Development of Constant-Force Tank-Testing Techniques and Associated Instrumentation

Part I Towing System Development

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Declaration

I certify that the work presented in this thesis, except where explicitly credited to others, is of my own commission in both substance and composition.

[redacted]

To: Peter, Isabelle,
Eric & Ingrid.

Abstract

Over the last ten years there has been a resurgence of interest in the detailed experimental testing of sailing yacht designs. Part 1 of this thesis reports work carried out with the aim of improving test realism and data quality. Reduction of the cost of testing by increasing the rate of data generation is also reported.

Constant-Force towing methods in oblique seas were developed. Both Constant-Force and Constant-Velocity data is presented along with a detailed comparison between the two modes. This comparison shows an apparent, frequency dependent, difference of up to +-2% between the two testing modes.

During the work on yacht testing techniques a novel stiff DC loadcell was developed to meet the specific requirements of the tank testing apparatus. Part 2 of this thesis reports the work carried out on the development of this transducer. The final prototype achieved a full scale deflection of 10 microns with an overload factor of more than 20 times the rated load.

Notes on the Layout of this Thesis

This thesis is presented in two separate parts.

Part I reports development of new tank-testing methods for the experimental prediction of yacht performance. This is the main topic of this thesis.

In the course of development of the final towing system there was a requirement for a very stiff loadcell. After assessing various options it was decided to develop a custom loadcell in-house. This took longer than expected and as such has become a major part of the overall project. The definitive loadcell prototype was used in the final towing system.

Part II of this thesis reports the development of the custom loadcell. A patent has been applied for so the information contained in Part II is of a commercially sensitive nature. For this reason the two parts are bound separately in order that Part II can be put on restricted access until the patent application has been processed.

All Figures are at the end of the appropriate Chapters.

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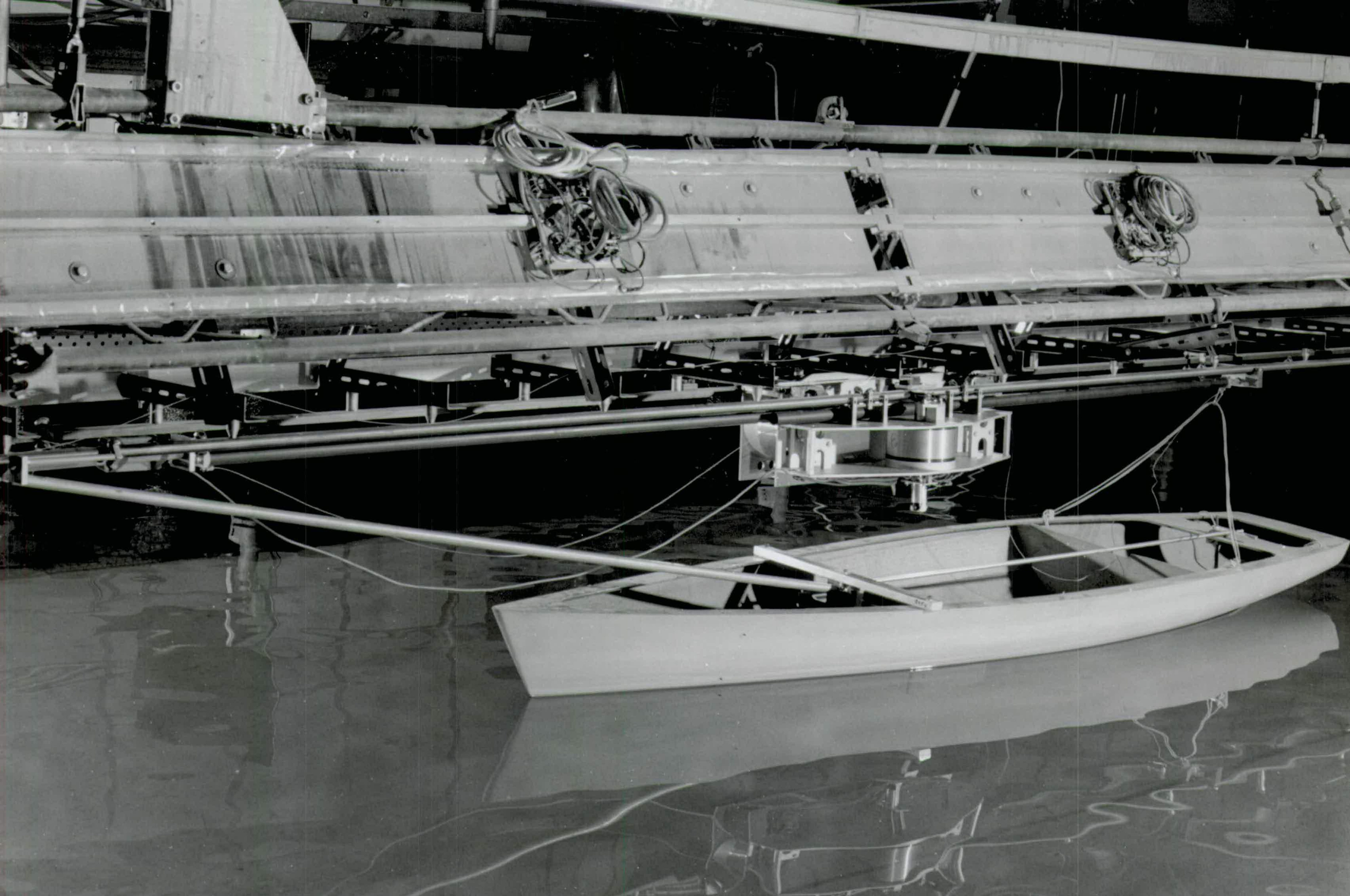
Frontispiece

The photograph overleaf shows the final towing system as described in Chapters 4 to 6.

The model under test is one of the Port Pendennis Challenge 1/9th scale AC Class models used extensively during the later stages of the project. Directly above the model can be seen the towing system central car. This car houses the main drive motor (axis vertical, between horizontal plates), the two lightweight car drive motors (axes horizontal, between vertical plates), the electronics bay (also between the vertical plates) and the assorted instrumentation for control of the system and data acquisition.

The front and rear lightweight cars and their carbon fibre drive rods can be seen on either side. The angled towing/roll damping reaction rod and elastic brakeline link the test model to the system. Also visible is the mechanical automatic steering system linking the tow rod to the tiller arm.

Note also the track construction details and (extreme top left corner) the lower end of one of the adjustable strops suspending the sections of wireless mast that form the backbone of the track.



Chapter 1. Introduction, Review of Field, Project Aims.

1.0 Chapter Summary

This opening Chapter begins with a short summary of the contents of Part I of this thesis. The Chapter then moves on to introduce the field of yacht resistance testing before summarising and formulating a broad set of aims for the project.

1.1 Chapter Breakdown

Chapter 1. is summarised above.

Chapter 2. describes the first Constant-Force towing system developed by the author as part of a submission for the Final Year Honours Degree course at the University of Edinburgh.

Chapter 3. details development of this first system on starting study towards this thesis. Development of the mechanical, control and testing techniques are reported and results are presented for the new configuration. Finally the Chapter describes further improvements that were investigated before a decision was made to build an entirely new system.

Chapter 4. discusses conceptual designs for such a new system and leads to a description of the layout finally chosen.

Chapter 5. describes all aspects of the new system's mechanical, electronic and computing design and development.

Chapter 6. reports the experimental programme carried out using the new system. Results are presented for both Constant-Force and Constant-Velocity testing modes. Also reported is a detailed comparative study of the two techniques.

Chapter 7. concludes Part I of the thesis and makes recommendations for future work.

1.2 Introduction

In order to arrive at the reasons behind the work reported in the main body of this thesis a short review of the historical development of this field of study is required.

1.2.1 Historical

All marine vehicles experience resistance to motion. The minimisation of this drag is often of prime importance to naval architects. Maximum speed is the objective of warship designers. In commercial applications drag reductions lead to cost savings and shorter transit times. In the world of yacht racing similar reductions can be the difference between success and failure.

Historically designers have iterated towards optimum solutions by bitter experience with full scale prototypes. However, over the past 300 years the various influences have resulted in the rapid development of techniques for the assessment and minimisation of resistance at the design stage.

The complete resistance equation is very complicated, thus initial investigations were entirely experimentally based. Leonardo de Vinci carried out tests on three different hull forms as early as the 15th century [1]. The next known use of models to investigate ship resistance were qualitative experiments made by Samuel Fortrey in the 17th century [1].

He used small wooden models towed by falling weights. From this time on there was a steady growth of interest in model experiment work. Colonel Beaufoy, under the auspices of the Society for the Improvement of Naval Architecture, founded in London in 1791, carried out between nine and ten thousand towing experiments in the Greenland Dock [1].

Throughout this period the method of gravity, or falling weight towing was used. The "father" of accurate tank testing, William Froude, was the first to become dissatisfied with this method. In 1868 the British Admiralty accepted his plans for the building of a large tank, probably the first purpose built facility for resistance testing. This pioneering facility was also the first to be equipped with a mechanically driven towing carriage and accurate instrumentation. As such it is usually regarded as the forerunner of all modern tanks today [1]. At the time of its inception model tests had fallen into widespread disrepute due to their apparent unreliability. With his new facility Froude showed that this was due to wrong methods of analysis and incorrect scaling up of model results. His resulting "law of comparison" has become the basis for all model work.

By the end of the 19th century there were perhaps five dedicated model testing tanks in the world. Now the number of such facilities runs into the hundreds, and they are regarded as vital tools for the research of all aspects of ship and yacht design.

Recently with the decline of the ship building industry the commercial need for so many tanks has reduced and many are faced with closure. However, the recent resurgence of interest from yacht designers and other new emerging areas of naval architecture has given many tanks a new lease of life.

The rapid advances in theoretical fluid dynamics brought about by the aviation industry allowed a new approach to be taken early this century. Modern computing power and continued improvement of numerical formulations has brought these techniques to the forefront of the field as discussed later in this review.

1.2.2 Components of Resistance

In order to minimise resistance some initial understanding of the various loss mechanisms associated with hydrodynamic drag is required. Froude was one of the first to try and separate these various components.

These can be summarised as follows:

- Frictional resistance, due to the viscous drag of fluid flowing over the surface of the vessel.

- Form drag, due to the physical displacement of water as the vessel moves.

- Wave resistance, due to the radiating pattern of waves generated by a vessel moving on or near the air/water interface.

- The added resistance incurred as a vessel moves through the wind generated wave system.

- In addition yachts experience a lift-induced drag due to the large side-force generated by the sails at most sailing angles.

Froude identified these separate components and was the first to realise that they must be scaled separately from model to full scale for accurate reliable, predictions to be made.

Each of these components has been the subject of much detailed research. This thesis is mainly concerned with the portion of added resistance due to the presence of wind generated surface waves. This area thus requires some additional discussion at this stage.

1.2.3 Expansion of Added Resistance component

Added resistance itself can be broken down into several different mechanisms as follows:

- Reflection or diffraction of the incident wave system by the vessel.

- Motions of the vessel in response to the waves dissipating energy, in particular heave and pitch.

- Side slip or drift angle due to wave action.

- Interaction of the driving forces with the waves and motions, for example propeller racing or the reduction of aerodynamic efficiency of sails.

- Increase in induced drag due to wave action.

- Motion of the control surfaces, for example the rudder, in response to the wave system.

- Wind induced drag on the superstructure and rigging.

Once again research teams have addressed all of these topics but generally in less detail than the more fundamental "calm water" resistance components. This is mainly due to the non-steady nature of wave interactions complicating the loss process.

Up until relatively recently it was assumed that a vessel that was fast in calm water would also be fast in rough conditions. This is clearly a dangerous assumption and over the last 30 years or so a renewed effort has been made to model and understand this complicated area.

1.2.4 Development of Modern Testing Methods

Since the early days of resistance testing towing methods have undergone a major revolution. In order to give a background to the reason for this project some detail of current "state-of-the-art" testing is required.

Towing systems fall into two main categories:

- Gravity or falling weight methods.
- Constant velocity methods.

As described earlier the first towing systems were exclusively of the gravity or falling weight type. A typical set up is shown in Figure 1.1. Wells are provided at both ends, and to limit their depth suitable gearing is introduced between the tow line and weight. The towing force is the difference between the tow line tension at either end minus the system friction. In order to accelerate and decelerate the model rapidly, and thus maximise run length for a given tank, an arrangement also shown in Figure 1.1 was often incorporated. Other methods included the floating off and on of accelerating and decelerating weights in mercury baths. Model speed is recorded between two fixed points after steady state has been achieved.

Later gravity systems were as shown in Figure 1.2. Here the towing weight is chosen to approximately match the model speed with the mechanically driven towing carriage. The difference force is applied by the low rate towing spring as indicated. Time and distance records along with the towing weight and spring deflection are used to generate resistance data.

Most modern towing systems use the constant velocity method. Here the model is attached to a towing carriage in such a way as to allow no speed variation. The model is free to pitch and heave but is restricted in all other degrees of freedom. The carriage is mechanically driven at a constant velocity. Resistance is measured using a dynamometer. There are many types of such dynamometer but nowadays most are of the electrical resistive strain gauge type. A typical "modern" set-up is shown in Figure 1.3.

For detailed testing of yachts various other features are included. Due to the action of the its sails the yacht hull will usually experience a sideways drift, or leeway, and a heel angle or tilt. Yacht dynamometers include mechanisms for presetting these, and associated instrumentation to measure the resulting side force and yaw moment. A typical yacht dynamometer is shown in Figure 1.4.

There have been a few attempts to develop towing systems specially designed to cope with the problems specific to yacht work. These will be discussed later in this review.

1.2.5 Application of Towing Methods to Testing in Waves

The use of the two methods of testing offer different problems when quantifying the added resistance of a vessel form. The added resistance of a ship hull is usually required to determine the extra power required to maintain service speed in a given sea state. This would seem to indicate that the constant velocity method would be the most appropriate technique [2]. Another attractive feature of this method is that the superposition techniques used for irregular wave predictions require data to be in the constant velocity form [3].

However, the gravity style towing system offers the model some freedom to surge in response to the incident waves. This clearly must lead to a different interaction with the wave system. In particular the phase relation between motions and waves may be altered. Gerritsma and Beukelman [4] demonstrated that this relation is the source of the added resistance itself. For this reason such free-to-surge methods are often considered to be the most accurate. However, existing gravity systems cannot be classified as true constant force systems due to the parasitic inertia of the tow weight, pulleys and wires. Thus gravity systems represent some compromise between constant velocity and true constant force methods. An additional concern with free to surge systems is that the velocity fluctuations, and the decrease in average speed, are Reynolds number dependent. However, unsteady motion of the water around the model due to the waves themselves are likely to cause larger variations.

The broad conclusion seems to be that the differences between the two methods are "negligible" [3,4] or within experimental error. Constant velocity methods are therefore used almost universally for convenience reasons. The comparison of the two methods is a main theme in this thesis and so will be dealt with in detail later.

1.2.6 Notes on Scaling of Results.

As mentioned earlier Froude discovered that prediction of full scale figures from model tests is not a straightforward task. This is due to the different scaling laws governing the different components of the overall resistance.

Viscous resistance scales according to the flow Reynolds number. That is to say it is dependent on the flow regime and fluid viscosity. Wave resistance and added resistance effects scale with the Froude number, that is effects are linear with scale. This means that careful separation of the different components is required before extrapolation to full scale figures.

This is usually achieved by separation of the model resistance into skin friction and what is known as the residuary resistance. This is all the non-Reynolds number dependent terms. This is done by applying a Reynolds number dependent friction coefficient over the calculated surface area of the model. These coefficients are empirical and come from a standard set of tests done originally by Froude with flat planks at many different Reynolds numbers [5].

The problem of scaling results is compounded by the large difference between the Reynolds numbers at model and full scale. This can lead to the incorrect flow regime over the model. To reduce this problem two methods are used. The first is simply to test at a very large scale. This is how the very large tanks around the world came about. The largest are over 1km in length and models over ten metres long are often used.

The other method is to deliberately force the flow regime to be representative on the model tests. This is done by tripping turbulence on the model. This is something of a black art with each test tank preferring different methods.

Recently there has been a convergence toward using small cylindrical studs spaced evenly around a section near the bow. These seem to be the most reliable method of ensuring tripping of flow over as much of the model as possible.

Some researchers [1] have warned that over stimulation can lead to other problems such as the position of flow separation over the stern sections varying with scale and method of turbulation.

Fortunately the added resistance due to incident waves scales exactly with the Froude number. This means that accurate predictions of added resistance can be made using small models as there is little dependence on flow regime.

1.2.7 Notes on Model Scales Used

The subject of model size, and thus scale, has recently been an area of much controversy. Confidence in the use of small models for resistance work was shattered by the much quoted "Mariner tank test disaster" [3]. The failure of this design was blamed on misleading results from small scale models tests. This was despite the fact that the same tank and scale were used to develop the outstanding yacht "Courageous".

This has resulted in the reliance on large scale tests, usually at 1/3 scale in the monster tanks. These tests are obviously prohibitively expensive and have restricted the use of extensive model testing to only the richest of design efforts. Large scale tests have a further drawback in that the available run rate is very low due to the long tank settling time required between tests. This can be over 30 minutes in the largest test tanks. This has the undesirable effect of either vastly increasing the turn-around time for testing out new ideas, or severely restricting the number of data points used and thus the reliability of the results.

It was not until 1983, when the Americas Cup was lost to a thoroughly tank tested model, that opinion began to sway back in the test tanks favour [6].

This lack of confidence in the use of small models, and test tanks in general, now seems to be due to a face saving exercise on the part of the syndicate involved, not due to misleading data. The true results from the Mariner test series were presented in 1986 in response to still growing criticism of the use of test tanks [7]. These clearly showed that the model tests had correctly indicated that the blanked off underwater body of "Mariner" was indeed a bad concept.

Recently there has been a steady move back to the use of intermediate scale models, typically 1/9th, with additional validification if required at 1/3rd scale.

As a result of the "Mariner" controversy several studies were carried out concerning the choice of model scale and the resulting confidence in the results that can be expected. Recent attention has also focused on the reliability of test data in general and a good examples are given in References [8,9,10,11,12].

It should be reiterated at this point that added resistance is almost exclusively Froude number dependent and so small models can be used with confidence as long as suitable precautions are taken.

1.2.8 Review of Computational Methods

As explained earlier, computational methods for the prediction of the main areas of resistance have advanced rapidly over the past 30 years. Although confidence in these methods, discussed below, is increasing there are still many problems to be solved and for the time being at least calibration of the programs with empirical coefficients from model testing and full scale data is usually still required.

Calm Water Resistance Methods

Detailed discussion of methods for the prediction of calm water resistance is not relevant to this thesis. A good treatise of the field can be found in [13].

Added Resistance Methods

One of the first calculations was performed by Havelock [14]. He obtained a simple and easy to use formula for the mean added resistance of a heaving and pitching ship, at any given frequency of encounter in head seas. The method is based on the summation of longitudinal pressure components on the hull assuming that the incident wave was undisturbed. Various researchers have modified or reduced Havelock's formulation (eg Joosen [15]), but their methods still rely on the same basic assumptions. These techniques formed a sound basis for the subsequent development of more rigorous methods and are often used for comparison purposes.

Modern methods of prediction follow one of three methods:

- Hull pressure (Maruo[16], Salveson[17])

- Momentum and Energy Methods (Lin and Reed[18], Maruo[19])

- Radiated Energy (Gerritsma and Beukelman[20], Loukakis and Sclavounos[21], Journee[22])

The first is a classical hydrodynamic solution using velocity potential techniques.

Momentum or Energy methods establish a control volume around the vessel and carry out either a momentum or energy balance.

The final method equates the work done against added resistance to the energy contained in the radiated damping waves.

All of the methods seem to have their relative strengths and weaknesses across the range of vessel speed and frequency of encounter. The Radiated Energy methods generally seem to produce the most acceptable results at present. The overriding problem seems to be the accuracy of the theory used to predict heave and pitch motions and their relative phase with the incident waves. Often results from model experiments are used to provide accurate data for use in the subsequent computations.

One aspect of added resistance theory which appears to offer large time savings over model testing is the linear superposition technique for estimation of added resistance in irregular sea spectra. Results from such computations appear to agree very well with experimental and full scale data for small wave amplitudes.

In conclusion it appears that at the present time no one theory for the prediction of added resistance in a seaway seems to offer accurate results over the entire range of vessel speed and frequency of encounter. Model tests are often required to provide accurate hydrodynamic coefficients for use in subsequent processing.

1.2.9 Full Scale Data

Various attempts at full scale testing have been carried out. These were mainly for calibration of model and computational testing [7]. Motion data and wave measurements from real vessels in service have been used to assess the accuracy of computed or experimentally derived motion coefficients.

Average speed and engine power measurements have been used to estimate mean added resistance [23].

Modern racing yachts usually carry much instrumentation to allow tuning of the boat to the conditions. These transducers in addition to others are often used to log full scale data during actual sailing [24]. The subsequent processing of this data allows appraisal of the various experimental and computational prediction techniques being used. This is an emerging area of research and is outwith the scope of this thesis.

1.2.10 Integration of Methods into a Modern Design Approach

The modern naval architect thus has many tools at his/her disposal to evaluate the likely performance of a candidate design. Used with care these can help avoid costly mistakes.

It is important at this stage to show how model tests still have an important role in modern integrated design approach, and thus why continued development of experimental techniques is justified.

Probably the best examples of integrated design programmes to date are the recent Americas Cup competitions.

In 1983, the Australian designer Ben Lexan and a team of Dutch scientists demonstrated beyond doubt that a strong technical effort could substantially improve the design of 12-metre yachts, a class which had been widely believed to have reached optimal design [6]. Their efforts in developing the radical winglet keel through extensive tank testing produced the unbeatable 12-metre Australia II, which dominated the 1983 challenge series and won the cup off the Americans for the first time in its 132 year history.

The response to this has been that all syndicates involved in such events since have spent an increasingly large percentage of their budget on exhaustive testing of ideas with model tests, computational methods and full scale appraisal all playing important roles. Good examples of such programmes are to be found in [25]. All test programmes to date have included a large amount of tank testing for assessing, and providing coefficients for, the various computational models.

The most extensively documented to date was the Sail America Foundations successful challenge that regained the Cup for America in 1987 [26,27,28,29,30]. It was interesting to note though that little or no attention was paid to added resistance in waves, despite the rough conditions in Perth, Australia, where added resistance can account for as much as 50% of total resistance [31].

The techniques developed by these teams are now finding wider application within the sailing world. The usual method adopted in a modern design programme is to combine the numerous different components under study into an overall Velocity Prediction Programme (VPP). This is used to assess the likely performance of candidate designs in real racing conditions. Such is the confidence in a carefully set up VPP that the recent International Measurement System (IMS) rating rule uses such a method for handicapping boats racing within this class. Individual handicaps are computed from a standard set of measurements for each boat, in each race, taking into account the race course and the prevailing conditions. The IMS VPP is being regularly updated with new data from all areas of research[32].

However, tank testing still remains an expensive luxury. Conventional systems can only generate a low run rate and therefore exhaustive testing takes a lot of time and money. The effect of this is to limit testing to a minimum. There are few documented records of detailed design evaluation of performance in waves in spite of many independent studies showing that for typical hull forms added resistance is a large proportion of total resistance [24,31,33,34,35,36]. In addition there are still reservations about the validity of the current methods being used for the performance prediction of yacht forms as discussed below.

1.2.11 Yachts Tank Testing as a Special Case

The problem with using conventional methods for testing yacht hull forms is the very small differences that constitute a significant margin. Resistance estimation is used to determine approximate powering requirements for ship forms. This means that error bounds can be large and relatively "hand waving" assumptions can be justified.

However, the difference between a race winning design and the rest of the field can be as little as 1 percent. This means that every assumption made during the design assessment phase must be looked at very closely indeed. Testing methods must be highly rigorous, use a large number of runs, and most importantly be as close an approximation to reality as possible.

Clearly, conventional systems provide a very accurate repeatable mechanism for testing the still water characteristics of hull forms. However, extension of design evaluation to include testing in waves leads to serious doubts over conventional testing methods as discussed below.

A major issue is whether the constant velocity approach to testing is a valid method in this case. This approximation, as explained earlier, is convenient in terms of experimental method and form of results, but may give misleading results when looking for very small performance differences between candidate designs. A rigorous comparison of the two methods is still required.

There have been several independent studies of methods of assessing the performance of yacht designs [24,31,33,34,35,36]. Many of the conclusions from these are contradictory and are based on very small data sets. The following section looks carefully at a series of articles which discuss the issues relating to testing of yacht forms in waves and the subsequent interpretation of the results.

1.2.12 Review of Work in Field

As explained earlier, there are several differences between the propulsion of ships and yachts. These will be reiterated here as most of them are considered by the papers discussed in this review. Major factors that must be considered when carrying out experimental studies of yachts for performance prediction include:

- Yachts rarely sail straight into a wave system, most towing tanks are only able to produce the extremes of pure head and following wave conditions.

- The forces driving yachts often include a significant force acting sideways as well as the driving force. This leads to a phenomenon known as leeway or sideslip, which in turn leads to induced drag from the hull and underwater lifting surfaces.

- The drive force act at the centre of effort, or lift, of the yachts sailplan leading to heeling or tilting and a bow down pitching moment.

- The motions of the yacht can be heavily affected by the aerodynamic damping provided by the sailplan. This in turn can affect the driving force.

- Due to the relatively small size of a typical racing yacht, the waves encountered often have a larger relative amplitude than for larger vessels. This means that added resistance is usually a significant proportion of the total resistance.

Many of these factors are often ignored in conventional testing because the towing system and test tank were originally designed for work on ship hulls. Numerical work also usually ignores the complications discussed above.

There are notable exceptions to this, examples of which are discussed in detail in this section.

Spens, de Saix and Brown, 1967 [33], were the first authors to publish experimental results for a sailing yacht in bow quartering seas. The towing system used was based on a system originally designed for measuring the motions of a free running model. This apparatus was modified to incorporate independent drive and side force components using weights. In this way the system becomes a good approximation of the real situation. The main problems not addressed by this system were towing the model from deck level not the centre of effort of the rig, and no simulated aerodynamic damping was applied. The raised centre of effort position was simulated using a torsion spring at the tow point. Full details of the rig can be found in Appendix 1 of the same paper. Experiments were carried out using a 1/13 scale 12-Metre at two radii of gyration. The results from these tests led to the following conclusions. Added resistance appeared to be proportional to the square of wave amplitude as expected from linear wave theory. It was noted that pitch inertia had a significant effect on added resistance. Finally it was noted that wave action seemed to have little effect on leeway angle, although doubts over this conclusion were expressed due to concerns about the accuracy of the system.

In the same study, Spens, de Saix and Brown also compared the results obtained in oblique seas with similar tests carried out in head waves. Several problems with this approximation were discussed including difficulties keeping the encounter frequency and wavelength consistent with the oblique case. The results obtained were seen to be "generally similar" but the paper concludes that head wave tests can only be used to give "a useful guide" to performance in oblique waves.

Pedrick, 1974 [34], showed how the results from added resistance experiments could be incorporated into a velocity prediction program. Speed loss, leeway increase and irregular wave spectra were incorporated in the model. It is interesting to note the large effect of a wave system on the velocity made good (VMG) to windward. Even for the small wave amplitudes used the loss in VMG corresponded to over 2 minutes 45 seconds on a single America's Cup windward leg. It is even more interesting to note that many more modern VPP's ignore added resistance completely.

Klaka and Penrose, 1987 [24], carried out tests in head seas for a 12-Metre hull form both upright and with 15 degrees of heel, without drift, using constant velocity towing techniques. The surprising result of this work was that the added resistance for the heeled condition was approximately 50% of the values for the upright case. It was concluded that upright tests only were inadequate for assessing added resistance in waves.

Gerritsma and Keuning, 1987 [35], carried out similar experiments with two markedly different hull forms, a light displacement hull with a high beam to draft ratio, and a heavy displacement design with a deep hull form. Experiments were carried out with and without heel and drift angle. The results for the heavy hull form seem to agree well with the results presented by Klaka and Penrose [24]. However the added resistance of the light displacement design seemed to be hardly influenced by the angle of heel. The paper also concludes that the effect of drift angle on added resistance is negligible. However, differences of over +-15% of peak values are evident from the results presented.

A broad conclusion derived from the articles described above can be formed as follows. The accurate assessment of added resistance in waves of a yacht form is a very complicated process. Many different factors seem to have a strong influence on the results. Much care should be exercised when using results from conventional testing for performance prediction, in some cases tests carried out may be at best misleading, at worst useless. All papers discussed above present only a few data points per graph, the overall level of repeatability demonstrated seems to be low, leading to uncertainty over conclusions. Far more rigorous investigation of the factors involved is required before any firm conclusions can be made. Thus most experimental facilities are inadequate for the accurate assessment of yacht added resistance.

The only sensible way to improve the situation seemed to be attempt to model the full-scale system as accurately as possible.

During the course of this project one serious attempt at this has been made at the Marin Seakeeping Basin in the Netherlands. The new techniques are reported by Kapsenberg, 1991 [31]. A system for applying a realistic constant tow force at the centre of effort of the rig was developed. A constant force winch was used in conjunction with careful towing carriage control and an active steering system to generate a realistic tow force vector. The model was completely free to move in all 6 degrees of freedom under the action of the wave system. This represents a massive leap in the technology available to yacht designers for assessing the performance of yacht hulls. However, there are still some shortfalls.

The data presented in the paper shows that still only a few runs are used to generate the graphs. No indication is given in the paper concerning the degree of automation of the testing rig, but the essentially free-running style of the system must make acceleration, deceleration and control of the model between tests difficult. Also, the tow force is a constant value representing the mean aerodynamic forces. The single constant-force winch configuration does not allow a full simulation of all transient aerodynamic loads acting on a yacht. Thus it can be concluded that although this system makes a vital step towards accurate added resistance prediction, much still remains to be done in the field.

1.3 Introduction to the Wide Tank

The basin used for the work carried out for this thesis was the Edinburgh University Wave-Power Wide Tank. This facility was constructed in 1977 for wave loading studies of the Edinburgh Duck wave energy converter. A schematic plan view of the tank is shown in Figure 1.5.

The external dimensions of the tank are; length 27.5m, width 11.0m and depth 1.2m. This is small compared with contemporary testing tanks. However, the tank confers a number of advantages for yacht testing work.

The wave generation system is one of the most advanced in the world. The waves are generated by a bank of eighty identical wave paddles, driven by individual DC servo motors. Each is independently controllable allowing a virtually infinite array of wave conditions to be generated. The paddles are also efficient wave absorbers. This in conjunction with the effective passive beaches along the tank walls gives the tank an extremely short settling time. The tank typically takes less than 1 minute to settle between runs. This feature is essential to allow a high test rate and to prevent low frequency standing waves affecting results.

The wave paddle control signals are generated by the wave synthesiser unit. This computes the individual drive signals for each wavemaker from a set of parameters down-loaded from the tank's computer system. Up to 1024 fronts can be generated simultaneously, each with a preset frequency, amplitude, angle and starting phase. This allows accurate simulation of the most complex wave systems.

Despite this capability the vast majority of the tests presented in this thesis were carried out in monochromatic seas. This was because most of the tests were designed to evaluate the new towing methods developed, rather than try and provide realistic performance prediction for individual designs.

1.4 Summary

The review of the field revealed a need to improve techniques available for the experimental testing of yacht hull forms in waves. It is envisaged that experimental methods will be used for the majority of work in this field for the foreseeable future because at present computational methods are not adequate for accurate performance prediction of a sailing yacht in waves.

1.5 Project Aims

With the results of this review the following project aims were formulated:

- It seemed important to improve the quality and repeatability of tank test data.

- Fully automated testing generating high run rates was seen as very important.

- A full investigation of accurate Constant-Force towing techniques was deemed to be necessary.

- Careful towing system design was required to attempt to model the full-scale system as closely as possible.

- Attempts to reduce the high cost of tank-testing were required to make the techniques more widely available to yacht designers.

