCC configurations.

Operator Control of Information Display Configuration

The fit between the presentation of information and the operator is a dynamic and interactive one. The operator can have different information requirements as a consequence of currently displayed information and different operators can have different preferences under essentially similar operating conditions.

"Every system design activity must contain the two complementary activities of adjusting the hardware interface to the man and adjusting the man to the interface by selection and training. Since training by experience continues indefinitely the presentation of information must be sufficiently adaptable to cope with these changes. There are two main ways of doing this. One is to provide a large and redundant array of data in the knowledge that the operator can select what he requires and can change his selection or his weighting when he so wishes. The other is to provide him with stored data and to allow him to interrogate the store according to his needs......Such a degree of flexibility can cope not only with differences within individuals but also differences between individuals." (21)

This requirement can be implemented rather easily in a computer- based CRT display system by allowing the operator to select any combination of machinery components or scan points for display on a CRT page called the 'operator monitor'. Furthermore, this technique relieves the operator of displaying machinery system parameters singly and serially, a problem which is described in an earlier section entitled 'Machinery Parameter Access'.

INFORMATION REQUIREMENTS OF MACHINERY CONTROL CONSOLE OPERATORS

The logical first step in developing a human engineered display system is to ascertain the information requirements of the watchkeeper and watchsupervisor. These requirements are determined by the set of tasks which each operator performs; however, any framework for information requirements (19,21) would include:

- information about policy and objectives;
- information about alternatives and consequences;
- information about the current state of the system.

Conventional display systems provide information about the current state of the system only. Information pertinent to policy, objectives, alternatives, and consequences is usually acquired during operator training. The only inputs which are received by the watchkeeper from other than machinery sources are ship's speed orders, station-in-control requests, and special instructions from the watchsupervisor.

The implementation of electronic displays in MCC design permits the display system designer to go beyond the constraints imposed by conventional instrumentation. Each display surface is inherently multipurpose and this capability, coupled with an effective data base management system, a machinery health monitoring system, and predictive alarm algorithms, allow the display system to present the watchkeeper and watchsupervisor with information about future alarm states, alternative courses of action or maintenance which would reduce the probability of occurrence, and consequences of not taking recommended corrective action.

In most systems, console displays serve their major purpose by providing information to the operator concerning current status of various machinery systems. Status information includes not only the current values of machinery parameters, however, but also the manner in which current values are changing. For example, a trend display of exhaust gas temperature (EGT) for a port main engine gives the operator information about what the EGT has been and where it is likely to go. The predictive alarm algorithm is, in this case, part of the internal model of the dynamics of the ship's machinery control characteristics stored in the operator's memory.

In work being undertaken in development of design guidelines for machinery control consoles of the Canadian Patrol Frigate, information requirements of both the watchkeeper and watchsupervisor have been tentatively identified. These information requirements have been based upon a review of DDH-280 machinery control room watchkeeper/watchsupervisor functions and upon examination of the control system requirements for the Canadian Patrol Frigate propulsion machinery.

The information requirements of the watchsupervisor differ somewhat from those of the watchkeeper. This difference is reflected in their respective roles. The central role of the MCC operator is that of supervisory controller.

"Supervisory control is control by a human operator of a computer which, at a lower level, is controlling a dynamic system. In such systems, computer control normally operates continuously or at high data rates in loops closed through electromechical sensors and motors. By contrast, the human operator normally signals or reprograms the computer intermittently or at a much slower pace. The human operator handles the higher level tasks and determines the goals of the overall system." (22)

The watchsupervisor, on the other hand, plays an executive supervisory control role in the machinery control room. He performs a general monitoring function for both machinery status and operator 'status' whilst using his comprehensive knowledge of system functions to solve intermitent and often complex problems.

The two different roles imply that:

- the watchsupervisor will access higher level information than the watchkeeper during executive supervisory tasks;

- the watchkeeper and watchsupervisor will access information in different orders and at different times; and

- the watchkeeper and watchsupervisor will require independent access to stored information.

Although the watchkeeper and watchsupervisor have different requirements for types of and access to information, the majority of displayed information is common to both.

The primary information requirement is to have continuously displayed a small amount of highly integrated information which permits the display user to quickly determine the status of main and ancillary propulsion systems. This requirement can be achieved with the implementation of two types of display previously discussed:

1. Propulsion Mimic Overview Display

2. Recent Alarms Display

The propulsion mimic overview could contain the following information integrated around a single graphics display:

- mimic display of main propulsion machinery;

- single block representation of major ancillary propulsion systems;

- general propulsion system alarms;

- multiparametric engine status display;

- alphanumeric display of
  :ordered, reply, and actual ship's speed
  :shaft RPM and propellor pitch
  :station-in-control

The *recent alarms* display is an alphanumeric presentation of the following kinds of information for the most recent, say, eight alarms:

- type of alarm;

- subsystem alarmed;

- cause of alarm;

- present status of alarm;

- time of occurrence of alarm.

The recent alarms display is updated as new alarms occur. In addition, it is envisioned that messages announcing new alarm conditions could be supplemented with alarmed subsystem graphics displays in order to reduce operator workload during alarm handling.

The *second major* information requirement of the watchkeeper and watchsupervisor is for displays pertinent to specific tasks which follow from monitoring of continuously available display pages. These displays include auxiliary machinery status graphics and alphanumerics displays, analogue engine instrumentaion displays for engine starting, trend displays, and a display of machinery parameters selected at the operator's discretion, (ie, the 'operator monitor').

CONCEPTUAL DESIGN OF CRT MCC DISPLAYS

Information requirements of the watchsupervisor and watchkeeper indicate a need for two operator consoles. These consoles have been designated as:

1. Main Machinery Control Console
   - assigned to the watchkeeper

2. Supervisory Machinery Control Console
   - assigned to the watchsupervisor

The Main MCC requires three CRT monitors. Two are for continuous display of the propulsion overview and the recent alarms list. The third is for display of additional task-specific information. (The monitor used for display of recent alarms can also be used for display of task-specific information when required.) Tentative assignment of pages to CRT monitors is as follows:

Monitor #1 (positioned at right side of console)

- recent alarms list
- new alarm message + subsystem graphic in split-screen format

Monitor #2 (positioned in centre of console)

- propulsion system status overview (continuously displayed and never overwritten)

Monitor #3 (positioned at left side of console)

- operator monitor
- trend displays
- analogue engine instrumentation display
- special task displays (eg. maintenance procedures for replacement of faulty PCB)

The Supervisory MCC requires two CRT monitors. They can be assigned display pages 'n any order desired by the watchsupervisor. Presumably, one will be used for display of the propulsion overview while the other will display trends, status reports, alarms, etc.

It is intended that the two consoles be completely independent insofar as operator access to page displays is concerned; however, it is conceivable that the watchkeeper will not have access to the complete library of pages available to the watchsupervisor. Conversely, control inputs at the Main MCC can be changed by the watchsupervisor from the Supervisory MCC but the watchkeeper will not be able to override any inputs at the Supervisory MCC. Console control and display interlocks in software are under control of the watchsupervisor in order to allow the watchkeeper to occupy the Supervisory MCC should the Main MCC displays or keyboards become unavailable. Much detailed analysis and simulation will be required to finalize page assignments and console interaction.

CONCLUSION

Existing consoles for surveillance and control of ship propulsion machinery are based upon the use of conventional process control instrumentation. Electronic display technology has not been implemented in naval machinery control consoles although merchant ships have been using CRT displays for a number of years.

Trends in propulsion machinery control system design and increasing sophistication in electronic display system hardware and software design suggest that CRT displays are feasible in machinery control consoles for future naval ships. It is argued not only that future MCCs will be smaller and lighter than their predecessors but that the human engineering advantages are considerable. The problem of the designer is to exploit the capabilities of modern display technology whilst implementing human engineering practices which are to a large extent a function of the technology itself.

It is concluded on the basis of an examination of the information requirements of both the machinery control room watchkeeper and watch-supervisor that CRTs are indeed feasible in MCC design and that they satisfy information requirements which cannot be satisfied by conventional display instrumentation. Further, it is concluded that the use of graphics displays offers the potential to reduce operator workload and to meet the primary requirements of machinery control operators, namely, continuous display of a small amount of highly integrated propulsion machinery status information and rapid response to an abnormal situation.

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THE IMPACT OF THE INTRODUCTION OF TRACKING RADARS AND SEMI-AUTOMATIC NAVIGATION SYSTEMS ON THE MERCHANT MARINE NAVIGATOR

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ABSTRACT

Recent advances in technology provide the navigator with a greatly increased amount of data, often in a form different from that that he is used to. Operational techniques have changed greatly: traditional skills requiring meticulous use of tables and arithmetic have been replaced by equally demanding skills of monitoring and checking computers or computer based equipment. Manufacturers produce equipment of their own design with perhaps not enough consideration given to its use in the total bridge environment. Computer based equipment operates programs the details of which are not known by the navigator so that he is unable to assess the quality of data obtained. These considerations demand a reappraisal of (i) the extent and design of navigation equipment and (ii) the knowledge and training required by the navigator to make the best use of his navigation equipment. This paper examines these considerations, indicates the scope of present investigation and suggests the way forward.

The technological explosion of the last few years has left the Merchant Marine largely stunned and bewildered. Urgent action is necessary to develop the full potential of technological innovations so that they will be accepted and used in an optimum manner. Otherwise they will be rejected either by resentful navigators or by disillusioned shipping companies who are becoming increasingly reluctant to spend large sums of money for what they see as doubtful benefit. Secondly, careful thought needs to be given to just what is the navigator's task and how he is to be trained to fulfil it properly. The sit ation is complicated by the fact that ships operate in a two dimensional environment to which all and sundry have free random access. The wide variations in the standard of fitting of equipment and in the training of the operators, the difficulties of obtaining international agreement on even minimal standards and the even greater difficulties of enforcing these standards mean that progress is unfortunately going to be slow. The lack of international, or even national agreement, has resulted in a situation where we usually do things the wrong way round: instead of producing a system specification and inviting manufacturers to produce it, we wait for manufacturers to make what they think we ought to have and then try to see if we have a use for it.

Until the very recent past the facilities available to the marine navigator were severely limited and quite often, for much of the time, he did not really have any precise knowledge of the position or velocity of his ship. Nevertheless, the methods used were tried and trusted and the necessary skills to use them had been developed over many years. The prime skills required were, firstly, making visual observations together with an unquantified assessment of their reliability acquired by long experience, and secondly the extraction of data from tables and the processing of this data by manual computation, requiring a certain meticulousness and orderliness of thought.

Then came the technological explosion and the confusion began: never before had the navigator had so much data available to him, so much capacity for processing that data and so little idea of how to get the best from it. Problems abounded: equipment was fitted or not at the whim of a shipping company or individual within that company, equipment was all too frequently unreliable and therefore suspect, training programs were non-existent or inadequate and consequently equipment was misused, and to a great extent, these problems still exist.

Fixing systems have proliferated with a variety of sponsors and manufacturers vying for the adoption and widespread use of their favoured equipment. The satellite system provided for the first time ever world-wide coverage with highly accurate but intermittent fixing. Omega provided for the first time continuous world-wide coverage, albeit with less accuracy. Loran C provides good accuracy with limited cover and the Decca Navigator likewise provides good accuracy with more limited cover still. In coastal regions radar is usually adequate and is moreover independent of external transmitters. The choice of systems for an individual ship is usually quite clearly indicated by the trade in which the ship is engaged and the area in which it operates but it is in the use of the equipment that difficulties arise. We have not yet reached the stage where these systems have been in operation for so long that navigators have received instruction in their use as part of their basic training. Some operators will have received very brief training provided by manufacturers; very rarely will navigators have been able to undertake refresher and updating courses. Hence situations such as the following, all too frequently occur. A manufacturer installs an Omega receiver in a ship and the navigators present are instructed in its use by the installers and all is well for a few months. Suddenly, reports arrive of unsatisfactory performance which on being investigated reveal that there is no fault, simply a change of crew and the new crew do not know how to use the equipment properly. Unfortunately it is all too common for students to arrive at my college with ingrained misconceptions which have been implanted by self-acquired on the job 'training'.

A current development is the provision of some computing facility as an integral part of a fixing system so that the readout is not just position line data but latitude and longitude. The provision of the final answer in this way can lead to far too much complacency: there is the feeling that, because a computer has done the job and the answer appears on a sophisticated display, it can be relied upon implicitly. When there is no need to manually acquire the data and consider its reliability the navigator's inbuilt caution disappears and he too readily accepts the final answer displayed. Furthermore there are frequently additional facilities such as real time computations of the estimated position between fixes, courses and distances made between way points, etc. These and the fixing computation itself require that data is fed to the computer, usually in a specific format. Simple though this is, it is a process quite different from that of traditional navigation and demands a quite different meticulous attention from what the navigator is used to.

Yet further development is occuring with the provision of automatic track following systems. In these the fixing data is processed and compared with a previously specified track and the auto pilot is automatically controlled to maintain the ship on that track. Again, there is a need on the part of the navigator to avoid complacency and to maintain adequate checks that the system is working properly. Further difficulties can arise because the navigator is not aware of what it is that the equipment is doing. A recently introduced Omega receiver acquires the data for four position lines and computes from these the Most Probable Position - a statistical estimate of where the ship is most likely to be. I would assert that few marine navigators at present understand the significance of this. The alleged advantages of such a sophisticated receiver are that plotting on small scale charts is avoided as is also the use and interpolation of propagation correction tables. This is no doubt true but with all the computing power available to determine position from lane counts, to compute propagation corrections and perform other navigational chores, it is perhaps surprising that the choice of stations to be used is still left to the navigator: he has to consider signal strengths, lane widths and angles of cut in making his choice. It would seem that the equipment could do this for him with no trouble at all, and yet it does not. Moreover after making a statistical estimate of the most probable position there is no information given as to the likely quality of the answer. This one example serves to illustrate that what is wanted is a careful review of the navigator's task and of the equipment that is necessary to accomplish it.

Radar provides yet another example of the way in which technical innovations are introduced in what is perhaps a not entirely satisfactory way. Some of these are good and well thought out but others leave much to be desired. Moreover the multiplicity of manufacturers' equipments with their numerous differences can be a source of confusion. The first improvement to be made to the original relative motion radar display was the provision of azimuth stabilisation. This was a very good improvement but has restricted the display to the North up mode in all but a very small number of ordinary (non-computer) radars. The next step was the introduction of so called true motion, a facility that can be extremely useful in certain circumstances but which, in my view, has never been sufficiently developed in non-computer radars. If the facility is required at all it must be capable of giving either a ground stabilised display or a sea stabilised display continuously and automatically and, as far as I am aware, there is no ordinary radar capable of doing this. To achieve proper stabilisation the ship's velocity must be fed to the radar correctly: heading data is obtained from the compass and speed from a two axis log and I am not aware of an ordinary radar that will accept a two axis log input.

The radar equipment that has aroused the most controversy is the so called collision avoidance radar, or tracking radar, as I prefer to call it. There are some half dozen manufacturers producing their various forms of tracking radar and, while they differ greatly in detail, in general they all do the same thing: namely to track targets and to display their velocity graphically as vectors on a synthetic radar display and also digitally. Variation of the length of the vectors enables the navigator to estimate the future development of a traffic situation. There is however one important exception and this is the system which displays predicted danger areas and is fundamentally different from all the others in that the 'future' scene on the radar does not indicate the situation for all tracked targets at a common instant of time. Data for these tracking radars is taken from an ordinary radar, processed by a computer and then displayed on a synthetic radar screen. This makes the presentation extremely versatile: it may be azimuth stabilised north up or ship's head up; it may be a relative display, a sea stabilised display or a ground stabilised display. What ever the mode of display, target vectors may be relative to own ship, the sea or the ground. Changeover from one to another of these possibilities is effected at the touch of a key. There is sometimes a manoeuvre simulation facility which permits the effect of a proposed manoeuvre to be seen on all traffic in the vicinity. Where the tracking radar is combined with a track following system a simulated manoeuvre, having been assessed and found satisfactory, can be executed at the touch of a key.

A number of difficulties arise with these systems. They do not give any indication of the quality of answers being obtained and once again it is very easy to forget that the input data is from an ordinary radar and to become deceived by the apparent precision of the output. Naturally enough manufacturers are reluctant to disclose the details of their computer programs and it is therefore difficult to know how the tracking is accomplished and the data filtered. While these tracking radars are extremely versatile and are able to process a vast amount of data and present it graphically to the navigator, these very attributes make much greater demands on the navigator. He must really understand the principles of radar displays and radar plotting in order to appreciate the information that is presented to him. He must also be completely familiar with operation of the equipment and must exercise a rigorous discipline to ensure that he is not being misled and to ensure that the equipment is working properly. It is important to remember that none of this sophisticated equipment is making decisions for the navigator, it is simply processing data at a very rapid rate, permitting the navigator to avoid much tedious work with his head

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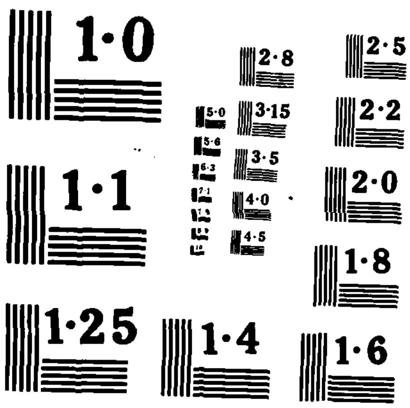
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down and enabling him to spend a much larger proportion of his time with his head up watching what is going on around him.

Recent research has indicated that a fully trained and competent navigator has his performance improved by the use of this equipment but that the less able navigator, with a less than adequate understanding of what he is doing, performs less satisfactorily than with an ordinary radar. If these findings are generally true, it would appear that the suggested compulsory fitting of tracking radars would produce the opposite effect to that intended.

CONCLUSIONS

It is suggested that a period of consolidation is required to permit navigators and their training to catch up with development. While manufacturers have made many praiseworthy attempts to improve navigation equipment there needs to be an international reappraisal of the navigator's task and the equipment that he needs to perform it. There needs to be international agreement on the design and specification of equipment to be fitted and on the standard of training to be undertaken by the navigator before he is permitted to use this equipment. These standards should be realistically set at the level demanded by safe operating requirements and not at the lowest common multiple for this would not achieve very much.

MANEUVERING CONTROL OF AMPHIBIOUS HOVERCRAFT

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ABSTRACT

The response of an amphibious air cushion supported vehicle or hovercraft during turning maneuvers with and without an automatic control system is examined. Over water response is predicted by a digital simulation program in which the plant model is based primarily on experimental model data. A digital simulation model is also used to study over land response. However, in this case, an analytical model is used for the plant. The results indicate significant overshoot and settling times in response to turn rate demand. By performing a gain sensitivity study it is demonstrated that improvements in craft response can be realized by using a speed dependent gain scheduling strategy.

INTRODUCTION

The maneuvering character of fully skirted air cushion supported vehicles (ACV's) is dominated by very low yaw and sway stability. Low transverse and rotational stability is the necessary price which must be paid to obtain the low resistance-to-displacement ratios which these vehicles exhibit. The typical ACV translates and spins almost as easily as it moves forward under most conditions and over most surfaces. This behavior can be used to advantage as a method of reversing thrust for deceleration through a 180 degree change in drift angle and with little change in direction of heading. Alternatively, the large drift angles achievable of up to 90 degrees allow the craft thrust to be vectored normal to the line of flight as a superb turning effector. On the negative side, the obvious disadvantage of the inherently low yaw and sway stability is the need for extremely sensitive course control systems and force effectors. The ACV must be designed through its maneuvering control system to compensate for its natural stability deficiencies.

The effects of various control system characteristics upon the maneuvering behavior of ACV's are explored in this analysis. The automatic control system of the Amphibious Assault Landing Craft ACV JEFF(A) is used as an example system to demonstrate the beneficial results of control system gains optimization. The nature of the supporting surface (either water or solid terrain) and the stability of the ACV is shown to have a significant effect upon desirable control system characteristics.

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Navy Programs

The U.S. Navy is actively involved in the development of a mission-oriented amphibious air cushion supported vehicle to be employed in amphibious assault and landing operations. This vehicle, designated LCAC (Landing Craft, Air Cushion); is being developed by the Naval Ship Engineering Center under the overall direction of the Naval Sea Systems Command. The proposed hovercraft uses the technological foundation developed under the Amphibious Assault Landing Craft (AALC) Program (see References 1, 2, and 3), at the David W. Taylor Naval Ship R&D Center.

The AALC ACV's, designated as the JEFF(A) and JEFF(B) are designed to operate over water, over land and through surf zones. The ability of these vehicles to operate out of the well-deck of an amphibious assault ship and over the beach at high speed has made these designs potentially excellent amphibious vehicles. The two configurations, JEFF(A) and JEFF(B) are shown in Figures 1 and 2; and the principal characteristics of the vehicles are listed in Table 1. These vehicles are presently undergoing Navy engineering trials by the AALC Trials Unit located on the facilities of the Naval Coastal System Center, Panama City, Florida.

JEFF(A) and (B) Lift and Propulsion Systems

The two AALC vehicles were designed to meet the same functional requirements for payload, speed, range, endurance and operating environment. The two hovercraft were, however, intentionally designed to incorporate different propulsion, lift, skirt/cushion system and control concepts in order to provide the Navy with an extensive engineering design and experience base. The major technological difference in the lift systems of the JEFF craft is in the skirt design. Both craft have a peripheral bag that distributes the air along the craft periphery. In the case of JEFF(B), the air is supplied by the peripheral bags to a series of "fingers". The "fingers" are "U" shaped triangular closures open on the inside and kept inflated by the cushion pressure. This bag-finger skirt system was evolved by the British Hovercraft Corporation and is in current use on all BHC craft. The JEFF(A) "Pericell" cushion system, patented by Aerojet General Corporation (AGC), was evolved by AGC during the AALC Program. The "Pericell" skirt system is composed of a series of conical cells extending downward from the peripheral bag. The JEFF(B) includes longitudinal and transverse stability bags to divide the cushion area into four compartments. The JEFF(A) pericell concept, based on model experiments, does not require internal subdivision of the cushion area.

The propulsion and maneuvering control force effector systems of the two JEFF craft also differ. For the JEFF(A), four shrouded propellers provide thrust and maneuvering control through vectoring of the thrusters from the normal propulsion position. Propeller RPM and propeller blade pitch angle can be controlled to provide forward and reverse variable thrust. The aft propellers in normal position are directed parallel to the craft centerline. The forward propulsor wakes, in normal position, are directed outboard 18 degrees. The forward propulsor travel has been restricted so that the wakes will not interfere with the aft propulsors. For the JEFF(B), an integrated propulsion system and common shaft requires that the propeller RPM be set by the required fan RPM. The thrust is then controlled by changes on propeller pitch angle to provide the necessary variation in forward and reverse thrust. Twin rudders are mounted behind each propeller. Part of the required forward thrust on the JEFF(B) is also provided by the two bow thrusters. The bow thrusters can be rotated 360 degrees and in the normal position are directed outboard 18 degrees, so that there is no interference with the stern propulsors.

Control Design of the JEFF(A) and (B)

The control system for maneuvering and directional stability of the JEFF(A) is a combination of automatic control of the aft propulsors with manual control of the forward propulsors. The forward propulsors are deflected (up to 36 degrees outboard) by moving the control handles to the side. Yaw rate demand for the automatic system is dependent on the amount of rotation of the control handles. Thus a simple movement of the control handles to the side which deflects the bow propulsors will result in a pure sway motion. Since the demanded yaw rate is zero, the stern propulsors will deflect to counteract any turning moment caused by this deflection of the bow propulsors. Rotating of the handles, combined with their movement to the side, will result in turns with large drift angles which is a good turning strategy for the craft. The control system in the JEFF(A) is sensitive to yaw rate demand, the heading angle demand (which is the time integral of yaw rate), and the integral of the heading angle demand. The gains for the system are fixed and were based on early test data and analysis for the AALC program. This paper examines the effect of these gains on craft response for various speeds and other conditions. Additional turning moment can be generated by changing the RPM and blade pitch of the individual propulsor. At high speed, RPM and blade pitch angle vary with throttle position. At low speeds, RPM is fixed and blade angle can be independently changed. It is even possible to reverse the thrust on a single propulsor. Such an action may be required if the demanded turn rate cannot be reached with a maximum deflection of the stern propulsors of 30 degrees. Basically the JEFF(A) can be considered highly stable when the control system is operating. For straight ahead operations, the control system can eliminate course changes due to random wind gusts or steady winds without affecting the pilot workload. This can be an important factor in performing operational missions. In turning, the best strategy may be to develop a sideslip condition with the bow thrusters then turn the craft by demanding a yaw rate. Speed loss and the possibility of plow-in must be monitored carefully during such a maneuver but simulation results indicate that the chance of success is good.

The JEFF(B) is controlled manually. The bow thrusters are moved by turning the wheel while the rudders behind the propulsors are moved by foot pedals. The propeller pitch can be modified on each propulsor to augment the turning. In addition, the control can be adjusted so that a movement of either rudder pedals or bow thruster wheel can actually move both devices. Nevertheless, this system required more pilot workload than the automatic JEFF(A) system. The JEFF(B) controllability will not be treated any further in this paper.

THE ACV OVERLAND SIMULATION

The maneuvering performance of hovercraft is determined by the competing and interacting effects of the supporting surface, craft drag, aerodynamic propulsion and the design and operation of control effectors. The prediction of ACV maneuvering characteristics via numerical simulation must address each of these considerations and their complex interrelations. There are three major methods through which maneuvering predictions may be effected: first, free running model experiments; second, data based modeling; and third, fully analytic modeling.

The use of a free-running model for maneuvering predictions is the most obvious and straightforward method but unfortunately this often proves to be the most expensive. The ACV model must be large enough to carry the bulky propulsion and control equipment and comparably large enough to minimize scaling effects according to a Reynolds criterion. Further, the model must be an accurate representation of a prototype design in dynamic as well as geometric characteristics. This requirement is most important in the representation of seal contact forces and the under-skirt jet characteristics. Detailed cushion dynamic modeling is seldom required for maneuvering applications but stage-discharge characteristics of the lift system and appropriate cushion compartmentization must be modeled exactly. The temporal responses of the control effectors must also be approximately modeled to aid the operator of the model with the intrinsic problem of flying the craft on an accelerated time scale compared with the full-scale ACV.

Data-based modeling of ACV maneuvering requires the measurement of craft stability derivatives in sway, yaw and surge through captive model experiments. This type of simulation has been developed for the over water maneuvering of two types of air cushion vehicles by Wachnik, Messalle and Fein. (See Reference 4.) This simulation has been used in this paper to evaluate the JEFF(A) control characteristics over water and is discussed in a later section. Unfortunately, the cost of experimental development of the corresponding stability derivatives for overland application has been prohibitive. A similar overland modeling would require horizontal plane oscillation or rotating arm experiments which would begin, of necessity, with the development of the required experimental facilities.

In the third method of analysis, which is used to evaluate the overland control characteristics in this paper, the overland maneuvering characteristics of an ACV is examined through a nonlinear time-domain maneuvering simulation in six degrees of freedom. The program is based upon analytic modeling of the propulsion and drag forces and moments imposed on the vehicle as a function of speed, heading, drift angle, propulsion and control conditions and environmental constraints. The propulsion forces and moments include components generated by propellors, ducts, rudders, cushion discharge jets, fan inlets and gravity (due to ground slope). Drag components include all hydraulic and aerodynamic drags as well as skirt drag. Each of these is modeled as a nonlinear function of craft trim, speed and heading with respect to wind and ground slope.

Skirt or cushion stability seal contact drag is modeled as a function of the computed contact areas for the known craft trim and skirt deformation of each computational time step. These computations yield the longitudinal and transverse drag forces (X,Y), and the yaw drag moment (N) as functions of all operational parameters. Similarly, the vertical force (Z), and the roll and pitch moments (K,M), are determined for the updated footprint area of the cushion and the modeled cushion and seal pressures. These pressures are functions of the underseal gap areas for each cushion compartment which are functions of craft trim, motion history and the specified ground surface. The jet forces resulting from the cushion discharge momentum are computed from the pressure and gap areas and are incorporated into the propulsive characteristics of the vehicle. Specifically, a component of yaw drag is determined from the rotational behavior of the skirt gap when the craft is rotating in yaw. Gross nonlinearities arise in all of these forces (propulsion and drag) from the discontinuous effects of nonhomogeneous skirt contact.

The force and moment derivatives obtained from the analytic models are cast in a coefficient matrix for the six degrees of freedom and solved simultaneously with the specified forcing functions. The analytic model is only quasi-dynamic in the vertical plane since the cushion flow equations are not solved, but rather, pressures are assumed to be functions of appropriate discharge areas with known discharge characteristics. The model is therefore based upon the assumption that the cushion pumping terms (rate of change of cushion volume due to pitch, heave and roll) are much smaller than the fan discharge. This seems intuitively true for maneuvering conditions which are only lightly disturbed in the vertical plane; and further, the assumption has proven to be valid in practice. The only disadvantage to this modeling is the need to introduce heave damping into the model since heave damping normally arises from variations in the rate of change of cushion discharge (dQ/dt). The model developed for the JEFF(A) peripheral cell system is similar to the JEFF(B). However, the calculation of force and moment derivatives is carried out for each cell.

One of the most interesting aspects of the analytic modeling is the treatment of the slope of the ground plane, a problem not realized in overwater maneuvering models. The horizontal plane dynamic response on a slope is simulated by specifying the three Eulerian angles that rotate the gravity vector away from the z-axis. This transformation yields the three components of the gravity force in the inertial coordinate system and is equivalent to specifying the roll and pitch of the ground plane. The application of the transformation matrix allows the computation of the gravity forces in the body axis system (including craft heading) from those computed in the inertial coordinate system.

THE ACV OVERWATER SIMULATION

The control simulation for overwater operation is described in detail in Reference 4. Since it was utilized to evaluate the response of the JEFF(A) to changes in gains, a brief overview of the simulation approach and content is in order. The forces and moments on the air cushion vehicle under investigation were divided into the following categories: propulsion and control forces, momentum forces, aerodynamic forces, and cushion and hydrodynamic forces. The largest force contributions came from the cushion and hydrodynamic components. These forces were obtained by the experimental test series.

The ACV Experiments

The ACV experiments are described in detail in Reference 4. Traditional techniques were employed to obtain the information needed to simulate the maneuvering of these high speed aerodynamically propelled craft. Horizontal PMM experiments and straight-line experiments with variation of speed, roll, drift, and trim were conducted. Also, an extensive rotating arm experimental program was conducted.

The experiments led to an extensive data base over a wide range of operating conditions. The forces showed a high degree of Froude number dependence. Also, variation of force with drift angle over a range of zero to 45 degrees was so nonlinear that even a fifth order polynomial did not provide reasonable representation. The high degree of nonlinearity made it very difficult to represent the data by generally accepted Taylor series coefficients. Consequently, rather than develop a whole series of high order coefficients, it was decided that a data table would be established which would have four dimensions and would contain the forces and moments as functions of speed, yaw rate, sway velocity and roll angle. The data base was sufficiently dense such that accurate representations of the forces and moments could be obtained by a four-dimensional linear interpolation between the points. This data base allowed for the development of an efficient, flexible simulation for each of the AALC designs.

The Design of the Simulation

The cushion and hydrodynamic forces were obtained from the data base. The aerodynamic forces were also arranged in an interpolation table using wind tunnel experimental results. The propulsion, rudder, and duct forces were derived from contractor data and are the same as those in the overland digital program. The momentum drag forces are computed based on the fan air flows, forward speed, and the assumption that the flow is distributed uniformly throughout the cushion system.

The six-degree-of-freedom calm water simulation is a standard time domain simulation in which the coupled differential equations of motion for the six degrees of freedom are solved simultaneously and integrated numerically to obtain velocities and displacements. The differential equations are solved in terms of variables defined in the body axis system. The computer program provides time histories of the craft motion, trajectories, force and moment components for each subsystem, and the status of various controllable items.

Numerical integration of the system of differential equations is accomplished by a fourth-order Runge-Kutta-Merson method. This method provides for an adjustable time step size depending on the relative error criteria set for each differential equation in the system. In most cases, during transient motion the resulting adjusted time step size required by the integration subroutine was 0.125 seconds. The six-degree-of-freedom version of the program runs in roughly 60 percent of real time on the CDC 6700 computer at the David W. Taylor Naval Ship Research and Development Center.

Simulation of craft control is possible in three ways. The first utilizes a table of motions for the various force effectors, such as wind, propulsor angle, or engine RPM. Linear interpolation between the data points in the table is used to compute required values for other times. This time history capability will be used to correlate the data from full-scale trials of the JEFF craft and the simulation. A second way that craft maneuvers can be controlled in the simulation is through a subroutine which provides a simplified model of a human operator. In this case various motion parameters can be observed to see if certain bounds are exceeded. If the parameters exceed these bounds, then prescribed control activities are adjusted to a given rate. The commands are not programmed into the control subroutine but are entered as data to the digital simulation program. This feature makes the model easy to use in attempting to optimize maneuvering strategies. The third way the craft can be controlled in the simulation is with an automatic feedback control system. This last approach was used in the analysis presented in the following sections.

Overwater Maneuvering Results

There have been extensive evaluations of maneuvering of the JEFF(A) using the manual control mode of the simulation program. These studies have been intended to identify problem areas prior to the full scale trials. The safety of planned maneuvers has been investigated as well as the effect of failures of various propulsors or control signals. Simulations to date have not revealed any major problem areas but do indicate that the craft control system is helpful in responding to subsystem failures. For example, when there is a failure of a single aft propulsor at 40 kts during straight-ahead flight the craft returns to straight ahead flight in 20 seconds when the autopilot is operating. Without pilot response a steady turn of 1.5 degrees per second results.

The effect of rate of deflection of the propulsors has also been investigated and it was found that high deflection rates can introduce transient responses that lead to possible spin out conditions. The recommended deflection rate is less than the hydraulic system capabilities of the JEFF(A), which is of interest for future designs.

The steady turning ability of the JEFF(A) is demonstrated by simulation in Reference 4. The craft can meet the AALC turn criteria for a 50 kt initial speed turn. The criteria are a 1980 meter transfer and 1210 meter advance with speed remaining above 40 knots. Table 2 shows the steady state yaw rates that can be developed using manual control of the bow and stern propulsors. The positive yaw rate indicates a turn to starboard. Positive deflection of the propulsors is leading edge to starboard. The forward propulsor deflection is on top of the manual deflection of 18 degrees leading edge inboard. The results show the turn rate does not increase linearly with control deflection as factors such as speed loss, roll angle, and drift angle affect the yaw rate.

The overwater simulation has been used in design studies for LCAC craft that include some combination of aspects of the JEFF(A) and JEFF(B). The program is flexible in regard to CG location, magnitude of control force, position of control device and even stall characteristics of the control device. This gives the LCAC designers the option of evaluating trade-offs between controllability and other design parameters. The large extent of the hydrodynamic data base assures the accuracy of the design evaluations done with the simulation.

OVERWATER CONTROL SYSTEM RESULTS

The results of experimental test programs indicated that the forces and moments acting on the ACV are highly nonlinear. Consequently, the use of an analysis in which linear models are used for these forces and moments is not appropriate. Hence, the analysis presented here is carried out by means of simulation studies using force and moment models based on the nonlinear force and moment tables acquired during experimental test programs.

The automatic control system on JEFF(A) is illustrated in the schematic presented in Figure 3. Three error terms are used: rate error, heading error, and the integral of the heading angle error. The corresponding gains are G1, G2, and G3 which have nominal values of -3.1, -1.9, and -0.1, respectively. The pylons, rotatable air propellers, are represented by a second order system. An automatic speed controller is not used. The plant is represented by the non-linear simulation model.

The mode of operation considered in this study involves application of a given turn rate demand curve. The curve consists of straight line segments of varying duration. Since speed is not controlled, there is typically a speed loss during a turn due to the increased drag arising from the change in craft orientation and motions during the turns. The version of the simulation program used here is adequate only at speeds above approximately 22 kts due to the available experimental data. Consequently, the time history plots presented in this paper are terminated if the speed drops below 22 kts.

The effectiveness of the automatic control system is illustrated in Figure 4, which contains a time history of craft turn rate and the demanded turn rate at speeds of 40 and 50 kts. The craft is stable for these two turns, which have a maximum demanded turn rate of 1.5°/sec. This particular type of control system will invariably exhibit overshoot behavior for a ramp demand because of the integral error terms. As seen from the figures, the overshoot and settling times are appreciable. There are several different ways to minimize the overshoot and decrease the settling time. One approach is to filter the demanded turn rate so that the sharp breaks in the demand curve are smoothed. Another approach is to determine a set of optimum gains via gain sensitivity studies. This last approach is considered in this paper.

The gain sensitivity study was conducted via the digital simulation program. Each of the gains was separately increased and decreased by a factor of 2. The results for 40 kts are illustrated in Figures 5, 6, and 7. The dashed line is the time history corresponding to a turn with nominal values for the gains. The demanded turn rate is also illustrated. The best performance (minimum overshoot and minimum settling time) is obtained for nominal values for G2 and G3 and a value of G1 equal to twice the nominal value. There may be problems with increasing the rate gain (G1) since the increased sensitivity may amplify rate sensor noise effects. Sensor noise is not modeled at this time.

At 50 kts the craft response is much more sensitive to changes in gains, as illustrated in Figures 8, 9, and 10. The greatest reduction in overshoot and settling time is achieved by increasing the magnitude of both G1 and G2 as illustrated in Figures 8 and 9, respectively. Note that at 40 kts the performance is not improved when G2 is increased. Consequently, these results imply that there may be some benefit in developing a gain scheduling strategy.

OVERLAND CONTROL RESULTS

The exercising of the analytic based simulation model has shown that the pilot must use care in maneuvering the craft at low speed overland. It has further shown that the pilot may expect to exercise the thrusters a great deal (specifically much more than the rudders) during low speed maneuvering if the cushion discharge is maintained at normal design value. If, however, the cushion discharge is reduced so that skirt contact and the resultant skirt drag are increased, the analytic model shows that the yaw stability of the craft is greatly increased and the craft becomes much easier to handle. Various pilot control strategies have been examined through the maneuvering model.

An important maneuver in the overland operation of ACV's is the 180° reorientation of the craft to depart from the shore. The primary requirement for this maneuver is to minimize the lateral and longitudinal movement of the craft and to spin the craft on its axis. One difficulty in this maneuver is the occurrence of yaw rate oscillations at the end of the maneuvering due to the lack of sufficient yaw rate damping. The operator wants a responsive craft and one that does not require more control actions to stop these yaw rate oscillations at the end of the maneuver.

In order to minimize horizontal plane motion during low speed maneuvers, the bow propulsors' thrust is reversed and opposes the stern propulsors. The thrust from the stern propulsors is reduced to cancel the bow thrust force. Several overland turns with different demand turn rates were simulated to determine the effectiveness of the control system at the reduced power levels and at very low speeds. Figure 11 shows the resulting craft yaw rate for demanded yaw rates of 1.5, 3.0, and 4.5 degrees per second. Note that the overshoot is almost twice the demanded yaw rate and almost no settling occurs for the nominal gains.

From these figures, it appears that the operator would have to initiate control actions midway through the maneuver to cancel out the craft yaw rate oscillations at the end of the maneuver. This is an extra burden on the operator. It also shows that a control system which follows the demanded turn rate very closely is required for this special overland maneuver.

Based on these studies it seems that a demanded turn rate of 3 degrees per second would be a reasonable goal since this would allow reorientation of the craft in approximately one minute without excessive operator actions.

Figure 12 shows the craft yaw rate for various gain adjustments. This shows that the currently installed (nominal) system is not optimal. In particular, doubling G1 shows an improvement in settling time and a reduction in the overshoot. The craft moved less than 40 feet laterally during this maneuver. When the value of G1 was increased by a factor of 4, the overshoot was reduced and the system settled to near the demand value in about 30 seconds. This would give the operator time to bring the yaw rate back to zero and stop the craft at the end of the maneuver.

From the simulation results, it appears that the control system gains should be altered as the speed and thrust are reduced. Increasing G1 by a factor of four may cause problems due to sensor noise. However, these problems, if any, may be tolerable for short term maneuvers. Further investigation of the effects of sensor noise awaits future study.

CONCLUSIONS

The following conclusions can be drawn from this analysis of maneuvering control of the ACV.

a.) Tools have been developed to evaluate control strategies for overland and overwater operation of ACVs. These tools can be applied to either the JEFF(A), JEFF(B), or LCAC designs. Control problems can be modeled and solutions evaluated.

b.) The JEFF(A) control approach is a means of providing efficient control of an ACV overwater in spite of the inherent yaw instabilities that may be present. The control approach relieves the operator of much of his workload in combating random wind gusts or in turning in restricted waters.

c.) The original control system gains for JEFF(A) were designed using linear theory. Based on the results presented in this paper, it is quite apparent that a gain scheduling scheme, where the gains are primarily a function of speed, would be desirable. Also, even though the restoring moments are small overland, the automatic control system does function. Again, a gain scheduling scheme may be desirable overland.

d.) In both overwater and overland operations, applying a suitable filter to the demanded turn rate may greatly minimize the turn rate overshoots. It is recommended that investigations into filtering strategies, particularly for overland operations, be conducted in future analysis efforts.

f.) The effects of sensor noise should be examined particularly for maneuvers requiring short time periods and high gains.

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TABLE 1

JEFF CRAFT PRINCIPAL CHARACTERISTICS

	JEFF (A)	JEFF (B)
LENGTH, HARD STRUCTURE	83 FT., 0 IN	83 FT., 0 IN
LENGTH, OVERALL (ON CUSHION)	96 FT., 0 IN	87 FT., 7 IN
BEAM, HARD STRUCTURE	43 FT., 0 IN	43 FT., 0 IN
BEAM, OVERALL (ON CUSHION)	48 FT., 0 IN	47 FT., 0 IN
HEIGHT, ON CUSHION	23 FT., 0 IN	23 FT., 6 IN
HEIGHT, OFF CUSHION	19 FT., 0 IN	19 FT., 0 IN
WEIGHT, DESIGN GROSS	340,000 LB	325,000 LB
LIGHT CRAFT (CREW, STORES)	190,000 LB	166,200 LB
FUEL	40,000 LB.	38,800 LB.
DESIGN PAYLOAD	120,000 LB.	120,000 LB
DESIGN OVERLOAD PAYLOAD	150,000 LB	150,000 LB.
WIDTH, FORWARD RAMP	20 FT., 0 IN	25 FT., 4 IN
WIDTH, AFT RAMP	27 FT., 4 IN.	14 FT., 6 IN.
AREA, CARGO DECK	2,100 SQ. FT	1,740 SQ. FT
DRAFT, OFF CUSHION (DESIGN WT.)	2 FT. 10 IN	3 FT., 4 IN
ENGINES	8 AVCO LYCOMING TF 40	6 AVCO LYCOMING TF 40
INSTALLED POWER, TOTAL	16,800 SHP	16,300 SHP
PROPULSORS	4 SHROUDED REVERSIBLE PITCH SHROUDED PROPELLERS OF 89.5 IN. DIAMETER	2 SHROUDED REVERSIBLE PITCH PROPELLERS OF 141 IN. DIAMETER. 2 BOW THRUSTERS (FROM LIFT FANS)
LIFT FANS	8 SINGLE CENTRIFUGAL FANS OF 48 IN. DIAMETER, 1,600 CFS PER UNIT AT 170 PSF	4 DOUBLE CENTRIFUGAL FANS OF 50 IN. DIAMETER, 4,750 CFS PER UNIT AT 170 PSF
CONTROL SYSTEM	4 ROTATABLE PROPULSORS, ARTIFICIAL FEEL, FLY BY WIRE CONTROL, YAW RATE FEEDBACK AUTO PILOT	2 ROTATABLE BOW THRUSTERS, 2 AERODYNAMIC RUDDERS, ARTIFICIAL FEEL, FLY BY WIRE CONTROLS
SKIRT SYSTEM	LOOPED PERICELL, 5 FT. HIGH	BAG/FINGER WITH STABILITY TRUNKS, 5 FT. HIGH
STRUCTURE	WELDED 5086 ALUMINUM CORRUGATED SHEET. GRP CREW CABIN HOUSING	WELDED 5086 ALUMINUM "HAT" STIFFENED SHEET, BALSA CORE SUPERSTRUCTURE DECKING, RIVETED 6051 8 TRUSS CORE CARGO DECK
DESIGN PERFORMANCE WITH 120,000 PAYLOAD ON 100 F DAY		
SPEED (SEA STATE 2 AND 25 KNOT HEADWIND)	50 KNOTS	50 KNOTS
RANGE (SEA STATE 2 AND 25 KNOT HEADWIND)	200 N. MILES	200 N. MILES
SURF CAPABILITY	8 FT. PLUNGING SURF	8 FT. PLUNGING SURG
MAXIMUM SLOPE	11.5%	13%

TABLE 2

Examples of Overwater Turning Simulations for JEFF(A) with Manual Control

[illegible]	Bow Propulsors Deflection	Stern Propulsors Deflection	Yaw Rate in deg/sec
[illegible]	10	0°	0.5
[illegible]	18°	0°	1.0
[illegible]	0°	10°	-1.0
[illegible]	0°	20°	-1.25
[illegible]	[illegible]	30°	-1.40
[illegible]	-1[illegible]	10°	-1.50
[illegible]	-10	15	-1.60
[illegible]	[illegible]	20	-1.80

rheostat resistance. Controlling fuel flow dictates prime mover (free turbine) power output. Field control establishes the generator internal voltage-speed characteristics providing for adjustable turbine-propeller speed reduction and torque conversion ratio. Controlling the polarity switches determines which machine combination is chosen, and also controls the direction of the motor (and thus propeller) rotation. Control of the rheostat is not independent of the switch control requirements since the resistor primary function is performed only during switching operations.

Design of the control system for this propulsion plant thus reduces to the question, "How should these variables be controlled and what parameters should be monitored to provide control feedback?" It is in answering this question that the simulation proves an invaluable design tool.

SIMULATION MODEL

Analog computer models of the basic electromechanical system, propeller thrust and torque characteristics, and single-degree-of-freedom ship dynamics have been developed and integrated to establish a simulation model. Simplified models of a turbine overspeed governor, a fuel flow scheduler and an automatic crash reverse controller were developed and incorporated in the simulation model.

The mathematical models are developed in three phases: (1) mechanical system and hydrodynamic models, (2) electrical circuit models, and (3) turbine control models. The equations describing the turbine output torque (as a function of turbine speed and fuel flow) are derived from actual performance data. The machine mechanical losses are included in the model but set to zero for preliminary investigations. The propeller dynamics are modeled as thrust and torque coefficients which can be expressed as functions of a modified advance coefficient, and programmed using function generators. The vehicle model is a simplified single-degree-of-freedom model, the effects of wake and thrust deduction factors being neglected. The ship drag is modeled as a function of speed using a function generator as opposed to a polynomial expression.

For machines of this type, the torque is proportional to current and the generated voltage is proportional to speed. The multiplying constants are determined by the field strength and machine internal circuit.

The equations which determine system currents and voltages are derived from the basic electrical circuit in Figure 2.

It is assumed that the capacitance and inductances in the circuit are negligible , and all electrical losses will be included as torque losses. The current in any on-line generator is equal to the differences between the internal generator voltage (E) and the terminal voltage (V_T) divided by the series resistance in that circuit. The motor current is the sum of all the on-line generator currents. The ramp rate of the control resistors is a control variable. The effect of reversing is handled by reversing the polarity of the generator voltage. Also, the opening and closing of the reversing switches occurs when the generator voltage equals the motor voltage, to minimize current flow during switching.

Table 1. Component Design Characteristics*

	Cruise Generator	Boost Generator	Motor
Output:	24400 A @ 150 V	48910 A @ 300 V	40000 hp @ 180 rpm
Input:	5000 hp @ 15200 rpm	20000 hp @ 3300 rpm	100000 A @ 300 V
Efficiency:	98.1%	98.3%	98%
Diameter:	.594 m (1.95 ft)	.966 m (3.17 ft)	1.98 m (6.5 ft)
Length:	.762 m (2.5 ft)	1.27 m (4.17 ft)	3.05 m (10 ft)
Weight:	1193 Kg (2630 lb)	4491 Kg (9900 lb)	38556 Kg (85000 lb)

* The data in this table do not reflect the latest machine designs. Machine design characteristics typical of more recent projections of the technology being developed by the Navy can be found in the references.

SWATH VEHICLE PROPULSION SYSTEM SIMULATION

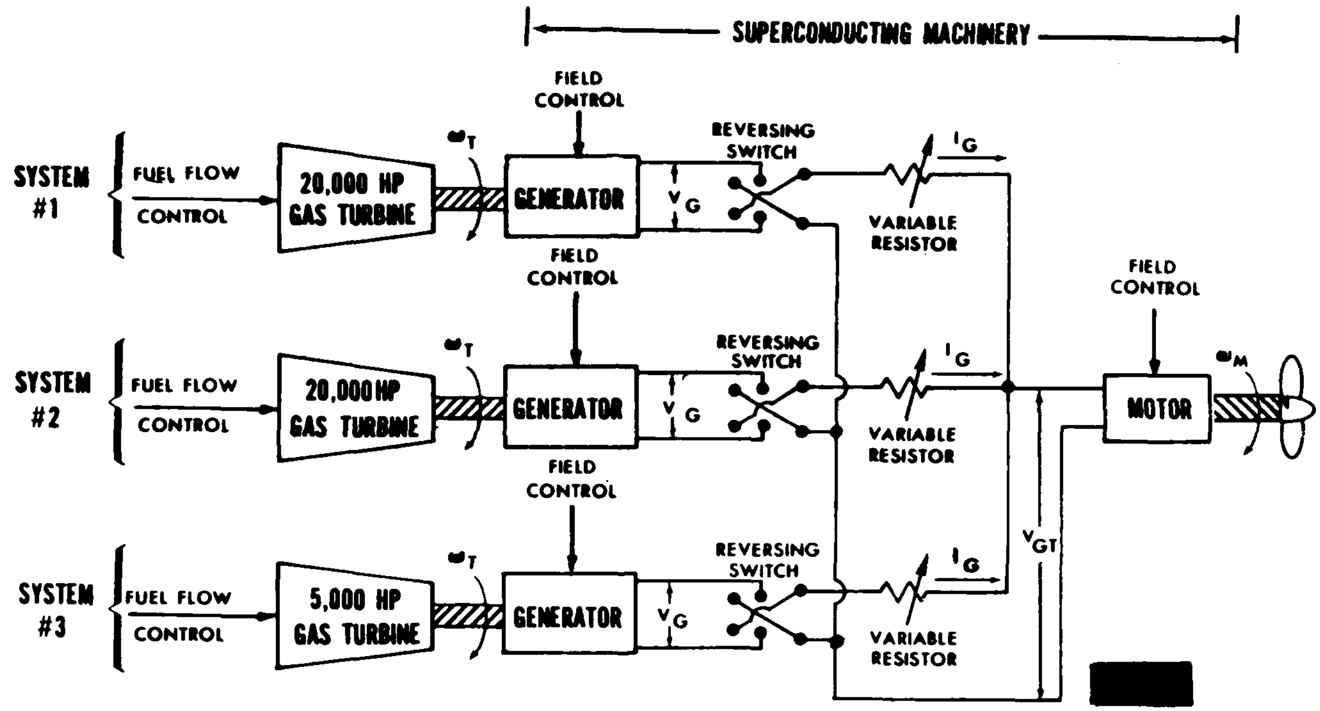


Figure 1. Single SWATH Vehicle Propulsion Plant

examining the ship model behavior in a series of basic maneuvers included in the mission profile adopted for this vehicle. Of particular interest were switching transients when bringing machines on line and during the crash reversal sequence.

The basic components of the propulsion plant and the computer model used in the simulation are described herein. Results of several simulation runs are discussed and conclusions summarized.

SYSTEM DESCRIPTION

The SWATH propulsion system studied consists of identical port and starboard propulsion plants. Gas turbine powered superconducting dc generators are connected by transmission lines and switchgear to superconducting dc motors which are connected directly to the propellers. Only one side of the plant is described and simulated to gain an understanding of the system and its dynamic performance. (In a practical ship installation switchgear allowing cross-connection of port and starboard plants would be included to provide the cruise fuel economy associated with one generator powering both motors.) The propulsion system under consideration is shown schematically as Figure 1. The component characteristics at design conditions are listed in Table 1.

The propulsion plant actually consists of three gas-turbine-driven generator systems which can be connected in any combination to a single motor. One turbo generator is rated at 3.55 MW (5000 hp) @ 16,200 rpm and is intended for use primarily during low speed cruise and station keeping operations. The other two units are each sized at 15 MW (20,000) hp @ 3600 rpm, and provide power for cruise and high speed operation. The 3.55 MW and 15 MW systems have overlapping operating regions to permit smooth transition between power levels.

