A Sail Force Dynamometer: Design, Implementation and Data Handling.

By

James Stackpole Herman Jr.

B.S. Mechanical Engineering
Yale University (1985)

Submitted to the department of Ocean Engineering in partial fulfillment of the requirements for the degree of

Master of Science in Naval Architecture and Marine Engineering

at the

Massachusetts Institute of Technology September, 1988

©Massachusetts Institute of Technology

Signature of Author

Department of Ocean Engineering
September 18, 1988

Certified by

Jerome H. Milgram
Thesis Supervisor

Accepted by

A. Douglas Carmichael
Chairman, Department Committee

JUN 15 1989
A Sail Force Dynamometer: Design, Implementation, and Data Handling.

by

James Stackpole Herman Jr.

Submitted to the Department of Ocean Engineering in partial fulfillment of the requirements for the degree of Master of Science.

ABSTRACT

A 35 foot sailboat was designed and constructed to measure sail forces with a six component load cell dynamometer. To remove large self canceling forces present in the rig, all the rigging components are attached to the dynamometer. Additional sensors which provide information on wind angle and speed at various heights, boat speed, heel and pitch angle, and heading are interfaced to a computer which averages the data and saves it for further analysis. The dynamometer and the instruments are calibrated, and some preliminary results are presented.

Thesis Supervisor: Dr. Jerome H. Milgram
Title: Professor of Naval Architecture
Contents

Introduction 6

1 Structural Members 10
   Introduction ........................................... 10
   1.1 Overview of Dynamometer ............................ 10
   1.2 Hull ............................................. 11
   1.3 Frame ........................................... 13
   1.4 Flexure Rods ..................................... 17
   1.5 Construction ..................................... 20

2 Sensors and Electronics 22
   Introduction ........................................... 22
   2.1 Dependant Sensors .................................. 23
      2.1.1 Load Sensors .................................... 23
      2.1.2 Mast Head Units ................................. 24
      2.1.3 Clinometers .................................... 26
   2.2 Independent Sensors ................................. 27
      2.2.1 Ockam Instruments .............................. 27
      2.2.2 Northstar Loran ................................. 28

3 Software 29
   Introduction ........................................... 29
   3.1 Reading the Sensors ................................ 29
      3.1.1 Sampling the Electrical Signals ............ 30
      3.1.2 Storing the Samples ........................... 31
      3.1.3 Numerical Considerations .................... 32
   3.2 Reading the Ockam Data ............................. 33
   3.3 Manipulating the Data - The Main Program ........ 33

4 Calibration 35
   4.1 Sensor Calibration .................................. 35
4.1.1 Load Cells .............................................. 35
4.1.2 Mast Head Units ...................................... 37
4.1.3 Clinometers ............................................. 39
4.2 Frame Calibration ......................................... 39
  4.2.1 6x6 Calibration Matrix ................................. 42
  4.2.2 Frame Weight Correction .............................. 46
List of Figures

1.1 Load cell mounting hardware ........................................ 12
1.2 Profile of the hull ...................................................... 13
1.3 Final design of the frame ................................................. 15

2.1 Load cell conditioning circuit ........................................ 25
2.2 Wind sensor circuit ...................................................... 26
2.3 Heel and pitch angle circuit ........................................... 27

4.1 Wind speed calibration curve ......................................... 39
4.2 Indicated airspeed vs. updraft angle ................................. 40
4.3 Indicated airspeed vs. apparent wind angle ...................... 40
4.4 Indicated airspeed vs. apparent wind angle and updraft angle . 41
4.5 Final wind speed calibration curve ................................... 41
4.6 Measuring the direction of gravity in the boat’s reference frame . 47
Introduction

The ability to predict the maximum potential speed of a sailboat in a given wind condition is a very strong asset, allowing the naval architect and the handicapper to compare two sailboat designs and the crew of a sailboat to gauge their performance at sea. The estimation of a sailboat’s potential speed based on its design alone began in the 1930’s [1], and though initially carried out by hand, the calculations involved were greatly eased with the advent of velocity prediction programs (VPPs) designed to run on computers. Today, many such programs are in use, and the design of both cruising and racing sailboats benefit from their existence.

The velocity prediction program uses mathematical models to predict the forces generated by the sails and the forces generated by the hull, keel, and rudder. By balancing these forces, the VPP is able to find the speed and attitude at which the sailboat is in equilibrium. Clearly, the results obtained with the VPP can only be as accurate as the mathematical models.

The forces generated by the hull of a sailboat moving through the water can be estimated by towing a model of the sailboat in a towing tank. By towing a variety of models at systematically varied speeds, heel angles, yaw angles, and rudder angles it is possible to create a mathematical model that predicts the hull
forces generated when any hull moves through the water (as long as that hull is reasonably similar to the models used for the data base). The mathematical model used in the VPP to predict hull forces is derived from several different series of hulls, and represents hundreds of hours of tow tank testing [5].