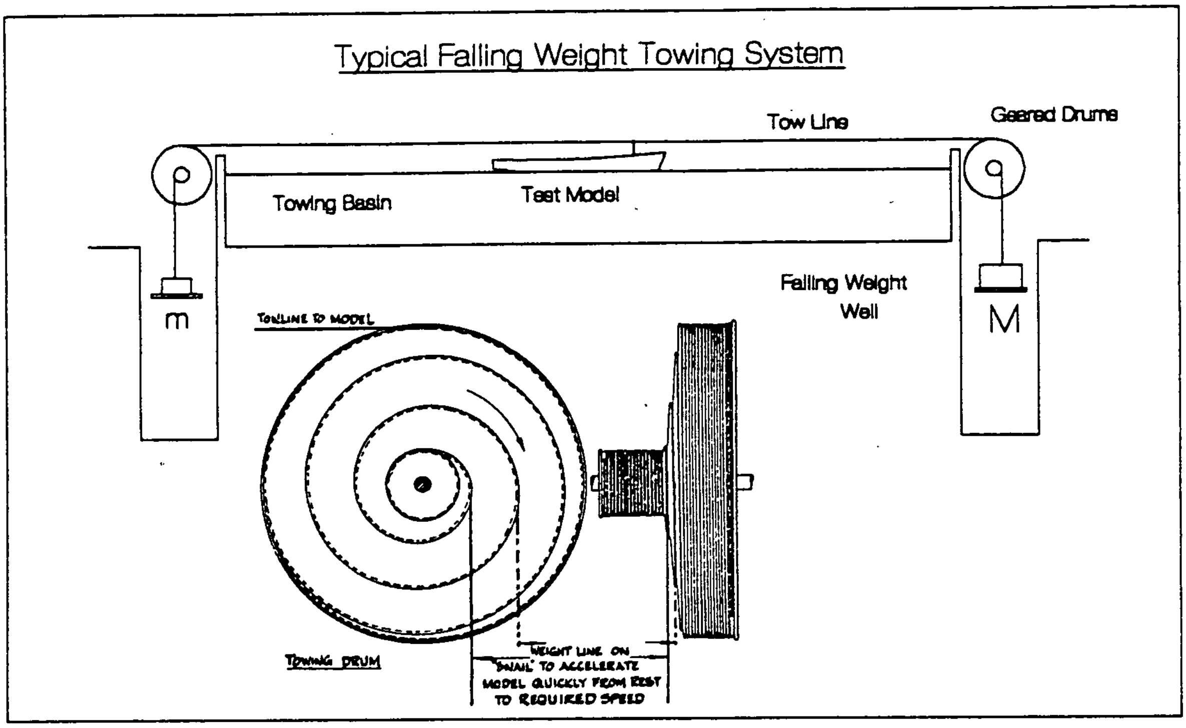


Figure 1.1 Typical Falling Weight Towing System

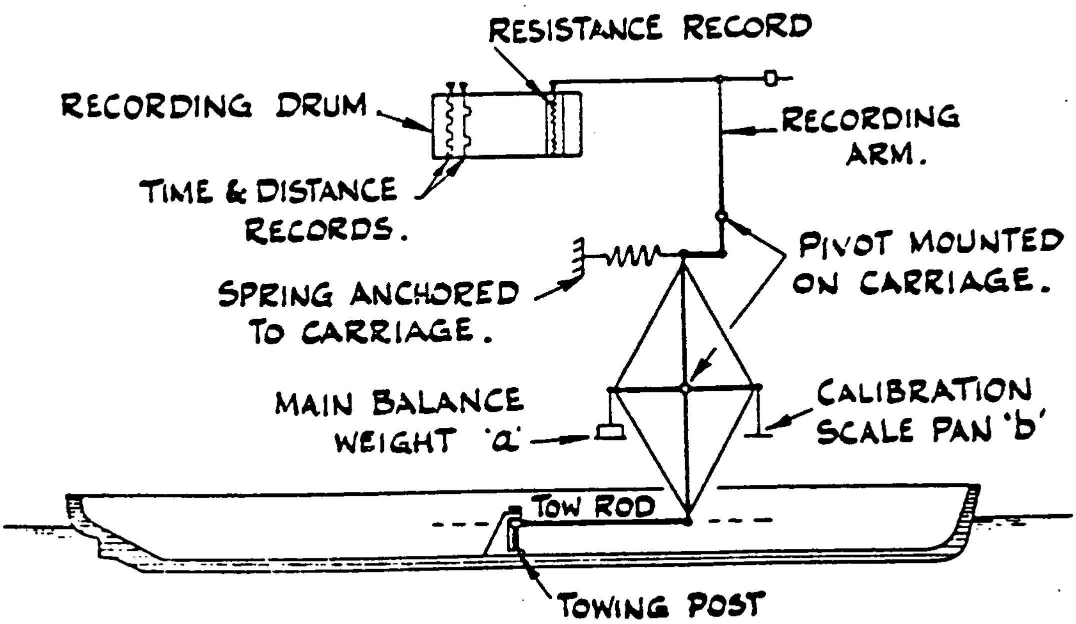


Figure 1.2 Servo-Carriage Gravity System

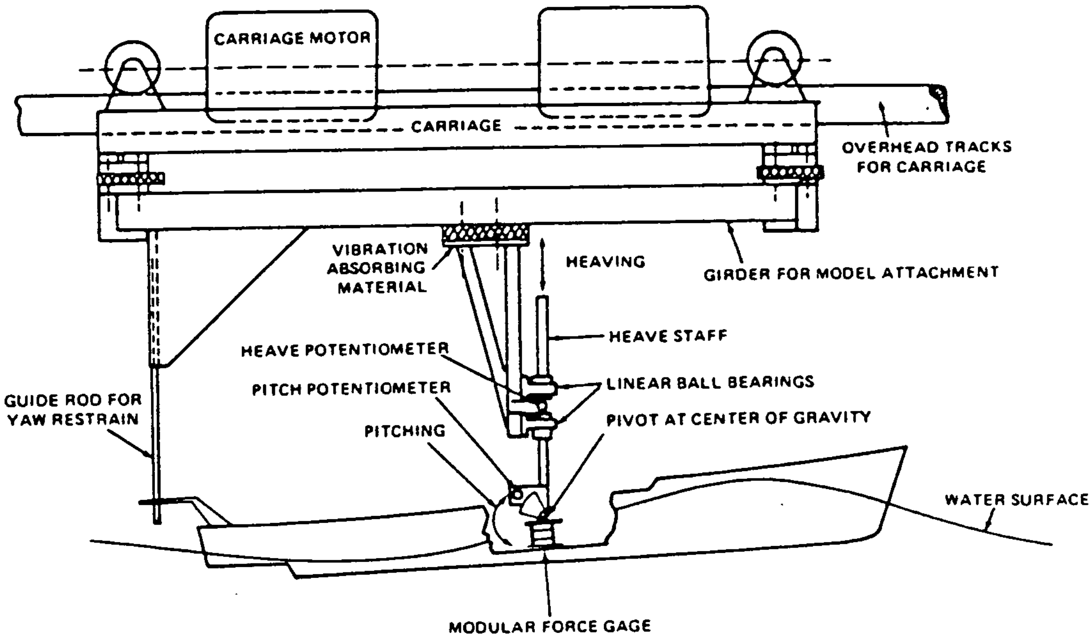


Figure 1.3 Typical Ship Dynamometer

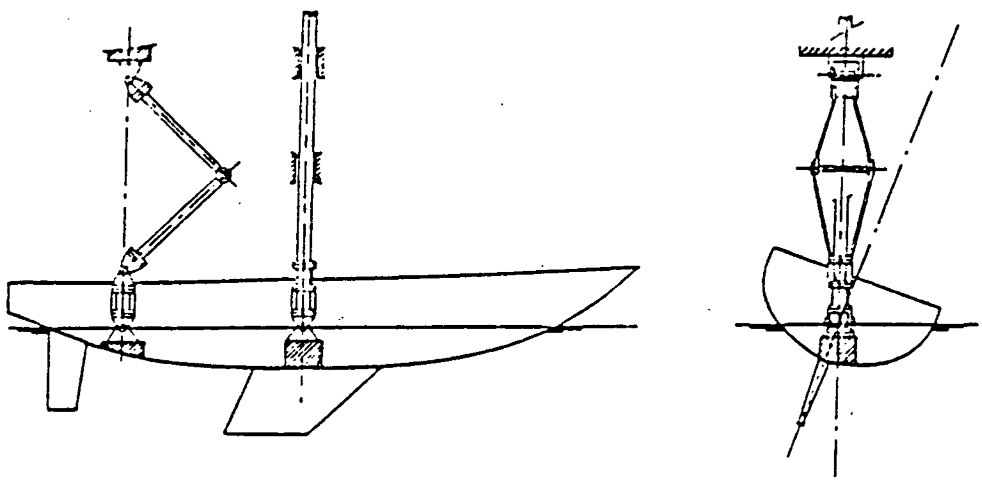


Figure 1.4 Typical Yacht Dynamometer

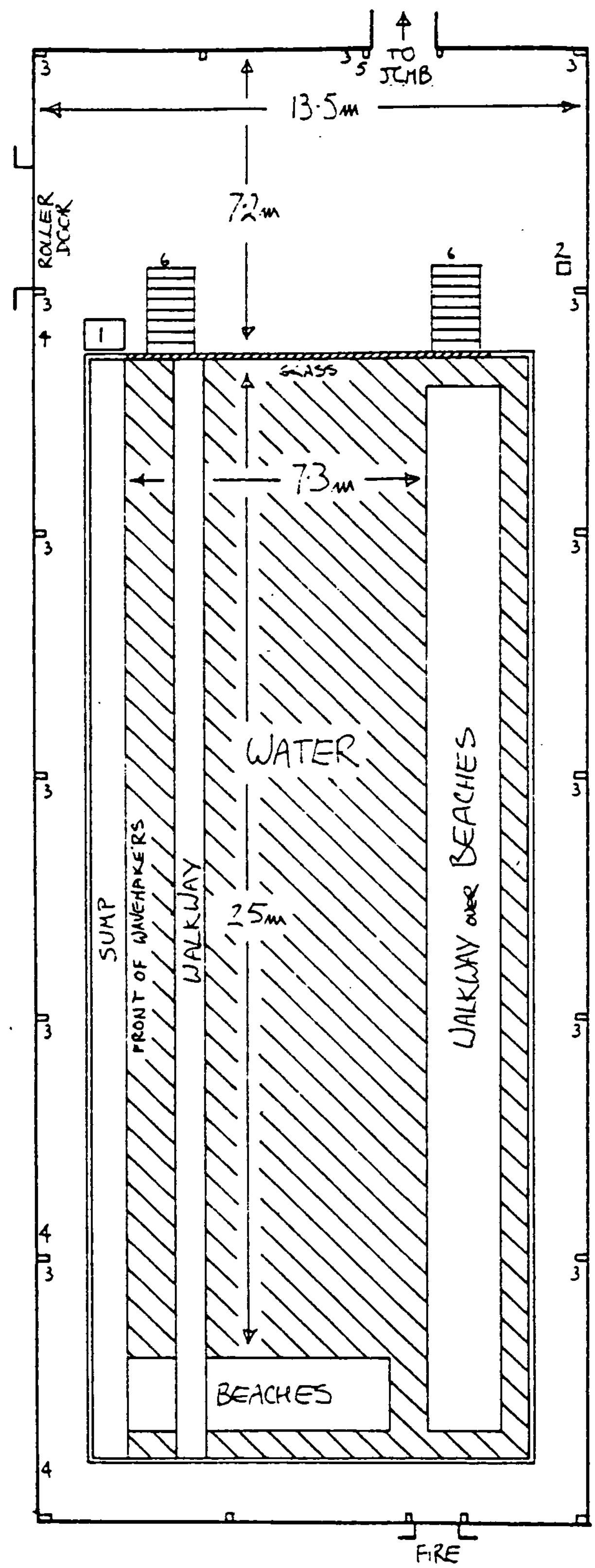


Figure 1.5 Schematic Plan View of Edinburgh Wide-Tank

Chapter 2. Final Year Project Work

2.0 Chapter Summary

This Chapter describes briefly the work carried out for the author's Final Year Undergraduate Project for the degree of Bsc(Hons) in Mechanical Enineering. As such this section is not submitted for examination but is included as it contains essential background material required later in the thesis. The Chapter describes the first towing system to be made and includes the recommendations for future development included in the Project Final Report. For fuller details please refer to this document directly [37].

2.1 System Description

Due to the short nature of the final year project the towing system built and tested was, out of necessity, a crude simple rig. As at all stages of the project several different configurations were considered.

As mentioned in the general introduction, it is desirable to try and simulate the forces on the yacht as accurately as possible. The major element of this which had to be addressed was provision for angled force towing, while allowing the model complete freedom of movement.

This is a difficult problem. Several proposals including ducted air fans and complex towing gantries were examined but discarded in favour of a crude 1-dimensional (1-D), straight ahead only system, based around a low inertia motor and winch drum.

The inherent simplicity belies the potential of such a system. The winch can be used to drive a pre-tensioned continuous wire loop, allowing fully automated testing, difficult with any free-running set up. If system inertia is kept low and wire stretch kept to a minimum, a motor mounted tachogenerator allows easy measurement of model velocity.

With the long wires closed loop force control would be difficult if not impossible. Thus some form of open loop drive with acceptable accuracy was needed.

Printed Armature DC servomotors have exceptionally low inertia and linear, velocity independent, torque/current characteristics. Such a motor driven by a current source gave a good first approximation to constant force towing.

The basic system was the subject of a great deal of future development and improvement, so it will be described in some detail.

2.2 System Components

An appropriately sized Printed Armature motor was found. The motor already incorporated a suitable tachogenerator.

A low inertia screw thread winch drum was made. The drum is mounted onto the motor using a split collet to minimise eccentricity. A ballraced low inertia pulley completed the drive hardware. The drum and pulley were mounted at opposite corners of the tank to maximise the available run length.

Multi-stranded stainless steel wire was chosen to make the pre-tensioned tow loop. The thickness was chosen to avoid fatigue failures due to drum diameter. The wire is wrapped around the drum with six complete turns preventing creep of the wire position.

The pre-tension of the wire loop was standardised by measuring the transverse deflection of the wires with an applied load.

The model was linked to the loop with a tow line and a brake line. The brake line was required to allow rapid braking of the model at the end of each run. This restricted towing to one direction only. With system automation this was not seen as a serious problem.

The wire loop was unsupported over the full length of the tank. This led to large transverse oscillations of the wires. Linking the two halves of the loop at the model attachment point forced the wire motions to be in phase with each other, reducing transient velocity errors due to differential torques on the winch drum from out of phase modes.

The motor was driven by a power-amplifier commandeered from the Wavepower Project.

The system was interfaced to a BBC microcomputer to allow automated testing. Lever operated micro-switches were mounted at either end of the drum, positioned so that the tow wire tripped each alternately as it wound back and forth across the winch. These allowed the computer to automate the test sequence.

Several failsafe features were incorporated to prevent damage to the model, towing system or wavetank in the event of the computer failing.

A schematic of the complete system is shown in Figure 2.1.

2.3 Control Software

The BBC micro had full control of the drive motor via a digital output channel. It also collected and stored the test data. Subsequently it was used to analyse and display the results. In its initial guise the system was not interfaced to the wave generation control hardware. The waves had to be changed manually during testing.

As mentioned earlier wire bounce was a potential source of errors. Wire snatch on start-up was the major cause of this phenomenon. With the motor under computer control it became simple to write a routine to gently take up tension on the tow line before the test run began.

The test program is shown using a flow chart in Figure 2.2.

The computer made the whole system, despite its simplicity, a credible unit. Without it, extensive testing giving accurate repeatable results would not have been possible.

2.4 Calibration Technique

As with all experimental apparatus, the accuracy of the results was heavily dependant on the calibration procedure.

The best scheme when using a computer to sample data is not to calibrate individual elements of the system, but to calibrate directly between the physical quantity and the sampled computer values. Where possible this procedure was used.

Calibration of model velocity was relatively simple. The tow loop was removed and the drum was driven at a range of speeds. For each speed angular velocity was measured using an optical tachometer and plotted directly against the average computer sampled value.

Calibration of the tow force was more difficult and the results more open to doubt.

The tow force was deduced directly from the power amplifier current, the motor's torque constant and the drum effective diameter. The manufacturer's torque specification was checked by looking at the e.m.f. characteristics of the motor when driven unloaded. A subsequent correction for the velocity dependent friction current was determined experimentally by driving the complete system with no model attached.

The two were then combined to give a direct relation between the digital number from the BBC and applied tow force.

2.5 Test Model

The model size is dictated by the frequency envelope of the wave generation system and the force range of the towing system.

There was insufficient time remaining on the 4th year project to build my own test model.

The model used for the system evaluation tests was kindly supplied by the British Hovercraft Corporation. She is a tank-test model built in the 1950's for 12-metre design analysis work for the British America's Cup challenge. The design has proved very suitable for towing system development on account of its size, rugged construction, heavy displacement and good directional stability. Model details are given in Appendix A.

The model is still used extensively for developing new features on the towing systems.

2.6 Experimental Work

The experimental work carried out for the final year project was mainly for system evaluation purposes. However, a lot of useful information concerning the test model's performance was obtained.

There was only sufficient tank time available for a limited number of parameter sweeps. These were chosen to evaluate and demonstrate the system's capability. Two principal wave encounter angles were used throughout testing. A 40 degree head sea simulating close-hauled on port-tack to examine pitch phenomena and a true beam sea, parallel to the model's track to investigate the model's heave response.

Several other short experiments were carried out to look more closely at certain aspects of the towing system and model design.

The important results are summarised and discussed below.

2.6.1 Still Water Control Series

The first set of experiments were a set of still water (no waves) control runs. These were to check that the system was giving sensible results across the speed range of the model. The results presented in Figure 2.3 show typical resistance trends for non-planing hull shapes. If inverted the trace shows the same characteristics as the output from conventional constant velocity towing system as expected.

The curve also shows the characteristic 'humping' and 'hollowing' caused by phase variation between bow and stern wave systems. The hollows appear when the bow and stern waves are in phase.

2.6.2 Frequency Sweeps

The next set of experiments were a series of monochromatic and mixed sea frequency sweeps for the two chosen approach angles.

40 degree head sea

The results from the monochromatic tests are plotted in Figure 2.4. The family of velocity curves show a large reduction in velocity at around 1Hz, corresponding to the pitch resonance of the test model. There is a smaller effect at 1.3Hz due to heave resonance. The percentage speed loss reduces sharply with increasing tow force.

The mixed sea tests show a much less marked effect. However, the same pattern is still evident. The result are shown in Figure 2.5 superimposed on the monochromatic curves for comparison.

Beam Sea

Figure 2.6 shows the regular and mixed sea test results. In monochromatic waves a significant speed loss is noted at heave resonance. The mixed sea tests show an erratic trace, but there is still an underlying downward trend as heave resonance is approached.

2.6.3 Amplitude Sweeps

The aim of this set of tests was to examine the relationship between wave amplitude and model velocity for the 40 degree head sea case. The test frequency was chosen to avoid resonance phenomena.

Figure 2.7 shows a plot of model velocity versus wave amplitude for a single tow force. The monochromatic curve follows the shape of a parabola demonstrating the expected inverse square law [3,34].

The mixed sea tests show a more linear trace. This may be due to wave breaking at the higher amplitudes leading to a lower effective r.m.s. amplitude. More work needed to be done to find out the cause.

2.6.4 Circle Wave Runs

The circle wave is an interference waveform analogous to the focusing of light through a lens. The incoming wave fronts form the arc of a circle as they converge to form an interference peak. The experiment was devised to see whether the measurement system could resolve individual wave interactions.

Figure 2.8 shows a typical trace. The numbered points correspond to the following events.

1. and 2. represent the velocity peaks as the model surges towards the centre of the wave system. The main crest is encountered at point 3. The remaining points 4. and 5. show the models retardation by the fronts approaching the interference zone.

2.6.5 Alteration of Resistance Characteristics

These tests were a quick evaluation of a model configuration change to see whether it led to a significant change in the model's performance in waves. A passive pitch-damping foil was mounted under the bow of the model.

Figure 2.9 shows that the device dramatically improves the performance of the model over most of the wave spectrum, without significant increase of the still water resistance. A similar study carried out in Canada [36] at approximately the same time, concluded that although added resistance was significantly reduced, the overall resistance was generally higher due to the parasitic drag of the foil.

The author of this thesis puts this down to the size and configuration used.

2.6.6 Repeatability Trials

The aim was to show that the velocity time-series' produced by the complete system were repeatable. The waves and towing system were crudely synchronised by simple manual key pressing.

Figure 2.10 shows two 'synchronised' runs.

2.7 System Evaluation

The experimental results can be used to analyse the systems' performance.

The towing system generates results that agree closely with established phenomena, for example, the still water and pitch resonance runs.

The system produces consistent and repeatable results. Typically all the average velocities from a set of runs are within 0.5% of each other.

The motor and drum assembly seem able to follow even the most violent disturbances as shown in the circle wave runs. A mathematical model of the effect of system inertia shows that the tow force should be constant to within +-5%.

The problem of wire bounce was initially considered more serious than it actually was. With the two halves linked at the tow point the vibrations damped themselves out rapidly.

The towing system carried out over 2000 trouble free automated runs during three weeks testing, indicating that reliability is excellent.

The various shortfalls and problems will be addressed in the next section.

2.8 Conclusions

- An automated constant force towing system was designed, constructed and tested.

- The system produces repeatable model velocity timeseries.

- A mathematical model predicts that the tow force is constant to within +-5%

- A large range of experimental results are presented that demonstrate the system's capability.

2.9 Recommendations for Further Development

- The towing system should be interfaced to the wave synthesiser to allow complete automation.

- A roll damping system should be developed.

- A brush-less motor should be considered to reduce system friction.

- A self compensating steering mechanism should be implemented.

- Methods of damping the tow loop should be re-examined.