The switchgear arrangement shown in Figure 2 will permit any combination of generators to be put on line with the motor, however, special operating procedures must be exercised when operating a 15 MW system in parallel with the 3.55 MW system. The reversing switchgear for each turbine/generator system consists of a double-pole, triple-throw (center off) polarity switch and a variable current-limiting resistor (rheostat). (A functionally identical system incorporating one set of reversing switch/rheostat hardware in the motor line could also be considered.) Under normal operating conditions, the rheostat resistance is set at zero, essentially off-line. During a switching operation, the rheostat is ramped up to its maximum resistance at a predetermined rate, and then, after the polarity switch has closed in the desired state, the rheostat is ramped down to zero again. The simulation which has been developed is expected to aid in establishing transient design characteristics for the switchgear. Of particular interest is the rate at which the rheostat resistance should be decreased. A fast rate minimized rheostat size but may result in excessive current and torque surges. A slower rate demands a larger energy absorbtive mass due to the increased power dissipation.

The superconducting motor is rated at 40,000 SHP and drives the propeller at a design speed of 180 rpm. The torque-speed requirements of the motor are determined by the propeller loading, which is in turn a function of propeller speed and ship dynamics.

The four basic controllable variables in the system are the turbine fuel flow, generator field strength, polarity switch position and

SUPERCONDUCTING SWATH PROPULSION SYSTEM SIMULATION

by Henry N. Robey
David W. Taylor Naval Ship Research and Development Center

ABSTRACT

A superconducting electrical propulsion system design for a hypothetical Small Waterplane Area Twin Hull (SWATH) vehicle has been simulated as an aid in evaluating the transient performance of full scale superconducting propulsion machinery. The twin-screw SWATH vehicle, which is propelled by identical port and starboard gas-turbine-driven power plants, was selected as representative of full scale application because it readily illustrates the flexibility of machinery arrangement available with electric drive systems. The basic power plant configuration is described and preliminary simulation results discussed. The simulation covers a range of ship maneuvers which comprise the mission profile adopted for this vehicle. The discussion includes suggestions for system operating procedures and associated control requirements.

INTRODUCTION

The Naval Sea Systems Command (NAVSEA) is sponsoring the development of superconducting machinery for shipboard main propulsion system applications. Program objectives are to establish the engineering capability for superconductive propulsion systems with motor power output up to 56 MW (75,000 hp), and to demonstrate the feasibility of constructing a 30 MW (40,000 hp) system for operational evaluation. Interest in superconductive machinery grew from the need for ship propulsion systems with high power and energy density, greater flexibility of ship installation and operation and reduced noise. Superconductive machinery offers the design, arrangement and operating flexibility of conventional electric drives with reduced size and weight obtainable by using superconductive field windings.

In support of the NAVSEA superconductive propulsion machinery development program, the David W. Taylor Naval Ship Research and Development Center (DTNSRDC) is performing design analyses of overall system performance in various full scale ship applications. These design studies include steady-state operation concerned principally with machine and system efficiencies, and transient operation which addresses maneuverability characteristics and system response to pertabations (switching transients, fault conditions, etc.). This paper deals with the transient performance of a superconductive gas turbine surface ship drive.

A hypothetical twin-screw Small Waterplane Area Twin Hull (SWATH) vehicle with 4000-ton displacement and 36 knot speed was chosen as representative of full-scale applications of superconducting machinery because of the power level (80,000 SHP) and because it readily demonstrates the flexibility of machinery arrangement available with electric drive systems. A model of the propulsion plant was developed and simulated on an analog computer. The transient analysis consisted of

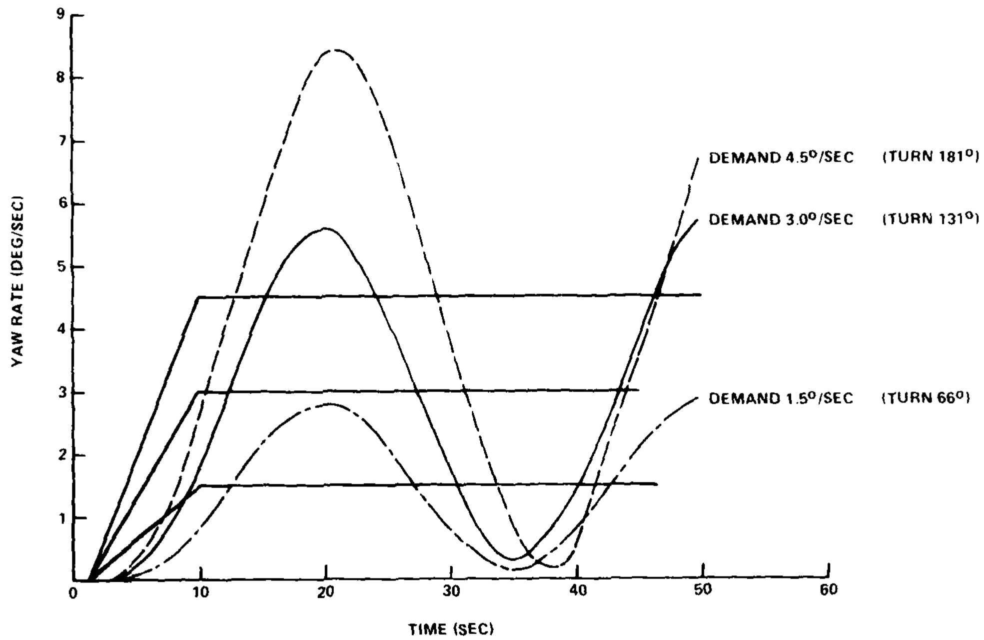


FIGURE 11. CRAFT OVERLAND YAW RATE AT NOMINAL GAIN SETTINGS FOR VARIOUS YAW RATE DEMANDS

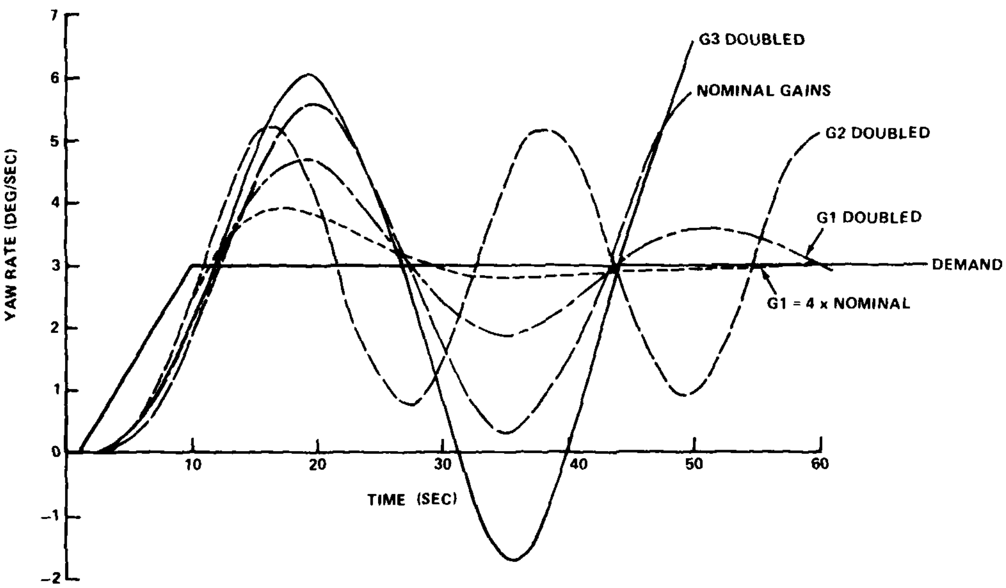


FIGURE 12. CRAFT OVERLAND YAW RATE FOR VARIOUS GAIN SETTINGS FOR A YAW RATE DEMAND OF 3DEG/SEC

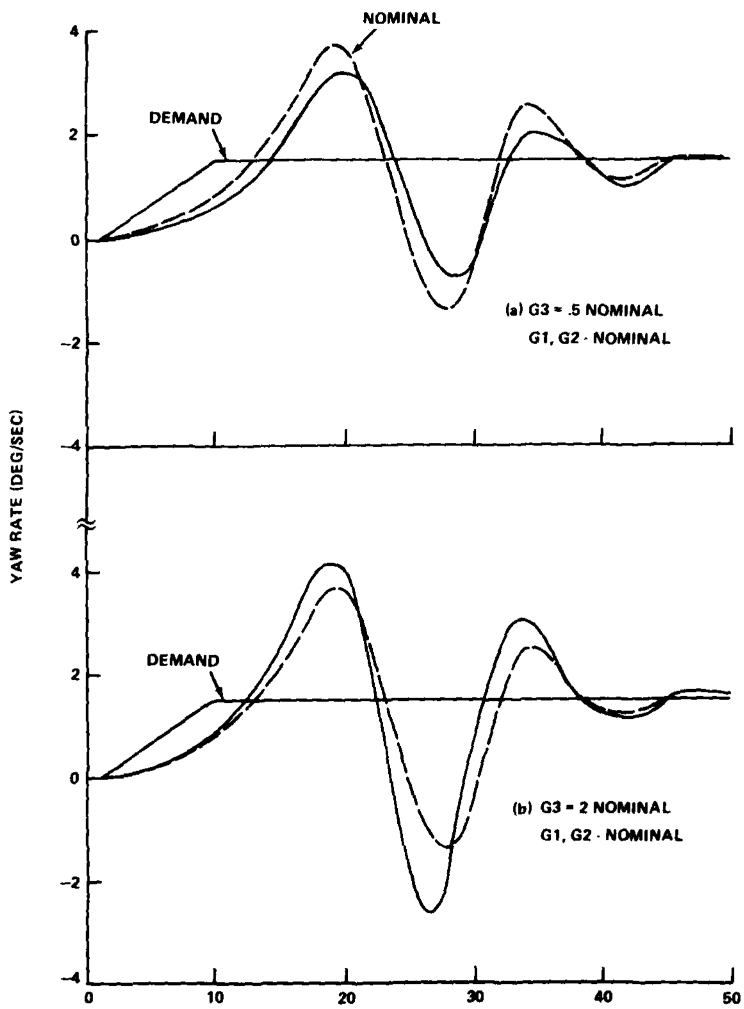


FIGURE 10. SENSITIVITY OF TURN RATE WITH RESPECT TO VARIATIONS IN G3 AT 50 kt

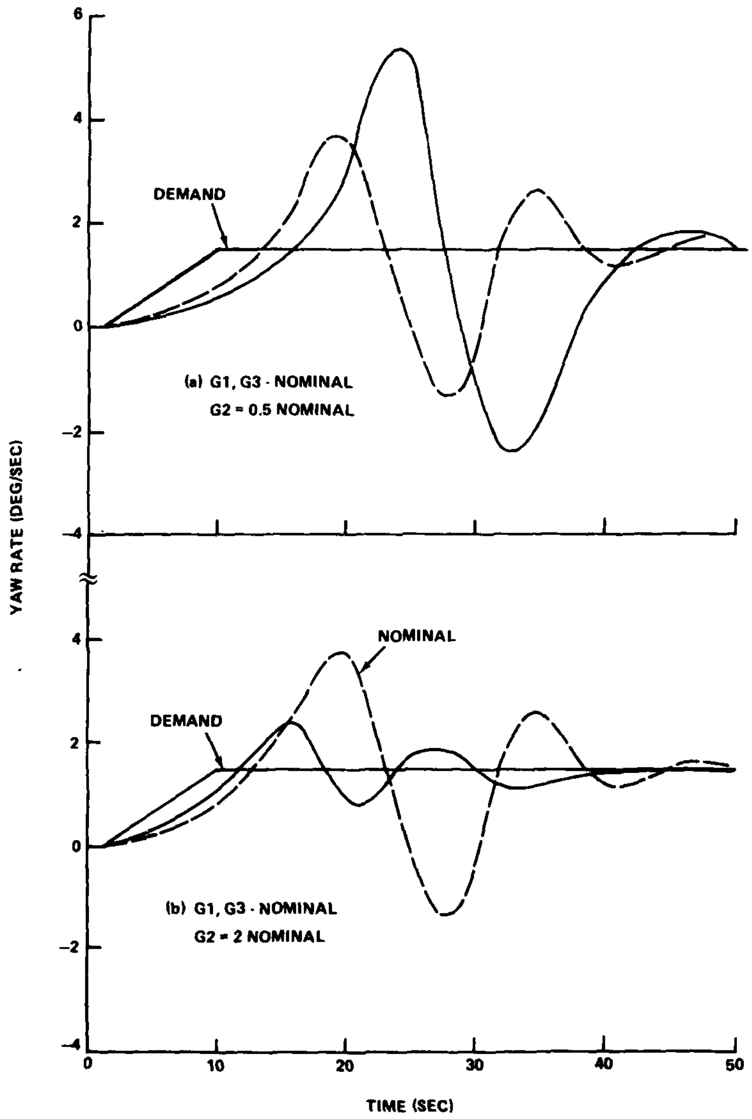


FIGURE 9. SENSITIVITY OF TURN RATE WITH RESPECT TO VARIATIONS IN G2 AT 50 kt

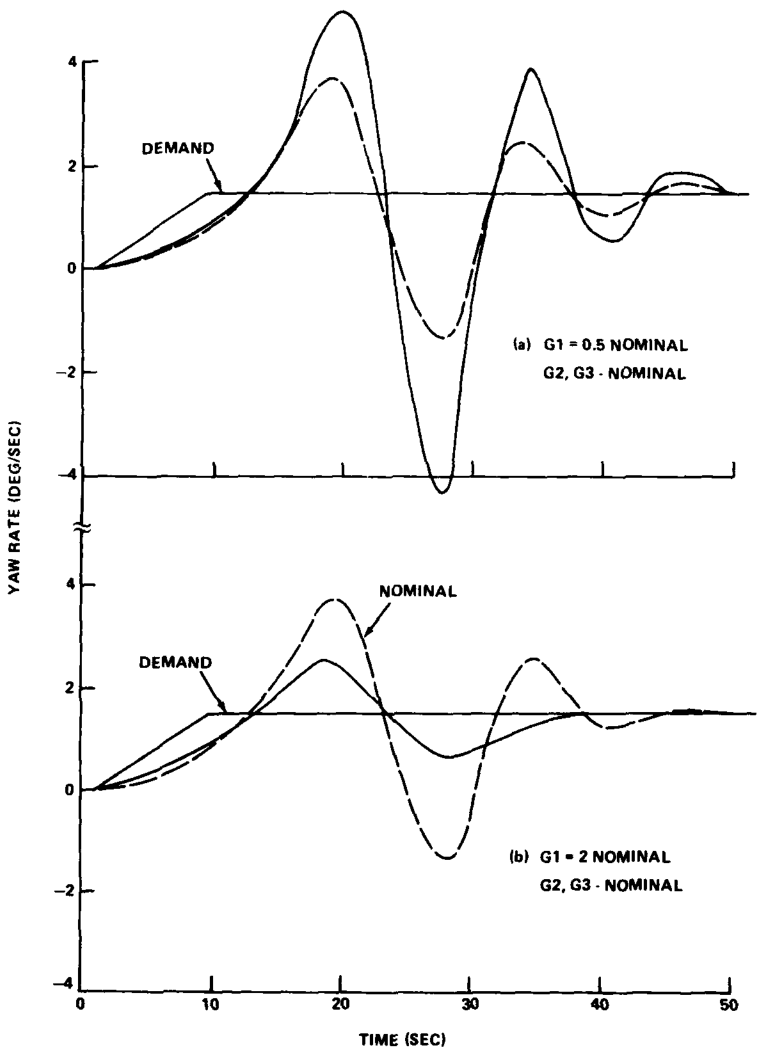


FIGURE 8. SENSITIVITY OF TURN RATE WITH RESPECT TO VARIATIONS IN G1 AT 50 kt

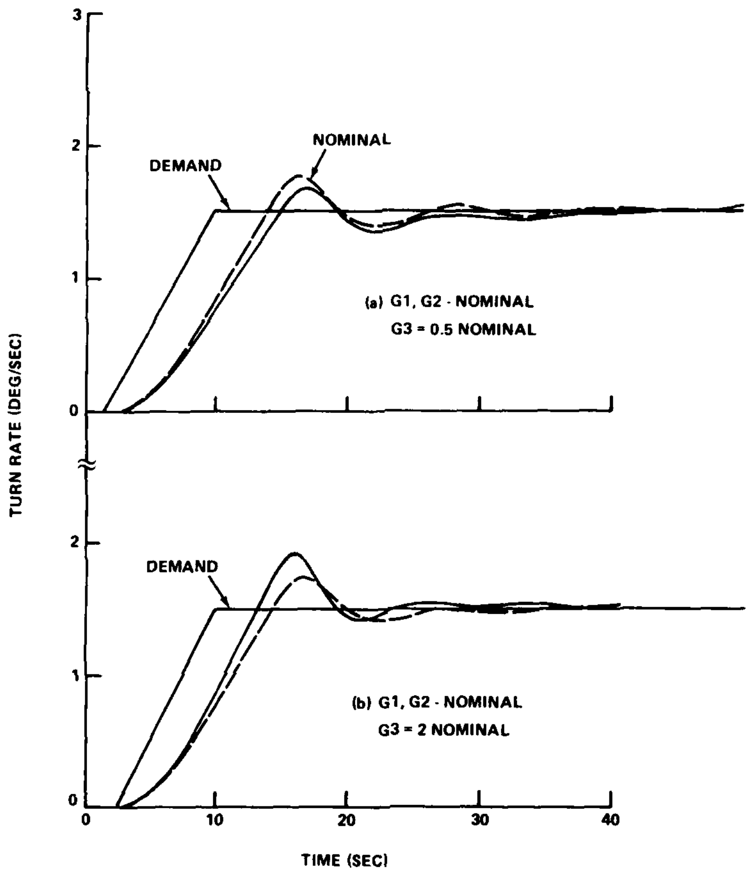


FIGURE 7. SENSITIVITY OF TURN RATE WITH RESPECT TO VARIATIONS IN G3 AT 40 kt

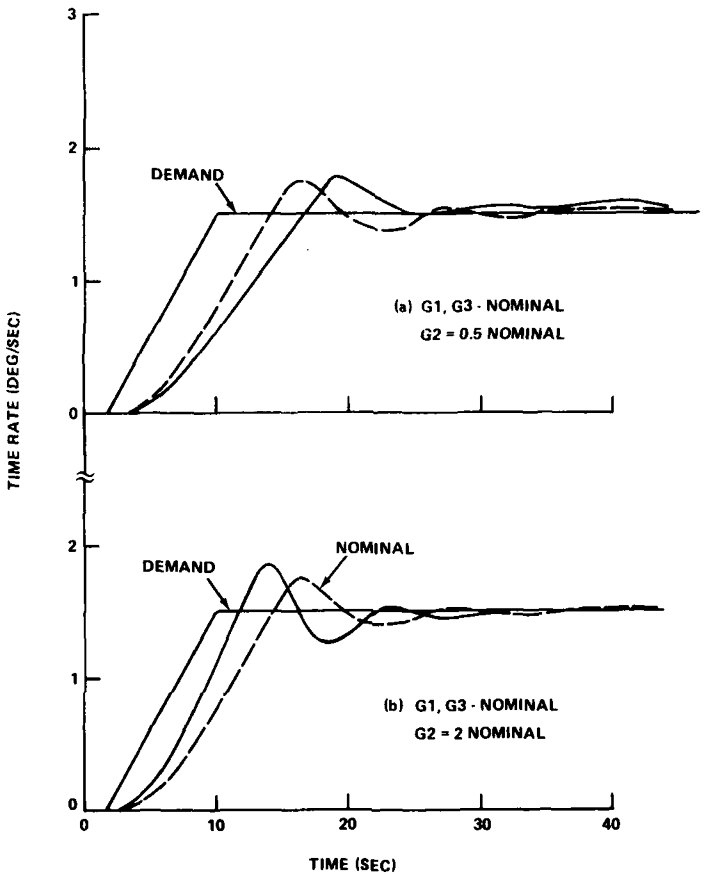


FIGURE 6. SENSITIVITY OF TURN RATE WITH RESPECT TO VARIATIONS IN G2 AT 40 kt

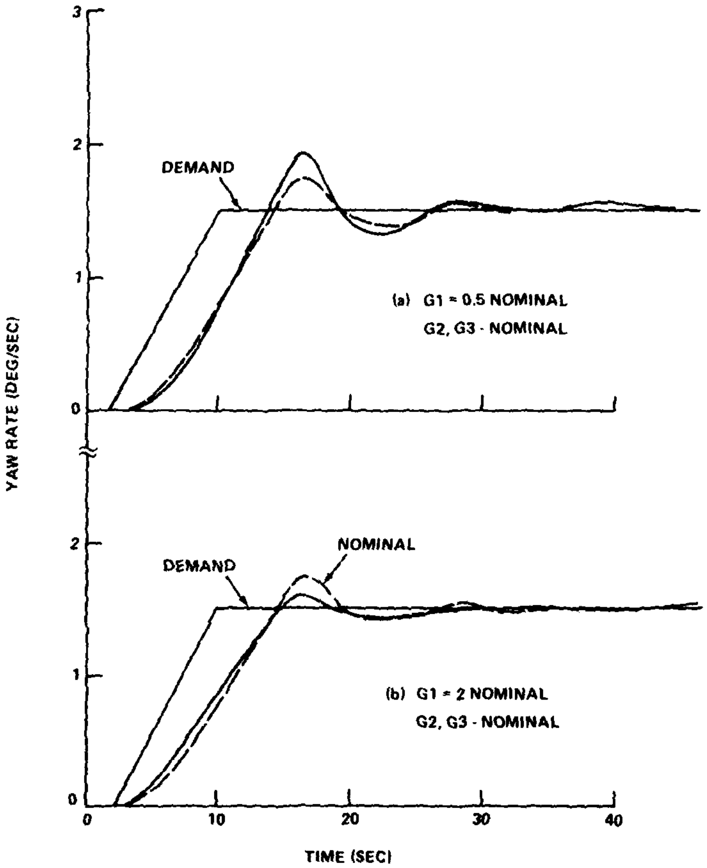


FIGURE 5. SENSITIVITY OF TURN RATE WITH RESPECT TO VARIATIONS IN G1 AT 40 kt

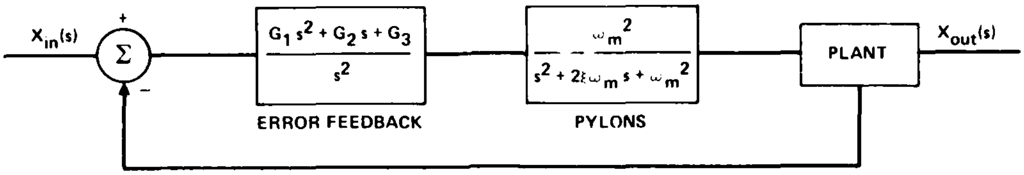


FIGURE 3. SCHEMATIC OF THE JEFF(A) AUTOMATIC CONTROL SYSTEM

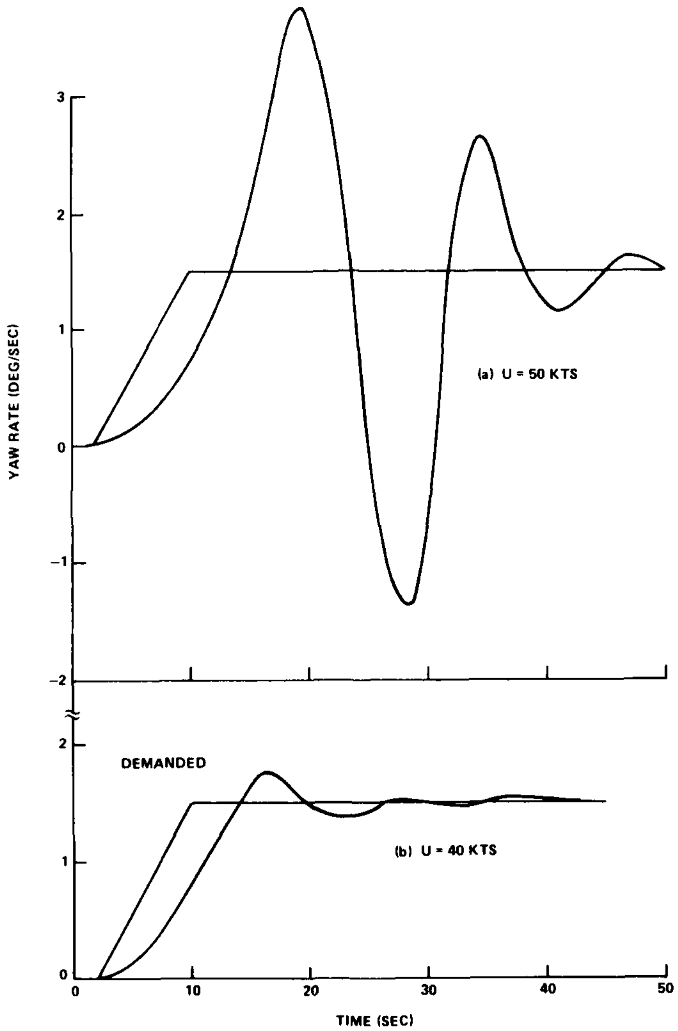


FIGURE 4. SIMULATED TURN RATES FOR JEFF(A) AUTOMATIC CONTROLLER TURNS AT 40 AND 50 kts WITH NOMINAL GAINS

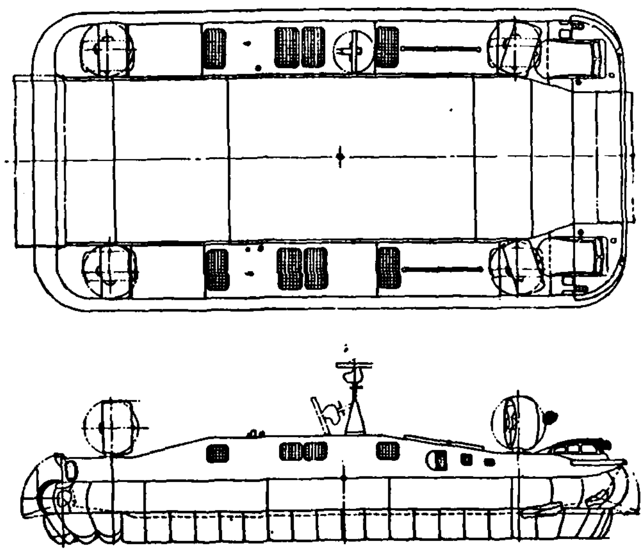


FIGURE 1. JEFF(A) LAYOUT

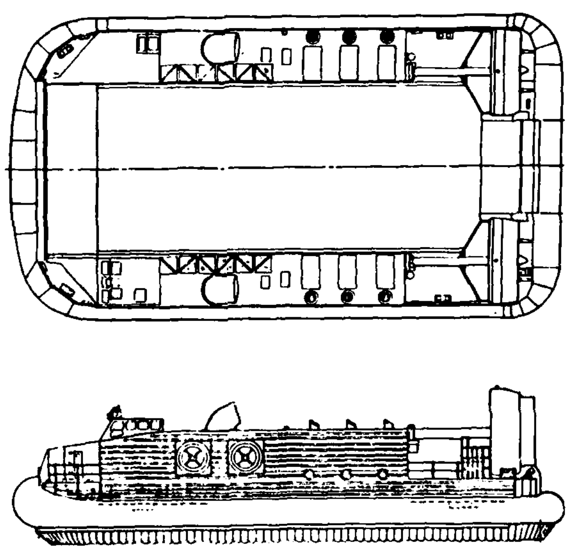
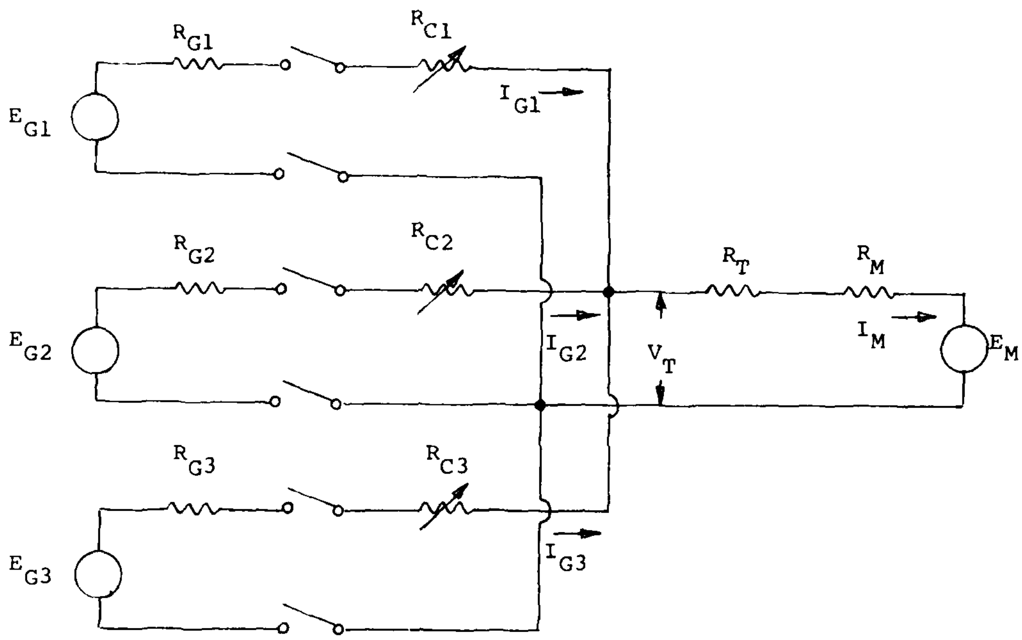


FIGURE 2. JEFF(B) LAYOUT



E_G - Generator internal voltage

E_M - Motor internal voltage

V_T - Terminal voltage

I_G - Generator current

I_M - Motor current

R_G - Generator resistance

R_C - Rheostat resistance

R_T - Transmission line resistance

R_M - Motor resistance

Figure 2. Basic Electrical Circuit

The turbine governor model is based on an operational envelope where the governor has no influence on turbine operation until the design speed is exceeded. Above design speed, the governor cuts back the throttle fuel flow command as required. If the turbine speed continues to increase, the governor will cut the fuel flow to idle command and hold it there until the turbine shutdown speed is reached. Normally, the fuel flow would be "killed" shutting down the turbine, but the turbine torque equations are not valid for zero fuel flow. Instead, a turbine abort signal will be generated to "flag" subsequent data output while the throttle is held at idle.

The fuel flow control model is defined as a simple time ramp function for fuel flow commands which tend to increase fuel flow. The fuel flow control model will immediately track commands which tend to decrease fuel flow (this could also be made a ramp). The computer model is set up to permit the maximum rate for increasing fuel flow to be easily changed between simulation runs, if desired.

The approach taken in developing the simulation model has been to provide a wide latitude in the types of investigations that can be performed. For example, the integrated analog computer model mentioned above is primarily a model for investigating system dynamic response and has been designed to permit the following control modes:

° Full manual control of all forward and reverse operations, including crash reverse.

° Full manual control with an automatic crash reverse control.

° Automatic control of basic forward and reverse operations with an independent automatic crash reverse control capability.

° Fully-integrated automatic control of all forward and reverse operations, including crash reverse.

Most investigations thus far have concentrated on the second mode of operation listed above. In this mode, the operator has manual control over the generator fields, the turbine throttle (fuel flow) commands, and the transfer switches used to put systems on line in the forward or reverse power mode. The ultimate control mode is the fully-integrated automatic control mode in which an automatic control system would exercise direct control over the generator fields and turbine throttles. The turbine on-line interlock, forward/reverse control, and automatic crash reverse functions would be incorporated in the automatic control system.

Since the analog computer model has been implemented on the analog portion of a hybrid computer, a wide range of automatic control philosophies and system designs can be implemented and investigated. Implementations of the model required the use of 75% of the analog and digital logic components contained in a fully expanded EAI 680*. On this basis, two 680's would be required to model the complete propulsion system for the SWATH vehicle.

The digital computer portion of the hybrid computer has been used to set up and document the analog computer model for each simulation run. Also, a data acquisition program was written in FORTRAN IV to

* A 10-volt analog computer system manufactured by EAI (Electronics Associates, Inc.)

demonstrate some of the additional capabilities and advantages of the hybrid computer simulation.

The equations describing the turbine and electrical machinery can be altered by changing the coefficients, and the propeller and ship dynamics can be modified by reprogramming the particular function generators. Thus, the use of the full capabilities of the hybrid computer has resulted in a very general simulation model which can be easily modified to simulate a wide range of propulsion system applications.

RESULTS AND DISCUSSION

During the simulation checkout phase, a specific operating sequence (standard run) was adopted to provide a consistent baseline for conducting simulation runs with different system parameters. The standard run consisted of bringing two turbine/generators (initially idling) on line one at a time, accelerating to high speed and then crash reversing from a steady state high speed condition. Additional runs included: repeating the standard run but with an increased ramp time for the rheostat; a transition run where duty is transferred from the 3.55 MW turbine/generator to a 15 MW turbine/generator; a crash reversal from a low power level; a field control start-up of the 3.55 MW turbine/generator; and a field transient run.

Figure 3 illustrates the standard run sequence for paralleling two 15 MW turbine/generator systems to the propulsion motor. After all switches are open, all magnetic fields are at design value, all machines are at rest and the rheostats are at maximum resistance. Idel fuel flow is applied to the turbines at t_1 and they reach idle speed of 1500 rpm. The generator open circuit voltage reaches 150 volts. At t_2 one turbine/generator is brought on line be closing its polarity switch (ahead mode) and ramping down the rheostat resistance to zero, it minimum value (one second ramp time). Due to the high inertia of the propeller and entrained water, the rheostat reaches zero resistance before the motor is at operating speed ($\sim$ 18 rpm). Thus, there is insufficient motor back EMF to prevent a current spike. However, at idle fuel flow rate there is insufficient energy to exceed current or torque ratings of the system during this transient (generator current spikes to 27,000 amperes). If fuel flow were above idle, a surge current in excess of design specifications might have occurred.

Once the motor is up to speed under idle fuel, the throttle is increased to accelerate the on-line turbine such that the generator voltage increases to match the open circuit voltage of the off-line idling generator (150 volts). At this point, t_3, the on-line generator current is 38,750 amperes, the motor speed increased to 78 rpm and the ship reaches a speed of 18.9 ft/sec. (5.8 m/sec.). The second turbine/generator is brought on-line as the polarity switch closes when the voltage differential across the switch is near zero, t_4, and the rheostat resistance is ramped down. By matching the voltage of the two gnerators, a current surge is avoided.

Fuel flow to both turbines is increased to design value to begin acceleration at t_5. Shortly thereafter, the generator currents peak then level out at slightly lower values. The peaks occur as propeller slip (or loading) changes during ship acceleration to a new steady state speed. After enough time has been allowed for stabilization, the system reaches the steady state high speed condition at t_6. The turbines are operating at 3500 rpm and the generator currents reach

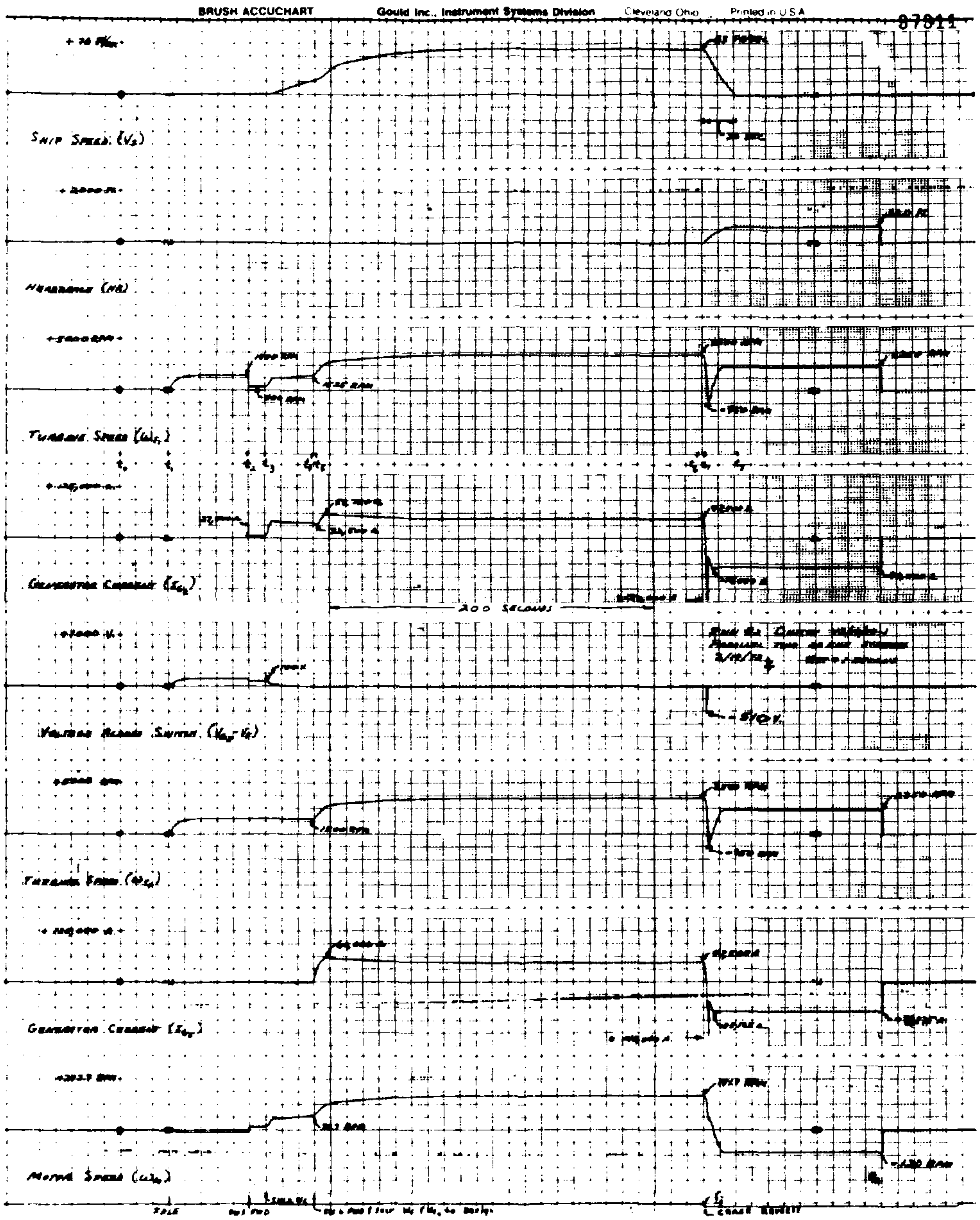


Figure 3. Standard Run: Two 15 MW Turbogenerator Systems

47,500 amperes each. The motor is running at 182 rpm and the ship speed is up to 63 ft/sec. (19.2 m/sec.).

The crash reversal is initiated at t_7, as the turbine fuel flows are cut to idle command and the rheostats are ramped up to maximum resistance. The polarity switches are reversed as the motor current reaches zero and then the rheostats are ramped down again to zero resistance. The fuel flow is then increased so as to maintain 150% rated torque on the motor. The motor eventually reverses and the ship is stopped at t_8.

In contrast to the smooth system response obtained when bringing a ship up to speed, the crash reversal causes several significant transient conditions. The kinetic energy of the ship and propeller drives the motor as a generator into the open circuit caused by the maximum resistance of the rheostat and the open polarity switch. As the polarity switch closes in reverse and the rheostat ramps down to zero resistance, the "generating" motor is loaded by the generator. Negative current flows in the motor and both machines are slowed down. The one second ramp time on the rheostat does not allow the machines to slow enough to reduce the voltage across the system resistance. Thus, current spikes exceeding maximum limits are generated. Additional runs showed that the current spikes can be eliminated by using longer rheostat ramp times, however, longer ramp times requires additional heat dissipation capacity in the rheostat.

As the motor continues to feed power back into the generator, the power is absorbed by the turbine which continues to slow down and then reverse. THe simulation was not designed to prevent reverse turbine rotation nor to determine if this action would damage the turbines or other system components. Reverse rotation spanned a two or three second interval during which fuel flow was increasing; however, the fuel flow increase was limited by constraints on maximum current and torque. The reverse turbine rotation was a result of the amount of power from the motor which had to be absorbed. If more power is dissipated in the rheostat, turbine reversal can be prevented.

A high-frequency, low-amplitude oscillation occurs in the generator currents as the motor accelerates in reverse, which is due to turbine governor response. During the rapid transient, the turbine/generator systems did not share the load equally causing a "push-pull" action between the turbine governors. Faster governor response can minimize this problem.

It should be noted that the propeller dynamics model used in the simulation neglects the effects of propeller cavitation. Thus, stopping the ship in 20 seconds with ahead reach of 620 feet (189 m) may be an optimistic projection. Also, if the control sequence is modified to limit the peak torques and currents to below maximum values, both stopping time and head reach will increase accordingly.

From other runs made, an operating procedure was established for transition from using the 3.55 MW turbine/generator system to a 15 MW system without losing power. The key factor in this transition is the overlap of generator output voltages. With the 3.55 MW system operating near maximum power output, a 15 MW system can match the line voltage to achieve a bumpless transition. Similarly, with a 15 MW system operating on line near idle turbine speed, the 3.55 MW turbine can be brought on line by matching voltages. An interlock must be included so that the 3.55 MW system is not on line when a 15 MW turbine is on line at high power.

The results of the generator field control runs indicate that it is possible to bring the motor up to design speed and then crashback without using the rheostat or polarity switches. However, should the generator field become too low, the motor will supply power to the generator and accelerate it independently of the turbine speed governor. If the generator speed gets too high, the turbine will shut down automatically. The system is so sensitive to small changes in generator field that using this kind of control would be extremely difficult and is not recommended.

The sensitivity of the system to changes in generator or motor tield is again illustrated by the field transient runs. Considering the case where two 15 MW turbine/generators drive the motor at design conditions, if one generator should lose field the second generator and the motor both supply power to what appears as a short across the generator. Their combined currents exceed the ratings of the shorted generator. Loss of motor field under the same operating conditions result in current surges in both generators which exceeds maximum ratings. Although magnet quenches are not anticipated, there must still be some protective measures included in the control system. The superconducting magnets designed for these full scale machines quench in 4-5 seconds for generators and 10-20 seconds for motors which should allow ample time for an automatic control sequence to open rheostats isolating the faulted machine before current limits are exceeded.

SUMMARY AND CONCLUSIONS

A simulation model of a superconducting propulsion system for a twin-screw SWATH has been developed, including turbines, generators, switchgear, motors and propellers as well as vehicle drag characteristics. Some assumptions and estimates were necessary in order to produce the model. As more accurate data becomes available, the model can be revised to reflect the new inputs. The model was exercised over a range of operating parameters to study system response and capabilities. Rheostat ramp time and governor response time were identified as critical parameters.

The simulated system performed well for normal operating modes, and behaved like a stiff gear train with high inertia. However, during crash reverse and certain fault conditions, it was subject to rapid transients which could be damaging if automatic protection is not incorporated. Such equipment is within the state-of-the-art and should represent no significant increase in system complexity. The following are recommended for incorporation as normal protection and safety devices for the propulsion system:

° a rheostat ramp rate controller which employs current sensing to reduce rheostat resistance as quickly as possible without exceeding safe current limits

° an interlock to prevent high fuel flow rates to the 15 MW turbines when on line with a 3.55 MW turbine

° an interlock to prevent closing polarity switches in the wrong direction

° a means to automatically remove a machine from the line if a magnetic quench occurs.

Allowing the free turbine to reverse rotation maximized the performance of the propulsion plant during crash reversal. However, if braking resistors are used to absorb the energy, the free turbine can be limited to some minimum forward speed. Further study of the power absorbing aspects of the free turbine should be considered including the following:

° investigate the ability of the free turbine to reverse rotation (there is no inherent reason why a gas turbine cannot be reversed, but there might be some specific problem with a particular turbine, for example, a lubrication system which works properly only with forward rotation)

° insure that the turbine governor used will respond fast enough to minimize oscillations between two paralleled turbine/generator systems (the existing system may be sufficient since the amplitude excursions are small.)

The simulation model has been developed such that modifications due to changes in system parameter values can be easily accomplished. For this reason, it is a simple process to update the model with the latest component data available. It is also possible to change all the parameters in order to simulate an entirely different propulsion plant and/or propeller characteristic and vehicle dynamics, without major modification to the general model.

Modifications to the simulation model that could be considered are:

° include propeller cavitation

° model a second motor/propeller system in order to evaluate cross-connect operation (specifically one generator powering both motors)

° employ a more accurate transient response model for the gas turbine.

These modifications could be incorporated in order to produce a detailed fault analysis of the full scale superconducting propulsion system.

ACKNOWLEDGEMENTS

The author would like to acknowledge the contributions of C. L. Patterson, who is chiefly responsible for developing the mathematical models and programming them on the analog computer.

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SPEED SENSORS FOR HIGH-SPEED SURFACE SHIPS

By H. K. Whitesel and L. W. Griswold

ABSTRACT

High-speed advanced surface ships require either water or ground speed sensing for navigation and ship control purposes. Since these ships, in general, either do not have contact or have only limited contact with the surface over which they move, they require a remote speed sensing capability. This paper summarizes the principles of operation and analyzes the capabilities of several types of remote speed sensors which might be applied to the requirements of high-speed advanced surface ships. Remote sensors considered here include inertial, doppler, and correlation types using radiated sonic or electromagnetic beams producing scattered return from the surface.

INTRODUCTION

High-speed surface ships (HSSS's) are defined for purposes of this paper as those which operate at high speeds at the air-water interface and have no contact or only limited contact with water. Many types of these craft may also operate over land or ice. Examples are hydrofoil craft, surface effect ships (SES), hovercraft, arctic surface effect vehicles (SEV), and amphibious assault landing craft (AALC) or the air cushion craft (ACC) using the air cushion principle. For vehicles of these types it may be difficult, extremely inconvenient, or impossible to use speed sensors which require contact with the water. The problems of using such sensors will not be discussed here. The purpose of this paper is to present the principles of operation and capabilities of three basic types of remote speed sensor which may be used to sense remotely vehicle speed. These are:

- Doppler
- Correlation
- Inertial

Doppler and correlation sensors involve the use of radar, optical (laser or other), infrared, or ultrasonic devices. Doppler sensors are active radiation emitters. Correlation sensors may be either active or passive (i.e., use self-generated or naturally occurring radiation for correlation). Inertial sensors derive their output from gyros, accelerometers, and integrators and are considered passive sensors.

FACTORS AFFECTING CHOICE OF SPEED SENSOR FUNCTION

Speed sensors for military high-speed surface ships (HSSS's) have five principle functions:

- Navigation

- Tactical Maneuvering
- Weapons Control
- Performance Monitoring
- Ship Control

The sensor requirements imposed by these different functions may be quite different, therefore, the choice of sensor type may be critically dependent on which function is to be emphasized.

SHIP CHARACTERISTICS

Ship characteristics affecting the choice of speed sensors include:

- Forward Speed (component along fore-aft axis) — 0-185 km/hr (0-100 knots)
- Drift Speed (athwartships component) — 0-74 km/hr (0-30 knots)
- Roll and Pitch — $\pm 10°$
- Yaw — $\pm 45°$
- Surface Environment - water, ice, snow, sand, mud, vegation, etc.
- Mission Time

Some designs of vehicle may permit use of in-water sensors. The decision to use them will have to be based on each individual craft designer. Vehicle configuration may even impact significantly on choice of remote sensor approach. In general it appears that the most difficult choices are faced when selecting a speed sensor for a ship which operates close to but not in contact (or in limited contact) with the water, ice, and land. Such a ship is exemplified by designs of ACCs for amphibious assault or for arctic operations. Considerations of energy form, beam configuration, beam angle, transducer size, sensor location, height above terrain, and shock and vibration all interact with ship configuration and operating mode in dictating the choice of design approach.

SENSOR EVALUATION PARAMETERS

The following characteristics and parameters for each speed sensor are discussed and evaluated:

- Operating principle.
- Predicted accuracy
- Size, shape, and weight of hardware
- Mounting configuration and location
- Type of velocity sensed (water or ground speeds)

- Outputs (heading speed, drift speed, and/or angle of attack).

- Inputs required.

- Ship motion effects (roll, pitch, yaw, and heave).

- Environmental effects (sea state, temperature, wind, fog, water, and mud sprays.

- Detectability (by hostile craft).

- Cost

DOPPLER SPEED SENSOR CHARACTERISTICS

Doppler speed sensors measure the frequency shift of a scattered radiated signal which may be sonic or electromagnetic (EM), either microwave or optical. All the doppler speed sensors considered in this report use the atmosphere as the transmission medium.

The doppler frequency (Δf) is given by:

$$\Delta f = \frac{2V \cos\theta}{C} f_o \qquad (1)$$

where

V is the velocity of the radiating source in the horizontal direction.

θ is the depression angle shown in Figure 1.

C is the radiation velocity.

f_o is the radiated frequency.

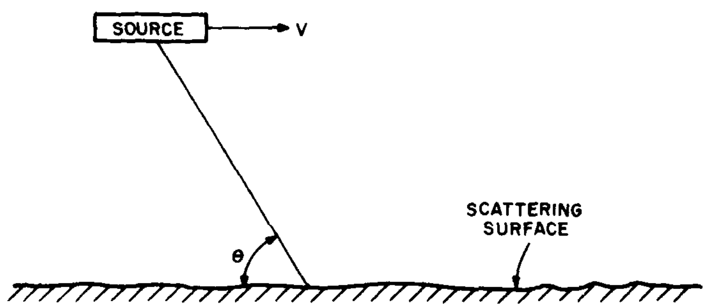

Figure 1. Doppler Speed Sensor Geometry

For the ACC two beams are required to provide two-speed components, one for forward speed and the other for drift speed.

GENERAL

Janus Configuration

A Janus doppler system (either ultrasonic or EM) uses two beams to sense a single-speed component or a total of four beams for two-speed components, shown in Figure 2.

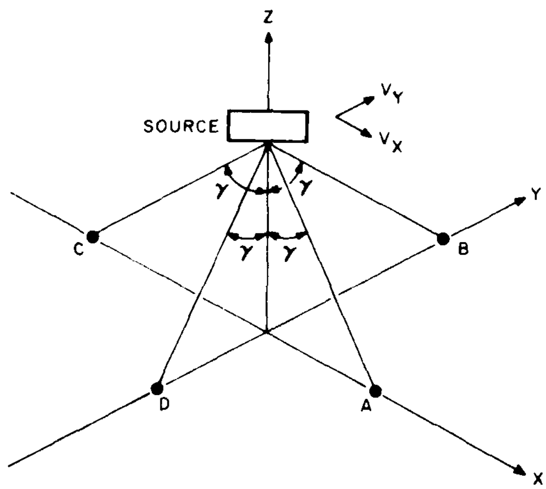

Figure 2. Geometry of a Janus Doppler System

For the Janus configuration, the doppler shift is given by

$$\Delta f = \frac{4Vf_0}{C} \sin \gamma$$

where γ is the incident angle with respect to the vertical, shown in Figure 2. Beam alignment with the X and Y axes is not required if appropriate electronic processing is used.

In general, the advantages of using a Janus configuration are the following:

- Vertical velocity causes no errors in the measurement.
- First-order compensation for trim and list variations is automatic.
- The instantaneous velocity signal excursion, due to roll and pitch, is reduced.

Figure 3 shows the amount of compensation for trim, list, roll, and pitch of the Janus configuration compared with a single beam system. The error in the Janus coppler system is not a function of the incident beam angle as it is for the single beam system. For the Janus system, trim and list angle values of $\pm 8^\circ$ and $\pm 10^\circ$ cause doppler frequency errors of 1% and 1.5%, respectively. The error for a single beam system, due to trim and list, is generally much larger and is dependent on the incident angle as shown in Figure 3. For example, a speed error of +13% to -15% results from a trim or list of ±8° for an incident beam angle of 45° (the angle used by the Ryan model 547).

