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The Computerized Planar Motion Carriage (CPMC) recently operationalized at Hamburg for better determination of ship maneuverability has several novel features. Carriage controls were specifically designed for generating transient motions. Superposition of four independent drives can generate any realistic horizontal planar motion. A freely programmable, process-control computer executes carriage control, model control, data acquisition and run evaluation. The CPMC can operate either in the towing mode (for measuring hydrodynamic forces on captive models moved along predetermined trajectories) or in the tracking mode (for following and recording trajectories of freely maneuvering models). Sample results of inaugural experiments in each mode are presented.

INTRODUCTION

Purpose

The purpose of this paper is to introduce to the international community of naval hydrodynamicists a novel model-testing facility recently operationalized at Hamburg with the aim of determining the maneuvering capabilities of ships more accurately and completely than was previously possible. We have chosen to call it CPMC, an abbreviation for Computerized Planar Motion Carriage. It incorporates the pertinent features of a conventional towing carriage and of special devices for conducting maneuvering experiments such as rotating arms, so-called planar motion mechanisms, and xy-carriages. Besides, it has several unique capabilities, not available to our knowledge in any existing single facility, such as independent multiple drives in 3 degrees of freedom, generation of precise transient motions, alternative towing and tracking modes, and flexible computer programming of carriage controls, model controls, data acquisiton and run evaluation. The audience will hopefully agree with the organizers that the CPMC is a suitable subject for the opening paper of a Symposium devoted to the "Unsteady Hydrodynamics of Marine Vehicles".

Brief history of PMMT

In the context of tankery the term planar motion model testing (PMMT) has come to connote in recent years the determination of velocity *and* acceleration dependent hydrodynamic forces on ocean vehicles by generating idealized planar motions (either horizontal or vertical) of captive models in a tank, recording the resulting forces and finally analysing and synthesizing the data in a manner appropriate to the chosen mathematical form for the equations of motion. Simply steady speed rectilinear oblique towing or even circular rotating arm tests are not considered PMMT. In this particular sense of the term the first published results of PMMT, to the authors' knowledge, were those of Horn and Walinski (1958, 1959).* They used a pair of cranks of 20 cm radius to generate nearly pure harmonic sway and yaw motions of a captive ship model in the *Versuchsanstalt für Wasserbau und Schiffbau*, Berlin. It is a pity that this pioneering and comprehensive paper, which contains a complete recipé for generating purer yaw motions by means of a pair of Scotch yokes rather than slider cranks and for analysing the recorded forces, has been almost totally ignored in the pertinent literature. The term planar motion mechnism (PMM) was of course coined by Gertler (1959) and Goodman (1960) to denote an ingenious two-point slider-crank oscillator of 1 inch radius designed mainly for testing submarine models at the David Taylor Model Basin (DTMB) near Washington, D.C. This PMM system and subsequent improved versions are reported to have been operating successfully at DTMB for almost two decades now, although nearly all the results seem to be classified. It was followed up by similar devices all over the world: Paulling and Sibul (1962) reported a PMM at the University of California, Berkeley (of 2 inch maximum sway amplitude); Keil and Thiemann (1963) at the *Institut für Schiffbau*, Hamburg (6 cm); Zunderdorp and Buitenhek (1963) at the Technological University, Delft (30 cm); Motora and Fujino (1965) at the University of Tokyo, Tokyo (35 cm); Strøm-Tejsen and Chislett (1966) at the *Hydro- og Aerodynamisk Laboratorium*, Lyngby (10 cm); Fujii (1969) at the Nagasaki Experimental Tank, Nagasaki (47 cm); Chislett and Smitt (1972) again at the *Hydro- og Aerodynamisk Laboratorium*, Lyngby (75 cm), and this list is certainly not complete. The most outstanding published results seem to be firstly by Leeuwen (1964), who thoroughly investigated frequency effects, Froude-number effects and effects of rudder and propeller on a standard Series 60 model, and secondly by Strøm-Tejsen and Chislett (1966) continued in Chislett and Smitt (1974) who demonstrated how to identify well over 50 empirical coefficients in the nonlinear equations of motion in 3 degrees of freedom and to obtain predictions for full scale maneuvers in fair conformity with actual trial trip results.

*References are identified by author(s) and year and listed in alphabetical order by first author, and if necessary in chronological suborder, at the end of the text.

In a series of papers Bishop et al. (1970,1973, 1973, 1974,1974) took a critical look at the PMMT technique and pointed up the need for neatly separating formally linear memory (motion history) effects from truly nonlinear effects in modelling hydrodynamic forces as functionals of motions. Using the classical technique of Fourier transforms they also demonstrated, although only within the linear range, the equivalence of representing memory effects either through impulse response or through frequency response functions. This had already been exploited earlier by Cummins (1962) and Kotik and Mangulis (1962), at least for ship motions in the vertical plane. According to Burcher (1975) and Nomoto (1975) the general consensus among maneuvering experimenters now seems to be that memory effects are not a genuine problem of any realistic ship maneuvers but rather an artifical problem of PMMT created by the use of low amplitude, high frequency devices. Leeuwen (1969) had already reached this conclusion several years ago and proposed a horizontal oscillator of extremely large amplitude (about 375 cm) and low frequency in order to simulate ship motions realistically.

Motivation for CPMC

When the *Institut für Schiffbau der Universität Hamburg* (IfS) joined hands with the *Hamburgische Schiffbau-Versuchsanstalt* (HSVA) - and with the *Technische Universität*, Hanover, and the *Germanischer Lloyd*, Hamburg - to form a special research pool for shipbuilding designated the *Sonderforschungsbereich Schiffbau* (SFB 98) in 1970, an opportunity arose at least to plan (and partially to build) new ship model testing facilities on a rather grand scale. In particular, one of the projects entitled "Safety of Ships against Collisions" called for a new technique of determining the maneuvering capability of ships more accurately and completely than hitherto considered possible. The team (Keil, Oltmann and Sharma) responsible for planning an experimental device for this purpose was at first considering a PMM of about 80 cm amplitude which would have fitted well into the trend of so-called large amplitude PMM which seemed to be roughly doubling in amplitude every eight years since 1958. However, a chance discussion with Professors Gerritsma, Glansdorp and Meier at Hamburg in early 1971 soon convinced us that such a device, while presumably acceptable for routine tests, would be totally inadequate as a research tool. Only a device of the magnitude proposed by Leeuwen (1969) could be expected to lead to an improved mathematical model for the hydrodynamic forces as functionals of motions, which was needed to calculate radical maneuvers typical of collision avoidance at the last minute. After thorough internal deliberations three firms were invited to submit feasibility studies for various alternatives which were scrutinized by independent reviewers appointed by the *Deutsche Forschungsgemeinschaft*, which was to share the financing with the HSVA. The ultimate product which emerged was CPMC, the subject of this paper. Just to complete the chronology we add that the building contracts were awarded to Messrs. Kempf & Remmers, Hamburg, and to Siemens AG, Erlangen and Hamburg, in 1972; the detailed designs were completed in 1973, and assembly began in 1974. The final adjustments and comprehensive acceptance trials took up the better part of 1975. The CPMC was formally declared operational on 8 October 1975. An advance announcement had been made in a note to the 14th ITTC by Krappinger and Sharma (1975).

NOTATION

Abbreviations

CPMC	Computerized planar motion carriage
HSVA	*Hamburgische Schiffbau-Versuchsanstalt*
IfS	*Institut für Schiffbau*, Hamburg
ITTC	International Towing Tank Conference
K	Amplifier
M	Servomotor
P	Proportional control
PI	Proportional-integral control
PMM	Planar motion mechanism
PMMT	Planar motion model testing
PT	Paper tape
PTI	Paper tape input
PTO	Paper tape output
SFB 98	*Sonderforschungsbereich Schiffbau*
T	Tachometer
TT	Teletype
TTC	Teletype console

Symbols

B	Beam of model
Dt	Basic control cycle (10 ms)
Dx_o etc.	Increments of x_o, y_o, ψ in time step Dt
f	Frequency in cycles per second
F_n	Froude number
g	Acceleration due to gravity
I_a	Armature current
I_{xx} etc.	Moment of inertia of model about the x, y or z axes
K,M,N	Hydrodynamic moments about the x,y,z axes
L	Length of model (between perpendiculars)
m	Mass of model
n	Rate of revolutions of propeller (motor)
O	Coordinate origin defined in the model, usually at midship in the waterplane
p,q,r	Rates of turn about the x, y, z axes
T	Period of oscillation; also model draft
t	Time
U	Resultant velocity of O in the horizontal plane
U_o	Initial value of U in the approach phase
u,v,w	Component of U along x, y, z axes
X,Y,Z	Hydrodynamic forces along x, y, z axes
x,y,z	Coordinate axes moving with the model
x_G,y_G,z_G	Coordinates of model center of gravity
x_o,y_o,z_o	Coordinates of O in a tank-fixed system
$\bar{x}_o(t)$	Position of main carriage with respect to tank
β	Drift angle
$\Delta x_o(t)$	Position of Δx_o-subcarriage with respect to its nullpoint
$\Delta\Delta x_o, \Delta y_o$	Tracking lag, i.e. position of O with respect to a reference point of the CPMC
δ	Rudder angle
δx_o etc.	Position errors (actual value minus required value) of x_o, y_o and ψ
ρ	Mass density of water
ϕ,θ,ψ	Angles of roll, pitch and yaw
ω	Circular frequency in radians per second

Note - ITTC standard symbols have been used as far as possible. A prime denotes that the quantity has been nondimensionalized using as fundamental units $\rho L^3/2$ for mass, L for length and L/U for time. A dot denotes time derivative as usual. A hat is used to denote some constant values of a motion variable, usually the amplitude.

DESIGN DECISIONS

General considerations

The basic philosophy leading to the design of the CPMC has already been outlined in the Introduction. The general idea was to build a device capable of generating on the model scale every realistic horizontal motion of which a merchant ship of any type is capable. The purpose of the following remarks is to justify some of the major choices which had to be made at the concept formulation stage and for drafting the technical specifications as a basis for the building contract. Naturally, many of these choices were dictated by the given features of the three main facilities, which were already present and had to be integrated into the total system, viz. the towing tank (280 m x 18 m x 6 m), the main carriage (a relatively soft framework structure of nearly 50 Mg mass), and the general purpose process-control computer (Siemens 301).

At this point it is useful to recall that a typical motion required in PMMT is a harmonic variation of pure yaw at constant forward speed and zero drift angle, e.g.

$$\psi(t) = \hat{\psi}\sin(\omega t),\ u(t) = \hat{U},\ v(t) = 0, \qquad (1)$$

alternatively in tank-fixed coordinates:

$$\dot{x}_0 = \hat{U}\cos(\hat{\psi}\sin\omega t),\ \dot{y}_0 = \hat{U}\sin(\hat{\psi}\sin\omega t),\ \psi = \hat{\psi}\sin(\omega t) \qquad (2)$$

Evidently, this involves the perfect combination of a constant speed in the tank longitudinal direction with periodic pure oscillations each in the tank longitudinal direction, the tank transverse direction and about a vertical axis. If maximum realistic yaw rates are to be attained *without* exceeding maximum realistic yaw accelerations, transverse amplitudes on the order of a model length are necessary, cf. Leeuwen (1969). With model lengths typically lying in the range of 6 to 8 m at the HSVA this implied utilizing essentially the entire available tank width for the transverse oscillation and established a rather gigantic magnitude for the projected device!

Carriage mounting versus trailer

It was decided to mount the oscillating device on an independent trailer rather than on the main towing carriage for the following reasons. Firstly, the carriage was not considered a sufficiently stiff base for measuring forces on captive models in unsteady motion. Due to a steady addition of equipment during the past 20 years it had also already reached its maximum designed weight of about 50 tons. Any permanent structural reinforcement of the carriage to increase its stiffness or carrying capacity would have caused a substantial increase in mass with deterioration of its dynamic performance (and hence a reduction of the useful length of run at high speeds) even for non-oscillatory tests. The trailer, of course, would be hooked to the carriage only for PMMT. Finally, the trailer mounted system would cause during its assembly, testing and servicing a minimum disturbance to the routine operation of the carriage, which is the primary source of income for the HSVA and normally in use 16 hours a day, 5 days a week.