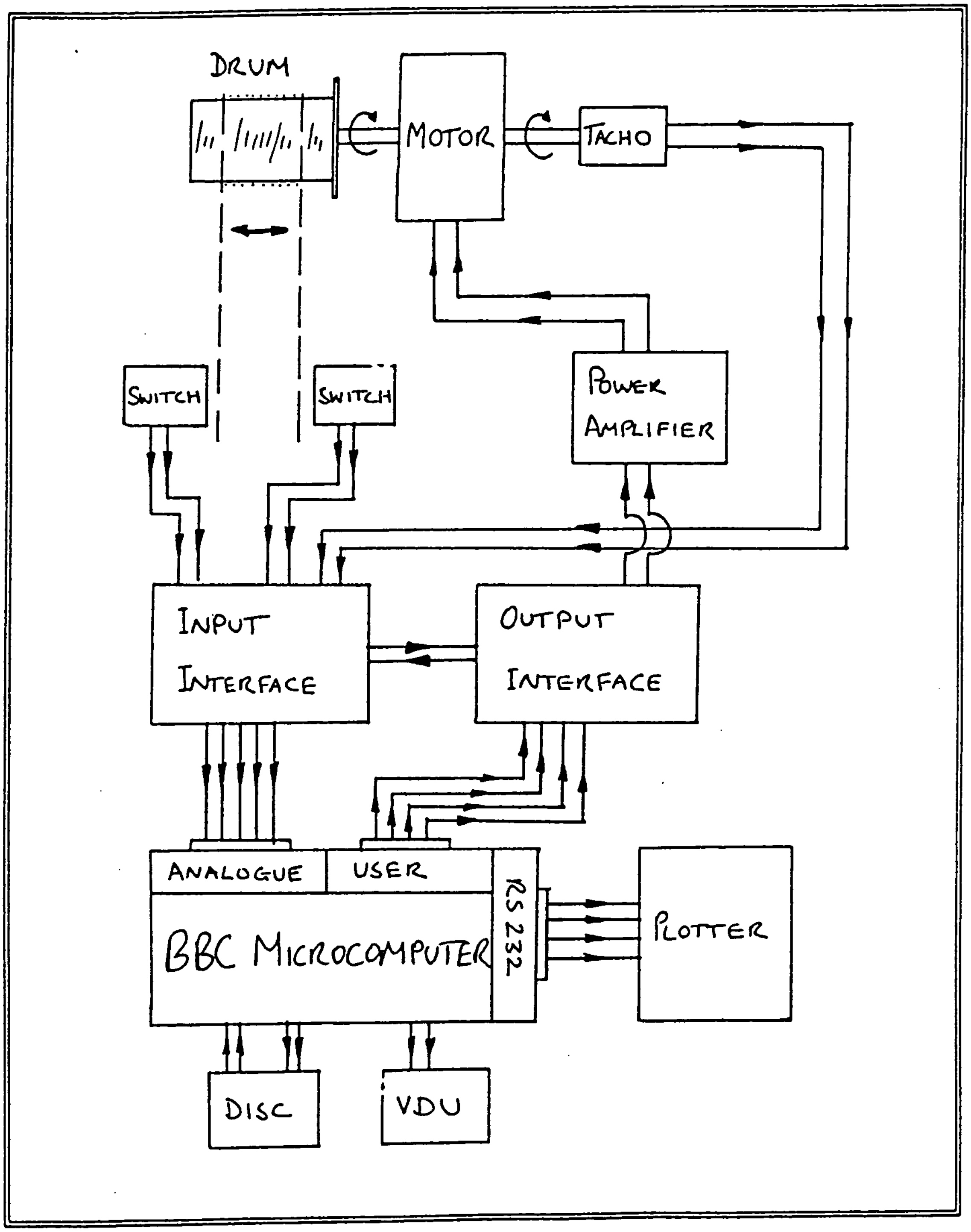


Figure 2.1 Schematic Towing System Layout

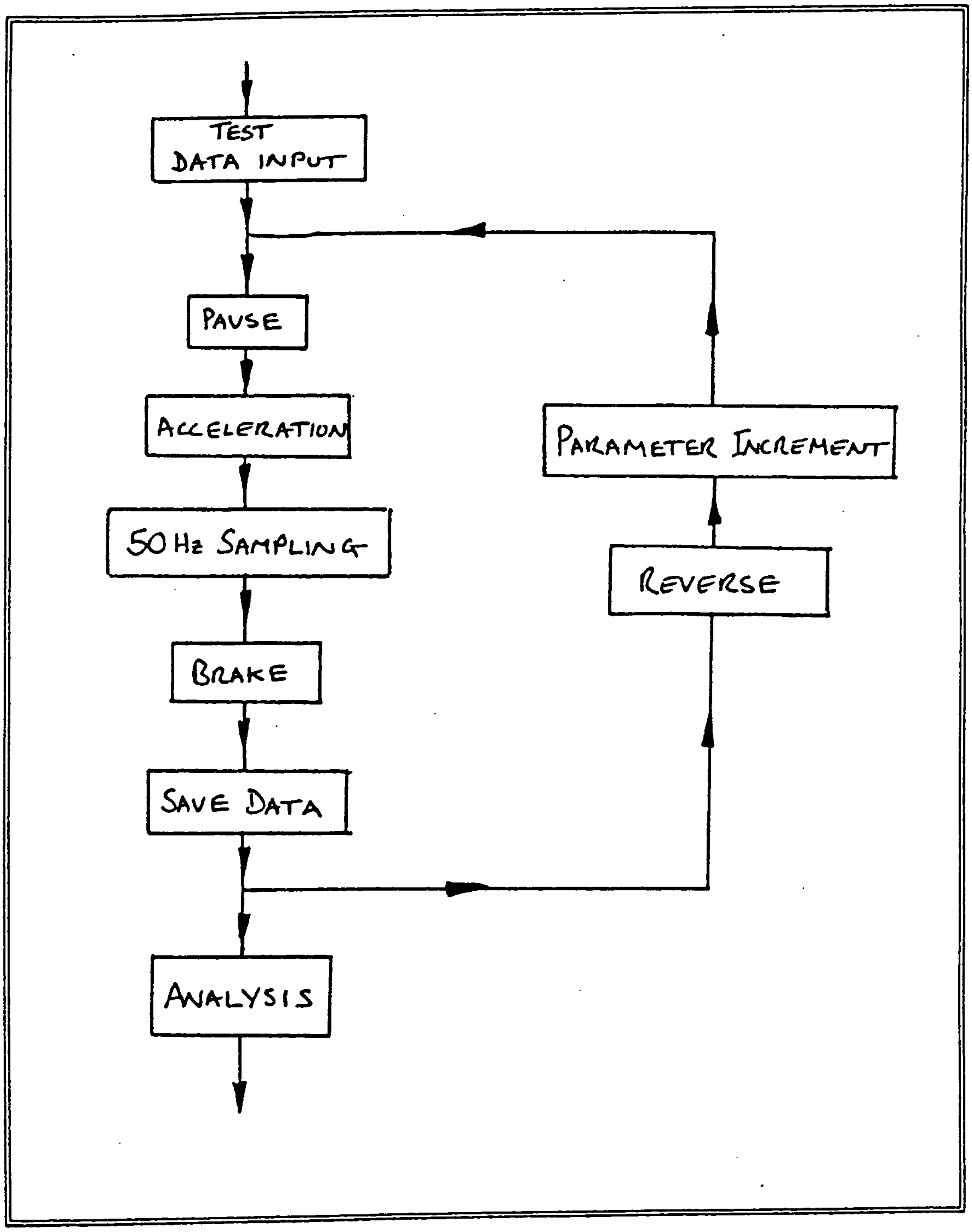


Figure 2.2 Test Cycle Flow Chart

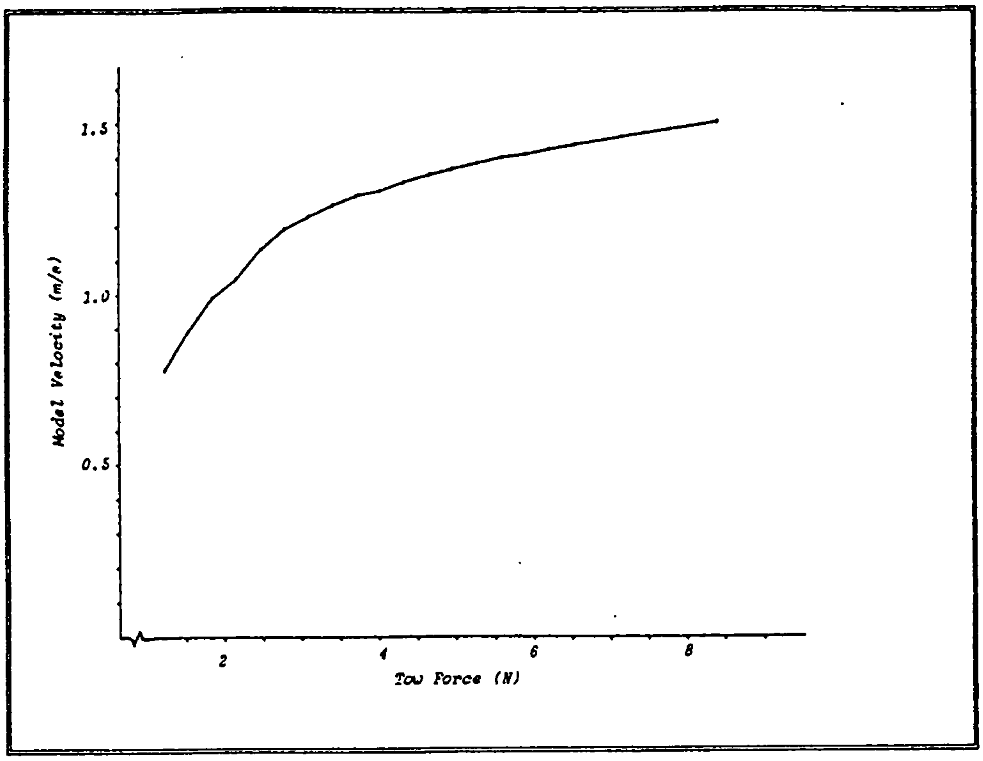


Figure 2.3 Still Water Control Series

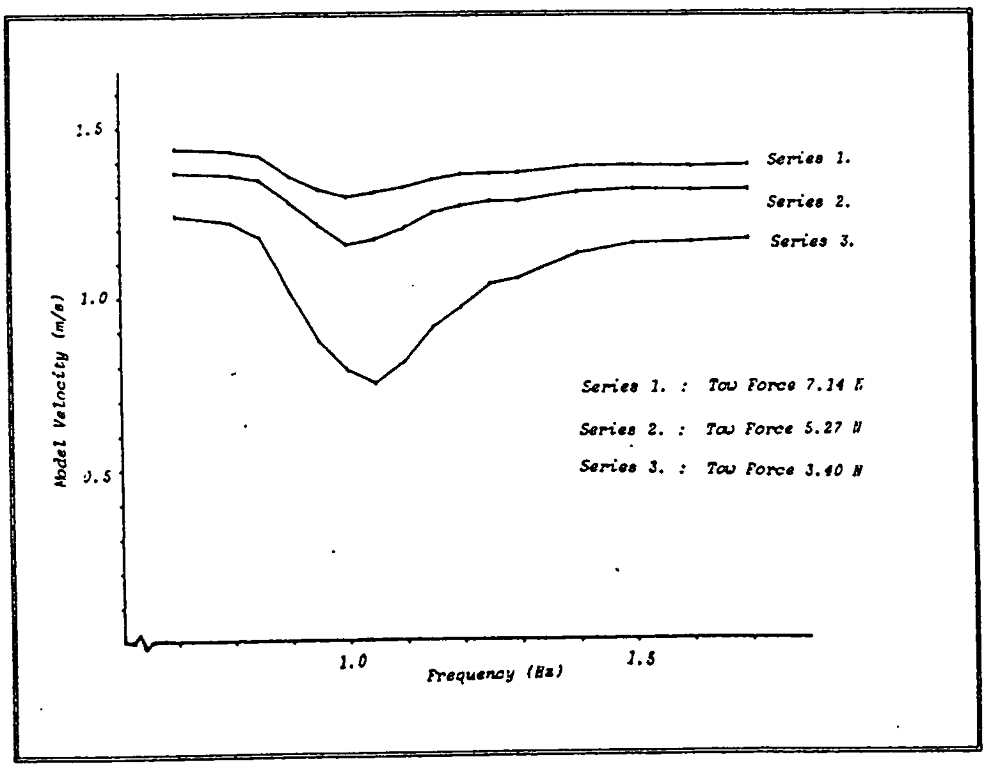


Figure 2.4 Monochromatic Frequency Sweeps

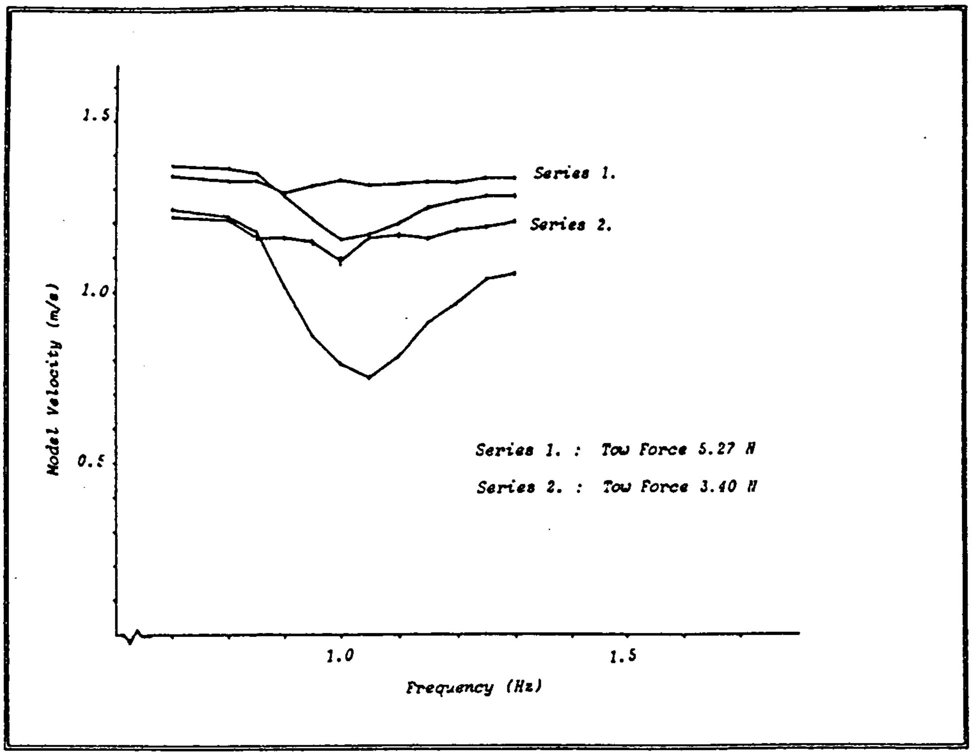


Figure 2.5 Mixed Sea Frequency Sweeps

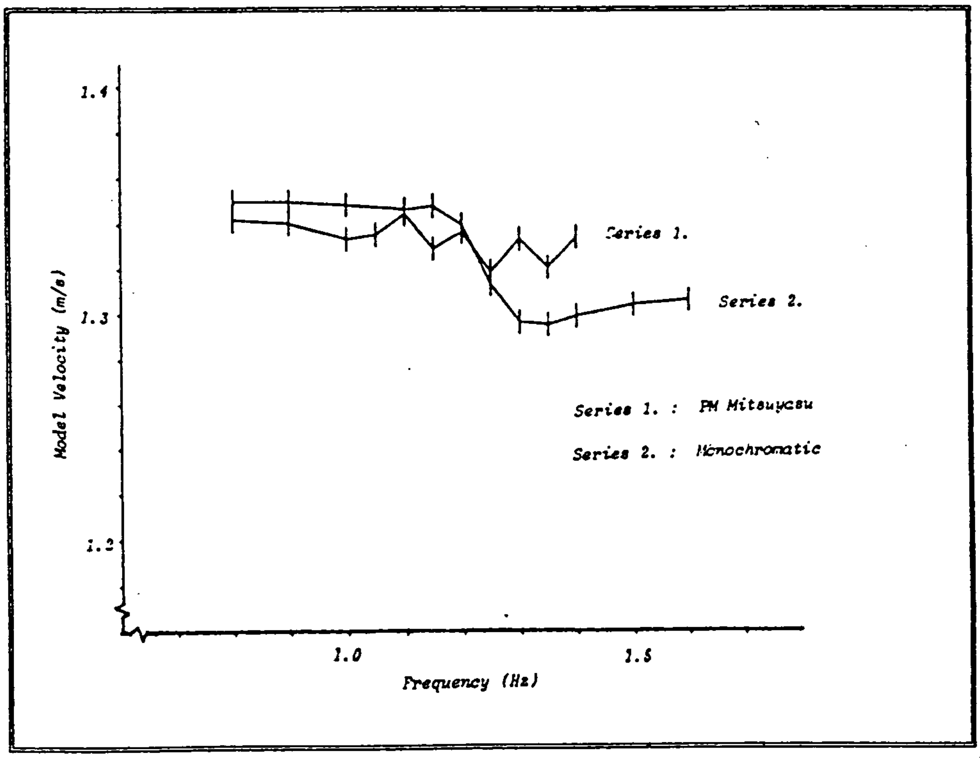


Figure 2.6 Beam Sea Frequency Sweeps

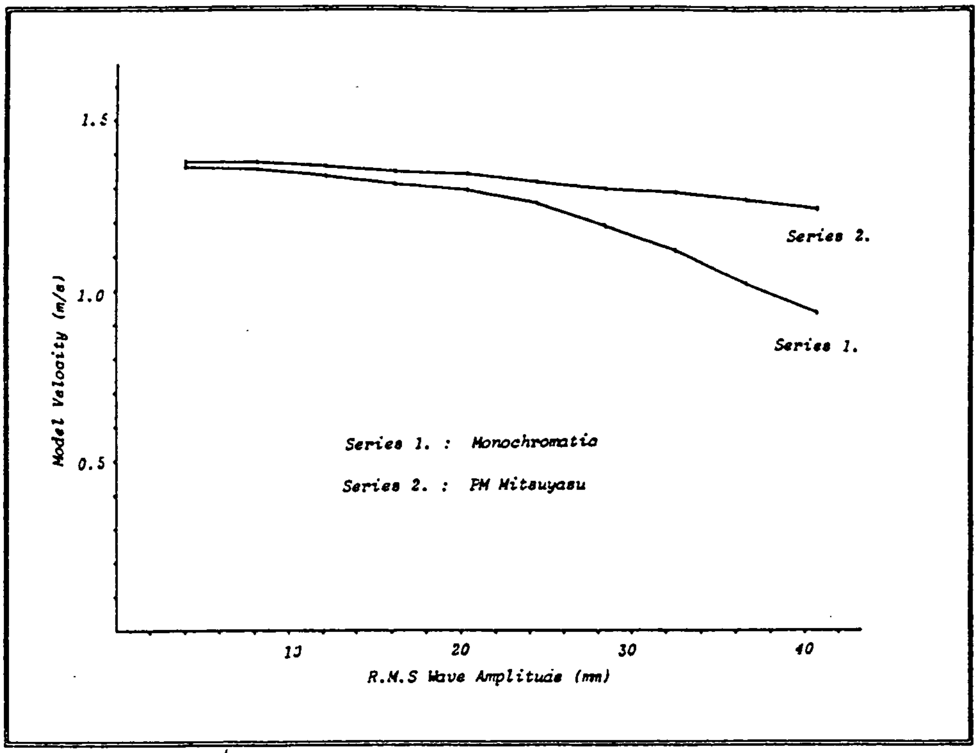


Figure 2.7 Monochromatic and Mixed Sea Amplitude Sweeps

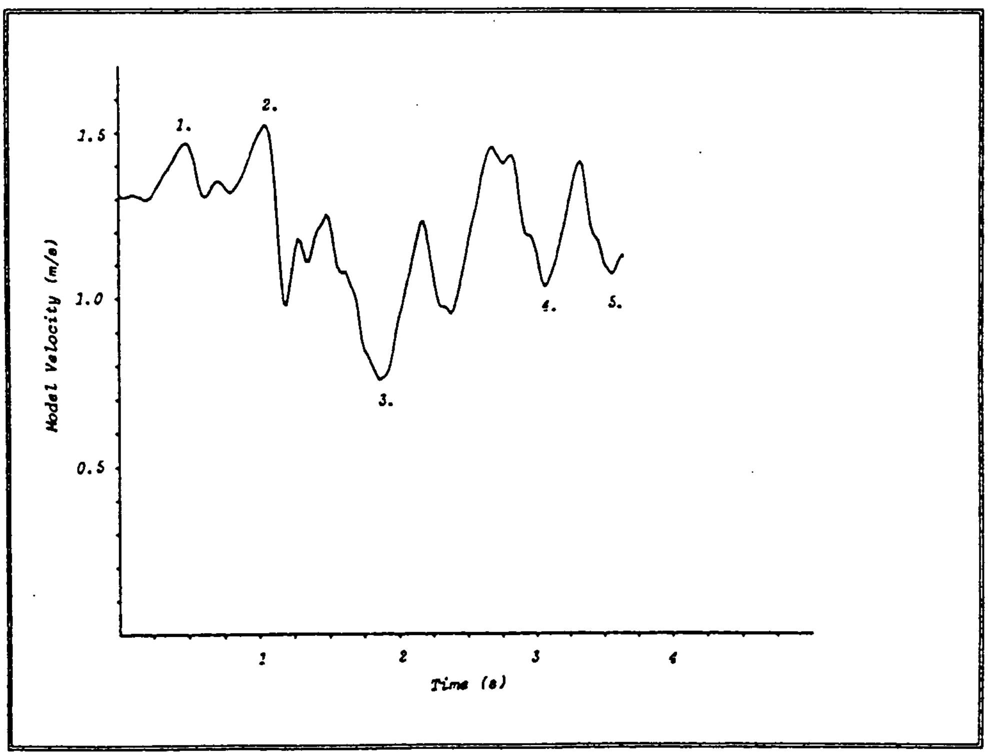


Figure 2.8 "Circle-Wave" Run

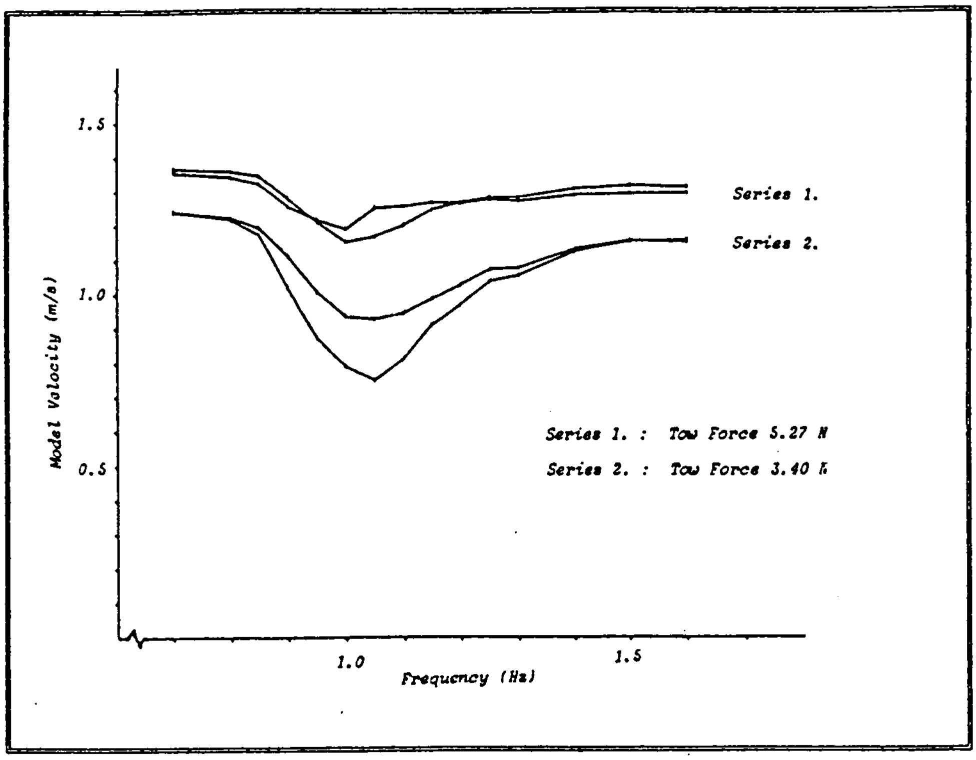


Figure 2.9 Pitch Damper Frequency Sweeps

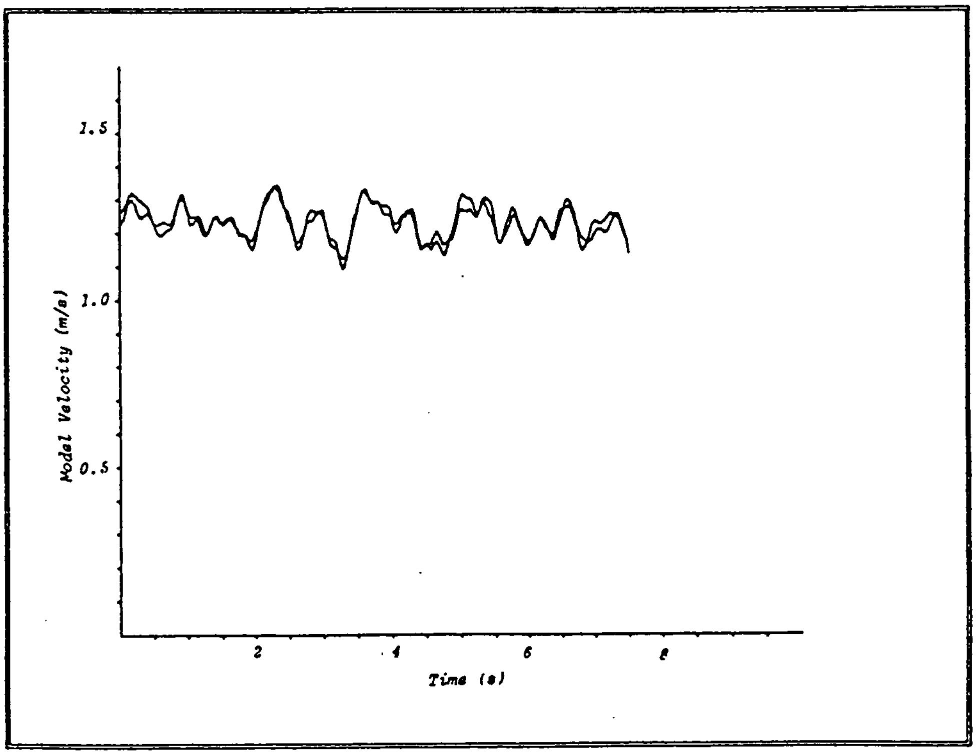


Figure 2.10 Repeatability Tests

Chapter 3. Subsequent Development

3.0 Chapter Summary

The Chapter starts with a description of the various system developments carried out on starting the PhD. The development of a new test format to improve repeatability is then explained. The Chapter then details an extensive set of tests carried out as a demonstration series for the Port Pendennis Americas Cup Challenge. The Chapter concludes with a discussion of further improvements considered before the decision was made to build an entirely new system.

3.1 Introduction

The testing carried out for the final year project proved that the towing system as it stood was capable of producing consistent, sensible results over a broad range of test conditions. However, there was room for much improvement. On starting the PhD the aim was to try and solve many of the shortcomings and problems.

This development was spurred on by the intention of the Port Pendennis Americas Cup Syndicate to carry out testing at the tank if certain improvements were implemented.

3.2 Roll Damping

3.2.1 System Configuration

Apart from the obvious lack of a side force component the most serious problem was the absence of realistic roll damping.

Yachts develop a large amount of aerodynamic damping due to their sails, especially when close hauled. The initial towing system had no provision for artificially applying such a constraint.

The magnitude of the required damping torque was estimated from simple aerodynamic theory and several candidate solutions evaluated. At this point the size of the Americas Cup models to be tested was uncertain. The calculations were therefore based on the 12-metre model to obtain a benchmark. They estimated the required damping torque to be 3.5 Nm/rad/s.

Some form of fixed reference against which to act was required. The tow loop itself proved to be far too floppy to be a serious candidate. However, a separate reference wire under a high pre-load may have worked. Two such wires arranged as a parallel track was another possibility. However, the most promising solutions all required a rigid track against which to act.

The tank has an overhead crane system, and suspended from this is an old section of galvanised radio mast. Although far from straight and fairly flexible, the mast could be used as a base for a rigid track.

After much discussion the roll damper configuration chosen was suggested by Peter Woodhead.

A tow tube transmits both the towing force and the damping torque between the rigid reference track and a model mounted roll damper. This system has many advantages including light-weight, good directional stability characteristics, and minimal off axis model constraint.

3.2.2 Track Design

The track itself required careful design. The prime considerations were; straightness, a smooth rolling surface, rigidity and mobility. Mounting the track on the relatively floppy radio mast provided an interesting design challenge.

The positioning repeatability of the mast was first improved by fitting adjustable strops in addition to the crane hoists. The length of each strop can be adjusted while under load by means of a yacht rigging screw, allowing coarse straightening of the mast. However, the track still needed to be fully adjustable to allow local alignment.

A fair degree of resilience was also required in the track mountings to prevent over stressing while it was being positioned.

The best combination of rails was deemed to be one round rail for directional guidance, with a flat reference track for rotational restraint.

The scheme finally chosen is shown in Figure 3.1, it proved to be simple and cheap to make, and efficient in operation. The two track rails were mounted via alloy spacers to horizontal 'Handy Angle' cross members. These were in turn attached to one of the lower mast longerons using alternating diagonal slinging braces, also made from 'Handy Angle'. The union was made using conventional toughened plastic pipe clips. Finally the track was rotationally restrained by occasional cross braces to the opposing mast longeron.

The round rail is made from 20 mm centreless ground EN1a steel rod, drilled and tapped to accept the spacers and joining dowels. For the flat rail cold rolled 40 x 12 mm flat stock was deemed to be of sufficient accuracy. These are again drilled and tapped and a smooth union is ensured by milled clamp blocks between sections. The assembled lengths of track are held firmly together by pre-stretched rope strops.

The support spacing was chosen to keep the deflection of the flat rail to less than 0.25 mm under a maximum applied torque of 10 Nm.

The resulting structure gave exactly the right mix of rigidity, resilience and adjustability.

After rough adjustment the track was accurately set up using a low power laser to an acceptable accuracy. On removal and replacement a fair degree of alignment was maintained. For serious testing the track needs to be quickly re-adjusted after coarse movement.

To prevent the mast developing low frequency oscillations under load, pre-stretched rope braces to the tank walls were included at intervals.

3.2.3 Reference Car Design

In order that the towing force and damping torque be transmitted efficiently, a small, rigid, light-weight car (L.W.C.) was designed. The car needed to be pre-loaded onto the track to prevent lost motion and backlash, and to allow a degree of dirt tolerance without jamming. It was accepted that the car and tow rod would add significant inertia, friction and stiction to the system. To this end a very lightweight solution was sought, with all rollers ballraced.

The T-configuration car shown in Figure 3.2 seemed to offer the best compromise. The pre-loading of the bearings is accomplished purely by deflection of the car struts. Radiused tyres are required for the flat rail rollers, to prevent scrabble and binding. The ballraces on the round rail bear directly on the steel track.

3.2.4 Damper Mechanism

The roll damping mechanism itself could take one of several forms. Desirable features included; the usual lightweight and small size along with adjustability of the damping coefficient.

Mechanical solutions using concentric cans and heavy grease were considered but might prove difficult and messy to adjust.

An active electronic damper seemed more appropriate. Several small motor/tachogenerator/gearbox combinations were available at the Wave tank. None of them were ideal but a good first pass was made using a Portescap servo drive system previously used in one of the Wavepower test models. The low inertia motor is fitted with an 18:1 low backlash gearbox. A simple active damping circuit was built and tested allowing the applied damping to be chosen in the range: 0-1.0 Nm/rad/s.

The full specification could have been met with the 52:1 gearbox available from Portescap. However, funds were not available at the time. It was anticipated that the gearbox would be upgraded when money became available.

The unit had to be self contained and no external power supply was available at the model. The damping circuit was designed to run off two 7.2 volt Nicad battery packs. These proved to last for approximately 2-3 hours in service, and could be recharged in less than 15 minutes using a special fast charger. The unit was then mounted on a convenient chassis and sealed using self-amalgamating tape and silicone sealant.

The tow rod is made from thin walled alloy tubing and is fitted with Huco zero-backlash universal joints at each end.

A schematic of the complete system is shown in Figure 3.3.

Tests were carried out to examine the performance of the damper. Visible roll motion was eliminated and on examining the tachogenerator output on an oscilloscope the roll velocities had been dramatically reduced. Unfortunately due to the rush no actual readings were ever plotted.

3.3 Magnetic Latch

The Mk.1 towing system had no provision for accurately positioning and aligning the model before the start of each test run. At the time this was not viewed as a serious problem because the testing was not synchronised with the wave generation. However, it was anticipated that this would be rectified so some form of simple, repeatable starting position and orientation were required.

Many alternative methods were examined. They ranged from simple mechanical end stops to active solenoid latches.

It was deemed desirable that the model be braced forwards against some form of restraint of the brake line. This was to allow a clean break off at the start of the test without undue snatching of the tow loop. Both tow car and brake line were thus made free to slide on the tow loop within the restrictions of end stops. This allowed a simple scheme shown in detail in Figure 3.3 to be implemented.

The brake line slider was equipped with a magnet keeper that was backed onto the permanent magnet latch. The model could then be pulled forward to it's starting position and braced with an appropriate holding force.

Although the mechanism proved to be very reliable, some form of latch status signal was required for the computer controller. Otherwise the computer would be unable to abort the test if the latch was inadvertently broken during the pre-tensioning or brace up phase. The bracing force was considerable and the model could have easily covered the length of the tank at high speed before any problem was detected.

Various proximity switches and hall effect sensors were looked at, but in the end a simple mechanical lever micro-switch proved to be adequate. The switch was connected to the BBC controller via a spare analogue channel.

Details of the software modifications to utilise this system are included later.

3.4 New Calibration Techniques

As mentioned earlier the accuracy of the original force calibration was open to doubt. Some form of load measurement for direct calibration was required.

The wave tank had some conveniently rated strain-gauge loadcells with a range of +-20 Newtons. This was used to directly calibrate the tow force. The loadcell was attached to the tow-wire loop near the winch drum. The computer then stepped through the appropriate range of output values and the loadcell output noted.

On plotting the results it appeared as though the first few data points were suspect, while the rest of the point showed a good linear relation.

The problem was attributed to stiction in the motor. A common technique to reduce stiction in open loop drives is to superimpose a high frequency "dither" signal. The frequency of the signal was chosen to avoid the natural resonant modes of the mechanical system. The amplitude is then trimmed so that the signal is just visible on an oscilloscope.

The calibration was then repeated revealing a completely different set of results. These were then quickly rechecked with a different dither frequency before being adopted as the calibration standard.

The velocity dependent friction current was then re-checked.

The velocity was recalibrated using the same scheme as before.

Now much more confidence could be placed on the accuracy of the apparatus.

3.5 Model Preparation

3.5.1 Ballasting

The model used in the 4th year project was roughly ballasted to float with a sensible trim and heel angle. Since the model was not being used for detailed performance prediction or comparative analyses the absolute values of displacement, moments of inertia, centre of gravity location and model attitude were not critical.

However, on receiving the two Americas cup models, some more accurate method of setting these various parameters was needed. The required fullscale characteristics for the new models are detailed in Appendix A.

The displacement of the models was easy to fix by simply weighing the various components and weights being used.

Ballast weights were split into four main categories: pure ballast, positioned at or near the required centre of gravity; inertia trimming weights; bow down moment weights; and a heel angle trim weight. The function of each is explained below.

The model was first loaded with all weights at some median position and the fore and aft trim set by moving the main ballast.

Pitching motion of a yacht is the dominant cause of added resistance. Since pitch response is heavily dependant on pitch inertia, it is very important that the models be ballasted accordingly. Some form of pitch inertia standardising rig was required.

Two tension springs were accurately calibrated over the relevant load range. The models were suspended from these, the period of free pitch oscillation of the system yields the pitch inertia. The pair of trim weights were then moved in opposing directions to set the pitch inertia to the required value, without affecting pitch trim.

The elevated position of the centre of effort of a yacht's sailplan gives rise to a bow down moment proportional to thrust. The towing system still towed from deck level. Two weights can be moved within the hull to artificially generate this couple without affecting the pitch inertia of the model. The positions required do not vary linearly. A simple BBC Basic program was written to calculate the weight positions for any bow down moment/pitch inertia combination.

Finally the heel trim weight can be moved as required to artificially generate the required heel angle.

No attempt was made to set the centre of gravity to a realistic vertical position because it was agreed that this was not relevant for the initial comparative study.

3.5.2 Turbulence Stimulation

As mentioned earlier, a critical part of ensuring that the results from small scale model tests are reliable, is the simulation of full scale Reynolds numbers. This is done by 'tripping' the flow round the hull from the laminar to turbulent regime, at a representative distance aft of the bow, and on all but minor underwater surfaces.

The recommended method at the time of the 4th year project was by applying bands of coarse sand at the relevant stations. This has been proven to be unreliable under certain circumstances.

A more satisfactory method has been found to be the application of small cylindrical studs at intervals in appropriate places. These were applied to the model as per instructions from Glasgow Ship Science.

There seem to be no hard and fast rules concerning this "black-art", but any unreliability in the test results is often traced back to this problem.

3.5.3 Steering

Conventional towing systems do not suffer from steering problems because the model is invariably restricted in yaw and sway. However, the freedom of motion supplied by the original and upgraded system necessitates a careful approach to this problem.

Various proposals for automatic steering systems will be discussed later in this thesis. However, all the tests carried out so far with the towing system have relied heavily on the directional stability of the model under test.

All models are fitted with adjustable rudders. A large number of dummy tests are carried out in varied conditions until a satisfactory rudder position is found.

Setting up the two Port Pendennis models gave no problems as they are conservative hull forms.

3.6 Control Development

The test control system was also developed considerably. As mentioned in the previous chapter the 4th Year project system was not synchronised with wave generation. This meant that fully automated testing was not possible. It was seen as a fundamental requirement that the system was fully automated.

Various alternatives were considered. In the end it was decided that the simplest solution was to control the tow cycle with the same BBC Micro that is interfaced to the wave synthesiser. Peter Woodhead wrote new communication software to allow the two functions to be performed by the same computer. This means that the Sun computers were now in full supervisory control of the towing system and waves.

Towing and wave parameters were down-loaded together over the RS232 link, then the Sun sent out a start signal to synchronously start the waves and towing cycle. After completion of the test run the sampled data was transferred directly back to the Sun before the next test was started.

The test cycle itself was extensively modified to incorporate the magnetic latch.

Peter Woodhead also wrote a set of analysis programs on the Sun system. These allowed much more rapid, in depth, analysis of the test results and were extensively used and modified.

A schematic of the new control system is shown in Figure 3.4.

3.7 Test format development

All the results presented in this and the following section are normalised values to preserve a data-confidentiality agreement with the Port-Pendennis Challenge. They are reproduced with the kind permission of this syndicate.

Initial tests were carried out with the Sceptre model used for the 4th Year Project work. Once again development of the acceleration phase was a process of manual iteration towards the optimum routine. Unfortunately this process had to be carried out for each change in model or tow force. However, the process became much quicker as experience grew.Control of the reverse and latching routines also required a fair amount of tweaking. However, once this had been done it was relatively independent of the model being tested.

With the system now working reliably throughout the full test cycle testing could begin in earnest. All the initial system development was carried out using the Sceptre model since the Port Pendennis models had not yet been delivered. As soon as the new models arrived one of them was used for all subsequent development to ensure that the system was optimised for models of their displacement and configuration. Testing concentrated exclusively on proving the repeatability of a sweep of 50 wave frequencies at an approach angle of 40 degrees, with a single amplitude and one tow force. The initial tests showed up several problems with the system.

The first most obvious feature of the results was a humping and hollowing of the mean velocity across the frequency sweep. This was a repeatable result. For a constant force towing system, precise prediction of the model's position during the test is not feasible. This means that it is difficult to sample over a complete number of wave cycles. The result is that as the wave frequency is incremented the start and end phases vary systematically. This effect was not observed in the results from the 4th year project system due to the fact that testing was not synchronised with wave generation.

This problem was solved by running each test frequency at 4 different wave starting phases. Figure 3.5 shows four such sweeps. The graph shows the expected trend that phase effects are large at lower frequencies where wave-by-wave velocity transients are significant and become progressively less important at higher frequencies where the model's velocity fluctuations are inertia dominated.

The towing system was now producing smooth output. However, repeatability trials revealed a slow drift of average velocity values. This can be seen in Figure 3.6. Figure 3.7 shows the same results on an expanded Y-axis for clarity. It was concluded that this was due to the diurnal air temperature fluctuations experienced in the tank. The precise cause of the observed drift was never ascertained.

The most likely cause was deemed to be either drift of the tacho conditioning circuit or fluctuation of the mechanical friction within the system due to grease viscosity or differential expansion.

To ensure that the tension of the tow loop remained as constant as possible, a pair of low rate tension springs were added to the loop behind the tow car forward stop. This seemed to have little effect on the drift. Resistors in the conditioning circuits were changed for 1% metal film if they were not already of this grade. Again little effect was noted.

The problem was eventually solved by carrying out still water control runs between each batch of four wave runs. The mean velocity from these control runs were then used to "correct" the adjacent wave runs. This led to a remarkable degree of repeatability. Figure 3.8 shows the same results as Figure 3.7 but this time after "correction" using the still water runs. The full test structure used is shown in Figure 3.9. The absolute value used in the "correction" process was the mean of all the still water runs carried out for that sweep.

3.8 Results

Over 5500 tests were run with the system in this configuration. Most of these runs were exhaustive repeatability tests. Each sweep of frequencies represents 250 test runs and took just over six and a half hours to complete.

Figure 3.10 shows four identical sweeps and gives an idea of the system repeatability. All the runs fall within a general band of less than +-0.5% of the still water resistance, with most of the curve showing a repeatability of better than +-0.3%.

During the main block of testing 4 sweeps were carried out for each model. The results from these were in turn averaged to give the curves shown in Figure 3.11. The absolute still water values for these curves were the average of all the still water runs carried out in the series, a total of over 200 runs.

As can be seen from Figure 3.11 the models show very similar resistance characteristics. Still water velocities are similar and any differences noted in speed loss are within the experimental error defined by Figure 3.10.

3.9 Planned Improvements

3.9.1 Automatic Steering

As mentioned earlier a major omission from the current towing system was any automatic steering facility. The problem is not system specific and indeed would be essential on the proposed new towing rig.

Stabilising a closed loop steering system is not an easy task. Some initial attempts were made on paper to solve the problem. These mainly centred around mechanical arrangements to make the various angles and velocities available to a controller.

Developing ideas for this area of the project threw up some interesting methods for attitude and motion measurement in general. Many of these ideas found uses in the new towing system design and will be discussed in greater detail later.

3.9.2 Portescap Closed Loop Force Correction

Several schemes for increasing the accuracy of the constant force drive with the existing set up were considered. The most promising is discussed in detail below.

The proposed modifications involved a local force correction drive on the light weight car, in parallel with the existing winch. A system very much analogous to a 'woofer-tweeter' hifi speaker set up.

The central bearing on the light weight car would be replaced with a small D.C. servo-motor. A loadcell would be fitted between the tow rod and the trolley. Closed loop control would thus be used to greatly increase the accuracy and consistency of the tow force.

The motor would have an integral tachometer allowing local velocity measurement.

The drive signal to the main motor would then be comprised of the original demand signal, plus an extra term based on the integral of the drive current to the closed loop correction motor. This would ensure that the main winch provided most of the D.C. load.

A schematic is shown in Figure 3.12.

To decide whether the idea was feasible, a force and velocity specification was drawn up.

Assuming a worst case perturbation velocity, and a total trolley mass of 1 kg, the maximum force required to drive the trolley alone turns out to be 2 Newtons. Friction and stiction would demand 1 Newton. Parasitic inertia from the tow loop and the communication/power wires a further 2 Newtons. This gave a maximum drive specification of +-5 Newtons.

A maximum model velocity of 2.5 m/s was deemed to be acceptable.

The best candidate motor was the un-geared version of the Portescap motor used in the roll damping unit. There were plenty of these spare at the tank and the compact lightweight casing with integral tachometer lent itself to this application.

With the drive pulley diameter set at a sensible minimum of 16 mm the velocity specification would be nicely met with a +-20 Volt supply. To generate the required force however, the motor would require a 3C/Watt heatsink. This would take the form of a finned, interference fit collar around the motor casing.

Drive to the motor would have been supplied by a similar circuit to the damper rig, except with a force rather than velocity feedback signal.

The tachometer e.m.f.constant of 0.84 V/1000 r.p.m. gives a convenient output of +-3 V at maximum velocity.

Drive would be transmitted to the track via a V-pulley. Several deigns for this were considered. Many polymers have friction coefficients on steel of 0.4-0.5, independent of lubrication. The most promising was a Lexan V-sleeve over an alloy boss. This had the advantage of reducing the required closing force. The sleeve would be post machined after assembly to reduce eccentricity.

All the associated electronics would be mounted on the trolley to reduce the number of wires to the shore.

The loadcell would initially have been a commercial unit similar to the one used for tow force calibration.

Finally, some form of latching system would have been required to lock the trolley to the tow loop during the test run. The required force would be low, so a permanent magnet latch would have been incorporated.

The decision to build a completely new towing system, avoiding the need for retro fit improvements, prevented this system from ever being realised.

3.10 Conclusions

The fourth year project system was extensively developed, the following improvements were made.

- Roll damping was introduced to simulate the damping action of a yacht's rig while sailing close-hauled.

- A latching system was implemented to ensure a repeatable starting position and orientation.

- The system was fully synchronised with the wave generation system allowing fully automated testing.

- A simple ballasting scheme was devised to allow accurate model trimming.

- A test structure was developed to ensure that results were highly repeatable.

A series of tests was carried out on two 1/9th scale AC Class models. The following conclusions were drawn.

- A comparative study was carried out for one tow force with a single set of wave conditions.

- Over 1000 runs were carried out for each model to ensure that the results of the comparison were reliable. This testing block took 4 days to complete.

- The two models exhibited very similar speed loss characteristics.

- Differences between the two models were of the order of the experimental repeatability bounds.

Various additional developments were considered but were rejected in favour of building a completely new system.