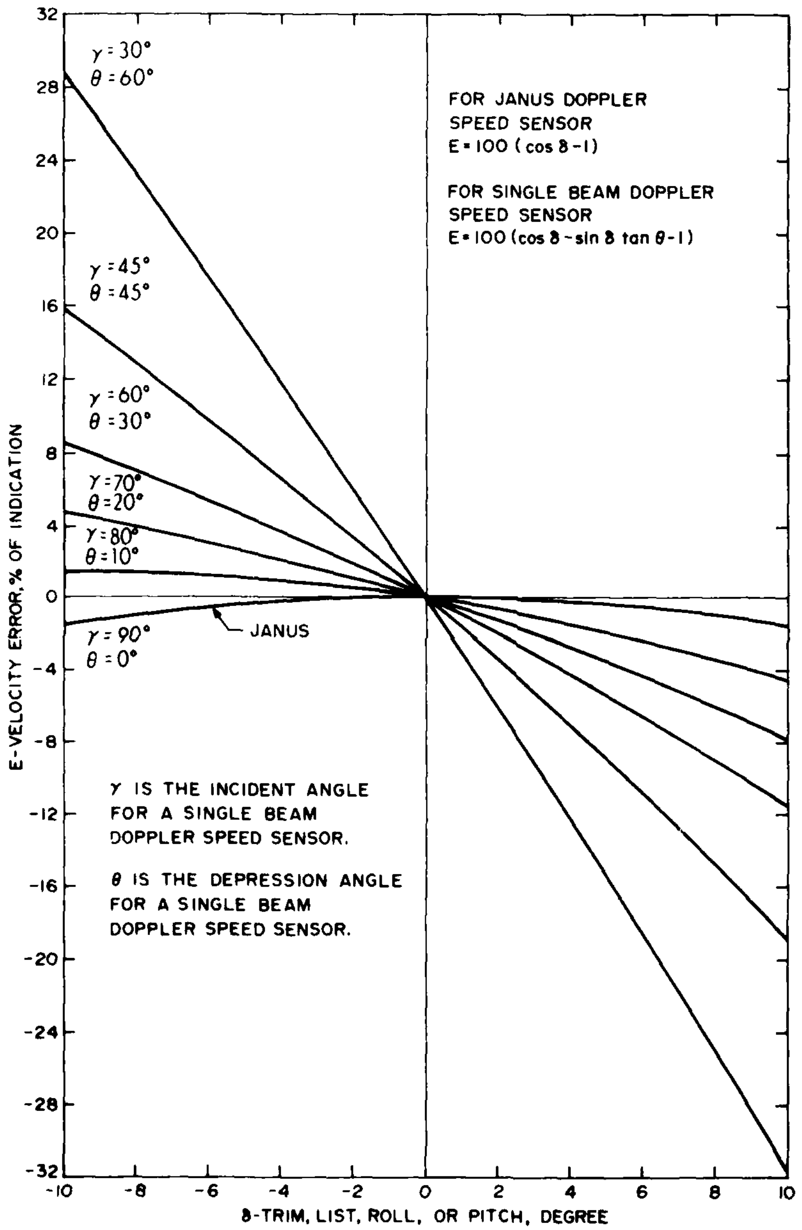

Figure 3. Velocity Error for Single Beam and Janus (Dual Beam) Doppler Sensors

Pitch and roll are dynamic motions which will cause both average and instantaneous errors. The average error due to pitch and roll is the same for both the single beam and Janus doppler systems; however, the velocity signal excursion is much less for the Janus system. Assuming pitch and roll to be sinusoidal, the average doppler error for ± 10 of pitch and roll will be -0.75%.

The Doppler Spectrum

The doppler frequency is not a single, well defined frequency. The radiated beams have a finite beam width which results in a return signal from a spread of incident angles. The doppler frequency shift thus spreads over a range of values. Assuming no frequency spread due to the transport medium, the form of the doppler spectrum is a function of the antenna pattern (usually Gaussian). The half-power width of the doppler spectrum (Δf) is given by:[1]

$$\Delta f \approx \frac{2V\, f_o}{C} \sin \theta_r\, \Delta\theta \qquad (2)$$

where θ_r is the center (average) depression angle, $\Delta\theta$ is the (round trip) half-power width (in radians) of the antenna pattern, V is the craft velocity in the horizontal direction, f_o is the radiated frequency, and C is the radiated velocity. A narrow beam reduces the width of the doppler spectrum and eases detection. Since antenna size increases as the beam width decreases, a trade off must be made between antenna size and beam width.

Wave Form, Pulsed or Continuous

The transmission mode employed in doppler speed sensing systems may be pulsed or continuous wave.

A continuous wave (CW) system usually has separate antennas for transmitting and receiving. There will always be some coupling between the transmitter and receiver which reduces the signal-to-noise (S/N) ratio. However, a doppler system on an HSSS will be operating close to the scattering surface where signal return will usually be high enough to overcome the reduction in the S/N ratio due to coupling.

Pulsed-wave systems eliminate the coupling between the transmitter and receiver because each is operated at separate times. However, if the pulse repetition frequency and range is such that the return signal occurs during the "on" time of the transmitter an ambiguity can result. For the HSSS application, where propagation ranges are very short, the problem will be to reduce the pulse duration below the transit time. For example, the round trip time for radar over a range of 3 meters (10 feet) is approximately 20 nanoseconds.

Use of a frequency modulated, continuous wave (FM-CW) doppler speed sensor gives several unique characteristics as compared with a conventional CW system. The FM technique makes it possible to achieve noise performance equivalent to that obtainable with the undisturbed receiver, as in the pulse doppler system when the transmitter is turned

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[1] Superscripts refer to similarly numbered entries in the References at the end of the text.

off.[2] Also, return from "close in" signals (water spray and rain) is discriminated against by the FM-CW doppler speed sensor.[2]

In general, it appears that CW wave forms will be adequate for a doppler speed sensor on an HSSS. However, use of an FM-CW wave form is probably the best because of discrimination against water spray.

To compare the performance characteristics of doppler speed sensors using sonic and EM waves each must be examined individually. Important parameters include transmission attenuation and interference, scattering surface effects, beam widths and incident angles commonly used, equipment size and cost, and antenna locations. These factors and available equipment designs are discussed in detail for doppler radar, doppler laser, and sonic doppler systems.

DOPPLER RADAR

Transmission Attenuation

Transmission of EM energy at microwave frequencies through clear, dry atmosphere is nearly 100%. Fog causes small attenuation of radar signals for ranges required by aircraft but should not cause significant signal loss over the short propagating ranges involved for an HSSS speed sensor. Rain and snow will cause higher attenuation of radar signals than fog but should also have little effect over the short ranges involved. In general, radars at lower frequencies are less attenuated by fog, rain, or snow. As the frequency increases, absorption and scattering increase. In rain, radar attenuation is usually below 10^{-1} dB/km; even in heavy rain fall, attenuation is below 3 dB/km at a 2 cm wavelength.[3]

Assuming no atmospheric attenuation, the power received at the antenna (for short range doppler radars) is inversely proportional to the range squared (distance to the scattering surface).

Scattering Surface

The surfaces over which HSSS's may operate include fresh water, sea water, sand, gravel, fallow ground, small vegetation, ice, and snow. The doppler radar speed sensor should be capable of operating over all these surfaces.

Figure 4 shows typical scattering coefficients for radar energy scattered from land (4a) and sea (4b). More detailed data is available in references 4 and 5. (The scattering coefficient is defined as the ratio of power backscattered by the target to that backscattered by a perfectly reflecting, hemispherical isotropic scatter). The scattering coefficient, as a function of incident angle, remains nearly constant over land but varies markedly over water as a function of sea state. The smoother the sea the lower the scattered return. For sea-state zero, nearly specular reflection occurs and no detectable energy is scattered back to the radar receiver. It has been estimated that the doppler radar will be inoperable 5% of the time due to calm seas.[6] However, various manufacturers of radar equipment gave estimates much greater than 5%. Aircraft data were used in estimating the 5% figure. An airplane flying at 1310 km/hr (400 knots) at several thousand feet can fly over a calm body of water and not lose signal because of the short transit time and the large area covered by the radar beam. However, the total mission time for an ACC may be on a smooth body of water where no doppler return signal is detectable.

Water Speed Indicated

Speed indicated by an acoustic doppler speed sensor will be surface water speed. If the wake is not used as the scattering surface the major source of error will be surface movement, as discussed in the doppler radar section. If the wake of an air cushion craft is used as the scattering surface, errors will probably be the same as in the doppler radar case, -5% zero offset plus ±2% to ±10% of indicated speed.

As in the doppler radar case, capillary waves will cause acoustic scattering and may contribute to an error in an acoustic doppler speed sensor. The speed of a capillary wave in cm/s is given by[9]

$$c \quad \frac{gL}{2\pi} + \frac{2\,T_1}{\rho L}$$

where

g acceleration of gravity cm/s^2

T surface tension of water (typical value is 74 dynes/cm)

ρ density of water g/cm^3

L capillary wavelength in cm.

The velocity of a 0.335 cm capillary wavelength is about 1.9 km/hr (0.75 knot). Thus, an acoustic doppler sensor, operating at 100 kHz (0.335 cm wavelength) may contain an indicated speed error of up to 1.4 km/hr (0.75 knot) when capillary waves are present.

Transmission Interference

Water and mud sprays will cause transmission interference in an acoustic doppler speed sensor as in radar and optical systems. Of the three types of doppler systems it appears that interference will be greatest in the optical system. Interference will be of the same order of magnitude for radar and acoustic systems. However, an acoustic system may have an advantage over radar and optical doppler speed sensors in that ultrasonic cleaning inherent in transducer operation may prevent water and mud accumulation on the transducers. Radar and optical transducers must incorporate cleaning techniques.

Beam Angle and Width

The incident beam angle for an acoustic doppler system should be as small as possible consistent with retaining enough doppler shift for sufficient resolution. A good design value for the incident beam angle would be about 20° (70° depression angle). At a frequency of 100 kHz and an incident angle of 20° the doppler shift is 1 Hz per 0.01 knot for a single beam system. This is a smaller incident angle than is typical for doppler radar and represents an advantage because it implies that specular reflection will occur a smaller percentage of the operating time.

The beam width should also be set as small as practical to reduce the land-water shift. A 3° beam width is readily obtainable for sonic transducers.

dry snow, ice, and slush. Weak signal returns were received from wet snow, wet smooth concrete, and large undisturbed puddles. The system using the 15° incident angle had higher strength signal returns than the system using the 45° incident angle.

Data on the backscattering of airborne acoustic energy from a water surface is scarce. However, there are considerable data on the scattering characteristics of the ocean surface using waterborne (subsurface generated) acoustic energy. (The terms airborne and waterborne will be used to indicate that the acoustic energy is generated in the air over the sea surface and in the water under the sea surface, respectively).

Scattering of waterborne sound from the air/water surface has been measured at low frequencies (20 Hz to 10 kHz).[32,33,34] The backscatter signals versus incident angle are similar to those curves shown in Figure 4 that apply to radar measurements. In general, as the wind speed increases (higher sea states), the acoustic backscatter energy varies less with incident angle.

In this discussion, it is assumed that airborne acoustic scattering off the sea surface is similar to waterborne acoustic scattering off the sea surface. In general, acoustic energy impinging on a boundary is divided into two wave components, one reflected into the first medium and the other transmitted into the second medium. The relative energies of the reflected and transmitted wave are determined by the acoustic impedances of the two media and the angles of incidence of the waves.[35] For the air/water boundary, 99.86% of the acoustic energy is reflected at normal incidence.[28] The coefficient of energy reflection is the same, independent of the side of the boundary where the sound originates.[28]

The above assumption implies that acoustic scattering from the ocean surface will cause similar errors in an acoustic doppler speed sensor to those caused by EM scattering in the doppler radar speed sensor.* Acoustic scattering will be affected by sea state. Specular reflection will probably occur over calm, smooth water for which an acoustic doppler speed sensor will be inoperable. However, acoustic specular reflection may be encountered for a smaller percentage of operating time than for radar because smaller wavelengths will likely be used, and the coefficient of acoustic reflection from water surfaces is larger. Smaller wavelengths (0.335 cm for 100kHz) yield larger scattered signals. At normal incidence, 99.86% of the acoustic energy is reflected from a water surface.[28] Electromagnetic energy, reflected from a water surface for the same conditions, is only 2%.[36]

Land-Water Shift

The land-water shift for an acoustic doppler speed sensor will probably be the same as for the doppler radar speed sensor (1% to 3%) because the scattering properties are similar. However, the land-water shift can also be reduced by using the same techniques as used for doppler radar.

* The similarity of scattering characteristics of acoustic beams and EM beams is approximate but still sufficient to determine approximate errors, discussed here. Additional details concerning the theory and measurements of acoustic scattering from the sea surface may be found in the references.

- Scattered power from an acoustic beam is due to wind and temperature fluctuation.

- No power to scattered at 90° to the direction of propagation.

- Wind fluctuations do not produce scatter at 180° (backscatter) to the direction of propagation.

- Most of the scattered sound is at angles less than 90° (forward scatter) to the direction of propagation.

Later experiements, based on Little's analytical work, have confirmed his conclusions. Beran and Willmarth have reported the design of a wind speed measuring system using the doppler shift in a bistatic acoustic sounder.[29] Hall, Wescott, and Simmons have reported the design of an acoustic sounder for measuring atmospheric temperature.[30] This means that temperature nonuniformities and wind turbulence must be considered in designing an acoustic doppler speed sensor but will not cause insurmountable problems. For example, the transmitter and receiver might be mounted to intersect at 90° where, theoretically, there is no scatter due to temperature and wind fluctuations.

Rain will cause scattering of acoustic waves. Scattering will be severe if the wavelength is on the order of the raindrop diameter.[28] Since acoustic reflection from an air-water interface is greater than for EM reflection, there should be greater scattering by rain for an acoustic beam than for a radar beam, assuming the wavelength is the same.

Even though transmission of acoustic energy through the atmosphere is subject to the above limitations, it is felt that an acoustic doppler speed sensor could be successfully designed to operate over the short transmission ranges involved on an HSSS to an accuracy of 1%. Members of industry, involved in manufacture of acoustic systems, support this viewpoint.*

Surface Scattering Properties

The acoustic scattering properties of the surfaces over which the HSSS will operate are similar to those observed for radar at similar wavelengths. Markson and Stern[31] used an airborne acoustic system to measure the backscatter from rough plane surfaces over the frequency range of 40 to 200 kHz. Scattering surfaces used included: No. 3 Snow Rock (stones of the type used in railroad track ballast), No. 1 Snow Rock (gravel), and dichondra (a lawn clover). The backscatter was nearly constant over incident angles ranging from 0° to 60°.[31]

Wachpress,[22] in attempting a design of an ultrasonic speed sensor for automobiles, also collected some data on backscattering of acoustic energy from plane surfaces. Incident angles used were 15° and 45° at a frequency of 200 kHz. Strong signal returns were received from surfaces including asphalt and concrete pavements, dirt roads, gravel, weeds,

*One of the authors discussed use of an airborne acoustic speed sensor for ACV's with Sperry Marine Systems Division and Burnett Electronics Laboratory Incorporated. Sperry has designed and marketed a waterborne sonic doppler speed sensor. Burnett has proposed designing an airborne acoustic speed sensor for ACV.

The nonlinearity for the acoustic Janus doppler at 150 km/hr (80 knots is 0.12% which is negligible compared to other errors present. In practice, the electronics of a single beam system could be designed to compensate for the nonlinearity.

Atmospheric Transmission of Acoustic Waves

Temperature will cause a significant variation in the speed of sound through the atmosphere. The velocity of sound in dry air is given by[22]

$$C = 20.05 \sqrt{T}\,(m/s) \quad 38.95 \sqrt{T} \text{ (knots)} \quad . \qquad (9)$$

where T is absolute temperature in degrees Kelvin. The variation of sound velocity in air at standard atmospheric pressure from 282 to 316 m/s (622 to 697 knots) over the temperature range -98 to 38 C (0 to 100 F) results in a doppler frequency shift of about 12%. This error can be compensated by transmitting a constant wavelength instead of a constant frequency.[24] This is accomplished by measuring the speed of sound at the transducer and assuming there is no significant variation over the transmitting path (as would usually be true on an HSSS). This will reduce the error due to variations in the speed of wound to less than 0.5%. Another compensation method involves directly sensing the air temperature and varying the incident angle of the beam.[25]

The speed of sound through the atmosphere also varies with humidity. The magnitude of this change is of the order of 0.3% and is considered to be insignificant compared to other errors.[22]

Wind will cause errors in the doppler frequency because it changes the acoustic velocity relative to the receiver. Wachspress computes an error, in doppler frequency due to wind, of -0.45% for a wind speed of 85 km/h (50 mi/h).[25] The Janus configuration reduces the wind errors by a factor of about ten.[25]

Power received by an airborne, acoustic doppler speed sensor is much smaller than the inverse second power of the range, predicted by geometry.[22] The atmosphere causes attenuation of a directional acoustic beam by absorption and scattering, which is a function of the molecular structure of air, turbulence, and local variations of temperature, wind, and humidity.

Operation of an acoustic doppler system at low frequencies (40-100 kHz) improves absorption characteristics. Massa[26] presents data showing that absorption increases markedly with frequency, increasing from 0.66 dB/m (0.2 db/ft) at 20 kHz, 16.5 dB/m (5.0 dB/ft) at 300 kHz for "average" atmospheric conditions of 75 F and 37% relative humidity. The presence of water vapor in the atmosphere will change the absorption of acoustic waves. An increase in relative humidity will cause an increase in acoustic absorption for frequencies below about 10 kHz.[27] The opposite is true for frequencies above 10 kHz. Absorption due to water in the form of fog is negligible.[27]

In an airborne acoustic doppler speed sensor, scattering of the sound beam will cause attenuation of the power returned to the receiver and broadening of the doppler spectrum.[22] (It appears that scattering is caused by nonuniformities dimensionally the order of the wavelength of the sound.)[28] Little discusses atmospheric scattering using a Kolmogorov spectrum and shows that:[22]

- Power per beam - 10 milliwatts.
- System power - 100 watts.
- Voltage - 115 volts, 60 Hz.

The power supply and electronics processing units are usually packaged separately from the laser.

ACOUSTIC DOPPLER

Acoustic waves provide an alternate means of designing a doppler speed sensor for HSSS's. The basic doppler principle is the same as for the EM wave doppler, but the transmission and scattering characteristics of acoustic waves offer several unique advantages. Since the speed of sound through the atmosphere is much less than that of EM waves, very small wavelengths are achievable at relatively low frequencies. However, the interaction of sound waves with the atmosphere is much greater than for most EM waves. For example, the scatter of acoustic waves due to atmospheric inhomogeneities is typically one million times greater than for EM waves (excepting the infrared and mm wavelengths).[22] Atmospheric parameters such as temperature, humidity, wind, and turbulence will have a pronounced effect on an acoustic doppler.

Acoustic doppler has short range and low-speed limitations relative to doppler radar due to the interaction of sonic waves with the atmosphere and their lower velocity. However, acoustic doppler systems using the atmosphere as the transmission medium are potentially applicable for all HSSS's Acoustic doppler speed sensors using water as the transmission medium are applicable for the SES and hydrofoil where a rigid mechanical contact with the water is maintained and bubble sweep-down is not present. Sperry, Raytheon, and Ametec/Straza (Marquadt) have designed and tested waterborne acoustic doppler speed sensors.[23,24]

Nonlinearity

The doppler frequency for the acoustic doppler system is a nonlinear function of velocity because the speed of sound is not large compared with the speed of the HSSS. For a single beam system the general equation for the doppler frequency is given by:

$$\Delta f \quad \frac{2Vf\cos\theta}{c}\left[1+\frac{V\cos\theta}{c}+\frac{V^2\cos^2\theta}{c^2}+\frac{V^3\cos^3\theta}{c^3}+\ldots\right] \quad (7)$$

Doppler frequency is considered as linear for EM waves because the velocity of these waves is so much greater than the vehicle speed. However, these higher order terms are significant for acoustic doppler systems. For example, for a ship speed of 150 km/hr (80 knots) the nonlinearity would be approximately 1.62%.

The Janus configuration results in some compensation because all odd order terms cancel resulting in[23]

$$\Delta f = \frac{4Vf\cos\theta}{c}\left[1+\frac{V^2\cos^2\theta}{c^2}+\frac{V^4\cos^4\theta}{c^4}+\ldots\right] \quad (8)$$

Beam Angle and Width

Over most surfaces the doppler laser return contains a Gaussian distribution of frequencies. As in the case for radar it is advantageous to keep the incident angle small and the beam width small. As implied in Figure 6, the incident angle should be in the range of 5° to 10° (from vertical) to reduce the probability of specular reflection. This is a smaller angle than is typical for doppler radar systems.

Typical beam widths for lasers range from 0.5° to 0.005°. For a doppler laser speed sensor on an HSSS, assuming the system is mounted 3 meters (10 feet) above the surface, beam widths of about 0.5° should be used to illuminate a large enough surface area to obtain some spatial averaging.

Shock and Vibration Problems

Practical problems arise for doppler laser systems used on board an HSSS because of the operating environment. Rough treatment of the equipment can be expected. Severe shock and vibration will occur which will cause alignment and modulation problems. If a laser doppler system is to be developed for use on any HSSS, the rugged operating environment of these ships must be considered throughout the design.

Sensor Location

A through-the-water laser doppler sensor could be mounted on the SES and the hydrofoil on any mechanical member which penetrates the water surface and is free from cavitation and bubble sweepdown. A simple system design would be a forward scatter system with the transmitting and receiving optics located on separate mechanical members. However, structural flexing and vibration would tend to cause misalignment which would have to be considered in the design.

Doppler Laser Systems Availability

Doppler laser systems which may be adaptable to speed sensor application on HSSS's have been designed by Raytheon,[14] several university laboratories,[15-20] and this Center.[21] Raytheon tested a CO_2 laser doppler system on an A-26 B aircraft at 200 meters (1000 feet).[14] Accuracy was not given but tests did show operability over plowed fields, trees, buildings, asphalt and concrete roads, snow, sand, and rocks. Effort at the universities is mainly directed toward developing the physics of laser doppler flowmeters using both forward scatter and backscatter from a volume within the fluid. This Center constructed a strut-mounted, forward scatter system to measure the speed of a ship. The system required mechanical contact with the water and scattering was taken from within the fluid volume not from the surface.

The general characteristics of a laser doppler sensor using surface reflection to measure HSSS speed are expected as follows:

- Accuracy - 2 km/hr (1 knot).
- Volume - 10 to 60 liters (0.4 to 2 ft^3).
- Weight - 7 to 18 kg (15 to 40 pounds).

Figure 6 shows that rough seas decrease the signal drop-out times. Curves shown are typical of data presented by Jelalian.[12] Signal return increases for incident angles near the vertical. However, even at an incident angle of 85° the inoperative time for calm seas was 25%. Additional data taken from Jelalian's results indicate that signal drop-out at a wavelength of 1.06 μm may be eliminated by increasing the beam width to illuminate a larger, more homogeneous area of the ocean surface.[12] An experiment using a neodymium-doped, yttrium-aluminum-garnet (YAG) laser (1.06 μm) resulted in no signal drop-out. This system was operated at 915 meters altitude with a beam width of 6 mrad, which illuminated a much larger scattering area than the 10.6 μm system. Operation of a laser system at altitudes of 3 to 6 meters (10 to 20 feet) should cause signal drop-out to occur more frequently because a smaller area of the ocean surface would be illuminated. This is a development problem which would have to be overcome.

Land-Water Shift

The land-water shift observed in doppler radar systems will not be significant in doppler laser systems because the beam width and the incident angle are normally smaller. (Smaller incident angles can be used because the higher optical frequencies increase the magnitude of the doppler frequency at a given incident angle.) If the beam is expanded to illuminate a larger surface area, as suggested above, the land-water shift will increase. In designing a doppler laser speed sensor, there is a trade off between beam width and signal return to decrease the land-water shift. There is probably sufficient adjustment range to keep the land-water shift error well below 1 knot and with a high signal return. Otherwise beam squinting or separated beams can be used to decrease the land-water shift as is done for doppler radar systems.

Water Speed Indicated

When the ship is traversing water, a doppler laser speed sensor measures speed with respect to the surface of the water. The water speed measured at any local point on the water surface will vary periodically and randomly with time. The doppler laser system will sense surface movement due to ocean waves and wind. If the wake is not used as the scattering surface, maximum errors in rough seas may be 4 to 11 km/hr (2 to 6 knots), as discussed for doppler radar. If the wake is used as the scattering surface, errors will probably be the same as in the doppler radar case (-5% zero offset plus ±2% to ±10% of indicated speed). The doppler laser speed sensor will probably not be sensitive to capillary waves because the optical wavelengths (μm) are much smaller than radar wavelengths.

Transmission Interference

The water and mud sprays generated by an ACC will cause interference in a doppler laser speed sensor. Relative to a doppler radar system the absorption is high. (EM energy at microwave frequencies penetrates fog and water droplets with less absorption than do the visible frequencies although 10.6 μm CO_2 laser energy also penetrates fog very well.)

The major problem caused by water and mud sprays will be coating of the optical surfaces, causing divergence of the laser beam and/or complete loss of transmission. As suggested for the doppler radar, it may be possible to design a cleaning system. Interference caused by water and mud sprays represents a major disadvantage of a laser doppler speed sensor.

Jelalian has reported reflectivity measurements, over the ocean, from two experimental laser systems operating at wavelengths of 1.06 and 10.6 micrometers.[12,13] (His objectives were to design a 1.06 μm laser altimeter and a 10.6 μm laser doppler radar.) Figure 6 shows the received data rate from a 10.6 μm laser experimental system for sea states 0 through 4.[12] The 10.6 μm system was operated at an altitude of 305 meters with a beam divergence of 0.1 mrad and vertical polarization. Optical heterodyning was used to detect the return signal. The received-signal data rate represents the probability of detecting return signals. The received-data return rate of Figure 6 is analogous to the radar scattering coefficient shown in Figure 4. The depression angles used are much closer to vertical than for doppler radar. (The higher carrier frequencies of optical systems produce adequate frequency resolution with beams closer to normal incidence.)

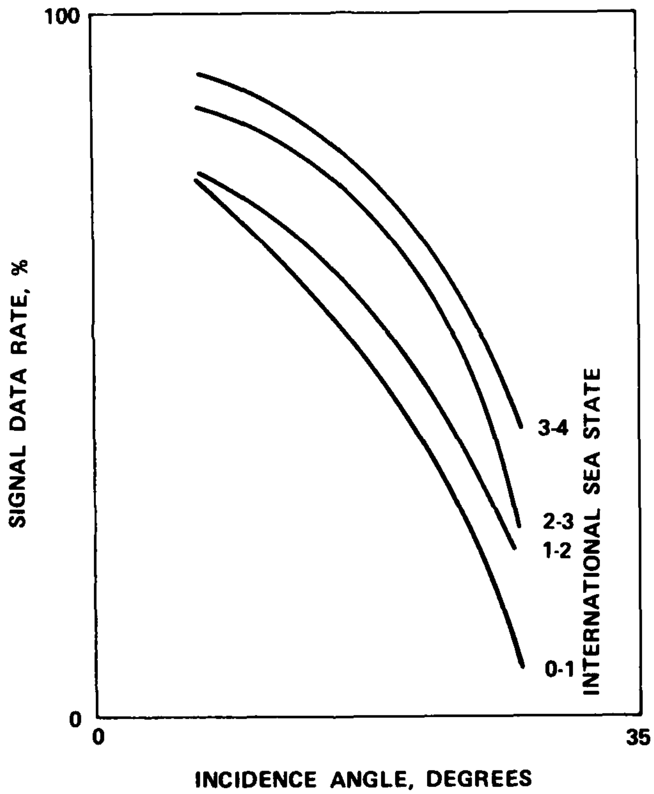


Figure 6. Illustrative Curves of Signal Data Rate for 10.6 Micron Laser as a Function of Incident Angle and Sea State

For the hydrofoil the best location is probably boom mounting or flush mounting on the hull as far forward as possible.

Doppler Radar System Availability

Doppler radar systems which may be adaptable to speed sensor application on HSSS's are fabricated by Teledyne Ryan, Canadian Marconi, Kearfott Division of Singer, and Bendix Navigation and Control Division. The general characterisitcs of these systems in production would be:

- Accuracy - 2 km/hr (1 knot).
- Volume - 19 liters (0.5 ft^3).
- Weight - 7 to 14 kg (15 to 30 pounds).
- Power per beam - 10 to 50 milliwatts.
- Power consumption - 50 watts.
- Voltage requirements - 115 volts (60 or 400 Hz).

The electronics and the antenna are usually included in the same package. The cost of adapting the first system is likely to be in the excess of $100 K. Cost of installation and evaluation must be added to that.

DOPPLER LASER

A doppler laser speed sensor for an HSSS is in many respects similar to a doppler radar system. EM energy, scattered from the traversing surface, is mixed with the transmitted frequency and doppler shift is observed. The frequencies used are much higher than radar and, if a laser source is used, the transmitted energy is more coherent. The following discussion considers visible frequencies, (wavelengths of 0.7 to 0.4 μm), as well as the infrared and ultraviolet regions. Only laser sources are considered because they generate a coherent collimated beam at a smaller cost than by using other optical sources and associated optics.

Transmission Attenuation

Geometric attenuation resulting from beam spreading obeys the same $range^2$ law as radar. However, since a laser beam has a much smaller beam angle (mrad), geometric attenuation is insignificant compared to other causes of attenuation.

Attenuation by the atmosphere of optical frequencies is tremendously variable because of molecular abosrption.[11] The most significant attenuation of a laser beam will be due to absorption and scattering by fog and rain. Over the range involved for an HSSS speed sensor the decrease in signal return will seldom be more than several decibels. Coherence may also be lost due to coating of the optical transmission surfaces.

Scattering Surface

Backscattering of laser light from solid surfaces over which an HSSS traverses will be sufficient to operate a doppler laser speed sensor. As in doppler radar, a problem arises when operating over water. Specular reflection occurs over calm water.

Antenna Location

The antenna of the doppler speed sensor for HSSS's should not protrude beyond the outline dimensions (plane view) of the ship because of possible mechanical damage during maneuvers and docking. Separate mounting locations for each beam for a Janus doppler radar should not be used because of installation alignment and coherence problems.

Possible mounting locations for the antenna of a doppler radar speed sensor on air cushion type ship include:

- Stern Mounting, When the Wake is Used as the Scattering Surface. The major disadvantage of stern mounting is that movement of the water surface in the wake will cause an error and a calibration shift when traversing from land to water and vice versa. Decca (of England) reports the following errors due to surface movement of the wake of a British hovercraft, using a CW, J-band, single-beam doppler radar. There was a -5% zero offset (difference from land readings) which reportedly varied with sea-state and trim. Additionally, there were random errors observed which varied from ±2% of indicated speed for low speeds to ±10% of indicated speed for high speeds, above 90 km/hr (50 knots). Accurate estimate of the wake error is impossible due to lack of knowledge of the precise characteristics of the wake and the dearth of test data on the performance of doppler radars using the wake as the scattering surface. Since only two beams can be used in a stern mounting, the velocity indication will be sensitive to trim and pitch of the ship, shown in Figure 3.

- Plenum Mounting (Under the Deck), Using the Surface Inside the Skirt as the Scattering Surface. Disadvantages of this mounting are the possibility of interference due to multiple reflections from the side skirts and of structural damage to the radar dome by debris. Surface movement of the water under the plenum may cause an error. The magnitude of this error is difficult to estimate because little is known about conditions under the skirt. Plenum mounting of a Janus doppler radar has the advantage that roll and pitch can be more easily compensated.

- Deck Corner Mounting. One corner of the deck could be used for mounting the antenna, using three beams directly over one end and side of the ship to sense speed. A Janus configuration providing compensation for roll and pitch could be used. The disadvantages include sensitivity to the land-water shift (this could be designed out using one of the techniques discussed in the land-water shift section) and the possibility of specular reflection over calm seas.

- Mast Mounting. The antenna could be mounted on a retractable mast which rises above the ship to a height where four beams, in a Janus configuration, could be used to sense speed. This would give compensation for roll and pitch. The disadvantages include sensitivity to the land-water shift and the possibility of specular reflection over calm seas.

The best mounting location for a doppler radar system on the SES is probably under the deck as far forward as convenient. The primary disadvantages are that specular reflection will sometimes occur, unless artificial means are used to roughen the water, and multiple reflections from the sidewalls may interfere.

output of a doppler radar speed sensor. Using an assumed wavelength of 2.25 cm and a 45° depression angle, the doppler error, due to capillary waves, would be equal to 92 rad/s, corresponding to a speed error of about 1 km/hr (0.5 knot). This error could have been calculated from the formula for the speed of a capillary wave as given by[8]

$$c_w = \sqrt{g/k + s\ k} \tag{5}$$

This implies, but does not prove, that movement of capillary waves directly cause a speed error.* The only safe statement is that capillary waves cause scattering of a radar beam and that speed errors are generated, according to equation (3), when capillary waves are present.

Transmission Interference

Based on reported performance with the arctic SEV, one difficulty encountered by the doppler radar was interference from water and mud spray between the scattering surface and the antenna. The interference may be of two types: backscatter from the flying particles of water and mud or coating of the radar dome. For the latter, it may be possible to design a cleaning mechanism using ultrasonics for a "windshield wiper" mechanism. In any event, the possible requirement for an antenna cleaning method represents a disadvantage for doppler radar speed sensors.

Beam Angle and Width

The bandwidth of the doppler return spectrum is controlled by the radiated beam width and the incident beam angle. For detection purposes it is desirable to keep the return beam width narrow, which implies a narrow radiated beam. A narrow radiated beam requires a large antenna, thus a trade off between small bandwidth and large antenna size must be made. The half power beam width (β) for a radar beam is given by

$$\beta = \lambda/L \tag{6}$$

where

β = half power (3 dB) beam width.

λ = wavelength.

L = antenna diameter.

For a 2.25 cm wavelength (13.3 GHz) and a 3° beam width, antenna diameter is 43 cm (17 inches).

There is also a trade off required in choosing the incident beam angle. A large incident angle, approaching the horizontal, aids detection by increasing the doppler frequency but also increases the probability of specular reflection. An incident angle of 45° for a carrier frequency of 13.3 GHz would have a doppler shift of 2583 Hz at a speed of 148 km/hr (80 knots). Incident angles from 20° to 60° are used in airborne doppler radars. Incident angles for a doppler speed sensor on an HSSS in this same range should provide sufficient doppler return at the vehicle speeds considered.

*Wright's expression, equation (3), can be arrived at by substituting equation (5) into equation (1), the doppler equation.

Water Speed Indicated

The speed indicated by a doppler radar is relative to the reflecting surface. If the "effective reflecting surface" has movement relative to the water mass as a whole then an error in speed indication will be induced. Some data reported by Grocott[7] indicates that errors due to water surface movement range as high as 6 knots for high wind conditions (60 knots). Grocott attributed this error entirely to surface water motion. In reality, the error producing mechanism was probably more complex and may have contained errors other than water motion due to radar interaction with the surface. The error attributable to surface movement only was probably significantly less than that shown by Grocott.

Scattering properties of a radar beam are determined by incident angle, polarization, surface roughness, and frequency. Smooth surfaces are characterized by an rms surface roughness of much less than a wavelength.[4]

The mechanism which controls radar scattering[8] from the sea surface is still being debated by scientists and engineers involved in radar development and application. It is generally well accepted that scattering occurs from irregularities (waves) roughly the size of the radar wavelengths.[9] A typical frequency for a radar doppler speed sensor is 13.3 GHz, corresponding to a wavelength of 2.25 cm. This is typical capillary wave size, where capillary waves are defined as having a wavelength less than 2.54 cm (1 inch).[8]

Experiments have shown that capillary waves act as scattering centers and that the doppler frequency is both broadened and shifted when capillary waves are present. Wright has observed that the doppler spectrum of mechanically generated waves peaks at an angular frequency given by[9,10]

$$\omega^2 = g\,k + s\,k^3 \qquad (3)$$

$$k = 2\,k_o \cos\theta \qquad (4)$$

where

g = acceleration of gravity.

s = ratio of the surface tension to water density.

$k = 2\pi/\lambda_w$ = water-wave propagation constant.

λ_w = water wavelength.

$k_o = 2\pi/\lambda_r$ = radar propagation constant.

λ_r = radar wavelength.

θ = depression angle.

Wright also observed wind generated waves in the water tank which showed a doppler spectrum peaked at a frequency related to the depression angle, according to equation (3), but much broadened and a function of windspeed.[10] The angular frequency given by equation (3), due to the presence of capillary waves, would result in an error in the

To eliminate specular reflection of the radar beam the wake of the HSSS can be used as the scattering surface, but more exact knowledge of the effect of water movement in the wake is needed to estimate the error.

Land-Water Shift

All doppler radars experience a land-water shift in the doppler return spectrum. The return spectrum is approximately Gaussian for both land and water but peaks at a lower frequency over water, shown in Figure 5. This is caused by the finite beam width of the radar beam and the fact that lower incident angles result in a higher amplitude return, shown in Figure 4. The shift may vary from 1% to 3% depending on sea state and the incident angle.

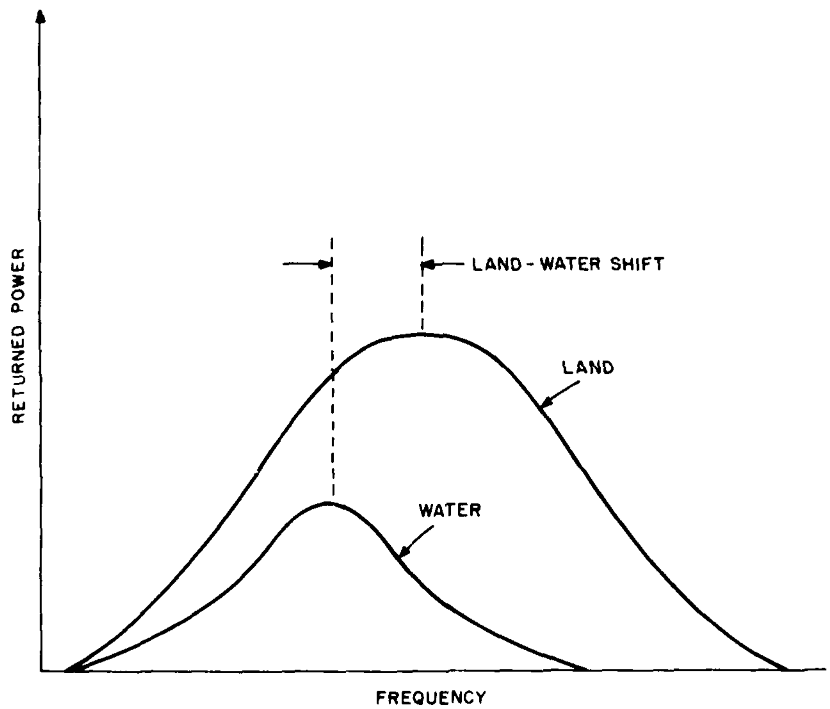
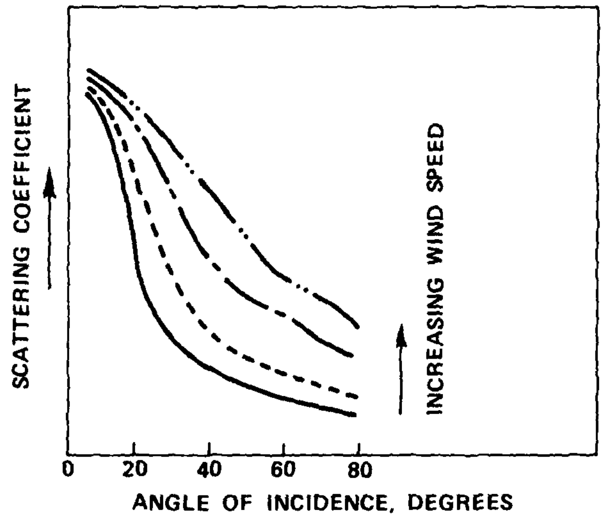
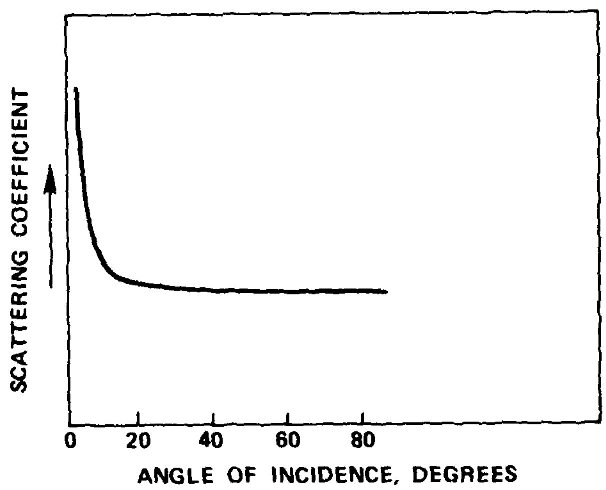


Figure 5. Doppler Frequency Spectrum for Land-Water Surfaces

Various techniques in antenna design have been used to reduce the land-water shift. One method uses two beams slightly separated in lieu of a single beam. For a two-speed component Janus system, a total of eight beams are required by this method. The intersection of the return spectra is only slightly affected by the sea state. Another technique uses a squinted radar beam from a rectangular array. Reduction of the land-water shift to 0.25% is claimed.



(a) Typical Scattering Coefficients for 1-3 cm Radar Over Water



(b) Typical Scattering Coefficients for 1-3 cm Radar over Wet Partially Grassy Terrain

Figure 4. Radar Scattering for Various Angles of Incidence

Shock and Vibration Problems

Structureborne shock and vibration may cause interference in acoustic doppler speed sensors. Transducers may have to be isolation mounted.

Transducer Size

Acoustic transducers for use in air include: vibrating piston, flexurally vibrating plate, magnetostriction devices, and piezo-electric devices.

The transducers will generally be smaller than radar antennas because smaller wavelengths will probably be used. For example, a vibrating piston producing a 100 kHz, acoustic wave of 3° width would have a diameter of 6.4 cm (2.5 inches). A 43 cm (17 in.) square area is required for a radar antenna operating at a 2.25 cm wavelength (13.3 GHz).

Sensor Location

Possible sensor locations for the transducers of an acoustic doppler speed sensor include the same sites as for the doppler radar. On an ACC, possible locations include stern mounting using the water surface under the ship. Multiple reflections may cause problems with the underside deck mounting. In the case of stern mounting, the sensors should not transmit through the "prop wash" as this would cause additional spread of the doppler spectrum. With an acoustic doppler speed sensor of an ACC an additional mounting configuration is possible. A Janus configuration could be located so that a single beam is transmitted from each side of the ship with the sensors separated by roughly the length and width of the ship. Since sonic frequencies are low, each transducer could be phase and frequency locked by a single electronic processing unit. A doppler laser system mounted in this fashion would probably require separate processing units because the high frequencies involved cannot be transmitted reliably to each antenna over the dimensions of the ship.

On the SES the best mounting location would be under the forward part of the deck. On the hydrofoil, the flush mounting on the forward part of the hull is probably the best location. A waterborne sonic doppler could be mounted on the SES and the hydrofoil on any mechanical member penetrating the water surface and free from cavitation and bubble sweepdown.

Acoustic Doppler Systems Availability

Through air acoustic doppler systems are not commercially available at the present time. A system was designed by Arma Division of America Bosch Arma Corporation for measuring the speed of a car.[25] This system was successfully tested by Arma but is not commercially available. Depression angles of 45° and 75° were used at a frequency of 200 kHz. Strong signal returns were received from rough surfaces such as asphalt and concrete pavements, dirt roads, gravel, weeds, dry snow, ice, and slush. Wet snow, wet smooth concrete, and large puddles gave weak returns. Signal drop-out occurred over undisturbed puddles at the 45° depression angle. Signal drop-out never occurred for the 75° depression angle. Maximum speed measured was 16 km/hr (10 mi/hr).

Waterborne (underwater) acoustic doppler systems which may be adapted to speed sensor application on some types of HSSS's are designed by Ametek/Straza, Raytheon,[24] and Sperry.[23] Raytheon has designed a complete acoustic doppler navigation system which is waterborne and operates at 200 kHz from bottom return.[24] Sperry has designed acoustic waterborne doppler speed sensors for docking of large vessels and for ship's speed measurement (ship's log). The acoustic ship's log operates at 2 mHz by using volume return from water scatterers about 2 meters (10 feet) out from the hull. Claimed accuracy is 0.2 km/hr (0.1 knot) or 1% whichever is greater. These systems must have the transducer in the water at all times and have a relatively bubble-free environment.

The estimated characteristics of an airborne acoustic doppler speed sensor for an HSSS, after development, would be:

- Accuracy - 2 km/hr (1 knot).
- Volume - 10 to 30 liters (0.4 to 1 ft^3).
- Weight - 7 to 18 kg (15 to 40 pounds).
- Power per beam - 1 watt.
- System power - 100 watts.
- Voltage - 115 volts, 60 Hz.

The basis for estimating accuracy is not as good as for the doppler radar and doppler laser. In airborne acoustic doppler systems there are more possible sources of error because of strong interaction with the atmosphere.

CORRELATION SPEED SENSOR CHARACTERISTICS

A correlation speed sensor performs a correlation analysis between two signals generated by receivers a known distance apart. The cross-correlation function of two signals is a measure of the similarity of those two signals. A correlation speed sensor may be designed by mounting two receivers in line with the velocity component to be measured. The signal from the upstream receiver is delayed in time until the correlation function is a maximum. The speed is then calculated as the receiver separation distance divided by the time delay.

A correlation speed sensor used on an HSSS will be a remote sensor and may be either active or passive. The incident angle can be set to zero so that specular reflection will increase signal strength at the receivers rather than cause signal drop-out. Most systems will be active and require a beam of energy to be reflected from the surface. The energy form may be acoustic or electromagnetic (including optical or infrared)

Operating Principle

A cross-correlation speed sensor uses the principle of measuring speed by timing the passage of a spot as it moves past two points of known separation.

The operating principle is illustrated in Figure 7. Sensors, labelled A and B, would be mounted to view the surface at a known separation, L, aligned with the heading direction of the ship. The beams intersect the scattering surface at a 90° depression angle. The reflected signal received at sensor B is similar to the signal received at sensor A, except it is delayed in time, τ . The cross-correlation function of these two signals maximizes at time delay τ because the best match of the two signals occurs for this time delay. The heading velocity, V, is given by the ratio of the separation, L, to the time delay, τ. Drift-speed may be sensed with two sensors aligned at 90° to the ship heading.

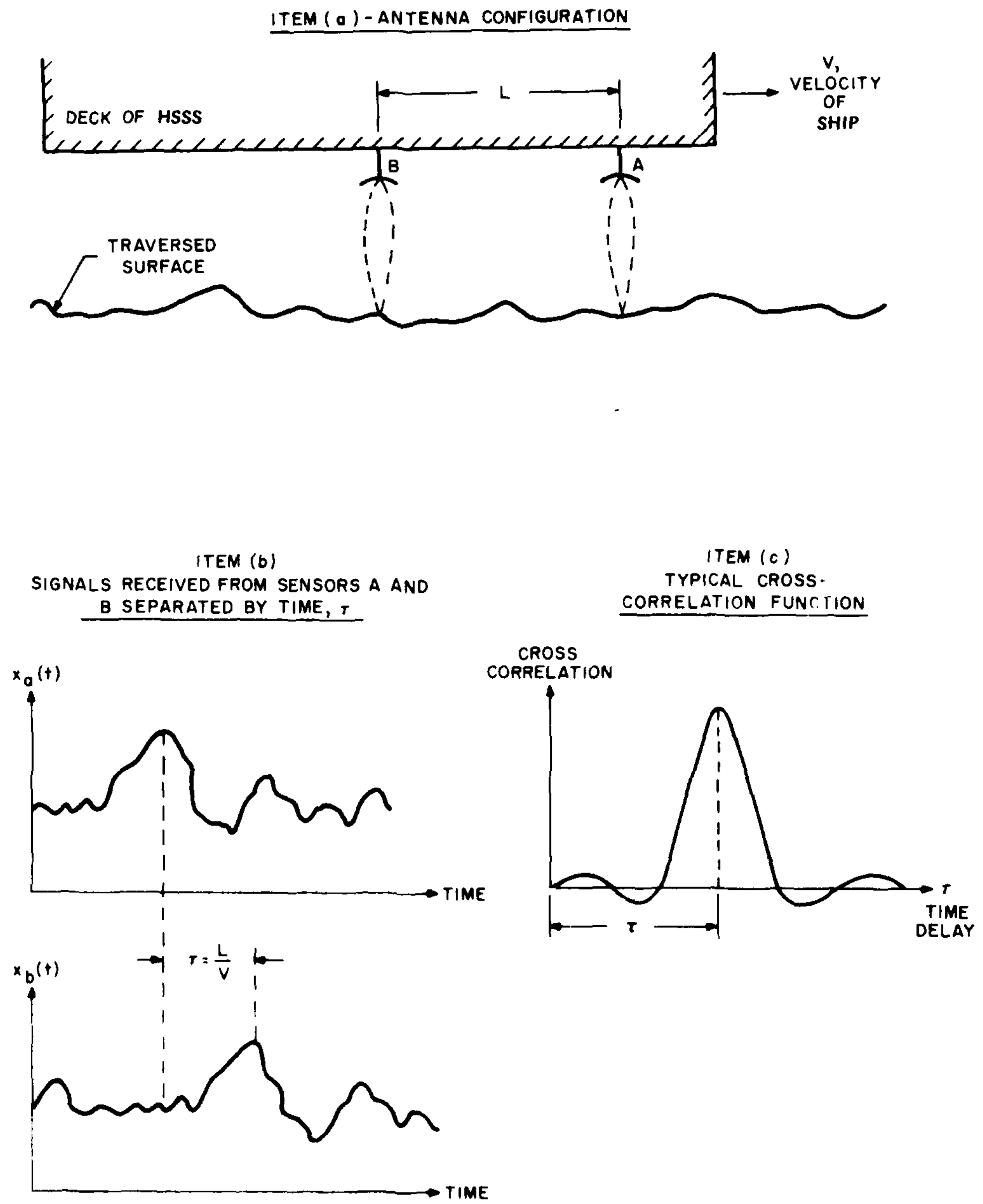


Figure 7. Cross-Correlation Speed Sensor

Signal Processing Circuitry

The signal processing circuitry required to perform the cross-correlation may be quite complex. A simple block diagram, illustrating the circuitry, is shown in Figure 8. This circuit was developed independently by the British Iron and Steel Research Association[37,38] (BISRA) and by Beck.[39,40] BISRA uses the meter to measure the rolling speed of hot steel; Beck applies the meter to the flow rate measurement of solids through a pneumatic conveyor. Currently developed circuitry requires manual manipulation of the time delay until the indicator shows a maximum. This adjustment could be automated to provide a continuous output of the speed.[30]

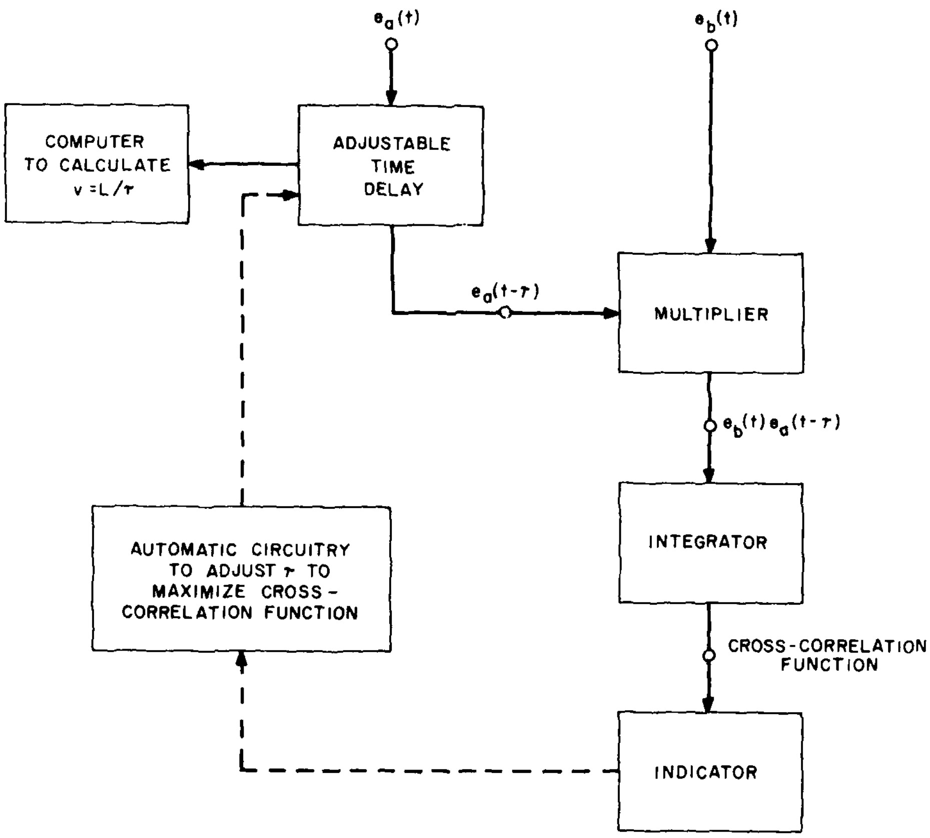


Figure 8. Simple Cross-Correlation Circuitry for Measuring Speed

Additional development on the cross-correlation circuitry has resulted in improvements whereby the multiplier is replaced with a subtractor, a squaring device, and an averager.

BISRA reports measuring the speed of rolling steel to an accuracy of 0.1% at a speed of 3.6 m/s.[38] Miller reports measuring aircraft speed to within 1.0% using a rigid antenna installation.[41]

Development work on fluid velocity and turbulence detectors has resulted in simple and fast correlators using optical devices which may be directly applicable to a cross-correlation speed sensor for HSSC.[42-45]

Effect of Roll, Pitch, and Trim

The effect of variations due to list, trim, roll, and pitch will depend on the mounting configuration. Figure 7 shows a mounting configuration under the deck of the HSSS with an incident angle of 0°. Four beams are required to sense two-speed components. Other mounting possibilities include location at the bow and sterm of the ship. These locations require a nonzero incident angle and result in higher errors for given values of list, trim, roll, and pitch. Figure 9 defines the terms required in calculating the magnitude of the error caused by nonzero list, trim, roll, and pitch; γ represents the mounting incident angle for zero trim angle of the antennas located at points A and B. δ represents the trim, list, pitch, or roll angles as the ship deviates from horizontal.