Single versus multiple drives

It is considered an ingenious feature of existing PMM that fully coordinated transverse and rotary motions of the model can be derived by means of mechanical linkages from the rotation of a single drive shaft. However, this works well only for small amplitudes. If the concept is extended to amplitudes on the order of a few meters, with the possible additional requirement of a longitudinal oscillation, the size and complexity of the linkages becomes simply ridiculous. Hence we chose mechanically independent drives in 3 degrees of freedom, i.e. 3 independent subcarriages for superimposing arbitrary motions $\Delta x_0(t)$, $y_0(t)$ and $\psi(t)$ on the generally uniform motion $\bar{x}_0(t)$ of the main carriage. The subdivision of the motion $x_0(t)$ in the tank longitudinal direction into an almost steady component $\bar{x}_0(t)$ and an oscillatory component $\Delta x_0(t)$ served the dual purpose of avoiding having to periodically accelerate the enormous mass of the main carriage and of achieving very accurate motion control using the light subcarriage as a corrective for compensating for unwanted fluctuations of main carriage speed.

Steady speed versus transient control

In principle any periodic motion can be derived mechanically from the uniform rotation of a drive shaft. Exploiting this fact control units of existing PMM are only required to maintain discrete steady speeds of the drive shaft in face of varying load during the run. Having already dispensed with the mechanical linkages for reasons just explained, we took another bold step and demanded precise transient motion control in face of varying loads. Each subcarriage would run on rails, driven by suitable rack and pinion mechanisms, so that the revolutions of the driving motors must, in general, continuously vary in proportion to the required subcarriage speed. Thus we sacrificed simplicity of controls for the sake of mechanical simplicity and complete flexibility of motion. Perfect phase coordination of the independent drives could now only be achieved by providing an input signal of extremely high temporal and spatial resolution and using a sophisticated hybrid control system. Theoretical calculations and simulation studies revealed that it would suffice to use a basic control cycle of 10 ms and feed in new trajectory coordinates from a process-control computer once in every cycle.

Additional tracking mode

Having demanded the capability of practically arbitrary horizontal planar motion within the entire tank area, it was a logical next step to introduce an additional operating mode in which the CPMC would physically follow a ship model maneuvering freely in the tank. Here the CPMC would form a base for exchange of energy and information with the model through a loose umbilical cable cord, thus eliminating the nasty problems of energy storage and wireless telemetry. Moreover, the model trajectory could be measured with high precision and resolution built into the control hardware for the towing mode.

Hydraulic versus electric drives

Careful feasibility studies carried out by the *Vereinigte Flugzeugbau-Werke*, Bremen, and Siemens AG, Erlangen, in 1971 indicated that the specified dynamic performance of the three subcarriages in both operating modes could be attained by servosystems using either hydraulic or electric motors. We chose electric drives for reasons of economy, compatability, and elimination of the risk of contaminating the tank water by an accidental leakage of oil.

HARDWARE

General features

The front view photograph (Fig. 1) conveys a general impression of the CPMC relative to the tank cross section. The tank is aligned roughly in the east-west direction and this picture was taken looking west with the trailer in front and the towing carriage in the background. The spine of the trailer is an extremely stiff closed box girder of steel measuring about 20 m in length, 2 m on the side and 1 cm wall thickness reinforced internally by web frames. Its lowest natural frequeny in any bending mode is above 6 Hz. At each end are platforms carrying racks of electrical hardware strictly separated into power processing on the north and signal processing on the south. The trailer runs on four wheels of hardened steel with a diameter of 70 cm, base of 4 m and span of over 18 m. It is constrained to the south rail by horizontal guide wheels at the center. When not in use it is parked at the east end of the tank and with its overall length of only 6 m it practically does not affect the routine operation of the 300 m towing tank. For use it can be mechanically hooked to the towing carriage by two tie rods, one on each side, each rod being provided with universal joints at both ends. For power and signal exchange there are multipoint cable connectors on the north and south side respectively. The entire coupling or decoupling operation takes no more than 15 min.

The general arrangement of the various subcarriages is depicted in Fig. 2. The y_o-carriage runs under the box girder practically over the entire width of the tank and carries two short longitudinal rails over which the Δx_o-carriage runs. The ψ-carriage, which is actually a turntable capable of rotation about a vertical axis going through the center of the Δx_o-carriage, is suspended from the Δx_o-carriage by a hydraulically operated elevator to accommodate models of different draft and freeboard. The principal particulars of all four carriages are compiled in Table 1. As already stated, the range and dynamics of the CPMC were so dimensioned as to be able to generate (within the constraints of tank size) on model scale every realistic horizontal planar motion of which any merchant ship is capable.

Process control

Fig. 3 is a simplified block diagram showing schematically the flux of information and interaction between essential components of the system. Normally the CPMC is operated under computer control either in Mode A (towing) or in Mode B (tracking). However, for inspection and maintenance as well as for the convenience of model attachment and detachment all carriages can also be moved slowly under manual control (Mode C) from an operating console on the towing carriage. This console contains position and crucial state displays as well as switches for various key functions such as turning power on or off, applying or releasing brakes, starting or stopping a run (under computer control), triggering an emergency halt etc.

The computer, installed in an air-conditioned cabin on the towing carriage, has a cycle time of 1.6 μs and a magnetic core storage capacity of 16 k words of 24 bit length. Communication between the human operator and the computer is via teletype, console and paper tape. The CPMC control unit on the trailer, the towing carriage control hardware and instrumentation in the model interact with the computer via a process control interface currently equipped with eight digital input and output channels of 24 bits each, four analog input channels and two analog output channels. The CPMC control is essentially digital with a basic cycle of 10 ms. In Mode A once every 10 ms the computer reads out a set of trajectory coordinates to the CPMC control unit, reads in (for later inspection) a set of coordinate errors (actual value minus required value), and senses and stores the suitably amplified force signals from the dynamometers in the captive model moving with the CPMC.

The control unit drives each individual carriage independently under digital position control with superimposed speed control as indicated in Fig. 4 for the case of the y_o-carriage. Digital position feedback is supplied by incremental transducers mounted on carriage wheels while analog speed feedback is supplied by tachometers mounted on drive shafts. Note that each carriage is driven by a symmetrical arrangement of four thyristor-controlled DC servomotors, two driving and two braking with their functions exchanged at each reversal of the sense of rotation, in order to avoid any canting or tooth bakclash in the gears, which might affect control accuracy or introduce noise in the force signals.

Towing hardware

The special towing hardware consists of a combined dynamometer and towing guide shown in Fig. 5. It is connected rigidly by four bolts at the top to the center of the ψ-carriage and by four bolts at the bottom to a metal base frame suitably anchored in the model, see also Fig. 2. The towing guide as shown constrains the model in surge, sway, yaw and roll while leaving it free to heave and pitch during the run, which is the normal mode of operation. However, it can also be rearranged to constrain the model to the CPMC in all six degrees of freedom if desired.* Moreover, traverse guides with scales at each end can be manually adjusted to constrain the model at any fixed angle of heel between -10^o and 10^o for special investigations. The dynamometer is a force balance consisting of two rigid beams, the lower one mounted to the model and the upper one to the carriage via the towing guide. The only mechanical connection between the two beams is by means of six slender tie rods so arranged as to accurately resolve the force and moment acting on the model in the components X,Y,Z,K,M and N. Each tie rod has double taper at both ends so that it can transmit only axial forces. At the upper-beam end of each tie rod are built in modular force sensors of the strain gauge type. The entire system is so stiff that the lowest natural frequency with a model of 3.5 Mg mass attached lies at about 5 Hz.

Tracking hardware

In Mode B the model is essentially free in all six degrees of freedom, is necessarily equipped with propulsion and steering gear, and is maneuvered by remote cable control either manually according to will from a command console on the towing carriage or automatically by computer according to a preprogrammed strategy of propeller and rudder com-

*For instance, this mode was used for a dynamic calibration of the force balance by attaching a steel weight (instead of a model) and swinging it in air at different frequencies by means of the CPMC.

mands. The special tracking hardware consists of electro-mechanical tracking lag transducers in all six degrees of freedom and a model locking device, see Fig. 6. The tracking lag is sensed by relative position transducers of pulse or inductance type mounted on a delicate system of almost frictionless slides and wheels interposed between the Δx_o-carriage and the model. The three digital transducers measuring the horizontal tracking lag provide the essential control inputs ($\Delta\Delta x_o$, Δy_o, $\psi_M - \psi_P$) which enable the CPMC to physically follow the model within narrow tolerances, see Fig. 3 and Table 1. The CPMC control unit digitally adds the tracking lag to its own position and transmits to the computer once every 10 ms the tank fixed coordinates of a reference point O in the model as increments Dx_o, Dy_o, $D\psi$ to economize on channel capacity. The other three tracking lag transducers generate analog signals z_o, θ, ϕ also sensed by the computer every 10 ms so that the trajectory of the free-running model can be recorded in six degrees of freedom. The analog channels can, of course, also be used alternatively to record any other quantities of interest, e.g. rudder angle and forces or propeller revolutions and forces.

The model locking device mentioned above consists of two hydraulically operated vertical arms mounted on the ψ-carriage and corresponding horizontal frames of pentagonal shape mounted in the model. By moving the arms out and in under manual or computer command the model can be engaged or released to serve various purposes. In a typical tracking experiment the model is initially engaged and accelerated from standstill to its approach speed with carriage assistance, then released and allowed to maneuver and finally engaged again and brought to a standstill with carriage assistance in order to make more efficient use of the available tank length. During the freerunning phase the locking device protects the tracking lag transducers and the power and signal exchange cables (hanging as loose umbilical cords from the CPMC into the model) by mechanically bounding the tracking lag despite a possible malfunction of the automatic tracking system. In this case the contact of the vertical arm with the horizontal frame generates a signal which does not automatically abort the run but is recorded by the computer and displayed on the operating console.

SOFTWARE

Comprehensive software has been developed by the authors for the freely programmable process control computer Siemens 301 to handle the multifarious tasks of carriage motion control, model motion control, hardware status monitoring, data acquisition and run evaluation. During CPMC experiments typically about 10 k words of core storage are loaded with programs (including the basic operating system) and 6 k words are available for data. As a measured quantity can usually be represented adequately by 12 bits whereas the word length is 24 bits, this means that up to about 12 k independent measurements can be stored in the core during a single run. The current status of the CPMC software has been fully documented in internal reports by Wolff (1974) and Oltmann (1975). For the present purpose it will suffice to give a brief general description of the software for Mode A. The block diagram (Fig. 7) is designed to show that the software is organized in a main program BDIE and five subprograms SWEZ, TEST, VORB, MESS and AUSW, and how these interact with each other and with the input/output and process-control periphery. The functions of the subprograms are best explained by following the chronological sequence of events during a run as illustrated in the logic flow diagram for Mode A (Fig. 8).