The principal sail forces for a given wind angle and speed can be expressed with three numbers: the sail lift, drag and heeling moment coefficients. Given these non-dimensional numbers, the sailboat’s attitude, and the apparent wind information, the forces generated by the sails can be found. In the past, the calculation of these sail force coefficients has been accomplished in several different ways.

The first method of calculating sail force coefficients is analogous to a towing tank test: a model of the sails is placed in a wind tunnel, and a dynamometer measures the sail forces. Though this technique has been used [4], several drawbacks cast doubt on the reliability of the results. First, a wind tunnel does not accurately duplicate the wind shear and turbulence experienced at sea. Second, the varying apparent wind angle with height caused by the sailboat’s forward velocity in the shear flow cannot be duplicated in a wind tunnel. Finally, proper Reynolds number scaling requires very high wind speeds and sail forces, making it difficult to properly scale the sail cloth and shape. For example, in the tests done by Herreshoff [4], wind speeds in excess of 50 m.p.h. were used on a sail with an area of 8 square feet, resulting in side forces in excess of 250 lbs.

The second method of calculating sail force coefficients avoids these limitations. The process involves first recording the actual maximum attainable speeds for a full scale yacht at various wind angles and speeds. Then, a scale model
of the yacht is used in a towing tank to calculate the lift and drag of the hull in all of the possible speed, heel angle, yaw angle, and rudder angle combinations. By matching the attitude of the hull in the full scale tests to the model tests, the forces generated by the sails can be found \cite{6,3}. Given these forces and the attitude of the sailboat, the sail force coefficients are quickly calculated. Though this method avoids the problems of scale associated with the wind tunnel, it introduces other problems associated with scale in the towing tank. First, close agreement of the forces predicted by the towing tank and those experienced in reality are not guaranteed. Second, the drag on the full scale hull includes added resistance due to the ocean waves, which may not be accurately accounted for in the model tests.

The third method used in the past to calculate the forces that a sail generates is based on theory alone and solved by a computer. However, in past efforts, the separation of flow from the sails has not accurately been accounted for, so that the results are only applicable when separation effects are small.

A fourth method of calculating sail force coefficients is based on empirical data obtained from as many as 50 strain gages mounted in the rigging components of a full scale sailboat\cite{7}. By observing the change in these loads due to the effect of the sails, it is possible to calculate the forces that the sails generate. Though this method is not limited by the problems associated with model testing, it is not very accurate because the sails cause only a very small change in some of the rigging loads. For example, consider the reading taken from the strain gage in the backstay. If the measured load is 10,000 lbs with no sails up, then an error of only 1 percent could change the calculated sail force by as much as 100 lbs. In
very light winds, this error could be more than half of the correct total sail force.

The purpose of this project is to measure sail forces directly on the sailboat, but to eliminate the problems of the full scale measurement scheme described above. This is accomplished by isolating the rigging hardware on a frame, separate from the hull, and then connecting the frame to the hull with six load cells. As such, the sailboat becomes a six component dynamometer.

Such a sailboat (the sail test boat) has been designed and built at M.I.T. under the sponsorship of the Marine Technology Advancement Project. This report describes the work done on the sail test boat between the fall of 1986 and the present, and is divided into four chapters: the design and construction of the structural members, the selection and installation of the sensors, the software development necessary to handle the signals generated by the sensors, and the calibration of the system.
Chapter 1

Structural Members

Introduction

This chapter describes the design work and some of the decisions made in the process of creating the structural members of the sail test boat. As mentioned in the introduction, the sail forces are isolated on a frame, separate from the hull, and the frame is connected to the hull through load sensors and connecting rods. The chapter begins with an overview of the dynamometer, and then details each of the three parts that make up the dynamometer: the hull, the frame, and the rods that connect the hull to the frame. The last section gives a brief history of the construction of the sail test boat.

1.1 Overview of Dynamometer

The six component sail force dynamometer measures the three forces and the three moments (henceforth referred to as simply the six forces) acting on the frame. Since all the rigging components are mounted on the frame, the dynamometer will therefore measure the forces generated by the sails. To do so, at least six load sensing transducers are needed between the hull and frame, oriented in such
a way that each of the six forces is sensed by at least one of the transducers.

The dynamometer design chosen for the sail test boat is shown in figure 1.3, and features six compression/tension load cells that support the frame in space with connecting (flexure) rods. Each of the six hull to frame connections consists of two flexure rods, two rod mounts, and a load cell, as shown in figure 1.1. The threads in the rods and the spherical washers against the mounts allow the rods to be aligned somewhat once the dynamometer has been built.

The location of the load cell mounts is chosen to balance the sail forces as directly as possible. The heeling moment, for example, is transmitted to the frame primarily through the shrouds. Therefore, two vertical load cells are placed on either side of the mast, at a longitudinal position as close to the shrouds as possible. As an other example, the side force and yaw moment are transmitted to the frame primarily through the shrouds, the headstay, and the jib and main sheet tracks. Two transverse load cells, one mounted near the shrouds, the other near the main sheet traveler, combine to balance these forces.