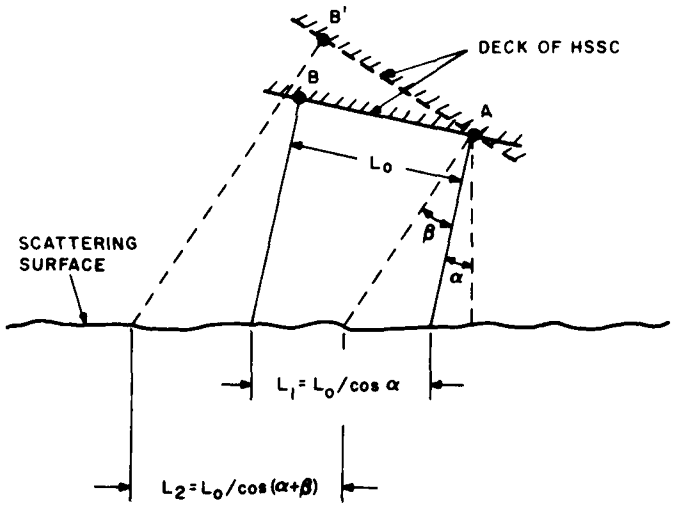


Figure 9. Beam Geometry for Cross-Correlation Speed Sensor

The velocity error for nonzero list, trim, roll, and pitch angles is shown in Figure 10. List and roll angles cause errors in the drift speed indication; trim and pitch angles cause errors in the forward speed indication. At a designed incident angle of 0°, trim angle errors of ±8°, and ±10° cause a heading speed indication error of +1% and +1.5%, respectively. The same value of list angles cause the same drift-speed error. (This is the same magnitude as the Janus doppler error.) Trim and list induced errors in correlation speed sensors vary with incident angle by exactly the same magnitude that trim and list induced errors in single-beam doppler speed sensors vary with depression angle.

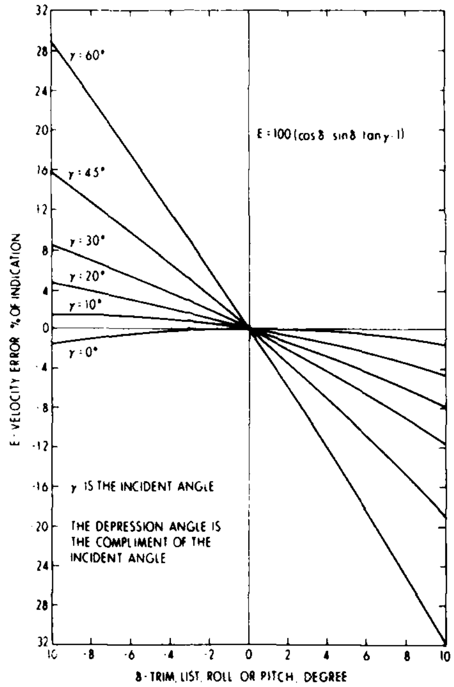


Figure 10. Correlation Speed Sensor Errors Caused by List, Trim, Roll, and Pitch

Pitch and roll will cause both average and instantaneous errors. Assuming pitch and roll to be sinusoidal in time, the indicated velocity errors would sweep back and forth on the error lines of Figure 10. Average errors due to pitch and roll are independent of incident angle. A +10° pitch and roll angle will cause an average error of +0.75% in each of the indicated components (drift and forward speeds). The velocity errors caused by list, trim, pitch, and roll are the same for a correlation speed sensor as for a Janus doppler speed sensor.

Transmission Attenuation and Interference

Transmission of each beam through the atmosphere for a correlation speed sensor will cause the same attenuation as for a doppler speed sensor. For a clear atmosphere the acoustic beam will be more attenuated than the radar beam. In the doppler speed sensor, transmission through a damp atmosphere may cause additional errors because the system may "lock on" the backscatter from fog and rain. The correlation speed sensor may experience similar errors because periodic or random attenuation of the beam due to fog and rain may correlate similarly as does signal level variation due to scattering on the traversed surface. However, this error may be designed out of the system by using appropriate electronics. For example, the frequency of the changes in attenuation due to atmospheric transmission through rain and fog will probably be low relative to the frequencies of interest in the correlator. The change in backscatter from the traversing surface will be more frequent than the attenuation changes due to fog and rain. The resulting errors can then be eliminated with high pass filters between the transducer receiver and the correlator. Transmission attenuation due to the atmosphere should cause no signficant error in a correlation speed sensor.

Scattering Surface Effects

The reflecting surface will affect correlation speed sensors differently than doppler speed sensors. When the ship is traversing over land or any solid surface, the scattering patterns sensed by the rear transducer of the correlation speed sensor will be the same as those sensed by the forward transducer except for a time delay and some decorrelation due to drift velocity. However, when the ship is traversing the sea, the constantly changing pattern of the surface will introduce more decorrelation directly proportional to the relative time delay of the signal received at the rear transducer. This means the two beams should intersect the surface at a separation distance as short as possible. Governing factors will be finite beam width and speed of electronic processing. The nature of the decorrelation caused by the changing sea surface will be a decrease in magnitude of the correlation function resulting in loss of resolution in the indicated speed.

Ideally the depression angle for a correlation speed sensor will be 90 to minimize errors due to list, trim, roll, and pitch of the HSSS. Specular reflection occurring over calm water at a 90° depression angle will increase the signal strength at each transducer and not result in loss of signal as in doppler speed sensors. However, a maximum in the correlation function depends on the presence of scattering centers on the reflecting surface to cause amplitude changes in the reflected signal. For calm sea states, the amplitude of the received signal at each transducer will be constant and it will be impossible to resolve the time delay between signals received at the forward and rear sensors.

Quantitative comparison of operation of doppler radar speed sensors and correlation speed sensors over calm seas is not possible with the information presently in hand. A more detailed study could resolve the comparison and would probably include:

- Mathematical analysis to predict amplitude of doppler return over calm seas.
- Mathematical analysis to predict amplitude variation of signal return for the correlation speed sensor.
- Comparison of electronic equipment detection capabilities, amplitude resolution of a correlator, and frequency resolution of doppler equipment.
- Experimental effort to confirm the mathematically predicted characteristics of reflection and scattering from a calm water surface.

If the wake of the HSSS is used as the scattering surface, specular reflection over calm water will never occur but errors will be induced by wake velocities.

The power requirements for each beam in a correlation speed sensor are expected to be less than for a doppler speed sensor because the depression angle may be set at 90° where most of the reflected power from the surface returns to the transducer. In general, airborne sound reflects better than airborne EM energy from the surfaces over which the ship will travel. For example, over a calm water surface at an incident angle of 0° (vertical), 99.86% of the sound energy is reflected;[28] only 2% of the EM energy is reflected under the same conditions.[36] If the depression angle is set at 80° or 70°, (where the antennas either transmit over the bow or stern) power requirements for each beam will be approximately the same as for doppler speed sensors.

Land-Water Shift

If the correlation speed sensor is operated with an incident angle of 0°, there will be no land-water shift as there is for doppler speed sensors. Even if the incident angle is set at nonzero values, the land-water shift would be a second order effect and well below an allowable error of 1%. This represents an advantage over the doppler speed sensors.

Water Speed Indicated

A correlation speed sensor will sense vehicle speed relative to the surface of the water. The speed sensed can be set by the designer to be relative to wave celerity, capillary waves, or even smaller areas depending on the frequency and type of transport beam (acoustic, EM) used and the filters between the receiving transducers and the correlator.

The amplitude of the signal received at each transducer depends on the distance between transducer and the reflecting surface and the angle of incidence. As a major depression or rise (examples are large ocean waves and sand dunes) passes beneath an HSSS the range will change and vary the level of the received signal. Since this variation would be low frequency, relative to other amplitude variations, a high pass filter will effectively filter out these variations and render the correlation speed sensor indication independent of the frequency of passing large waves (celerity) or sand dunes. It appears from a cursory examination of the problem, that a band-pass filter could be used to refer the indicated velocity of a correlation speed sensor to nearly any size group of scattering centers on the traversed surface; the lower limitation being small areas roughly the same dimension as the wavelength of the carrier beam. Thus, a correlation radar speed sensor might be designed to discriminate against all waves present on the ocean surface. Since correlation sensors have not yet been developed significantly for water speed measurement, data are not available on system errors, but it is estimated that the errors will not be higher than those expected from doppler sensors.

Doppler Spreading

Any frequency deviation from the carrier frequency of the return signal will tend to cause decorrelation between the two signals required to resolve one component of speed.[46] Doppler frequency spread occurring for nonzero incident angles is one example. This would arise from beam spread and list, trim, roll, or pitch of the ship. It is difficult to state quantitatively how much the correlation function will be attenuated. It is mentioned here as a reminder to potential developers of correlation speed sensors that doppler spreading could cause a decrease in sensitivity.

Sensor Location

Important factors to consider in selecting the mounting location are:

- Incident angle should be small to minimize ship attitude errors.

- Interference of ship structure with sensor beams should be minimized.

- Influence of ship presence and motion on the water surface should be minimized if maximum accuracy is to be achieved.

- Effects of roll, pitch, and yaw rates should be minimized.

- Sensor should be protected from damage.

For the bow and side installation, specular reflection may be encountered during calm sea states. For the stern installation with scattering off the wake, specular reflection will never be encountered, but the speed indicated over the sea will contain an error due to wake velocity similar to that encountered by doppler radar. The possibility of plenum installation needs to be investigated.

CHOICE OF CARRIER BEAM

Correlation speed sensors may use radar, lasers, conventional optical sources or acoustic devices to generate the carrier beam. Each method possesses unique characteristics.

An acoustic correlation speed sensor has more limitations than an EM type because of the strong interaction of the acoustic beam with the atmosphere. Local variations in wind, temperature, and humidity will cause variations in the speed of sound as it transmits through the atmosphere, causing time spreading in the return of signals which will result in decorrelation and a loss in velocity resolution. Also, the scattering from atmospheric nonuniformities arising from turbulence, temperature, and humidity will cause amplitude variations which could have a stronger correlation than the scattered signals from the traversed surface, resulting in a speed indication relative to the atmosphere rather tha to the traversed surface. The primary advantages of the acoustic beam are the stronger reflection from the traversed surface and the self-cleaning potential of the transducers in a water and mud spray environment. The transducer involved should also be cheaper to manufacture than EM transducers.

An optical correlation speed sensor would have the limitation of being unable to penetrate water and mud sprays which are sometimes generated by an ACC. Advantages of an optical system result from the narrow beam width and coherent frequency generated by a laser. Narrow beam width and coherence will minimize doppler spreading and result in a sharper correlation function which increases resolution and accuracy.

Use of radar frequencies in designing a correlation speed sensor offers the advantage over the use of optical frequencies of being able to better penetrate fog, water drops, and mud spray.

Developed Instruments Availability

By comparison with doppler speed sensors, correlation speed sensors remain undeveloped. The few correlation speed sensors developed to date include solid and fluid flowmeters,[37-42] one system developed by General Electric (GE)[41] for measuring aircraft speed, and a correlating speed sensor for river craft developed in the USSR.[38]

The flowmeter developed by Beck has been applied to the measurement of granular and powdered solids transported in pneumatic conveyors using capacitance transducers. Additional implemented and planned applications by Beck include flow measurement of:[39]

- Nonconducting liquid slurries using capacitance transducers.
- Conducting liquid slurries using conductivity transducers.
- Liquids and gases using ultrasonic transducers to detect density changes caused by turbulence.
- Liquids using thermocouples to detect small changes in temperature.
- Blood using ultrasonic transducers on the surface of the skin.

Accuracy is not reported. However, direct communication with Dr. Beck indicates that 1% accuracy is likely for most installations.

The correlation speed sensor developed by BISRA has been applied to measuring rolling-mill speeds. Conventional light sources (automotive light bulbs) were used as the beam generator during laboratory tests of this instrument. Accuracy achieved was 0.1% at a metal moving rate of 3.6 m/s.[38] The correlation system developed by GE for aircraft speed measurement used microwave beams at approximately 0° incidence. The system was flown in a small aircraft for testing purposes and demonstrated operability. Reported error did not exceed 1.0% using rigid antennas.[38] The correlation speed measuring sensor for river craft developed in the USSR uses acoustic beams reflected off the river bottom. Accuracy is not reported.

General Characteristics

The general characteristics of the hardware associated with correlation speed sensors for HSSS will be similar to those for doppler speed sensors. Physical dimensions and weight will be about the same. Power transmitted may be less because of the higher reflectivity at incident angles of 0°.

Inertial Systems

Inertial systems were considered as a speed sensor on HSSS's because they are passive and measure land referenced speed. A passive sensor is desirable for minimizing ship detection. Land referenced speed is better for navigation purposes in most cases. For example, on an AALC land referenced speed is desirable for input to the true motion radar.

However, there are some cases where it is desirable to know water speed. A land referenced speed sensor does not solve all the speed sensor problems.

Most inertial systems are primarily complete navigation systems as opposed to more simple speed sensors like the radar and correlation type. Inertial navigation systems display a multitude of outputs including:

- Position (latitude and longitude).
- Ground speed (north-south and east-west).
- Heading angle.
- Roll.
- Pitch.

Accelerometers

Nearly all accelerometers are composed of some form of seismic mass restrained by a spring with motion damping. When subjected to an acceleration, the mass moves an amount proportional to the acceleration. Displacement or force can be sensed. Types of pickoff devices include: piezoelectric crystals, potentiometric transducers (the mass is mechanically linked to the wiper arm of a potentiometer), differential transformer, strain gage, force balance, torque balance, null balance, and a vibrating string. Another type of accelerometer is the "pendulous integrating gyroscope accelerometer" which uses an unbalanced gyro rotor.[47] Acceleration produces precession about the output axis due to the mass unbalance.

Gyros

A gyroscope can be thought of as a flywheel rotating at large angular velocity, ω, about an axis. The gyro is supported within gimbals which allow it to rotate about an axis perpendicular to the spin axis. The number of orthogonal axis about which the spin axis can rotate may be selected as one or two, creating one or two degrees of freedom, respectively. The direction of the spin axis remains fixed in inertial space until a torque is exerted on the mechanism. An applied torque, about any axis except the spin axis causes the spin axis to rotate in a direction perpendicular to the spin axis and to the torque axis.

Gyros are referenced to geographic north and vertical. Several types of conventional gyros are briefly described in the following section to illustrate inertial navigation system design.

Rate Gyros

A rate gyroscope is constrained to one-degree-of-freedom, and its displacement about the output axis is proportional to the angular rate about the input axis.[47] It is called a rate gyro because it is used tomeasure angular rates of motion about a selected axis. The rate gyro is used also to introduce artificial damping into a vehicle's control system.

Rate Integrating Gyros

A rate integrating gyroscope is constrained to one-degree-of-freedom, and its displacement about the axis is proportional to the integral of the angular rate input.[47] Rate integrating gyros are used as basic sensors of angular rate and angular position. They have been applied in flight and fire control systems in both aircraft and ships.

Free Gyros

A free gyro is a two-degree-of-freedom gyro whose gimbal displacements about the output axis are a measure of input angular motion from a predetermined reference about corresponding axes; i.e., the roll axis motion is a measure of roll input.[47] Free gyros are short-term devices in which error buildup limits operation to less than 5 minutes.

Vertical Gyros

A vertical gyro is a two-degree-of-freedom instrument whose gimbal displacements about each output axis constitute a measure of angular deviation from the local vertical axis.[47] Vertical gyros are applied where a gravity reference is used to control or stabilize systems such as antenna platform stabilization and gunfire control. A vertical gyro cannot distinguish between gravity and acceleration; thus errors in vertical alignment result when the craft on which it is mounted maneuvers. This error can be compensated for in practical devices but it does represent an important error source.

Directional Gyros

A directional gyro is a two-degree-of-freedom gyro which indicates the motion of a flight vehicle in azimuth measured from a heading reference.[47] The gyro spin axis is usually maintained level and aligned to point to geographic north. If the directional gyro is precessed exactly the amount required to cancel out the earth's rotation, it is called a latitude corrected gyro. If the directional gyro is torqued to always align with north, it is called a north slaved system. A directional gyro which does not maintain north alignment is called a free azimuth gyro. Any inertial navigation system which operates in the vicinity of the north or south pole is usually a free azimuth system. In this area a north slaved system experiences excessive azimuth gyro torquing that results in large errors.[48]

Typical Systems

There are basically two methods of designing inertial navigation systems, the stable platform method and a "strap-down" method. A stable platform is an assembly of gyros mounted within gimbals. The gyro outputs control the gimbals by means of a servo loop. The stable platform maintains the same angular orientation relative to inertial space or some other reference coordinate system. The accelerometers then sense the three components of acceleration and are free from the angular motions of the ship. In a "strap-down" inertial navigation system the components are rigidly attached to the ship. The angular outputs of the gyros are used, via a computer, to compensate the outputs of the accelerometers for roll, pitch, and yaw. In general, the stable platform approach has been used more frequently than the "strap-down" method. More computer capability is required with the "strap-down" method.

It is possible to improve the accuracy of simple displacement gyros (discussed above) by assembling gyroscopic components into a stable platform. Stable platforms provide a mounting for stabilizing accelerometers and provide accurate azimuth, pitch, and roll attitude information. They are essentially a cluster of gyros mounted within gimbals, the gyro outputs controlling the gimbals by means of a servo loop. The accuracy improvement with a stable platform results from the smaller operating range on the gyros, since they are acting only as error transducers in this application. The introduction of stable platforms complicates the inertial navigation system because arrangement of gimbals and platforms produce a variety of platform types, each with unique error characteristics. Drift rates can be produced in the range 0.001° per hour to 0.5° per hour as a function of design and cost.[47]

The three- and the four-gimbal stable platforms are in common use. The advantages of the three-gimbal system are smaller size and weight and lower cost.[47] Disadvantages of the three-gimbal system are lower system accuracy, higher susceptibility to effects of vibration, and limited maneuverability of the craft.[47] The three-gimbal system cannot handle pitch angles exceeding ±85. The four-gimbal system has 360° of freedom about all three axes. A three-gimbal stable platform should probably be adequate in any inertial speed measuring system applied to the HSSS's.

Inertial System Errors

Inertial navigation systems experience errors as follows:[48]

- Accelerometer and gyro inaccuracies.
- Misalignment of the instruments on the platform.
- Initial errors in platform erection and alignment.
- Computational errors.
- Approximations in the mechanization of the system equations.

Accelerometer and gyro inaccuracies include uncompensated drift and scale-factor drift. Uncompensated gyrodrift acts on the inertial system in the cumulative manner so that the resulting position error oscillations tend to grow with time.[49] In general, errors in the output velocity and position from the above sources consist of 84-minute and 24-hour oscillations and a linear buildup. The 84-minute oscillations result from the natural frequency of the vertical loop. The 24-hour oscillations arise from azimuth inaccuracies. As the inertial system rotates with respect to a point on the surface of the earth, azimuth errors oscillate with a 24-hour period. The linear buildup with time arises from the integration processes in computing velocity and position from acceleration.

Schuler-Tuned System

It is convenient when traveling on the earth's surface to employ a vertical reference which constrains the stable platform to the vertical so that horizontal acceleration may be directly sensed. The pendulum is a mechanism by which vertical can be sensed and maintained. However, a pendulum when accelerated indicates a false vertical unless its length is equal to the radius of the earth. Such a pendulum will always indicate the vertical, independent of any accelerations the support point experiences. This pendulum has a period of 84.4 minutes and is called a Schuler pendulum. It is possible to design an accelerometer-integrator-platform system which behaves like a Schuler pendulum.

A Schuler-tuned system may be damped or undamped. When an undamped Schuler-tuned system is perturbed, it will continue to oscillate with with an 84.4-minute period so that the velocity and position outputs will include an 84.4-minute oscillation. Damping is introduced into the system to produce transient decay. However, damping will result in perfect alignment with the vertical only after an excessively long time. The errors previously mentioned also prevent perfect system operation. This means that the 84.4-minute oscillations will always appear in the velocity and position outputs to some degree. The more acceleration experienced by the platform the greater the magnitude of the 84.4-minute oscillations.

For illustrative purposes, an example of an undamped Schuler-tuned feedback loop for sensing one of the two horizontal components of velocity is shown in Figure 11. The accelerometer is mounted, level with the earth's surface, thus sensing one of the horizontal components of acceleration. Since an accelerometer measures acceleration with respect to inertial space, its output must be corrected for Coriolis and Centripetal accelerations which arise from the earth's rotation. These corrections are known functions of latitude, altitude, earth-rate, and ship velocity. The acceleration is integrated into computed velocity, V_C. The amplifier, 1/R, divides the computed velocity by the earth's radius, R, making the loop a Schuler-tuned system. The gyro then senses vertical which is compared with the reference vertical. The gravity feedback arises because any error in vertical will cause the accelerometer to sense a component of the earth's gravitational acceleration. Solution of the differential equations, applying to this loop, show that accelerometer and gyro errors are propagated undamped at the Schuler frequency.[50] The accelerometer and gyro errors are larger when the system is subjected to acceleration, causing the 84.4-minute oscillations to grow larger when the craft maneuvers. In an inertial system of this type velocity can only be obtained by using extremely accurate gyros and accelerometers which are carefully calibrated, aligned, and erected prior to launch of the ship. This is costly, time consuming, and does not provide for system operation through a temporary power failure.

In a pure inertial system the oscillations that tend to buildup in the vertical loop can be damped by using the outputs of the accelerometer or vertical integrator during straight and level flight (no accelerations) of the ship. However, the steady-state errors now depend on the aircraft steady-state velocity, acceleration, and turning rates. Any changes in these dynamic quantities cause the Schuler loops to again undergo transient changes.

Damping, Reset, and Computation Improvements

The errors in pure inertial sensors have been reduced significantly by using other systems or sensors, external to the inertial system, and by improving the processing schemes using digital computers. External velocity damping and position reset use inputs from other sensing systems to improve accuracy of the inertial velocity and position outputs. Predictive filtering uses digital filtering to improve the accuracy of the inertial velocity and position outputs.

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• Both the doppler and correlation radar speed sensors appear capable of meeting the desired 2 km/hr (1 knot) accuracy. An acoustic doppler speed sensor may also be expected to meet the 2 km/hr (1 knot) accuracy. Neither the acoustic doppler nor the correlation radar speed sensor systems are as developed as doppler radar systems which have many aerospace applications.

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- The principal error in a pure undamped inertial navigation system arises from Schuler oscillations caused by ship maneuvering. External velocity signals are used to damp the Schuler oscillations so that a combination external velocity/inertial navigation system has insignificant error due to pitch, roll, and heave of a maneuvering s' ip.

- Inertial sensors have been designed as complete navigation systems or as gyrocompass systems. Inertial sensors have not been designed to function solely as speed sensors.

- Alignment and erection of an inertial navigation system from a moving platform require precise knowledge of velocity and position of that platform.

- Inertial navigation systems are generally much heavier and bulkier than other systems considered.

- Inertial sensors cannot be operated with sufficient accuracy during long mission times without an external velocity input for damping. Thus, if inertial sensors are used on the arctic SEV, the SES, or the hydrofoil, an addition velocity sensor will be required for damping.

Summary Comparison of Inertial, Correlation, and Doppler Speed Sensors

- All three types of speed sensors can be designed to com pensate for ship motion effects (roll, pitch, yaw, and heave).

- Inertial navigation systems are passive; doppler and correlation speed sensors are active (scattered energy must be received from the traversed surface). Thus, an inertial navigation system can be mounted nearly anywhere on the ship while a doppler and correlation speed sensor must be located to have a clear line of sight to a portion of the traversed surface which is undisturbed by the craft.

- The best design choice of speed sensor for the ACC appears to be doppler radar. Acoustic doppler and correlation speed sensors also offer excellent possibilities of successful application on the ACC.

- The best speed sensors for the arctic ACC are doppler radar, acoustic doppler, and correlation radar. The severe operating environment for the arctic ACC will call for development effort for any sensor chosen.

- Where mission times are short a pure inertial system may be a feasible approach.

- Application of a combination external velocitv, inertial navigation system is possible on HSSS's with larger mission times. A waterborne or airborne speed sensor can be used to provide the required external velocity signal.

- Successful application of an inertial navigation system on arctic ACC would require substantial development and additional cost because of the severe environmental conditions.

- The production cost of an inertial navigation system ranges five to 50 times higher than a doppler or correlation speed sensor depending on the quality and complexity of the inertial system.

- Specular reflection occurring over calm, smooth seas will cause operational problems if the wake is not used as the reflecting surface. A quantitive statement about the percentage of operating time cannot be made without more detailed investigation into the correlation speed sensor.

- If the wake is used as the scattering surface, excessive errors will result because of momentum imparted to the water surface by the ship.

- A radar correlation speed sensor appears more feasible than a laser or acoustic correlation speed sensor. Both the laser and acoustic devices will have larger atmosphere-induced errors than radar.

- Radar and laser correlation speed sensors will probably require a cleaning mechanism for the antennas and transmission surfaces because of water and mud buildup.

- A land-water shift will not be apparent in any of the correlation speed sensors.

- Error caused by water surface movement due to wind will frequently exceed 4 km/hr (2 knots) relative to the water immediately beneath the surface over which the HSSS is traversing.

- Errors caused by different reflecting and scattering properties of microwave and optical EM waves and acoustic waves do not vary much compared to the error caused by wind-driven surface movement.

- Correlation speed sensors will probably require less energy per beam than doppler speed sensors because they can operate at normal incident angles.

- There are no correlation speed sensors commercially available which can be immediately applied to HSSS. Development would be based on flow sensor designs and experimental models for aircraft and ship correlation speed sensors.

Inertial Sensors

- Inertial sensors can be considered as replacement navigation systems fof those presently existing on HSSS's or as supplementary navigation systems.

- Inertial navigation systems provide a multitude of outputs compared to doppler and correlation speed sensors. Outputs include: velocity, position, roll, pitch, and heading.

- Pure inertial systems can only be used for short mission times, (4 to 10 hours). Beyond that time errors will become excessive.

- Exact error figures and cost of an inertial navigation system depend on the complexity and precision of the components. Costs can vary by a factor of ten.

- Errors in inertial navigation systems generally contain oscillating components with periods of 84 minutes and 24 hours and a d-c component which continually gets larger with time.

- List, trim, roll, and pitch variations will not cause errors which exceed 2 km/hr (1 knot) if a well-designed Janus configuration is used.

- Specular reflection will cause signal drop-out problems with all doppler speed sensors under calm water conditions unless the wake or other artifically perturbed surface is used. Estimates of drop-out time range from 2% to about 50% of operation time.

- The acoustic doppler speed sensor may have a higher percentage of operation time over calm, smooth water surfaces because a greater percentage of energy is reflected and scattered back.

- If the wake is used as the scattering surface, excessive errors will result because of momentum imparted to the water surface by the HSSS. Detailed calibration can alleviate these errors to some extent.

- Most doppler laser speed sensors will experience problems because of poor transmission through fog, water and mud sprays. (CO_2 (10.6 μm) lasers will work through fog.)

- Doppler laser and doppler radar speed sensors will probably require a cleaning mechanism for the antennas and transmission surfaces because of water and mud coating under adverse conditions. The acoustic doppler speed sensor may be self-cleaning.

- For acoustic and radar doppler speed sensors, the land-water shift errors which may be as high as 3% can be reduced to 0.25% through the use of special design techniques.

- Errors caused by the land-water shift will be much smaller for the laser doppler speed sensor than for the radar and acoustic doppler speed sensors.

- Errors caused by water surface movement due to wind will frequently exceed 4 km/hr (2 knots) (Figure 6) relative to the water immediately beneath the surface over which the HSSS is traversing.

- Errors caused by different scattering properties of microwave and optical EM waves and acoustic waves do not vary much compared to the error caused by wind driven surface movement.

- There are no doppler speed sensors commercially available which can be immediately applied to the HSSS. However, several aircraft doppler radar systems might be modified for this application. The acoustic and laser doppler speed sensors would require a greater cost and longer development time.

- The acoustic doppler speed sensor may result in a small production cost because of the lower cost of transducers.

Correlation Speed Sensors

- Correlation speed sensors are not as developed as doppler speed sensors but apparently could provide similar accuracy.

- Errors caused by list, trim, roll, and pitch will not exceed ±2 km/hr (±1 knot). These errors will be identical (for an incident of 0°) to these occurring for a Janus doppler speed sensor.

AIR SPEED SENSORS FOR CALM CONDITIONS

There are several approaches to making remote speed sensors work better over water under calm wind conditions. But the most viable immediate solution to the calm wind problem is to provide a suitable airspeed sensor as a supplementary sensor for calm wind conditions. Under calm conditions relative wind and water velocity will be identical and a well designed airspeed sensor should meet all the accuracy requirements of high speed surface ships.

CURRENT WORK AND APPLICATIONS

All three speed sensors; doppler, correlation, and inertial continue to be developed for various applications.

DOPPLER RADAR

The Naval Air Development Center, in a cooperative effort with the David W. Taylor Naval Ship R&D Center, has evaluated a four-beam, FM-CW, doppler radar (made by Kearfott Division of the Singer Company) on board the hydrofoil, USS HIGH POINT (PCH-1). Results were encouraging. Additional evaluation is currently being conducted on the JEFF-B Amphibious Assault Landing Craft, an air cushion type ship, at the Naval Coastal Systems Laboratory at Panama City, Florida.

Development of doppler radar is continuing by several private companies in the areas of improved antenna and electronic processing designs. These additional developments should help solve the speed sensor problems on most HSSS's if Navy activities continue to be funded for development and evaluation of doppler radars as improvements are made by private industry.

CORRELATION RADAR

Correlation radars are also being developed because of their potential for overcoming two of the disadvantages of doppler radar systems. Correlation radars are expected to be less susceptible to signal dropout over calm water because the beams can be transmitted straight down and reflected straight. Also, correlation radars should not be subject to the land-water calibration shift exhibited by doppler radars. A development program is currently underway by the Kearfott Division of Singer Company and NADC for application to helicopters. Performance is not yet competative with that of most doppler radar systems. The authors do not yet know of any application of correlation radars on HSSS.

INERTIAL SENSORS

Many companies and government agencies are working on the development of inertial navigation systems but none are known to be specifically aimed at HSSS applications.

CONCLUSIONS

Doppler Speed Sensors

- Trim, roll, and pitch variations will frequently cause errors larger than 2 km/hr (1 knot) if a single beam is used to sense each component of HSSS speed.

The sensor depends on maintaining correlation of the scattering centers across the area examined by the detector. Decorrelation caused by random water movement on the surface will cause loss of resolution as it does in the correlation speed sensor. A calm water surface may not contain sufficient scattering centers to detect a frequency shift similar to the operational problems expected for the doppler and correlation speed sensors. The sensitivity of the system to list, trim, roll, and pitch, would be the same as for the correlation speed sensor.

In general, this optical speed sensor using spatial filtering has all the advantages and disadvantages of the optical correlation speed sensor. Thus, it appears more reasonable to develop a system for the HSSS using microwave frequencies rather than optical frequencies because of the better transmission through water and mud sprays.

INFRARED TEMPERATURE CORRELATION SPEED SRNSOR

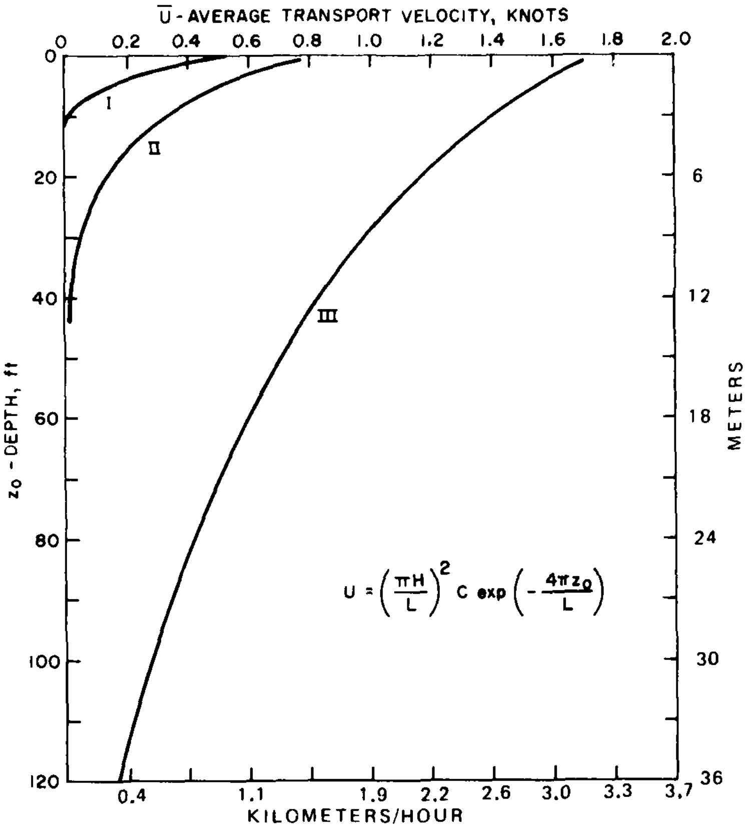
Both the correlation and doppler speed sensors, discussed in this report, have operational limitations over calm, smooth water surfaces. An operational improvement would be realized if a system could be designed to be sensitive to a parameter which has a longer diffusion rate than do waves. One potentially useful parameter is temperature. When the surface wind over the ocean stops, waves gradually decrease in magnitude as their energy dissipates. Since the wave energy dis sipates in the form of heat, it is reasonable to assume that local nonuniformities in temperature may exist longer on the ocean surface than do waves. The temperature nonuniformities could be detected with infrared detectors and processed with circuitry described in the correlation speed sensor section. Performance properties would be the same as for the correlation speed sensors except that better operation could be expected over calm, smooth water surfaces. The first steps in the development of an infrared correlation speed sensor would be to obtain data on the diffusion time of temperature nonuniformities, relative to surface waves, and the sensitivity of infrared detectors relative to optical detectors.

ACOUSTIC DISTURBANCE DOPPLER OR CORRELATION SPEED SENSOR

As mentioned previously, both the correlation and doppler speed sensors have operational limitations over calm, smooth water surfaces because of specular reflection. This problem might be alleviated by using a high intensity, acoustic beam to sufficiently disturb the water surface to cause scattering.

WATERBORNE SPEED SENSORS FOR THE HYDROFOIL AND THE SES

To mount any waterborne speed sensor on the ACC would be difficult because there is no rigid mechanical structure which penetrates the water. Such a design might involve a retractable streamlined sensor or a towed sensor. However, the hydrofoil and the SES do have a rigid mechanical structure which penetrates the water surface and can be used as a mounting surface for a waterborne sensor. The hydrofoil has successfully used a flush-mounted EM log for years. The SES also can use an EM log flush-mounted on one of the sidewalls in a forward location where no cavitation occurs. The EM log system has the advantage of being a proven system. The doppler sonar log presently being tested by the Navy is another possible waterborne sensor for the SES and hydrofoil.



CURVE	WAVE HEIGHT H, M(FT)	WAVE LENGTH L, M(FT)	WAVE CELERITY C, M/S(FT/S)
I	1.2(4)	18(59)	5.3(17.4)
II	3(10)	44(144)	8(27)
III	15(50)	214(700)	18(60)

Figure 12. Water Transport Velocities Due to Waves as Function of Depth

SPATIAL FILTER OPTICAL SURFACE VELOCITY SENSOR

An optical method for measuring surface velocity has been developed by Marconi Company Ltd. of Great Britain.[56] The principle of operation is similar to that of the correlation speed sensor discussed previously. Basically, the speed sensor consits of a grating mounted in front of a photo detector which spatially filters light from the uniformly illuminated surface over which the vehicle is traveling. Scattering centers on the traversed surface move through shaded areas of the grating causing a modulation of the light intensity at a frequency governed by the spacing between the cross members of the grating and the vehicle velocity. This frequency can be detected with electronics similar to those used in a doppler speed sensor.

- Automatically compensates for roll, pitch, and yaw.
- Speed accuracy is generally ±1 knot or better.

Disadvantages of inertial speed sensors are:

- Systems are relatively expensive.
- Systems are usually physically large.
- Accuracy is usually reduced significantly in polar regions.
- Electronics package is complex.
- For long mission times external sensors are required for damping and position reset.

OTHER METHODS OF SENSING SPEED ON A HSSS

There are several other methods of designing speed sensors for a HSSS which have not been completely developed as yet but which appear to have promise. They could be developed in a short time (1 to 3 years), given sufficient monetary support.

THROUGH-THE-SURFACE DOPPLER LASER

One of the principal problems in the doppler laser speed sensor discussed previously is that speed measured over the water surface contains errors produced by surface movement and particle and wake velocities. In general, the errors due to water movement decrease as depth increases. An alternate doppler laser system might be designed in which green light is transmitted through the water surface and backscattered from the water volume located at some depth below the surface. The sensor could be made sensitive to a scattering from this volume below the water surface by time gating to prevent surface scatter from being received. The velocity signal improvement, as a function of depth, was calculated and plotted in Figure 12,55 which shows that water transport velocity is significantly reduced even at a depth of 3 meters (10 feet). The major advantage of the laser doppler speed sensor, using a scattering signal from a volume of water below the surface, is that the velocity indicated is probably closer to ground referenced speed than with a sensor using surface scatter. The disadvantages and problem areas are:

- A mode switch (manual or automatic) must be included on the equipment for operation over shallow water and land.
- Much greater power (than required for the surface doppler laser) will be required to overcome losses in signal strength due to attenuation and divergence in the water and at the air-water interface. This will also produce a greater frequency spread on the doppler spectrum. Potential eye hazard would limit installation to a "protected" location such as under the deck on HSSS's.

AVAILABILITY

Present inertial systems are used mostly for navigation, although speed sensing is inherent in these systems. Most inertial navigation systems have been designed for space, aircraft, or large ship applications.

The big expensive systems accept inputs from external sensors for reset and damping purposes and use predictive filters to reduce error. Accuracies of these systems varies over a large range as does cost. It is not unreasonable to expect errors of 1.6 kilometer (1 mile) over a 12-hour period for position and 1.8 kilometer/hr (1 knot) for velocity.

Several companies make essentially self-contained inertial navigation systems which have a velocity accuracy of the order of 1.8 kilometer/hr (1 knot) and position error accumulation of about 1 kilometer/hr (0.5 nautical-mile/hr). These systems are best applied where mission times are low or where at least position up date are frequently available.

APPLICATION ON HSSS's

Speed Measurement on AAC with an Inertial System

Use of an inertial system on AAC to measure ground speed would be possible if a suitable alignment procedure could be developed. Short mission times of the AAC (4 hours) would allow use of a pure inertial system as a speed sensor (no external velocity input required).

The major disadvantage of an inertial speed sensor on ACC will be the alignment problem. If the ACC will be launched from a "mother" ship, the inertial system must be aligned on a moving platform, and accurate velocity and position information for alignment would have to be provided before launch.

Speed Measurement on Large HSSS's Using an Inertial System

An inertial navigation system can be used on large HSSS's with permanent structure continuously in the water, not only as a speed sensor but as a complete navigation system. The most accurate system would be a combination inertial system and external speed sensor with provision for position resets using radio, satellite, or celestial observations. This does not solve the speed-sensing problem since the inertial system would be used as a navigation system. The logical choice for a water speed sensor would be a doppler, correlation speed sensor, or an EM log or doppler sonar mounted in one of the sidewalls. Alignment may present some problems because these ships may be subject to rolling and pitching to some degree most of the time. Examples of this type of ship are surface effect ships and hydrofoils.

Inertial Sensors Summary

Advantages of inertial speed sensors are:

- System is passive.
- Speed measured is land referenced.
- A multitude of outputs (position, ground speed, heading, roll, and pitch) are available.

Gas bearing gyros use gas to support the rotor and eliminate wear and consequently drag which reduces the drift.

Cryogenic gyros generally use magnets to suspend the gyro rotor, replacing conventional bearings and thereby reducing the drift characteristics of the gyro.

Electrostatic gyros use an electric field to suspend the gyro rotor and reduce drift. This is one of the most important unconventional gyros and will likely find application on ships.

INITIAL CONDITIONS AND ALIGNMENT

Velocity and position outputs from an inertial navigation system depend on integration of acceleration. Initial values of the integrals (velocity and position) must be known at the start of operation. For a ship at rest on the earth's surface providing initial velocity and position is not difficult. However, if a ship is moving, or is launched from a "mother" ship as in the case of the AALC, an accurate estimate of velocity and position is difficult to achieve.

The inertial navigation system must be aligned in the proper coordinate system. This involves leveling of the stable platform (commonly referred to as erection of the vertical) and azimuth alignment. (Even in a free azimuth system a knowledge of north direction is required). Alignment may be accomplished by using externally provided attitude information or by a self-alignment mode where the inertial instruments sense their misalignments. Generally, the self-alignment procedure takes a longer time for a given accuracy. The erection of the vertical and azimuth alignment is usually done simultaneously.

Self-alignment procedures require the insertion of a damping term into the vertical feedback loops of the system, reducing the transient decay time and increasing the natural frequency above the Schuler tuned value. With self-alignment using a reduced natural period, damping alignment can be completed in less than 30 minutes. Since the alignment errors are usually a continuous function of time and gradually approach zero with time, the inertial systems should remain in the align mode of operation as long as possible for greatest accuracy. The alignment error, as a function of time, varies with the system manufacturer who must be consulted for precise data on alignment time and accuracies involved.

Alignments using externally provided attitude information include plumb bobs, bubble levels, star trackers, radio sextants, and gyrocompasses.

When an inertial navigation system is aligned in a vehicle already in motion with respect to the earth's surface, auxiliary external sources of velocity and position information are required. Time required for alignment is of the order of a few minutes and is generally faster than self-alignment.

small position errors. Designers have found that greatest accuracy can be obtained through the use of both external velocity compensation and position fixes. Sources for resetting position outputs include:[52,53]

- LORAN C
- OMEGA
- Celestial sightings
- Satellites

These position fixes are used to reset the longitude and latitute outputs of the inertial system. The use of an external velocity input for an inertial system has been demonstrated to reduce the position error by 40 to 50 percent.

Predictive Filtering

The development of Kalman filters has also increased the position accuracy of inertial navigation systems.[51-53] Kalman filtering uses a statistical approach to estimate error magnitude from a knowledge of system dynamics and error sources depending on past performance of the combined systems. This technique has been applied to aircraft systems where a combination of doppler radar and inertial navigation systems is used. Videla[51] shows that better than an order of magnitude reduction in error can be achieved using Kalman filtering. Achievable position accuracies are better than 0.4 kilometers (0.2 nautical mile) in a 6-hour reset period.

Gyro Improvements

Manufacturers of inertial systems are constantly working on equipment improvements, particularly in the area of designing lower drift gyros. There have also been improvements in system accuracy by rotating the platform; steady (d-c) errors are converted into oscillating (a-c) errors which are then filtered, thereby reducing drift effects. However, the more important area of improvement is in the area of using new gyro designs to reduce drift.

The various kinds of unconvential gyros are:

- Ring laser gyro.
- Nuclear magnetic resonance gyro.
- Gas bearing.
- Cryogenic gyros.
- Electrostatic gyros.

The ring laser gyro uses light beams transmitted around a closed loop. As the laser system rotates the round trip travel time increases or decreases as a direct function of rotation rate.

The nuclear magnetic resonance gyro utilizes the natural angular momentum possessed by nuclei and optical pumping to align the nuclei.

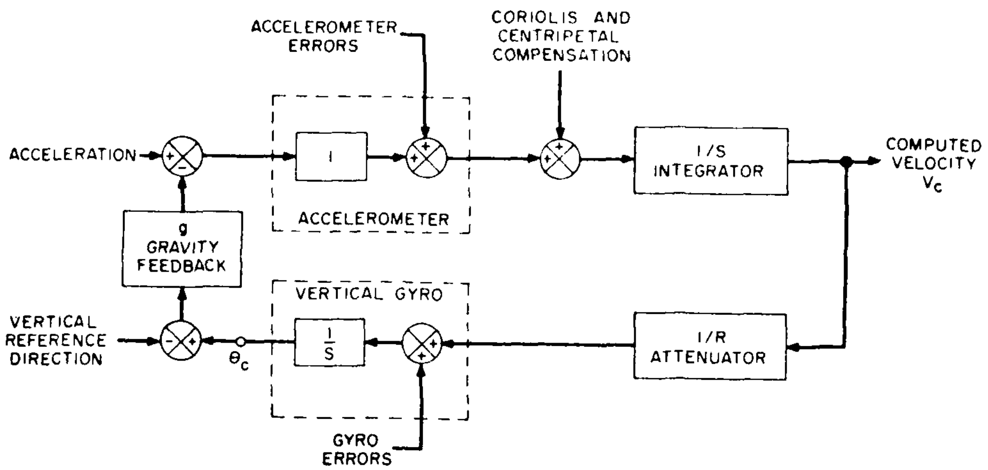


Figure 11. Pure Inertial System for Computing One Horizontal Velocity Component

External Velocity Damping

The introduction of damping into the inertial navigation systems changes the 84.4-minute Schuler oscillations on the velocity and position outputs. The source of the damping signal is an external velocity sensor. Sufficiently precise velocity information, properly injected into the inertial navigation system, can place the system in a state where errors are not excited by ship maneuvers. However, the accuracy of the external velocity information must be near that of the velocity signal produced by the pure inertial system. This implies use of a completely redundant system whose principal function is to provide compensation for the dynamic effects of damping. This procedure is questionable at best in terms of providing a speed sensoring system for HSSS's, but it can be shown that greater position accuracy can be achieved by using two systems than could be achieved by using a single system. For ships using the SINS the usual source of external velocity information is the EM log, for aircraft, the doppler radar.

If an external independent source of velocity information is provided, the two measurements can be compared and their difference used to damp the inertial system. Under this condition the error in the velocity output arising from the accelerometer and gyro errors are damped-out. Using external velocity information also permits the inertial system to be erected in a shorter time with greater accuracy.

There are various methods of connecting an external velocity signal to the inertial system. In general, the result is that the combined system error is reduced (about a factor of two) below that available from either system separately. Bias errors of the external velocity sensor can be compensated, but transient errors in the external velocity sensor can cause large combined system error.[51]

Position Reset

The relation between the position error and the velocity error in an inertial navigation system is not always linear (in the mathematical sense). It is not uncommon to have large velocity errors and

A NEW AUTOPILOT SYSTEM WITH CONDITION ADAPTIVITY

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Mitsubishi Heavy Industries, Ltd.

ABSTRACT

When a ship sails with the yawing of prolonged periodicity, its fuel consumption is said to increase by several percentage or more than when it sails without such yawing. The conventional autopilot systems have the weather adjusting deadband which is manually adjusted by the operator to avoid high frequency disturbances in rough seas. However, the increase in the deadband leads to the decrease in low frequency performance and eventually to the yawing of prolonged periodicity.

Our new autopilot system which employs DDC is aimed to solve the problems inherent in the conventional autopilot systems and offers the following advantages.

1. The high frequency component of gyro signal is filtered by Kalman filter, and the stable rudder control is secured without increasing the deadband.
2. The function to estimate the wave condition from the gyro signal is added.
3. According to the various conditions such as cargo load, ship speed, and estimated wave condition, this system automatically selects the optimum control gain and filter parameters given in advance.

It is evident from the computer simulation that this system is far more capable of minimizing the yawing under various ship conditions as compared with the conventional autopilot systems.

At the present time, this system is actually being used on board a 230 KDWT tanker. The field data to substantiate the above-mentioned advantages are expected in near future.

1. INTRODUCTION

The ship steering has to deal with the problems of safety and economy at the same time. In narrow waters a ship must be steered in such a manner as to ensure the maximum safety, and at the open sea an emphasis must be placed on achieving the maximum economy of ship operation.

The autopilot system now in prevalent ship application is indeed a useful navigational aid from the standpoint of both safety and economy. It is felt, however, that the existing autopilot system has a plenty of room for improvement if still greater safety and economy are to be assured for ocean-going ships. The yawing of prolonged periodicity is often experienced by ships steered by the conventional autopilot system; the weather-adjusting practice of broadening the deadband is principally to blame for such a yawing problem.

Accordingly, the authors devised a new autopilot system which enables a ship to automatically select and follow a course that assures the shortest, hence most economical, run to reach its destination, avoiding at the same time both known and unexpected dangers on the way. By adding an adaptive control function to computerized autopilot function and therein incorporating a specially programmed Kalman filter, the system is made capable of screening out the undesirable high frequency fluctuation from gyro signal to ensure the maximum directional stability for a ship even in a heavy sea.

Simulation tests with the new autopilot system coupled to a hybrid computer serving as ship steering hardware proved the excellent course controllability of the system.

2. CONVENTIONAL AUTOPILOT SYSTEM

Nearly all ocean-going ships are equipped with the autopilot system of the type shown in Fig. 1 to keep them on predetermined courses while underway at sea. For ships whose voyage days are spent mostly out at the open sea, the importance of the autopilot system as a navigational aid cannot be overemphasized since keeping to a desired course is a vital factor in the saving of fuel cost. In fact it is claimed that navigation with the yawing of prolonged periodicity will result in higher fuel cost by several percent as compared with the sailing on a straight course without such a yawing[1)]. Indeed, with the yawing of prolonged periodicity, the ship sailing distance increases and the ship propulsion efficiency decreases due to resistance resulting from centrifugal force generated by the yawing motion. Also, the frequent steering for course correction increases the rudder resistance.

The yawing, which occurs when steered by the conventional autopilot system, may be attributed to the following.

(1) Though what is called "rate rudder" steering, in which the gyro signal is differentiated for rudder control purposes, is adopted by large ships like oil tankers to ensure the directional stability considering the effects of their geometrical hull characteristics such as fullness of hull form, etc., the resolution of the gyro signal is approximately 1/6 deg. at the best with the step-motor type gyro repeaters and it is therefore difficult to obtain clear, distinctive differentiated signal in

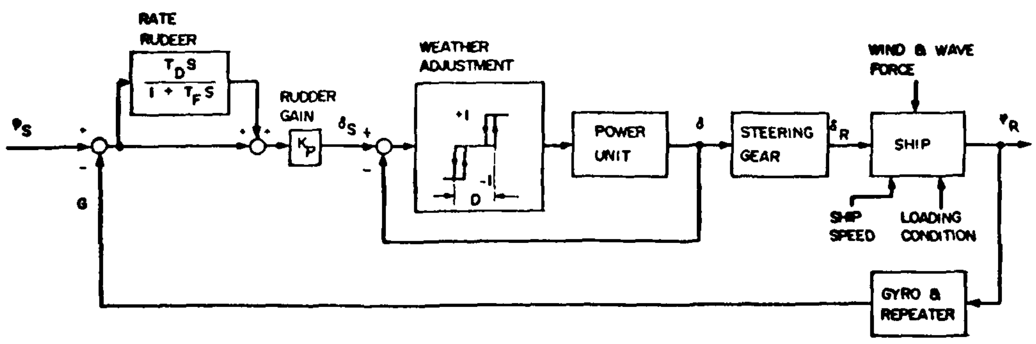

Fig.1 Block Diagram of Conventional APS

case where the course error is minor. Consequently the "rate rudder" steering fails to exert the desired damping effect, giving rise to the yawing.

(2) When navigating in a rough sea, the gyro signal becomes loaded with high-frequency fluctuation from the effects of raging waters. Since steering the ship faithfully to such a gyro signal only increases the rudder resistance and even shortens the life of the steering gear, the conventional autopilot system copes with the situation by broadening its "weather-adjusting" deadband. The deadband, however, adversely affects the course control function of the autopilot system, thereby providing a cause for the yawing of prolonged periodicity.

(3) Ship dynamic characteristics to be dealt with by the autopilot system vary greatly with loading conditions and running speeds, and there are varying degrees of disturbances to the autopilot system due to changing sea conditions. For successful course control under such circumstances, it is highly desirable that optimum control parameters be selected and used from time to time according to the changing ship and sea conditions. With the conventional autopilot system, however, the control parameters are manually selected and hence often turn out to be inadequate to prevent the yawing.

It is considered imperative, therefore, that all the above deficiencies of the conventional autopilot system be remedied to assure the directional stability and saving of fuel for ocean-going ships.

3. NEW AUTOPILOT SYSTEM

The new autopilot system Mitsubishi has been considering employs a control computer for direct-digital control function in order to minimize the yawing that results from the use of the conventional autopilot system. Functional features of this system may be summarized as follows.

(1) The Kalman filter employed for this system as part of its software is so devised as to remove from the gyro signal the undesirable high-frequency fluctuation caused by sea waves so that the stabilized, consistent steering can be accomplished without broadening the weather-adjusting deadband as with the conventional autopilot system.

(2) The system diagnoses the wave conditions from the gyro signal received.

(3) The system automatically selects from its memory the optimum control parameters, such as control gain, filter, etc., in response to loading-condition, ship-speed, and wave-condition data input.

3.1 Constitution and Functions

Figure 2 shows a block diagram of this system. As is clear from the diagram, the system includes the closed-loop-autopilot circuit and adaptive-function circuit which are indicated divided by a dotted line. Vertically divided by a dot-dash line, the left-hand side represents the control computer and the right-hand side the conventional hardware.