The first step is to read appropriate run parameters into the computer, usually from prepared paper tape. These consist mainly of up to 100 Fourier coefficients which define uniquely the functions $\dot{x}_o(t)$, $\dot{y}_o(t)$ and $\psi(t)$ over a finite interval of time T. From these the subprogram SWEZ generates the required trajectory coordinates (at 10 ms intervals) in three phases: a periodic phase determined by synthesis of the given Fourier coefficients, a run-in phase calculated to move the subcarriages from their inital resting positions to the synchronous entry points of the periodic phase, and a run-out phase to decelerate the subcarriages from their synchronous exit points of the periodic phase to the final resting positions. The complete trajectory (along with some auxiliary control information) is stored in a highly compact form in the core in a so-called run control list which uniquely determines the course of the run to follow. Now the subprogram TEST checks the run control list by simulating the required run to see for instance whether it is compatible with the available range and dynamics of the four carriages. If no errors are detected the subprogram VORB1 is started to initialize core storage and check the status of the control hardware. After errors, if any, have been corrected a lamp "ready to start" lights up on the console. The human operator may now press the START button. The subprogram VORB2 now accelerates the towing carriage to the desired mean speed $\bar{\dot{x}}_o$ and releases the brakes of the subcarriages. The basic 10 ms control loop MESS now takes over, reading out required coordinates (now calculated in real time from the run control list), reading in path errors and force data, and monitoring control hardware until the predetermined trajectory has been completed. Unless the run has been aborted owing to a bad control state triggering an automatic CPMC-HALT, which immediately brings all carriages to a standstill with the maximum permissible deceleration, the control is gracefully transferred to the subprogram VORB3 which now applies brakes to the subcarriages already at rest and decelerates the towing carriage to a standstill thereby terminating the run. During the return trip of the towing carriage and while waiting for the tank water to calm down for the next run the computer is free to evaluate the stored information and print out the results. This is handled by the subprogram AUSW, the main jobs being a systematic survey of the recorded trajectory errors and a Fourier analysis of the forces recorded during the periodic phase. Although the maximum sampling rate is 100 Hz per channel, it has been found more advantageous to record each error and force at 25 Hz over a correspondingly longer interval of time. Usually the path errors are only documented on teletype whereas the force coefficients (or the entire raw data) are punched out on paper tape for final analysis off site.

We note in passing that for the protection of the CPMC, model and instrumentation against accidental damage following wrong input, improper commands or system component failure an entire hierarchy of sophisticated safety features has been incorporated into the hardware and software, e.g. joint position and speed monitoring for preventing the carriages from transgressing their operating ranges and the model from hitting a tank wall.

SAMPLE RESULTS IN MODE A

General remarks

The following sample results in Mode A were obtained with a 1 : 25 scale model of the Mariner class vessel (Table 2) tested in the ITTC standard condition as defined by Gertler (1969). Speed U corresponded to 20 kn full-scale and propeller revolutions n were held at the self-propulsion point of the ship. The forces sensed by the dynamometer were corrected for inertial effects arising from the mass and moment of inertia of the model to the extent effective at the transducers, see Equations (5a) in Mandel (1967). The net hydrodynamic forces are plotted in Figs. 9-12 in standard nondimensional form X', Y', N' as functions of rudder angle δ, drift angle β and nondimensional yaw rate r', cf. Notation. For further discussion we must distinguish the two types of tests involved.

Steady state tests

In the steady state tests the quantity δ, β or r was held constant at selected values for a finite period of time ranging from 5 to 60 s. As the analog force signals were plotted in parallel on a strip-chart recorder it was easy to tell in each case when a nearly steady state had been achieved. Hence the interpretation of measured forces was straightforward. One only had to average the values sampled over the final portion of the steady state run. These results are indicated as discrete points in Figs. 9-12. Of course, the steady state rudder angle and drift angle tests are trivial and might as well have been done using the towing carriage alone. However, the steady state yaw rate tests were accomplished by actually varying the yaw rate periodically as a *trapezoidal* function of time, thus effectively simulating steady state rotating arm tests. This is a benefit accruing from the complete flexibility of motion control in the CPMC. In fact, not only can we simulate rotating arms of any radius from zero to infinity but also realize via input parameters different definitions of pure yaw as illustrated by the circles (zero drift angle at origin O) and crosses (zero drift angle at centre of gravity x_G) in Fig. 12. As the yaw moment is always taken about the origin O these two sets of points reflect the genuine hydrodynamic difference in the two definitions of *pure* yaw! Two other interesting exhibits in our steady state results are the appreciable asymmetry of rudder induced forces (Fig. 9) apparently caused by the single screw, and the significant dependence of X on r (Fig. 11). The former effect has been seldom observed as most experimenters were content with tests on one side only, and the latter effect contradicts previous findings of Strøm-Tejsen and Chislett (1966) as well as Chislett and Smitt (1974).

Harmonic motion tests

Here the pertinent motion variable v or r was varied as a cosinusoidal function of time for several periods $T = 2\pi/\omega$ of about 20 to 30 s. The measured hydrodynamic forces were analysed into a Fourier series in ωt. Assuming quasi-steady flow, the obvious interpretation would be that the first order sine coefficient yields the hydrodynamic mass, while all significant cosine coefficients taken together yield the complete nonlinear damping. (A Fourier analysis in the t-domain is tantamount to a least squares fit of orthogonal Chebyshev polynomials in the v- or r-domain with a natural weighting function.) This view is substantiated by the results as seen in Figs. 10-11. Each of these six diagrams contains three independent curves, each generated by synthesizing the measured cosine coefficients of the corresponding force from single runs of different amplitudes, viz. $\hat{\beta} = 6, 12, 18^\circ$ (Fig. 10) and $\hat{r}' = 0.3, 0.6, 0.9$ (Fig. 11). The mutual consistency of each triplet of curves and their almost perfect agreement with the discrete points from steady state tests supports the above interpretation and demonstrates how the CPMC yields from each single run an invariant nonlinear force response to velocity for which other techniques require a whole series of runs!

However, one puzzling feature did emerge from these experiments, in that a significant sine coefficient of third order was detected for Y in the sway tests and N in the yaw tests. Evidently this coefficient cannot be interpreted as a pure velocity response. Hence the possibility of higher order acceleration or mixed terms had to be examined as shown in Table 3. To date the most plausible explanation seems to be invoking mixed terms of the type $Y_{vv\dot{v}} \cdot v^2\dot{v}$ and $N_{rr\dot{r}} \cdot r^2\dot{r}$. Contrary to popular belief there is no theoretical difficulty in visualizing such terms as arising from viscous or wave effects. The traditional argument of Abkowitz (1964) that the only force response to acceleration is linear and independent of velocity is theoretically founded only for ideal flow in an unbounded medium. However, further research is warranted before arriving at a definite conclusion.

SAMPLE RESULTS IN MODE B

The trajectories of two zigzag maneuvers, each constructed from original computer recordings of incremental digital coordinates $Dx_o, Dy_o, D\psi$ at 100 Hz and samplings of analog signals $\delta, z_o, \theta, \phi$ at 20 Hz over a period exceeding 60 s, are reproduced graphically in Figs. 13-16 and numerically in Tables 5 and 6 in order to illustrate the capabilities of the CPMC in the tracking mode. Both runs were part of the comprehensive acceptance trials in Mode B performed with a 1 : 9 scale model of a twin-screw tug-boat (Table 4) for the compelling reason that it happend to be the most agile model tested at the HSVA during the past 20 years! If the CPMC could cope with a model yawing wildly up to ±15 °/s and zigzagging along the tank at over 1.5 m/s with course angles approaching ±50°, it could easily track any ordinary ship model. Actually the y_o- and ψ-carriages had no difficulty in tracking this model with negligible lag! The only serious difficulty was encountered following the first rudder deflection when the model decelerated in the x_o-direction at over 0.2 m/s² which was almost too much for the limited acceleration of the towing carriage and the limited range of the Δx_c-subcarriage. The problem was finally mastered by improvements in the hardware and software.

Although originally conceived as an afterthought, the tracking mode is now yielding trajectory data of unprecedented resolution ideally suited for the application of system identification techniques as discussed for example by Oltmann (1973). The task ahead is to find a judicious combination of Modes A and B for identifying a general mathematical model for simulating arbitrary ship maneuvers on the basis of a few runs with a physical model under the CPMC.

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Table 1 Principal particulars of computerized planar motion carriage (CPMC)

Drive	Main carriage with trailer	Δx_o–subcarriage	y_o–subcarriage	ψ–subcarriage
Empty mass	73000 kg	1370 kg	3250 kg	1500 kg
Mechanical range	280.0 m	±1.00 m	±7.30 m	±180.0°
Operating range	200.0 m	±0.85 m	±6.50 m	±150.0°
Maximum speed	3.0 m/s	0.80 m/s	1.90 m/s	24.0°/s
Maximum acceleration	0.2 m/s²	0.60 m/s²	0.90 m/s²	12.0°/s²
Trajectory resolution:				
Input every 10 ms	5×10^{-5} m	5×10^{-5} m	2.5×10^{-5} m	1.25×10^{-6} rev
Output every 10 ms	← 10^{-4} m →		10^{-4} m	10^{-5} rev
Tolerances:				
Trajectory error (Mode A)	← 0.01 m →		0.01 m	0.1°
Tracking lag (Mode B)	← 0.20 m →		0.20 m	8.0°

Table 2 Main dimensions of HSVA Model No. 2654 Mariner class vessel, Model scale 1:25

Length between perpendiculars	6.437 m
Beam	0.927 m
Draft forward	0.274 m
Draft aft	0.323 m
Displacement	1.064 m³
Coordinate origin aft of FP	3.170 m
Center of gravity aft of origin	0.142 m
HSVA Model Propeller No. 1379	
Right-handed single screw	
Diameter	0.268 m
Pitch ratio	0.964
Expanded area ratio	0.660
Number of blades	4

Table 4 Main dimensions of HSVA Model No. 2509 Twin-screw tug-boat, Model scale 1:9

Length between perpendiculars	3.792 m
Length of waterline	4.113 m
Beam	1.016 m
Draft mean	0.389 m
Displacement	0.852 m³
Coordinate origin aft of FP	1.896 m
Center of gravity aft of origin	0.050 m
HSVA Model Propellers No. 1707/1708	
Rotatable Kort nozzle rudders	
Diameter	0.217 m
Pitch ratio	1.260
Expanded area ratio	0.580
Number of blades	4

Table 3 Interpretation of hydrodynamic forces measured in forced harmonic motion

U = const, $v = \hat{v}\cos(\omega t)$, $r = 0$, n = const, $\delta = 0$

Possible term		Measured force components $\cos(2k\omega t)$	$\cos\{(2k+1)\omega t\}$	$\sin(2k\omega t)$	$\sin\{(2k+1)\omega t\}$
v^{2i}		x			
v^{2i+1}			x		
	$\dot{v}^{2j}$	x			
	$\dot{v}^{2j+1}$				x
v^{2i}	$\dot{v}^{2j}$	x			
v^{2i}	$\dot{v}^{2j+1}$				x
v^{2i+1}	$\dot{v}^{2j}$		x		
v^{2i+1}	$\dot{v}^{2j+1}$			x	
Significant k observed in					
$X(t)$		0,2	1		1
$Y(t)$			1,3		1,3
$N(t)$		0	1		1

Table 5 Trajectory of $30^{o}/10^{o}$ zigzag maneuver

PROGRAM: PO/LSTR3

MSVA-MODEL NO. 2509, RUN NO. 7 (24.10.75)