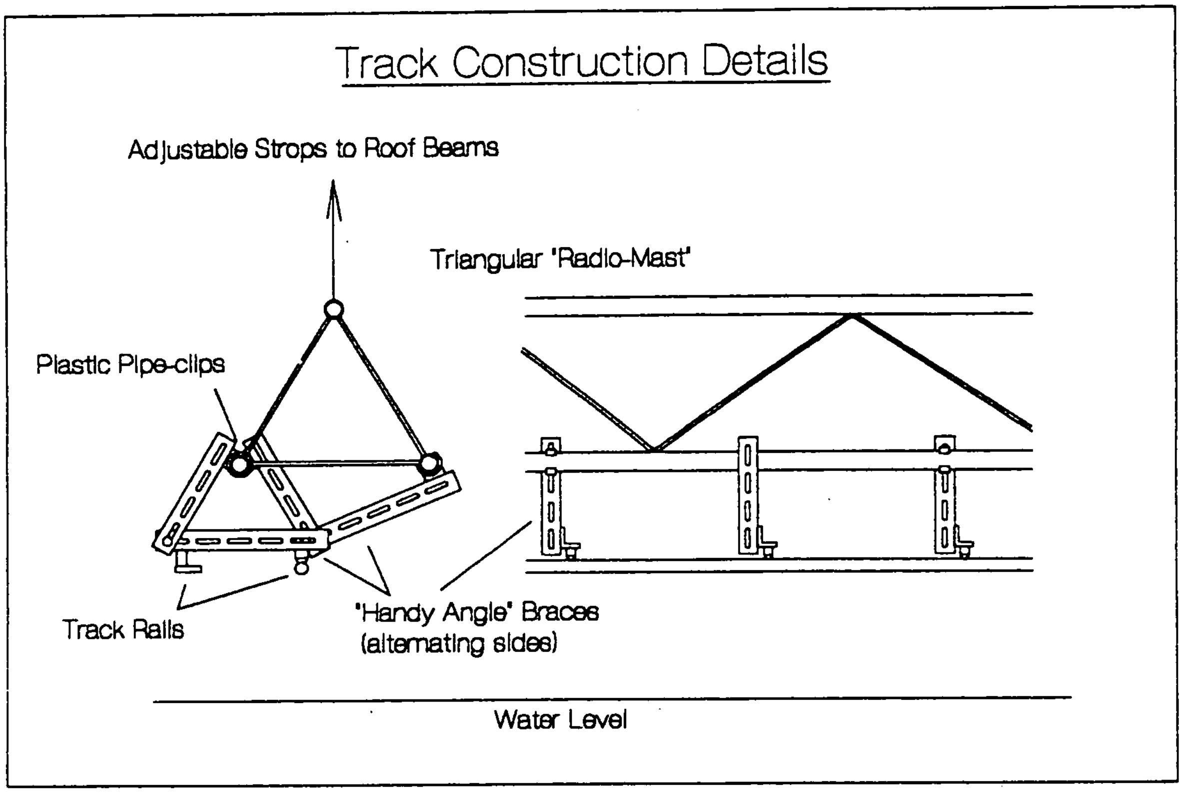


Figure 3.1 Track Fabrication Details

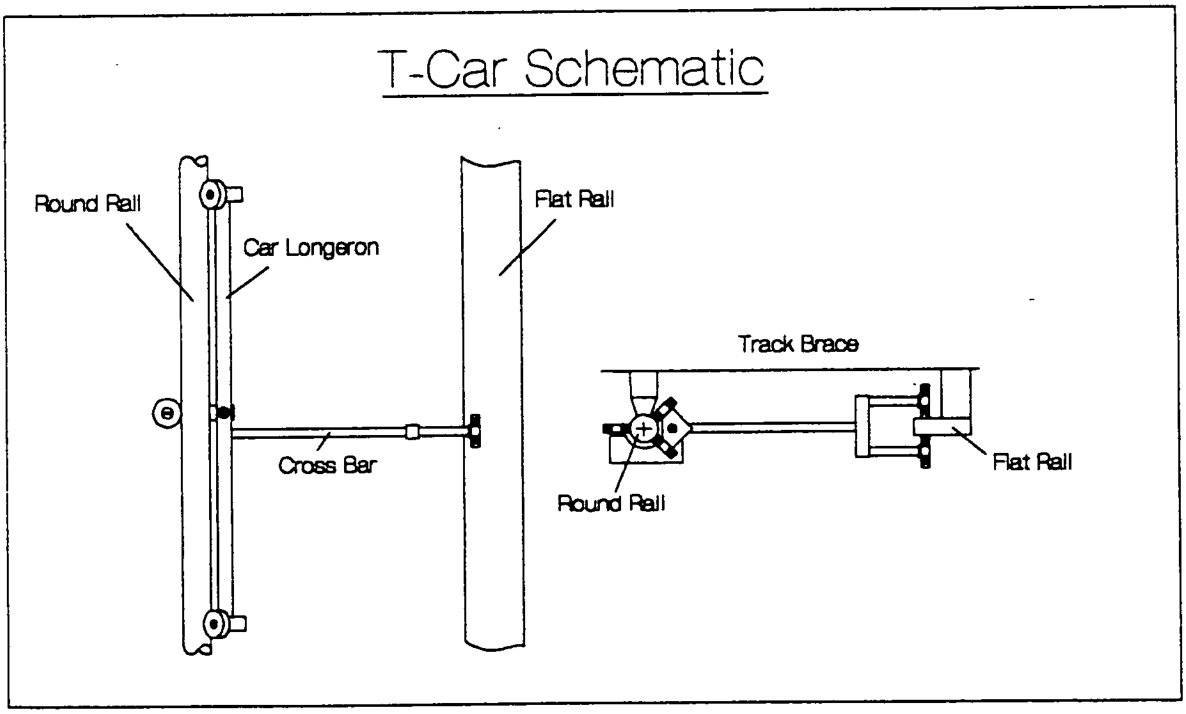


Figure 3.2 T-Configuration Light-Weight-Car

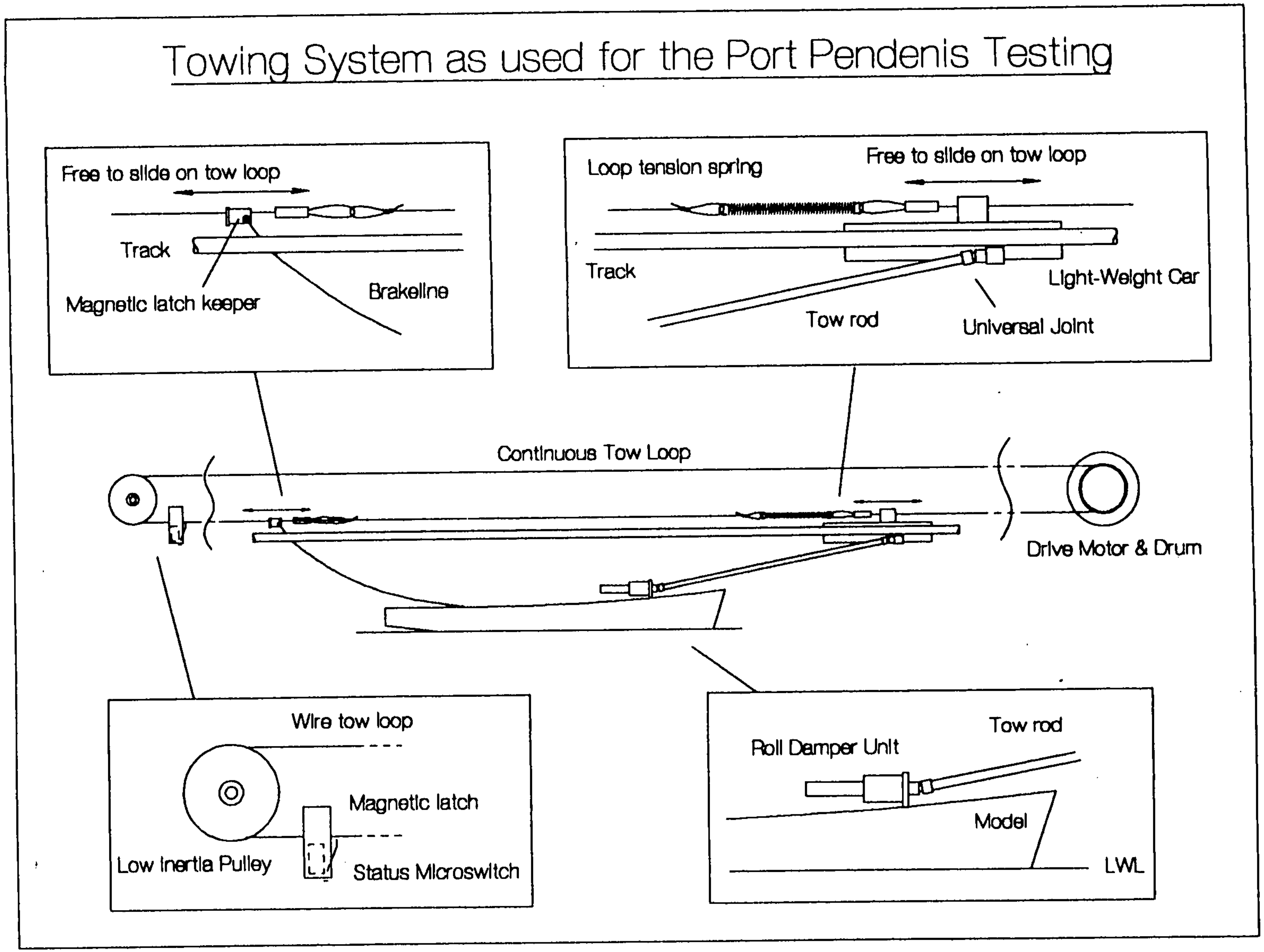


Figure 3.3 Schematic of Upgraded Towing System

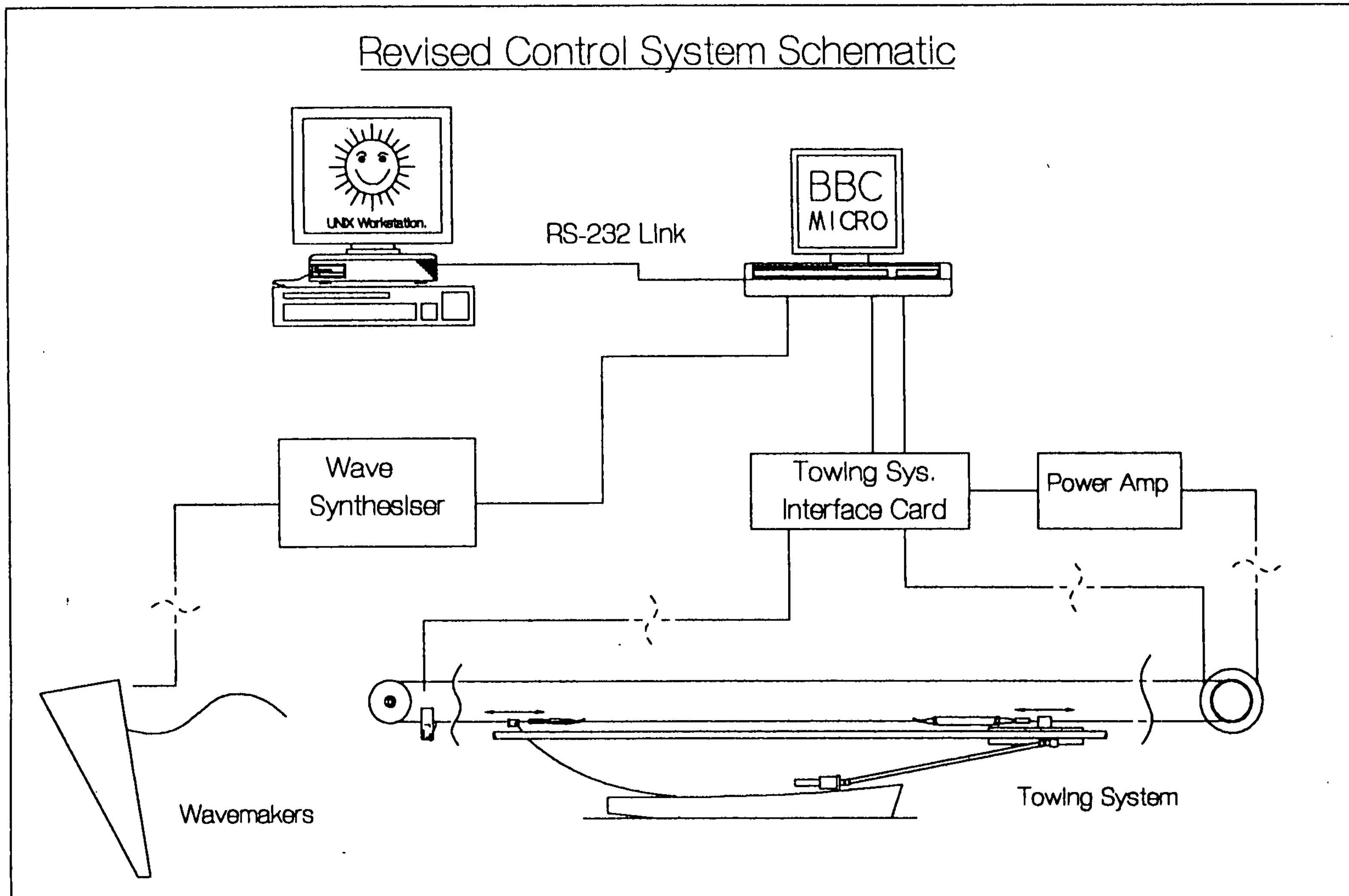


Figure 3.4 Schematic of Revised Control System

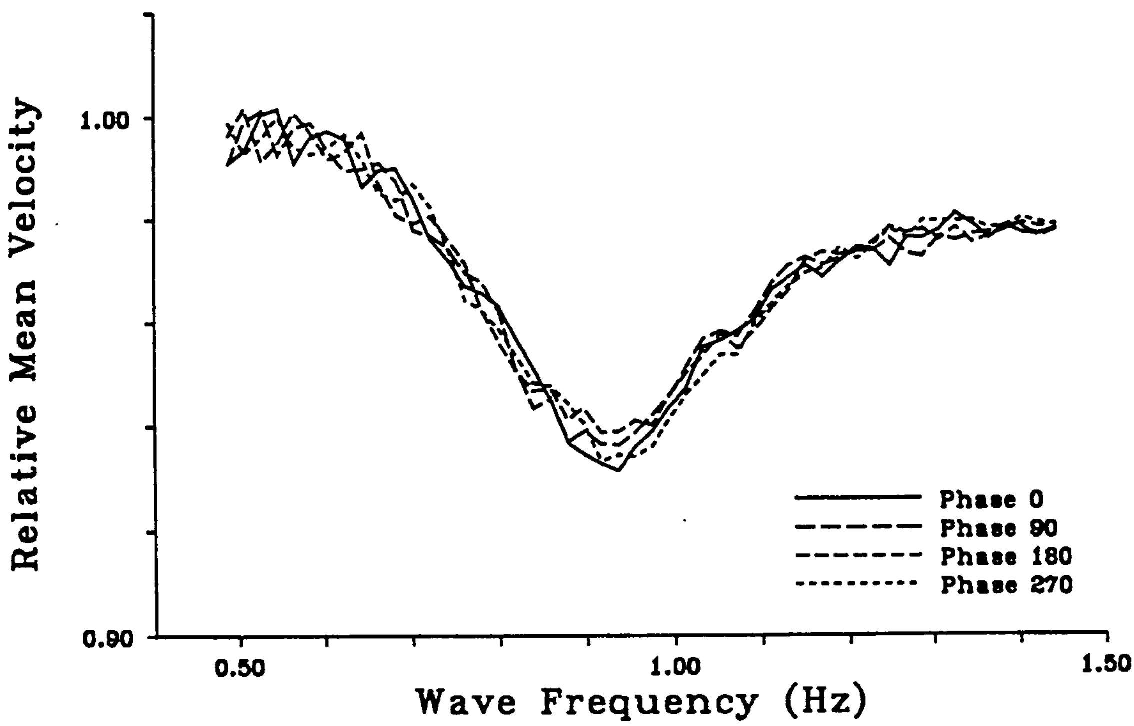


Figure 3.5 Effect of Wave Starting Phase

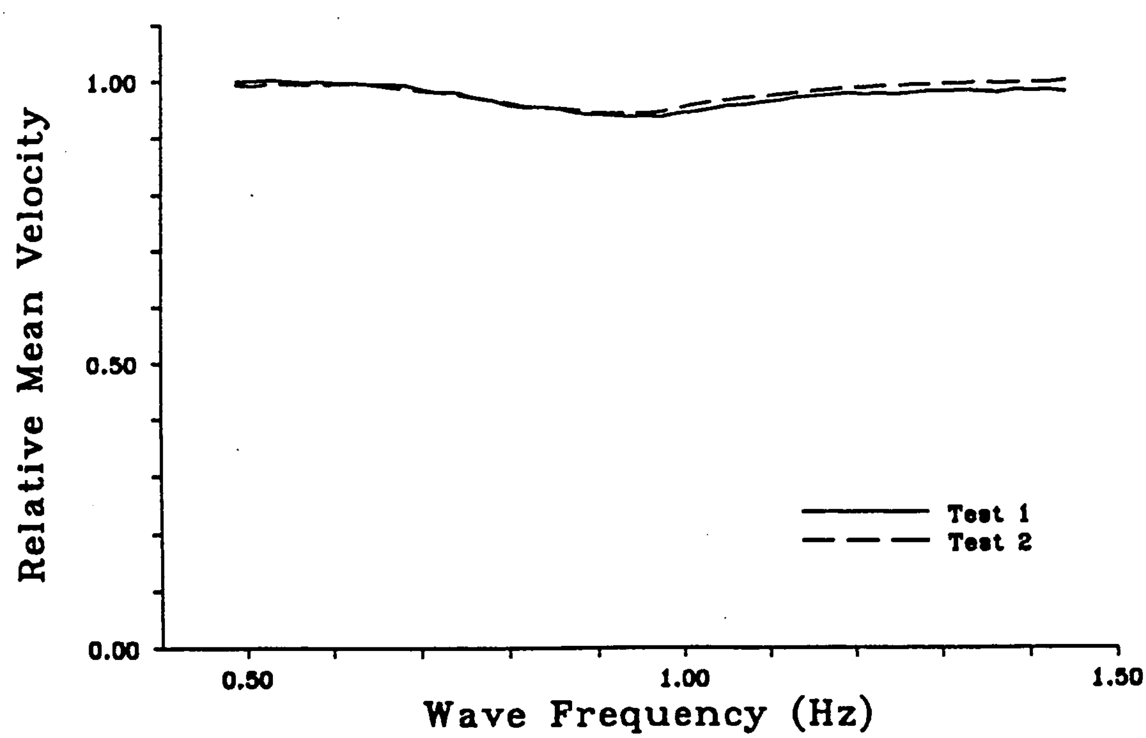


Figure 3.6 Temperature Drift of Results

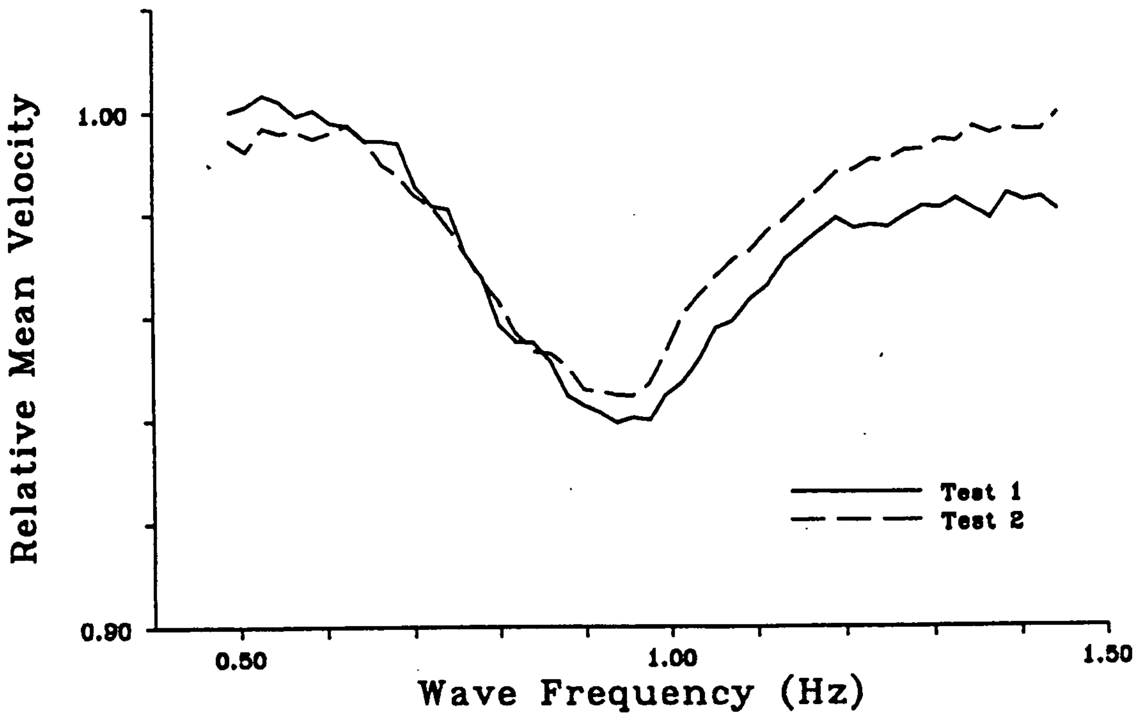


Figure 3.7 As Figure 3.6 but with Expanded Y-axis

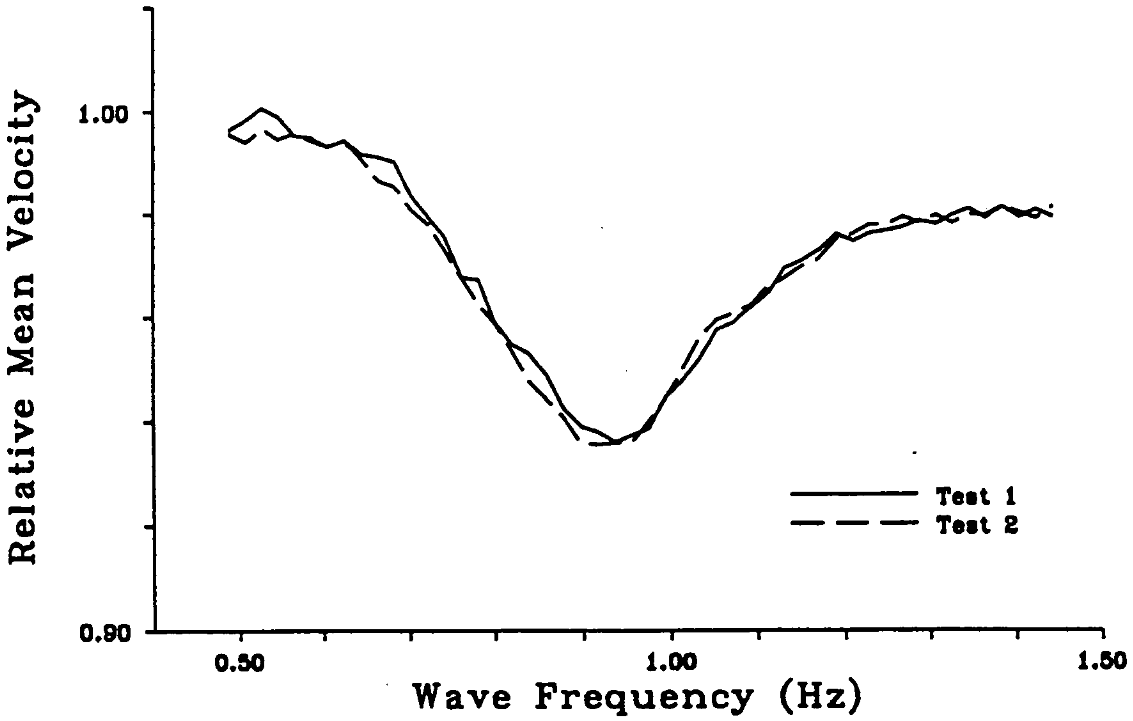


Figure 3.8 Effect of Still Water Correction Runs

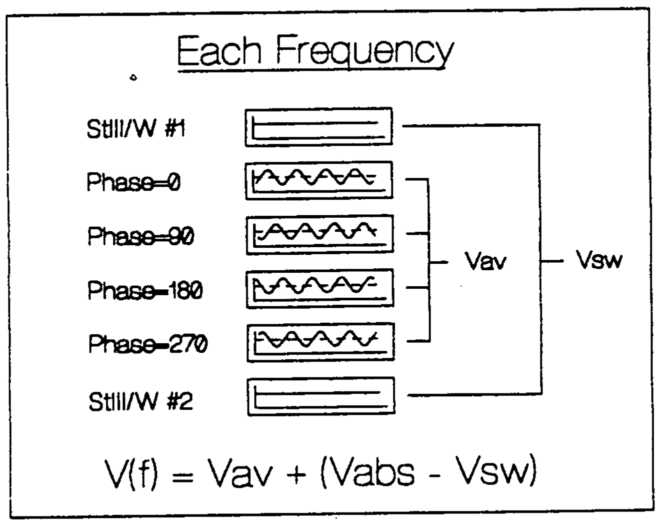


Figure 3.9 Full Test Format

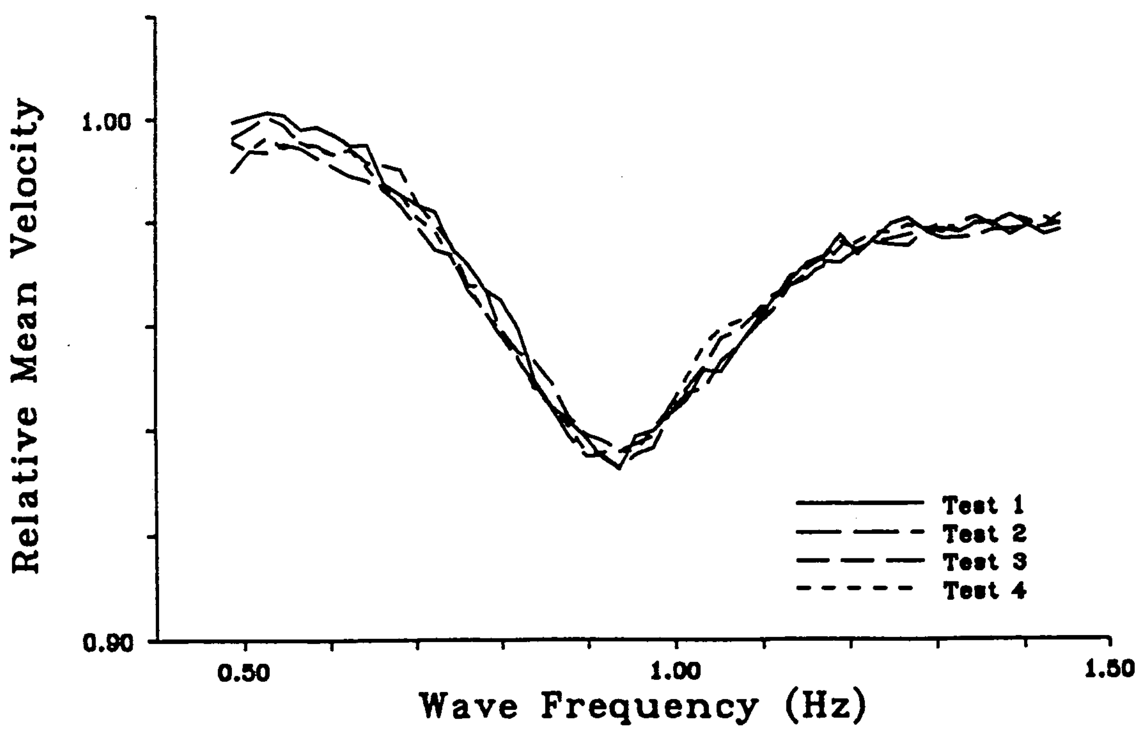


Figure 3.10 Repeatability Tests

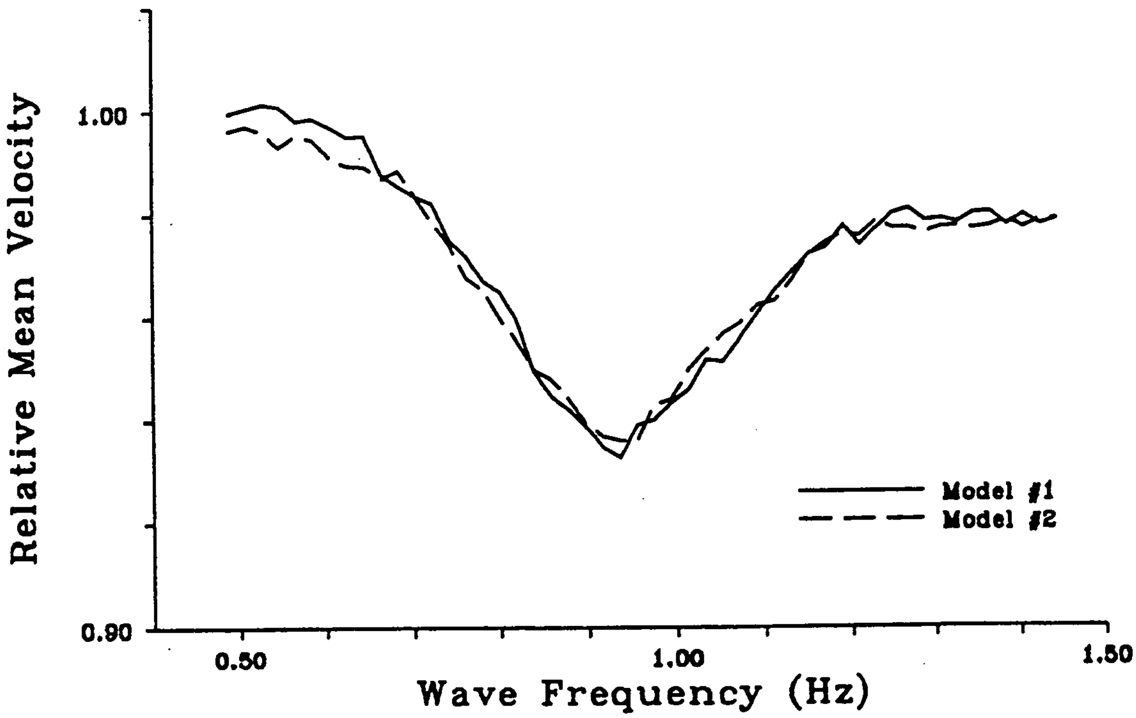


Figure 3.11 Absolute Comparison Between Test Models

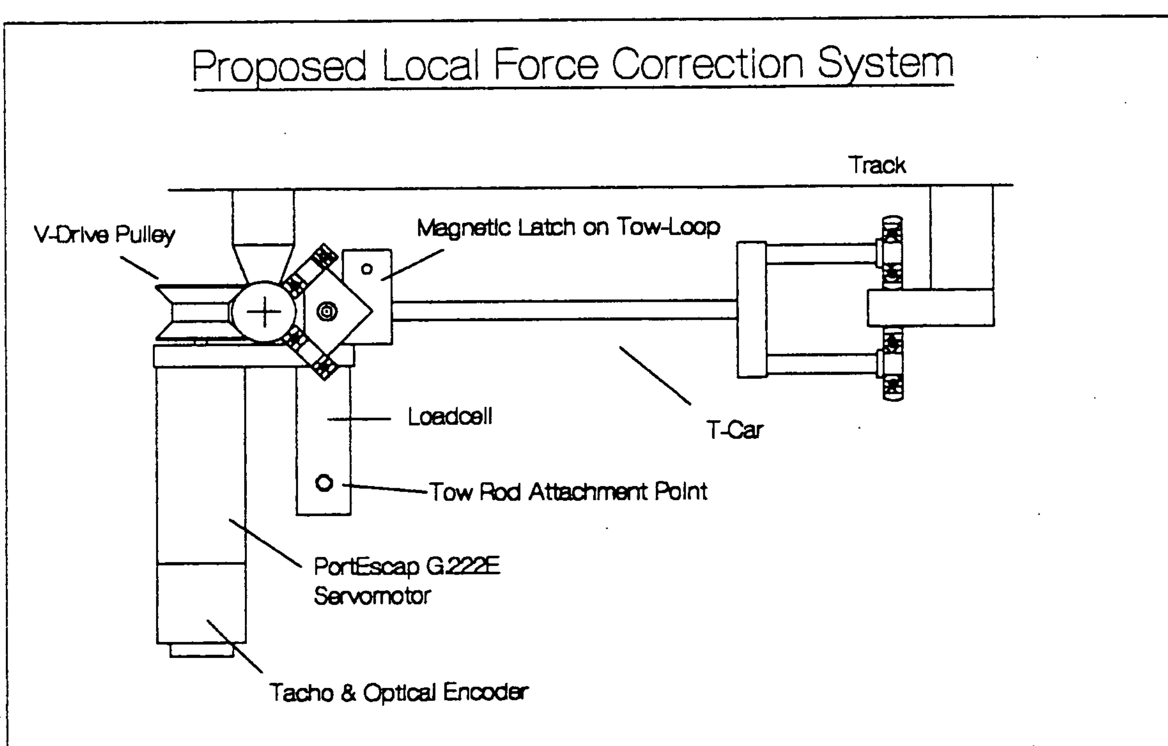


Figure 3.12 Schematic of Local Force Correction System

Chapter 4. New System Concepts

4.0 Chapter Summary

This Chapter starts with a summary of the specifications that were laid down to aid design of the new towing system. It then moves on to discuss several different configurations that were studied. The Chapter ends with a conceptual description of the system chosen to be implemented.

4.1 Introduction

Despite the promising study of the local force correction system described at the end of the previous Chapter, it was decided that the best way ahead was to make a fresh start by designing a completely new system, rather than continual upgrading of the old winch drum set-up.

The system envisaged was to be designed to tackle the difficult problem of angled force towing and was to use full closed-loop control throughout. A study of the force and dynamic performance required was undertaken. This allowed candidate configurations to be critically evaluated.

4.2 Specifications for the New System.

The specifications for the new system fell into four main areas:

- General Conceptual Requirements

- Technical Specification

- Control Requirements

and,

- Practical Considerations

4.2.1 Conceptual

The new system must be capable of applying a controlled force on the test model, preferably in any direction in 3-D space, but certainly in any direction on a horizontal 2-D plane.

This force will be made up of various components designed to simulate the real forces on a yacht as closely as possible.

Some form of staged development path seemed desirable to prove system components on a simple configuration before the full multi-degree-of-freedom was attempted.

4.2.2 Technical Specifications

The size of models used for previous tests seemed to match the wave tank wave envelope nicely. It was anticipated that the new towing system would be used with models in the same size range.

Thus the required tow forces and velocities would be similar.

However, the angled component of the tow force leads to greater absolute force requirements. In addition the rig would have to supply the roll damping force previously provided by a separate unit.

Towing from the centre of effort of the sailplan leads to another problem. The velocity transients dealt with by the old, deck level towing system were dominantly surge fluctuations. With its raised tow point the new system would have to cope with much larger transient velocities due to pitch and roll motions.

To design for these criteria some form of combined force and velocity specification was required, to try and define a sensible worst case situation.

The following individual worst cases were identified:

Towing Loads

For a typical model of length 2 metres, the maximum likely tow force for simulated close hauled work would be in the region of 20-25 Newtons acting at approximately 70 degrees off the bow.

As the towing angle is reduced, the required force diminishes to around 10 Newtons for straight ahead work.

Roll damping requirements

For the same model peak damping torques of 10 Nm through the centre of gravity would be required. This figure was considerably more than the original requirement for the old system. The revised specification was chosen as a result of the new configuration, which allows large torques to be applied easily. With a tow height of between 0.75-1.0 metres this translates into a peak damping force of 10 Newtons.

Maximum Motion Specifications

From experience, and the tank bandwidth limits, a sensible set of worst case pitch roll and surge motions were arrived at. They are as follows:

- Pitch: +-10 degrees at 1.0 Hz
  +-5 degrees at 1.5 Hz

- Roll: +-5 degrees at 1.0 Hz
  +-2.5 degrees at 1.5 Hz

- Surge: +-0.03 metres at 1.0 hz

These generate the following maximum accelerations at the tow point:

- Pitch: +-7.0 m/s^2

- Roll: +-3.5 m/s^2

- Surge: +-2.0 m/s^2

The towing rig would have to be capable of driving its own inertia at these accelerations.

The figures above give rise to a further criterion and that is the size of the operating zone. A sensible operating region over which the stipulated technical specification should be met was chosen to be square of side 0.5 metres.

Maximum Velocity

The new towing system was mainly intended for testing non-planing designs. For models of the suggested size a sensible maximum model velocity of 3 m/s was deemed to be ample.

Other

It was deemed desirable to be able to measure the model motion in all six degrees of freedom.

A loose target was set for the maximum tow force fluctuations. The heady goal of maximum transients of less than 1% of the tow force was chosen.

4.2.3 Control Requirements

Many details of the control specifications were difficult to determine before the system configuration was finalised. However, certain key requirements needed to be defined.

- All forces must be applied using closed loop feedback systems to improve accuracy and reduce drift.

- The system must be fully automated and synchronised with wave generation.

- The control scheme to be implemented should be designed to fit in with any long term plans for tank control, to allow easy upgrading.

- A real time control system must be implemented that has the capacity to compute and apply realistic damping, excitation and other types of control terms.

4.2.4 Practical Considerations

In addition to the technical specifications there were a number of practical considerations to be taken into account:

- Due to budget limits a total cost for the system of o3000 was set.

- The apparatus should, where possible, use as much of the hardware and technology that had already been developed to reduce time and cost.

- The new rig must not compromise the accessibility or performance of the tank in any way. Thus, the rig must be as compact as possible to allow rapid commissioning and removal.

- The system should be designed to allow a staged development path, with interim operational modes, in case time runs short.

4.3 Discussion of Several Proposed Configurations

Many different possible configurations were evaluated. Several of the more promising ideas are discussed below.

4.3.1 Ducted air fan

One of the ideas for the original final year project was a ducted air fan, Figure 4.1(ii). The concept was re-examined at this stage. Though conceptually elegant the system would have severe practical limitations. The free running nature of the system would make test automation impossible.

The force specifications would be hard to meet without using a very heavy fan unit. The problem would be compounded by the need for contra-rotating inertia to cancel out gyroscopic effects. This would mean that any realistic centre of gravity location would be impossible.

The rotational inertia of the fan would severely restrict the bandwidth of the system, making the application of representative damping forces difficult.

The concept was therefore rejected.

4.3.2 Tow Wire cage.

One scheme that initially looked promising was an array of winch motors mounted on a travelling frame, Figure 4.1(iii). The tension on each wire would be controlled to generate the required thrust vector. The system frame could possibly be carried along a widened version of the existing track.

However, the angle and operating range specifications would be hard to meet because side forces are hard to generate when the system moves significantly away from the median position. The available loop gain, and hence accuracy, attainable would be low unless the wire pretension was high. This would lead to further problems resolving the relatively small tow forces. The outer frame would have to be large and bulky, and the required rigidity would make it heavy.

In the end the idea was rejected.

4.3.3 X-Y Plotter

This solution was a serious contender until the final decision was made.

An angled tow rod joins the tow mast to an X-Y drive system, Figure 4.1(iv). The 'plotter' moves in relation to the model to maintain the required magnitude and angle of the applied force. The rig would have very little parasitic inertia and so would be fast in operation. The compactness and light weight of the frame would allow it to run on the existing track.

The main problem was that of braking, and controlling the model between runs. No fixed reference was available against which to act. Possible problems may have been encountered if a force reversal was required. The plotter may have to move unfeasibly fast as the load was reversed.

This promising solution was kept in mind while other avenues were explored.

4.3.4 Three Rocking Arm Solution

Stephen Salter suggested an interesting idea based on a 3-D wave gauge built for the wavepower project many years ago.

A schematic is shown in Figure 4.1(v). Three rocking arms transmit force to the tow point via pushrods. This allows a vector in any direction in 3-D space to be generated. With knowledge of the system geometry, the three arm angles dictate all the unknowns in the system, allowing easy computation of the component vectors.

The sectors would be driven by a pre-tensioned, push-pull belt drive. This would have no problem generating the required forces and would allow a very high loop gain to be achieved.

However, problems occur when the pushrod angles become too obtuse or acute. The only solution to this is to limit the angles by increasing the size of the whole rig. The size required to meet the system specifications became very large and un-wieldy. A purpose built gantry spanning the whole tank would have been required. This would be very expensive, intrusive and the available run length would be limited.

The initially exciting idea thus turned out to be too bulky for the operating range required and so was rejected.

4.3.5 Coupled linear Actuators

Another possible solution involved the use of coupled linear actuators. These could be used in either a 3-D or 2-D configuration. The 2-D case was chosen for serious examination. Figure 4.1(vi) shows a likely configuration.

This system could make use of the present track without extensive modification.

Several forms of linear actuators were considered. All forms incorporated two axis swivelling with pre-loaded pivot bearings to eliminate backlash and lost motion.

The most promising was the friction drive shown in Figure 4.2. The pushrod is pinched between opposing rollers to generate the necessary friction. The closing force is set by adjusting the pre-load on the pinch spring. Position and velocity would be measured on the squeeze rollers to prevent creep or slipping leading to errors. The entire pinch mechanism is pivoted on the motor shaft. This means that only one extra degree of rotation is required. Backlash is removed from the motor bearings by pre-loading them with the auxiliary rollers shown.

The unit is very compact. Most of the mass is concentrated at the point of rotation vastly reducing its effect. Very high linear velocities would be achievable, the only limiting factor being the rod mass. Such a drive system provides unrivalled rigidity, allowing very high loop gains and bandwidths to be achieved. This would almost certainly offset the extra friction and stiction incurred. The overall rig itself is simple when compared with the alternatives studied.