The closed-loop-autopilot circuit performs the following functions.

1) The rudder angle δ transmitted by way of the analog-digital converter (A/D) and sampled as digital signal and also the gyro signal φ_G transmitted by way of the gyro repeater and shaft encoder (S.E.) are fed into the Kalman filter. The Kalman filter

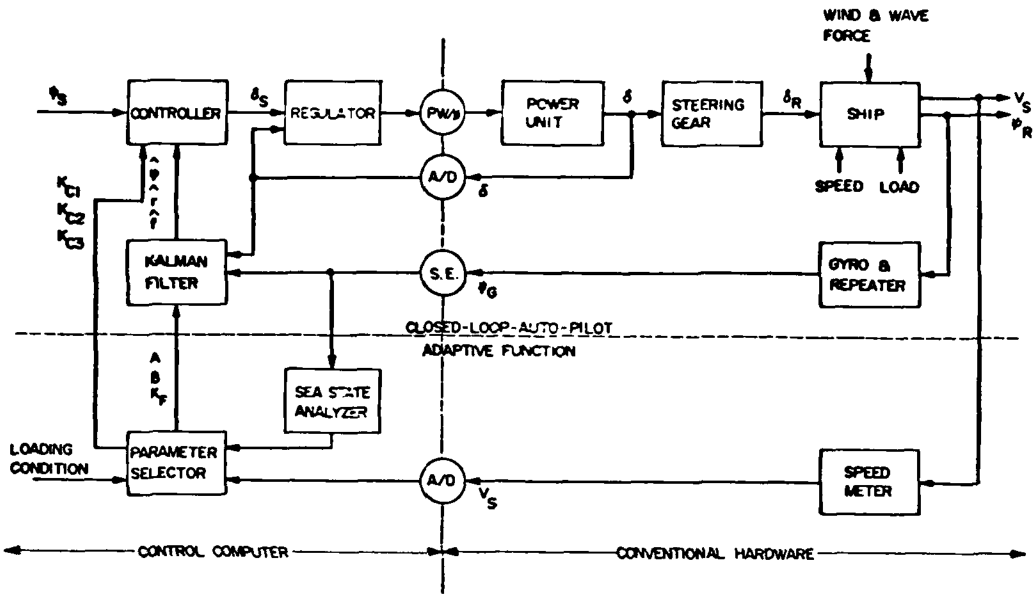

Fig.2 Block Diagram of New Auto Pilot System

screens out undesirable high-frequency fluctuation from the gyro signal and turns out the estimated $\hat{\varphi}$, $\hat{r}$ and $\hat{f}$, which correspond to the heading φ, rate of turn r and low-frequency disturbance f, respectively.

2) The controller receives from the Kalman filter the estimated $\hat{\varphi}$, $\hat{r}$ and $\hat{f}$, plus desired heading φ_s, and performs the following calculation, to turn out the rudder-angle command δ_s.

$$\delta_s = K_{c1}(\varphi_s - \hat{\varphi}) - K_{c2}\hat{r} - K_{c3}\hat{f} \qquad (3.1)$$

The above equation corresponds to PID control, the first, second and third terms on the right-hand side respectively being equivalent to proportional, derivative, and integral control functions; K_{c1}, K_{c2}, and K_{c3} are their control gains and each value is automatically changed by dint of the system adaptive function.

3) The regulator converts the $\delta_s - \delta$ difference into solenoid valve operating time as pulse-width output (PW/Ø).

4) The power unit integrates the computer pulse-width output in the same manner as it does in the conventional autopilot system and produces the rudder angle δ as an output.

5) The steering gear operates exactly in the same manner as it does in the conventional autopilot system, amplifying by hydraulic servo mechanism the rudder angle δ produced by the power unit and actually moving the rudder to that angle.

The adaptive-function circuit performs the following functions.

1) The sea-state analyzer stores 1200 samples of gyro signal φ_G collected at a rate of one piece every second over a period of 20 minutes, determines from the collected samples the periods and amplitudes of the gyro signal, and consequently feeds the wave-condition index into the parameter selector.

2) The ship-speed meter, which is the conventional doppler sonar or electro-magnetic log, measures the ship-speed and feeds the ship-speed signal V_S into the computer via the A/D converter.
3) The parameter selector receives the wave-condition index from the sea-state analyzer, ship-speed signal V_S from the ship-speed meter, and manually fed loading-condition signal, and processes all these data to select optimum control parameters from the computer memory. The optimum control parameters selected are then given to the controller and Kalman filter in the closed-loop autopilot circuit.

3.2 Kalman Filter

The Kalman filter receives gyro signal φ_G and rudder angle δ and, as stated earlier, removes from the former the undesirable high-frequency fluctuation, to feed the estimated $\hat{\varphi}$, $\hat{r}$ and $\hat{f}$ into the controller.

The Kalman filter technique is a filter design method based on the time domain approach and, once given a mathematical model of measured signal, formulates an optimum filter for that mathematical model.

The Kalman filter employed for this system is designed to deal with the following mathematical models.

1) Measurement Equation: Assuming that the gyro signal φ_G is composed of heading φ, high-frequency fluctuation w_1, and measurement noise $\lambda_0 \xi_0$, the following measurement equation is adopted.

$$\varphi_G = \varphi + w_1 + \lambda_0 \xi_0 \tag{3.2}$$

In the above equation, λ_0 is coefficient, ξ_0 is normal white Gaussian noise, and φ and w_1 respectively are made up of low-frequency and high-frequency models as discussed in 2) and 3) below.

2) Low-Frequency Model: The heading φ is given by the integral of the rate of turn r. The rate of turn r can be given as the first-order lag of rudder angle δ, low-frequency disturbance f, and white Gaussian noise $\lambda_2 \xi_2$ which is considered as disturbance.

$$\varphi = \frac{1}{S} \cdot r \tag{3.3}$$

$$r = \frac{K}{1+TS} \left\{ \delta + f + \frac{T}{K} \cdot \lambda_2 \xi_2 \right\} \tag{3.4}$$

$$f = \frac{1}{S} \cdot \lambda_3 \xi_3 \tag{3.5}$$

In the above equations, S represents the Laplace operator and K and T the ship dynamic characteristics coefficients. The ship dynamic characteristics coefficients vary with loading conditions and ship speeds. The low-frequency disturbance f, representing the turning moment which is caused by wind, etc., and which approximates direct-current component, is used converted into rudder angle. $\lambda_2 \xi_2$ represents all disturbances except for f and other disturbances responsible for undesirable high-frequency fluctuation. ξ_2 and ξ_3 denote white normal Gaussian noise, and λ_2 and λ_3 are coefficients.

3) High-Frequency Model: The high-frequency model assumes, as its characteristics, that the power spectrum of high-frequency fluctuation w_1 exists in the range of 5 - 20 sec periods. Accordingly, as the model of sea wave w_h the following equation is adopted[7].

$$w_h = \frac{b_0 S}{S^2 + b_2 S + b_1} \xi_1 \tag{3.6}$$

In the above equation, ξ_1 represents normal white Gaussian noise and b_0 , b_1 , and b_2 are coefficients. The sea wave w_h acts on the ship hull and transforms into undesirable high-frequency fluctuation w_1 of gyro signal, the relation of which is given by the following equation.

$$w_1 = \frac{a_0 S^2}{(S+a_1)(S+a_2)} w_h \tag{3.7}$$

In the above equation, a_0 , a_1 , and a_2 are coefficients. From Eqs. (3.6) and (3.7), state variables w_1, w_2, w_3, and w_4 of high-frequency model are expressed as follows.

$$w_1 = \frac{1}{S+a_1} \cdot S w_2 \tag{3.8}$$

$$w_2 = \frac{1}{S+a_2} \cdot \frac{S^2}{S^2+b_2 S+b_1} (-\lambda_1) \xi_1 \tag{3.9}$$

$$w_3 = \frac{b_1}{S^2+b_2 S+b_1} \cdot \lambda_1 \xi_1 \tag{3.10}$$

$$w_4 = \frac{b_2 S}{S^2+b_2 S+b_1} \cdot \lambda_1 \xi_1 \tag{3.11}$$

In the above equation, $\lambda_1 = -a_0 b_0$. Let $\lambda_1 = 0$, then $w_1 = 0$, this representing the condition without high-frequency fluctuation. The high-frequency fluctuation w_1 of large amplitude can be expressed by increasing λ_1.

From Eqs. (3.2)-(3.5) and (3.8)-(3.11), the mathematical models which the Kalman filter is designed to deal with are expressed by the following measurement and state equations.

$$\underset{\substack{\| \\ y}}{\varphi_G} = \underset{\substack{\| \\ M}}{(1,0,0,1,0,0,0)} \cdot \underset{\substack{\| \\ X}}{(\varphi, r, f, w_1, w_2, w_3, w_4)^t} + \lambda_0 \xi_0 \tag{3.12}$$

$$\frac{d}{dt}\begin{pmatrix}\varphi\\ r\\ f\\ w_1\\ w_2\\ w_3\\ w_4\end{pmatrix}=\begin{pmatrix}0 & 1 & 0 & 0 & 0 & 0 & 0\\ 0 & -\frac{1}{T} & \frac{K}{T} & 0 & 0 & 0 & 0\\ 0 & 0 & 0 & 0 & 0 & 0 & 0\\ 0 & 0 & 0 & -a_1 & -a_2 & 1 & 1\\ 0 & 0 & 0 & 0 & -a_2 & 1 & 1\\ 0 & 0 & 0 & 0 & 0 & 0 & \frac{b_1}{b_2}\\ 0 & 0 & 0 & 0 & 0 & -b_2 & b_2\end{pmatrix}\cdot\begin{pmatrix}\varphi\\ r\\ f\\ w_1\\ w_2\\ w_3\\ w_4\end{pmatrix}+\begin{pmatrix}0\\ \frac{K}{T}\\ 0\\ 0\\ 0\\ 0\\ 0\end{pmatrix}\cdot(\delta)+\begin{pmatrix}0 & 0 & 0\\ 0 & \lambda_2 & 0\\ 0 & 0 & \lambda_3\\ -\lambda_1 & 0 & 0\\ -\lambda_1 & 0 & 0\\ 0 & 0 & 0\\ b_2\lambda_1 & 0 & 0\end{pmatrix}\cdot\begin{pmatrix}\xi_1\\ \xi_2\\ \xi_3\end{pmatrix} \tag{3.13}$$

$$\mathbb{X}\qquad\qquad \mathbb{A}\qquad\qquad \mathbb{X}\qquad \mathbb{B}\quad u\qquad \mathbb{C}\qquad \xi$$

In Eq. (3.12), t denotes transposition. Using vector symbols, Eqs. (3.12) and (3.13) are rewritten as follows.

$$y = \mathbb{M}\mathbb{X}+\lambda_0\xi_0 \tag{3.14}$$

$$\frac{d}{dt}\mathbb{X}=\mathbb{A}\mathbb{X}+\mathbb{B}u+\mathbb{C}\xi \tag{3.15}$$

For these mathematical models, the steady-state Kalman filtering formulates a filter which keeps to the minimum the error variance of estimated value $\hat{\mathbb{X}}$ of the state variable $\mathbb{X}$. The formulated filter is expressed by the following equation.

$$\frac{d}{dt}\hat{\mathbb{X}}=\mathbb{A}\hat{\mathbb{X}}+\mathbb{B}u+\mathbb{K}_F(y-\mathbb{M}\hat{\mathbb{X}}) \tag{3.16}$$

$\mathbb{K}_F$ is what is called "filter gain" and can be obtained through steady-state solution of the covariance equation (see Appendix). Eq. (3.16) is the Kalman filter used in the closed-loop-autopilot circuit and, using u and y, or δ and φ_G, as input, turns out $\hat{\varphi}$, $\hat{r}$, and $\hat{f}$ which are components of $\hat{\mathbb{X}}$. Each of $\mathbb{A}$, $\mathbb{B}$, and $\mathbb{K}_F$ values is automatically changed by this system's adaptive function. Though Eq. (3.16) is indicated as a continuous time process, actual arithmetical operation is performed in a discrete time process.

3.3 Adaptive Function

The selection of optimum parameters according to changing ship dynamic and sea conditions is a vital factor in accomplishing successful course control. The parameter selector in the system adaptive-function circuit adjusts the control gains K_{C1}, K_{C2}, and K_{C3} for the controller and $\mathbb{A}$, $\mathbb{B}$, and filter gain $\mathbb{K}_F$ for the Kalman filter in the following manner.

1) Loading Conditions: When a ship navigates under ballast, its dynamic course-keeping characteristics remain relatively stabilized because the ship assumes the trim-by-stern attitude which tends to provide a larger ratio of rudder area to the hull in water. Under full load, however, the ship's autopilot system must execute adequate derivative action to ensure the stabilized course keeping since the ship tends to lose directional stability. For this reason, the greater K_{C2} value is employed when

under full load than under ballast.

2) Ship Speed: Since the ship steerability improves with the increase in ship-speed V_s, control gains K_{c1} and K_{c2} that are employed in the low-speed sailing must be reduced as the ship picks up speed. Otherwise, self-excited yawing will results.

3) Sea State: When gyro signal becomes loaded with high-frequency fluctuation from the effects of sea waves, the coefficient λ_1 in the high-frequency model equation (3.13) is increased according to the wave-condition index from the sea-state analyzer for solution of the covariance equation in order to obtain the filter gain $\mathbb{K}_F$. The filter gain $\mathbb{K}_F$ is then fed into the Kalman filter in the closed-loop-autopilot circuit, so that the Kalman filter formulates a filter which screens out the high-frequency fluctuation from the gyro signal. Figure 3 shows examples of filter gain $\mathbb{K}_F$ for the coefficient λ_1 corresponding to sea state, in which K_{F1} is the filter gain for the estimated heading $\hat{\varphi}$ and K_{F4} the filter gain for the estimated high-frequency fluctuation ω_1. With the increase in λ_1, K_{F1} becomes small and K_{F4} large, indicating that the high-frequency fluctuation of the gyro signal φ_G is absorbed by $\hat{\omega}_1$, and $\hat{\varphi}$ reflects the low-frequency component.

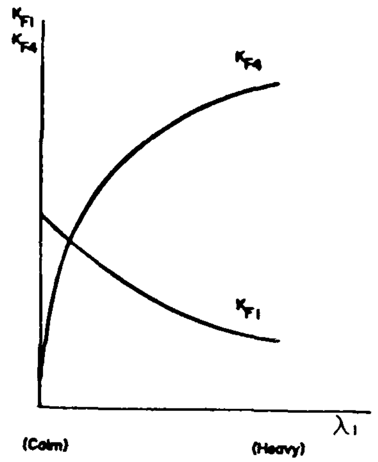

Fig. 3 Filter Gains vs. λ_1 (Sea State)

4. SIMULATION

Adjustment of optimum control parameters to be used for varying ship dynamic and sea conditions was made by means of simulation test. In this simulation test the new autopilot unit was coupled to a simulator made of a hybrid computer which was arranged to simulate necessary characteristics of power unit, ship motions, disturbances, and gyro/repeater unit as shown in Fig. 4.

As examples of the simulation test results, responses of actual rudder angle δ_R, actual heading ψ_R, and gyro signal ψ_G are indicated in Figs. 5 - 9. The ship for which the simulation was carried out was a 237,000 deadweight tons tanker navigating at 10.2 knot under full load. The results with the new autopilot system and those with the conventional autopilot system shown in Fig. 1 are indicated as obtained for calm sea and heavy sea conditions, for comparison.

As regards the responses in calm sea conditions, the use of the conventional autopilot system generates a self-excited yawing of the period of about 150 seconds (Fig. 5) while the use of the new autopilot system produces no such yawing (Fig. 6). Regarding the responses in heavy sea conditions, the conventional autopilot system causes the ship rudder to move very frequently in response to high-frequency fluctuation unless the deadband is broadened for weather adjustment (Fig. 7) and yet the broadening of the deadband leads to the generation of yawing of prolonged periodicity (Fig. 8). The responses of the new autopilot system are characterized by stabilized, consistent steering, with the optimum control parameters being automatically modified to cancel any high-frequency fluctuation that appears and thereby successfully avoiding the yawing of prolonged periodicity (Fig. 9).

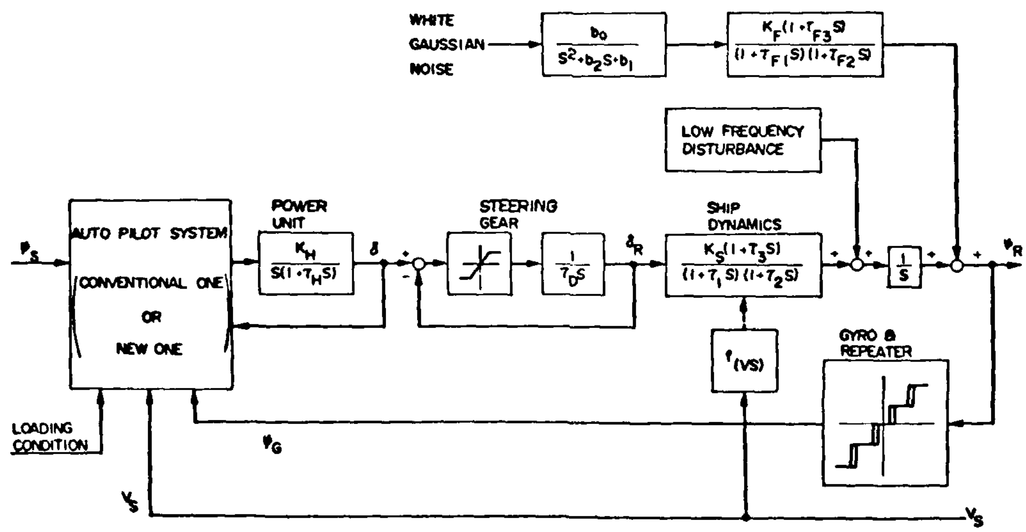

Fig. 4 Block Diagram of Simulation Model

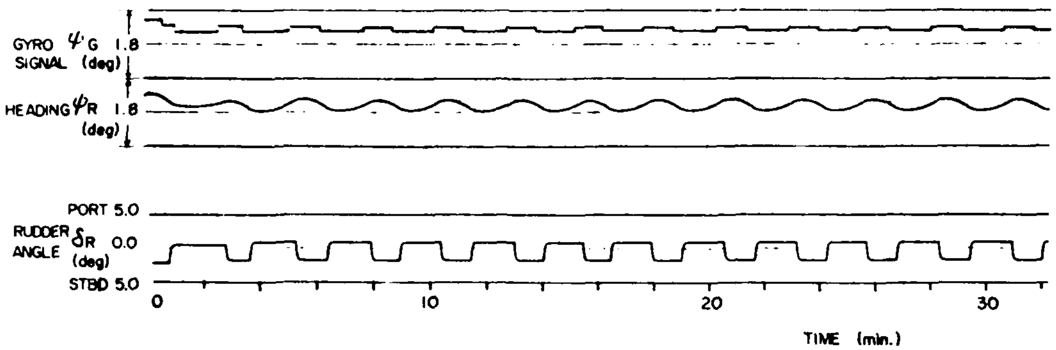

Fig. 5 Conventional APS at Calm Sea

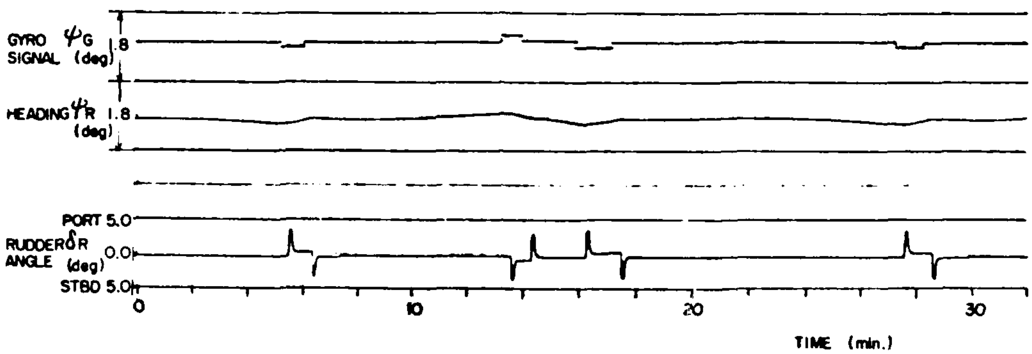

Fig. 6 New Auto PILOT at Calm Sea

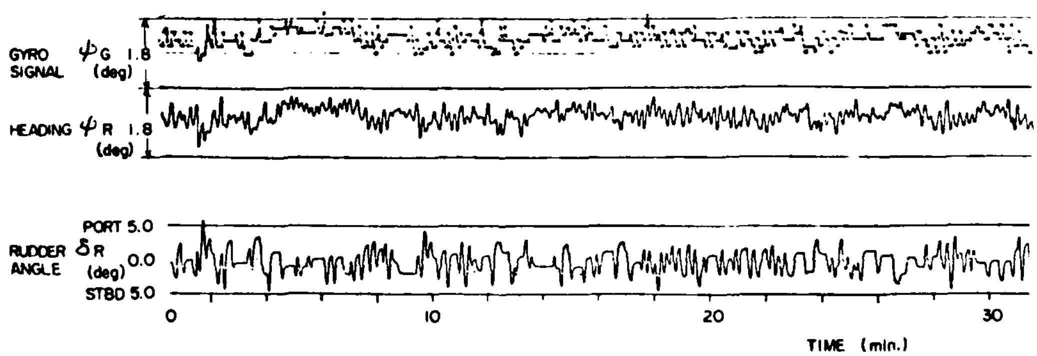

Fig. 7 Conventional APS without Weather Adjust at Heavy Sea

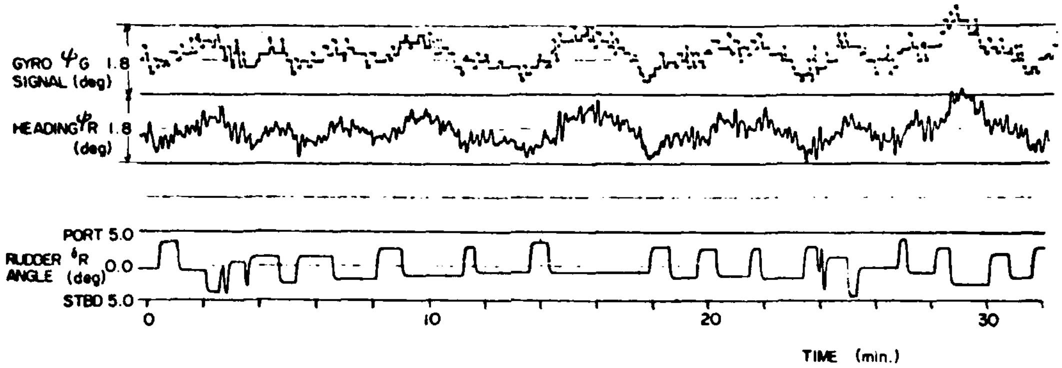

Fig. 8 Conventional APS with Weather Adjust at Heavy Sea

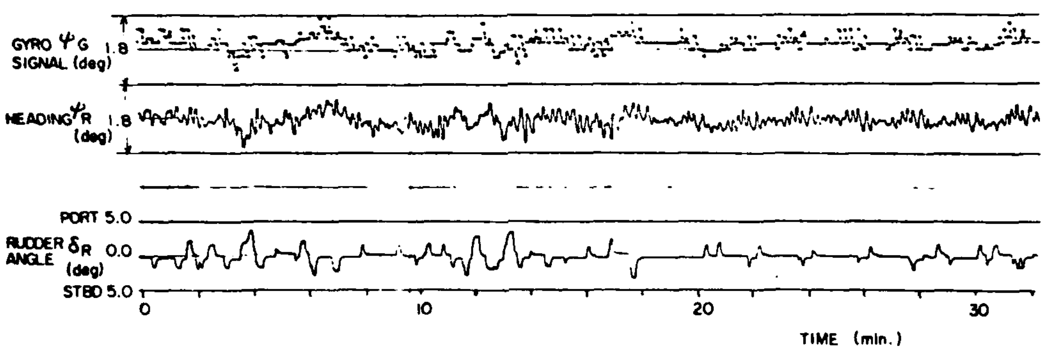

Fig. 9 New Auto PILOT at Heavy Sea

5. CONCLUSION

The new autopilot system is designed to offer an improved low-frequency control characteristics in a heavy sea and ability to automatically select optimum control parameters according to changing ship dynamic and sea conditions, in order to assure the higher reliability for ship course control than with the conventional autopilot system. It is believed that the use of the new autopilot system will lead to the saving in fuel cost.

The progress in digital devices has made it possible to employ advanced data processing technique like the control by adaptivity with this system. However, there remain much to be improved about the this system in specific regards to functions of its closed-loop-autopilot-circuit elements, such as ability to detect the rate of turn, accuracy of steering-gear operation, etc. Furthe. engineering efforts will therefore be made in these respects.

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APPENDIX

Discrete Square Root Filtering

Though discussed in this paper as a continuous time process, it is as a discrete time process suitable for computer processing that the Kalman filtering is employed for the new autopilot system. The algorithm used for determining the filter gain $\mathbb{K}_F$ (covariance equation) is a square root filtering algorithm which is free of divergence in iteration calculation. Eqs. (3.14) and (3.15), when expressed as discrete time processes, are written as follows.

$$\mathbb{X}(k+1)=\Phi\mathbb{X}(k)+\mathbb{D}U(k)+\mathbb{G}\,\xi(k) \qquad (A.1)$$

$$y(k)=\mathbb{M}\mathbb{X}(k)+\lambda_0\xi_0(k) \qquad (A.2)$$

For each of Eqs. (A.1) and (A.2), the Kalman filter is expressed as follows.

$$\hat{\mathbb{X}}(k|k-1)=\Phi\hat{\mathbb{X}}(k-1|k-1)+\mathbb{D}U(k-1) \qquad (A.3)$$

$$\hat{\mathbb{X}}(k|k)=\hat{\mathbb{X}}(k|k-1)+\mathbb{K}_F(y(k)-\mathbb{M}\hat{\mathbb{X}}(k|k-1)) \qquad (A.4)$$

In Eq. (A.4), $\mathbb{K}_F$ is a converged value of $\mathbb{K}_F(k)$ obtained by the following iteration calculation (square root filtering).

$$\Pi_s=\Phi\Pi(k-1|k-1) \qquad (A.5)$$

$$\Pi(k|k-1)=F(\Pi_s,G) \qquad (A.6)$$

$$r=(\mathbb{M}\Pi(k|k-1))(\mathbb{M}\Pi(k|k-1))^t+\lambda_0^2 \qquad (A.7)$$

$$\mathbb{K}_F(k)=\Pi(k|k-1)\Pi(k|k-1)^t\mathbb{M}^t/r \qquad (A.8)$$

$$1/\alpha=1+\sqrt{\lambda_0/r} \qquad (A.9)$$

$$\Pi(k|k)=\Pi(k|k-1)-\alpha\mathbb{K}_F(k)\mathbb{M}\Pi(k|k-1) \qquad (A.10)$$

In Eq. (A.10), $\Pi(k/k)$ represents the square root matrix of covariance matrix of an error of $\hat{x}(k/k)$. $F(\Pi_s, G)$ in Eq. (A.6) represents the Householder transformation[4] and can be rewritten as follows.

$$\Pi(k|k-1)\,\Pi(k|k-1)^t = (\Pi_s : G)\cdot(\Pi_s : G)^t \qquad \text{(A.11)}$$

SHIP RESPONSE CONTROL DURING HEAVY WEATHER OPERATIONS

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ABSTRACT

The paper reviews the environmental and operational problems encountered in heavy weather, the applicable seakeeping theory and developing technology. On board response assessment and the corresponding prediction are discussed as they may be integrated into a ship response control system. Work underway on the development of satisfactory hardware and software for bringing this technology to bear is discussed. The requirements for successful application are proposed including special man/machine interaction, problem solutions and training requirements. Areas of current investigation are identified and system effectiveness is discussed.

INTRODUCTION

With diminishing supplies of natural resources ashore and increasing needs for ocean transport between countries seas apart, the ability to work or move over and through the world's oceans has taken on an ever larger role in today's maritime technology. To effectively use the sea, it is necessary that marine systems be designed, built and operated in a manner that takes into account the true character of the sea and the response of structures therein.

Ships operating on the high seas are subject to forces of several types, most of which are dynamic and cause ship responses of concern. Externally, forces experienced are primarily aero or hydrodynamic in nature, while internal forces experienced are mostly structural, mechanical or inertial in character. While the external forces are readily appreciated in a qualitative sense, the internal forces are generally assessed only through indirect processes. Seafarers have, for generations, tried to gain a sensual appreciation of these forces with only modest success while the naval architect has tried mightily to deduce them through logic and imagination. Success is nearly at hand.

All who have lived by the sea understand and appreciate that as the waves are generated by the wind, so the ship pitches, heaves, rolls, etc., with the waves. In the days of sail, not only were these rigid body motions noted but the straining of the ship's structure and outfit were identified as well by the creaking sounds of strained planks and rigging and the rising level of bilge water as strained seams were opened; rudder forces were immediately sensed by the helmsmen. By reducing sail or changing course, the ship's motion and strains were eased to the point where master and crew could feel more secure as the wind and waves roared on.

The use of all welded steel ship hulls, propelling machinery and

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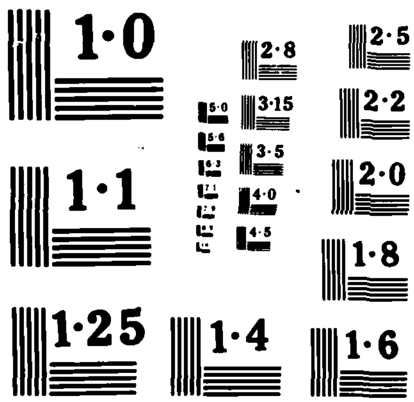
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powered steering gear, the enclosing of the ship's officers and crew in remote and climate-controlled spaces, has made it most difficult for ship's officers to assess the ship's loads, much less the responses of concern. It has, at the same time, made it more important than ever that the naval architect deduce the means for reliably assessing the forces on and of the ship, and the important responses resulting therefrom. Whereas in an earlier time heavy weather ship operations were generally restricted to easing the ship's motions and insuring her survival, today there is an intense demand for extended ship operational performance in all sea and weather conditions. In the commercial transport services it is important that the cargo be delivered intact and on time, that the vessel's schedule be maintained and injuries to the ship, its outfit and crew kept to a minimum. In naval services it is important that fleet units be able to operate so as to serve their mission. In the offshore industries where remote, long time, on-site operations must continue under wide ranging weather and sea conditions, the support craft must be able to perform effectively or costly operations may be interrupted.

It is well known that ships in a seaway experience pitching, heaving, rolling, yaw, sway, etc. Further, it is generally appreciated that ship structure flexes, twists and otherwise distorts in complex fashion. All of these responses to the seaway loadings have generally adverse effects on a ship system's ability to perform its assigned mission. Large pitch and heave motions cause bow emergence or submergence. With bow emergence, underwater electronic devices such as sonar may be exposed and made ineffective; upon bow re-entry into the water large impact pressures may be generated locally causing structural damage. Bow submergence often results in the carriage of massive amounts of green seas over the fore deck, sweeping cargo overboard, damaging upper works and ship's outfit; there have been numerous accounts of personnel being swept overboard as well. Heavy rolling and sway motions make performance of duties by ship's personnel difficult and degrades operational efficiencies. Further, large accelerations associated with heavy rolling give rise to cargo damage, personnel injuries, inability to land helicopters on shipboard landing pads, launch or recover rescue craft or conduct underway replenishment activities. Large yawing motions make course keeping difficult and underway replenishment impossibly dangerous; the heavy rudder action necessary to correct for excessive yawing causes resistance to the ship and thus reduces speed made good for a given power level. Internal to the ship, stresses and strains develop as the hull distorts under its loadings. Hull flexure and twist effect the alignment and operation of weapons, power and control systems, neutralizing their effectiveness in severe sea state conditions. Hatch closure distortion may result in loss of weather tightness with consequent damage of cargoes; hull flexure and twist changes alignment of weapon system fire control devices; propulsion gearing and shafting alignment can be disturbed to such an extent that excessive wear occurs; stresses in critical locations may become excessive and induce fractures.

While there are many ramifications to the broader range of ship response, the more important from an operational point of view are the rigid body responses and the primary stresses and strains of the hull girder. If ships were always to operate by themselves, the control of important ship responses would be a difficult enough task. More and more, however, ships and other marine vehicles are operating in consort. Commercial ships such as the Lash and Seabee designs load and discharge lighters, naval task groups operate in increasingly complex fashion, offshore drillships and drillrigs are operating with supply and support craft of several types.

If all marine vehicles responded to the sea in the same manner, the problem would be eased somewhat, but because of the variations in size, shape, structure and operational requirements, it is seldom, if ever, that any two craft will respond to a given sea in like fashion. Furthermore, because in most cases no two ships occupy the same location in the sea at a given time, they are acted upon differently at any instant in time, even if on average the loadings may be the same. Thus, there will always be relative differences in ship responses even under the most favorable of conditions.

All of this has been so from the beginning of time, but only recently has major importance been attached to understanding the response of ships to seaway loadings for purposes of their control. This has come about because lesser operational problems have been solved, making it possible to thrust more deeply into the ocean environment under operating conditions previously considered most difficult. Today, the needs for ocean transport, resource development, energy conservation and security of the nation's access to the sea make controlled ship operation at sea under the widest range of conditions most important. The ability to effectively conduct operations in or on the sea, commercially or militarily, requires an understanding of the sea loadings, ship system responses thereto and means by which they can be kept under control for the purposes at hand. It is the purpose of this paper to explore this problem at the present state-of-the-art level, to identify the technology which is available for use and to make some suggestions concerning the manner of bringing it to bear on the identified problems. It will be seen in the following sections that first and foremost one must have an adequate description of the state of the sea and the resulting ship responses; it is necessary to be able to reliably assess critical ship responses in a seaway so as to determine what should be done, if anything, when responses are judged excessive or dangerous and there must be a consistent approach to the problem solution so that adequate operational decisions will be made and applied in the solution of the problems. Several examples illustrating the state-of-the-art in implementing different types of vessel Response Monitoring and Guidance Systems are discussed and their effectiveness illustrated. Further, because of advances currently being made in ship-to-shore communications and sea state forecasting, a look ahead is taken at possible future modes of operation for ships in heavy weather.

THE OPERATIONAL ENVIRONMENT

The description of the environment can assume different degrees of detail and accuracy depending on the specific application for which it is intended to be used. Though environmental conditions generally include, in addition to waves, such conditions as wind, current, tide, ice, rain, fog, humidity and others, the ship responses discussed here are primarily wave induced and the discussion of the environmental conditions is therefore limited to the sea state in general or wave data in particular.

The most crude and general description of the sea is in terms of a numerical Sea State Scale or even the less descriptive Beaufort No. Since ships underway cannot as yet constitute a platform for instantaneous wave measurements, sea state conditions can only be determined by visual observation or, alternatively, forecasted or hindcasted analytically. In specific cases measured data are available while the ship is passing through a location where waves are being measured experimentally or routinely. Wave conditions must be defined in order to control operations under limiting conditions and the efficiency of such control is often directly related to the quality of the input wave data.

Early full scale measurement on commercial vessels such as the "WOLVERINE STATE" (1) and the "UNIVERSE IRELAND" (2) were identified with a Beaufort No. scale as routinely recorded onboard in the ship's log book entries. Results obtained under these circumstances exhibited a rather large scatter though the mean trend provided useful information for verification of model tests and theoretical analyses. Other full scale measurements made with special observers onboard (3) (4) were classified by sea state or observed wave height and these naturally resulted in somewhat less scatter of the data, at least to the extent the ship response was depended upon for the definition of the sea only and not other questionable parameters such as ship speed, heading, loading condition, etc. The definition of the sea in terms of the Beaufort No. is a function of the wind speed only and fails to take into account swell, wave propagation from other areas, or the level of development of the storm. The Sea State Scale, however, provides for a better definition of the sea condition directly rather than by means of the wind intensity. Relating wind speed to wave height, though not without merit under specific circumstances, can be rather misleading if no other information, such as the duration and character of the wind build-up and local topographical conditions, is known. By the same token, observed wave height is often equated to significant wave height thereby allowing observed data to be substituted for measurements. Both sea state and wave height observations constitute poor definitions of the sea for the purposes of evaluating or predicting ship responses, other related seakeeping events or associated forces.

Ideally, measured data, digitized and subsequently mathematically or statistically analyzed, provides the desired comprehensive wave data, though forecast or hindcast data are often considered as an acceptable substitute. Experiences gained in running and evaluating the SOWM model (5) over the past three years have led to the conclusion that such wave data, which are available regularly in a directional spectra format for the entire Northern Hemisphere, constitute a most valuable substitute for the unavailable measured data which, at best, are limited to a small non-uniform sample of point spectra. The forecast spectra are generated on shore and updated every twelve hours for up to 72 hours in advance, for some 3000 grid points covering the entire ocean area lying between the equator and 60th north latitude.

The example illustrated in Figure 1 clearly shows the range of frequencies covered and the directional resolution of the present SOWM product, though it is by no means limited to this format nor to the present distance between the grid points, which is approximately 100-200 miles. In the forecasting mode, such data can, in principle, be transmitted to shipboard and directly into a heavy weather response control system, hence allowing generation of information regarding expected behavior of the ship for up to three days ahead. The hindcast spectra are available for the same ocean areas and provide the back-up for statistical climatology. Hindcast spectra will eventually be generated for a full twenty year continuous period, arranged in the same frequency and directional format. Such data are presently being used for the planning of sea operations in general, using the

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	.0	.0	.0	.010	.040	.040	.670	.530	.020	.010	.020	.020	.0	.0	.0	1.360	29.51
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	.190	.200	.220	.190	.020	.010	.0	.0	.0	.0	.0	.0	.0	.0	.0	.830	269.51
	.320	.200	.190	.280	.170	.130	.060	.010	.010	.0	.0	.0	.0	.0	.0	1.370	239.51
	.240	.130	.280	.060	.050	.010	.0	.0	.0	.0	.0	.0	.0	.0	.0	.770	209.51
	.070	.0	.0	.0	.010	.0	.0	.0	.0	.0	.090	.0	.0	.0	.0	.170	179.51
	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	149.51
	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	119.51
	.860	.980	.900	.820	.440	.300	.740	.630	.510	.040	.110	.030	.0	.0	.0	6.360	

$H_{1/3}$ 10.09 ft.

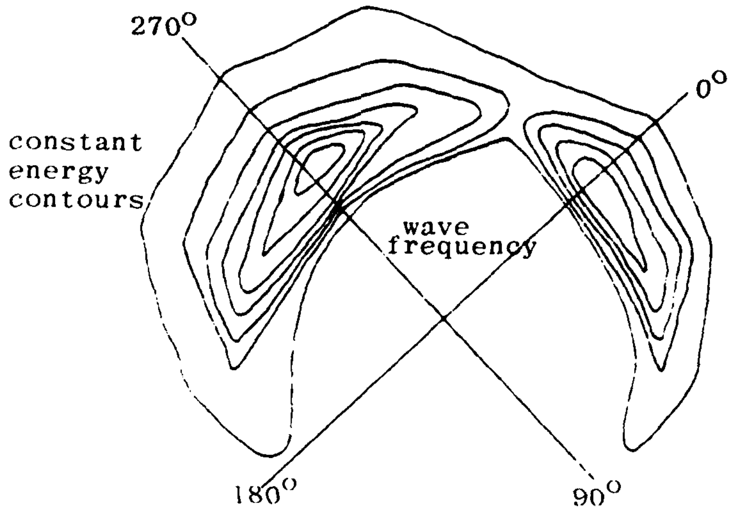

Figure 1. Tabular and Schematic Representation of a Directional Spectrum.

appropriate statistics for the specific location and calendar period to extend the forecast beyond the current forecast limitation of 72 hours. The quality of the SOWM has been proven to be reliable (6) though future improvements are not beyond expectation. The introduction of the Global model in 1979 should extend the coverage of the grid points to the southern hemisphere and, in conjunction with wind data obtained from SEASAT, should provide a more reliable output. From the operational viewpoint, the forecast provides the ideal input to any onboard heavy weather response control system which allows on-line adjustment to be entered by the navigator to account for actual observed conditions. The availability of a detailed wave pattern surrounding the ship, such as in a forecasted directional spectrum, which includes the separation into sea, swell and their corresponding directions and periods, can in many cases help the onboard observer define the condition of the sea more accurately. The climatology data from hindcast studies are not presently practical onboard tools since they are rather voluminous and therefore best utilized by a shore-based station which can communicate with the ship and provide it with the specific wave data required well ahead of its likely encounter. Furthermore, due to the statistical nature of this type of forecast, the predicted ship behavior must be given in a non-deterministic format and include the worst and the best expected in the vicinity of the grid point investigated for the specific two week calendar period under investigation.

While measured data have not been mentioned thus far, they have been ignored only because of their general non-availability. However, in many offshore construction sites, waves are being measured regularly and such data are available in real time onboard. If the computation means are available, such as with a heavy weather response control system, the real time wave spectra can be obtained and used on-line to calculate predicted responses under any desired operating conditions, which may or may not differ from the one for which actual responses are being monitored onboard. Figure 2 illustrates an example of such an on-line analysis indicating a distinct separation of sea and swell for the point spectra shown. If such data are available to the officer on the bridge in numerical and graphical form and they are supplemented by observations of directions from which each of the wave systems is approaching the ship, the point spectrum can be analytically extended to a directional spectrum by using theoretical spreading functions for each wave system as built into the control unit algorithm. In the absence of wave spectra and if only wave height, period and direction of each wave system present is available from measurement or observation, the same approach can be used for analytically constructing a directional spectrum. However, under these circumstances the reliability of the data is not likely to be as good as in the previous case.

The number of locations in the oceans where measurements of wave conditions are being regularly made is rapidly increasing due to the national data buoy projects, increasing numbers of offshore construction sites and other experimental projects. However, data these provide are a mere "drop in the bucket" in relation to the need and their value is greatest for checking the reliability of theoretically generated forecast and hindcast wave data.