INITIAL SPEED U0= 2,004 M/S

T	DELTA	U	V	R	PSI	BETA	X0	Y0
S	DEG	M/S	M/S	DEG/S	DEG	DEG	M	M
0	-,156	2,004	,000	,000	,000	,000	,000	,000
1	-,133	1,990	-,009	,216	,144	,259	1,999	-,001
2	-,244	1,991	-,016	,324	,432	,461	3,986	-,002
3	-,133	1,985	-,014	,468	,792	,417	5,973	,003
4	-11,200	1,970	-,074	3,852	2,520	2.142	7,954	,013
5	-23,933	1,910	-,196	10,332	9,720	5,868	9,903	,075
6	-20,711	1,797	-,284	13,716	22,500	8,993	11,748	,339
7	-8,511	1,749	-,288	10,368	34,884	9,364	13,433	,937
8	3,711	1,738	-,202	3,924	42,084	6,626	14,941	1,839
9	16,133	1,756	-,045	-3,852	41,976	1.478	16,312	2,929
10	28,000	1,755	,139	-11,484	33,948	-4,524	17,660	4,066
11	28,111	1,704	,268	-15,048	20,196	-8,939	19,181	5,047
12	28,889	1,613	,328	-15,804	4,608	-11,481	20,653	5,702
13	28,822	1,516	,348	-15,516	-11,052	-12,931	22,231	5,953
14	20,933	1,454	,343	-14,796	-26,352	-13,275	23,741	5,810
15	8,867	1,457	,312	-11,124	-39,564	-12,107	25,132	5,292
16	-3,178	1,496	,220	-4,644	-47,448	-8,382	26,377	4,462
17	-15,489	1,568	,080	3,096	-48,132	-2,926	27,508	3,417
18	-27,044	1,606	-,098	11,258	-40,644	3.475	28,631	2,283
19	-27,889	1,568	-,236	14,832	-27,036	8,570	29,651	1,247
20	-28,067	1,509	-,302	15,480	-11,736	11,332	31,204	,475
21	-28,178	1,439	-,334	15,408	3.708	13,058	32,648	,052
22	-25,778	1,371	-,345	15,444	19,152	14,121	34,084	-,004
23	-14,778	1,343	-,320	13,104	33,696	13.402	35,436	,298
24	-2,733	1,383	-,258	8,208	44,352	10,577	36,667	,935
25	9,622	1,468	-,165	1,224	49,104	6,411	37,790	1,034
26	21,756	1,538	,005	-6,948	46,044	-,194	38,854	2,903
27	28,533	1,568	,164	-13,212	35,352	-5,971	39,967	3,997
28	28,778	1,531	,272	-15,516	20,592	-10,062	41,229	4,925
29	28,733	1,473	,318	-15,480	4.932	-12,168	42,622	5,548
30	28,956	1,423	,332	-15,372	-10,440	-13,146	44,078	5,805
31	20,622	1,387	,335	-14,616	-25,632	-13,574	45,511	5,692
32	8,622	1,395	,288	-10,620	-38,412	-11,669	46,846	5,217
33	-3,489	1,462	,203	-4,536	-46,008	-7,914	48,058	4,433
34	-15,567	1,544	,067	3,348	-46,548	-2,492	49,190	3,432
35	-27,156	1,575	-,113	11,268	-38,844	4,103	50,313	2,341
36	-27,933	1,545	-,241	14,760	-25,344	8,866	51,535	1,350
37	-28,111	1,481	-,305	15,516	-10,044	11,553	52,890	,622
38	-28,133	1,413	-,328	15,516	5,508	13,076	54,321	,245
39	-24,667	1,347	-,333	15,336	20,952	13,875	55,735	,236
40	-12,578	1,340	-,303	12,384	35,064	12,741	57,062	,585
41	-,489	1,392	-,234	6,876	44,748	9,532	58,273	1,259
42	11,778	1,483	-,125	-,216	48,060	4,805	59,387	2,182
43	24,067	1,544	,043	-8,532	43,452	-1,601	60,465	3,252
44	28,244	1,547	,199	-13,932	31,572	-7,340	61,610	4,306
45	28,733	1,513	,297	-15,372	16,668	-11,094	62,903	5,163
46	28,822	1,458	,327	-15,300	1,296	-12,647	64,328	5,704
47	28,756	1,399	,340	-15,408	-14,040	-13,642	65,773	5,880
48	17,844	1,369	,334	-13,968	-29,052	-13,705	67,176	5,689
49	5,711	1,394	,278	-9,288	-40,788	-11,285	68,474	5,146
50	-6,400	1,471	,186	-2,700	-46,800	-7,194	69,658	4,314
51	-18,733	1,544	,033	5,328	-45,432	-1,219	70,774	3,292
52	-27,378	1,566	-,141	12,492	-35,964	5,143	71,910	2,218
53	-27,489	1,532	-,257	14,940	-21,924	9,516	73,166	1,284
54	-28,222	1,470	-,314	15,408	-6,588	12,042	74,545	,633
55	-28,289	1,398	-,337	15,372	8,820	13,569	75,979	,333
56	-23,600	1,336	-,340	15,156	24,192	14,295	77,380	,395
57	-11,444	1,334	-,301	11,916	37,944	12,714	78,683	,808
58	,711	1,392	-,231	6,480	47,124	9,439	79,857	1,535
59	13,178	1,482	-,119	-,900	49,932	4,607	80,933	2,502
60	25,489	1,544	,058	-9,108	44,640	-2,144	81,973	3,607

Table 6 Trajectory of $20^{o}/20^{o}$ zigzag maneuver

PROGRAM: PO/LSTR3

MSVA-MODEL NO. 2509, RUN NO. 15 (24.10.75)

INITIAL SPEED U0= 2,005 M/S

T S	DELTA DEG	U M/S	V M/S	R DEG/S	PSI DEG	BETA DEG	X0 M	Y0 M
0	-,133	2,005	-,001	,000	,000	,029	,000	,000
1	-,156	1,994	-,009	,252	,180	,266	2,001	-,001
2	-,133	1,997	-,014	,360	,468	,411	3,993	-,002
3	-,400	1,991	-,022	,396	,828	,627	5,987	,001
4	-13,022	1,978	-,085	4,320	2,772	2,454	7,976	,005
5	-18,756	1,926	-,196	9,792	10,296	5,822	9,931	,070
6	-18,600	1,855	-,265	11,628	21,240	8,132	11,813	,354
7	-10,911	1,804	-,280	10,556	32,832	8,825	13,565	,936
8	1,444	1,787	-,216	5,040	40,824	6,881	15,143	1,818
9	13,889	1,802	-,074	-2,664	41,940	2,363	16,575	2,907
10	19,111	1,817	,094	-9,108	35,388	-2,962	17,965	4,062
11	19,178	1,809	,210	-11,592	24,696	-6,629	19,450	5,112
12	19,244	1,773	,274	-12,168	12,672	-8,774	21,066	5,918
13	19,222	1,733	,298	-12,204	,432	-9,754	22,772	6,404
14	19,267	1,698	,305	-12,096	-11,700	-10,173	24,504	6,536
15	17,889	1,673	,309	-12,060	-23,760	-10,476	26,200	6,316
16	7,911	1,666	,282	-9,684	-35,028	-9,618	27,795	5,757
17	-4,311	1,692	,197	-3,780	-41,632	-6,632	29,250	4,894
18	-16,911	1,730	,045	4,032	-41,616	-1,493	30,598	3,833
19	-18,667	1,758	-,110	9,648	-34,236	3,571	31,947	2,721
20	-18,800	1,746	-,216	11,016	-23,148	7,046	33,402	1,726
21	-18,844	1,714	-,268	12,240	-10,944	8,872	34,983	,982
22	-18,911	1,680	-,292	12,132	1,260	9,863	36,648	,560
23	-18,867	1,654	-,295	11,916	13,320	10,114	38,335	,481
24	-17,622	1,625	-,302	12,096	25,308	10,517	39,978	,739
25	-7,378	1,619	-,273	9,792	36,648	9,588	41,509	1,325
26	5,222	1,652	-,190	3,816	43,560	6,556	42,900	2,207
27	18,089	1,701	-,043	-4,140	43,308	1,454	44,188	3,287
28	19,289	1,733	,110	-9,828	35,748	-3,644	45,402	4,418
29	19,400	1,735	,209	-11,844	24,624	-6,874	46,895	5,437
30	19,489	1,721	,266	-12,132	12,528	-8,791	48,455	6,217
31	19,556	1,692	,285	-12,024	,432	-9,569	50,117	6,584
32	19,467	1,668	,298	-12,024	-11,556	-10,136	51,815	6,812
33	19,533	1,650	,304	-12,240	-23,688	-10,423	53,484	6,596
34	9,400	1,639	,276	-10,260	-35,316	-9,539	55,052	6,043
35	-3,067	1,666	,197	-4,428	-42,768	-6,740	56,476	5,180
36	-15,844	1,707	,056	3,240	-43,344	-1,681	57,787	4,108
37	-18,756	1,735	-,097	9,432	-36,396	3,211	59,083	2,971
38	-18,667	1,727	-,215	11,772	-25,452	7,088	60,483	1,935
39	-18,578	1,703	-,270	12,132	-13,392	9,010	62,017	1,132
40	-18,689	1,671	-,287	11,952	-1,332	9,755	63,651	,638
41	-18,556	1,642	-,298	11,916	10,584	10,278	65,323	,480
42	-18,133	1,610	-,305	11,988	22,572	10,709	66,965	,654
43	-10,000	1,600	-,288	10,620	34,236	10,213	68,512	1,153
44	2,156	1,626	-,220	5,400	42,444	7,703	69,925	1,958
45	14,467	1,672	-,088	-2,232	43,992	3,014	71,223	2,985
46	18,600	1,712	,072	-8,748	37,944	-2,393	72,497	4,103
47	18,711	1,729	,185	-11,268	27,612	-6,106	73,867	5,151
48	19,289	1,715	,259	-11,952	15,840	-8,582	75,382	5,999
49	19,356	1,689	,293	-12,060	3,816	-9,848	77,012	6,562
50	19,257	1,662	,301	-12,096	-8,280	-10,248	78,697	6,794
51	19,289	1,642	,308	-12,168	-20,448	-10,625	80,370	6,681
52	12,244	1,623	,292	-11,124	-32,364	-10,202	81,963	6,227
53	,022	1,644	,229	-6,156	-41,184	-7,936	83,420	5,453
54	-12,467	1,687	,102	1,044	-43,704	-3,456	84,751	4,441
55	-18,733	1,722	-,058	8,244	-38,520	1,937	86,036	3,321
56	-18,800	1,724	-,183	11,304	-28,404	6,047	87,400	2,258
57	-18,911	1,705	-,257	12,024	-16,560	8,565	88,900	1,394
58	-18,889	1,678	-,286	12,024	-4,536	9,682	90,510	,815
59	-18,933	1,640	-,297	11,952	7,452	10,274	92,174	,564
60	-18,933	1,609	-,302	12,024	19,440	10,625	93,822	,649