1.2 Hull

An 83 foot “maxi yacht” was chosen to be the parent for the sail test boat because a 22 foot scale model of this hull was available for tank testing. Also, an overall length of 35 feet was chosen, which is large enough to obtain good sail force data and yet small enough so that it can be easily handled by just one or two people.

Initial analysis of the scaled down hull revealed that significant changes were needed to accommodate the dynamometer structure. The initial estimated weight of the sail test boat indicated that the displacement of the scaled down
Figure 1.1: The hardware that connects the frame to the hull.

hull had to be almost doubled, and the initial estimate of the center of gravity indicated that the stability had to be increased as well.

Three modifications were made to the hull to increase the stability and the displacement. First, the hull was sunk roughly 6 inches (at sail test boat scale), which essentially doubled the displacement. Second, the freeboard was increased 18 inches, thereby increasing the stability at high heel angles, and also creating more space belowdecks for the frame. And third, the hull was extended roughly 20 inches aft to compensate for the new waterline. The change in appearance due to these modifications can be seen in figure 1.2, which shows the profile of the scaled down parent superimposed on the profile of the sail test boat.

Given the changes made to the parent hull, it is clear that the 22 foot
Figure 1.2: Superimposed profiles of the scaled parent hull and the sail test boat. The changes were made to increase the stability and displacement of the sail test boat.

model of the maxi yacht no longer accurately represented the sail test boat. As a result, the tank test model was modified to be geometrically similar to the sail test boat below the waterline.

1.3 Frame

In theory, the dynamometer design presented earlier is both elegant and simple. However, before building, there are two sources of error that must be addressed: bending of the frame, and interference of the deck with the frame and the sail related hardware.

Referring back to figure 1.3, it is clear that applying a load to the frame causes distortion of the frame. This distortion, in turn, will bend some of the flexure rods that support the frame. Since the load cells only measure the force that is parallel to the flexure rod, the force required to bend one rod may
not be fully transmitted to any of the other load cells. This error, referred to here as flexure bending error, can be partly corrected in the calibration procedure. However, since practical corrections are linear only, it is desirable to keep this error to a minimum.

One way to minimize the flexure bending error is to make the rods more flexible. However, as explained in the next section, there is a minimum flexure size needed to support the expected loads acting on the frame. Another possibility is to increase the stiffness of the frame. Unfortunately, due to the relatively low stability of the hull, the frame must also be as light as possible, since it sits up quite high in the boat.

The final design of the frame, shown in figure 1.3, represents a compromise between stiffness and weight. The frame is made of a balsa wood and fiberglass sandwich. In regions of extreme tension, such as the bottom of the frame, carbon fiber is added to increase the stiffness. In regions of high localized loading, such as the load cell mounting locations, the balsa is replaced by plywood. The heart of the frame consists of a torsion box around the mast, that acts as a stiff connection between the shrouds and the mast. The mast is fixed at the bottom of the torsion box by the mast step, and also at the top by the partners. The shrouds tie into the side of the torsion box through the chain plates. As a precautionary measure, reinforcing wires tie the chain plates back to the mast step. The frame extends aft to allow mounting of the standoffs, winches, and backstay. Forward, an aluminum/stainless steel wire truss connects the frame to the headstay.

An other source of error to be addressed is interference of the deck with
Figure 1.3: The final design of the frame, showing the load cell mounting locations, the outboard profile of the hull, and the deck.
the frame. Because the rigging components are primarily above the deck, whereas the frame lies below the deck, penetrations are needed in the deck to allow the hardware to pass through and connect to the frame. These penetrations should be waterproof, but at the same time be as flexible as possible to minimize the force that the deck exerts on the frame. Since the dynamometer measures all the forces acting on the frame, the force exerted by the deck will be incorrectly attributed to the sails.

To connect the various tracks to the frame, aluminum cylindrical stand-offs pass through the deck, attaching to the frame and the tracks. To keep out the water, circular rubber bellows seal the space between the standoffs and the deck, but do not significantly restrict motion of the frame relative to the deck. Similarly, the chainplates for all the stays extend through holes in the deck, and are also sealed with rubber bellows.

To connect the winches to the frame, circular winch bases mount to the frame and pass through holes in the deck, so that the top of the winch base is even with the deck. When in place, there is roughly one half inch of clearance between the outside of the winch base and the inside of the hole in the deck. Before mounting the winches, a thin rubber sheet is fixed to the deck around the winch base opening, covering the winch base and the space between the winch base and the deck. Then, the winches are mounted to their bases, with the rubber sheet between the base and the winch. Like the rubber bellows, the rubber sheet prevents the water from entering, and also allows the winch (and thus the frame) to move freely relative to the deck.