The main negative aspects are that the main car is large and therefore heavy. The wide base would mean that track alignment would be critical. The large mass of the car would mean that all but very low frequency corrections would be impossible. This would mean that the linear actuators would have to work very hard to maintain the demand force.

The inertia of the pushrod itself, although small would lead to significant tow force errors at high accelerations, unless compensated for. To allow the actuator room to pivot it would have to hang down below the track. This may cause track oscillation problems. Counter balancing of the pushrods would be difficult and parasitic load on the model would be significant at full extension. The 3-D configuration would of course cure this.

A staged development path first using the simpler 1-D configuration would have proved difficult due to possible confliction between the pushrod and the track

This configuration would probably have been chosen if Peter Woodhead had not come up with the elegant improvement described below.

4.3.6 Chosen System

A functionally similar system can be realised using two fixed length tow rods forming an isosceles triangle. The inboard ends of the rods would be mounted on independent lightweight trolleys. Tow point displacements parallel to the track would be followed by common mode motion of these two cars. Transverse displacements by anti-phase motions.

Controlling the force on each of the tow rod cars allows two force vectors to be demanded.

The trolleys would be driven from a much more compact main car, running on the same rails. The tow rod lengths required to meet the operating range specifications would leave enough room for the main power car to sit between the two tow rod trolleys. All drive motors would be mounted in the central car. Drive to the trolleys would be via a fixed version of the drive described in the preceding section.

The configuration is shown schematically in Figure 4.3.

This arrangement has several important advantages over the previous design.

The system would certainly be able to use the track as it stood, and maximum possible use of the tank length would be made.

The size of the central car is dramatically reduced. This would make accurate control much more straight forward. However, the clearance between the lightweight trollies and the central car means that control of the latter could be sluggish, without increasing the load on the tow rod drives. The reduced length of the centre car would also lower the accuracy with which the track would have to be aligned.

With the motors mounted on a rigid chassis the available loop gains should be higher still. The parasitic inertia of the tow rods themselves would be much less than that of the slewing drives, further increasing accuracy. The drives could be of a lower rating than the slewing system because of vector magnification due to the tow rod angle. The lightweight tow rods would also be very simple to counterbalance. A simple counterweight would work for all rod angles.

Yaw control of the model between test would be effected by a simple torque motor mounted in the model. During testing the system would be disabled.

One of the most important advantages is that the system allows a very progressive development path. Testing of most components can begin with a replacement 1-D system, to iron out control problems, before the full 2-D configuration is attempted. This would have been much harder with the previous system due to confliction between the track and the rear portion of the driven pushrod.

This gives the added major bonus that in 1-D mode the rig could be designed to be software configurable as either a constant force or constant velocity system. This would allow very accurate comparison of the two methods under controlled conditions.

With all these factors considered the rig described above was chosen as the configuration to be used.

4.4 Conclusions

A study of various potential configurations for the new system was undertaken.

A configuration suggested by Peter Woodhead was chosen to be implemented.

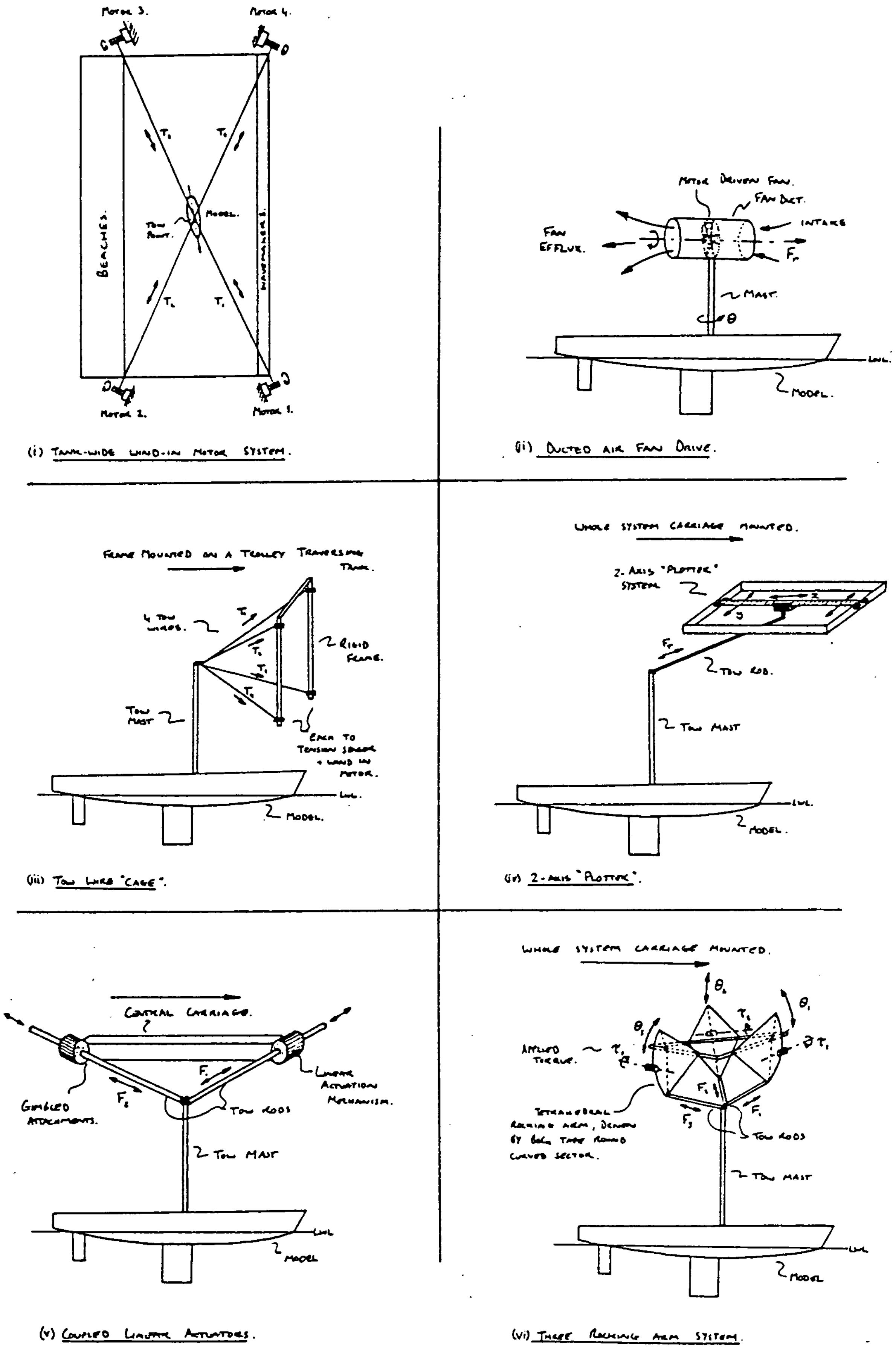


Figure 4.1 Alternative System Configurations Considered

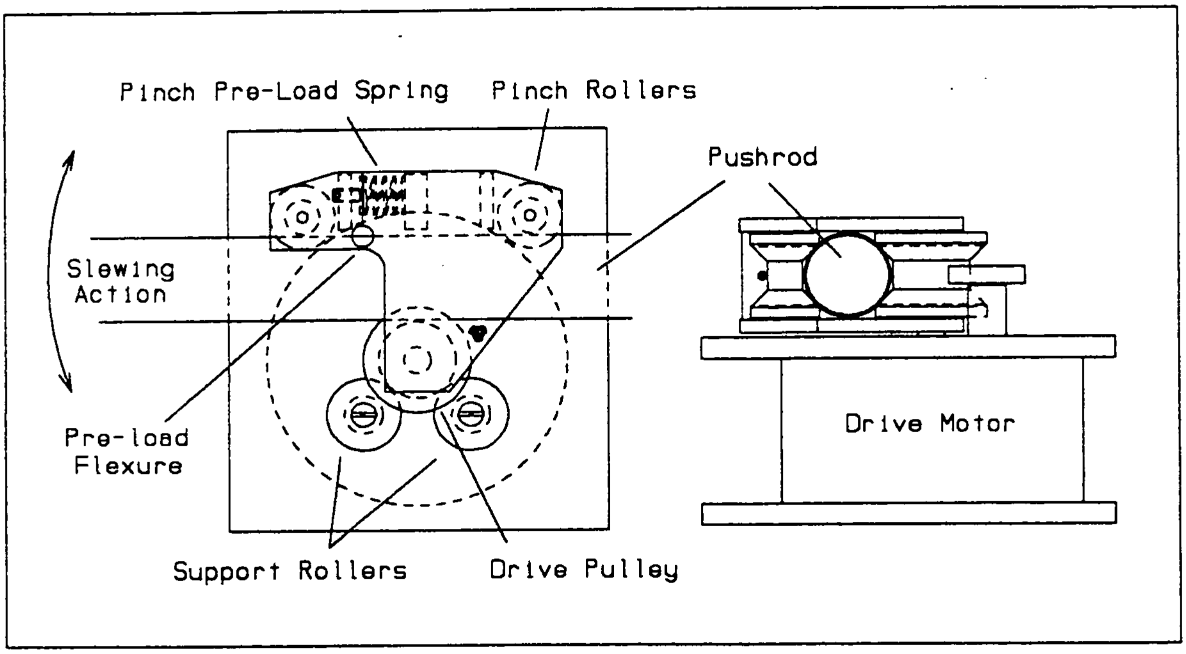


Figure 4.2 Slewing Pushrod Drive Detail

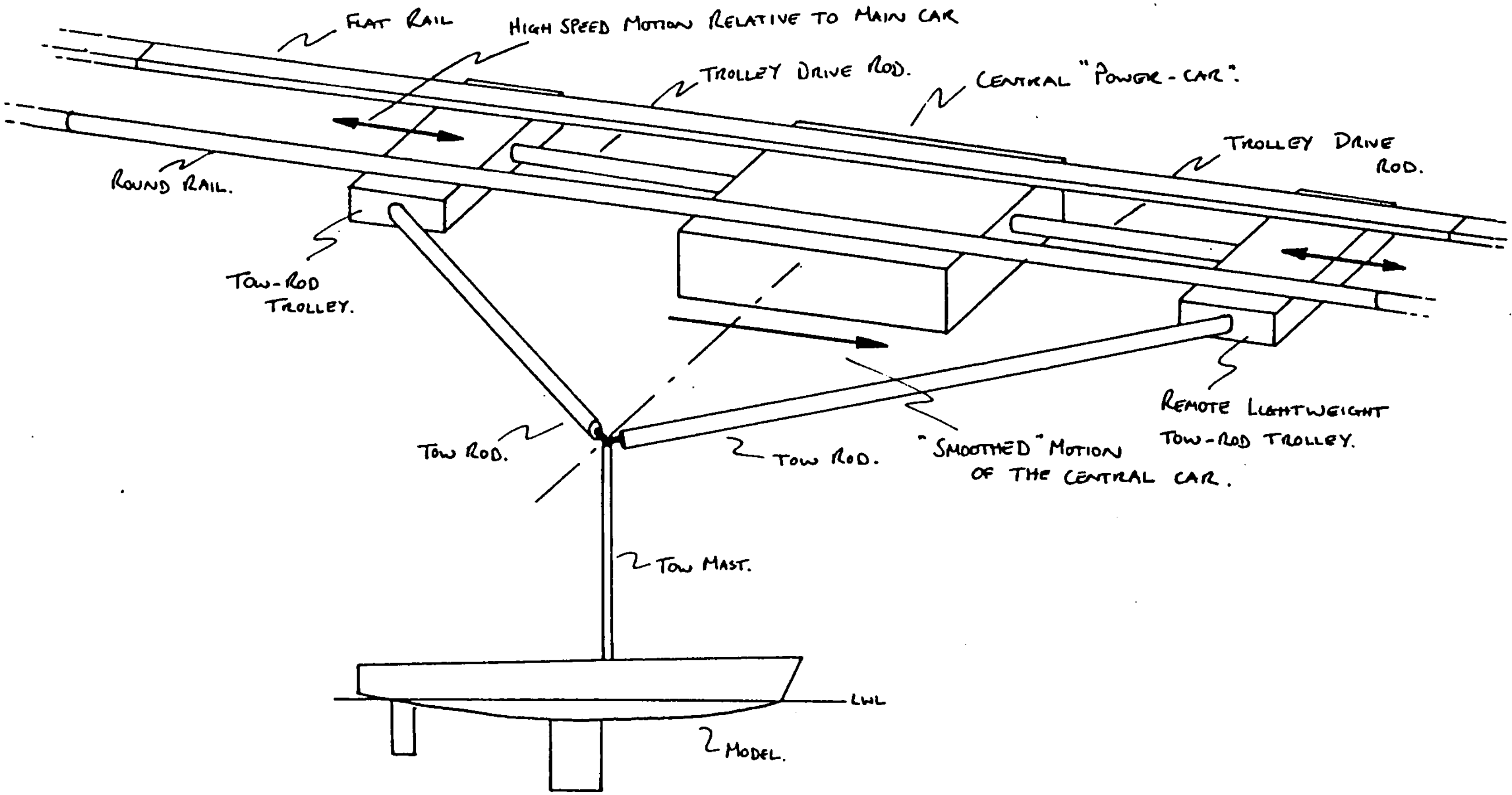


Figure 4.3 Schematic of Chosen System

Chapter 5. Detailed System Design.

5.0 Chapter Summary

This Chapter first discusses the way in which the complete system design evolved. Various aspects of the mechanical and control systems are then described in some detail.

5.1 Notes on Description of System

The system was designed from the outset to allow a staged development via a 1-D combined Constant-Force/Constant-Velocity system. This was viewed as important because it would allow a detailed comparison of the two testing regimes. As such the mechanical and control systems were designed with this aim as a prime objective. In line with this the following description of the system developed explains the features relevant to both configurations.

In the event only the 1-D system was implemented due to time spent developing the optical loadcell reported in Part II of this thesis.

5.2 Detailed Discussion of System Layout

5.2.1 Mechanical Considerations

The first task was to fix the system dimensions and drive requirements to allow serious design to begin.

To avoid problems with extreme angles sensible tow rod limits of 30 degrees and 60 degrees were set. This, coupled with the operating range dictated the tow rod lengths to be a convenient 1.5 metres long. The limits are shown in Figure 5.1. At the extremes of operation the minimum gap between the two lightweight trolleys would be 1 metre. To give a generous amount of clearance central car length was set to 0.6 metres.

Assuming a total drive mass of 1 kg the force/motion specification could be met with an actuator force of 30 N. The case chosen was deemed as being very unlikely so the requirement was de-rated to 25 N. This was still ample for the 1-D case.

The main car drive was harder to specify. The maximum tow force would be around 10 N but a high degree of overdrive would be required for acceleration and control purposes, due to the considerable weight. It was decided that the car should be able to generate an acceleration of approximately 1.5 m/s^2. An estimated car mass of 25 kg gave an required drive force of approximately 50 N.

Due to budget restrictions it was necessary to use motors that were already available to the project. Printed armature motors were again viewed as ideal for the task due to their low inertia and high torque/weight and size ratios. There were several different models spare at the wide tank.

In the end a pair of Printed Motor G12M4-T units were chosen for the pushrod drives. These had the added advantage of having an integral tachometer. A PM G12M4-H high-torque model was chosen for the main drive.

With the motors' torque and e.m.f. constants taken into account, all force and velocity specifications were met by carefully choosing appropriate drive pulley diameters.

Since the lightweight cars use the round rail as their position reference, it was deemed important that the push rods drove the cars as close to this as possible.

Minimum weight of the central car was seen to be desirable though not critical. More important was that the bulk of the weight of the main car was carried by the stiffer round rail.

Another important consideration with a 2-D rather than 3-D system is the towing sense. The track height would be set so that the tow rods were horizontal with the model upright. Figure 5.2 shows the correct sense of towing to keep the tow rods and tow mast as orthogonal as possible when heel is generated. Much consideration was given to making the natural development path be via an interim 1-D towing system. Pushrod lengths were chosen accordingly.

At this stage a careful study of transducer requirements was undertaken. The choice of sensors would have a heavy influence on the design of the individual components.

The following conclusions were drawn:

The applied load would initially only be measured at the end of the pushrods. This would not take the trolley mass and friction into account but would allow a much higher loop gain to be achieved. A subsequent, lower gain second loop could be closed around a transducer nearer the model at a later date.

Tow point position can be easily computed from the distance between the tow trolleys and system geometry. Accurate position information would also be required for model control between tests. Thus some form of accurate position measuring system was required for the pushrod and main drives. Rotary optical encoders of appropriate resolution on the squeeze pulleys were deemed to be the most satisfactory solution.

Velocity information was already available for the pushrod drives. A discrete tachometer would be used for the main car.

Attitude measurement of the model could be achieved using similar technology originally intended for automatic steering systems. Tow point position and the angle of the tow mast relative to the rods is sufficient information to calculate the position of the test model in all 6 degrees of freedom. Custom universal joints with servo-potentiometers for angle measurement were envisaged. These same units could easily be reconfigured to allow 6 degree of freedom information for the 1-D case.

5.2.2 Electronics & Control Considerations

The control scheme to be used would have certain influences on the design of the towing system. Although details were impossible finalise at this stage, certain philosophies and concepts had to be fixed.

The main consideration was whether to mount control systems on the car itself, or have most of the hardware shore based. Local control reduces the demand on the shore communication side. Only a few low bandwidth control and data transmission lines would be required. However, testing of the system would be very difficult, the hardware would be more exposed to damage, shock and vibration and the weight of the central car would rise significantly.

Thus it was decided to go for an entirely shore based control system. Only necessary conditioning and communication electronics would be mounted on the car. All data channels would be brought ashore on screened cables to a Eurocard rack. This rack would be directly interfaced to the control computer via a data bus. Shore based power amplifiers would drive the motors via screened power cables.

A small rack would also be designed into the central car to keep loose wires to a minimum and to afford maximum protection to the sensitive circuitry. A comprehensive, enclosed cable routing system would also be included to tidy up the car and reduce chances of wires snagging or fatiguing.

5.3 Chosen Layout

Within these guidelines a compact configuration was finalised.

A complete schematic of the layout is shown in Figure 5.3 for reference.

5.3.1 Chassis Layout

The most compact layout satisfying the various factors was the union of two vertical plates separated by the two pushrod motors, with two horizontal plates locating the main drive motor. They are joined along the edges of the horizontal pair to give an extremely rigid chassis. Various extra spacers as required complete the basic shell.

The two vertical plates rise up between the two track rails making efficient use of space and keeping the pushrods as close to the round rail as practical.

The chassis plates are made from 3/16" alloy plate keeping weight to an acceptable minimum, while retaining enough thickness to allow direct tapping of threads into the plates.

Attached to the bottom of the lower horizontal plate is a set of general purpose equipment rails to allow easy mounting of ancillary items such as cameras.

5.3.2 Electronics Bay

Sufficient space was available between the two vertical plates to for a small electronics rack holding up to six 230x160mm Eurocards. These are plugged into a pair of 96-way backplanes to allow easy collection and distribution of channels. This layout give a total of 192 connections which are allocated between shore, car and test model communication lines. Such a large number of channels was required because many of the signals are differential pairs.

The rack is made from standard 19" components mounted between the vertical plates.

Enough space was left for power supply regulation and decoupling and a general purpose patch panel for channel commitment if required.

Overall cover plates are included to shield and protect the sensitive conditioning electronics.

A cable routing system was designed in at the lower union of the horizontal and vertical plates. Plenty of entry and exit slots were included, each with tyrap points to allow secure location of all signal wiring. Sliding covers allow easy access.

5.3.3 Drive Systems

It was decided that the best way of providing drive, closing force and location was using a pair of V-pulleys for all drives. This also has the added benefits of reducing the magnitude of the closing force that has to be applied and the resulting Hertzian contact stress. All drive pulleys are mounted using split collets to reduce eccentricity to a minimum.

The main drive train is shown in detail in Figure 5.4. The pushrod drives are functionally similar. Both and are discussed below.

Main Drive

The round rail was thus chosen for the main drive because of its superior dimensional tolerances and stiffness. A more accurate assessment of the car mass reduced the drive requirement to 40N. Assuming a friction coefficient of 0.15 the required closing force is approximately 250 N. This means that steel drive and pinch rollers would be required to withstand the contact stress. To reduce the inertia of the drive pulley it was decided that a hardened steel tyre over an alloy boss was the best solution. The squeeze pulleys are solid steel on account of their small size.

To prevent the motor bearings from being subjected to undue load, and to reduce backlash, drive pulley support rollers were included as shown. The rollers are mounted on eccentrics to allow easy adjustment. As an extra precaution, the squeeze pulley is mounted on an axially fixed bearing to isolate vertical loads from the motor.

To minimise friction and stiction while preventing any slipping of the drive the squeeze force needs to be accurately adjustable. The same mechanism used in the slewing-drive pinch rollers was seen to be the best solution.

As already stated, it is very important that the position measurement be immune to scrabble and slipping of the drive if errors and control problems are to be avoided. Therefore the output shaft for the optical encoder used for position measurement is also incorporated in this assembly.

The main drive tachogenerator is mounted onto the rear of the motor under the lower horizontal chassis plate.

The various components were arranged to prevent confliction with the pushrods as shown.

Pushrod Drives

Before this drive train could be finalised the pushrod material had to be chosen. The pushrods had to be as stiff as possible to raise the first resonant mode while being light to reduce drive inertia. Straightness was also of prime importance as was a smooth accurate outer dimension.

The best solution found was centreless-ground unidirectional epoxy/carbon-fibre tube. Pulltrex had some 'cheap' end-of-line 25mm O.D. tubing of the required specifications. Unfortunately it was only available in 0.75 m lengths but joining was not seen as a problem.

The main drawback of such a scheme was the low yield stress of the epoxy resin matrix. However, the contact stresses worked out to be acceptable and the advantages outweighed this problem. It was anticipated that the pushrods would be rotated periodically to get maximum use as wear accumulated. In service this was not in fact a problem. The epoxy soon wore to expose the much harder carbon fibres which have lasted well.

A minimum clearance of 5 mm was set between adjacent pushrods and the car components to cope with the inevitable small misalignments.

With the pushrod material fixed a drive train very similar to the main drive was designed. The important differences are as follows: The lower contact stresses allow all alloy components to be used, reducing inertia to a minimum. The squeeze pulley can be mounted on to free bearings as large off axis loads are not present. A maximum axial motion of +-0.25mm was chosen in line with the position encoder limits. This makes the system self aligning.

The inner and outer pushrod drives differ only in the overhang of the pulleys.

5.3.4 Track Rollers

Enough of the system had now been designed to allow an accurate assessment of central car mass and centre of gravity position to be made. This information was required before the track rollers for the car could be designed. Several other factors had to be considered.

It was desirable to incorporate some form of suspension springing into the roller mounts to allow track misalignments and dirt to be tolerated. The springing of the main round rail rollers would preferably by fairly low rate to prevent significant variations of the main drive squeeze force as a result of such perturbations.

The main rollers also have to provide directional location and absorb any torques applied to the car by the pushrod drives. Careful design was also required to keep the track-pushrod separation to a minimum.

Main Rollers

The configuration that best satisfied the requirements is shown in Figure 5.5. The set of ballraces is flexibly mounted onto the car using Beryllium-Copper spring rods. These form good low-rate springs with the same stiffness in all directions. The pair of springs are arranged in the same plane as the track to prevent side loads twisting the bearing block, leading to confliction between the track spacers and bearings.

A pair of rigid endstops were provided as shown to protect the springs from overload. The various blocks are cut away to provide the appropriate clearance for the pushrods.

The roller sets are mounted close to the main drive pulleys to reduce further undesired loading.

The pinch bearings were designed in as a precaution in case the weight on the car proved insufficient to generate the required side-loads. They turned out to be un-necessary.

Flat-rail Rollers

The flat-rail rollers are much simpler. A single ball-race is mounted on a cantilever spring. The rollers are as far apart as practical to increase the pitch stability of the car. The third pinch roller turned out to be un-necessary.

5.3.5 Light-Weight Car Design

The lightweight tow rod cars were initially of similar design to the configuration used for the previous system. Minor differences were incorporated to allow for the different drive and tow rod arrangements. After initial testing it was found that the existing design had insufficient stiffness and was severely affecting the stability of the force feedback loop. A new front car was made using a thin-walled tube space frame dramatically increasing stiffness.

5.3.6 Universal Joints

As mentioned in the general discussion of mechanical requirements an instrumented universal joint was required for motion data. Although these units were never made their design is included as they would be made as the first part of a continuing development programme.

The joint had to have the following properties:

- Angular limits of +-60 degrees on both axes.

- The unit must give accurate readout of angle in the two axes of rotation.

- The joint must have a low friction torque to avoid undemanded forces being applied to the model.

- Zero backlash is vitally important, any play would severely affect the force control loops.

- Finally it must be capable of carrying a direct load of 30 N and torques of up to 3 Nm.

This is a very demanding specification to meet. After consideration it was decided that a ball raced unit was required. The design shown in Figure 5.6 is easy to make and meets the performance criteria.

The opposing ball races are pre-loaded with sprung washers on assembly. The unit is made more compact by using the shaft of the Spectrol servo-potentiometer as one of the bearing axles. The ball races themselves are rated to meet the load specification. They would however be working near their limit and so would be changed frequently.

5.3.7 Yaw Control Rig

If the full 2-D configuration had been realised some form of yaw control between and during tests would be required. Although never made the following system was designed:

The chosen layout for the yaw control rig is shown in Figure 5.7. The geared Portescap motor similar to the roll damper unit would use both torque and position control loops. Between runs and during acceleration/deceleration phases the unit would maintain the demand yaw angle. During testing the unit would apply a demand yaw torque. This may be zero, a constant, a damping term or an active steering demand.

5.4 Rig Control Overview

5.4.1 Introduction

The rig control system was also designed with both 1-D and 2-D configurations in mind, the aim being to make the system as general purpose as possible. All the hardware built would be common to both. Only the control program is specific to the particular configuration used.

5.4.2 Technologies Considered

As discussed earlier it was envisaged that the rig would be under the control of a supervisory computer. However, the degree to which this computer is involved with controlling the test cycle had to be chosen. To this end a full study of the various control options was undertaken.

Full Hardware control

Worries concerning the availability of a suitable real time control computer at the Tank lead to an investigation into whether the vast majority of cycle control could be accomplished by using dedicated electronic hardware.

It was decided that although this was achievable it would be very difficult to fully automate the test cycle and a great deal of complex circuitry would be required. System flexibility would be severely compromised.

Full Software

State of the art DSP computers would certainly be capable of full software control of the towing system. All feedback loops would be closed in software with the computer having control of all aspects of the test cycle. This would allow ultimate flexibility and the required interfacing would be fairly straightforward.

Unfortunately there were insufficient funds available to buy a suitable DSP board so reluctantly this scheme had to be dropped in favour of a compromise solution.

Hybrid

The chosen solution was, inevitably, a hybrid compromise of the two schemes discussed above. A large degree of test flexibility can be retained using a much slower real time controller if all the high bandwidth tasks, such as feedback loops, are done in hardware. This leaves the computer to calculate the required demand signals at a much lower data rate. The increase in hardware complexity would mainly be the addition of some relatively simple analogue circuitry. The lower data sampling rate required would allow the use of cheap conventional sampling hardware such as PC labcards.

This method allows a much wider variety of computers to be used as discussed later.

5.4.3 Choice of Controller

Dedicated Embedded Controllers

There exists a vast array of single board embedded controllers. Although powerful these often make software development difficult because all code has to be written and compiled on a remote host before down loading. Unless an expensive emulation package is used debugging can be difficult. The severely restricted budget put most controllers of this type out of reach.

P.C.

The PC equipped with a Labcard is a powerful laboratory tool. The wavetank already possessed such a set-up. The only concerns were that the PC is not designed for real-time control and as such may not be able to meet the speed requirements.

Sun Based DSP

The most desirable solution would have been a Sun based DSP controller. This would have afforded the ultimate speed and power. However, suitable DSP's were far too expensive. As mentioned later the towing system interface electronics were designed to be as non-host specific as possible to allow rapid interfacing to such a controller if it had become available.

5.4.4 Chosen System

In the end the natural choice was a PC/Labcard based system as this was already available at the Tank. The Tank's Sun network has overall control of the system, and synchronisation with the wave generation was achieved using the synthesiser generated sampling clock as discussed later.

This was viewed as a compromise and there were concerns over whether the still significant computational requirements could be met. With this in mind the electronics were designed to be as non-specific as possible to allow subsequent use of a more powerful controller if it became available.

5.4.5 Description of Chosen System

The equipment available at the tank necessitated a somewhat "Heath-Robinson" overall control system. A schematic representation of the complete system is shown in Figure 5.8.

The Sun system was again in overall control of the test cycle. However, with the towing system now being controlled by the PC instead of the BBC some form of synchronisation link was required.

After long discussions with Peter Woodhead it was decided that the best solution was to use the sampling clock on the synthesiser rack to drive the PC control program. This was seen as being far from ideal because the Sun had no indication of the status of the test cycle. The sampling clock had to be allowed to run for an extra "safety" period to ensure that the test cycle had been given ample time to finish. Also a large amount of failsafe code was incorporated in the PC program to abort failed test cycles and hold the system stationary until the next test began. This worked satisfactorily.

The PC was interfaced to the towing system via a PCL-812 Labcard as described in detail below. The full 812 specifications can be found in the manufacturers manual.

5.5 Interface & Control Electronics

The interfacing and control electronics are discussed in detail below.

5.5.1 Clocked Bus

The most convenient way of dealing with complex digital and analogue I/O with the Labcard was deemed to be the use of a purpose made clocked bus. Synchronous operation is always much more straightforward than asynchronous methods or relying on the software to prevent timing problems.

This approach allows multiplexing of channels to reduce the number of data lines required. Power supply schemes are simplified and fewer problems with grounding should be encountered. A 4-bit addressing scheme was used to allow reading and writing of multiple digital I/O channels. There was no analogue multiplexing as 16 inputs were sufficient and additional outputs were provided via latching digital-to-analogue converters (DAC's) written to from the digital outputs.

A convenient solution to the problem of communication between the transducers on the towing rig and the computer is described below.

All interconnection between the Labcard and the towing system is accomplished through an uncommitted backplane mounted in a double height eurocard subrack. A second backplane is connected to the central car communication cables. This allows the various cards to distribute, and operate on, digital and analogue signals as required. The separation of the two backplane functions allows easy modification of the front end interfacing, making transition to different controllers more straightforward.

Communication around the bus is synchronised using a 1.25MHz clock. The wavemaker synthesiser clock is synchronised to this before being passed to the PC to generate the real time program interrupts.

Individual interface cards were designed to handle analogue and digital channels separately. These would have served to isolate the rack and PC power supplies to prevent damage on power-up. Also included would have been a watchdog circuit that continuously monitors the status of both the computer and the electronics. If an anomaly were to be detected the towing system would be shut down into a heavily damped failsafe mode.

However, in the interests of rapid commissioning of the towing system these interface cards were not built. The absence of the failsafe circuits meant that the system could not be left unattended, also the lack of supply isolation meant that the +5V supply for the rack must come from the PC. Both of these are undesirable and the interface cards would be built as the first stage of a continuing development programme.

5.5.2 Notes on Grounding Scheme

The wide separation of the control rack and central car meant that significant differences in the local ground voltage may be encountered. This meant that careful consideration had to be given to the overall grounding and analogue 0V reference scheme if ground loops were to be avoided. After discussions with Peter Woodhead the following method was adopted.

The central car, track and power amplifiers were all single point grounded to the terminals of the main power supply. All electronic circuitry isolated from this ground and was made relative to one of three "0V" references.

To reduce the number of data lines required between the central car and the shore based rack single ended data transmission is desirable. This can be achieved by buffering the local 0V level of the central car rack and sending this ashore as a reference voltage. This signal was then re-buffered in the main control rack and used as the second input to a differential input stage for each data signal.

All subsequent analogue circuitry was relative to a local rack Analogue 0V. This was set by buffering the Labcard's own Analogue reference.

Finally, as mentioned above, the digital electronics in the shore rack were all powered by the PC +5V supply and digital ground.

5.5.3 Central Car

The central car position control loop is closed in software on the PC. This is possible because of the car's long time constant.

Position is measured using an optical-incremental encoder mounted on the main drive squeeze pulley. It is important that the encoder is not mounted on the motor shaft because any slip under load would have led to unacceptable drift. In operation drift was very low. Over a typical three hour testing block the position drift was less than 150mm. The encoder drives a 16-bit binary counter giving a resolution of better than 0.4mm.

Velocity information is provided by a tachogenerator mounted on the rear motor shaft extension. As explained later the tacho signal was only used for damping terms, all test data was computed from digital position information to reduce calibration errors.

The software loop uses this position and velocity information to apply stiffness and damping terms. These were chosen to give the best response.

The computed main motor drive command was sent directly to the power amplifier through one of a pair of 8-bit analogue outputs in the interface rack.

5.5.4 Pushrods

The high bandwidth of the pushrod drives necessitated a different approach. The two pushrod drives required a force feedback loop in addition to the position control mode. The computer had to be able to switch smoothly between the two modes. The pair of analogue control loops were set up with the feedback path and loop gain set using an array of 8-bit multiplying DAC's addressed on the interface bus. To change modes the overall loop gain was reduced to zero while the feedback paths and gains were selected. The master gain was then ramped up to enable the control loop. This scheme allowed smooth transition between modes.

Again, position information was supplied by optical incremental encoders this time driving 12-bit counters giving the pushrod drives a resolution of better than 0.2mm. The motors used included integral tachos for velocity information. The force signal used for both control and measurement was supplied by the final custom loadcell prototype. Development of this unit is described in full in Part II of this thesis.

5.5.5 Power Amplifierss

The original intention was to make a set of custom +- 10Amp power amplifiers mounted in the back of the control rack. However, to save time it was necessary to use three spare wavemaker drive amplifiers. These have a DC current limit of less than seven Amps. The result was that the system could not be run at the design specification force limits, reducing peak accelerations and necessitating a lower testing velocity.

5.6 1-D System Implementation

Figure 5.9 shows the 1-D configuration that was implemented. The overall layout is similar to the arrangement used by the previous system. A tow rod provided both the towing force and roll damping reference. This time however it was attached to the front L.W.C. of the new towing system. Due the absence of large off-axis forces in the 1-D case the tow-rod could be attached directly to the loadcell without leading to problems with crosstalk. This meant that trolley mass and friction were within the feedback loop dramatically reducing their parasitic effects. The elastic brake-line bridle was attached to the rear L.W.C.

Much thought was devoted to finding a simple solution to the problem of automatic steering. In the end a simple mechanical solution was found. The operation of this is shown in Figure 5.10. The system confers a surprising amount of dynamic stability because of the coupling between sway and yaw afforded by the simple linkage.

With the mechanical system set up in this way the control program was developed to allow both Constant-Force and Constant-Velocity testing modes.

5.7 Calibration

Once again careful consideration was given to calibration techniques to ensure accurate absolute data values. All calibrations were again carried out directly between the physical quantity and the sampled computer data.

All velocity data was to be computed from the position history of the front L.W.C. to reduce the effect of temperature drift on the results. The flat track rail was marked off at 100mm intervals over a range of 3 metres using a precision 3m steel rule. A set square was then used to position the central car at each of these marks while the computer sampled for 5 seconds. The pushrod encoders were calibrated in the same way while the central car was held fixed.

The three tachogenerators were calibrated in much the same way as the previous rig. A hand held digital tachometer was used to calibrate motor R.P.M. against the sampled drive-tacho output at a range of speeds. Accurate measurement of the rolling radius of the drive pulleys was then used to arrive at a linear velocity calibration. This was deemed to be of sufficient accuracy to be used for computation of damping factors.