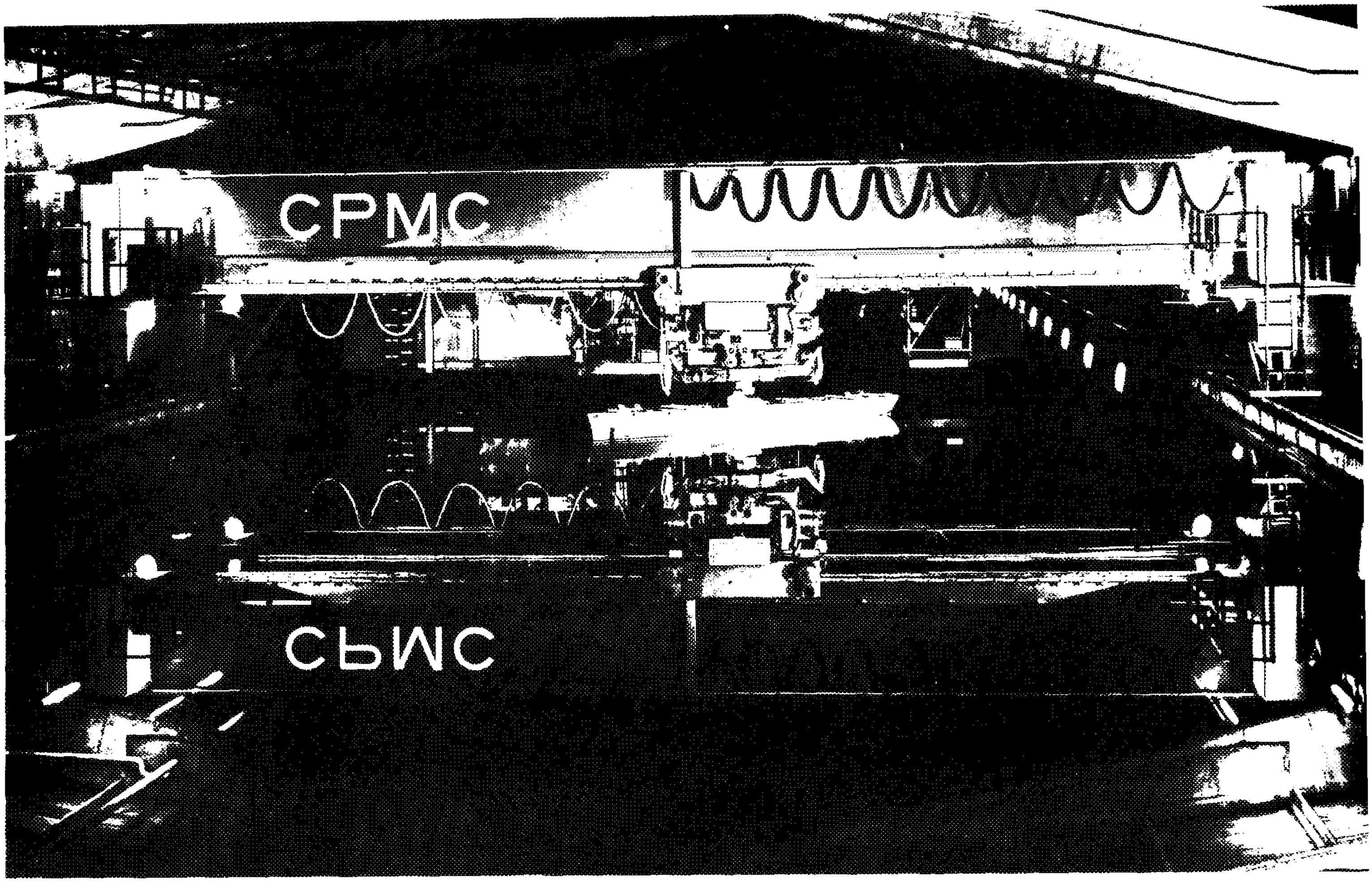

Fig. 1 Front view of CPMC

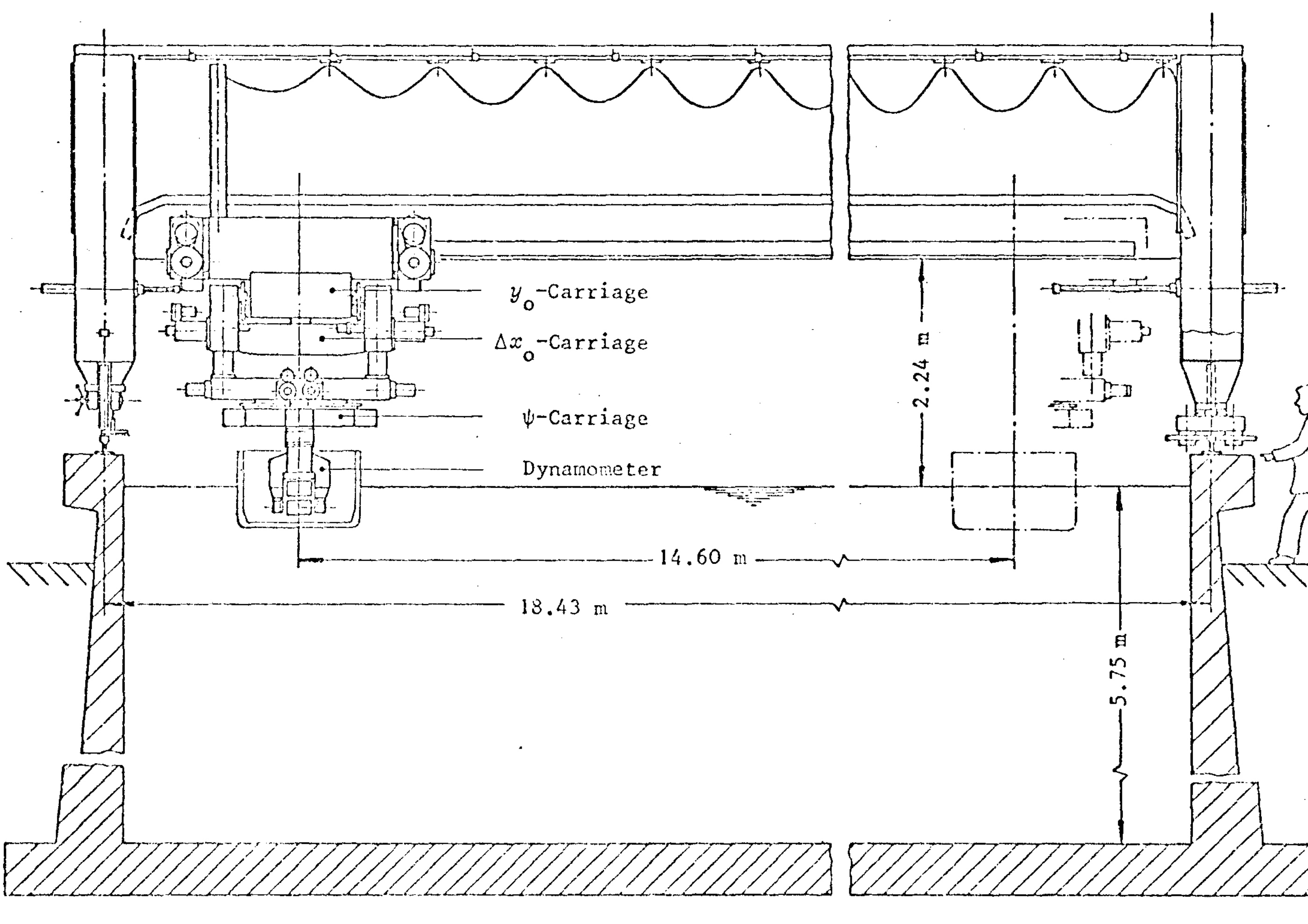

Fig. 2 Schematic of CPMC

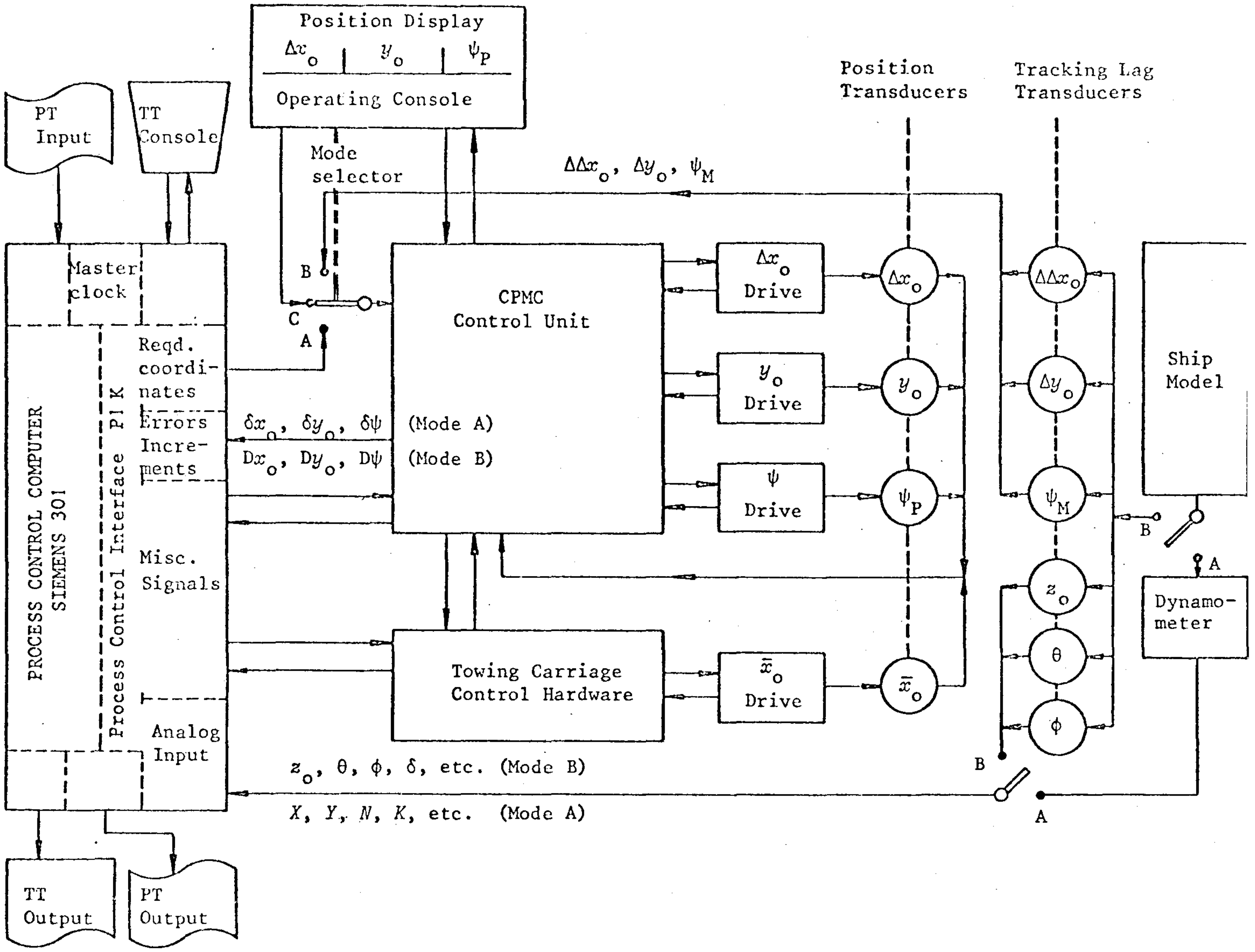

Fig. 3 System flow chart

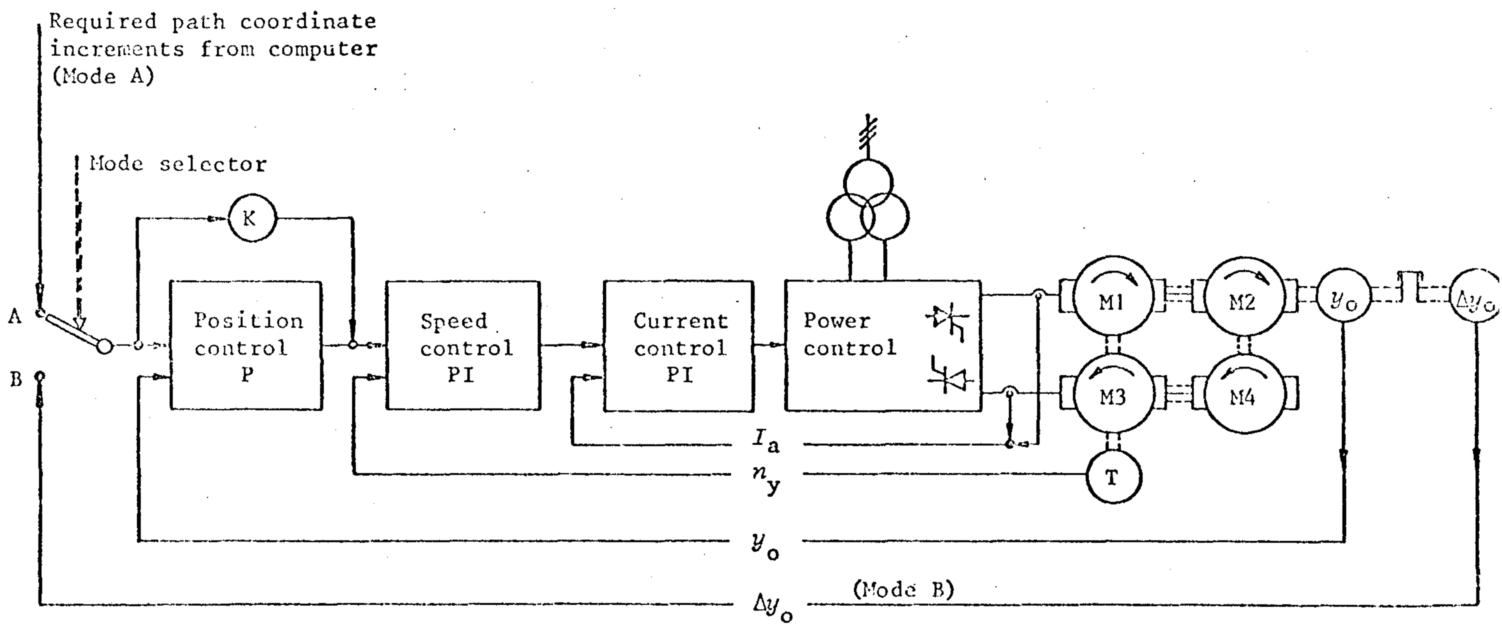

Fig. 4 Schematic of control loop (y_o-subcarriage)