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 21/ 5/78   2346        SPECTRAL ORDINATES (M**2/RPS)          H1/3=  1.32 MTRS
PERIOD   FREQ              0.10       0.20       0.30       0.40       0.50
 (SEC)   (RPS) X----------X----------X----------X----------X----------X
  25.6   0.246 !
  20.5   0.307 !
  17.1   0.368 !
  14.6   0.430 !
  12.8   0.491 !++++++++++
  11.4   0.553 !+++++++++++++++++++++++++++++++++++++++++
  10.2   0.614 !++++++++++++++++++++++++++++++
   9.3   0.675 !+++++++++++++++++
   8.5   0.737 !++++++++++
   7.9   0.798 !+++
   7.3   0.860 !+++
   6.8   0.921 !++
   6.4   0.982 !+++++
               !+++++++++++
   5.7   1.105 !++++++++++++++
               !++++++++++++
   5.1   1.228 !+++++++++
               !++++++++
               !+++++++
   4.4   1.412 !++++
```

Figure 2. On Line Bi-Directional Measured Point Spectrum.

The development of an onboard wave measuring device would lead to a major breakthrough in the operation of weather-limited ships using shipboard response control systems. Though the likelihood that there soon will be an inexpensive, commercially available wave measuring unit for onboard use is not yet promising, the availability of an experimental unit could also substantially contribute to the testing and evaluation of onboard ship response control systems.

It is not the purpose of this presentation to elaborate on the format of wave data ideally desired onboard. This has been discussed in detail in (7) and it has been assumed in this presentation that no substitution for spectral wave data can be accepted as a useful input for onboard systems (8). Wave data can be transformed into some sort of spectral format even if only the observed wave height is available. As expected, the reliability of the data will vary substantially with the kind of input parameters used to generate the spectra. For the long range forecast, for which climatology has been suggested, in addition to spectral data representing various wave height ranges at different locations, the probabilities of exceeding different levels of wave height and the likelihood that each level of wave height would persist for a specific duration is also required.

The availability of such data onboard or in a shore-based station capable of communicating regularly with the ship would provide the input data required to "drive" the ship response model. The following section discusses the current status of seakeeping prediction theory as it relates to the operational aspect onboard.

SEAKEEPING PREDICTION - STATE OF THE ART ASSESSMENT

The problems associated with the determination of the seakeeping characteristics of vessels are wide ranging and vary in their solution depending upon whether the problem relates to the design of the ship, its systems and instrumentation, or its operational aspects. Any of these solutions require use of some form of data bank specifically identifying or defining the ship motions and loads induced by the environment in general and the waves in particular.

Since the generation of this data bank, theoretically or experimentally, is a necessary prerequisite for any application of seakeeping, the state-of-the-art assessment must be clearly presented in order to allow a periodic re-assessment as it is advanced with time.

The problem of wave-induced loads on a ship at sea is that of determining successive conditions of dynamic equilibrium of forces and moments acting in and on an elastic body moving in the irregularly disturbed interface of two different fluids. This problem can be simplified by considering external loads only, those acting on the underwater part of the ship which is considered to be a rigid body in an ideal fluid. Motions and other ship responses in waves are often regarded as linear functions of the wave height, and both the irregular waves and the irregular responses can be considered as the sum of money sinusoidal components. Hence, the analysis begins with the study of harmonic oscillations of a rigid body, moving at forward speed on the surface of an ideal fluid under the action of regular gravity waves.

Since the concern here is with successive conditions of dynamic equilibrium, it should be noted that a complete solution of the problem of wave loads cannot be obtained without first determining the motions.

Though in principle the ship motion problem has in some cases been solved for the three-dimensional case (9)(10), the analytical solution is, however, limited to forms such as a sphere or an ellipsoid. In view of this, a less rigorous strip theory solution has been developed which is ideally suitable for long, slender bodies, where each cross section of the ship is considered to be part of an infinitely long cylinder; it has also been successfully used in cases not conforming with the above definition with or without some modification in the approach. Hence, a series of individual two-dimensional problems can be solved separately and then combined to give a solution for the vessel as a whole. The idea was originally introduced by Korvin-Kroukovsky (11) and has since been endorsed, criticized and modified by several others (12)(13)(14).

The main drawback of the strip theory is that it neglects the mutual interactions between the various cross sections, which are of particular importance for certain frequency ranges, depending on the size of the body. Hence, in waves that are either very long or very short relative to a ship, the theoretical justification of strip theory is somewhat questionable. This statement is particularly applicable to lateral motions, since the hydrostatic restoring force is small or non-existant under these circumstances. In spite of these reservations, the basic strip theory has been found to be satisfactory for motions, forces and moments (15)(16), and it is the only practical method for numerical computation available to date.

It is felt that as long as the assumptions and restrictions of the theory are clearly stated, the use of mathematical solutions for ship motion analysis is perfectly justified and offers a much more extensive coverage than the one expected from model tests, which are usually limited to the restrictions imposed by the size of the tank, its instrumentation and economical considerations due to the large number of variables subject to changes of interest. Hence, a minimal model test program is highly recommended, particularly in regular waves; it should be designed to verify the theory by spot checking and providing certain inputs otherwise not available. Tests in irregular waves are only practical as a means of identifying problem areas in general, evaluating procedures visually, etc. Verification of results in irregular waves are best obtained by spot checks in the real environment through full scale instrumentation since model tanks cannot generally reproduce realistic short-crested seas.

In order to help evaluate the state of development of ship motion and load calculation in waves, a short analysis of the basic approach to the problem will first be given.

The mathematical formulation of the problem, i.e., a ship advancing at constant mean speed with arbitrary heading into regular sinusoidal waves, can be presented in a most general form by defining the velocity potential so as to satisfy the Laplace equation, as well as several boundary conditions within the assumptions of the ideal fluid linearized theory. At this stage, no strip theory assumption is required. The time-dependent part of the potential can be decomposed into three components representing the potentials due to incident wave, defraction and the mode of motion considered, as in the original theory of Korvin Kroukovsky (11). An additional time-independent term due to steady forward motion of the ship has been added in more recent theories (14).

In order to obtain a numerical solution, the application of strip theory approximations are necessary for the integration of the sectional exciting and motion-related forces over the length of the ship; these sectional forces involve two-dimensional added mass, damping and displacement terms. The speed-dependent coefficients are expressed in terms of a speed-dependent variable, which is evaluated by means of a strip theory and of a speed-dependent term which is obtained from a line integral along the waterline as given by Stoke's theorem.

Once the formulation of the component potentials is completed, the hydrodynamic forces and moments acting on the hull can be determined. Using the Bernoulli equation, the pressures in the fluid are defined and expanded in a Taylor series about the undisturbed still water position of the hull. Ignoring steady pressure terms, at first, the linearized time-dependent pressure on the hull can be formulated and integrated over the hull surface. The hydrodynamic forces and moments can be obtained in two superimposable parts: those associated with a wave passing a restrained ship (excitation) and those acting on a body forced to oscillate in calm water.

The formulation of hydrodynamic forces and moments permits the equations of motion to be solved and the amplitudes and phase angles of the motions determined as well as velocities and accelerations. Then the longitudinal distribution of all forces, including those that are dependent on the motions and forward speed, can be evaluated and shearing forces and bending moments calculated for any instant in the motion cycle, usually at midship. In general, the solutions for two instants of time suffice to determine the amplitudes and phase angles of these quantities.

In general, this is a convenient stage for a man-machine interaction since the transfer function which could be defined at this stage may be evaluated by comparing with model tests, or other theories based on similar assumptions.

The extension of regular wave results to short-crested irregular seas, by means of a superposition principle, was first suggested by St. Denis and Pierson (17) on the assumption that both the irregular waves and the ship short-term responses are stationary stochastic processes.

Though the method of extending the calculations to irregular waves is universal, the techniques vary considerably depending on the wave input data used and the statistical model applied to the data for long-term predictions. In most cases, spectral representation of the wave is used if available, though it varies between mathematical formulation, actually measured data, single spectrum, etc. The state of the art knowledge with regard to wave data and its application to the prediction of ship responses constitutes a separate subject which has been discussed at some length in (18)(19).

Once the wave spectrum is linearly superimposed on the specific response transfer function, at a constant speed and heading, that response is expressed statistically in terms of the root-mean-square of the process and its multiples representing the $1/n^{th}$ highest expected values. However, the extrapolation of the rms to extreme values as expressed by the $1/n^{th}$ highest is usually limited to return periods characterized as steady state conditions of the sea. Such periods are limited in time and cannot be extended beyond four hours or approximately

5000 reversals. A more reliable extrapolation to longer periods of time is, therefore, required and the use of order statics, or combined cumulative distribution is, therefore, called for. Such extrapolation can be applied to periods representing a storm, two weeks of operation or the lifetime of the vessel, depending on the specific application for which the data are being generated.

A detailed description of the statistical models was presented in (20)(21).

Due to the indeterministic nature of the above conditions, the selection of a single design value to represent a specific response under operational conditions is not always easy. A typical way of presenting motion and load for design purposes is by referring to the level of response expected to be exceeded once in a lifetime of the ship or approximately 10^8 reversals in the case of design values or a two week period in case of a typical ocean voyage. Such a definition requires the use of assumptions with regard to the loading of the ship relative to the waves; the speed expected at each sea state, the specific route or location in question, etc. The response is also largely dependent on the basic design of the ship and particular loading condition as expressed in terms of the metacentric height, GM; the longitudinal, transverse and vertical weight distribution often influences the results significantly.

For operational purposes, a much shorter return period is usually required such as 1-day operation, a voyage, etc. Furthermore, under these circumstances, it is often necessary to determine the expected operability of the vessel under specific sea conditions in terms of the duration of an uninterrupted operation. For this purpose, wave persistance data must be available over and above the wave exceedance data which are usually significant for most design-oriented purposes. A system diagram for a typical ship motion analysis procedure is illustrated in Figure 3.

Several programs are now generally available to determine ship motion loads in regular and irregular waves (13)(22)(23). All three programs are based on similar principles (11) with slight variations in the approach used in solving the problem. The results in regular waves are generally in fair agreement.

The program used in several cases described in this paper is based on a combination of these programs with certain modifications and improvements introduced over the past several years as a result of applying the program and analyzing the dynamic respones of many different hulls. In addition, several full scale measuremens and a variety of model test results have also been analyzed. The modifications, relating to the regular wave parts of the program, include an option for the selection of the best suited hull geometry representation for each transverse section, a sensitivity analysis of input parameters, bilge keel, damping effects and others. Improvements in the statistical analysis include a unique wave spectral file for several ocean locations with an application technique, recently developed statistical prediction methods and design and operational parameter identification logic.

The effectiveness of an analytical seakeeping prediction technology can only be assessed in terms of its ability to reproduce results measured onboard under realistic sea conditions where both the basic theory and the wave application technique are being evaluated or

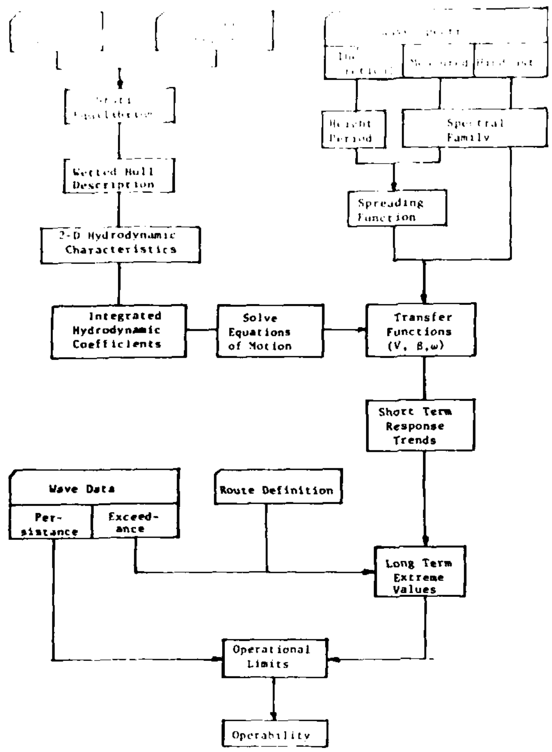

Figure 3. Typical Seakeeping Analysis Procedure.

in the simpler case to reproduce regular wave results measured in a model tank. While results for the latter are available, the comparison of full scale measurement and theoretically derived results based on actual simulated environmental conditions are rather scarce. An onboard heavy weather control system is an excellent tool for providing such data, as shown in (24), and hence serves as its own tuning device for having a more effective and accurate system onboard. Since model tests are limited in their effectiveness, particularly when simulating realistic sea conditions, the onboard system allows an effective full-scale test which can be achieved at a minimum cost as a bv-product of an operational tool.

Further, such onboard monitoring is the only means available for evaluating present operational limits on seakeeping events and responses. A more detailed discussion on the subject is provided in the following sections.

ONBOARD RESPONSE ASSESSMENT

Ship's officers have a natural concern for the responses of their ship in a seaway. It only requires experience in a good storm or two for one to appreciate that structure, cargo, outfit and special devices can be severely damaged when ship motions become large in pitch or roll, or that the vessel may become virtually uncontrollable when its yawing motion in a following sea is excessive; coupled yaw-roll motions can lead to capsizing.

Subjectively, the ship's officer tries to determine for himself when a level of ship response is too severe, just as he tries to master the Beaufort Scale so that his assessment of wind and sea conditions will be the same as that of an experienced master. Without reliable shipboard instrumentation, however, the consistency of such judgments is questionable and subject to bias.

Such a manner of ship management may have been adequate in earlier times, but with today's larger, faster, more complex and costly ship systems this state of affairs is quite inadequate. It is known that the subjective evaluation of the sea state in terms of significant wave height, period and direction and a ship's responses thereto leads to inconsistent conclusions and therefore decisions based upon them. A ship officer's assessment of the sea state is some unknown function of his experience (all cannot have 50 years at sea behind them), the size of the vessel, the observers location on the ship, the speed and course of the ship in relation to the magnitude and predominant direction of the waves (assessing sea and swell, their direction and period can be a most difficult task), the time of the day and perhaps other factors (experienced observers are relatively few in number and usually specialists of one type or other). Under such circumstances it is quite understandable that some ship masters develop reputations for timidity or boldness depending upon their personalities as well as their experiences. The result often is that some ship operations are costly in terms of damages incurred while others are costly because of unnecessary delays or inability to perform assigned tasks as required; a task group commander on a carrier could well induce imprudent action on the part of a young, inexperienced skipper of a screen vessel, resulting in loss of operational effectiveness, damage to vessels and equipment and even personnel, if conditions were severe enough. To avoid such possibilities it is necessary that there be means for reliable assessments of ship responses during heavy weather operations, that they be evaluated in the light of established operating requirements and recognized prudent levels of operation. This can make possible reliably good decisions by ship's officers.

The last two decades have seen the successful application of a growing number of ship response sensors suited to the routine monitoring of important responses (1)(2). Accelerometers, gyro stabilized roll indicators, strain gages, velocity gages and relative motion sensors have all been used at one time or other for particular purposes. See Figure 4 for a generalized sensor system schematic. In most work, the intention has been to gather data for analysis ashore, not to provide meaningful information for shipboard use. The data have been collected using strip chart or magnetic tape recorders and taken ashore where analysis by manual or electronic data processing means was possible. The recent development of mini-computers suitable for use at sea has changed the situation completely. It is now not only feasible, but reasonable to carry out both data collection and analysis on board. The results of analysis can therefore be made available for onboard assessment and decision-making and at the same time be compactly stored on computer subsystems for further study and evaluation work ashore.

Responses of concern are generally several, each requiring particular attention for adequate treatment. As noted earlier, pitching and heaving motions are most frequently of concern because they lead to bow emergence, with exposure of forward mounted electronic detection devices and generation of slam loadings on the bow structure, and submergence of the foredeck with green seas carried aboard and subsequent dangerous loadings of topside structure, fittings and cargoes; large

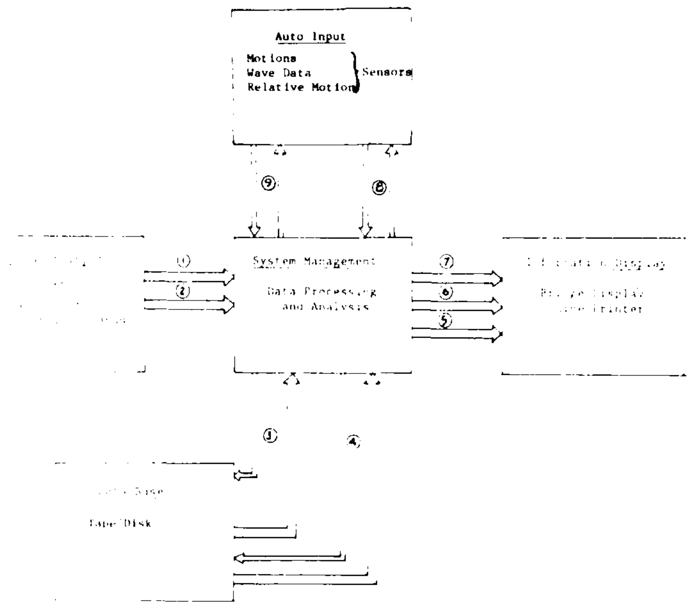

Figure 4. Schematic Diagram of a Generalized Ship Response Control System.

vertical and transverse accelerations can also be experienced with consequent damage to inadequately secured items and impairment of the crew's ability to work the ship. By installing accelerometers at the ship's bow, it is possible to measure continuously the accelerations being experienced in time. By installing strain gages on bow frames, a counting of pitching motion can be accomplished; with judicious selection of their location, the magnitude of loading can be consistently assessed and related to bow emergence or submergence events. Strain gages have likewise been used for sensing hull girder flexure and torsional stresses in critical locations. Use of a gyro stabilized roll indicator allows measurement of true roll or pitch angles so that helicopter landing or search and rescue operations can be judged as feasible, safe, etc., when such motions are within established limits.

Whatever the response of interest, it is now possible to reliably sense it at sea and make the information of interest available to the ship's staff in real time. The data can be continuously acquired, periodically processed, analyzed and presented in a meaningful format for routine use in ship operational decision-making. In a later section, some examples of recent work illustrate the point.

Midship bending stresses in a ship's hull girder are recognized as of primary concern to vessel safety. The hull girder stress variation while underway can arise from several sources:

a) Consumption of fuel, water and stores will change the still water stress, a stress that changes but little hour by hour.
b) Changes in course, speed and sea state will change the low frequency variations of stress caused by the vessel's passage through wave systems.
c) Dynamic response of the hull girder, with relatively higher frequency variations, will result from wave impacts due to slamming or carrying green seas aboard.

data analysis, storage and automatic printout, the important information on critical responses can be prepared for transmission ashore either in hard copy or through high speed data links such as by satellite communication. In this way important responses can be related to the operating environment, the shore staff can determine limits of ship operability for purposes of future route planning or cargo operations in road steads. Cargo damages can be kept under closer scrutiny as to probable cause and heavy weather damage occurrences can be related to conditions experienced. Such data on routine operations is of great value in providing a foundation for setting design requirements for future fleet procurement; the absence of such experience data has been a serious lacking in the past and which has been a major deterrent to innovative development in ship design. There no longer is any excuse for such a deficiency.

In the case of the HELM system,management ashore is kept abreast of construction progress offshore, and it has been shown that data monitored by the system is used to update the various scheduling programs so as to provide a projection to the end of the project, predict problems ahead of time due to slippage in scheduling, etc. The wave data recorded onboard is also used to update long range forecast of waves as well as tune the on-line model.

No nation has, today, any means at their disposal for reliably assessing the ship handling expertise of shipboard personnel as it may relate to minimum standards of seakeeping for ship classes. Similarly, no means are available for determining the level of ship officer skills in the quantitative assessment of sea state. Consequently, traditional practice is to let each ship Master try to control his ship as best he can. Some have better judgment than others; some have better luck than others. The most successful efforts usually involve the avoidance of waters having a high frequency of heavy weather or sea states that would be detrimental to safe, efficient vessel operations. The result of this situation is that ships most frequently travel extra distances as a general strategy. Even so, cargo damage in commercial services typically amounts to around 1% of revenue on normally travelled fair weather routes and has been as high as 5% and more on trade routes with a high frequency of heavy weather occurrence.

If one were to turn such figures into dollars, they would vary of course with the size and type of ship considered as well as their service. A 30,000 deadweight ton dry cargo vessel will cost in excess of $1,500 per hour while operating normally at sea, about $500 of which would be for fuel costs alone.

A new $130,000,000 LNG carrier will cost about twice that amount in capital charges alone. Offshore drill rigs operating in the Gulf of Alaska or the North Sea can have costs in the range of $5,000 or more, which go on whether they are conducting operations normally or not. Todays naval task group can easily represent a capital cost of over three billion and a manpower commitment of five thousand or more. The loss of utility of such a force even for an hour represents a staggering sum.

What then can be justified in the way of investment to minimize such potential losses. Recent cost estimates for the installation of systems on large ships such that the advances in seakeeping and shipboard instrumentation discussed above can be made available for improved ship performance in heavy weather operations suggest the following costs for oceangoing ships on a one of a kind basis.

made by the officers on the bridge during heavy weather operations, such as in slamming frequency observation, or registering of shipping of water events, has been proven effective. The system counts the number of slams in each fifteen-minute interval by sensing with midship gauges, the high frequency stress above a predetermined threshold level and noting the intensity of the highest. Hence it provides an unbiased account of the significant slams and also shows trends over time. Other technological benefits which have become available through the implementation of such onboard systems relate to the ability of setting threshold values and limits of operation which are otherwise determined by "gut feeling". The level of roll at which a Master decides to take preventative action, or the number of slams per hour which would induce him to slow down, are easily determined for the system output and can be averaged to obtain the reaction of several Masters or to correlate with damage experienced.

In the case of offshore construction work, the limits of operation of crane ships as a function of the boom tip displacement and velocity were determined soon after the installation of onboard monitoring and predicting system and were later used for assessing progress, settling disputes and performing pre-analysis of offshore construction scheduling

As an operating system under test and evaluation, the one aboard the LASH ITALIA has been used by numerous persons of varying experience. One of the Masters had 35 years of sea experience to his credit and was initially reluctant to admit a need for guidance, must less quantitative information on his ship's responses. A midshipman of the U.S. Merchant Marine Academy, assigned to the vessel as part of his sea training, was given a one day introduction to the system ashore, an onboard system checkout and provided with instruction manuals for reference. He and others similarly trained on subsequent assignment became the initial system users while at sea. As experience aboard grew, even the most experienced Master began to rely on the system, first to confirm his judgments and later to provide guidance when sea conditions were more severe. In subsequent program reviews, the utility of the system in assessing ship performance, expected future operating conditions and the possibilities for multiple uses of the system computing capabilities for other ship business were of high interest to the ship's younger Master and the shoreside technical staff. It was suggested that each Master and Chief Officer be given special introductory training as part of any future work with the system.

Because it is not necessary that the shipboard personnel fully understand the theory behind the capability being made available to them, current assessment of training needs suggest that a two day introduction should be given to the ship's officers who will use the system. In this effort, it should be remembered that ship's officers have a general appreciation of ship responses but do not have an understanding of the particular relations between events or the importance of the statistical values which can be used for operational guidance. Accordingly, a qualitative introduction to the phenomena of concern and the manner of their treatment should be given along with a detailed familiarization with the system operation through an onboard system checkout. While this is necessary to insure proper system utilization, confidence in system utility takes a longer time and in-service experience is required for that to develop.

As a tool for the improvement of ship operations the system can simultaneously provide shoreside management information for use in performance evaluation and planning of future operations. With onboard systems are not only scientifically reliable, but technically effective, operationally efficient and economically beneficial, as well. As a side benefit it would also be most useful if such systems could be used as management tools to enhance planning and control of resource utilization.

Scientifically, the systems installed to date have made major contributions to the state of knowledge concerning ship stresses and motions on regular trade routes and variations in hull loads due to cargo operations. There has been reinforcement of assessments by senior ship masters and guidance given to younger, less experienced officers. As a tool for evaluation, the system on the LASH ITALIA has provided excellent correlations between predicted and measured ship response spectra. It has been used to asses the validity of sea state predictions in the mid-Atlantic for 24, 48 and 72 hour forecasts of the Spectral Ocean Wave Model (28). As a tool for route planning, SOWM forecast and hindcast data have been used on board to compare expected sea conditions and ship responses with those experienced (28), thus allowing an opportunity to prepare for rising sea conditions or the altering of the vessel's planned route.

A typical by-product of the LASH ITALIA system has been the generation of still water stress data covering 30 months of operation. The results were plotted as a histogram showing the distribution of various levels of stress as a function of east and westbound crossings (27). Other data of scientific value was obtained from the LASH ITALIA system showing the general relationship between various responses and wave heights as obtained from theoretical calculations using general spectral families and as measured on board using observed and hindcast wave data (26)(27).

The analysis of other data collected so far by the several HELM systems in use in the North Sea has been particularly beneficial since the responses are measured simultaneously with the corresponding wave data and hence provide an excellent basis for comparison of the theoretically derived and the measured data. Furthermore, the HELM system provides for an absolute prediction rather than a relative one as has been the case so far with the Ship Response Monitoring and Guidance System. It should also be noted that since all HELM systems are provided with a static analysis capabilities of the ship loading condition, the responses measured are defined for the specific condition and can be reproduced theoretically without the need to resort to generalization such as light or loaded condition. Instead, the drafts, GM, and radius of gyration, longitudinally and transversely, are defined. Some of the results obtained so far are illustrated in references (24)(31).

In a technological sense, the down time of the system on the LASH ITALIA has been minimal and caused by a noisy fan bearing in one case, a faulty computer circuit board in another, and a faulty sensor in a third. Since all system service was accomplished in the ship's home port, down time was significantly greater than would normally be expected if the system were a standard element of the ship's instrumentation. It should be noted that all system components reflect established technology and standard application techniques. Therefore, were such a system designed into the ship's instrumentation system, it could be expected that service requirements would be at the same level as presently is the marine electronic experience.

The use of a computerized system onboard to supplement observations

An adaptation of the system designed for oceangoing ships and offshore vessels but for specific use on Great Lakes Bulk Carriers is currently being investigated under another project also sponsored by the Maritime Administration. This work is being carried out by Webb Institute of Naval Architecture and Hoffman Maritime Consultants Inc. with additional support from Marine Consultants & Designers in Cleveland, Ohio, and Teledyne Engineering Services in Waltham, Mass. Though ship motions on the Great Lakes are definitely secondary in importance to the stress problem, the recommended system emerging from the study seems most likely to include sensors for stress measurements at midship and possibly elsewhere along the ship's length, as well as vertical and lateral accelerometers at the bow and bridge, respectively. The software for the proposed Great Lakes system is likely to be more sophisticated when dealing with the separation of the quasi static low frequency wave bending stress and the higher frequency stresses due to the wave induced "springing" under certain operational conditions. The Great Lakes Hull Stress Warning System will most likely also be provided with an extensive graphical display though a CRT on the bridge and a numerical display available as an alternative option.

The experience with these systems discussed thus far has been limited to efforts underway in the U.S.A. over the past five years. Parallel work has been underway in Norway over the past few years as well (29)(30), under the title of Hull Surveillance System and supported by the Norwegian Counsel of Scientific and Industrial Research (NTNF). It has been carried out by Det Norske Veritas, the Norwegian Maritime Directorate and Norcontrol. Several first generation systems were tested during the first half of the seventies and a second generation system is currently being evaluated. Though the stated objectives of the project are somewhat different from those in the U.S.A., the product emerging for the two independent projects is not too different and it is likely that a mutual benefit could materialize in future.

The most recent awakening to the problem of shipboard monitoring and response predictions has resulted in a joint British-Dutch effort to design and evaluate such systems for commercial and other applications. This project is just getting underway and the information available is rather limited as yet.

It is evident from the preceeding survey that the experiences which are being gained onboard relevant to the application of such systems is of major importance and it is expected that by the beginning of the next decade the world maritime community will likely have at its disposal several versions of such systems having a wide variety of capabilities and providing a wealth of full scale data relative to the behavior of ships at sea, which today is rather scarce. However, the only way such systems can be accepted commercially is through their undisputed effectiveness, which must be clearly demonstrated in terms of technical, economical and safety accomplishments.

SYSTEM EFFECTIVENESS

Regardless of how great the need, or how sound the theory, "The proof of the pudding is in the tasting". The design, construction, installation and operation of shipboard instrumentation is not without added procurement and support costs, competition for critically short shipboard space, weight and manpower requirements. In order that the above-discussed ideas be accepted for implementation in a shipboard Heavy Weather Response Monitoring and Guidance System, there must be adequate justification. There must be sound demonstration that such which are operation limited by heave motion of the riser, etc.

Experience on the LASH ITALIA and with the various HELM installations is being applied to two projects currently in the final stages of implementation. The USNS FURMAN which has been routed by FNWC over the past few years will be equipped with the latest generation Ship Response Monitoring and Guidance System during the fall of this year. The system will include standard off the shelf hardware which has been successfully tested on the LASH ITALIA. Further substantial changes will be made to this hardware by adding a roll gyro, floppy disc facility, CRT, etc. A totally new software system will be developed for the FURMAN system. This software will be updated and extended onboard as the test and evaluation project requires. The project is jointly sponsored by the USCG, NMRC and FNWC with each of the above participants hoping to achieve some or all of the following goals:

- To provide for the test and evaluation of a ship response monitoring and guidance instrumentation system having shipboard response analysis and information display for utilization in cargo vessel operation while under Optimum Track Ship Routing on North Pacific trade routes.
- To use the system to assist in the development of principles and tools for shore-based vessel routing using communication and information display onboard and ashore for interaction between routing and operations personnel.
- To carry out assessment of human understanding and response to important ship response data on heavy weather ship performance.
- To evaluate the reliability and suitability of employing ship responses computed ashore using ship motion theory and the Spectral Ocean Wave Model for making considerations effecting vessel routing.
- To evaluate basic considerations of tactical routing.
- To identify inadequacies of supporting theoretical techniques.
- To assess system performance and hardware reliability in a North Pacific environment and thus expand the range of experience with shipboard heavy weather response measurement and data processing systems.
- To determine training requirements for effective utilization of ship response monitoring and guidance systems.
- To establish installation and maintenance specifications for reliable operation of such systems.

While for NMRC the FURMAN project is a logical followup of the LASH ITALIA evaluation and includes the implementation of some of the conclusions reached during 1976-78 test and evaluation program, the USCG hope to gain sufficient experience with the system that they may better understand its potential for enhancing vessel safety. FNWC are expected to operate a simultaneous shore-based system of the Ship Response Monitoring and Guidance System so as to facilitate easy comparison of data predicted onshore and monitored and adjusted onboard. The FURMAN installation is likely to be operational for 2-3 years and, in view of the forecasted and hindcasted directional spectra available for the routes previously followed by the ship, the benefits of such a study may go well beyond the nominal evaluation of an onboard system and allow a significant advance in the state-of-the-art knowledge of ship response prediction, of wave data utilization for such purposes, effectiveness of forecasting and hindcasting and, in general, the seakeeping behavior of such a vessel.

MEASUREMENT

Time	Wave - M $H_{1/3}$	Wave - M $T_{1/3}$	Wave - M Max	Boom Tip Motion - M $H_{1/3}$	Boom Tip Motion - M $T_{1/3}$	Boom Tip Motion - M Max	Boom Tip Motion - M Vel	Roll-Deg $H_{1/3}$
1738	2.4	10.4	6.2	1.5	15.7	2.1	1.43	4.4