The loadcell calibration values were viewed to be the most critical element of the transducer system and as such the calibration system was made as simple as possible to encourage frequent checks. The method adopted was to calibrate the loadcell in-situ using a removable "frictionless" pulley and a standard 10 Newton load. This was weighed on a precision Mettler balance. The entire procedure took only 2-3 minutes and was carried out a total of 27 times during the 5 week test programme. After each calibration the new gain and offset values were entered into the test program.

The loadcell was found to be remarkably stable with the standard deviation of the gain values representing less than 0.9% of the mean values. The standard deviation of zero point drift was found to be less than 0.4% of the full scale output (FSO). Combined linearity and hysteresis were estimated at less than 0.1% FSO. This was despite continuous use, temperature fluctuations estimated to be over a range of more than 10 C and several severe shock loads. Full loadcell performance details are discussed in Part II of this thesis.

5.8 Control Program

The towing system required two main programs to be written. The Sun end for control of the waves and sampling clock and the PC real-time control program. The Sun program required was similar to the code used to drive the previous towing system. The PC program was however completely new and took a considerable amount of time and effort to develop.

The control program is of necessity complex. The PC is not only having to make many real-time decisions but also has to handle all the sampling and bus control functions. A detailed flow chart was drawn up to define the sequence of operations required to run both constant-force and constant-velocity tests. This was refined and simplified through discussion with Peter Woodhead before being finalised. The control program itself was written exclusively by Peter Woodhead.

5.9 Description of test cycle

The complete control cycle for both Constant-Velocity (C.V.) and Constant-Force (C.F.) modes is shown schematically in Figure 5.11.

The PC program is driven entirely by the wave synthesiser sampling clock. The clock is connected to the Labcard external clock input which generates the program interrupts. The program, and thus the towing system, sits dormant until the clock signal appears. As soon as the clock appears the program goes into a "wake-up" routine during which the current position is sampled to allow the control loops to be turned on safely.

The three position control loops are then activated with the demands being the previously sampled initial positions. The three degrees of freedom then move to what was known as the "brace-up" position. This is where the model is held under load between the front tow rod and the elastic brake-line. This allows the model to be moved around under full control.

The model then moves from its arbitrary wake-up position to the start position for the next test. During this period the waves have been started by the synthesiser and there follows a wait period of sufficient length to ensure that the waves have reached the model before the test run begins.

Next the rear brake line is retracted and the acceleration phase begins. The program calculates and outputs the required stream of position demands to accelerate the model to the demand initial velocity by a defined start position and time. This ensures that the actual test phase always starts at the same time and place. This process is identical for both C.V. and C.F. runs. However, for the C.F. tests the required initial velocity is read from a batch file. The choice of this initial velocity is discussed in more detail the next Chapter.

After a short settling phase to allow the system to reach steady-state the test begins. For C.V. runs the program then simply outputs a stream of position demands to maintain a constant velocity until the "end-of-test" position is reached. The system then enters the braking phase. For C.F. runs the process is slightly more complex. The system switches smoothly into the controlled force mode and then maintains a constant force on the front tow rod. The central car trys to maintain a constant distance between itself and the front L.W.C. This mode is maintained until the end position is reached where-upon the program switches the system back into the controlled velocity mode before entering the braking phase. Much care is required when switching back into the position control loop. The central car is only capable of following small transients so the L.W.C. accommodates most of the velocity fluctuations. The initial position demand at the time of changeover must take account of this. Through trial and error the most reliable method was to keep a moving average of the system's position history over the preceding half-second. This gave an accurate position estimate while preventing noise spikes from leading to large offsets.

To allow the model to be decelerated rapidly under control the system then returns to the brace-up mode. The deceleration routine then calculates the stream of position demands required to bring the model to a halt at the end of the track.

The test data is then saved on the PC's hard disk before the model is returned to the start position where the system "relaxes" with all loops disabled. The sampling clock then stops and the program "sleeps" until the clock signal is detected again.

5.10 Conclusions

A towing system capable of realistic controlled force testing of yacht models was designed and built.

The full 2-degree-of-freedom capability was not realised. Instead a simpler 1-D configuration was implemented that allowed either C.F. or C.V. testing to be carried out.

The control systems were developed over many hundreds of runs to realise a reliable and repeatable testing system.

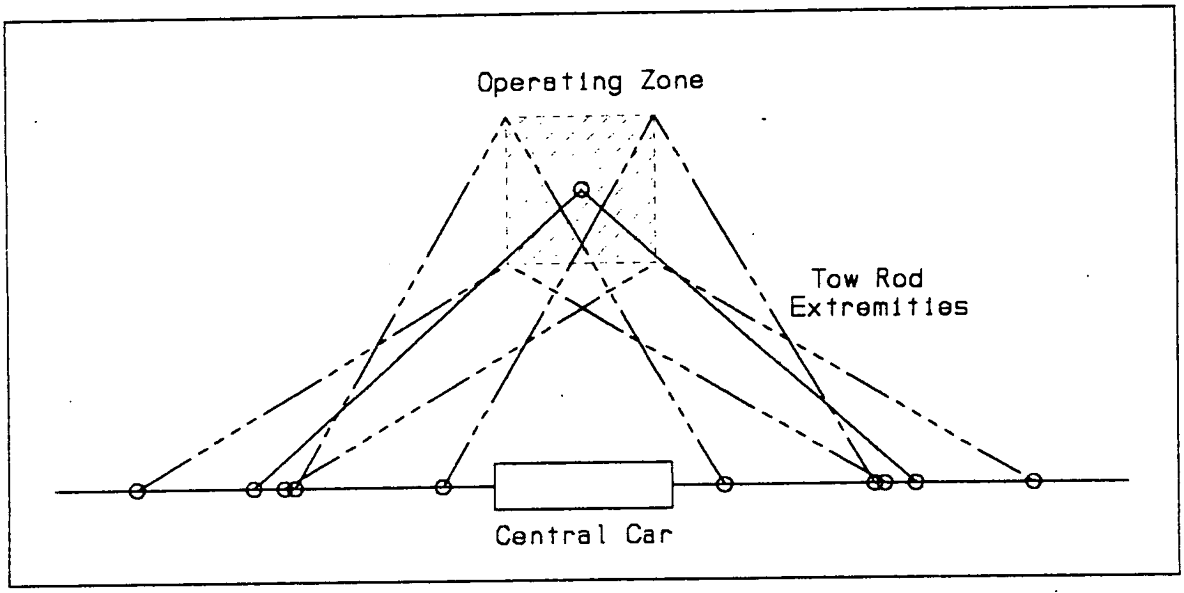


Figure 5.1 Tow Rod Limits

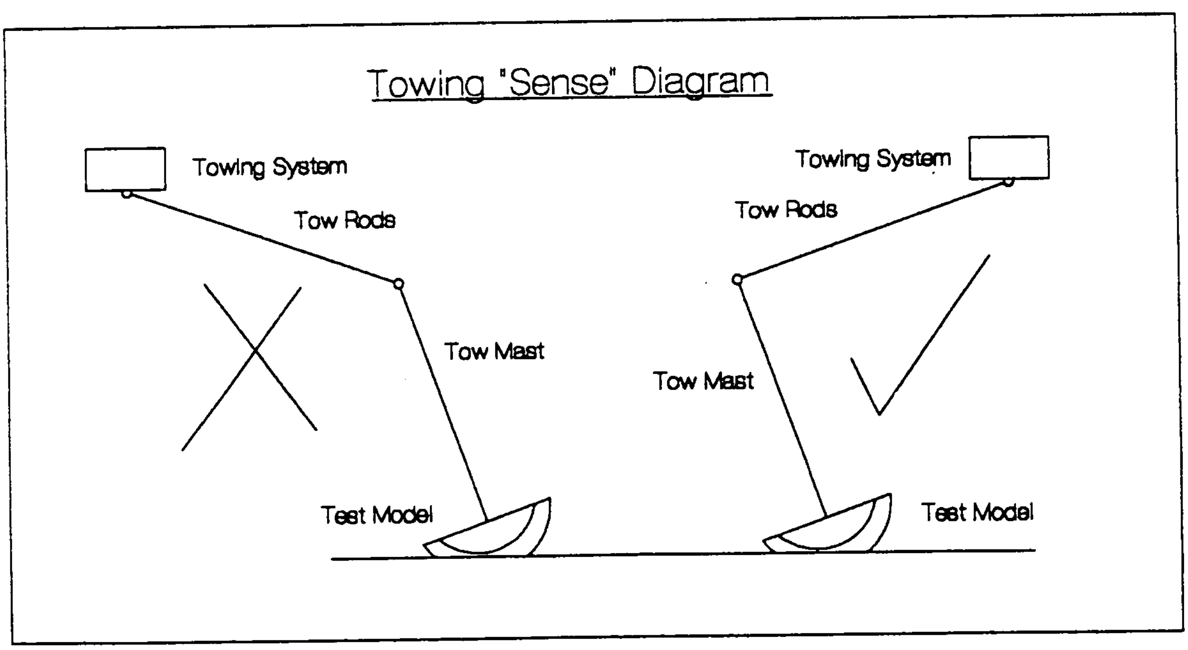
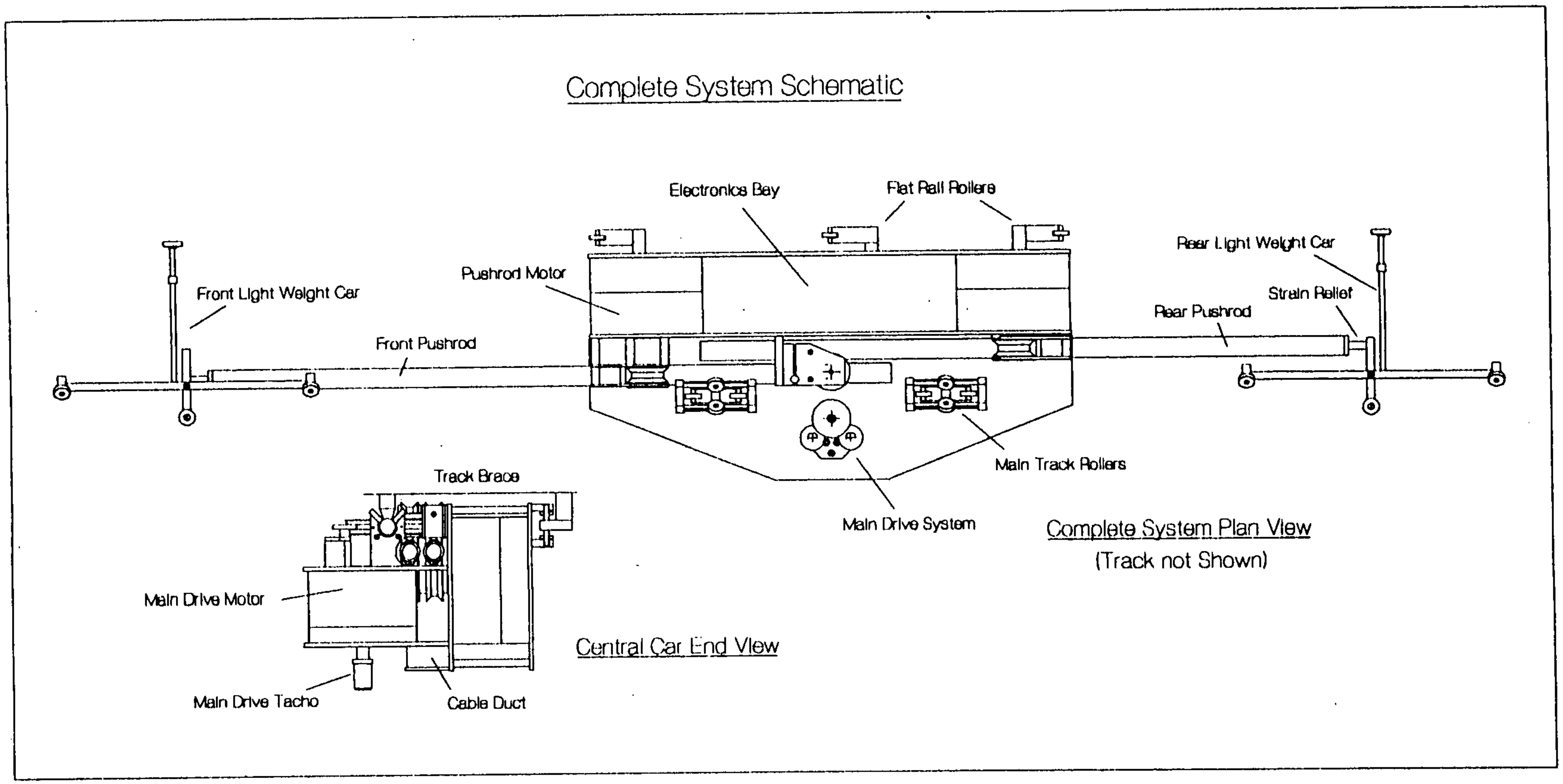


Figure 5.2 Towing "Sense" Diagram

Figure 5.3 Full Schematic of Chosen Layout



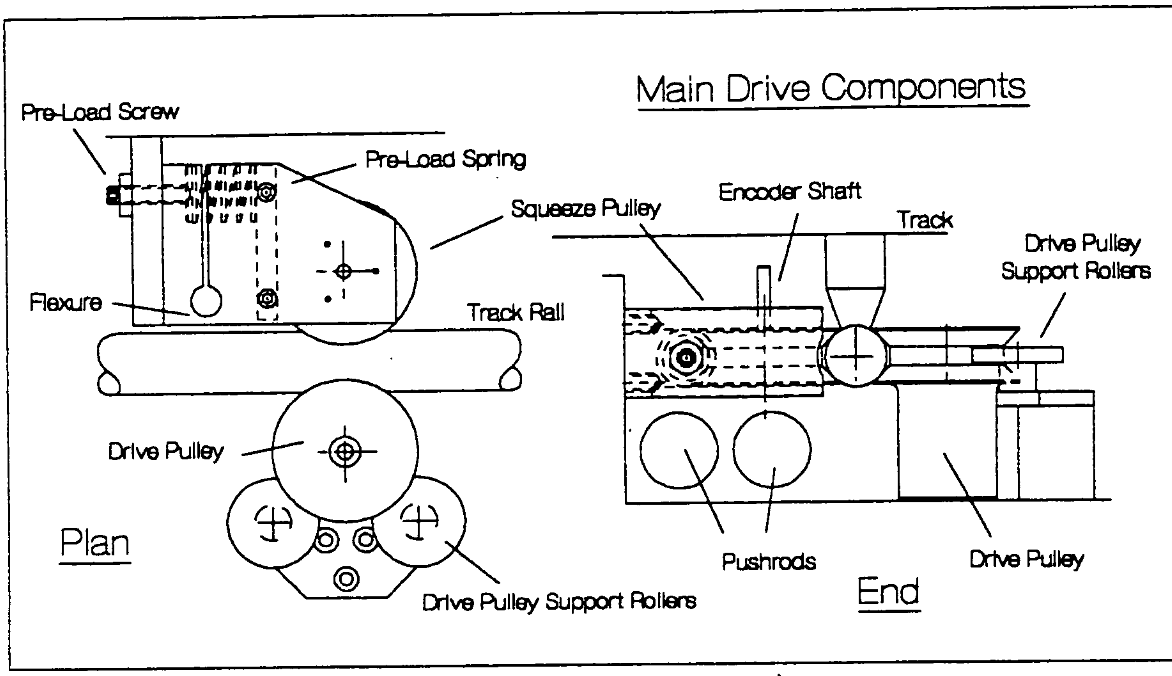


Figure 5.4 Main Drive System Detail

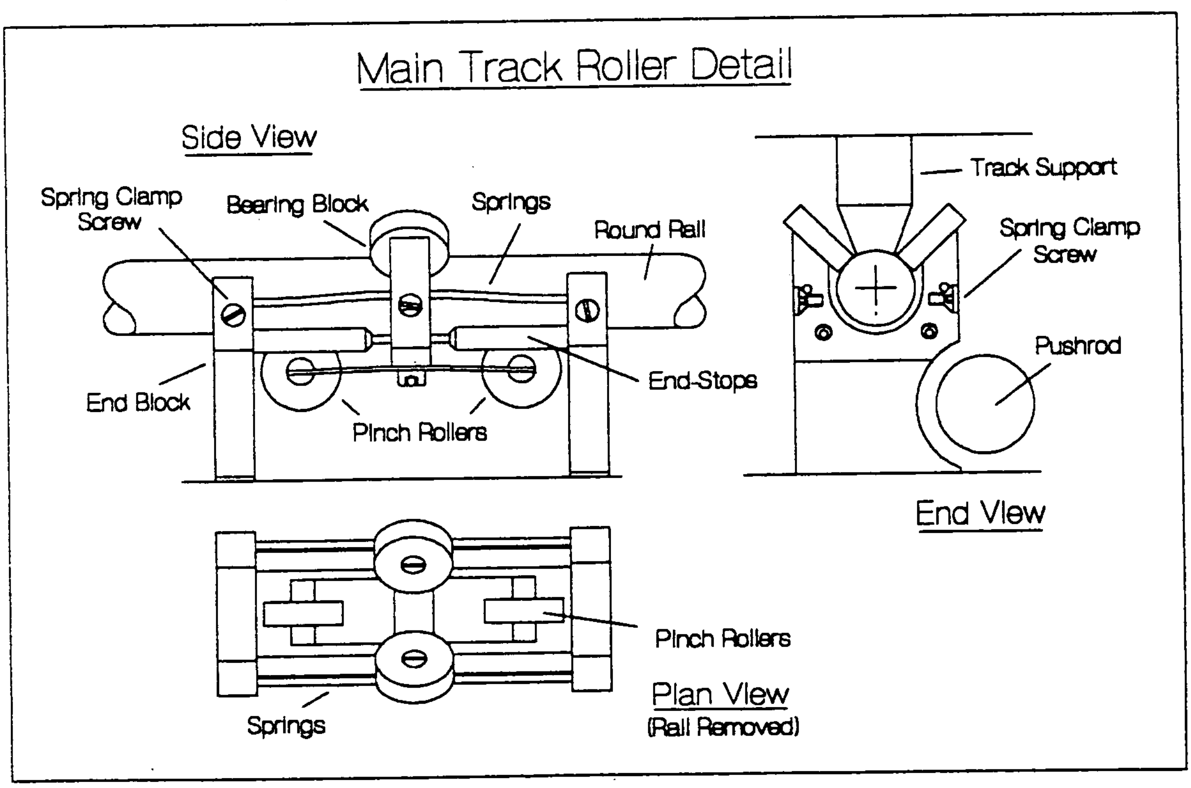


Figure 5.5 Main Track Roller Detail

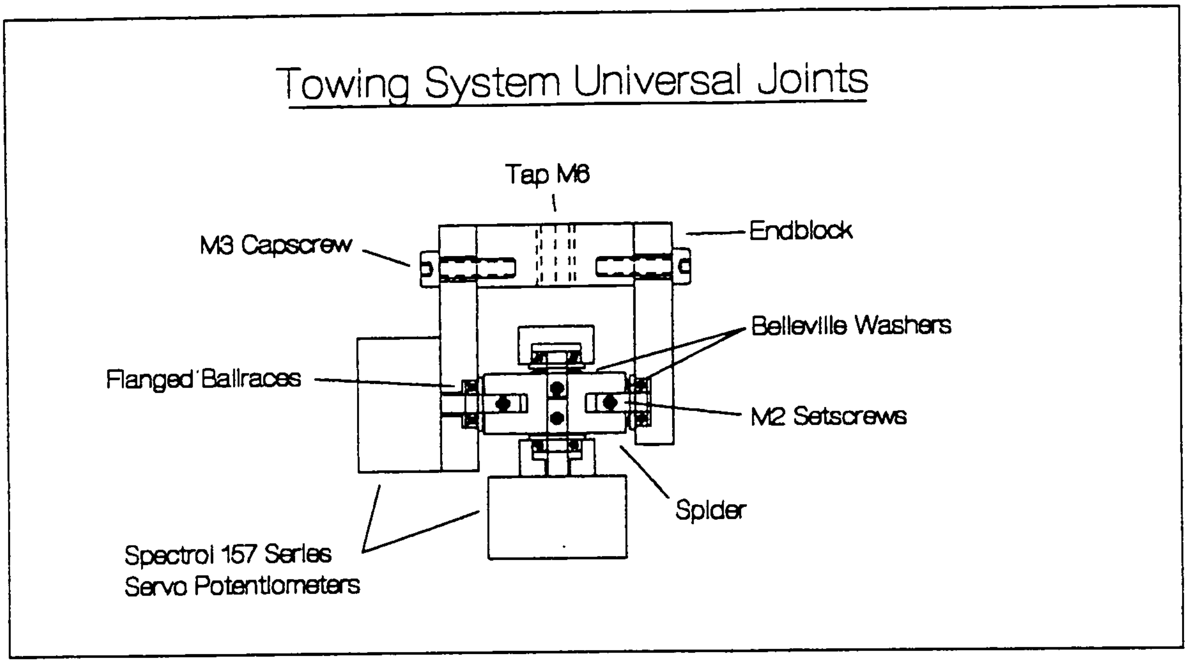


Figure 5.6 Instrumented Universal Joint Design

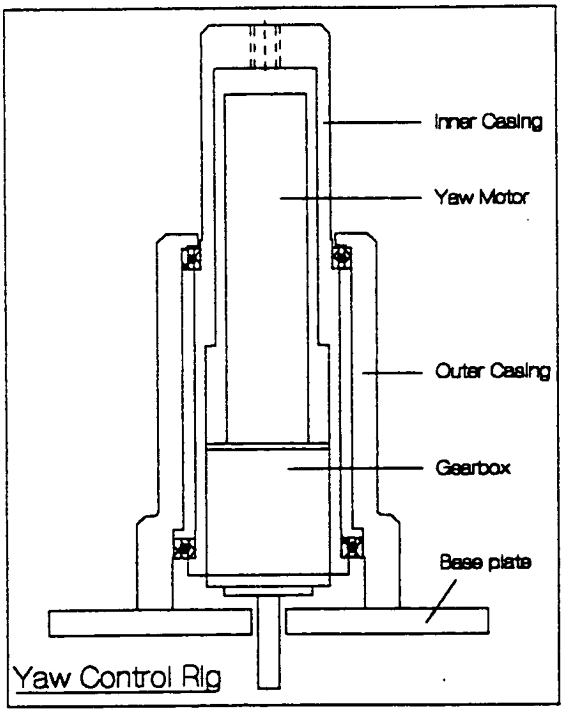


Figure 5.7 Yaw Control Rig

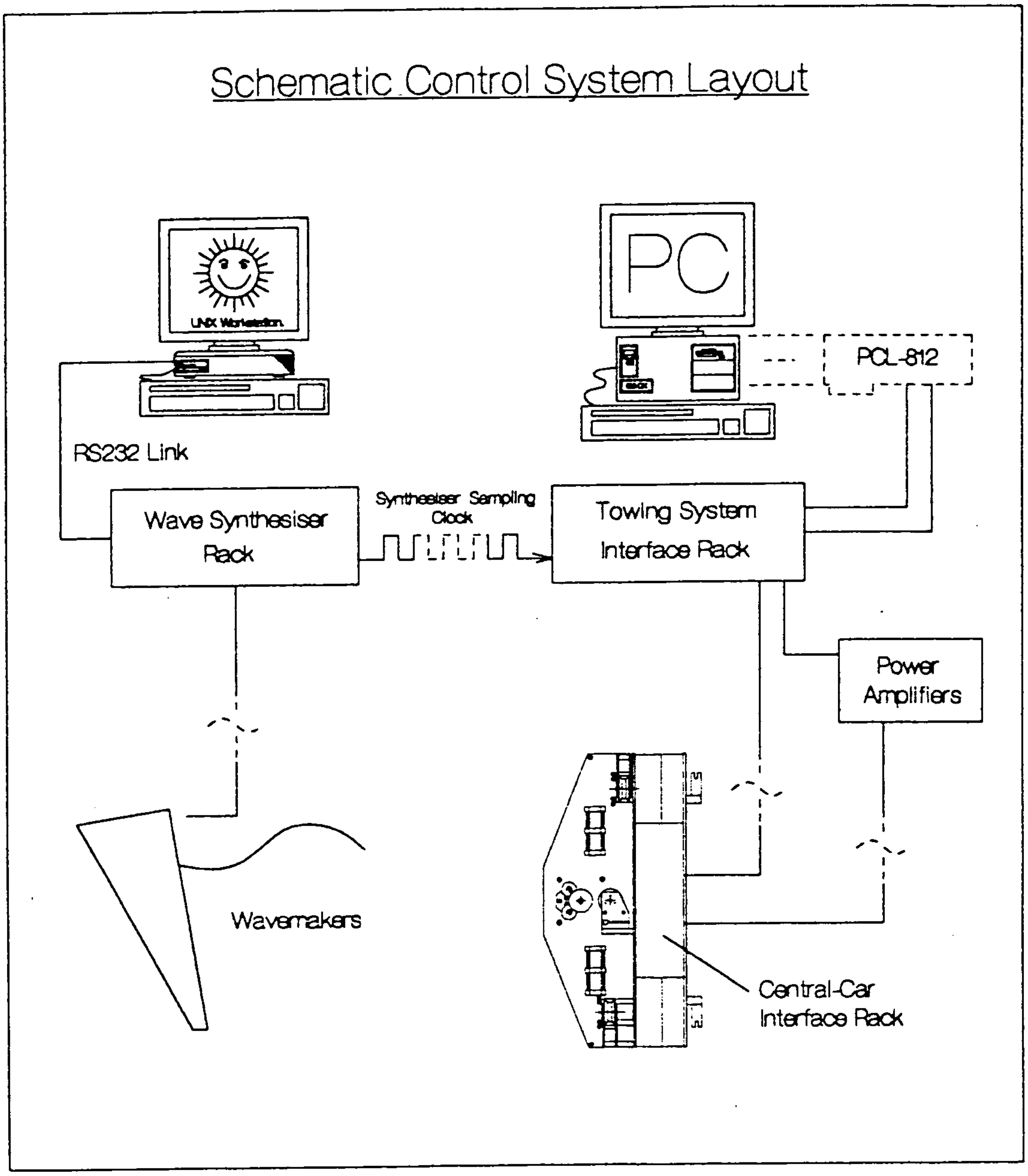


Figure 5.8 Control System Schematic

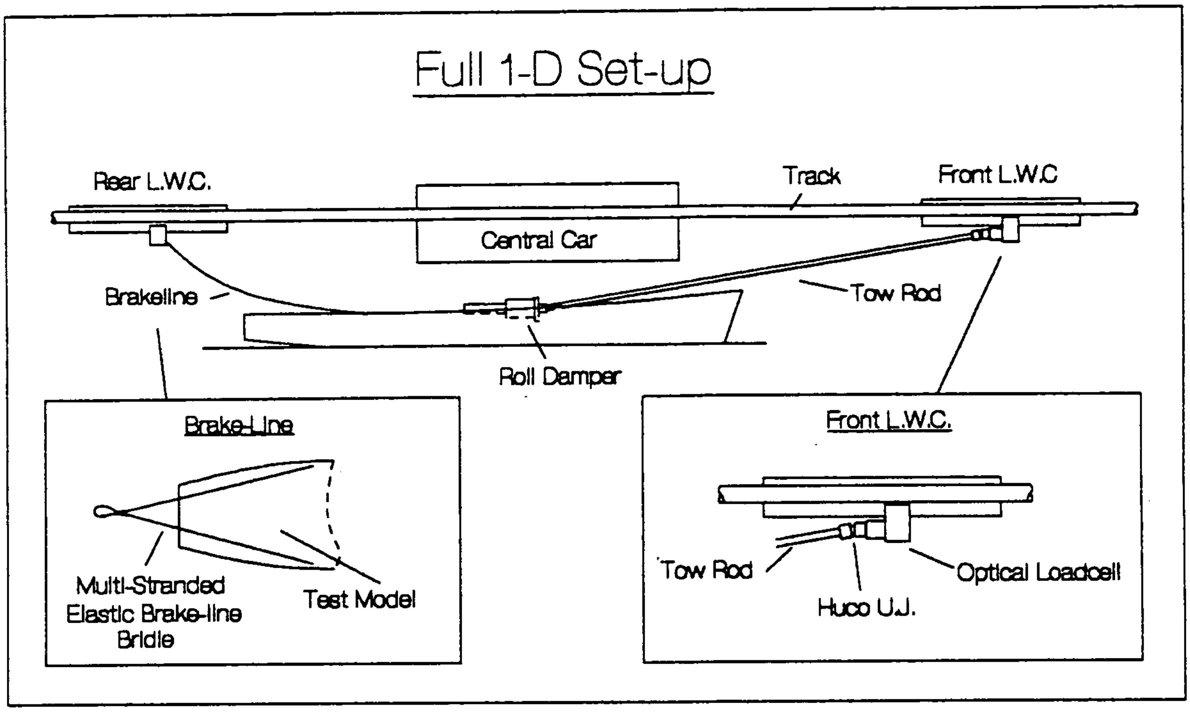


Figure 5.9 1-D System Schematic

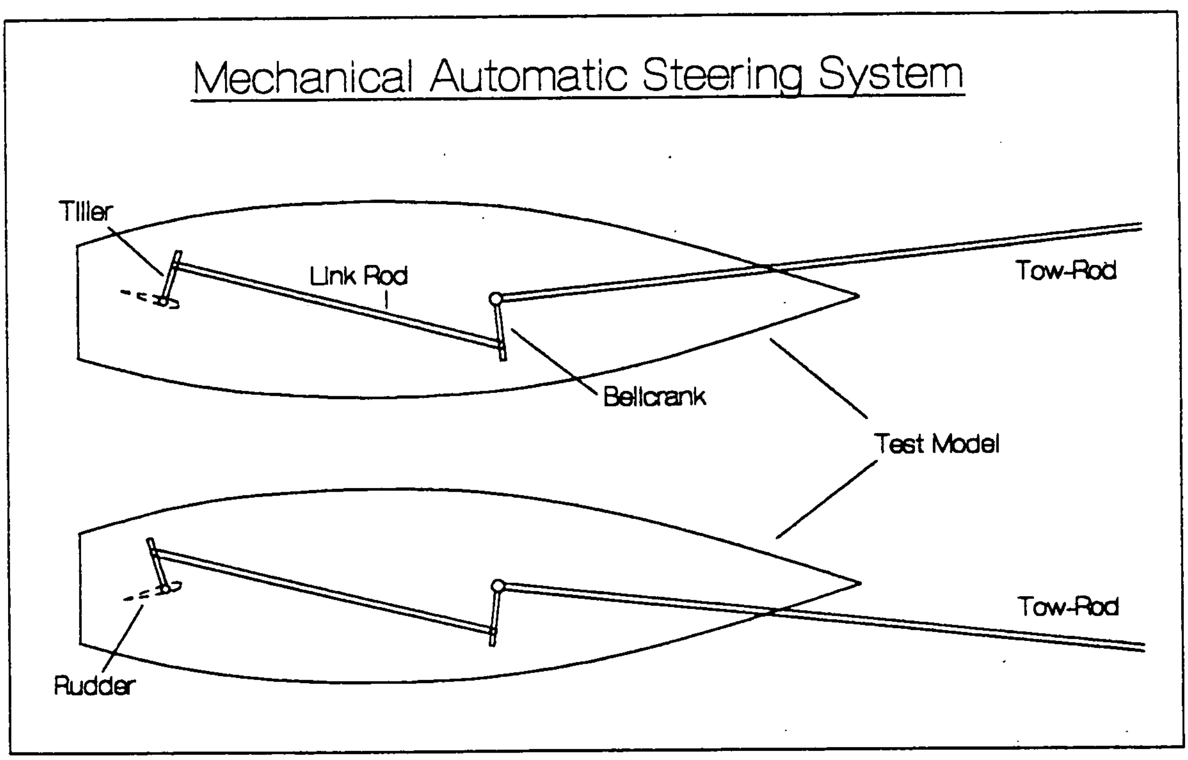


Figure 5.10 Mechanical Automatic Steering System

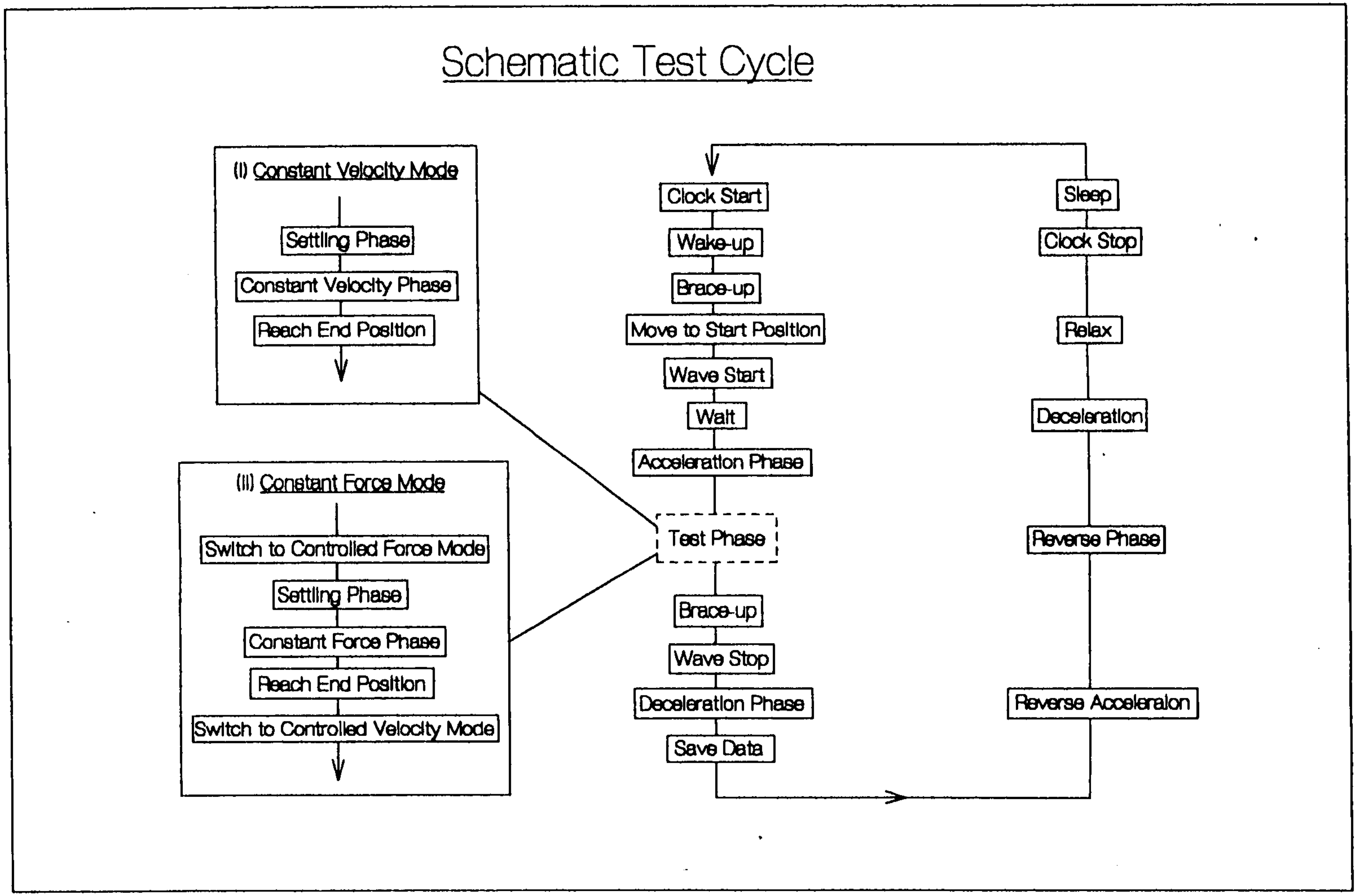


Figure 5.11 Dual Mode PC Control Program Flow Schematic

Chapter 6. Experimental Work

6.0 Chapter Summary

This Chapter begins with an explanation of the way the experimental results from the new system will be presented. Results from both Constant-Velocity and Constant-Force experiments are then reported along with explanations of the test methodology. A detailed comparison between the two modes is then described before conclusions are drawn.