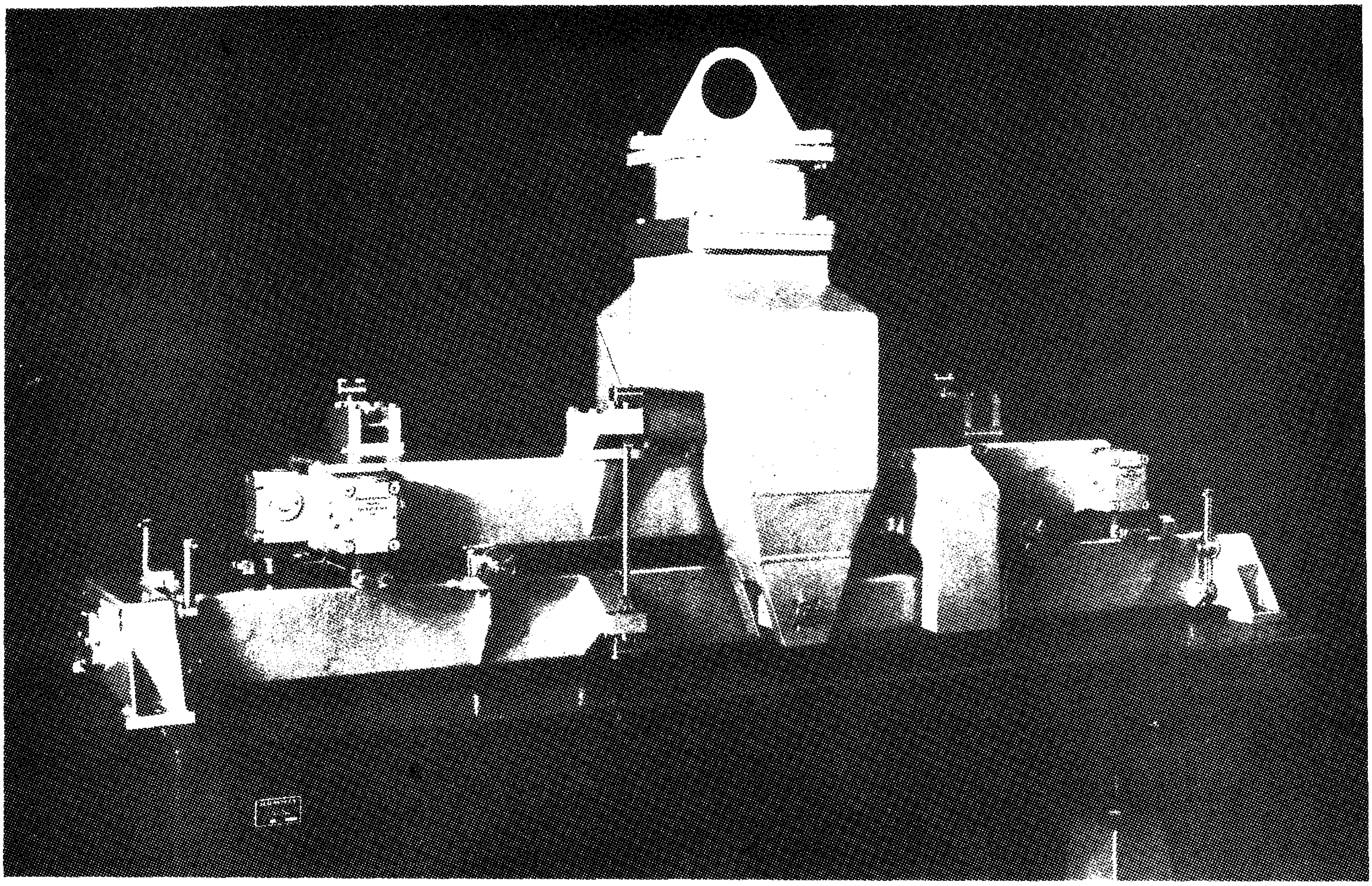

Fig. 5 Dynamometer and guide (Mode A)