The mast penetration is sealed with a rubber boot, like on most normal
sailboats.

1.4 Flexure Rods

Since forces required to bend one rod are not fully transmitted to the other load cells, it is imperative that the hull to frame connections be able to support large tensile and compressive forces, but not significant side loads. The error due to the rod bending, previously referred to as flexure bending error, is minimized on the sail test boat with flexures cut into the rods as close to the mounts as possible (see figure 1.1).

To determine the size of the flexure needed to support the loads acting on the frame, we must first estimate the maximum expected sail forces. This is easily accomplished using the stability curve of the hull combined with sail force calculations from the VPP. For example, the maximum forces expected in the forward vertical load cells are due to the righting moment of the hull balancing the heeling moment of the sails. The maximum righting moment for the hull is 13,300 ft-lbs, at 70 degrees heel, and the load cells are 4.6 feet apart. Including the component of the frame’s 800 lb estimated weight acting parallel to the rods, $800\frac{3}{4}\cos(70)$, balanced evenly between the two load cells, we find that the leeward rods experiences 3020 lbs in compression, while the windward rods experiences 2750 lbs in tension.

Similarly, when sailing off the wind with a spinnaker set, the maximum forward force generated by the sails is 800 lbs, acting at a height of roughly 20 ft above the waterline. Clearly, the maximum load expected in the longitudinal load cell is 800 lbs. Also, the pitch moment generated by this forward force causes a
tensile force in the aft vertical load cell of 1390 lbs. Subtracting the weight of the frame acting on this load cell, we find that the maximum expected load is 1120 lbs tension.

Finally, the maximum side force generated by the sails is 1050 lbs, experienced when the boat is heeling at 30 degrees. Adding the component of the frame’s estimated weight of 800 lbs that acts in the transverse direction, the maximum expected side force is 1450 lbs, to be opposed by the two transverse load cells. In addition, there is a considerable yaw moment generated by the sails, that is not calculated by the VPP. To be safe, it was assumed that each of the two side load cells could experience 1500 lbs.

Based on these maximum expected loads, the twelve flexure rods are grouped into two load ranges for the sake of convenience. The first group consists of the four rods used in the two vertical load cells which have a maximum expected load of 3000 lbs. With a safety factor of 3, these rods must therefore withstand 9000 lbs without failing. The other group consists of the eight rods used in the other load cells, and are designed for a maximum expected load of 1500 lbs and a safety factor of 2.5.

The maximum diameter of the rods is set at 1/2 inch, which allows them to thread directly into the load cells. Then, the minimum diameter of the rods, the flexure diameter, is set so that the flexure will not quite yield in tension. Finally, the flexure length is set so that the rod does not buckle in compression.

If we assume that the flexure is much more flexible than the 1/2 inch rod, then we can use the Euler buckling load to find the flexure length. That is, the critical buckling load for a column clamped at both ends in the fundamental
mode is

\[ P_{cr} = \frac{\pi^2EI}{l^2} \]  \hspace{1cm} (1.1)

where \( I \) is the moment of inertia \( \left( \frac{\pi^4}{4} \right) \) of the flexure, \( l \) is the length of the flexure, and \( E \) is the modulus of elasticity of the rod material. Clearly, this model requires that in the buckled state all of the bending takes place in the flexure. For flexure diameters close to 1/2 inch this will not be a good approximation.

Based on equation 1.1, the eight rods for the 1500 lb connections have .25 inch diameter flexures that are 2.9 inches long. In an Instron material testing machine, these rods supported 3800 lb without signs of buckling. The four rods for the 3000 lb connections, however, began to distort considerably at roughly 6500 lbs with 3.9 inch long, .33 inch diameter flexures.

A model based on the Raleigh quotient [2, for example] was used to redesign the four rods that failed. In this model, bending of the entire rod is accounted for, and the critical buckling load is found from

\[ P_{cr} = \frac{\int_0^L EI(v''(x))^2dx}{\int_0^L (v'(x))^2dx} \]  \hspace{1cm} (1.2)

where \( L \) is the length of the column, and \( v(x) \) is a trial function of the column displacement in the buckled state. To account for the different section moduli of the rod, two cubic trial functions are used: the first, \( v_1(x) \), \( x \in (0, l) \), gives the displacement of the flexure, and the second, \( v_2(x) \), \( x \in (l, L/2) \), gives the displacement of the 1/2 inch rod. \( l \) is the length of the flexure, and as above, \( L \) is the length of the rod. The following boundary conditions are used to determine the coefficients of the two cubics:
\[ v_1(0) = 0 \]  
(1.3)
\[ v'_1(0) = 0 \]  
(1.4)
\[ v_1(l) = v_2(l) \]  
(1.5)
\[ v'_1(l) = v'_2(l) \]  
(1.6)
\[ E_1v''_1(l) = E_2v''_2(l) \]  
(1.7)
\[ E_1v'''_1(l) = E_2v'''_2(l) \]  
(1.8)
\[ v_2(L/2) = \delta \]  
(1.9)
\[ v'_2(L/2) = 0 \]  
(1.10)

\( \delta \) is the displacement of the column at its midpoint, and cancels out of equation 1.2. The first two boundary conditions and the last two boundary conditions are specified by the fundamental buckling mode shape. Equations 1.5 through 1.8 specify the conditions when \( x = l \), and are derived by noting that the displacement, the slope, the sheer force and the bending moment must all be continuous across the end of the flexure.