6.1 Notes on the Presentation of Results

It is useful at this stage to make some general notes on the presentation of the results in this Chapter.

All experiments were carried out using one model, at a single radius of gyration. This was a 1/9th scale AC Class yacht, kindly on continued loan from the ex-Port Pendennis Americas Cup Challenge. The model was prepared and ballasted as before (see Appendix A).

All testing was done using a wave approach angle of 40 degrees off the bow to simulate the close-hauled condition. A single monochromatic wave frequency sweep was used in all cases. This sweep was through 25 values over the band between 0.6Hz and 1.33Hz. All frequency data plotted on the graphs is the absolute wave frequency rather than the more commonly used frequency of encounter. The author feels that this is more useful for prediction of full scale data from a measured wave climate. Three different wave amplitudes were used.

All tests were based on a single still water model velocity to allow easy comparison between tests. As explained earlier this could not be the same velocity as used for the previous tests on this model due to the choice of power-amplifiers.

The results are presented with the kind permission of the Port Pendennis Challenge. All results are presented as normalised values at model scale to avoid infringing a data confidentiality agreement. For the same reason no specific model details or photographs of testing can be included.

Many of the graphs use expanded axes for clarity and careful note should be taken of the relevant scales. All graphs are plotted as straight lines between data points, no smoothing or curve fitting was used.

Finally, to avoid verbosity the following abbreviations will normally be used:

- Constant-Force: C.F.

- Constant-Velocity: C.V.

- Still-Water-Control: S.W.C.

- The results from a C.F. test will be called "Constant-Force-Velocities": C.F.V.'s

- The results form a C.V. test will be called "Constant-Velocity-Forces": C.V.F.'s

6.2 Experimental Results

6.2.1 Constant-Velocity Tests

The test format used for C.V. tests is shown in Figure 6.1. The format is identical to that used for the previous towing system and is made up of the same mix of four wave runs at 90 phase intervals between two S.W.C.'s. The "absolute" value used to standardise all wave runs was derived from the mean of all S.W.C. tests carried out in this series.

Figure 6.2 shows the total model resistance for the three wave amplitudes. The curves show the typical resistance peak as the model passes through pitch resonance before recovery at higher frequencies. Figure 6.3 shows the same results plotted on an expanded Y-axis starting at the still water resistance. This can thus be taken to be the normalised added resistance coefficient.

Figure 6.4 shows the same results again but this time displayed along with two combinations of phase component. From this we can conclude that, for all but the most exacting experiments, two 180 degree phase shifted runs would be sufficient to stay within the system repeatability bounds implied by Figure 6.5. This shows three identical sweeps done near the beginning, middle and end of the 5 week testing period by way of a repeatability check.

This high degree of long term repeatability is largely due to the S.W.C. system. A particulary poor set of results is shown in Figure 6.6. As can be seen the introduction of the S.W.C. correction goes a long way to improving the situation.

The results from the three sweeps can be cross-checked by using the assumption made by linear theory that added resistance should be proportional to the square of wave amplitude [3,34]. Figures 6.7 and 6.8 show both upward and downward square-law predictions for the set of results shown in Figure 6.3. As can be seen the results agree well with the predictions.

6.2.2 Constant-Force Tests

Next a set of C.F. tests was carried for the same wave conditions. The test format for the C.F. runs was similar to the C.V. case apart from the addition of techniques for choosing the initial velocity. As explained earlier it is important to accelerate the model to an initial velocity close to the expected result from the test. This is to ensure that the model is not accelerating or decelerating significantly over the length of the run.

The previous system did not have a facility for accelerating to a known velocity. Very careful development of the acceleration routine was required to ensure that a reasonable initial velocity was attained. As explained in the previous Chapter the control program for the new system had been set up to take advantage of the multi-mode control system of the new towing rig. The acceleration phase of the C.F. run is carried out in the controlled velocity mode to allow precise selection of initial velocity.

This introduces the problem of choosing a suitable value. The method adopted is shown in Figure 6.9. The model is accelerated to the average velocity of the previous wave run. For the very first test the model was accelerated to the still water speed. This was a good approximation because the speed loss at low wave frequencies is small. Passing on the mean velocity of the previous frequencies last run to the first of the next was also seen as a good approximation due the small frequency interval between tests. Again "absolute" values were derived from the mean of all S.W.C.'s.

Figure 6.10 shows normalised average velocities for frequency sweeps at the three wave amplitudes. Figure 6.11 shows the same results on an expanded Y-axis. As expected the model shows maximum speed loss at a wave frequency corresponding to pitch resonance with a subsequent recovery at higher frequencies. As the wave amplitude is increased there is a perceivable shift of the absolute wave frequency that corresponds to pitch resonance. This is due to the speed loss effecting the frequency of encounter.

There is no simple relation between speed loss and the wave amplitude so no predictions similar to the square-law values presented for the C.V. case can be made. This is due to the fact that the still water resistance curve can be highly non-linear complicating the relationship. However a qualitative assessment reveals the expected approximate correlation between speed loss and wave amplitude squared.

Figure 6.12 shows the same results displayed with two combinations of phase component. Once again it can be seen that two phases would certainly be adequate for most experiments.

6.2.3 Comparison of the Two Modes

The third Phase of the experimental programme was the direct comparison of the two testing methods. Intuition would suggest that there is a difference between the two modes, in particular it is likely that the relative phase of hull motions and wave action may be different. Added resistance is known to be very sensitive to small variations of this phase angle [4]. The system as it stood had no facility for measuring hull motions so the only way of comparing the two modes was directly by the total resistance for the same conditions.

This could be achieved in two ways:

- C.F.V.'s could be used as test velocities for a C.V. series.

- C.V.F.'s could be used as demand forces for a C.F. sweep.

There was only sufficient testing time to conduct one of these trials. The additional uncertainty surrounding the choice of initial velocities for the second case led to a series using the former method.

The test format used is shown in Figure 6.13. A standard C.F. test was used to generate individual mean velocity values for each frequency and its component phases. These were then passed on as demand velocities for the corresponding frequency and phase of a C.V. sweep. The results from this series were averaged in the usual way to give mean resistance data for the C.V. case. If these values were not the same as the original demand force used for the C.F. sweep then a difference between the two methods would be implied.

Tests of this format were carried out for the three wave amplitudes. The results are shown in Figure 6.14. The graph shows a frequency dependent difference of up to +-2%. C.V. methods appear to over-estimate resistance at frequencies below pitch resonance, and under-estimate at frequencies above the natural pitching period. There appears to be some amplitude dependence but the trends are not clear enough to form any firm conclusions.

These results initially appeared very exciting. Two features point towards the difference being due to a motion-phase related effect. At low and high frequencies the two methods seem to converge. This would be expected because at low frequencies motions are small due to low wave steepness, while at high frequencies motions are again small as the system becomes inertia dominated. The second is that the crossover occurs at pitch resonance where the pitch motion undergoes phase reversal.

The apparent lack of amplitude dependence is surprising. One would expect the magnitude of the difference to be proportional to wave amplitude squared in common with other wave induced drag phenomena. This lack of a clear amplitude dependence raised suspicions that the differences could be due to other effects caused by the testing system.

The first potential cause investigated was the possibility that the model motions were being influenced by the system geometry. It was possible that the large force fluctuations present in the C.V. mode were coupling into the pitch response. A series of experiments was conducted using three different tow points. The results are shown in Figure 6.15. The position of the tow point seems to have little effect on the results. The high degree of repeatability of the system is again demonstrated by the three sweeps carried out with the second tow point.

It was concluded that this effect was not responsible for the observed differences.

The second potential cause looked at was possible errors in the original C.F. data due to incorrect initial velocities. To investigate this the C.F.V.'s for the two larger wave amplitudes were iterated as shown in Figure 6.16. The original and iterated C.F.V.'s are shown in Figure 6.17. As can be seen there are significant differences between the two data sets. These differences are of the expected form. The method used for choosing the initial velocity for each test introduces the following error. As the mean velocity decreases towards pitch resonance a small amount of over acceleration can be expected, the reverse being true after resonance. Both iterated data sets show this trend clearly although the magnitude of the differences is small.

These new C.F.V.'s were then fed into a C.V. sweep as before. The results along with the original data are shown in Figures 6.18 and Figure 6.19. There seems to have been a significant effect on the magnitude of the observed differences between the two modes. There is a reasonable degree of correlation between the differences observed in the C.F.V.'s (Figure 6.17) and the resulting C.V.F.'s (Figures 6.18 and 6.19). However, the overall trends remain present. Further iterations of the C.V.F's were not possible due to time constraints. It seems reasonable to assume that the magnitudes of the differences between iterations would progressively decrease. Thus it seems unlikely that the entire observed differences can be due to incorrect initial velocities.

A system for measuring the relative phase and amplitudes of vessel motion and wave action should be implemented. Further tests with such a system would ascertain whether variation of either amplitude or phase between the two modes was responsible for observed difference in resistance.

Further tests are required using the other mode of comparison, namely running C.F. experiments using C.V.F.'s as demand values.

Much further testing is required before firm conclusions can be drawn.

6.3 Conclusions

- A software configurable Constant-Force or Constant-Velocity towing system was developed.

- Over 6000 test runs were carried out using the system over a 5 week testing period.

- The loadcell developed as part of the work for this thesis was used for all testing. The transducer, described in Part II of this thesis, proved to be adequately stiff, very rugged and exhibited low drift characteristics..

- Techniques for improving the accuracy and repeatability of test results were developed and checked.

- The results generated were highly repeatable.

- Tests carried out in the Constant-Velocity mode demonstrate the dependence of added-resistance on wave amplitude squared.

- Tests carried out in the Constant-Force mode qualitatively show a similar dependence.

- A detailed comparison between the two modes was carried out.

- This comparative study showed a frequency-dependent difference between the two modes of up to +-2%.

- From the limited tests conducted the magnitude of the difference seems to show no clear amplitude dependence.

- Tests were carried out to confirm that system geometry was not responsible for this difference.

- Tests were carried out that showed that incorrect initial velocities in Constant-Force tests may be responsible for the observed discrepancy.

- A system for measuring the phase angle between wave action and vessel motions should be implemented in order to determine whether a difference in this phase is responsible for the results.

- This system should also measure vessel motions to determine whether differences in the amplitude of these motions are responsible for the observed effect.

- Tests should be carried out using the results from a Constant-Velocity sweep as demand forces for a Constant-Force series to determine whether similar differences are present for this case.

- Much further work is required to determine the true cause of this phenomenon.