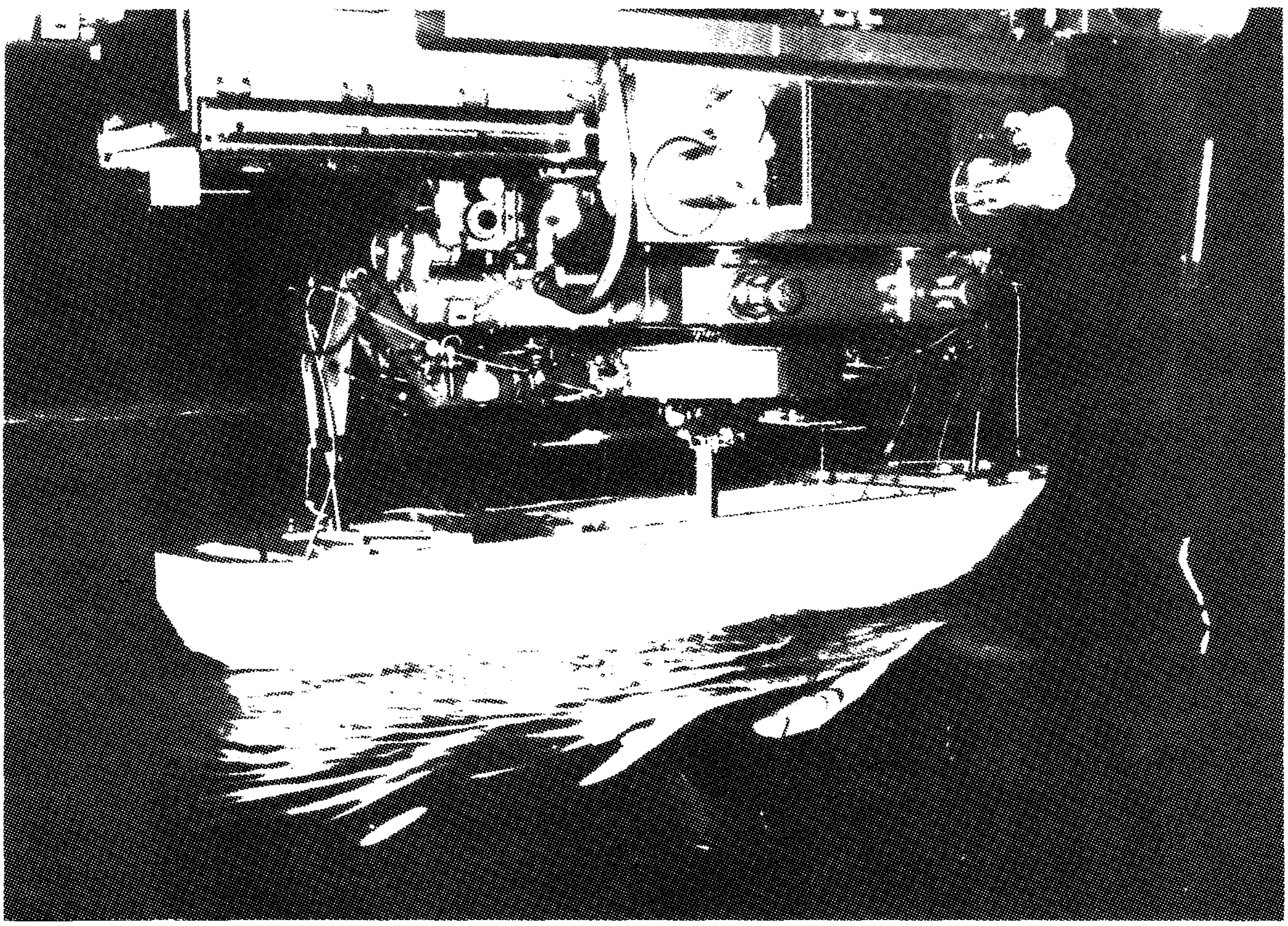

Fig. 6 Typical run in Mode B

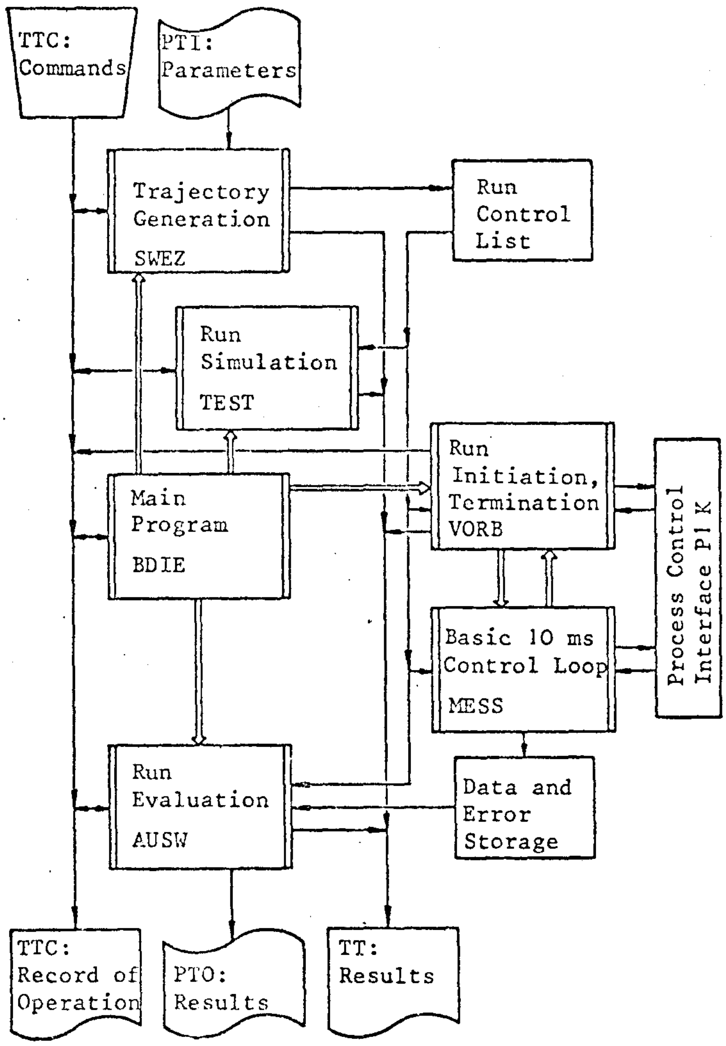

Fig. 7 Organization of software for Mode A

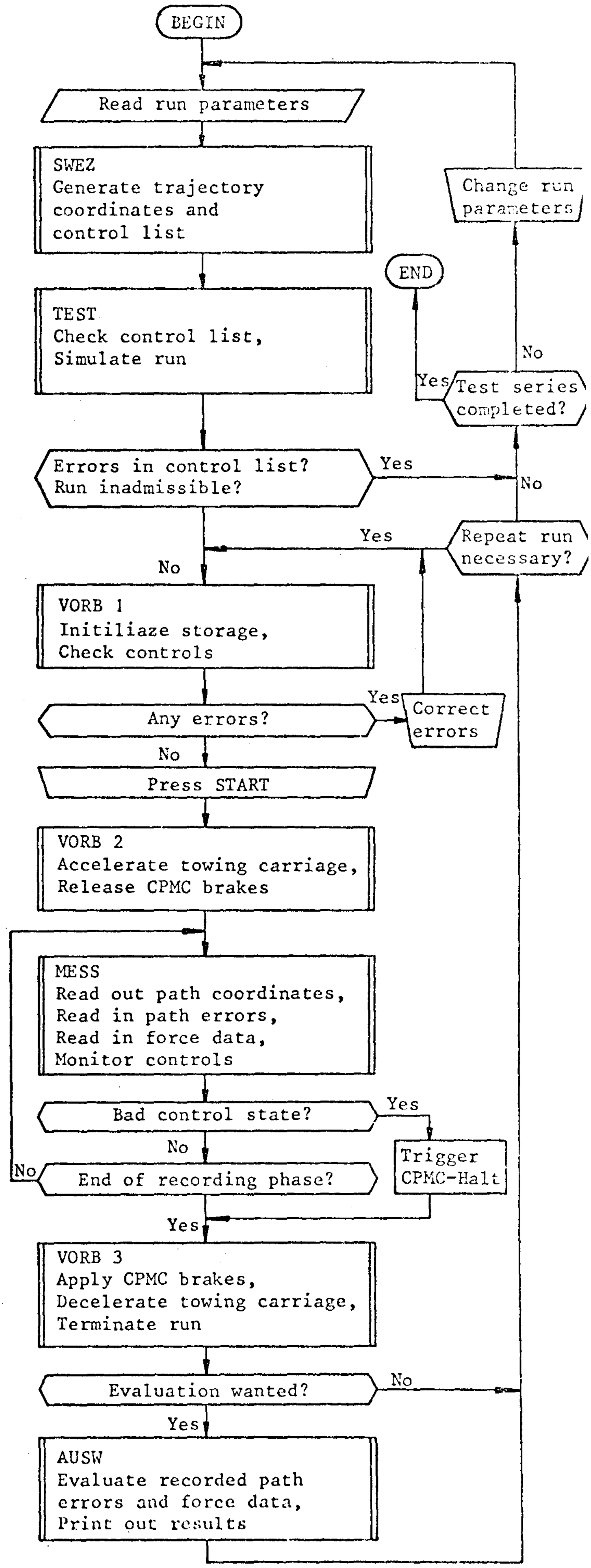

Fig. 8 Logic flow diagram for Mode A

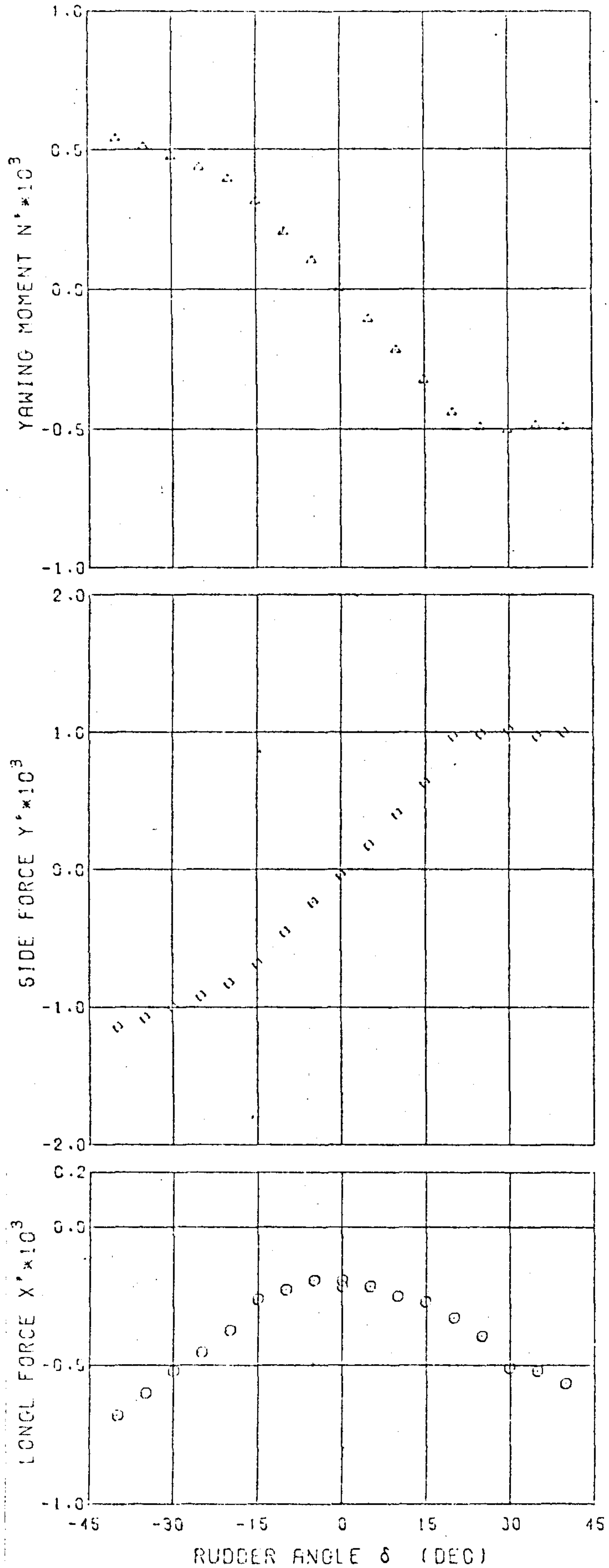

Fig. 9 Results from static rudder angle test, HSVA-Model No. 2654

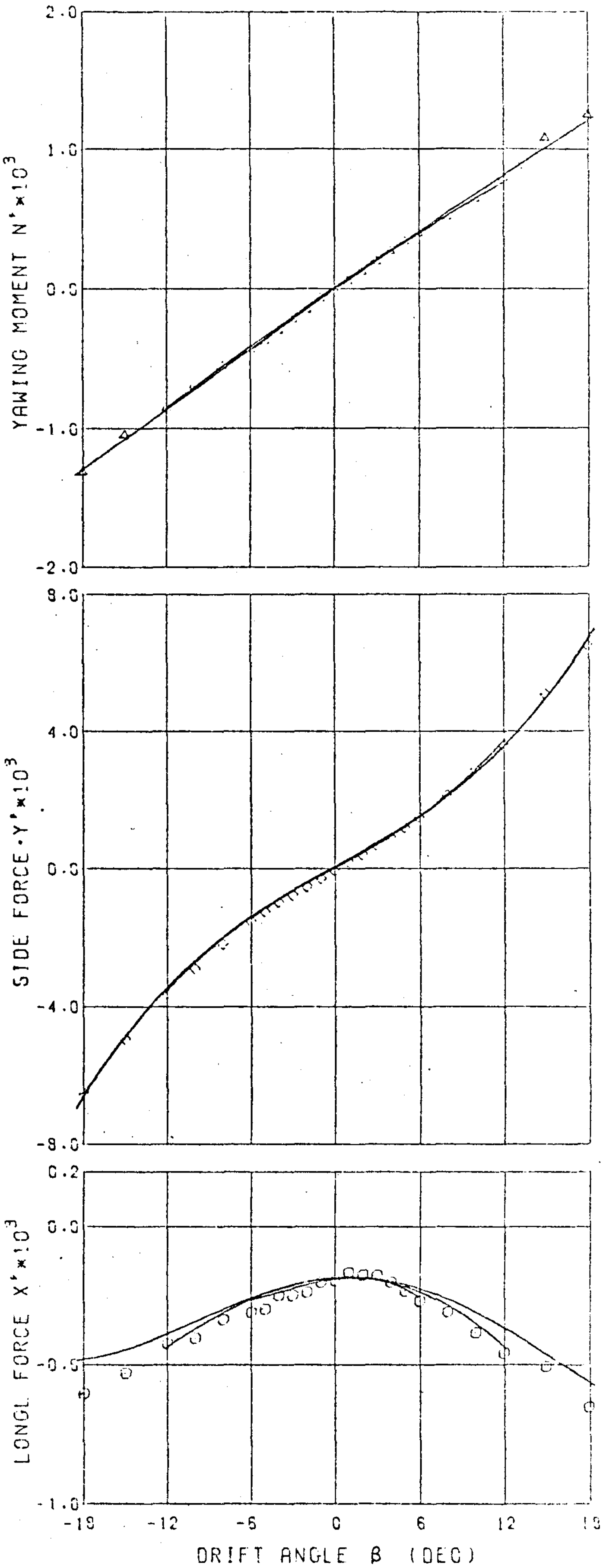

Fig. 10 Results from pure sway test, HSVA-Model No. 2654

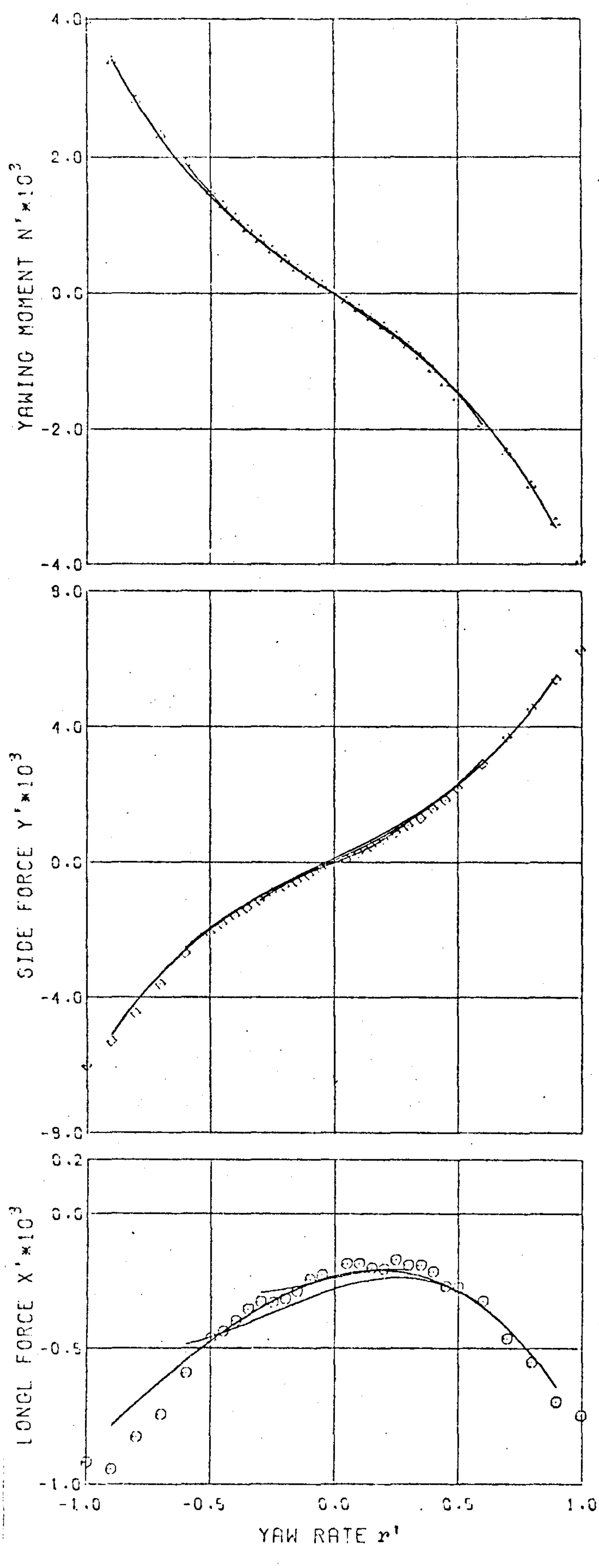

Fig. 11 Results from pure yaw test, HSVA-Model No. 2654