The buckling load is found by solving for the coefficients in the trial functions, and then performing the integration in equation 1.2. The calculations led us to choose .4 inch diameter, 3 inch flexures which were able to support 10,000 lbs without buckling.

### 1.5 Construction

This chapter has so far detailed the design of the hull, frame and flexure rods. This section briefly outlines the construction of these and other parts as they have
occurred over the last two years.

Design of the sail test boat began in the fall of 1986. By the spring of 1987, the design for the hull and frame were complete: the plans for the hull and deck were sent Greene Marine in Yarmouth Maine, and the plans for the frame were sent to Lindsey Boat Works, in Gloucester Massachusetts. Shortly thereafter, the design of the aluminum fittings needed to connect the frame, hardware and hull was complete. These plans were sent to Metalmast Inc. in Putnam Connecticut.

In the fall of 1987, the design of the electronics desk was complete, and the plans were sent to Greene Marine.

During the winter, the electronics were selected and purchased, and construction of the interface electronics began. Also, the electric panel was installed in the electronics desk, and the desk was wired up to ease the installation of the desk in the boat.

In the early spring of 1988, the hull was complete, and the installation procedure began. First, the engine and associated hardware went in. Then, the frame was temporarily installed so that the deck penetrations could be completed. Once the penetrations were done, the frame and deck were taken off, and the electronics desk was installed. Then, the frame went back in, this time with the load cells and flexure rods. Lastly, the deck was fixed to the hull.

By the end of April, the hull was ready for painting and the installation of the keel. On May 11th, the sail test boat was launched, and the rig was put in for the first time.
Chapter 2

Sensors and Electronics

Introduction

The design of the electronics system for the sail test boat is done with standard circuits and commercially available transducers. At the heart of the electronics system is an IBM PC-XT computer. In addition to two serial communications ports the computer is equipped with a 16 channel analog to digital converter (A to D board) and a 5 channel counter/timer board, both of which are manufactured by the Metrabyte Corporation.

Dash-16 The Dash-16 is a 12 bit A to D board capable of reading eight channels in differential mode, or sixteen channels in single ended mode, and in either unipolar or bipolar voltage ranges. Thus with a signal of +/- 5 volts we are able to resolve roughly 2.5 millivolts.

Ctm 05 The ctm05 counter board has 5 programmable counter/timers. For the sail test boat, the channels are set up as event counters. By keeping track of the time, the computer can calculate the frequency of square wave or pulse train signals coming to the counter board.
Two types of sensors supply information to the computer: those that are directly connected to the computer, and those that are a part of their own electronic subsystem. The next section of this chapter describes the sensors that send a signal directly to the computer, either as an analog voltage, or as a pulse train. These sensors include the load cells, the wind sensors, and the heel and pitch sensors. The second section describes the self contained subsystems, which have their own microcomputer and communicate with the IBM XT over serial lines. The two self contained subsystems on the sail test boat are the Ockam instrument system and the Northstar LORAN.

2.1 Dependant Sensors

2.1.1 Load Sensors

Several guidelines are used to select the load cell most appropriate for measuring the forces acting on the dynamometer. First, to maximize the resolution of the readings, the rated capacity of the load cell should be close to the maximum expected load. Second, the load cells must be able to withstand roughly the same force as the flexure rods without failing. Third, the load cells must read the axial load only, and reject any side forces due to bending of the flexure rods. Fourth, the load cell must have a much higher flexural rigidity than the flexure rods, so that the load cell does not cause the rods to buckle. And finally, the load cell should be capable of withstanding the environment of the ocean, which means that it must be waterproof, somewhat rustproof, and send the same signal for a given force in widely varying temperatures.

The sensor best suited for this job is the shear beam load cell, one
of which is shown in figure 1.1. The two models used on the sail test boat are both manufactured by Hottinger Baldwin Measurements Inc. Two USB-5K load cells (5000 lb rated capacity, 12500 lb failure) are used for the forward vertical connections, and four USB-2K load cells (2000 lb capacity, 5000 lb failure) are used for the other connections. These force transducers consist of a wheatstone bridge with live gages in all four arms, and have both temperature compensation and modulus compensation built in. (The temperature correction is needed because the strain gages change resistance with temperature, and the modulus correction is needed because the elasticity of the steel used in the load cell changes with temperature.) The output of the load cell is linear to within .03 percent of full scale, changes less than .08 percent with a 100 deg F, and senses only .2 percent of the side load on the cell.