1935	2.7	11.4	5.6	3.9	15.5	5.9	1.10	2.4

1738 PREDICTION

Ship Heading = 80.0 deg.
Crane and -Bow = 0 = 175.0 deg.
Module Weight = 0.0 Mt.
Draft = 9.5 M Fwd 10.5 M Aft

Time Period = 0.2 hours
Crane Outreach = 74.0 M
Hook Ht. above WL = 10.0 M
GM = 6.3 M

RECORDED SPECTRA
1738 14/12/77 Sea 190 Swell 260 Split Prd 10.5 Sec.

Ship Hdg.	ROLL Avg.	ROLL High	ROLL Prd.	VERTICAL BOOM TIP MOTION Avg.	VERTICAL BOOM TIP MOTION High	VERTICAL BOOM TIP MOTION Prd.	VERTICAL BOOM TIP MOTION Vel.
60.0	4.33	6.91	12.3	1.63	2.55	15.3	0.43
70.0	4.55	7.27	12.2	1.65	2.60	14.9	0.44
80.0	4.76	7.62	12.1	1.68	-2.65	14.6	0.46
90.0	4.97	7.96	12.1	1.73	2.72	14.5	0.47
100.0	5.14	8.21	12.2	1.78	2.80	14.6	0.49

1935 PREDICTION

Ship Heading = 80.0 deg.
Crane and -Bow = 10.0 deg.
Module Weight = 180.0 Mt.
Draft = 9.5 M Fwd. 10.5 M Aft

Time Period = 0.2 Hours
Crane Outreach = 45.0 M
Hook Ht. above WL = 45.0 M
GM = 6.3 M

RECORDED SPECTRA 14/12/77
1935 Sea 190 Swell 260 Split Prd. 10.5 Sec.

Ship Hdg.	ROLL Avg.	ROLL High	ROLL Prd.	VERTICAL BOOM TIP MOTION Avg.	VERTICAL BOOM TIP MOTION High	VERTICAL BOOM TIP MOTION Prd.	VERTICAL BOOM TIP MOTION Vel.
60.0	3.02	4.77	14.0	3.53	5.51	15.5	0.89
70.0	2.72	4.31	13.4	3.54	5.53	15.4	0.90
80.0	2.65	4.21	13.1	3.56	5.57	15.4	0.91
90.0	2.80	4.45	13.2	3.59	5.62	15.3	0.93
100.0	3.31	5.23	13.7	3.62	5.67	15.2	0.94

Figure 10. Measured and Predicted Responses of a HELM System.

predictive mode of the system provides the operator with the ability to simulate any condition that the vessel is likely to encounter during the operation without having to expose the ship to such hazards as losing the boom due to excessive angular displacement, destruction of a module due to impact, etc. The system assists in the selection of the ideal heading for the ship in the prevailing wave system as well as indicates the relative advantage of slight variations in heading under moored conditions. Documentation of optional ballasting arrangements, preparation of daily progress reports to management and generation of a data bank of wave and ship responses second to none are additional benefits to be obtained as by-products of this system for use by both offshore operational personnel and shore-based management and engineering staff. Since the ship is not underway, actual measured wave data are available onboard. Further, the ship's proximity to shore makes communication less complicated. Provisions have been made for the input of forecasted directional spectra which are available for up to 72 hours in advance. The actual mathematical modelling of the crane and ship motions are carried out in real time to provide for actual motion simulation rather than interpolated results which one has to be content with in other shipboard systems. The system capability is now being extended to other offshore applications including pipelaying operation where the stinger motions are of a critical nature, diving vessels, which have to control the lowering of vehicles through the moon pool, support vessels, with their fire-fighting capabilities, drilling vessels.

TIME	BENDING				ACCELERATION				TRANSIENTS					
				STD	VERT		LATERAL		SLAM		BOW		AFT	
	AVG	MAX	MIN	DEV	RMS	MAX	RMS	MAX	#	MAX	#	MAX	#	MAX
0017	1.4	7.5	9.[illegible]	2.2	.10	.32	.02	.10	2	2.9•	3	3.5•	0	.0
0032	1.5	7.9	9.1	2.0	.09	.37	.02	.10	1	2.4•	3	5.1•	1	.7
0047	1.5	5.8	6.5	1.9	.09	.27	.02	.08	0	.0	0	.0	0	.0
0102	1.5	6.4	8.5	2.1	.09	.34	.02	.09	0	.0	5	3.0•	0	.0
0117	1.5	7.7	8.3	2.1	.09	.27	.02	.11	0	.0	5	4.6•	0	.0
0132	1.5	6.0	7.3	2.1	.10	.29	.02	.08	0	.0	4	3.1•	0	.0
0147	1.5	7.5	9.4	2.4	.11	.40	.02	.09	1	2.4•	6	3.5•	0	.0
0202	1.6	6.9	8.7	2.2	.10	.34	.02	.10	1	2.3•	6	4.2•	0	.0
0217	1.5	6.2	7.8	2.1	.10	.32	.02	.09	0	.0	5	3.0•	0	.0
0232	1.5	5.8	6.4	1.9	.09	.27	.02	.08	0	.0	1	2.5•	0	.0
0247	1.5	6.4	8.3	2.1	.09	.30	.02	.09	0	.0	1	3.1•	0	.0
0316	1.6	6.1	6.9	2.0	.09	.30	.01	.09	0	.0	1	2.9•	0	.0
0331	1.7	7.9	9.9	2.0	.09	.40	.02	.09	0	.0	8	5.6•	0	.0
0346	1.8	6.2	8.0	2.0	.09	.33	.01	.07	0	.0	2	2.6•	0	.0
0401	1.8	5.2	6.5	1.8	.08	.27	.02	.06	0	.0	3	2.9•	0	.0
0416	1.9	6.8	7.7	1.9	.09	.30	.02	.07	0	.0	4	4.0•	0	.0
0432	2.0	5.5	6.7	1.8	.08	.27	.02	.07	0	.0	0	.0	0	.0
0447	2.0	5.6	7.8	1.8	.09	.30	.02	.08	0	.0	3	3.3•	0	.0
0502	2.0	7.4	9.2	2.0	.09	.34	.01	.09	0	.0	3	3.1•	0	.0
0517	2.0	5.5	7.2	1.8	.08	.25	.02	.08	0	.0	2	4.7•	0	.0
0532	2.0	5.9	7.3	2.0	.09	.28	.01	.08	0	.0	4	3.4•	0	.0
0547	2.0	5.6	6.4	1.6	.08	.26	.02	.08	0	.0	1	2.4•	0	.0
0616	2.0	5.8	7.7	1.9	.09	.31	.01	.07	0	.0	2	3.1•	0	.0
0631	2.0	5.7	7.4	1.7	.08	.28	.01	.07	0	.0	2	4.0•	0	.0
0646	2.0	4.6	5.6	1.5	.07	.24	.01	.06	0	.0	1	3.6•	0	.0
0701	2.0	5.1	6.5	1.6	.08	.26	.02	.08	0	.0	0	.0	0	.0
0716	2.1	5.7	7.0	1.6	.08	.28	.01	.07	0	.0	1	4.1•	0	.0
0732	2.1	5.9	7.4	1.7	.08	.32	.01	.07	0	.0	1	2.7•	0	.0
0747	2.1	4.9	5.5	1.5	.07	.25	.02	.08	0	.0	1	2.9•	0	.0
0802	2.0	5.1	6.2	1.5	.08	.27	.02	.07	0	.0	1	2.8•	0	.0
0817	2.0	4.2	5.2	1.5	.07	.24	.02	.07	0	.0	0	.0	0	.0
0832	1.9	5.3	6.4	1.5	.08	.34	.02	.08	0	.0	1	2.6•	0	.0
0847	1.9	5.2	6.5	1.6	.09	.26	.02	.08	0	.0	2	3.4•	0	.0
0916	2.0	5.0	5.6	1.5	.07	.26	.02	.08	0	.0	1	4.1•	0	.0
0931	2.0	3.1	4.3	1.3	.06	.22	.02	.09	0	.0	0	.0	0	.0
0946	2.0	5.2	5.9	1.5	.08	.27	.02	.10	0	.0	1	2.7•	0	.0

Figure 9. Onboard Data Display in a Monitoring Mode

The need for a Heavy Weather System with an integral static loader has been repeatedly indicated by shipboard personnel in order to provide correct hydrostatic and initial stability data input while also taking advantage of the onboard computer availability for loading calculation purposes.

An entirely different type of response control system consisting of fundamentally different hardware and software systems designed for application to weather-bound offshore construction operation was introduced in early 1977 in the North Sea. This system is software oriented using standard off-the-shelf hardware and software generally associated with large main-frame computers. The Heavy Lift Monitoring and Predicting System (HELM) represents a fundamental breakthrough in computerized marine systems. Its monitoring functions serve merely as input to its predictive capabilities which allows system simulation of operating scenarios and enables operating personnel to adjust proposed operations to conform to the vessel's predicted ability to perform acceptably. HELM System effectiveness have been so striking that 4 systems have been installed this year on major crane vessels operating in the North Sea; sever oil companies have required the system for use in offshore construction work in hostile environments. The HELM system was designed by Hoffman Maritime Consultants Inc. as an aid in the conduct of heavy lift work offshore, which is one of the most weather restricted operations carried out in the ocean environment. The system monitors key responses, such as roll, boom tip, absolute and relative motion, and the sea state. All signals are analyzed on-line and displayed numerically and graphically. (See Figure 10). The

Though the system on the ANTONIA JOHNSON performed according to specifications, it failed to answer the needs of the Captain who specifically requested additional monitoring and guidance information relevant to pitching and rolling. The Master of the 500' long ANTONIA JOHNSON indeed had cause for concern, particularly when travelling along the west coast of the U.S. during the winter. The system was removed after 4-5 months of operation for hardware modifications and was never re-installed. The installation on the Sea Land ECONOMY (U.S. Gulf to Northern Europe) included an accelerometer at the bow and a pair of strain gauges amidship. This system was plagued with sensor problems and was removed after several voyages without providing feedback of significant value. In spite of the relatively short period of installation onboard the ANTONIA JOHNSON, the results obtained while there indicated that the technology incorporated in the system was adequate, the prediction and guidance was reliable and that, with certain modifications, the system could become an important tool for use on the bridge during periods of heavy weather. Other than one additional system installed on a tanker, which has been laid-up following its completion in early 1975, no additional experience with the system was obtained until late in 1975 when the implementations of the recommendations in reference (25) lead to the installation of a newly modified and extended system on board the LASH ITALIA. This project was sponsored by NMRC and was carried out at Webb Institute of Naval Architecture with participation by Edo Corp. and Prudential Lines Inc. Modifications and additions to the system installed on the LASH ITALIA affected both hardware and software and the new system was expanded well beyond the original scope of the work. The new system specification for the hardware and software was formulated at Webb Institute and implemented by Edo under subcontract to Prudential Lines, Inc. and NMRC. The installation onboard the LASH ITALIA took place in November 1975 and the system has been continuously in use through May 1978, i.e., approximately 30 months. The route of the ship has been consistently from the eastern Mediterranean. In addition to the basic bow accelerometer and midship strain gauges port and starboard, a lateral accelerometer was installed close to the bridge, several strain gauges were installed on the bow and the deck forward (26) to indicate possible magnitude and frequency of shipping of water. The software was extended to separate low and high frequency stress at midship and hence provide for detection of a slam or bow impact event with the midship strain gauges. Data display in the monitoring mode was reformatted, to that shown in Figure 9, so as to include the added sensors and present the data in a format more appealing to the user onboard. An on-line spectral analysis of measured data was also incorporated to provide for a frequency domain representation. Results obtained on the LASH ITALIA have been presented in several papers (26)(27)(28) which summarizes four manned voyages during the winters of 1976, 1977 and 1978, as well as continuously monitored data collected over fifteen return voyages across the Atlantic.

The system onboard the LASH ITALIA has been used by the officers on the bridge during heavy weather to evaluate trends of accelerations and bending stress, to provide a comparative level for response evaluation so as to establish a performance index for the vessel and in the guidance mode to predict the expected variations in the different responses as a result of changes in vessel speed and heading.

The degree to which each of these modes of operation was found to be adequate vary somewhat. In some cases, it was concluded that changes in format only are necessary, in other cases it has been found that additional information should be displayed and changes should be conclusion that no system then available met minimum requirements for such an on-board tool. A research program aimed at the design and evaluation of a new ideal system was initiated under National Maritime Research Center sponsorship. Since it was established in (25) that many of the existing systems used similar hardware, it was concluded some of these systems could be used as a basis for the design of a new, improved system. To minimize duplication of effort in pursuing this goal, it was decided then to select the hardware of the EDO Heavy Weather Damage Avoidance System as the basic hardware configuration, to modify, improve and eventually replace its software in conformance with new system requirements.

Shipboard experience with EDO's system was limited to two experimental installations on the S S ANTONIA JOHNSON (see Figure 8) and the S S SEALAND ECONOMY. The former installation (during the winter season 1974/75, Northern Europe to Western U S A) led to system hardware redesign so that it could withstand conditions of heavy weather operation. The sensors on the S S ANTONIA JOHNSON consisted of two strain gage pairs, at side amidships and the forward quarter point. .This system was designed to analyze the signals and print out the mean static bending moments and the mean and maximum dynamic variations about them. A short term prediction feature provided the operator with maximum expected values in the period following measurements; this was repeated each 3 hours for 15 minutes. A maneuvering analysis using precalculated values provided assessment of changes expected to result from modifying ship heading, speed and draft.

Figure 8. Computer and Electronics Cabinet as installed on S S ANTONIA JOHNSON

Load Condition 1   Speed = 12 MPH   Wave Dir. = 18 Deg.

	Heading	(DEG)	65	80	Present Hdg.	110	125
	Speed M.G.	(KN)	10.40	11.60	12.80	11.60	10.40
	Acc. Fwd.	(G/S)	.10	.09	.07	.05	.06
6	Lat. Acc.	(G/S)	.07	.11	.14	.20	.15
KN	Roll	(DEG)	10.00	9.00	10.00	12.00	15.00
	Slams	(NO)	0.00	1.00	2.00	1.00	0.00
	Bow Imp.	(NO)	0.00	1.00	4.00	1.00	0.00
	Mid Stress	(TPSI)	3.80	3.40	2.80	2.00	2.50
	% Allowable Stress		71	66	57	47	54
	Speed M.G.	(KN)	8.70	9.60	10.00	9.60	8.70
	Acc. Fwd.	(G/S)	.07	.05	.03	.03	.03
12	Lat. Acc.	(G/S)	.08	.13	.18	.20	.16
KN	Roll	(DEG)	10.00	9.00	10.00	12.00	14.00
	Slams	(NO)			0.00		
	Bow Imp.	(NO)			0.00		
	Mid Stress	(TPSI)	4.10	3.90	3.50	2.90	3.50
	% Allowable Stress		59	55	50	41	50
8 KN	% Allowable B.M.		56	53	48	40	49

Figure 7. Digital Display of Predictions.

It should be the function of shipboard instrumentation systems to provide reliable data in meaningful form with an identified range of performance parameters available for use in their evaluation. There needs to be interaction capability such that for a response of interest the input of minimum data on proposed changes will allow query for effects of proposed changes. The ship's officer may only have course and speed change under his control, thus the system must require only this at most. On this basis, it should be possible to obtain by query any change in response of interest, roll, pitch, midship stress, etc, so that selection of the most favorable action will be readily accomplished. In multi-vessel operations it is necessary that those in operational command have similar, and parallel capability for all units operating together in order that there be realistic expectations of ship performance at hand.

Data display for effective evaluation should include, if possible, maximum, minimum, average, significant or rms and event counts of interest and time intervals for data update should be related to time intervals needed for obtaining stable results; for instance about 15 minutes seems generally required for obtaining stable stress data on large ships. All predictions should provide comparable data projections for proper evaluation and decision-making.

OPERATIONAL EXPERIENCE WITH HEAVY WEATHER RESPONSE CONTROL SYSTEM

Heavy weather response control systems of various degrees of sophistication have been suggested and in some cases tried onboard different vessels over the past several years. A survey of available systems was performed by Hoffman and Lewis (25) in 1975, which lead to the

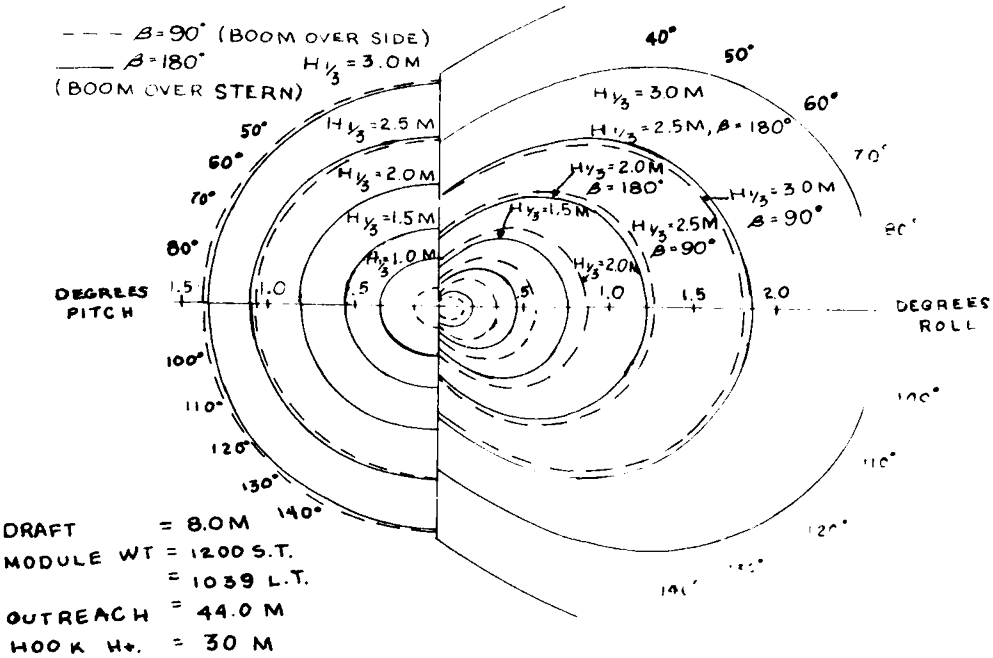

Figure 6. Onboard Guidance Chart.

board simulation of ship maneuvers is possible and the needed information can be displayed on demand prior to decision. Then upon execution, there should be confirmation that the prediction was valid and that indeed the maneuver resulted in bringing the troublesome response under the desired level of control without creating another problem in the process. An illustration of a display in answer to these requirements is given in Figure 7.

In the above, it is seen that there is required input by the ship's officer in order that desired information be derived and displayed. The required interactive nature of the system constitutes a key element in determining its effectiveness in ship response control. It is not presently possible to automatically sense and assess all factors of importance to ship response control. Similarly, it is neither reasonable nor desirable that the judgment factors in ship operational decision-making be preprogrammed and used automatically for such purposes. The abilities and responsibilities of ship's officers are most important ingredients to proper ship operation. The flexibility of a sound system would be seriously impaired if they were not utilized effectively.

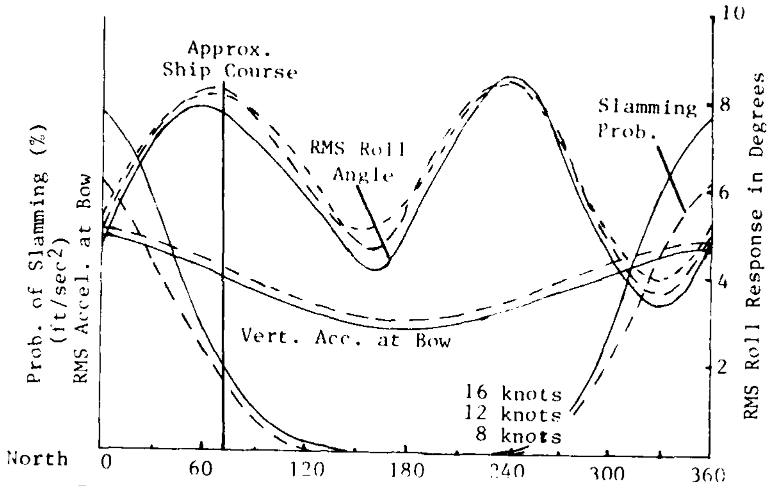

Figure 5. Variations of vertical acceleration at the bow, slamming probability and roll responses due to changes in heading and speed.

will improve if the course change is sufficient.

With the basic characteristics of a ship defined while in its design stage, only minor changes can be made during its operating life, such as result in specific draft, trim, GM, etc. and therefore ship dynamic and quasistatic response characteristics can be computed initially for expected conditions of ship loading and be provided onboard as basic ship response information such as in Figure 6. It has been suggested by some that this type information should be provided onboard ship in booklets of charts and tables. However, when operating at sea in heavy weather, there is seldom time or inclination to search through dusty pages of ship response information to find that which may be useful in solving a seen response problem. Further, depending upon the characteristics of the sea, whether in the Mediterranean, Atlantic or Pacific, differences in sea state may require more specific treatment of ship response than fixed charts and graphs can conveniently include. Thus there is a need for onboard computation and evaluation of response variation, quickly and reliably made, for use in operational decision-making.

Step 4 suggests that multiple options need to be evaluated before it is possible to select an optimum action. To do this based solely on the exercise of judgment would require an inordinate amount of time. Experience has shown that 10-15 minutes of operation in a particular condition is required before a reliable assessment of response can be made. Since a sea state can change significantly in a two hour time span, trial and error assessment of the optimum course of action seems highly unrealistic as a basis for ship response control. If a task group is operating in a prescribed ocean area, the problem is complicated still further by the differing characteristics of the various ship types. In such situations, a trial and error solution is totally unacceptable for efficient operations.

On the basis of the above, it seems clear that a suitable system for ship response control must be capable of predicting the response changes to be expected under extant conditions, given a reasonable selection of speed and course changes. This implies that not only the ship's characteristics, but also those of the sea be modelled so that on-generally too complex to be solved through experience alone. In any case, a solution is not usually recognized until after it has been tried and under test for a significant period of time. This is a costly, inefficient and often ineffective approach to ship response control problems. The proper solution requires the following steps be carried out:

1. Identification of the response problem area.
2. Establishment of the severity of response.
3. Identification of corrective measures.
4. Determination of the most favorable measure for problem solution.

Step 1 is frequently, though not always, obvious to shipboard personnel. The rigid body responses, of pitch, roll and yaw, are most readily judged to be serious by an observer; structural responses such as hull girder torsion and flexure must be identified by adequate instrumentation. Step 2, however, requires a consistent measure of response and a bench mark against which the measurement can be evaluated. The measure must also be meaningful in a comparative sense, not a transient event which may not be repeated. Thus, the data on a particular response of interest must be analyzed and quantified as a stable and meaningful measure.

Since, as noted earlier, ship responses are random in their occurrence, the measures must be statistically meaningful and reliable. They must be displayed or made available to an observer in such a manner that not only the value but also its importance is clearly appreciated.

Steps 1 and 2 can be readily satisfied by data display in analog form such as with dial gages having levels of severity marked in colors or with the use of alarm devices. Such devices, however, may be of little use beyond the identification of problem areas and in a complex operation it is conceivable that a large number of gages, colors or alarms may be required.

Step 3, the identification of suitable corrective actions, is operationally restricted to four feasible options; changing vessel course, changing vessel speed, changing vessel load condition in respect to transverse stability or in respect to longitudinal weight distribution. Only the first two options, changing vessel speed and course can usually be accomplished in a short time period and they are therefore of primary concern here. The latter two would generally be limited to the taking on, shifting or discharging of fuel, water or ballast and cannot be considered as valid exercises for short term heavy weather response control, even though they may have importance in determining the basic response characteristics of a ship and in changing the frequency of slamming or shipping of water at the bow.

Changes of ship speed and course in a seaway produce effects on the responses of a ship which cannot be precisely determined by means of experienced judgment as such actions basically alter the magnitude and encounter frequency of seaway loadings. Since response characteristics differ for each mode, such changes will induce response variations, the magnitudes of which will be different for each mode. Thus, improving one response can result in another becoming serious. Consider the case illustrated in Figure 5. Here the ship has excessive motions in pitch and roll, speed change has little or no effect on either while course change in one direction will improve roll response at the expense of pitch, yet in the other direction both roll and pitch

All of these stress variations will be superimposed, one upon another at a given time. However, through signal filtering, the hull dynamic stresses can be separated out for separate event counting and magnitude assessment, the remaining signal can then be digitized and analyzed for statistical properties such as the mean stress, the significant variation of stress about the mean, the maximum and minimum values in a given sample period, etc. This information can then be displayed in a suitable format on a CRT or teletypewriter for use by the ship's officers in making comparative response evaluations for their operational decision-making.

The response of a ship to seaway loads is random in nature, just as the wave systems which excite them. Thus, knowing the exact state of response at a given time is of limited value since a decision to carry out an operation cannot be executed at the level of response identified. If, however, one defines the responses of concern in statistical terms, the level of response under given conditions can be compared with a value of recognized importance and they may be used meaningfully for decision-making since the risk that an excessive specific response will occur during or after an operation execution can be predicted and judged as within acceptable limits or not prior to execution. Through in-service experience, safe levels of response can be established for particular ship systems and thereby become available generally for fleet wide use in particular operations. For example, in replenishment exercises the rolling and yawing motions of replenishing ships are of serious concern. With a suitable sensor package, the motions can be sensed, analyzed and response level statistics compared with qualitative evaluations of operational performance. From such evaluations, acceptable levels of response can be established and later used by all similar ships for such operations. The data can be of further use for generalization of experience and design guidance for new ship systems.

From the above, it seems clear that reliable onboard decision-making requires consistent acquisition of data, its analysis and presentation in meaningful measures of performance, the setting of operational limits for responses of concern and the recording of results of operational changes. A system suitable for use in satisfying these needs can significantly enhance operational effectiveness and safety during heavy weather operations. The selection of suitable sensors and locations for their installation is important to the development of a reliable system. The sensing of critical parameters will allow the minimization of system complexity and maximization of system reliability and utility. Thus, for a given ship system, careful identification of important responses is needed; locations of maximum measurement sensitivity should be sought for greatest system effectiveness.

SHIP RESPONSE PREDICTION

As noted earlier, a ship operating in high sea states may experience ship responses which limit its ability to operate efficiently and carry out an assigned mission, safely or otherwise. If the ship is fitted with response sensors, an onboard data analysis system and information display devices, there should be ample confirmation of the seriousness of the situation. The ship commander, however, is left with only a clear statement of his problem and a challenge to extricate himself and the ship, if he can. An experienced commander may be consistently able to do so, if conditions aren't too severe, but in general a trial and error solution would be required with no guarantees of success. In multiple ship operations, ship response problems are

System Installation	$ 17,500
Ship Response Characteristics	10,000
Basic Data Analysis Console and Peripheral Devices	50,000
Initial Software Development	25,000
	$102,500
Contingincies	7,500
Total Cost	$110,000

For multiple ships of a single class there would be reductions possible in the amount of perhaps $30,000 for 2nd and later installations. For multiple production of individualized, non-class ship systems an approximate $20,000 reduction can be expected. For ultimate system definition where system configuration is well defined, microprocessors will possibly replace the mini-computers with a further cost reduction of $5-15,000. However, in such an event the added flexibility of the mini-computer based system will be compromised.

How effective can such systems be in improving heavy weather performance? Can the improved performance justify the costs of such systems? In voyage analysis it has been found that it is typical practice to travel about 5% further on a North Atlantic crossing in an attempt to avoid unfavorable sea states. This amounts to 150-250 miles for many Northern Hemisphere trade routes. At $1,500 per hour for a 30,000 DWT vessel at 20 kts. this means $11,000-$18,000 per passage. If vessels still encounter heavy weather somewhere enroute on half their passages, as is often the case, system costs can be recovered within less than two years. In regard to heavy weather damage to cargo, 5% of revenue for a 30,000 DWT dry cargo vessel can amount of $500,000 in a year. Reducing such losses for one ship to the 1% that is normal, could result in recovery of system costs for a six ship fleet in less than two years. Between these two possibilities there is ample opportunity for short term payout on a properly configured and utilized commercial system. While it is true that all of the above represents rather casual estimates, it seems quite clear that potential savings are large in relation to costs and in multi-ship operations, improved performance can simultaneously and significantly enhance fleet effectiveness and maintenance of fleet schedules, reducing revenue losses thereby as shippers are better served and service reliability is established. A case by case study of proposed installations would be necessary if a more definitive evaluation of cost effectiveness is required.

The cost benefits of the HELM system are much easier to demonstrate. Though the cost of this system is generally somewhat greater the daily rate of typical crane vessels is such that saving just a few days of on-site operating time a year is enough to more than offset the initial cost of a system. Since typical yearly down time for such vessels in an environment such as the North Sea or the east coast of the U.S.A. is nearly 180 days the ability to salvage any of these lost days could no doubt effect the economics of the operation significantly. It should be noted that the cost benefit in the case of offshore construction is not just with respect to the daily rate of the vessel, but it also effects significantly the starting date of production which is often the more critical factor of the two since if a well is commissioned by the end of the summer, a whole six months of production can be achieved which otherwise would be delayed due to interruption in work during the winter.

FUTURE WORK

While the experience with shipboard response monitoring and guidance systems is growing, the supporting technology is becoming more useful as well. The introduction of satellite communication systems for ships at sea (32) greatly enhances the ability to provide reliable shoreside support for at-sea operations. The refinement and expansion of the U.S. Navy's Spectral Ocean Wave Model to a global capability similarly extends the range of at-sea operations that can be supported.

Further, since shoreside computation is still a best assessment effort, feedback from on site sensors as confirmation or correction data will provide a closing of the loop in the total system. Additional development of algorithms for wave forecasting definition of ship seakeeping characteristics, inclusion of consideration of voyage costs, constraints and objectives will allow the implementation of system operation optimization (33).

From the above it would seem that all one has to do is go out and install a suitable system and improve ship operation performance. Such is not quite the case, however, as much of what has been discussed above needs better definition of information before full effectiveness can be realized. In regard to response sensors and data analysis other than waves, the present technology can be used with confidence. The signal processing, filtering, digitizing, statistical and spectral analysis, data storage, etc. represent state-of-the-art capabilities that have and can be used freely. The setting of sound operational limits, the validation of prediction and guidance models, the determination of the most suitable means for information display and interaction with ship's officers all require further study. However, it is not yet clear that any particular format is better than another from a human engineering point of view or that the appropriate presentation and interpretation of data should be in scientific, seaman's or some new set of terms.

For ship routing and response control, it is necessary that there be confirmation of sea state prediction beyond the present capability. In high sea states, non-linearities may become more important than presently found to be the case. There is a need for interpolation of spectra between grid points of the SOWM, or grid point refinement, and ship data feedback may be highly valuable in such a development.

It is anticipated that ships at sea will seldom be in locations where sea conditions will be independently known or predicted by SOWM. Thus, there will most likely be discrepancies between the sea states ships at sea are advised to expect and those they will experience. In order to build confidence between those at sea and those ashore, a sound communication link will be needed, one which allows reliable data obtained at sea to be send ashore and vice versa for data bank updates and use for supporting assessments of sea conditions. With satellite communication system which can be used for clear channel, high speed data transmission, such possibilities are not far from realization.

One can see in the above the possibilities for a greatly improved system of ship operational control. With a shore-based weather and seastate forecasting station having in its data files the important seakeeping and response characteristics of fleet elements, operational objectives and data provided by the fleet managers, vessel route planning and progress monitoring can be accomplished ashore through ship to shore satellite data links. With suitable shipboard instrumentation, response data could be sent ashore along with other ship operational data such as speed, course, position, etc. Through routine progress update and route reanalysis, periodic guidance could be transmitted for optimum achievement of voyage objectives.

Though the critical decisions would and should continue to be made onboard, the close guidance possible through such linkage would allow the power of a central shore-based facility to be readily available for improving at sea operations with enhanced productivity, safety and efficiency in the fleet.

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ADAPTIVE AUTOMATIC COURSE-KEEPING CONTROL OF A SUPERTANKER AND A CONTAINERSHIP- A SIMULATION STUDY

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ABSTRACT

Many aspects of the dynamic behaviour of ships, and consequently many problems concerning their control, require analysis by means of experiments and simulation. The paper presents the results of a simulation study on adaptive control, carried out by the Italian Laboratorio per l'Automazione Navale together with the Dutch TNO-Institute for Mechanical Constructions.
The L.A.N. has acquired much experience in the field of identification and adaptive control of ships during the last ten years, while the Dutch Institute has developed and used a number of simulations-models of different ships.
The simulation study covered the en-route course keeping under widely varying weather conditions. The ships to be controlled were a supertanker in loaded and in ballast condition and a containership.
Simulation is a very powerful tool, because of the great flexibility and saving of time, compared with sea trials.
The performance of the adaptive control system for course-keeping in deep water looks promising, also in rough weather. Further development may lead to the practical application of modern control theory on board in order to ensure safe and efficient operation of modern ships.

INTRODUCTION

State of the Art

During the last years a relevant amount of work has been developed all over the world in order to study, experiment and put into practice different types of ship autopilots designed according to modern criteria of advanced control theory techniques.

Much has been published on the subject in many countries, and comprehensive reports are available in the Proceedings of the last International Symposia on Ship Automation:(1), (2), (3), (4).
It must be pointed out, however, that a complete and fully satisfactory solution has not yet been found for the problem. The possible solutions are indeed numerous and, besides economic considerations, depend on the type of considered ships and their operational characteristics, on the available measuring equipment, and on the level of integration of the automatic steering with other bridge area functions. For a general analysis of the problems connected with the ship automatic guidance control system implementation by means of an onboard digital computer, we suggest (5).

From the point of view of industrial realization, let us observe that actually a great number of steering devices is commercially available, either consisting of independent autopilots: ANSCHUTZ, ATEW, DECCA, SPERRY, etc., or as an integrated function of more complete navigation systems: KOCKUMS, IBM MABS, TONAC, etc.

The performance of such autopilots during real operation at sea has generally been advantageous, on account of the obtainable reduction in cruising time and propulsion power.

Yet, it must be pointed out that the performance of conventional autopilots has been very far from the one theoretically obtainable by optimal steering. This discrepancy is mainly due to the impossibility for conventional autopilots of an automatic regulation with respect to the frequent variations of the environmental conditions (sea waves, wind, currents, etc.), as well as of the ship's dynamic behaviour (speed, draught, loading conditions, etc.).
In fact, even if such autopilots are all provided with certain "weather adjusters", designed in order to obtain a sort of "empirical adaption" with respect to the changing characteristics of the disturbances, the adjustment procedure is actually carried out by human operators who manipulate the various control knobs according to their subjective evaluation rather than to a real knowledge of the controlled process.

This explains the growing interest towards adaptive autopilots, because such autopilots need less or no human interference at all.

In principle, adaptive control systems might be schematically diveded into two groups: To the first one belong those systems which adapt the controller's structure or their parameters according to a predetermined open-loop scheme.
Such regulators are generally constituted by three-terms controllers of the P.I.D. type, capable of automatically adjusting the control-loop coefficienst as a function of some measures of the external disturbances. This is the case, for example, of the autopilots proposed by OLDENBURG (6), and SCHILLING (7).

These open-loop adaptive autopilots, however, although they may show a slight improvement with respect to conventional ones, can not yet be considered a completetely satisfactory solution to the automatic ship steering problem, since they work according to a fixed program which contains unavoidable errors and reductive assumptions and therefore cannot adequately treat unscheduled situations.

The second class of adaptive autopilots is constituted by those regulators designed according to modern adaptive and stochastic control techniques, on the basis of mathematical models of the ship steering dynamics and of the external environment. This class which, at least in principle, is rather wide, must be drastically limited if we introduce some realizability constraints for the resultant onboard computer by which the adaptive autopilot is implemented.

We can mention, for example. the research of VAN AMERONGEN and UDINK TEN CATE (8) dedicated to the implementation of an adaptive autopilot by means of model reference techniques.
Two different types of autopilots have been widely investigated by simulations and fullscale experiments by KÄLLSTRÖM and ÅSTRÖM (9) based on self-tuning of the parameters and control by means of a minimum variance control law.
The simple system uses only heading measurements, while the more complicated one is provided with a Kalman filtering of heading, yaw rate and sway velocity.
A new type of autopilot has been recently proposed by OHTSU and HORIGOME (10), which relies on a stochastic multivariable autoregressive model, through which not only the yawing motion but also rolling, pitching and lateral acceleration are considered.

Scope of the Study

The simulation study which is the subject of the present paper is the result of a joint-research between the Italian Laboratorio per l'Automazione Navale (L.A.N.) and the Dutch TNO - Institute for Mechanical Constructions (abbreviation in Dutch: TNO - IWECO) devoted to the adaptive control of the ship's steering process.

In the recent years the L.A.N. has studied the particular field of experimental identification and adaptive control of the ship steering dynamics (11), (12). A large number of experiments have been carried out, since 1974, on board the oceanographic vessel "Bannock", with the purpose of testing the performance of computerized adaptive autopilots.