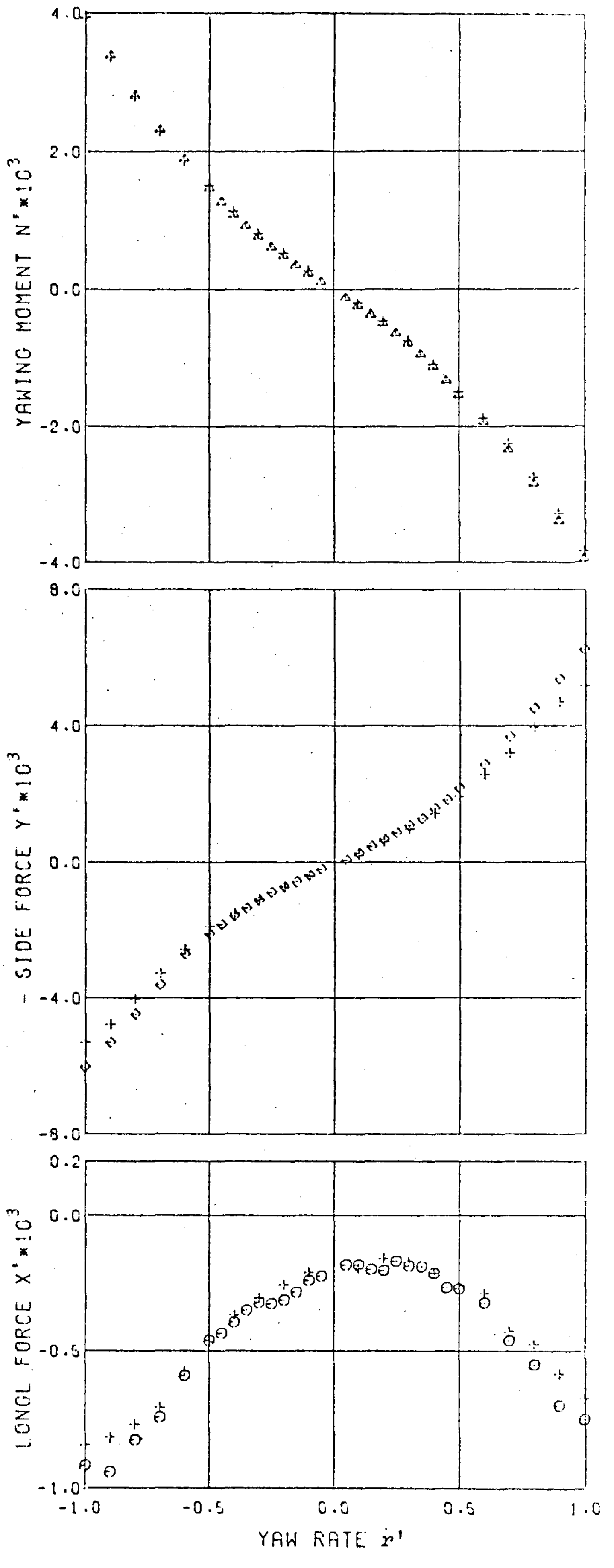

Fig. 12 Results from pure yaw test, HSVA-Model No. 2654

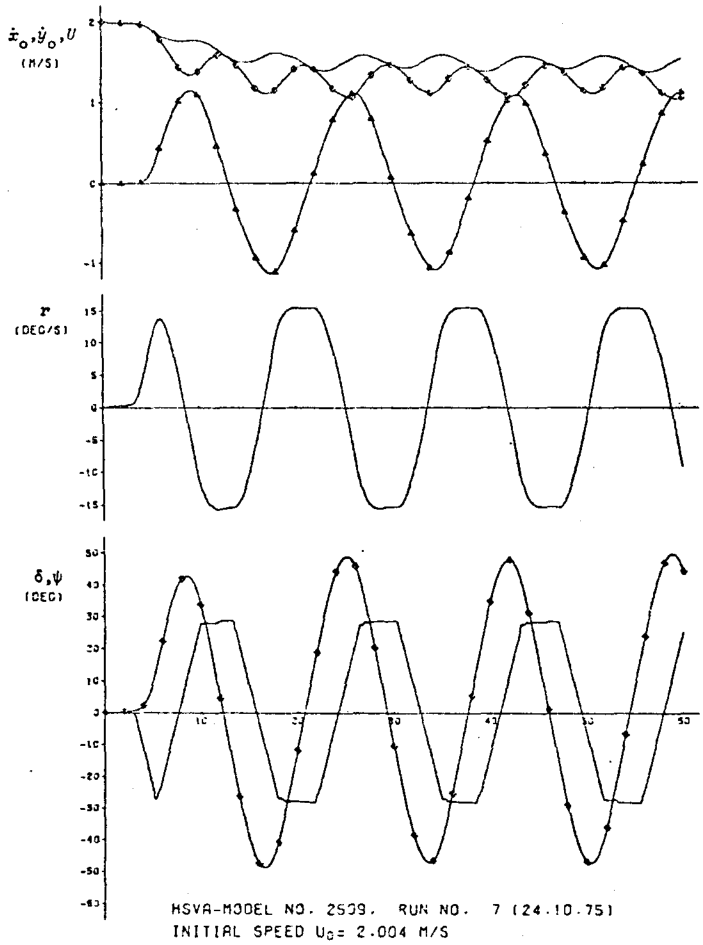

Fig. 13 Trajectory of $30^{\circ}/10^{\circ}$ zigzag maneuver

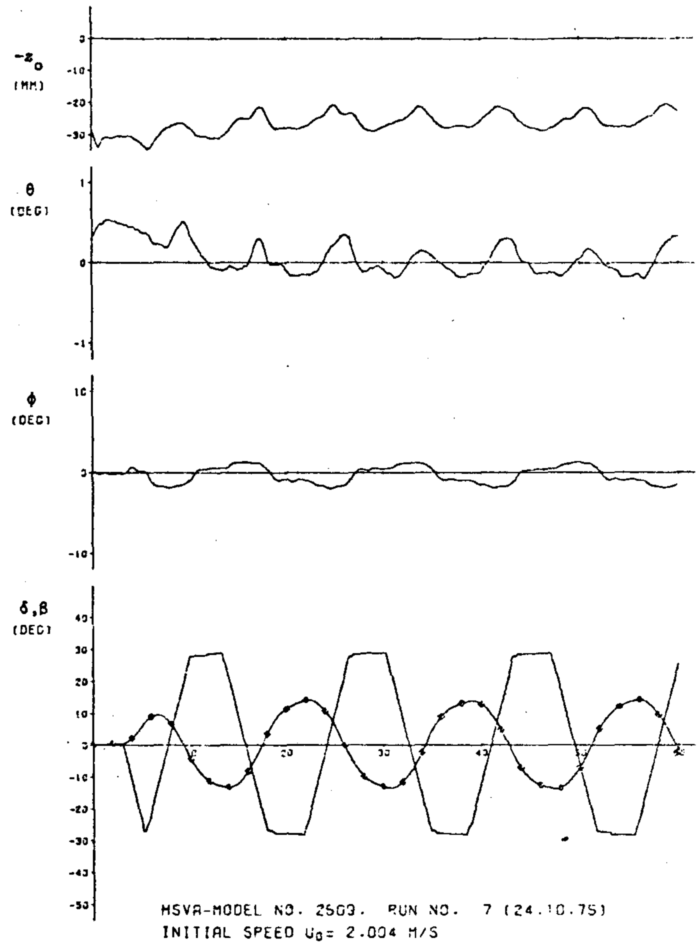

Fig. 14 Trajectory of $30^{\circ}/10^{\circ}$ zigzag maneuver

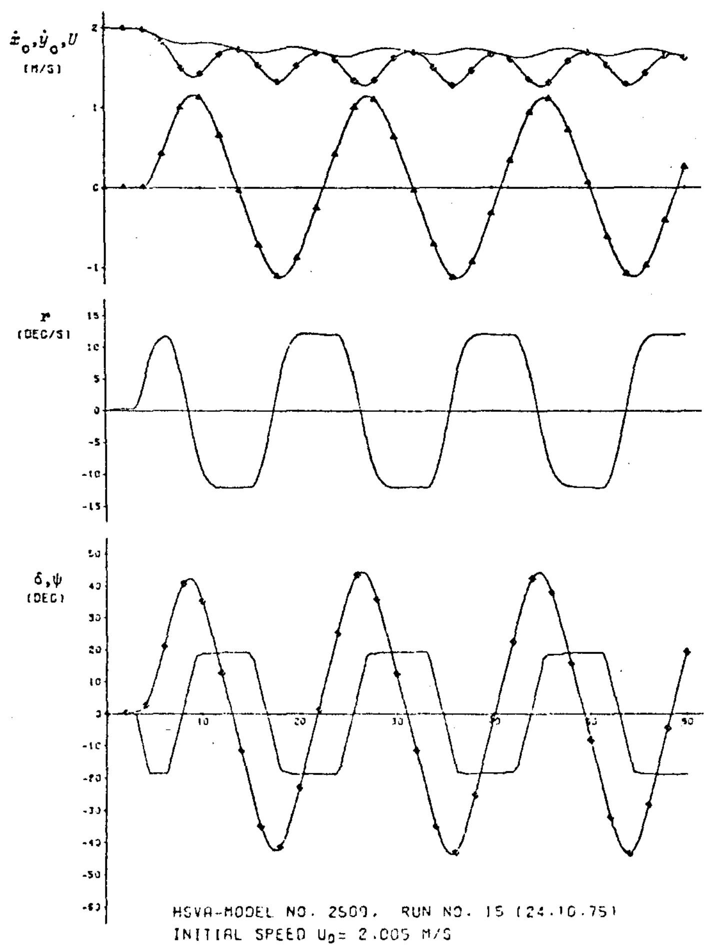

Fig. 15 Trajectory of $20^{\circ}/20^{\circ}$ zigzag maneuver

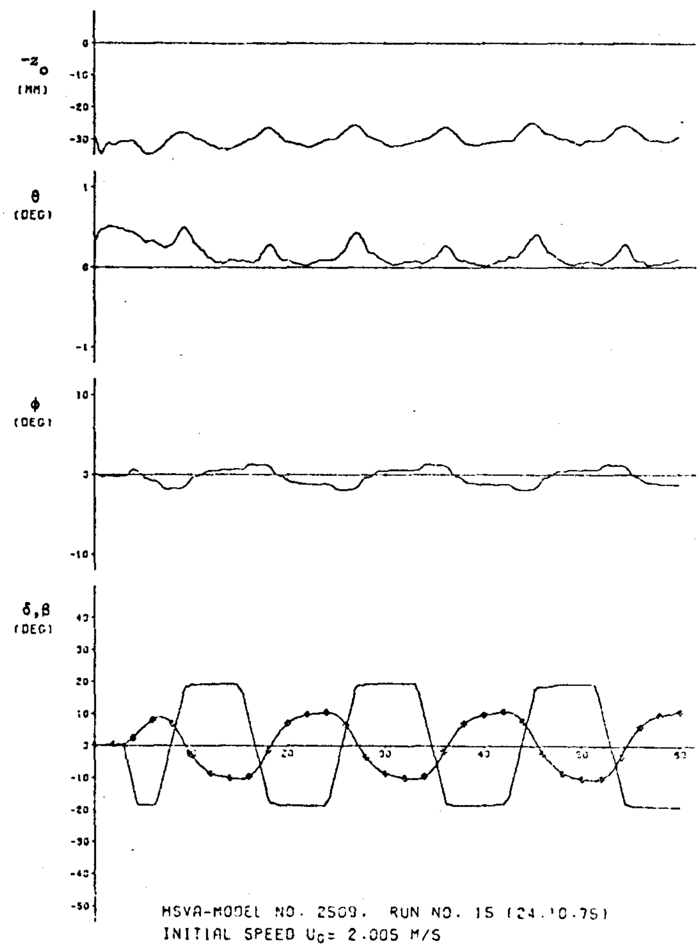

Fig. 16 Trajectory of $20^{\circ}/20^{\circ}$ zigzag maneuver