Both the USB-2K and the USB-5K generate an output of 2 millivolts for each volt of bridge excitation, at full scale loading. Analog Devices 2B31K strain gage signal conditioners amplify this signal so that an output of ±5 volts results when the maximum expected compression/tension acts on the load cells. In addition, a 2 Hz low pass filter attenuates unwanted high frequency signals. The load cell conditioning circuit is given in figure 2.1.

2.1.2 Mast Head Units

In the past, the wind speed and direction have been derived from a single sensor located at the top of the mast. To improve the sail force coefficients, the wind profile as a function of height will be recorded on the sail test boat, derived from 5 sensors mounted on a specially made 27 ft mast.
Figure 2.1: The circuitry that generates the load signal to be read by the A to D board.

Selection of the wind sensors is dictated by several factors: the transducers must provide accurate and repeatable readings, they must be able to withstand the ocean environment, and they must be relatively easy to interface to the computer. Stowe Microl ine wind sensors were chosen due to their simplicity and relatively low price. These sensors feature a single slider potentiometer for the wind angle sensor, and a Hall-effect generator for the wind speed sensor.

The output from the potentiometer is between 0 and 6.9 Volts, whereas the computer’s A to D board reads +/- 5 volts. An op-amp network is used to amplify and shift the input to the desired level.

The output from the Hall effect generator, on the other hand, is a
Figure 2.2: The circuitry that generates the wind speed and direction signals to be read by the computer.

square wave with an amplitude equal to that of the power supply. A resistor network attenuates the signal so that it triggers the TTL level CtM-05 counter. The complete wind sensor circuit is given in figure 2.2.

2.1.3 Clinometers

To calculate the forces exerted by the sails, the computer starts with the forces acting on the frame and then subtracts the forces due to the weight of the frame and rig. The heel and pitch angles needed to determine this correction are measured by two Accustar clinometers, manufactured by Schaevitz Sensing Systems Inc. These sensors, in addition to being completely encapsulated, have well filtered outputs with a time constant of about 2 seconds. Also, the sensors are gimbaled in only one direction, so that they measure just the component of gravity acting in the plane of the sensor.

The output of the Accustar clinometer is a linear function of the inclined angle, and has a sensitivity of 60 millivolts per degree. Before going to the
Figure 2.3: The circuit that generates the heel and pitch angle signals to be read by the computer.

computer, this signal is amplified so that the full range of the A to D is used. Clearly, the expected range of heel angles is much larger than the expected range of pitch angles, so the pitch angle signal is amplified quite a bit more than the heel angle. The interface circuitry that amplifies the signals is shown in figure 2.3, and enables the A to D board to read heel angles between -40 and 40 degrees, and pitch angles between -6 and 6 degrees.

2.2 Independent Sensors

2.2.1 Ockam Instruments

The Ockam system is state of the art yacht instrument system, controlled by a central CPU that receives signals from various sensors, and displaying data derived from the sensors on repeaters located anywhere on the yacht. As with most electronics manufactured for sailing yachts, the Ockam instrument system can be configured in many different ways.

The system implemented on the sail test boat has a central processing
unit (CPU) that collects data from three sensors. The mast head unit provides the Ockam with the apparent wind angle and apparent wind speed at the top of the mast. The boat speed transducer (impeller), located on the centerline of the boat forward of the keel, provides the system with the boat’s speed through the water. The compass, mounted on the frame forward of the mast, provides the system with the boat’s compass heading.

To view the data calculated by the Ockam CPU, six repeaters are mounted in the electronics desk, and four repeaters are mounted on deck to be viewed from the cockpit.

Finally, two way communication between the IBM computer and the Ockam is made possible by an RS232 module connected between the Ockam multiplexing line and the IBM-XT’s serial port. In addition to sending the Ockam data to the IBM-XT, this module allows the IBM-XT to send commands to the Ockam. For example, the IBM-XT can change the time constants for the Ockam averaging schemes. Also, the computer can send data to any repeater, allowing it to display information in the cockpit, from where it is difficult to see the IBM-XT’s screen.

2.2.2 Northstar Loran

The other independent subsystem on the sail test boat is the loran. The loran uses radio signals from land based transmitters to determine its location. Though not immediately useful for sail force measurements, the loran will be essential in calculating the current in future tests.
Chapter 3

Software

Introduction

The data acquisition software on the sail test boat has three main components. The first two consist of routines that sample the data from the electrical signals and from the Ockam instrument system. They are both interrupt driven and therefore run as background processes. The third is the main program which runs as the foreground process. It averages the samples taken by the interrupt handlers, displays the results on the screen, and saves the results on the disk. Additionally, it allows the user to send commands to the Ockam system, change various timing parameters, and calibrate some of the sensors.