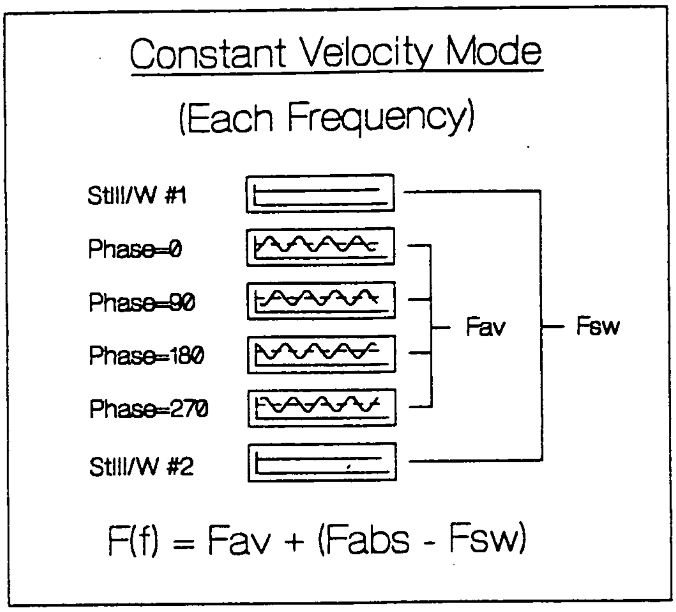


Figure 6.1 Constant-Velocity Test Format

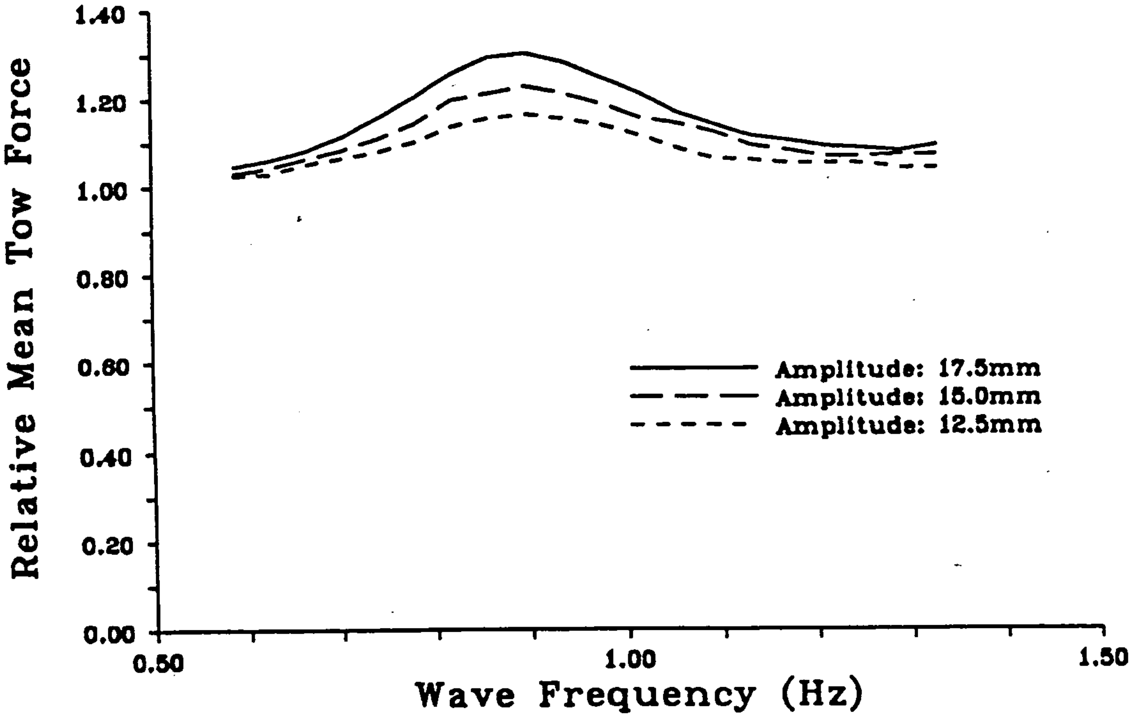


Figure 6.2 Normalised Resistance for 3 Amplitudes

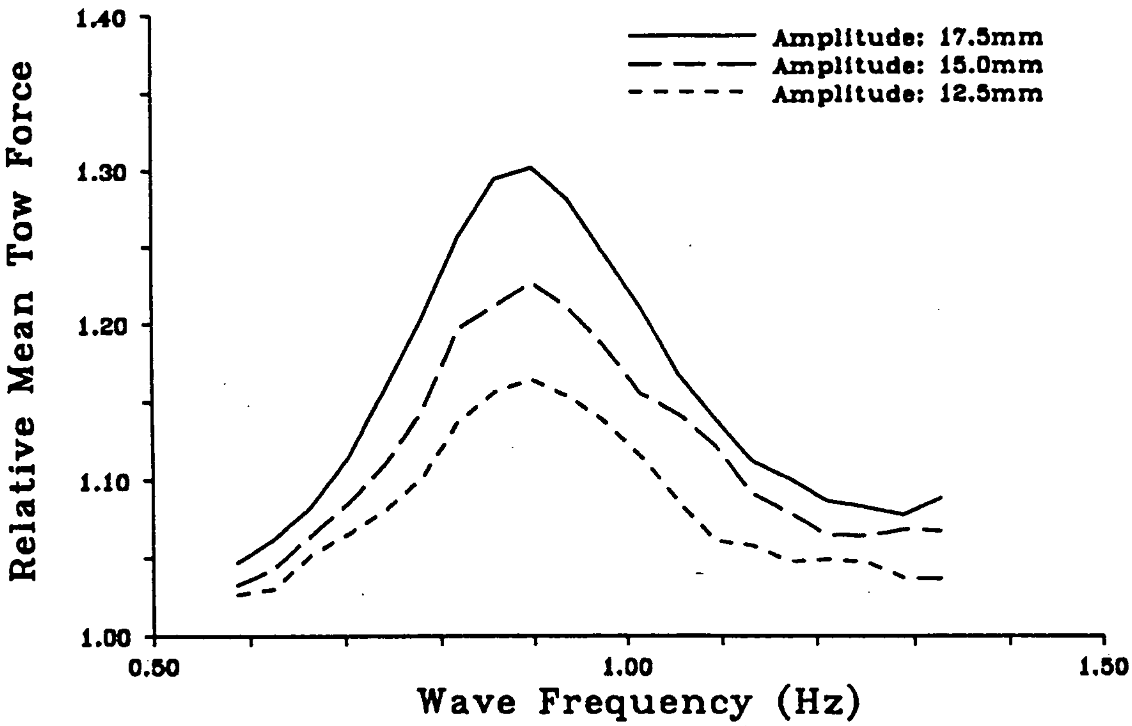


Figure 6.3 Normalised Resistance with Expanded Y-axis

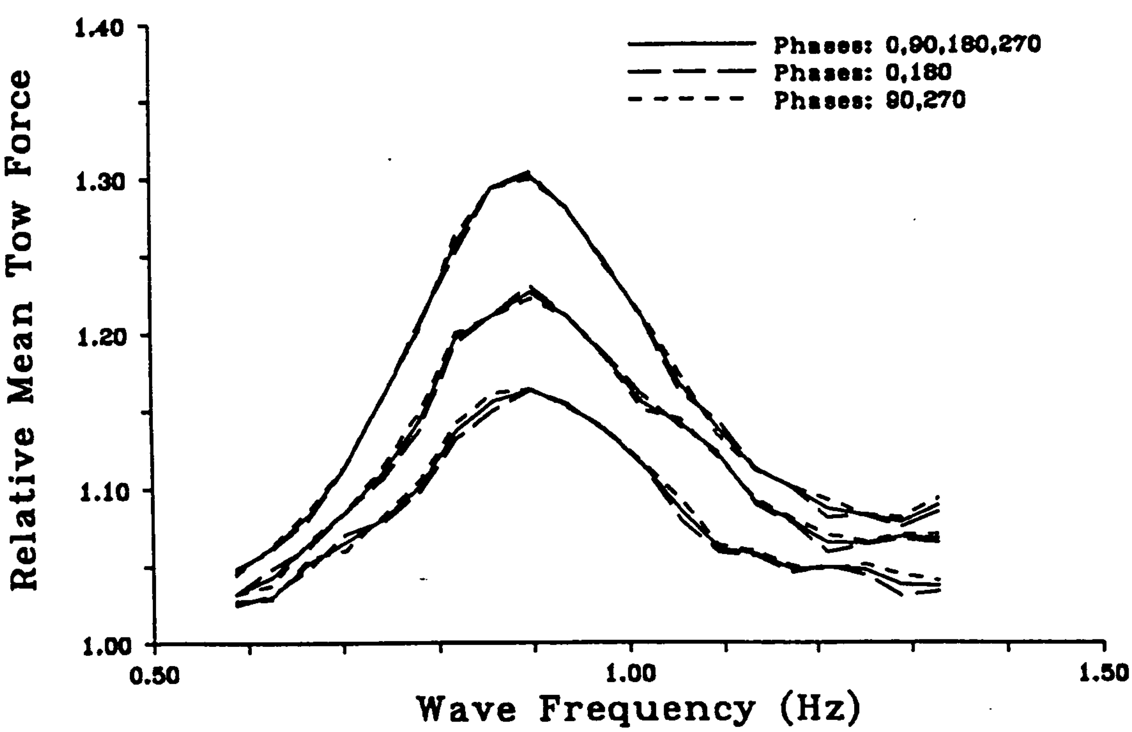


Figure 6.4 Different Phase Combinations

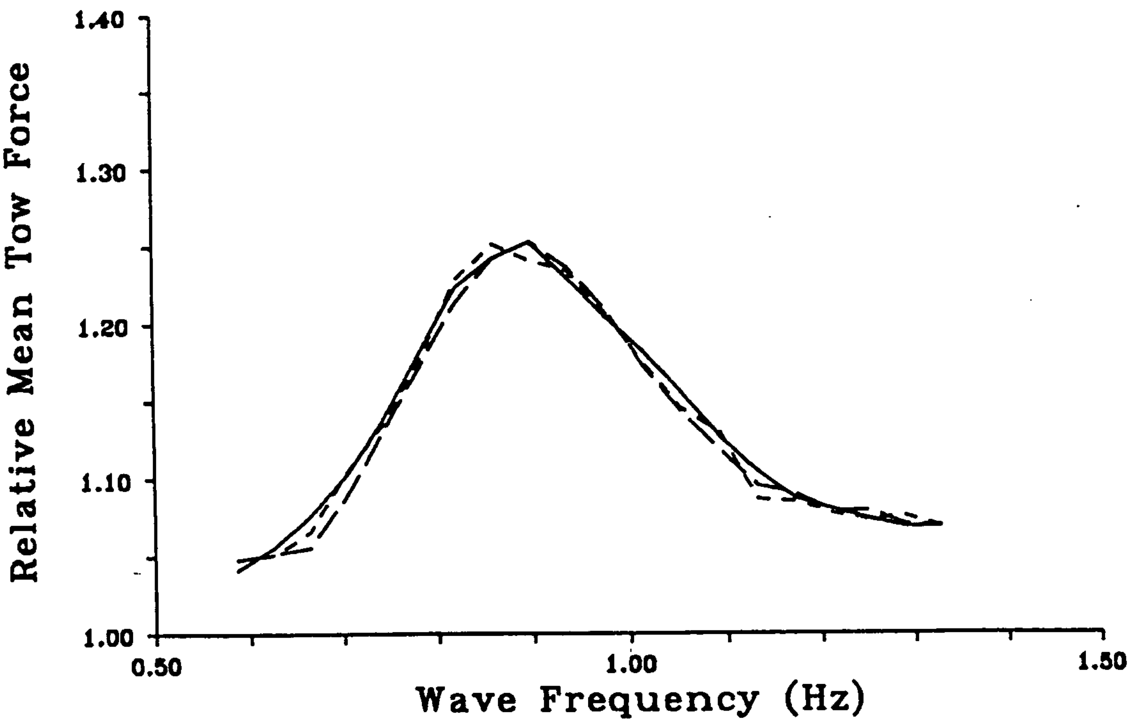


Figure 6.5 Repeatability Trials

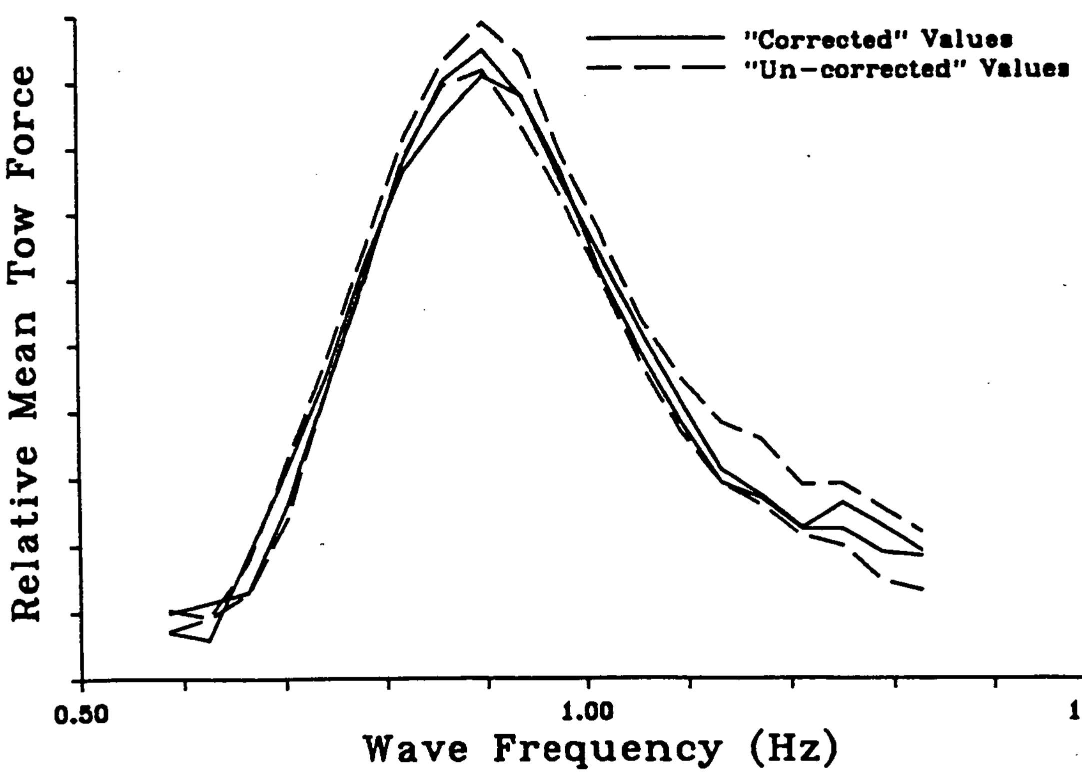


Figure 6.6 Effect of Still Water Controls

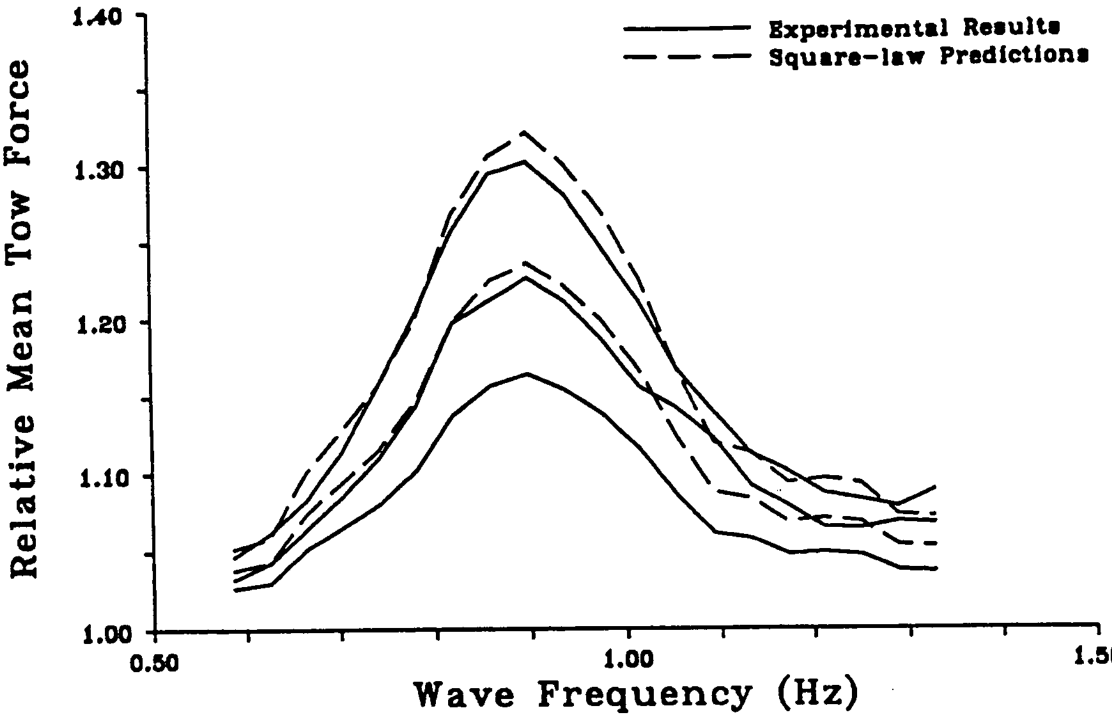


Figure 6.7 "Upward" Square-Law Predictions

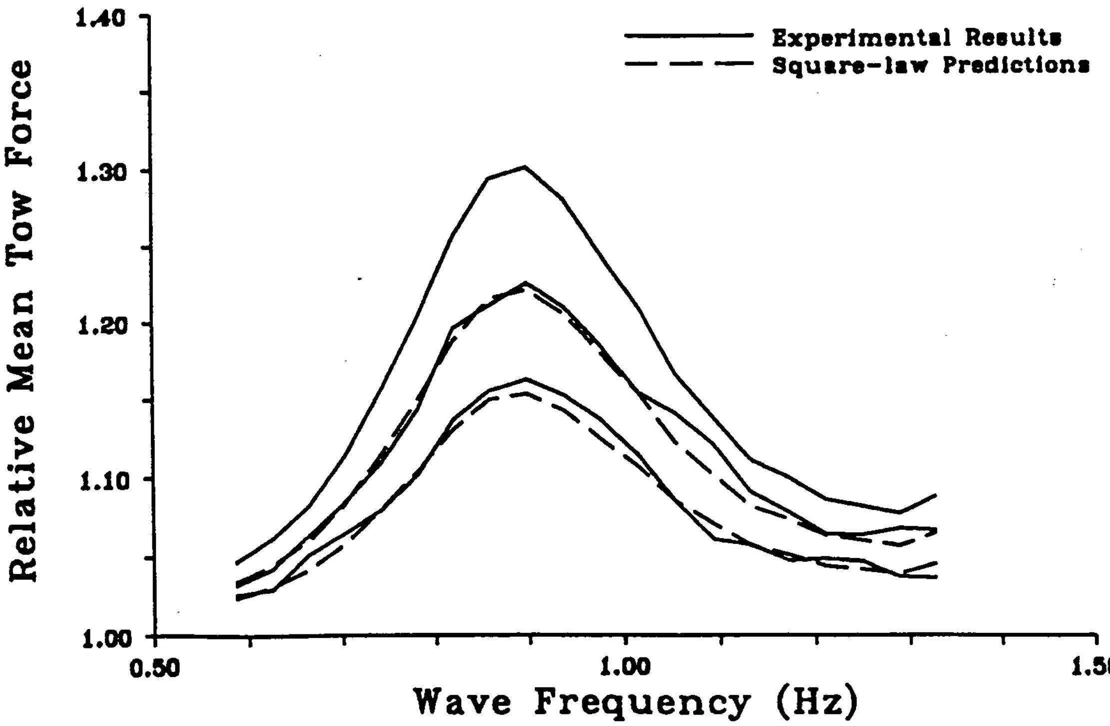


Figure 6.8 "Downward" Square-Law Predictions

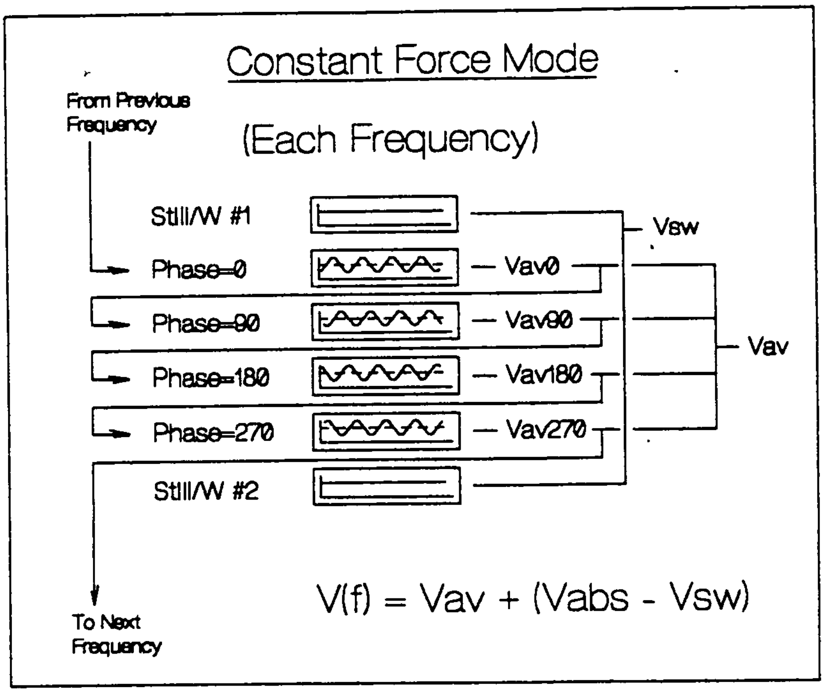


Figure 6.9 Constant-Force Test Format

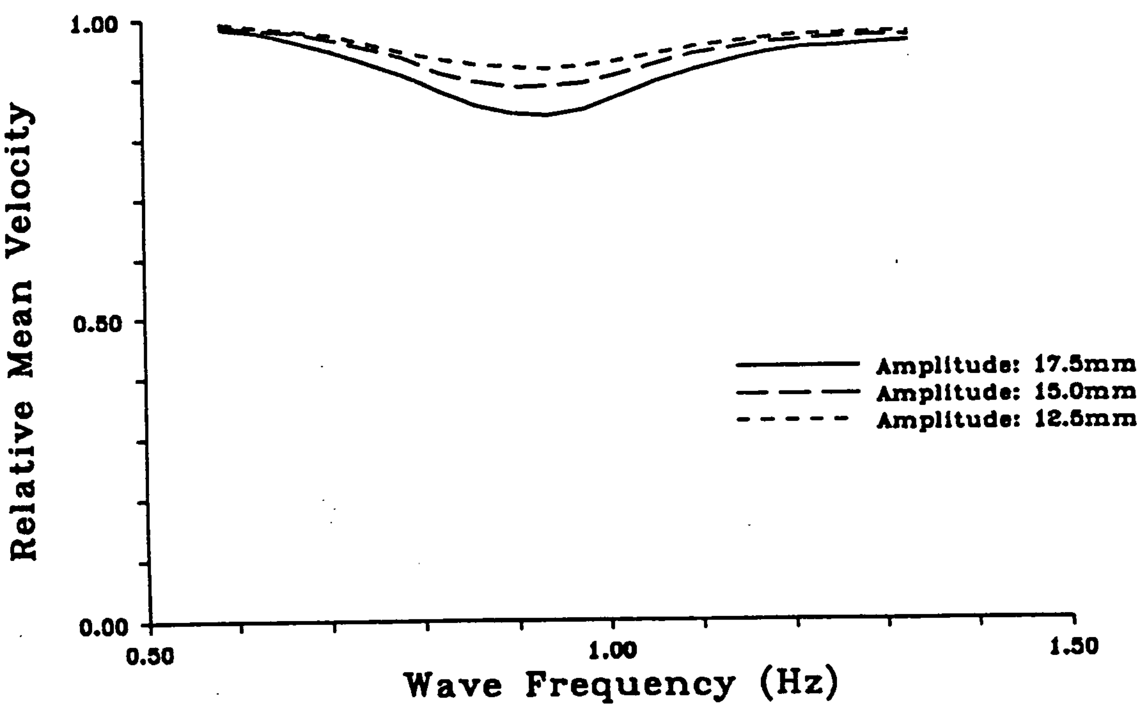


Figure 6.10 Normalised Velocities for 3 Amplitudes

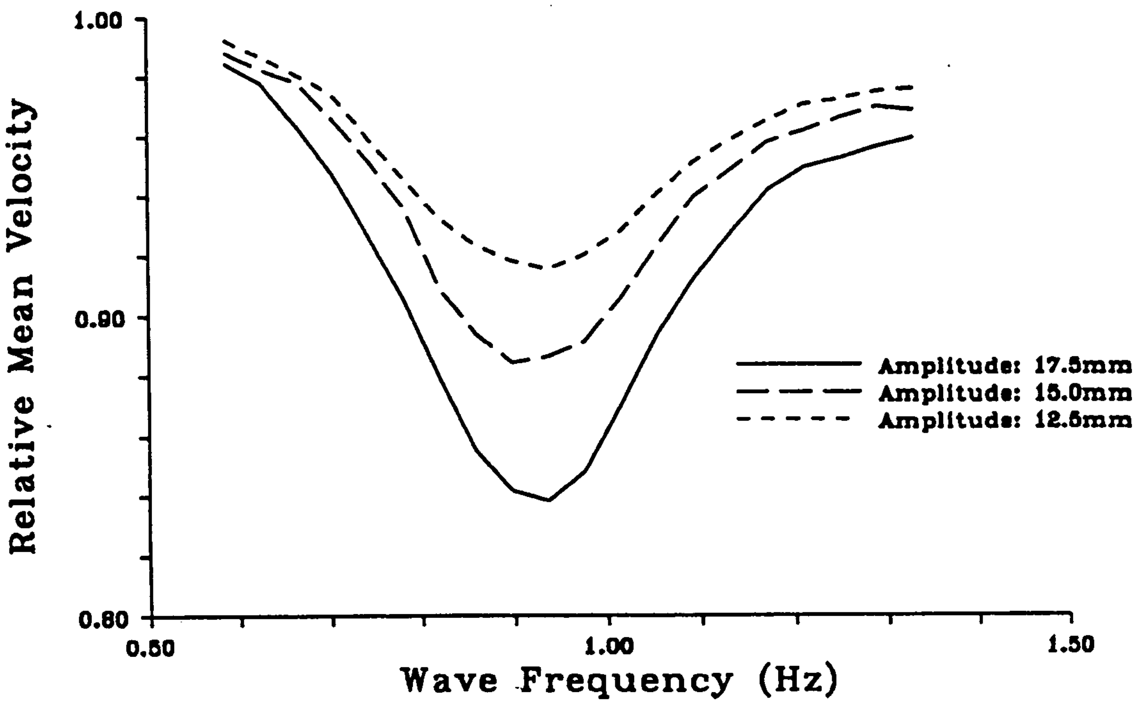


Figure 6.11 Normalised Velocities with Expanded Y-axis

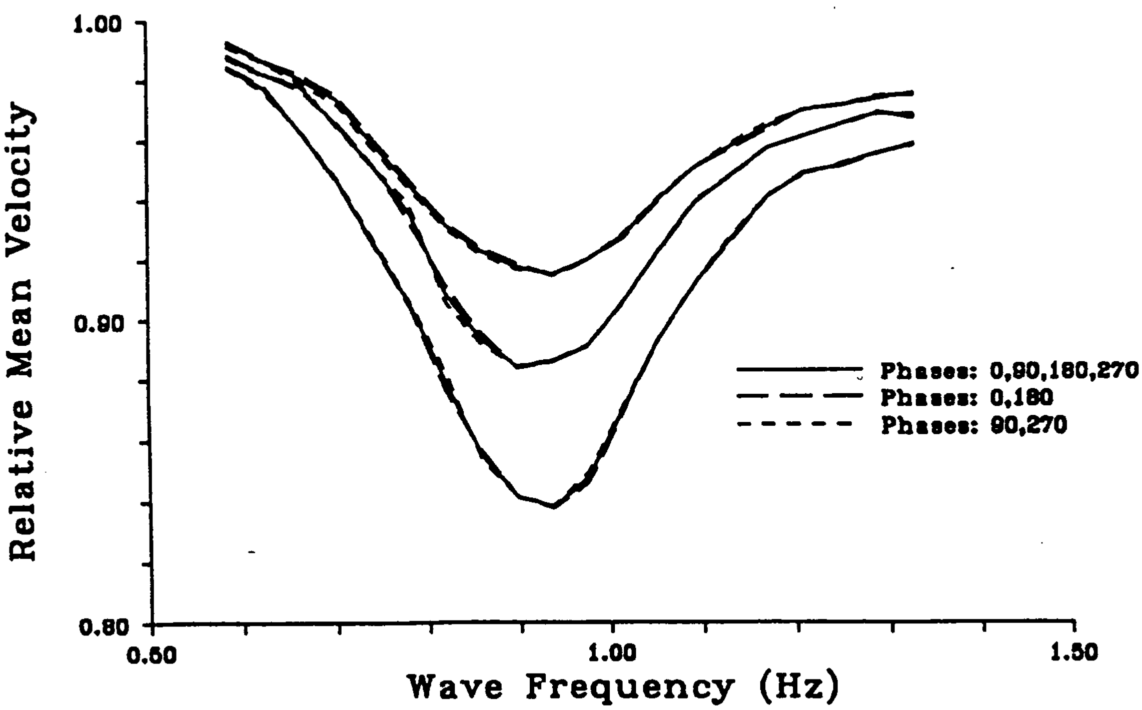


Figure 6.12 Different Phase Combinations

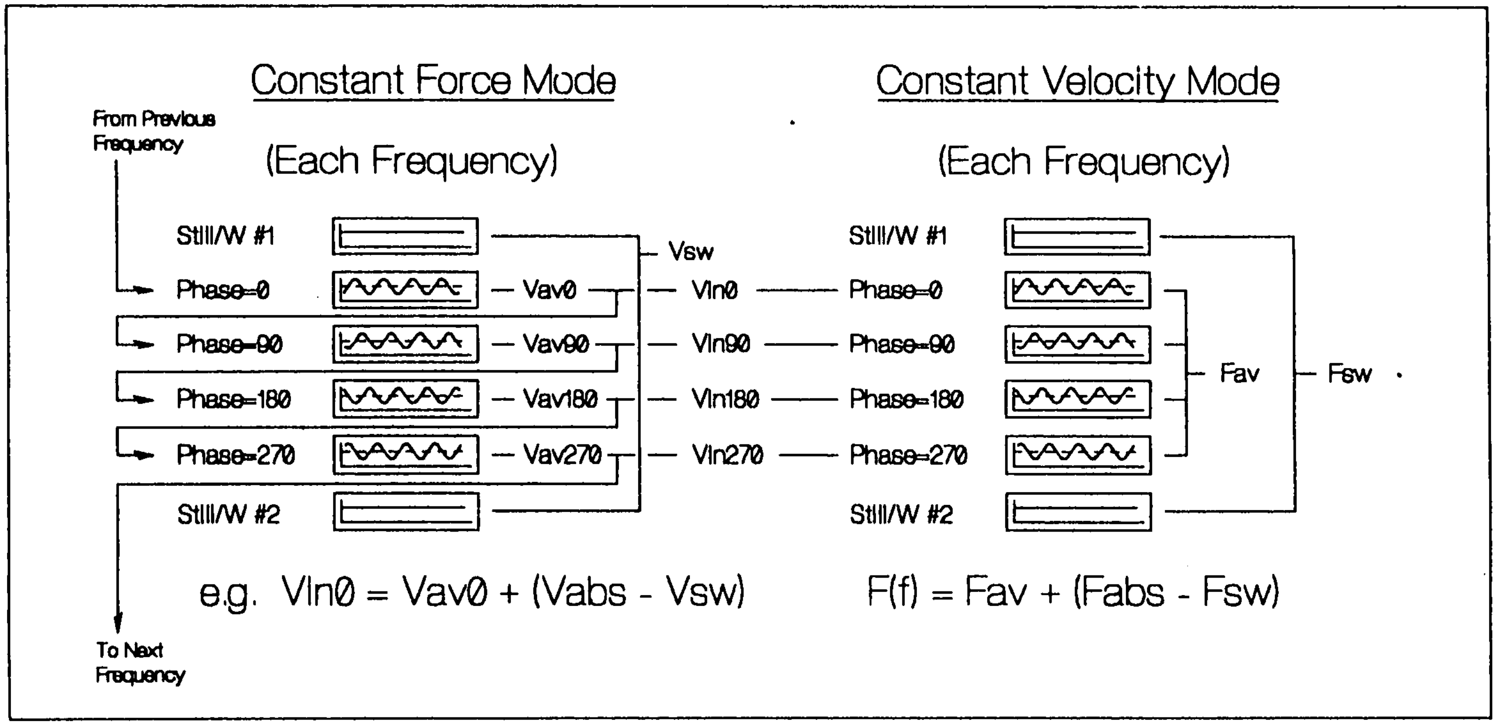


Figure 6.13 C.V./C.F. Comparison Test Format

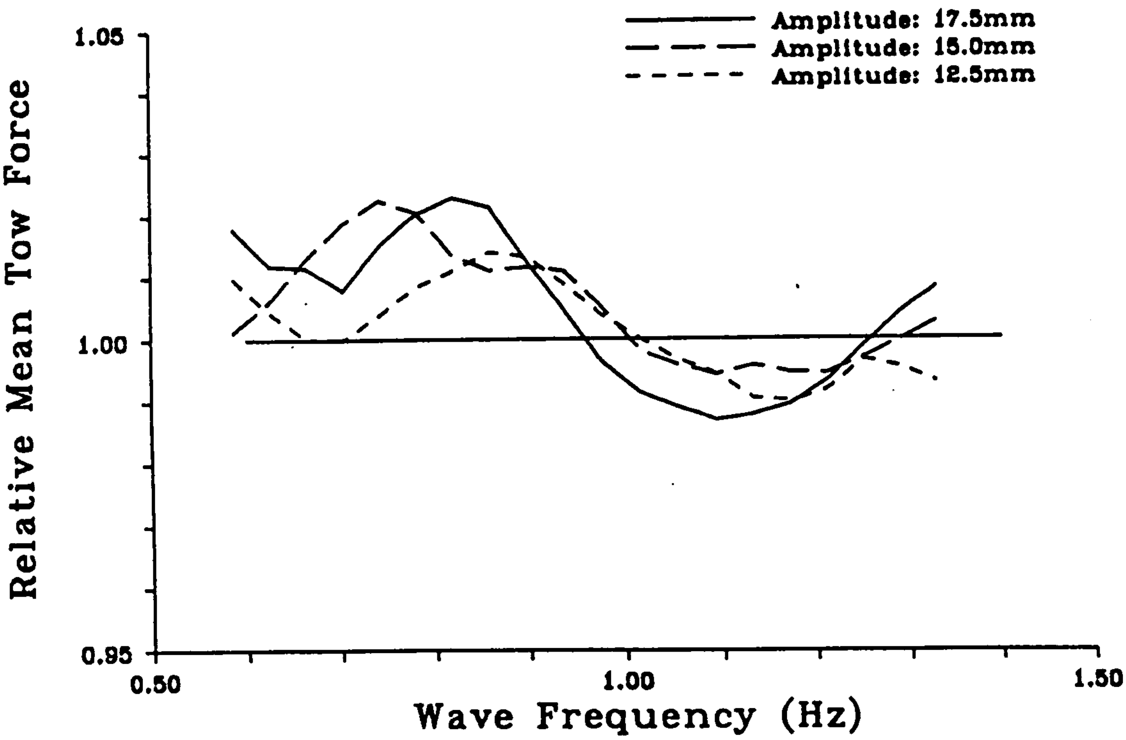


Figure 6.14 C.V./C.F. Comparison at 3 Amplitudes

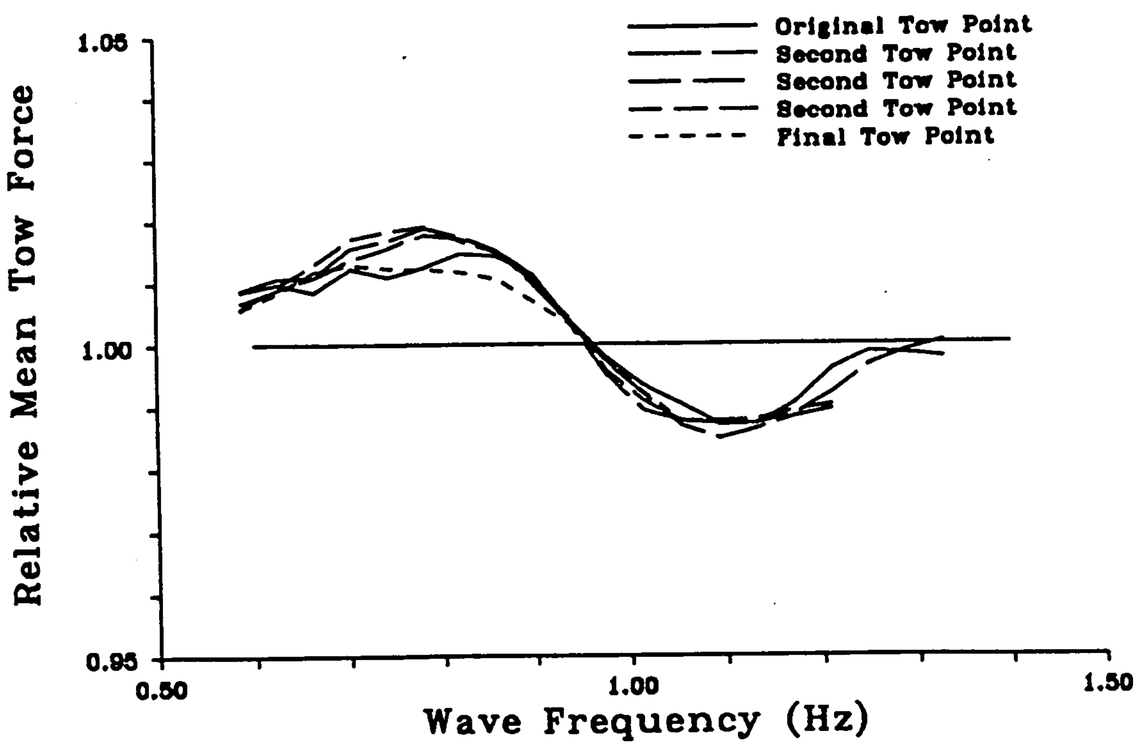


Figure 6.15 Effect of Variation of Towing Geometry

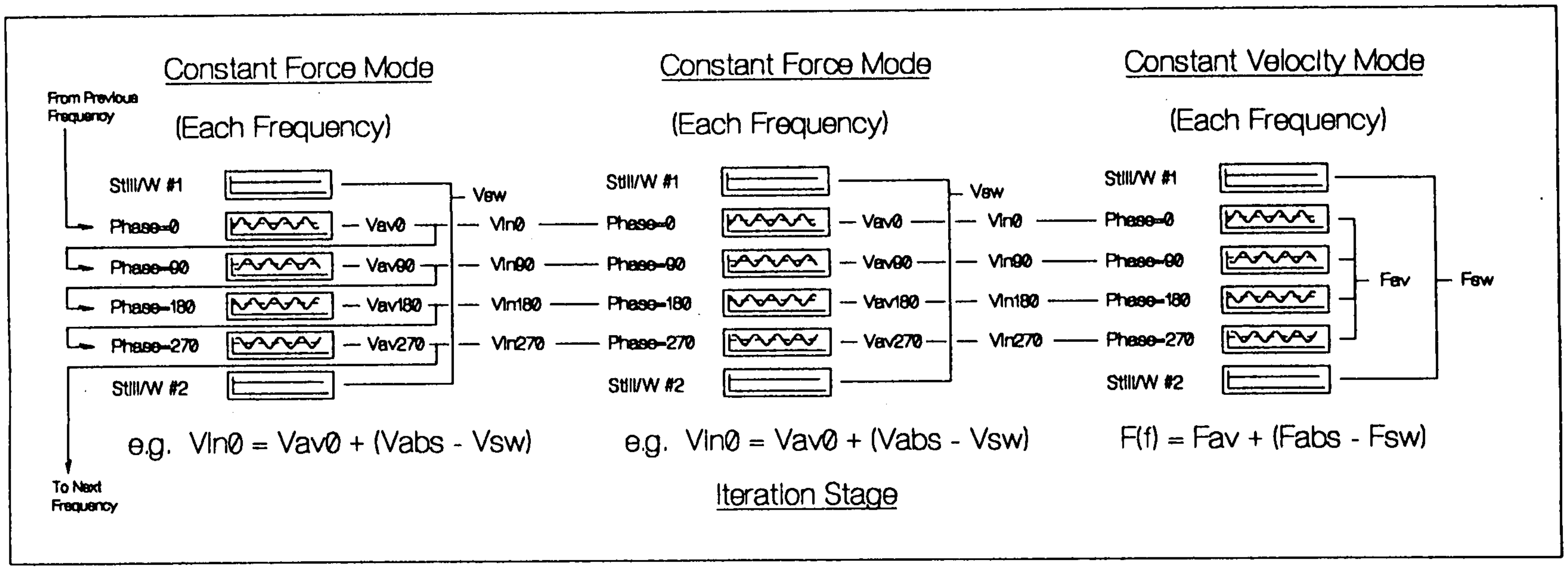


Figure 6.16 Iterated Comparison Test Format

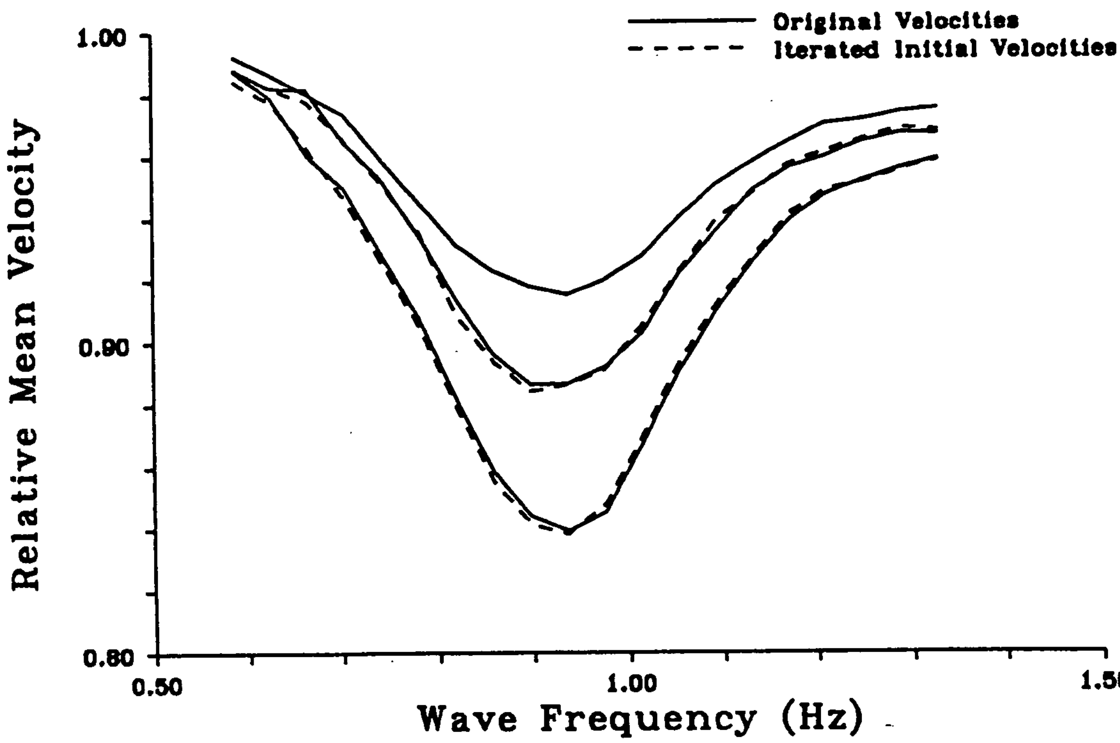


Figure 6.17 Original and Iterated C.F.V.'s

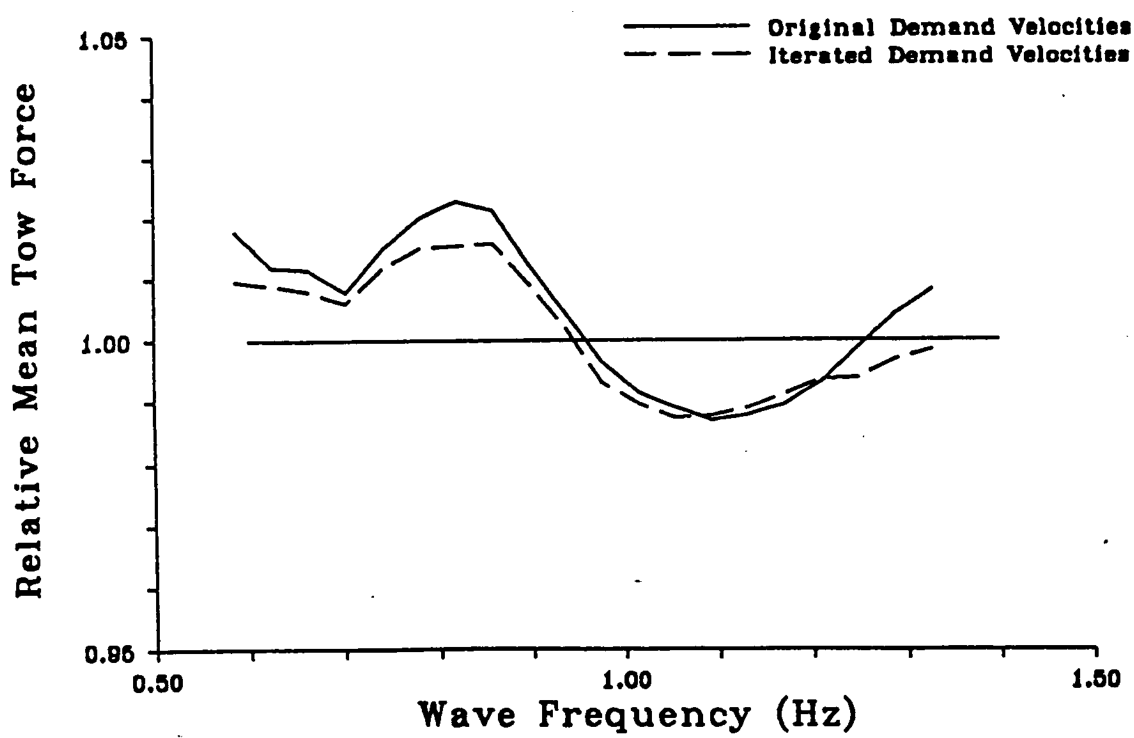


Figure 6.18 Iterated Results for Largest Wave Amplitude

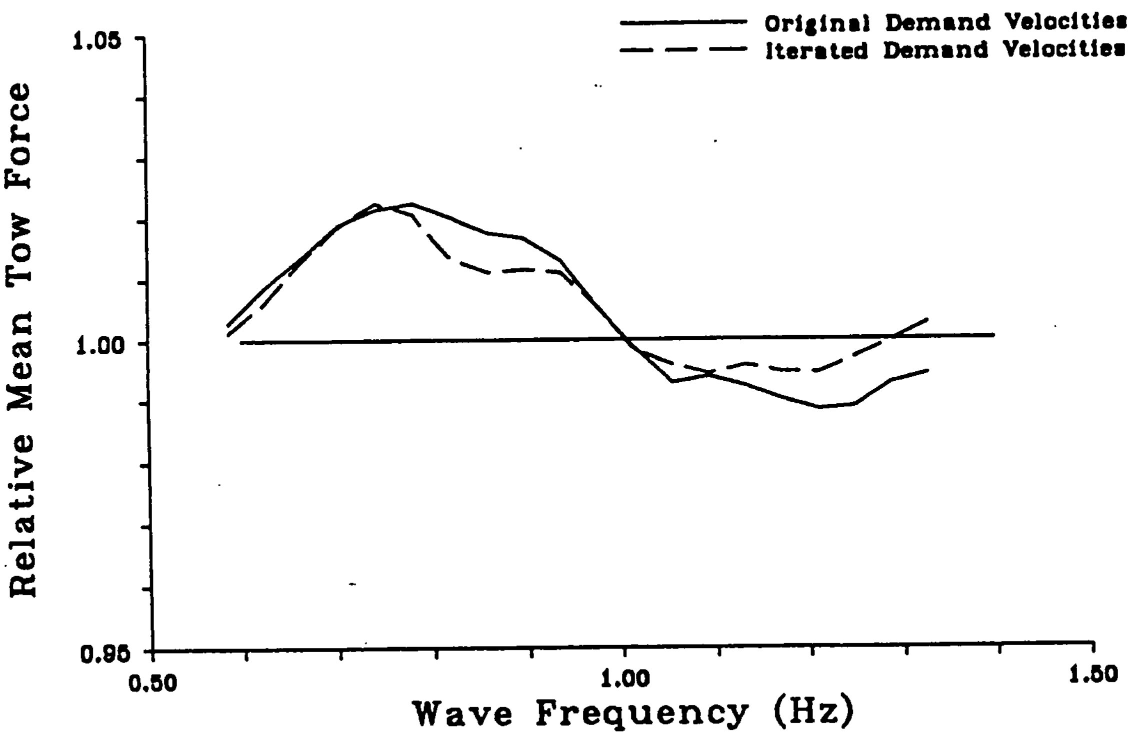


Figure 6.19 Iterated Results for Medium Wave Amplitude

Chapter 7. Project Conclusions

7.0 Chapter Summary

All the Chapters in this thesis contain their own individual conclusions but an overall summing up is still required, with special consideration given to the original project aims listed in Chapter 1. This will be presented in the form of a short overview of the whole project followed by a number of broad conclusions and recommendations for future work.

7.1 Project Overview

Over the four and a half years the project has been running three towing systems have been developed, with over 13,500 test runs carried out. The technology involved has progressively improved from open-loop non-synchronised testing, to a fully automated, closed-loop, multi-degree-of-freedom system. Although the final rig did not reach its design conclusion due to time spent developing the loadcells described in Part II of this thesis, all the ground work has been done for rapid implementation of the full 2-D controlled-force configuration. Most of the hardware has been thoroughly proven and control and testing techniques developed. A great deal of interesting data has already been generated by the interim 1-D configurations, opening up potential areas of study as described in the previous Chapters.

From the outset the three main objectives have been to improve data quality, data productivity and test realism. The three systems have gone a significant way towards achieving these goals. In particular the quality and speed with which results can be generated has met all expectations.

The high run rate generated due to the fully automated test-cycle and short tank settling time allows far more runs to be completed within a given testing period. This allows the use of many hundreds of runs where only a few tens would have been practical with other systems. This in turn means that the techniques developed to improve repeatability can be used without incurring unacceptable time overheads. In conjunction with the highly repeatable wave generation system this leads to significant improvements in terms of both data quality and rate.

Although the ultimate goal of full simulation of the various forces acting on a yacht has not yet been achieved, significant advances towards test realism have been made. Constant-Force techniques have been extensively developed and many of the associated problems addressed. Models are tested in oblique wave conditions while having complete freedom in all degrees-of-freedom except roll, where realistic damping is applied. The wave generation system at the Wide Tank, in conjunction with the fully automated towing system opens up new opportunities for the study of yacht performance in realistic sea states over a large, statistically significant number of runs.

The decision to develop a complex, multi-mode, software configurable towing system was justified by the results from a detailed comparison carried out between Constant-Velocity and Constant-Force techniques. Although the limited testing carried out for this thesis means that few firm conclusions can be drawn, the results have opened up a whole new area of fundamental research into the unsteady dynamics of yachts in waves. Implementation of the full 2-D system would take this research a stage further. Possible areas of study could include; interaction of hull motions with rig aerodynamics; systematic studies of the effects of rig configuration and centre-of-effort location; investigation of optimum steering techniques and aspects of controllability on all points of sailing.

The output from a set of standard test cases could be fed directly into a much simpler Velocity-Prediction-Program incorporating fewer approximations or assumptions. Testing time could be dramatically reduced, allowing cheaper design evaluation, thus making the techniques more widely accessible.

7.2 Project Conclusions

The following broad conclusions can drawn:

- Constant-Force techniques were developed and evaluated.

- Significant advances have been made in terms of data quality.

- High data productivity rates were achieved, reducing testing times.

- Significant steps towards improving test realism have been made.

- Initial tests have indicated an apparent difference between Constant-Force and Constant-Velocity towing methods.

7.3 Recommendations for Future Work

The following areas for further work have been identified:

- A full study of the differences between Constant-Force and Constant-Velocity techniques should be carried out. This should include studies of vessel motions for the two cases.

- Extensive tests should be carried out with a wider variety of wave conditions and model configurations.

- The full 2-D controlled-force system should be implemented and a similar set of development and evaluation trials conducted for this configuration.

- Performance predictions from all three systems should be compared to assess the validity of current techniques.

Appendix A. Model Details

Only basic details of the test models are supplied here to allow comparison of the results from this work with new or established data. In the case of the AC Class models the author is bound by a data confidentiality agreement.

If more detail of the test models is required please contact the author who will attempt to obtain the necessary clearance from the appropriate parties.

1. 12 Metre Model

The 12 Metre model was of 1950's vintage and was lent to the project by the British Hovercraft Corporation. Full details are available on request. Approximate model data is given below.

LOA:	1.75m
LWL:	1.21m
Bmax:	0.30m
Mass:	16.5kg
Radius of Gyration:	35%lwl

2. AC Class Models

The two 1/9th scale AC Class models were supplied by the Port Pendennis Challenge and were part of their systematic design series. They had identical bow and stern forms but significantly differed in mid-section shape. The following approximate full scale data applies to both models.

LOA:	21.5m
LWL:	17.8m
Bmax:	5.5m
Mass:	22,000kg
Radius of Gyration:	25%lwl

List of References

[1] Barnaby, K.C., "Basic Naval Architecture", Chapter 14, Hutchinson Scientific & Technical, London 1969.

[2] Bhattacharyya, R.,"Dynamics of Marine Vehicles", Chapter 11, John Wiley & Son, 1978.

[3] Strom-Tejsen, J., Yeh, H.Y.H, Moran, D.D., "Added Resistance in Waves", SNAME, 1973.

[4] Gerritsma, J., Beukelman, W., "Analysis of the Resistance Increase in Waves of a Fast Cargo Ship", International Shipbuilding Progress, Vol. 19, No. 217, Sept 1972.

[5] Comstock, J.P.(Editor), "Principles of Naval Architecture", Chapter 7, SNAME, New York 1969.

[6] Kirkman, K.L., "Gentlemen, Choose Your Weapons - The Race Conditions for the America's Cup", Marine Technology, March 1990.

[7] Van Hemmen, R.F., "12-Meter Design: State of the Art in 1986", Marine Technology, Vol. 23, No. 4, October 1986.

[8] Miller, R.T., Kirkman, K.L., "Sailing Yacht Design - A New Appreciation of a Fine Art", SNAME, Vol. 98 1990.

[9] DeBord, F. Jr, Teeters, J., "Accuracy, Test Planning and Quality Control in Sailing Yacht Performance Model Testing", NEYSYS, 1990.

[10] Kirkman, K.L., Pedrick, D.R., "Scale Effects in Sailing Yacht Hydrodynamic Testing", SNAME, 1974.

[11] Kirkman, K.L., "The Evolving Role of the Tank", 4th Chesapeake Sailing Yacht Symposium, 1979.

[12] Fryer, D.K., Gilbert, G.G., "The Useful Length of a Ship Tank", International Shipbuilding Progress, Vol. 36, No.408, 1989.

[13] Gadd, G.E., "Improved Methods for the Calculation of Ship Resistance", CADMO 86, September 1986.

[14] Havelock, T.H., "Drifting Force on a Ship Among Waves", Philosophical Magazine, Vol. 33, 1942.

[15] Joosen, W.P.A., "Added Resistance in Waves", 6th Symposium on Naval Hydrodynamics, Washington, 1966.

[16] Maruo, H., "The Theory of the Wave Resistance of a Ship in a Regular Seaway", Bull. of Fac. Eng., Yokohama University, Vol.6, 1957.

[17] Salveson, N., "Second Order Steady Forces and Moments on Surface Ships in Oblique Regular Waves", International Symposium on Dynamics of Marine Vehicles and Structures in Waves, University College London, April 1974.

[18] Lin, W.C., Reed, N.A., "The Second Order Steady Force and Moment on a Ship Moving in an Oblique Wave", 11th ONR, 1976.

[19] Maruo, H., "Resistance in Waves", S.N.A.J., Vol.8, 1963.

[20] Gerritsma, J., Beukelman, W., "Analysis of the Resistance Increase in Waves of a Fast Cargo Ship", International Shipbuilding Progress, Vol. 19, No. 217, Sept 1972.

[21] Loukakis, T.A., Sclavounos, P.D., "Some Extensions of the Classical Approach to Strip Theory of Ship Motions, Including the Calculation of Mean Forces and Moments", J.S.R., Vol.22, No.1, March 1978.

[22] Journee, J.M.J., "Motions and Resistance of a Ship in Regular Following Seas", Delft University of Technology, SHL Report, No.440, 1976.

[23] Gerritsma, J., Van Den Bosch, J.J., Beukelman, W., "Propulsion in Regular and Irregular Waves", International Shipbuilding Progress, Vol.8, No.82, June 1961.

[24] Klaka, K., Penrose, J.D., "Performance Prediction of Sailing Yachts in Waves", Conference on Yachting Technology, 1987.

[25] Salveson, N., Fritts, M., Meinhold, M., "Technology and Design for Stars & Stripes", Finite Elements in Analysis and Design, Vol.4, No.1, 1988.

[26] Letcher, J.S. Jr, Cressy, C., Oliver, C., Fritts, M., "Hydro-numeric Design of Winglet Keels for Stars & Stripes" SNAME Chesapeake Section Meeting, February 1987.

[27] Letcher, J.S. Jr, Marshall, J.K., Oliver, C., Salveson, N., "The Design and Performance of Stars & Stripes", Scientific American, August 1987.

[28] Letcher, J.S. Jr, McCurdy, R.C., "Data Collection and Analysis for the 1987 Stars & Stripes Campaign", 8th Chesapeake Sailing Yacht Symposium, March 1987.

[29] Oliver J., Letcher, J.S. Jr, Salveson, N., "Performance Prediction for Stars & Stripes", Society of Naval Architects and Marine Engineers Annual Meeting, November 1987.

[30] Scragg, C.A., Chance, B. Jr, Talcott, J.C., Wyatt, D.C., "The Analysis of Wave Resistance in the Design of the 12-Meter Yacht Stars & Stripes", 8th Chesapeake Sailing Yacht Symposium, March 1987.

[31] Kapsenberg, G.K., "A New Technique for Testing a Sailing Yacht in Waves", 10th Chesapeake Sailing Yacht Symposium, 1991.

[32] Kerwin, J.E., "A Velocity Prediction Program for Ocean Racing Yachts, Revised to February 1978", Report 78-11, H.Irving Pratt Ocean Racing Handicapping Project, MIT 1978.

[33] Spens, P.G., DeSaix, P., Brown, P.W., "Some Further Experimental Studies of the Sailing Yacht", SNAME, 1967.

[34] Pedrick, D.R., "The Performance of Sailing Yachts in Oblique Seas", Marine Technology, October 1974.

[35] Gerritsma, J., Keuning, J.A., "Performance of Light- and Heavy-Displacement Sailing Yachts in Waves", Marine Technology, Vol.26, No.1, Jan 1989.

[36] Avis, J., "The Use of an Anti-Pitch Hydrofoil to Reduce the Added Resistance of a Yacht in Waves", Sailtech-89, 18th Annual Conference on Sailing Technology, Vol.35, October 1989.

[37] Yemm, R.W., "The Design, Construction and Testing of a Constant-Force Yacht Testing System", 4th Year Project Final Report, Dept. of Mechanical Engineering, University of Edinburgh, May 1989.