After the first promising results, the need has been felt of extending them to different and more important classes of ships, as well as to different operational conditions.
TNO-IWECO has acquired, with the realization and during the operation of the Delft Ship Manoeuvring Simulator, a very wide experience in the simulation of ships having different characteristics and navigating in different situations of wind and currents, (19), (20).

The complementary experience of the L.A.N. and TNO-IWECO in their respective fields has been the starting-point for the present study.
The fundamental purpose of the joint research is to compare different procedures of adaptive control for the following cases:
- course-keeping
- course-changing
- track-keeping

The study is not yet finished, since only the course keeping in deep water has been analysed for three different ships. The results obtained during this first phase, however, look promising and indicate that the performance of the considered control system is very good. To get an idea about the relative performance, a comparison has been made with an "ad hoc" designed PID-autopilot.

MATHEMATICAL MODEL OF SHIPS

Co-ordinate systems

The ship path is referred to an earth fixed right hand orthogonal co-ordinate system 0, x_0, y_0, z_0. The ship motion is described relative to a ship fixed orthogonal co-ordinate system c.g., x, y, z. The origin of this axis system coincides with the centre of gravity of the ship. The positive directions of the principal axes are indicated in figure 1.
All symbols used in this report are explained in the list of symbols at the end of this paper, just before the Appendix.

Kinematic relationships

The kinematic relations convert the ship velocity relative to the water $\underline{U}$ into the corresponding velocity relative to the ground $\underline{V}$, taking into account the ocean-current velocity $\underline{V}_c$, which is assumed constant: $\underline{V} = \underline{U} + \underline{V}_c$.

The components $(\dot{x}_0, \dot{y}_0)$ of $\underline{V}$ in the direction of the x_0-, y_0-axes are computed from the components of $\underline{U}$ and $\underline{V}_c$:

$$\begin{aligned} \dot{x}_0 &= u \cos\psi - v \sin\psi + V_c \cos\psi_c \\ \dot{y}_0 &= u \sin\psi + v \cos\psi + V_c \sin\psi_c \end{aligned} \qquad (1)$$

The third kinematic equation relates the yawing motion ($\dot{\psi}$) about the z_0-axis to the yawing motion (r) about the ships vertical axis:
$\dot{\psi} = r$

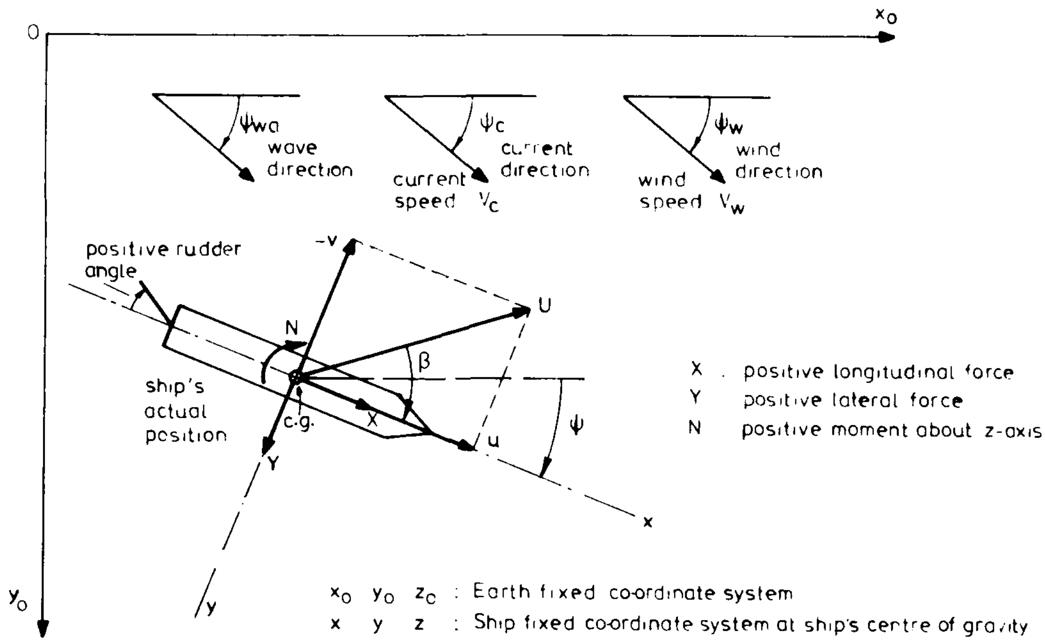


Figure 1: Co-ordinate systems and sign convention.

Equations of Motion

The low frequency equations of motion relative to the moving or Eulerian axes are, the ship being considered as a rigid body, for the three degrees of freedom which are relevant for manoeuvring:

$$\begin{aligned} X &= m(\dot{u} - rv) \\ Y &= m(\dot{v} + ru) \\ N &= I_z \dot{r} \end{aligned} \tag{2}$$

The forces and the moment exerted on the ship can be split up into contributions of the ships hull (without propeller, without rudder) cf the propeller, of the rudder, of waves, and of the superstructure due to the wind.
Hence:

$$\begin{aligned} m(\dot{u} - rv) &= X = X_{hull} + X_{prop} + X_{rud} + X_{waves} + X_{wind} \\ m(\dot{v} + ru) &= Y = Y_{hull} + Y_{prop} + Y_{rud} + Y_{waves} + Y_{wind} \\ I_z \dot{r} &= N = N_{hull} + N_{prop} + N_{rud} + N_{waves} + N_{wind} \end{aligned} \tag{3}$$

A description of the contributing forces and moments is included in Appendix A, both for the tanker and for the containership.

Data of the supertanker, fully laden and in ballast, and of the containership are presented in the following table.

Table 1 Ship Data

SIMULATED SHIP	SUPERTANKER (FULLY LADEN)	SUPERTANKER (BALLAST)	2ND GENERATION CONTAINERSHIP
Ship dimensions			
lenght	315 m	315 m	220 m
beam	48 m	48 m	30.5 m
draught fore	19 m	7 m	10.04 m
draught midship	19 m	9 m	10.62 m
draught after	19 m	11 m	11.20 m
displacement	250,000 tons	110,000 tons	42,984 tons
Propeller			
type	fixed, righthanded	fixed, righthanded	fixed, righthanded
Propulsion			
type	turbine	turbine	turbine
S.H.P.	30,000 S.H.P.	30,000 S.H.P.	32,450 S.H.P.
Nominal service speed	14.5 kts	16.5 kts	22 kts

Steering Gear

The steering gear or rudder control system is basically a hydraulic servo system, which behaviour can be described by

$$\dot{K} = \frac{1}{T_k} (K_r - K) \tag{4}$$

in which K_r and K are the required and actual percentage of servo valve opening. respectively. T_k is a time constant, while the obtainable value of k is limited. The rudder rate $\dot{\delta}$ is determined by the servo valve opening k, while the maximum rudder rate ($\dot{\delta}_{max}$) occurs if the servo valve is fully open (k = 100 %). So

$$\dot{\delta} = \dot{\delta}_{max} \cdot \frac{k}{100} \tag{5}$$

$\dot{\delta}_{max}$ equals 2.5 degree per second for the supertanker and 2.32 degree for the containership.

Turbine

The turbine is simulated in such a way that the percentage steam is regulated automatically in order to keep the engine's rpm constant.
The required rpm is set by the telegraph.

Performance Curves

Figures 2,3 and 4 show a few performance curves for the supertanker, fully laden and in ballast, and for the containership. The performance tests have been carried out on the Delft Ship Manoeuvring Simulator.

MATHEMATICAL MODEL OF THE ENVIRONMENT

A realistic simulation of the environment is considered very important. Therefore, the simulation model includes all the effects due to wind, waves and current.

Wind

Only the horizontal windvector-component is of interest. It can be split up in a part along the mean wind direction and a part perpendicular to it. The first part is composed of the mean wind speed and a randomly fluctuating part, being the longitudinal gust. The second part is the lateral gust.
The gust components are uncorrelated random phenomena, which follow the Gaussian probability distribution.

For simulation purposes, the power spectra of both the longitudinal and lateral gust are modelled by means of shaping filters with inputs from white noise generators. The power spectral density functions are obtained from literature: For the longitudinal gust the Van der Hoven - spectrum is used (21), for the lateral gust a spectrum is used according to (22).

Time series are generated, sampled and stored on disc and retrieved from disc during the simulation. The instantaneous windspeed is the sum of the mean speed and the longitudinal gust. The instantaneous winddirection is calculated as the sum of the mean wind direction plus the "directional" gust, being the lateral gust divided by the mean wind speed.
Figure 5 shows a record of the longitudinal and directional gust.

Waves

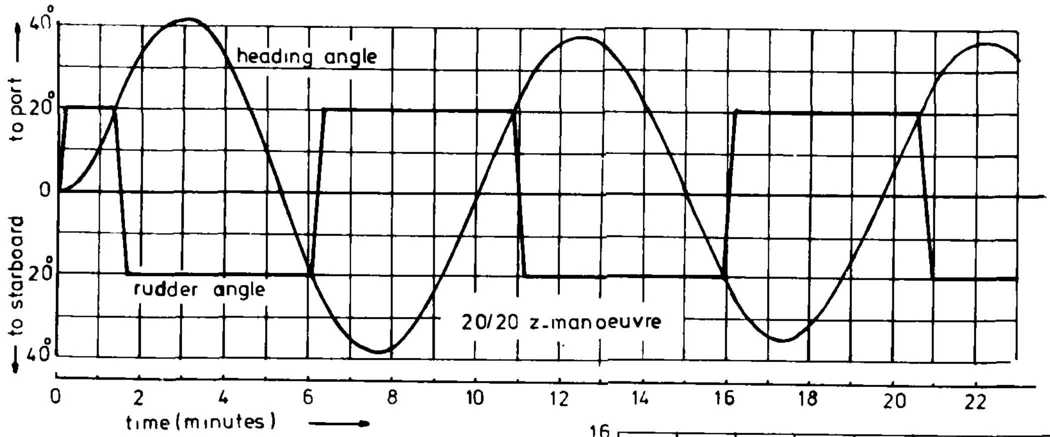
The effect of the waves (wave induced motions and wave drift forces) are computed off-line using special computer programmes.

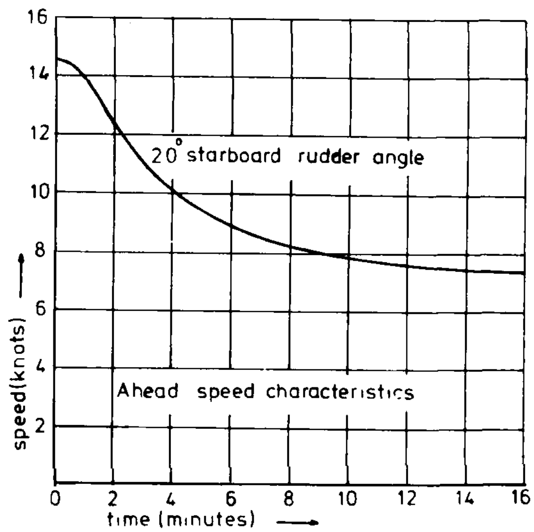
The "high frequency" motions induced by the waves are oscillatory with zero mean and with frequencies equal to the wave frequencies (roughly between .05 Hz and .35 Hz). At the same time the ship drifts off its position as the result of the second-order wave drift force.

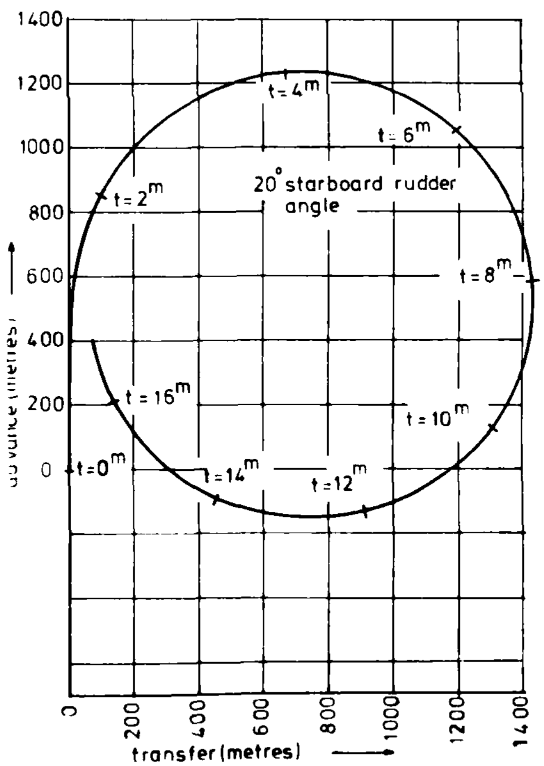
To calculate the motions in 6 degrees of freedom, linear coupled equations of motions are used of which the coefficients are frequency-dependent. (23)

As a result, transfer functions in regular waves and motion spectra in irregular waves are obtained. Also the mean wave driftforces and moment in regular and irregular waves are computed.

For simulation purposes, time series are needed of both the wave induced motions and the wave driftforces and moment in irregular waves.







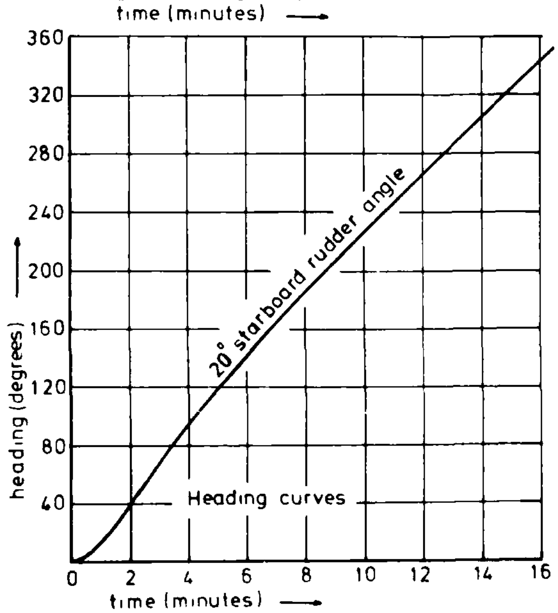
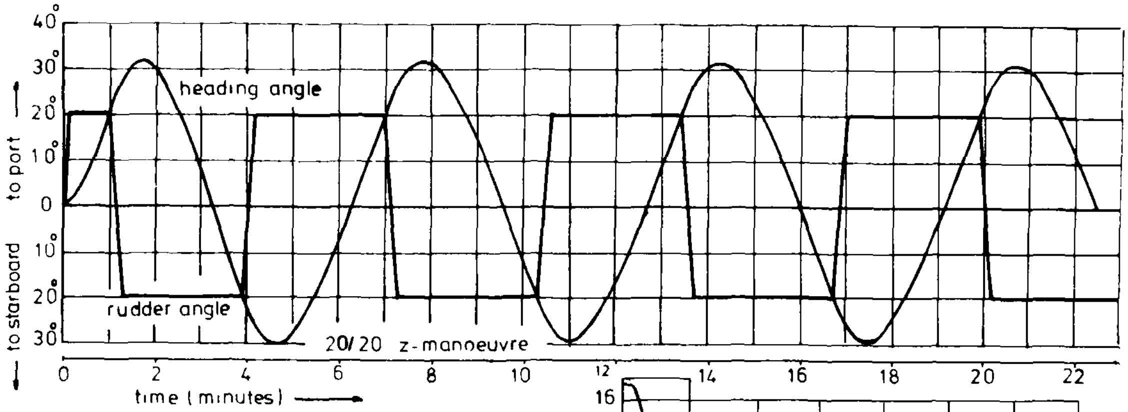


Figure 2: VLCC, fully laden:
zig-zag manoeuvre and
turning circle.



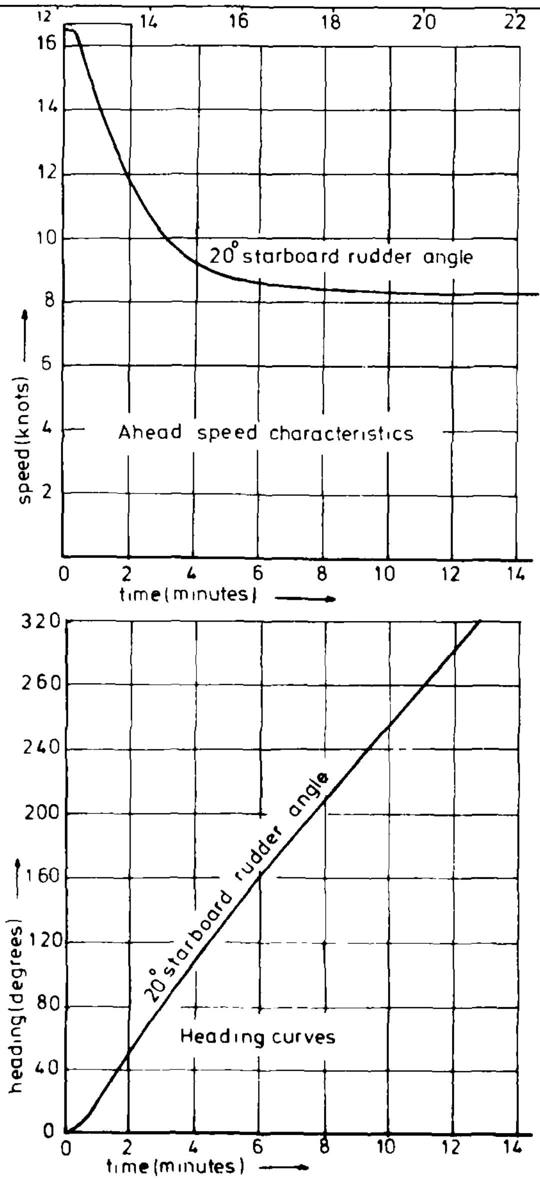
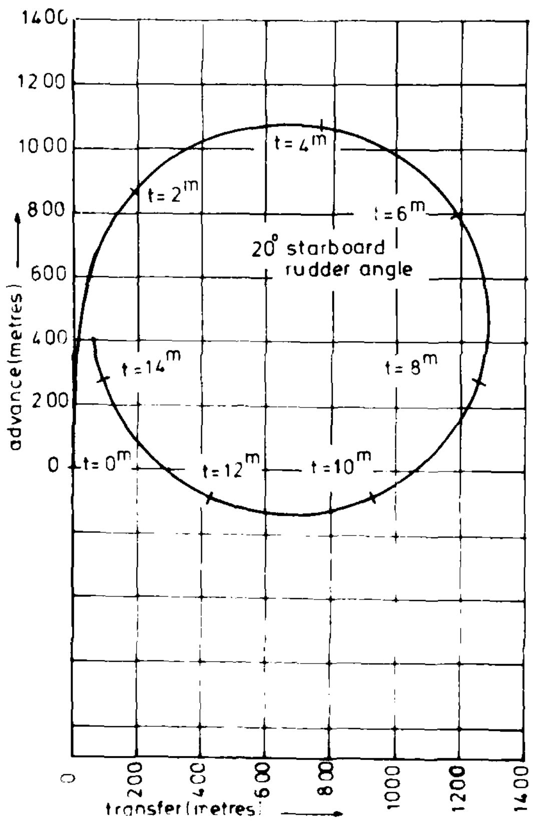
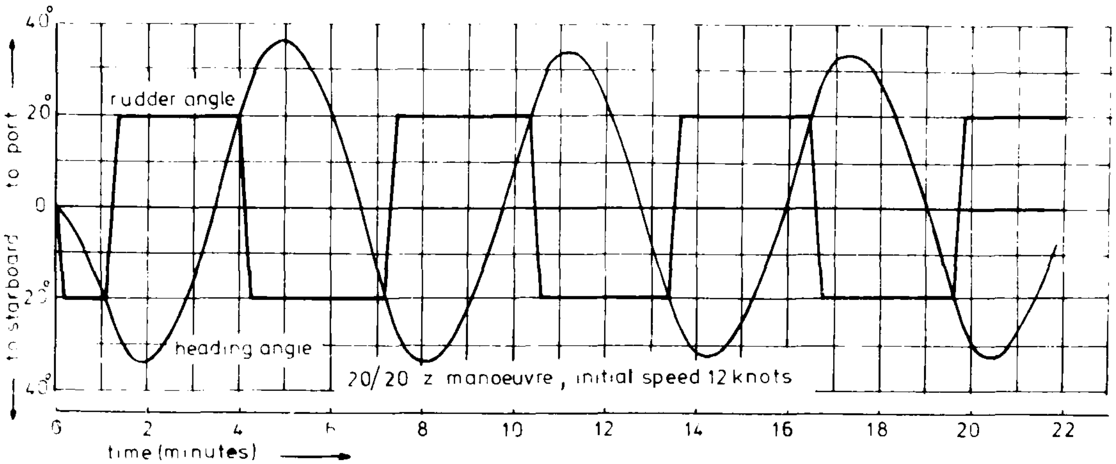


Figure 3: VLCC, ballast conditions: zig-zag manoeuvre and turning circle.





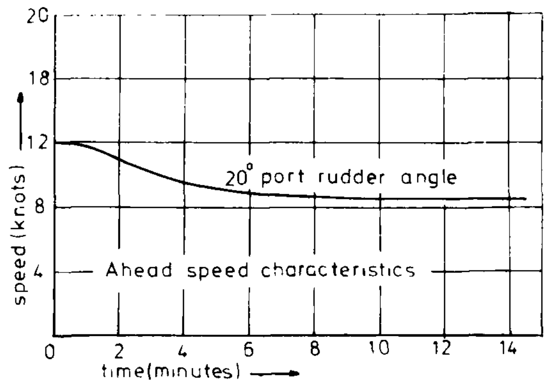
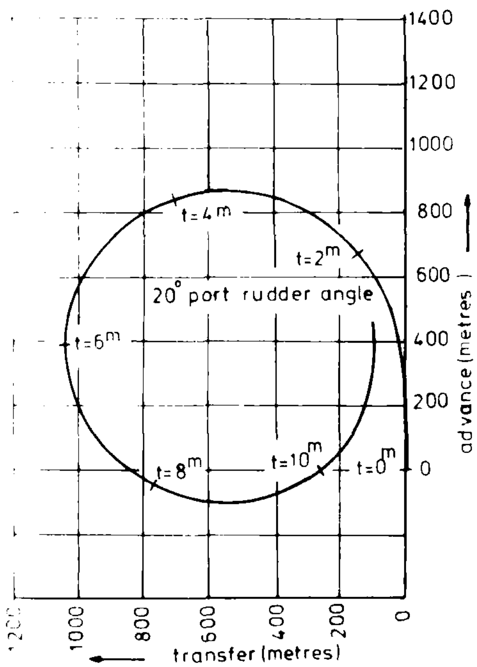
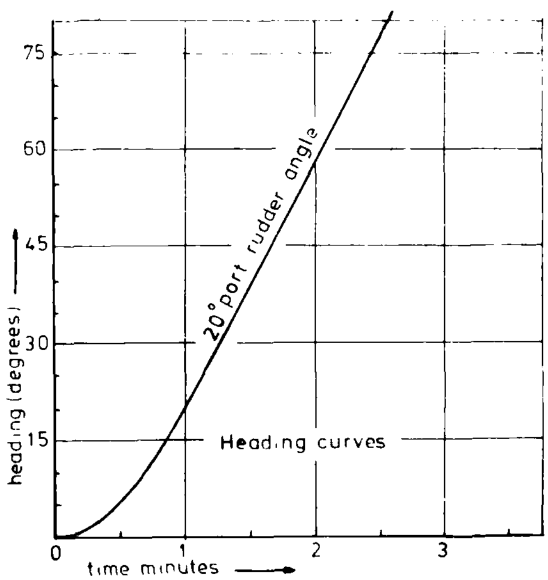


Figure 4: Second generation containership: zig-zag manoeuvre and turning circle.





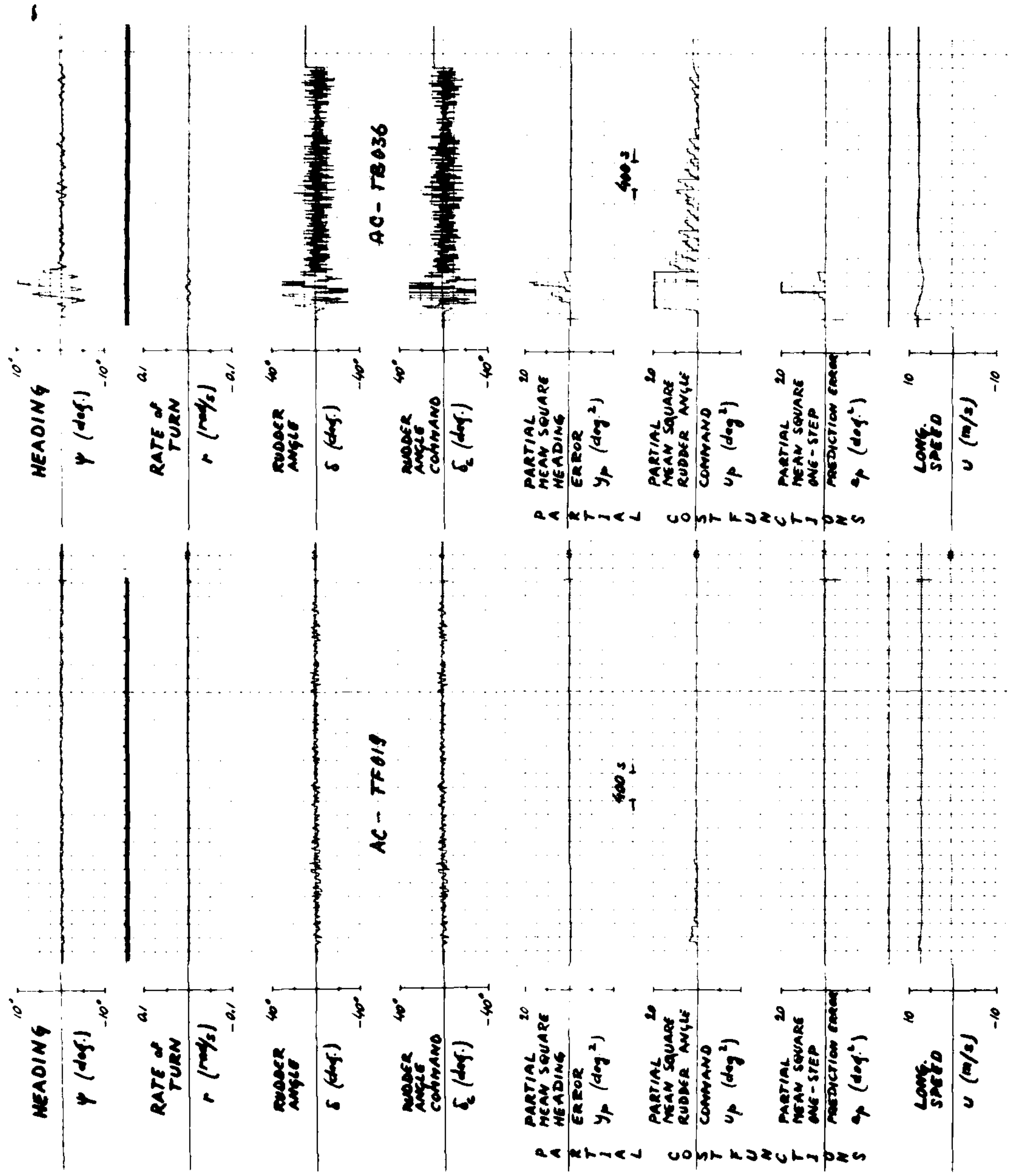


Figure 12: Time histories of runs AC - TF 019 (tanker, full) and AC - TB 036 (tanker, ballast).

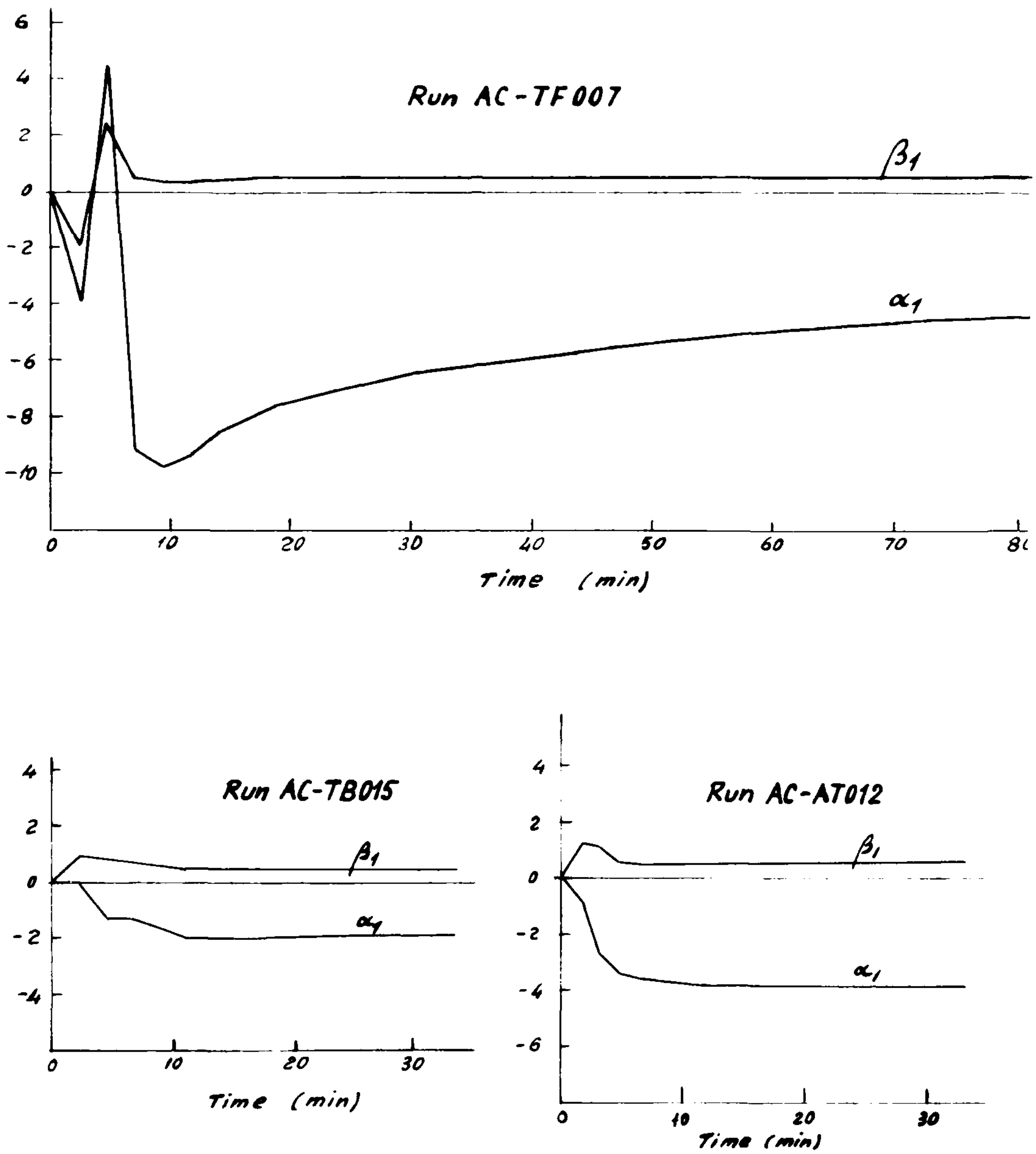


Figure 11: Rate of convergence of model coefficients α_1 and β_1. Runs AC - TF 007 (tanker, full), AC - TB 015 (tanker, ballast), AC - AT 012 (containership).

supertanker, fully laden

supertanker, ballast

containership

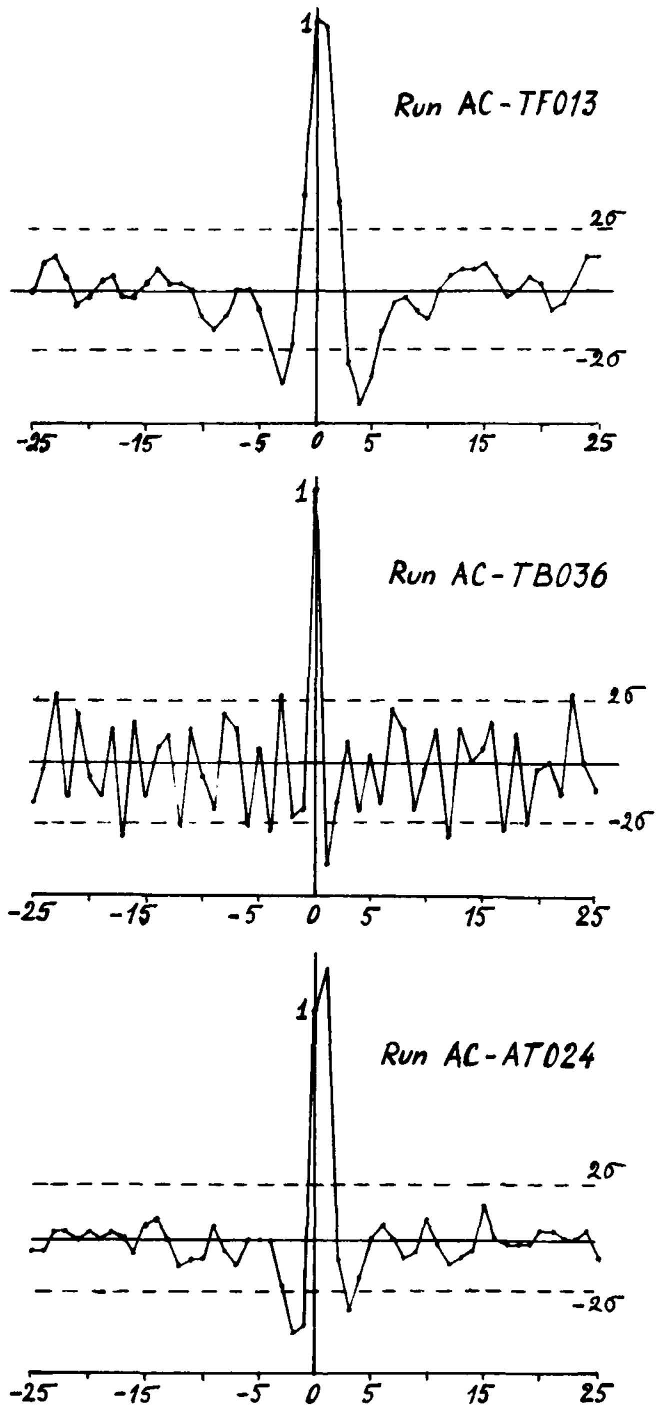


Figure 10: Cross-correlation function of systems input and output signal for the three considered ships.

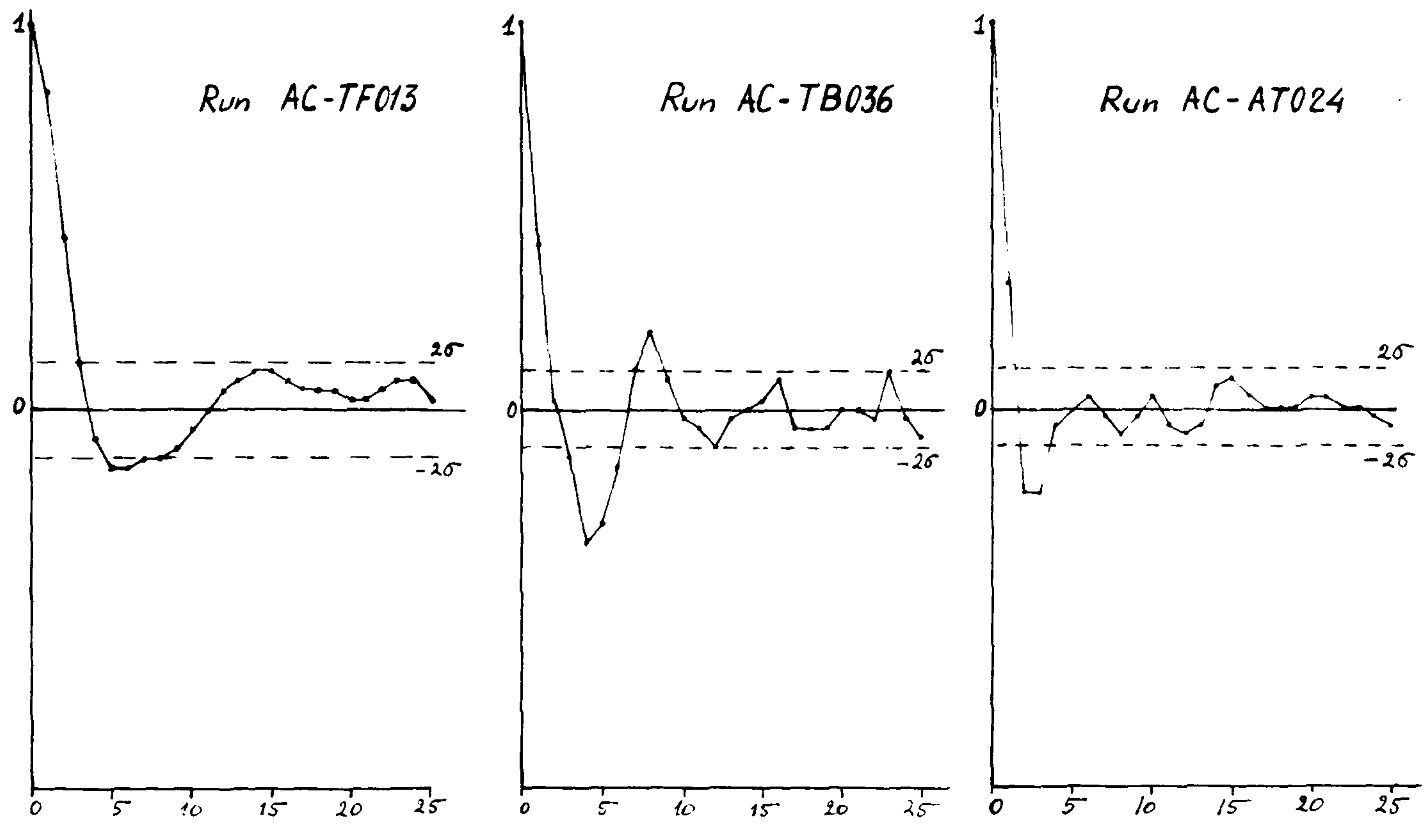


Figure 9: Auto-correlation function of the system output signal. From left to right: supertanker, full; supertanker, ballast; containership.

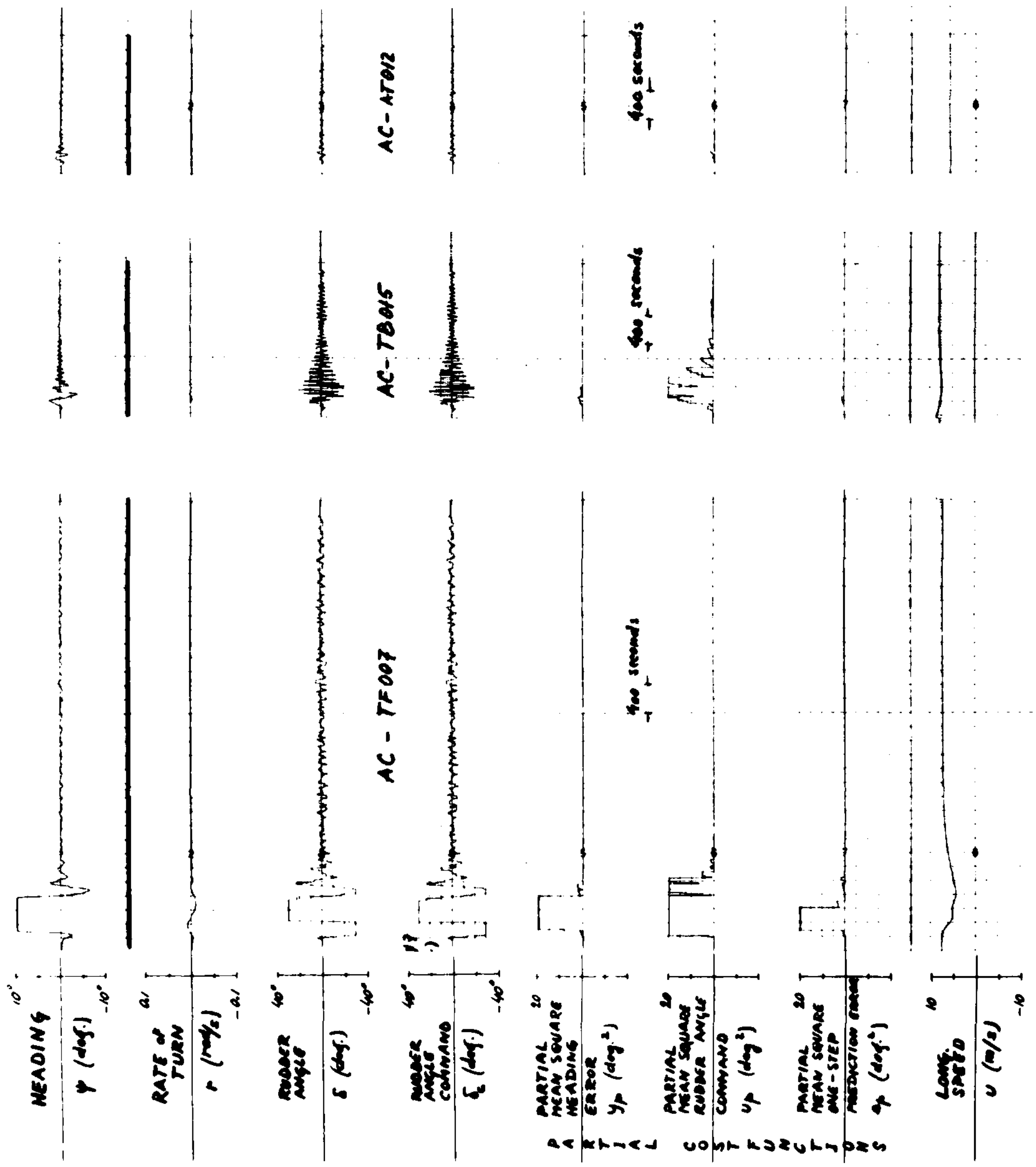


Figure 8: Time histories of runs AC - TF 007 (tanker, full), AC - TB 015 (tanker, ballast), and AC - AT 012 (containership).

RESULTS

Adaptive Control

The first task has been the tuning, for each considered ship, of the structural control system parameters (model order and time delay) in calm weather and for a given sampling time. Due to the limited amount of time, which was available for the simulation, no particular effort has been devoted to establish the optimal sampling time and gain factor β_o. This should be done in later stage.

The following table shows the results:

Table 3 Control system parameters

SHIP TYPE	Sampling time	Model order	Time delay
Supertanker, fully laden	14 seconds	4	3
Supertanker, ballast	12 seconds	4	2
Container-ship	10 seconds	3	1

In Figure 8 the time histories of the input-output variables are shown for the three considered ships. Also some performance cost functions associated with the algorithm are shown: the partial variance of the heading error, of the commanded rudder angle, and of the one-step prediction error. The analysis of the convergence of the adaptive controller is based on a necessary condition, which is stated in (16), according to which the convergence of the algorithm implies that the auto-covariance of the output $R_y(t)$ and the cross-covariance between the input and the output $R_{yu}(t)$ are zero for all the values of the lag t greater than the time delay in the process. This theorem has been used to determine if the parameters have been tuned properly.

Figures 9 and 10 present the corresponding sample autocorrelation and cross-correlation functions, which all comply with the zero condition within the confidence limits - indicated by the broken lines.

From Figure 11 it is possible to get an idea about the rate of convergence of the two first coefficients α_1 and β_1, starting from an initial condition of complete ignorance.

The behaviour of the three ships in varying weather conditions is presented in figure 12 and figure 13. The run conditions of the runs discussed here, are summarized in Table 4.

- Deep water
- No ocean current
- Varying wind- and wave conditions

Adaptive Controller. In the executed simulation programme, the following aspects have been considered
- Tuning of control system parameters under calm weather conditions, while the measured heading was contaminated with noise.
- Checking of control system parameters under moderate weather conditions, and adjustment of parameters, if necessary.
- Performance assesment under weather conditions varying from moderate up to severe.

PID-Controller. In the executed simulation programme, the following aspects have been considered:
- Tuning of control system parameters during a few runs under varying weather conditions, the measured signals (heading and speed) being contaminated with noise.
- Checking of the parameters and performance assessment under weather conditions, varying from calm up to severe. Adjustment of the heading control gain factor, but only in case significant improvement of the performance would result.

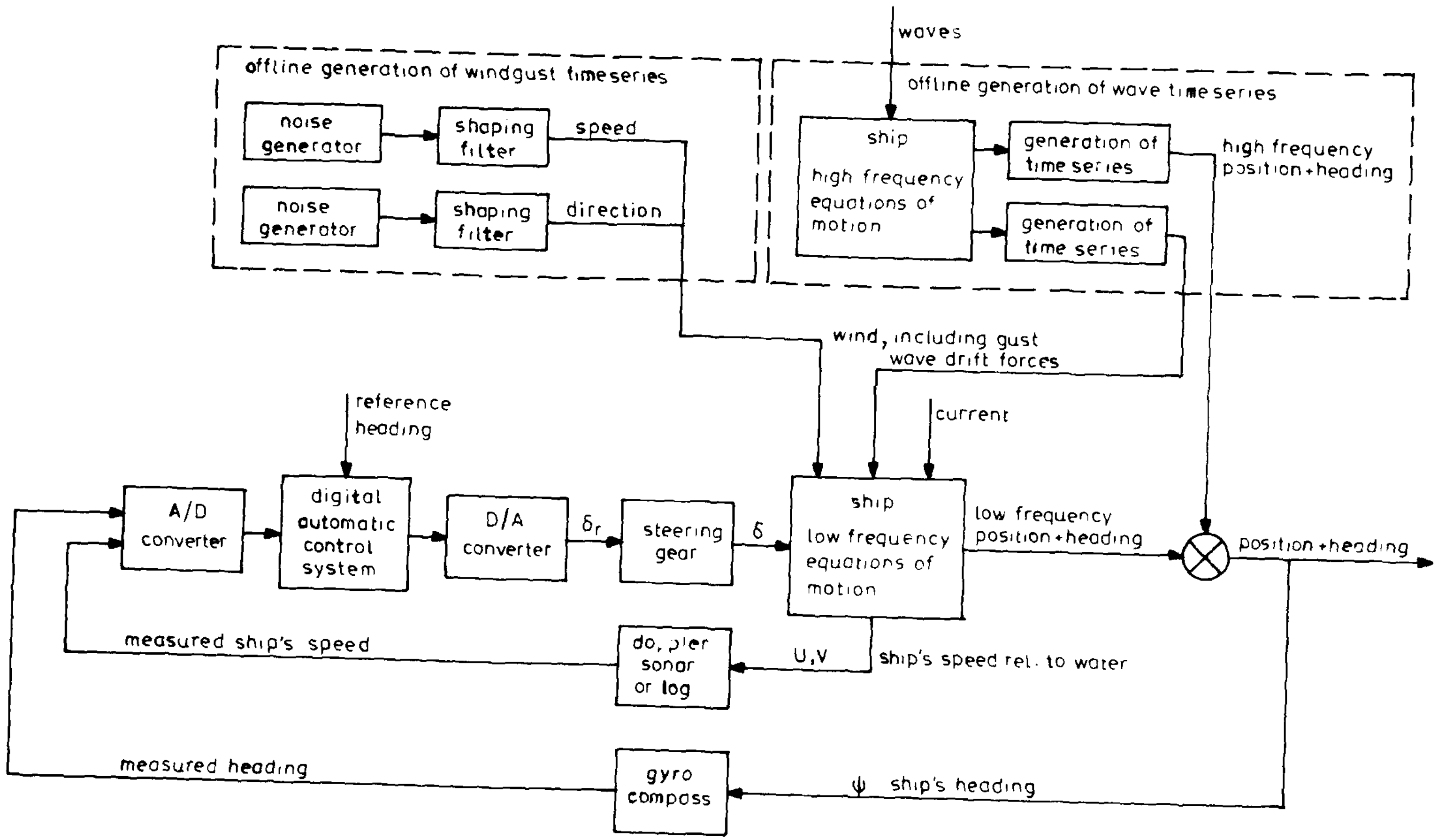
Identical environmental conditions have been selected for the ships under adaptive and under PID-control to guarantee a realistic comparison:

Weather Conditions. Apart from the calm weather-case, the following wind-waves conditions have been considered for each ship.

Table 2 Weather conditions

	WIND		WAVES	
	MEAN SPEED (m/s)	MEAN DIRECTION (deg.)	SIGN. WAVE HEIGHT (m)	DIRECTION (deg.)
WEATHER ON THE BOW	10 20	150 *) 180	4 8	150 150
WEATHER FROM ABEAM	10 10 20	90 90 90	4 6 8	90 90 90
WEATHER ON THE STERN	10 10 20	30 30 30	4 6 8	30 30 30

*) The direction of wind and waves is relative to the ship's longitudal axis:
0 degree = on the stern, 180 degree = on the bow.



NOTE : Measured speed is only used in case of PID - control

Figure 7: Block diagram of the simulation model

Therefore a 2nd-order derivative-filter has been applied, with a bounded gain beyond the frequency range of interest.

In parallel, an integrator takes care of the reduction of steady state heading errors. By limiting the input to the integrator and its output, overloading is avoided.

The gain factor $K_{p\psi}$ has been made a function of the speed $U_f = \sqrt{u_f^2+v_f^2}$, where the index f denotes that the measurements of the Doppler Sonar are smoothed by a low pass (1st-order) filter:

$$\text{For} \quad U_f < U_o \quad \rightarrow \quad K_{p\psi} = (K_{p\psi})_o$$
$$U_f > U_o \quad \rightarrow \quad K_{p\psi} = (K_{p\psi})_o \cdot \left(\frac{U_o}{U_f}\right)^2 \tag{11}$$

By adopting this function, the rudder efficiency is made largely independent from the ship's speed.

Summarizing, the PID-controller has the following transfer function:

$$\frac{\delta_c(z)}{\psi_m(z)} = - \left[\frac{\sum_{m=0}^{3} a_{fm} z^{-m}}{\sum_{n=0}^{3} b_{fn} z^{-n}}\right] * \left[K_{p\psi} \frac{\sum_{m=0}^{2} a_{dm} z^{-m}}{\sum_{n=0}^{2} b_{dn} z^{-n}} + K_{i\psi} \frac{1+z^{-1}}{1-z^{-1}}\right] \tag{12}$$

in which $K_{p\psi} = f(U)$

SIMULATION MODEL AND PROGRAMME OF EXPERIMENTS

Simulation Model

The general set-up of the simulation model is presented in figure 7.
The main blocks are:
- the controller and the measuring units (i.e. gyrocompass and dopplar-sonar)
- the steering gear
- the ship, including the steam turbine, and
- the invironment

The off-line part represents the generation of time-series of wave induced motions, wave drift forces and wind gust signals.

The individual blocks have been described separately. It should be noted that the measured ship's speed is used only for the PID controller. The measurement system is assumed to be a Dopplar-Sonar. However if such a system is not available on board, the speed signal as measured by the log could be used.

The computer which has been used for this simulation is an EAI Pacer 100 digital computer with a 32K 16-bits words memory. The computing time needed per cycle turned out to be 4-5 times less than real time. This is important as to limit the computer time as much as possible.

Programme of Experiments

General Conditions

- Straight course to be followed
- Speed at the start of each run equal to service speed

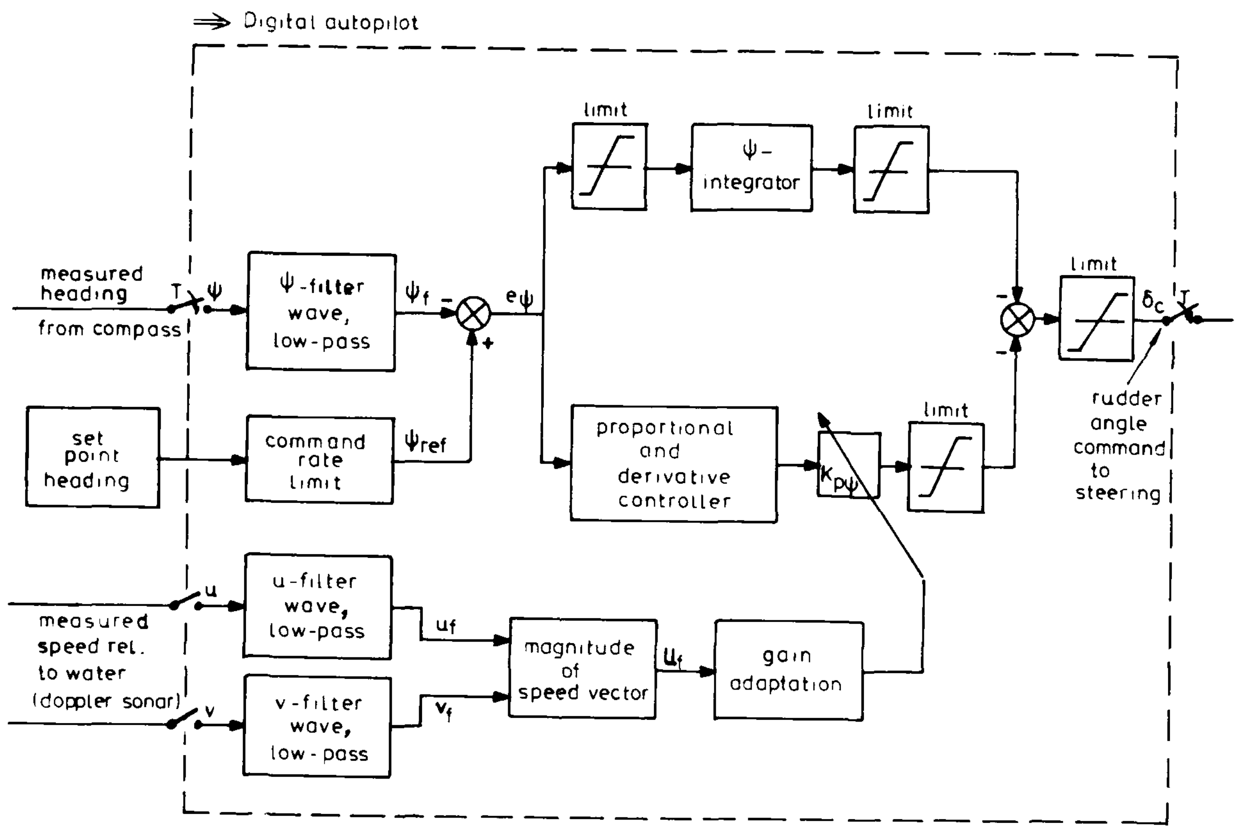


Figure 6: Structure of PID-type autopilot

the least squares estimator results in biased estimates of the actual model parameters. However, it can be shown that, if the parameter vector $[\alpha_1 \ldots \alpha \; \beta_1 \ldots \beta_1]$ used in the predictive model converge asymptotically for $t \to \infty$, then the self-tuning regulator will converge to the optimal one.

The most important properties, which must be analysed when applying self-tuning regulators to the optimal control of the course-keeping process are:
- overall stability of the closed-loop system
- numerical stability of the least squares estimation
- convergence of the regulator

There is a certain number of factors which influence the above properties. They can be summarized as:
(a) - Initial degree of ignorance about the actual system model:
It may be assumed to be represented by: model order n, time delay k, initial value of the parameters estimate and associated covariance matrix,
(b) - Gain factor β_0
(c) - Rate of exponential forgetting of past data in the estimation algorithm
(d) - Sampling time interval

These control systems parameters must be specified in advance to the adaptive control subroutine.

Since no a-priori knowledge of the ship's behaviour was assumed, it was necessary to tune the control system parameters in.
Initially, the parameters mentioned under (a) and (d) have been tuned during calm weather conditions. They have been subsequently checked under moderate to severe weather conditions and adjusted if necessary.

Digital PID - Controller

A PID - type course-keeping controller has been implemented to compare its performance with the performance of the adaptive autopilot.

The PID - type controller is not like the conventional autopilots. The controller parameters have been chosen with a good knowledge of the hydrodynamic characteristics of each individual ship. The aim was also to keep the need of adjusting the heading control gain factor under varying weather conditions at a minimum.

The control system sampling period was chosen equal to two seconds for all three ships.

The structure of the controller is illustrated in figure 6. The same controller structure has been applied to each ship.

The main functions are:
- low pass filtering of measured heading and speed
- calculation of the rate-limited heading command and of the heading error
- calculation of the rudder angle command

The rate-limited heading command procedure is only activated in case of course changes. In this study only course-keeping has been investigated, so the procedure has not been operational.

The low pass heading filter is a 3rd-order filter. It suppresses the influence of the wave induced yawing motion and of measurement noise.
Different transfer functions have been selected for the supertanker and containership.

It has been assumed that the measured rate-of-turn was not available.

navigation.
As shown by NORRBIN (14), the two main contributions to propulsion losses consist of the increase of resistance due to the periodic yawing of the ship about the straight course and of the added resistance caused by the rudder actions.
Consequently a cost function is considered, constituted by the weighed sum of the mean square values of the heading error and of the rudder angle:

$$\mathcal{J} = [\overline{y^2} + \lambda\overline{u^2}] \tag{8}$$

Starting from different considerations, the same cost function was previously proposed by MOTORA (15), in order to take into account the loss due to increase of the path length during the yawing motion.

The optimal control of a linear stochastic system like the one described by eq. (7), which involves the minimization of the quadratic cost function expressed by eq. (8), however, gives rise to excessive computational difficulties, even in the simple case $\lambda = 0$.
It seems therefore more convenient, for the practical implementation of the adaptive autopilot, to resort to suboptimal control strategies.

An important class of methods, called "self-tuning" regulators, has been developed in the recent years, (16), (17), by the Lund University, in order to minimize the variance of the output variable $y(t)$ of the A.R.M.A. model represented by (7).
Further modifications (18) of the basic self tuning method allow the minimization of a quadratic cost function of the type expressed by eq. (8).
During the first phase of the simulation study, however, we have concentrated our attention only on the simple case $\lambda = 0$.
As already said in the introduction, this case has also been successfully studied by KÄLLSTRÖM and ÅSTRÖM (9) on three different tankers.

According to this approach, the two functions of the self-tuning regulator are:
- On-line identification of the predictive model:

$$y(t) + \alpha_1 y(t-k-1) + \ldots + \alpha_n y(t-k-n) = \beta_0[u(t-k-1) + \beta_1 u(t-k-2) + \ldots\ldots + \beta_l u(t-k-l-1)] + \varepsilon(t) \tag{9}$$

Where $l = n+k-1$ and the disturbance $\varepsilon(t)$ is a moving average of order k of the noise $e(t)$.
This model can be obtained by substitution of a minimum variance control law in the system equation (7), and different from it, is suitable for on-line parameters estimation.

- Actuation of the minimium variance control law:

$$u(t) = \frac{1}{\beta_0}\,[\alpha_1 y(t) + \ldots\ldots + \alpha_n y(t-n+1) - \beta_1 u(t-1) - \ldots\ldots \beta_l u(t-l)] \tag{10}$$

In such a way, it is possible to realize a self-tuning regulator consisting, at each sampling time, of an estimator of the parameter vector $[\alpha_1 \ldots\ldots \alpha_n \beta_1 \quad \beta_l]$, followed by an actuation of the optimal control law.
The parameters estimation can be economically done by a recursive least square techniques using exponential weighing of past data (17).

The gain parameter β_0 is assumed to be constant, in order to avoid, during closed-loop operation, a correlation of the inputs with the disturbances $\varepsilon(t)$, which results in a biased parameter estimation.
In practical autopilot implementation, when large variations with respect to the cruising speed should be expected, it is convenient to regard β_0 as a scheduled function of the ship's speed.

A remarkable feature of the regulator described above is that, owing to the structural difference between the predictive model (9) and the original one (7),

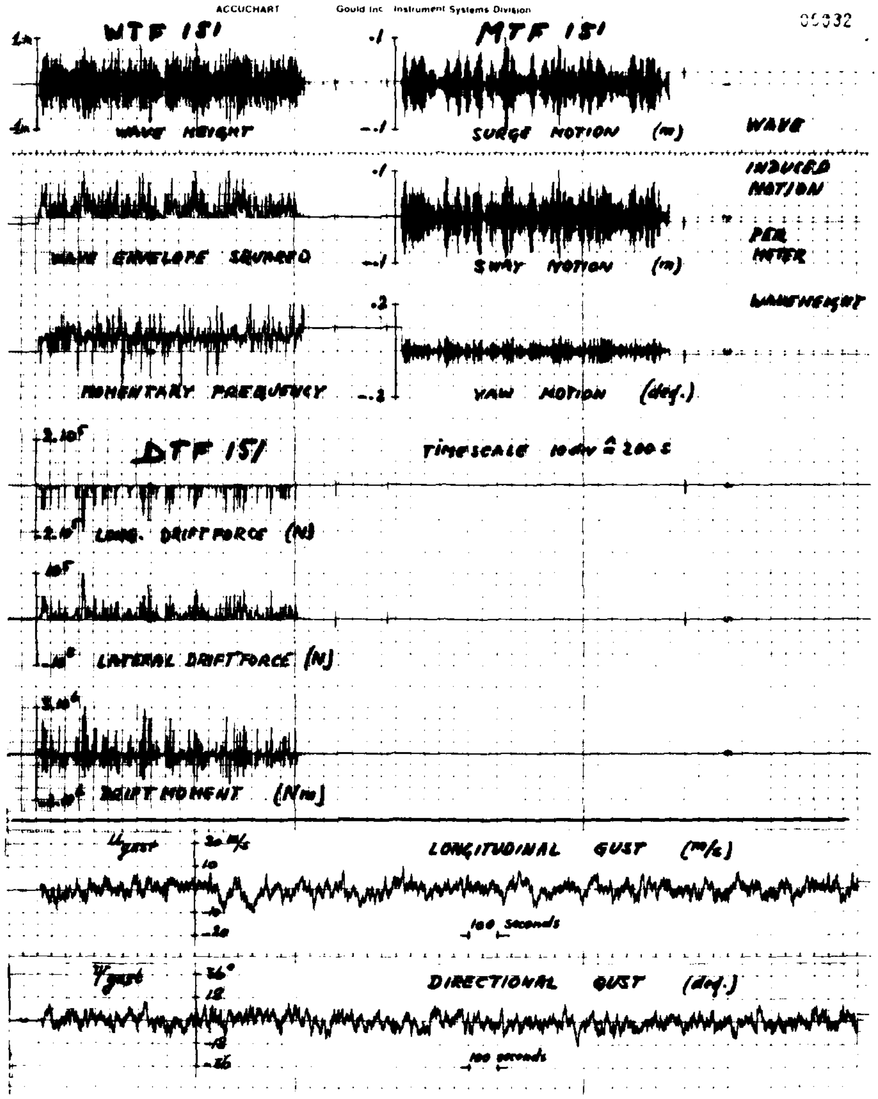


Figure 5: Timeseries of wave induced motions, driftforces and -moment of VLCC, fully laden. Wave direction: 30° off bow. Time series of windgust signals.

Programmes are available, which convert the motion transfer functions into time series, for a given wave spectrum and which generate time series of the drift forces and moment for a given wave spectrum, on the basis of the mean drift forces and moment in regular waves.
Figure 5 shows a record of the motion and driftforce time-series (waves on the bow).

On line, the wave induced motions are superimposed on the calculated low frequency position and heading, while the wave driftforces and moment are introduced as external disturbances, in the same way as the wind.

The wave spectrum used is the modified Pierson - Moskowitz spectrum. The two parameter spectral density function is:

$$S_{\zeta\zeta}(\omega) = A\omega^{-5} \exp(-B/\omega^{4}), \text{ in which}$$
$$A = 172.79\, H_{1/3}^{2}\, T^{-4} \qquad \text{and } B = 691.17\, T^{-4} \tag{6}$$

The characteristic parameters are the significant wave height $H_{1/3}$ and the average wave period T.

Current

The (ocean) current comes only into play - as far as the vessel is concerned - when the velocity relative to water is converted into velocity relative to ground. The accompanying forces are part of the total hydrodynamic forces, because the vessel's velocity relative to the water includes the current velocity.

During the simulation only zero-current conditions have been observed.

CONTROL SYSTEMS

Adaptive Controller

Two major problems must be considered when designing an adaptive autopilot:
1.- Assessment of a suitable mathematical model of the ship steering dynamics at different operational conditions.
2.- Determination,for such a model, of optimal control strategies in order to minimize a preset cost function.

On the basis of the results of a large number of identification experiments, (11), (12), (13), a discrete-time A.R.M.A. (Auto Regressive Moving Average) equation has been assumed, in the course keeping case, as a mathematical model of the ship response to the rudder and to the external disturbances:

$$y(t) + \sum_{i=1}^{n} a_i y(t-i) = \sum_{i=1}^{n} b_i u(t-k-i) + \lambda\left[e(t) + \sum_{i=1}^{n} c_i e(t-i)\right] \tag{7}$$

Where y(t) represents the heading error deviation and u(t) the commanded rudder angle, while λ is a parameter depending on the external noise level and e(t) is a sequence of independent equidistributed random variables with zero mean and standard deviation equal to one.

If sufficient information is available, the model order n can be assigned by a-priori kwowledge of the ship dynamics; in any case it can always be determined by experimental identification.
The time delay k, which takes the inertia of the steering gear hydraulic actuation system into account, depends on the chosen sampling interval.

As concerns the cost function associated to the steering control system, it is generally accepted, that a minimization of the propulsion losses caused by the ship's course-keeping control should be adopted as a performance index of autopilot

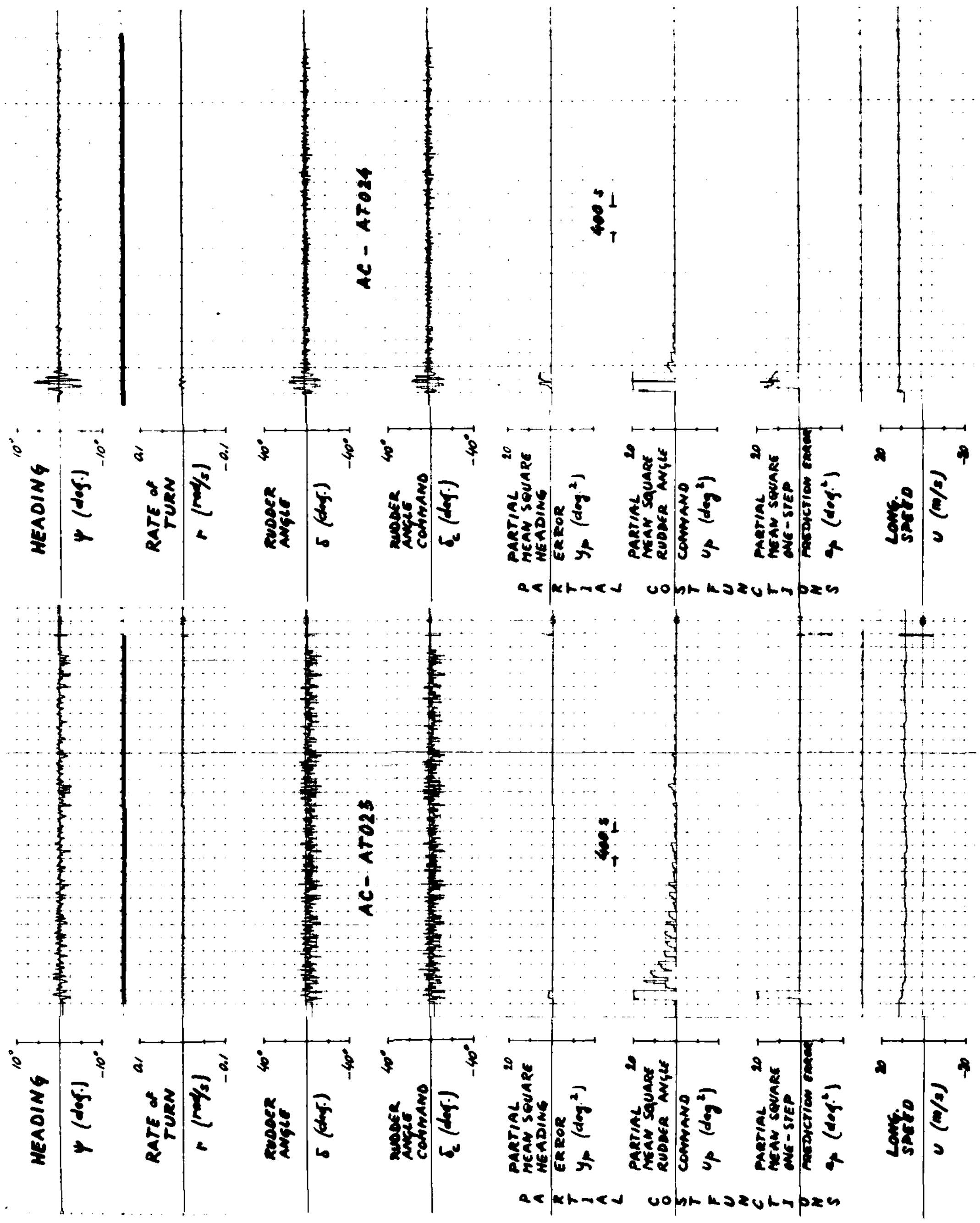


Figure 13: Time histories of runs AC - AT 023 and AC - AT 024 (containership).

Table 4: Conditions of runs with adaptive controller.

RUNNR.	WIND		WAVES	
	Speed (m/s)	Direction (deg)	Height (m)	Direction (deg.)
AC - TF 007	-	-	-	-
AC - TF 013	20	30	8	30
AC - TF 019	10	150	4	150
AC - TB 015	-	-	-	-
AC - TB 036	20	30	8	30
AC - AT 012	-	-	-	-
AC - AT 023	20	180	8	150
AC - AT 024	10	150	4	150
The wind and wave direction is relative to the ship's longitudinal axis: 150 degrees means 30 degrees off bow				

The robustness of the algorithm can be appreciated in Figure 14. It can be noted that starting with a parameter-estimate obtained during trials with different weather conditions, the system is able to tune the parameter-values within a few minutes.

The so-called exponential forgetting factor has been varied from .98 to .99 during the trials for moderate up to severe weather conditions.

In one of the next paragraphs the results, obtained with the adaptive-controller, will be compared with those, obtained with the PID-controller.

PID-Control

Although the tuning of the control system parameters has not been done extensively, the results are quite acceptable, except in very bad weather conditions (especially in beam seas).

A few typical runs are presented in figures 15, 16 and 17. The corresponding run condition are given below:

Table 5: Conditions of runs with PID-controller

RUNNR.	WIND		WAVES	
	Speed (m/s)	Direction (deg.)	Height (m)	Direction (deg.)
PID -TF 013	20	30	8	30
PID -TF 017	20	180	8	150
PID -TF 019	10	150	4	150
PID -TF 029	10	90	6	90
PID -TF 036	20	30	8	30
PID -TF 037	10	150	4	150
PID -TF 023	20	180	8	150
PID -TF 024	10	150	4	150

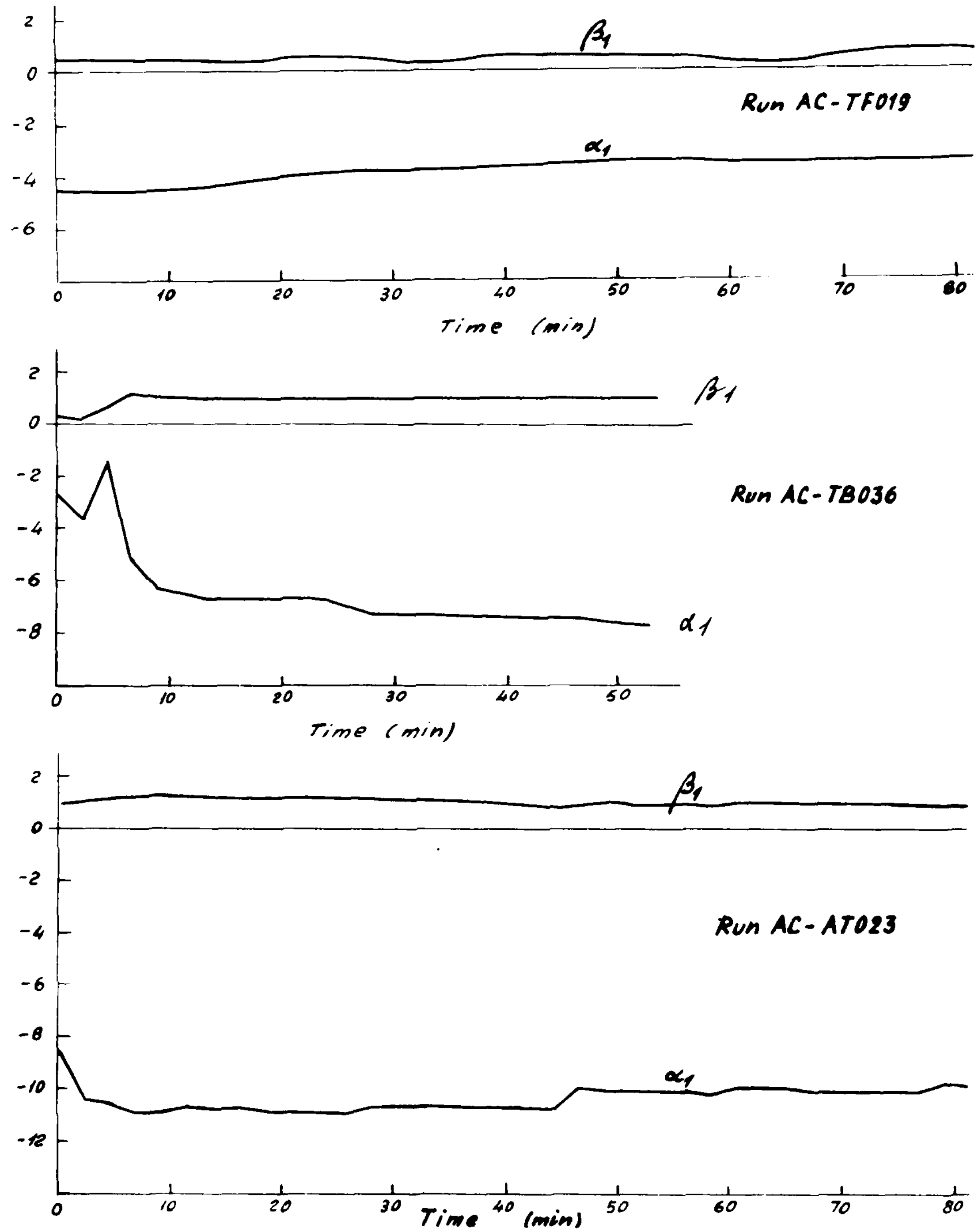


Figure 14: Model coefficients α_1 and β_1 to show robustness of algorithm. Runs AC-TF 019 (tanker, full) AC - TB 036 (tanker, ballast) AC - AT 023 (containership).

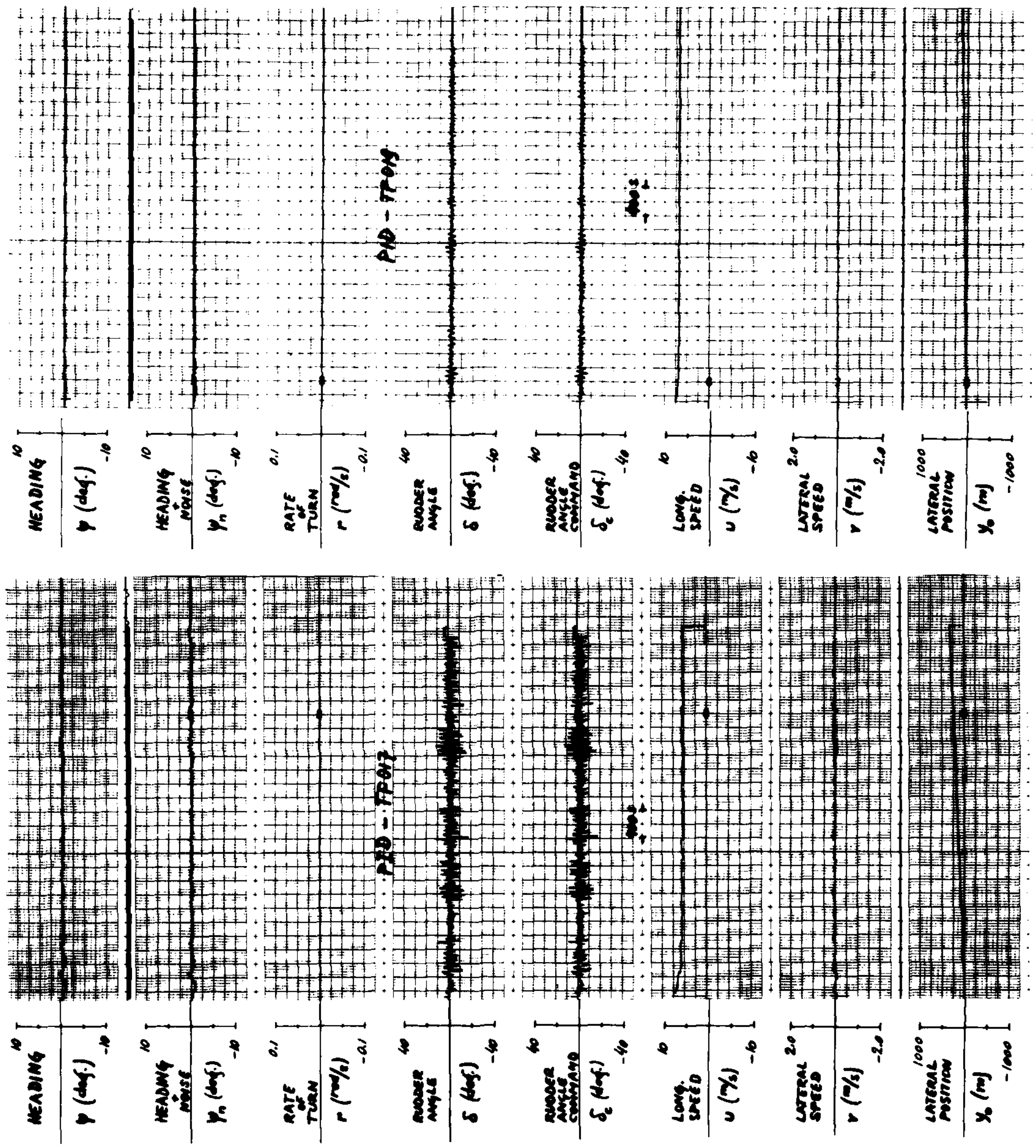


Figure 15: Time histories of runs PID - TF 017 and PID - TF 019 (tanker, full).

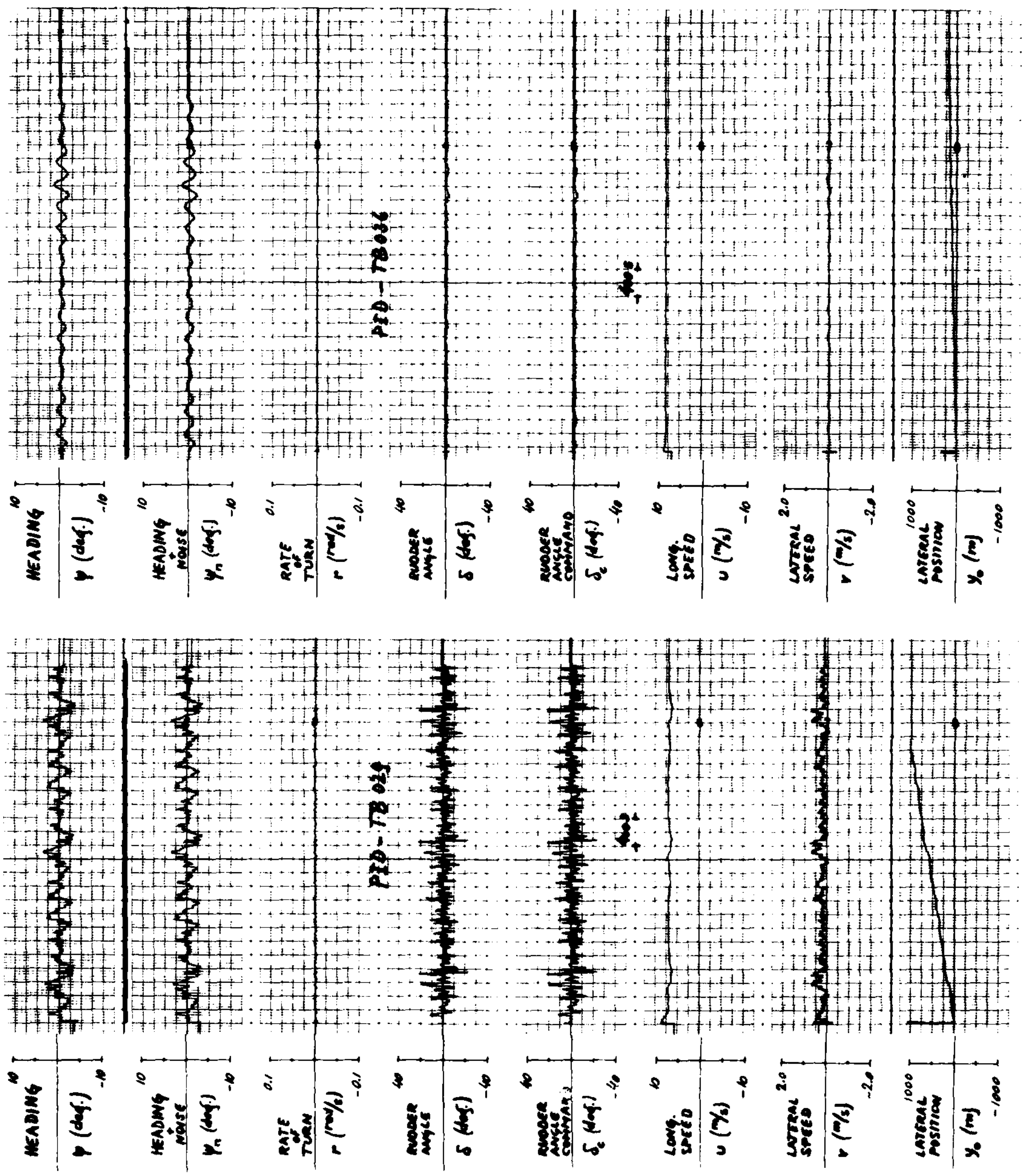


Figure 16: Time histories of runs PID - TB 029 and PID - TB 036 (tanker, ballast)

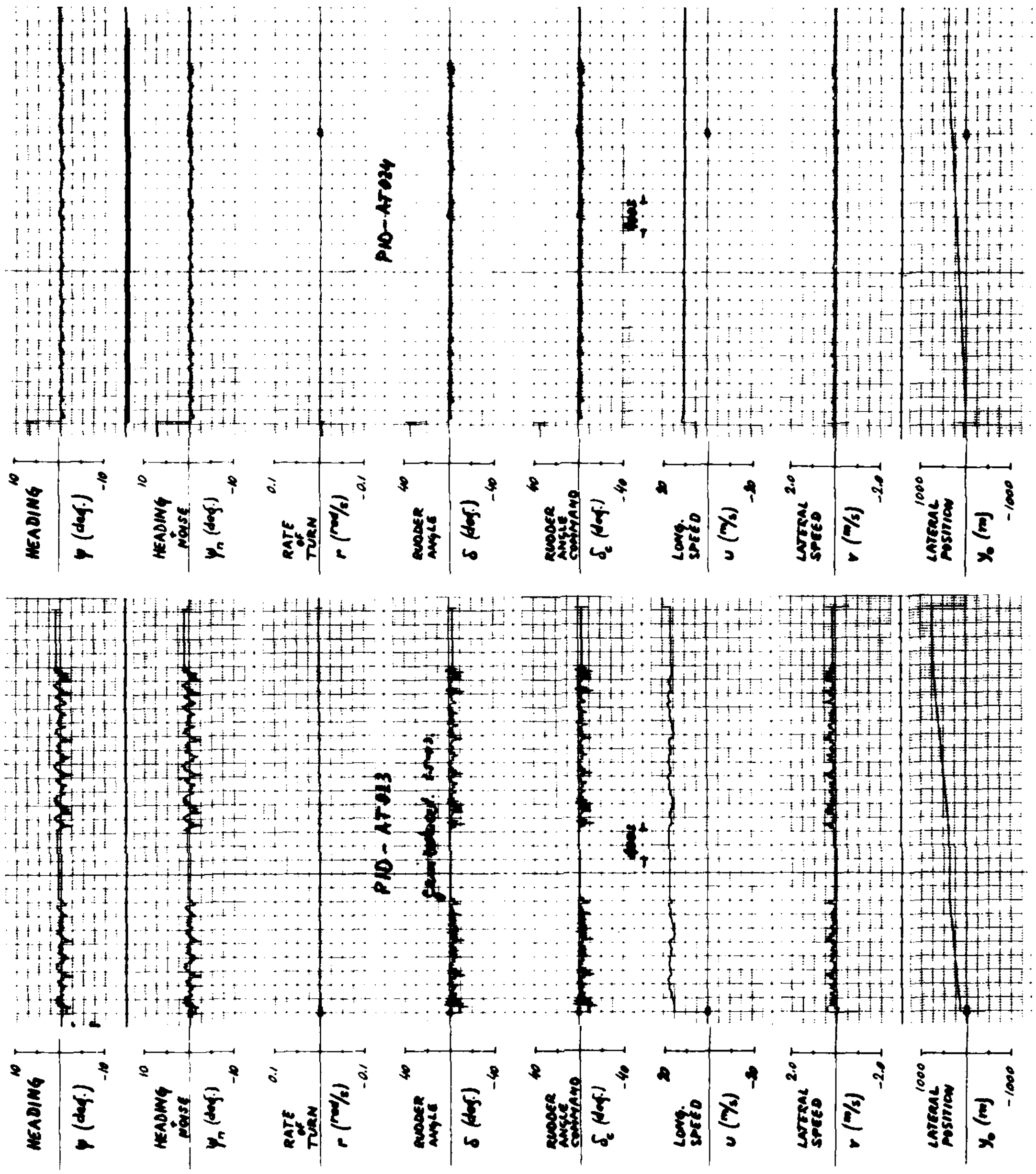


Figure 17. Time histories of runs PID - AT 023 and PID - AT 024 (containership).

As said before, the control system sampling period has been chosen equal to two seconds, which is different from the adaptive controller sampling period. It means that the steering gear will be activated almost every two seconds although the commanded rudder angle is small during most of the time.

For both loading conditions of the supertanker, the same transfer function for the low-pass and the derivative filter could be applied. Only the nominal heading control gain factor $K_{p\psi}$ has been reduced for the supertanker in ballast. However, it proved not necessary to adjust the chosen gain factor under different weather conditions.

The transfer functions to both the low-pass and derivative filter have been tailored to the characteristics of the smaller and faster containership. During most runs the nominal gain factor has been kept constant, expect during run PID-AT023. Those conditions asked for a smaller gain factor. As a matter of fact, this means that adjustment of the gain factor should also be possible on board.

Table 6 lists the results of all runs obtained with a statistical analysis programme.
Also a discussion of the results compared with those obtained with the adaptive controller is presented.

Comparison of the Results.

The performance of both controllers can be compared to a certain extent on the basis of the statistical analysis, presented in Table 6.

Since all the compared trials have been simulated at the same r.p.m., it seems that the effective speed is the most suitable performance index to judge the autopilot.

It should be taken into account that the study has not been concluded. Nevertheless, some comment can be given:

1) As far as the supertanker is concerned in the two loading conditions, it seems that the minimum variance adaptive autopilot and the well-tuned PID-autopilot attain, in the different considered conditions a comparable performance, in terms of heading error variance and effective speed, while the adaptive autopilot is somewhat better in the most severe weather conditions (beam sea).
2) For the containership, it seems that the adaptive autopilot works globally better at all the different conditions, with a significant improvement at moderate to rough weather conditions.
3) It must be noted that, during all the considered trials, the adaptive autopilot generally started with a complete ignorance about the system parameters, which were to be estimated during the experiment. For the ship having very slow dynamics (supertanker), it often happened that, over the considered time interval, the autopilot did not reach a stationary steady state behaviour.
   It has been observed, in fact, that during the subsequent times, the adaptive autopilot was generally able to further improve.

Table 6: Comparison of performance of controllers

	Run-number	Heading error		Effective speed		Commanded rudder angle		Rudder angle	
		Mean (deg.)	Mean square ($deg.^2$)	Mean (knots)	Mean square ($knots^2$)	Mean (deg.)	Mean square ($deg.^2$)	Mean (deg.)	Mean square ($deg.^2$)
A	AC - TF 13	.03	.03	12.67	160.69	-1.60	6.03	-1.60	5.99
D	AC - TF 17	-.01	.05	10.58	111.74	-2.25	44.14	-2.16	37.60
A	AC - TF 19	-.05	.03	13.15	172.90	-.88	4.35	-.88	4.39
P	AC - TF 22	.01	.54	13.53	183.41	-4.20	[illegible]9.60	-4.20	99.67
T	AC - TF 23	-.007	4.64	13.14	175.40	-4.20	112.93	-4.20	120.21
I									
V	AC - TB 29	.07	1.45	13.19	172.81	-1.49	122.25	-1.38	85.15
E	AC - TB 30	.46	8.05	11.07	123.03	-.50	169.69	-.54	144.00
	AC - TB 36	-.0002	.15	15.43	238.50	-1.42	62.76	-1.37	42.95
C	AC - TB 37	-.004	.026	13.01	169.10	-.23	18.58	-.17	15.94
O									
N	AC - AT 23	-.11	.48	16.40	269.50	-3.32	16.42	-3.30	16.40
T	AC - AT 24	.004	.06	20.64	452.90	-1.28	5.39	-1.27	5.40
R	AC - AT 27	.01	.12	21.87	478.12	-2.54	18.63	-2.52	16.46
O	AC - AT 28	-.3	1.41	21.37	456.46	-7.89	80.12	-7.87	79.27
L	AC - AT 34	.003	.06	22.47	504.86	-1.39	19.99	-1.37	11.70

Table 6: continued

P	PI - TF 13	-.01	.008	12.68	160.81	-1.65	4.77	-1.65	4.79
I	PI - TF 17	-.01	.047	10.57	111.78	-2.30	40.71	-2.30	40.88
D	PI - TF 19	-.001	.008	13.15	173.00	-.87	2.41	-.87	2.43
	PI - TF 22	-.01	.60	13.58	184.51	-4.06	30.48	-4.06	30.54
	PI - TF 23	-.38	6.94	12.94	167.69	-9.15	185.60	-9.10	185.86
	PI - TB 29	.02	1.96	13.26	176.02	-1.30	40.73	-1.30	40.79
	PI - TB 30	.70	20.76	10.67	114.30	-1.88	407.25	-1.37	339.00
	PI - TB 36	-.02	.26	15.61	243.70	-1.52	2.74	-1.52	2.74
C	PI - TB 37	-.01	.044	13.01	169.40	-.16	.75	-.16	.76
O									
N	PI - AT 23	-.01	.82	16.15	261.16	-3.22	18.99	-3.22	19.00
T	PI - AT 24	-.01	.04	20.58	423.73	-1.32	3.08	-1.32	3.08
R	PI - AT 27	-.02	.14	21.86	477.97	-2.62	8.43	-2.62	8.44
O	PI - AT 28	-.04	1.60	21.23	450.78	-8.27	88.97	-8.27	89.00
L	PI - AT 34	-.01	.04	22.50	506.30	-1.37	2.23	-1.37	2.23

CONCLUSIONS

The results have shown that the self-tuning minimum variance regulator has good stability and convergence characteristics for all three considered ships.

The performance of the controllers is comparable in many cases. However, in rough weather, the adaptive controller works better. Moreover, the duration of the simulation runs was not always sufficiently long, as to quarantee that the adaptive controller reached a steady state. In other words: in many cases the performance would still improve as time goes on.

One difference between the adaptive- and PID-controller has not been taken into account in the performance-analysis. This is the difference in sampling period: The smaller sampling period of the PID-controller means more steering gear actuations per unit of time.

As said before, the paper reports of the status of the work at the end of 1977. Further study is needed. Aspects to be considered are:
- investigation of other adaptive control schemes.
- economic considerations to be observed with regard to the design of adaptive controllers.
- operational aspects of the implementation of the adaptive controller
- extension of the research towards course changing, trackkeeping, shallow water and non-homogeneous current effects
- evaluation of factors to be taken into account when defining cost functions, such as: heading error, rudder angle, increase of resistance which reduces the effective speed,actuations of the steering gear
- influence of the sampling time on the performance of the PID-controller
- combining digital filtering techniques with the self-tuning regulator
- constraints introduced by the steering gear (i.e. the allowable number of actuations per unit of time).

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LIST OF SYMBOLS

<u>Ship, Ship Motions and Environment</u>.

C_D, C_L	drag and lift coefficient of rudder
C_Q, C_T	torque and thrust coefficient of propeller
c_a	speed increase in propeller race
D	diameter of propeller
D_r, L_r	drag and lift force of rudder
L	ship length
l	distance between c. g. and 50% - point of rudder chord
m, I_z	mass of ship and polar mass moment of inertia about vertical axis
n	rpm of propeller
S_r	rudder area
S_w	reference wind area
T_p	propeller thrust
U_{sr}	water speed relative to rudder
V_A	intake watervelocity into propeller
X,Y,N	forces and moment acting on ship in x-, y- direction and about vertical z-axis, respectively
β	drift angle
δ, δ_e	geometric and effective rudder angle, respectively
δ_v	direction of U_{sr} relative to x-axis
U,V	ship's speed rel. to water and to ground, resp.
u,v,r	speed rel. to water in x-direction, y-direction and about the vertical z-axis, resp.
$\dot{u},\dot{v},\dot{r}$	accelerations; derivatives of u,v,r, resp.
$\dot{x}_o,\dot{y}_o,\dot{\psi}$	speed rel. to ground in x_o-direction, y_o-direction and about the vertical z_o-axis, resp.
ψ	ship's heading
$H_{1/3}$, T	signifant wave height and average wave period
$S_{\zeta\zeta}$	wave spectral density
Ψ_{wave}	direction of the waves
V_c,Ψ_c	current speed and -direction
V_w,Ψ_w	wind speed and -direction
ρ,ρ_{air}	density of (sea) water and of air

<u>Adaptive Controller and PID-Controller</u>

a,b	system parameters
k	system time delay
J	cost function
R_y,R_{yu}	auto-covariance of system output and cross-covariance of system input and output
α,β	model parameters
a_f, b_f	coefficients of heading signal filter of PID-controller
a_d, b_d	coefficients of derivative filter of PID-controller
$K_{p\psi}$, K_{iz}	proportional and integral control gain factors
Δ_c, Ψ_m	z - transforms of rudder angle command δ_c and ship's heading ψ

APPENDIX A

Forces and Moments Exerted on Tanker and on Containership

Hull Contribution. The forces and moment on the ship's hull (without propeller(s) and rudder(s)) are expressed as follows:
For the 250.000 tons VLCC, fully laden as well as in ballast condition:

$$X_{hull} = D_1\dot{u} + D_2u^2 + D_3v^2 + D_4r^2 + D_5vr + D_6u/v/$$

$$Y_{hull} = E_1\dot{v} + E_2uv + E_3v/v/ + E_4ur + E_5r/v/$$

$$N_{hull} = F_1\dot{r} + F_2uv + F_3v/v/ + F_4ur + F_5r/r/$$

Coupling effects of roll, pitch and heave motions into the horizontal motion have been neglected.
The values of the coefficients D_1 trough F_5 are different for the two loading conditions.

For the second generation container ship:

$$X_{hull} = A_1\dot{u} + A_2u^2 + A_3v^2 + A_4r^2 + A_5vr$$

$$Y_{hull} = B_1\dot{v} + B_2\dot{r} + B_3uv + B_4v^3/u + B_5ur + B_6r^3/u + B_7v^2r/u + B_8vr^2/u$$

$$N_{hull} = C_1\dot{r} + C_2\dot{v} + C_3uv + C_4v^3/u + C_5ur + C_6r^3/u + C_7v^2r/u + C_8vr^2/u$$

Propeller Contribution. The propeller thrust and torque are written as

$$T_p = C_T\rho\ D^2\ [V_A^2 + (nD)^2]$$
$$Q_p = C_Q\rho\ D^3\ [V_A^2 + (nD)^2]$$

It is assumed that C_T and C_Q are known as a function of the propeller flow parameter

$$\sigma = \frac{nD}{\sqrt{V_A^2 + (nD)^2}}$$

Now the propeller contribution to the X-force can be calculated, assuming the wake fraction (w) and the thrust deductionfactor (t) are known. The contribution to the Y-force and the moment is assumed to be linear with the X-force.
Hence:

$$X_{prop} = (1-t)C_T\rho D^2 * [u^2(1-w)+(nD)^2] \qquad \text{with } V_A = u\ (1-w)$$
$$Y_{prop} = K_{yp}\ X_{prop}$$
$$N_{prop} = K_{np}\ X_{prop}$$

Rudder Contribution. The velocity along the x-axis in the slipstream of the propeller at the position of the rudder amounts to $u_{sr} = V_A + c_a$, in which the sign of the speed increase c_a is directly related to the sign of the thrust (C_T). For the different ship types u_{sr} is calculated as follows:
For the VLCC, fully laden as well as in ballast condition, c_a is assumed to be proportional with the square root of the thrust. Thus:

$$u_{sr} = (1-w)\ u + K_{ca}\frac{Tp}{|Tp|}\ \frac{\sqrt{|Tp|}}{\rho\frac{\pi}{4}D^2}$$

For the second generation containership, u_{sr} is calculated in the following way

The application of momentum theory yields for the thrust

$$T_p = \rho\frac{\pi}{4}D^2(V_A+\tfrac{1}{2}c_a)c_a$$

from which

$$V_A+c_a=\sqrt{V_A^2+\frac{2Tp}{\rho\frac{\pi}{4}D^2}}$$

or

$$u_{sr} = V_A+c_a = (1-w)u\sqrt{1+\frac{8}{\pi}C_T\frac{1}{1-\sigma^2}}$$

It is thought reasonable to assume u_{sr} being zero for

$$1 + \frac{8}{\pi}C_T\ \frac{1}{1-\sigma^2} < 0$$

The flow field at the position of the rudder is not only determined by u_{sr}, but also by the sideslip velocity v and the yawing velocity lr (l being the distance between the centre of gravity and the 50 percent chord point of the mean geometric rudder length).
The total on coming speed relative to the rudder becomes:

$$U_{sr} = \sqrt{u_{sr}^2 + (v-lr)^2}$$

The forces on the rudder (lift and drag) are:

$$L_R = C_L\ \tfrac{1}{2}\rho U_{sr}^2\ S_r$$
$$D_R = C_D\ \tfrac{1}{2}\rho U_{sr}^2\ S_r$$

These forces become along the x- and y-axis of the ship:

$$X_{rud} = -L_R\sin\delta_v - D_R\cos\delta_v$$
$$Y_{rud} = L_R\cos\delta_v - D_R\sin\delta_v$$

The rudder moment N_{rud} is: $N_{rud} = -lY_{rud}$

The coefficients C_L and C_D have to be known as functions of the effective rudder angle δ_e for the respective rudder configurations.

The definition of the angle δ_v and δ_e follows from the figure below.

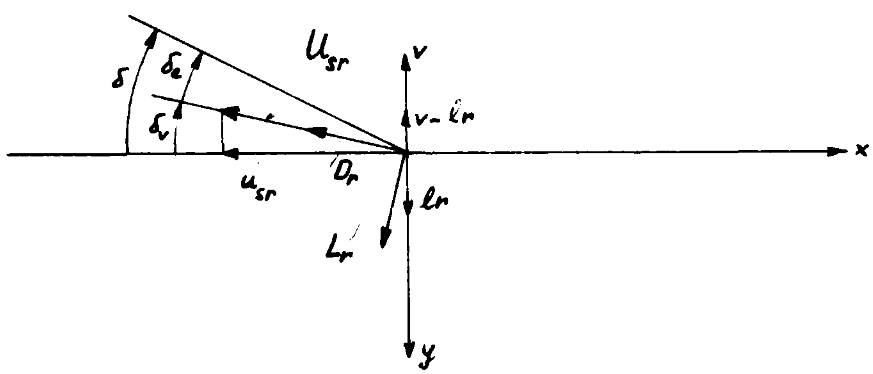

Figure 18: Relative speeds with respect to the rudder and forces on the rudder.

Wind Contribution. The components of the relative wind velocity in the x-y axis system are given by:

$$u_{wr} = V_w \cos(\psi_w - \psi) - u - V_c \cos(\psi_c - \psi)$$

$$v_{wr} = V_w \sin(\psi_w - \psi) - v - V_c \sin(\psi_c - \psi)$$

The resultant relative wind velocity and direction are given by:

$$V_{wr} = \sqrt{u_{wr}^2 + v_{wr}^2}\,,\quad \psi_{wr} = \arctan \frac{v_{wr}}{u_{wr}}$$

The wind forces and moment are:

$$X_{wind} = C_{xwind}\ \tfrac{1}{2}\rho_{air}\ V_{wr}^2\ S_w$$

$$Y_{wind} = C_{ywind}\ \tfrac{1}{2}\rho_{air}\ V_{wr}^2\ S_w$$

$$N_{wind} = C_{nwind}\ \tfrac{1}{2}\rho_{air}\ V_{wr}^2\ S_w L$$

The wind speed V_w is composed of a constant part V_{w1} and a gust part u_g;
the same for wind angle $\psi_w = \psi_{wg} + \psi_g$.
The gust signals are generated off-line and stored on disc as a function of time. During the simulation these parameters are read from disc and added to the constant components.

Low Frequency Wave Drift Forces and Moment. The low frequency wave drift forces and moment are generated off-line and stored on disc as a function of time. During the simulation these forces and moment are read from disc.

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