3.1 Reading the Sensors

The routines that measure the electrical signals generated by the dependent sensors must sample the signal and store the result in memory to be accessed by the main program. This section describes the function of these routines, and details two techniques used to make them more efficient.
3.1.1 Sampling the Electrical Signals

Analog Signals

By definition, the analog signals that come to the computer are continuous time signals. On the other hand, the computer is a discrete time machine, so that the computer can only sample the electrical signal. This process of converting a continuous signal into a series of discrete samples introduces error, which is minimized in two ways: the sampling period is kept as consistent as possible, and the sampling period is set high enough to avoid aliasing of the analog signal.

The first sampling problem is very conveniently solved by the IBM-XT's internal clock, which generates a software interrupt through the time of day services 18.2 times per second. By trapping this interrupt, (1C hex) the data acquisition software is able to consistently sample the data with any period that is an integer multiply of 1/18 th of a second.

The second sampling problem is solved by combining analog filters and a suitable sampling period. For example, the load cell signal conditioners have low-pass filters with a cutoff frequency of 2 Hz. To sample this signal, and not loose any information, we must sample at or above the Nyquist frequency which in this case is 4 Hz. This does not present a problem, as the computer can sample at any frequency up to 18.2 Hz. The signals from the pitch and heel meters have similar filters built into them, and therefore do not require any additional analog filtering. The wind angle sensors, on the other hand, did not have any analog filtering, and required the addition of first order RC lowpass filters.
Digital Signals

Whereas the process of sampling an analog signal requires some care, the process of sampling a pulse train, or square wave, is very straightforward. Every time the computer samples the data, it saves the number of pulses that have occurred since the last sample, and the elapsed time since the last sample. By dividing the number of pulses by the elapsed time it determines the frequency of the pulses. Of course, a large number of pulses should be averaged so that the effect of a pulse occurring immediately after a sample was made does not significantly change the result.

3.1.2 Storing the Samples

By sampling the data with an interrupt driver, the sampling process runs in the background, while the calculations, disk operations, user interaction, and screen display all run in the foreground. Separating the sampling process from the rest of the program allows the foreground process to ignore the timing problems associated with sampling the data. For example, if the computer is sitting waiting for user input, and the sampling process runs in the foreground, then no samples could be taken during the time that the user decides what to type. However, as a background process, the sampling is completely unaffected by anything that the foreground process does, short of crashing the computer.

Obviously, the sampling process must store the samples that it gathers in such a way that the foreground process can average them. This is most efficiently done with a circular queue, in which elements are placed in consecutive spots on a carousel, or circular array. This means that the most recent sample
always overwrites the oldest sample, so that at any point in time, the queue has the last $N$ samples, where $N$ is the size of the queue.

To read the signals and store the samples in the queue, the data acquisition software uses two routines, one for the A to D board, and the other for the counter board. The A to D routine saves an integer in the range 2047 to -2048 for each channel, corresponding to the ±5 volt input range. The counter board routine saves an integer for each channel, corresponding to the number of pulses measured by each counter. It also saves the time that has passed since the last reading to allow the main program to determine the frequency of the signal.

### 3.1.3 Numerical Considerations

Since the sampling process ties up the computer while it gathers data, it is essential that this process be as quick as possible. Two steps are taken to optimize the speed of the sampling functions.

First, the sampling functions deal with just integers. Both of the sampling functions return integers that correspond to voltages and counts, and since it is much faster to manipulate integers than floating point numbers, the sampling functions just save the integers. This leaves the conversion process to the averaging scheme which runs in the foreground.

Second, the software allows only queue sizes that are an even power of two so that fast logical operations can be used to determine the next queue location. After a sample is taken, and the result stored in memory, the index counter is incremented so that the routine will know where to place the next sample. When the counter is greater than the queue size, the remainder of the

32
counter divided by the queue size represents the next location. This remainder is most quickly calculated by masking off the lowest order \( n \) bits, where the queue size is \( 2^n \). Though this technique limits us to queue sizes that have 1, 2, 4, 8, 16, \ldots elements in them, it is the fastest method of storing the samples.

3.2 Reading the Ockam Data

The other background process is a device driver supplied with the Ockam RS232 interface module. This driver reads the data at the serial port, sent by the Ockam instruments, and places it in memory for access by other programs. Additionally, the driver can be called by other programs with instructions for the Ockam CPU.

3.3 Manipulating the Data - The Main Program

The previous sections explain how the data that we wish to acquire is placed in the computer's memory. From there, it is the job of the main program to average these numbers and manipulate them as needed. This section describes the actions taken by the main program in a typical data acquisition session, which can be grouped into two categories: the start up procedures, and the main loop.

The main program's start up procedures prepare the computer for data logging. First, the calibration files needed to convert the voltages and frequencies into lbs, degrees and knots are loaded from the disk. Then the hardware items such as the screen, the D to A board, the counter board, and the Ockam interface are initialized. Finally, the sampling procedure is initiated by changing interrupt vector 1C hex so that it points to the data sampling routines.
The first step in the main loop of the program is to average the samples taken by the A to D board and the counter board. For the analog voltages, this is accomplished by adding the samples in each queue, and dividing by the number of items in the queue. Then, the average (which is a floating point number) is scaled and offset to represent a force in the case of the load cells, or an angle in the case of the wind angle sensors and clinometer sensors. The average of one of the counter board signals is found by adding the total number of pulses and dividing the result by the time over which the samples were made. The resulting frequency (in pulses per clock tick) is then scaled to represent a wind speed in knots.

After the averages from the A to D board and the counter board are calculated, the program transfers the character data read by the Ockam driver into local floating point variables.

At this point, the wind angle and speed information obtained from the wind sensor mast is corrected for heel. Then, the sail forces acting on the dynamometer are calculated from the six load cell readings (this calculation is explained in detail in chapter 4).

With all the calculations complete, the data on the screen is updated, and data is stored on the disk. Finally, the program responds to requests from the user.
Chapter 4

Calibration

Introduction

The physical quantity derived from the electrical signal that a sensor generates can only be accurate if the sensor has been properly calibrated. This is not only true for the load cells, the wind sensors, and the attitude sensors, but is also true for the dynamometer: we must calibrate the dynamometer to derive the sail forces from the load cell readings.

This chapter describes the calibration of the systems on the sail test boat, and the software used to simplify the calibrations. The first section describes the calibration of the sensors: the load cells, the wind sensors, and the attitude sensors, while the second section describes the calibration of the dynamometer.

4.1 Sensor Calibration

4.1.1 Load Cells

The output of the signal conditioner is a voltage proportional to the force acting on the load cell. Thus, we can convert the voltage measured by the computer to a force with a linear gain and a constant offset. That is, the load $l$ is given by
\[ l = ax + b \]

where \( a \) is the gain, \( b \) is the offset, and \( x \) is the measured voltage. Two methods are typically used to find these calibration numbers: a physical calibration, or an electrical (shunt) calibration. In the first method, the output of the signal conditioner is plotted against a varying applied load (such as a weight). By fitting a line to the data points, the values of \( a \) and \( b \) can be found. Due to the difficulty in removing the load cells from the sail test boat, however, this method is not practically feasible.

The shunt calibration is accomplished by shunting a resistor across an arm of the Wheatstone bridge. Given the mechanical and electrical characteristics of the load cell, it is possible to calculate the physical load that a particular shunt resistance represents. Thus, the shunt calibration simulates one known physical load and allows us to find the gain \( a \).

A relay is wired into the load cell circuit so that the shunt resistor can be included or removed under control of the main program. By calculating the change in the load cell reading caused by the shunt resistor, compared to the correct value stored in a data file, the gain \( a \) can be found.

Unlike the physical calibration, the shunt calibration does not determine the offset \( b \) since only one data point is measured. Thus, the data acquisition program on the sail test boat allows the user to zero the load cell readings when the boat is at rest.
4.1.2 Mast Head Units

Wind Angle Sensor

Like the load cell signal, the voltage generated by the wind angle sensor is a linear function of the apparent wind angle, so that the calibration must produce a gain and an offset.

The wind angle sensors generate a voltage which has a maximum and a minimum value. The difference between these represents the range of the wind angle sensor, which, accounting for the flat spot on the potentiometer, corresponds to roughly 350 degrees. By recording the maximum and minimum output from the wind sensor circuit as the wind vane moves from -180 to +180 degrees, it is possible to calculate the correct gain.

Like the shunt calibration, the calibration described above does not determine the offset for the wind angle. Thus, the data acquisition program allows the user to zero the wind vanes when they are aligned on a known apparent wind angle.

Wind Speed Sensors

The wind speed sensors generate a square wave that has a frequency that is roughly a linear function of the wind speed, and according to the manufacturers is given by a linear relationship of .45 Hz/Knot. To check this recommended value and to experimentally measure the effect of heel on the measured wind speed, wind tunnel tests were performed that provided some very useful information.

First, the wind speed is not quite a linear function of the pulse fre-
quency, but can be accurately described by a quadratic. Figure 4.1 shows both the suggested linear fit, and the quadratic fit based on the experimental results for an apparent wind angle of 0 degrees.

Second, the effect of heel on the measured apparent wind speed is not correctly accounted for using simple geometry. Figure 4.2 shows the change in measured wind speed as a function of the updraft angle $\phi$ for an apparent wind angle of 0 degrees. Also shown on the graph is $\cos(\phi)$, which is the component of the wind speed in the plane of the anemometer cups. Assuming the trigonometric relationship holds results in an error on the order of 10 percent at 30 degrees, whereas a cubic fit to the experimental data has essentially no error.

Finally, it was found that the flow around the support tube had a dramatic effect on the indicated airspeed. With the upwash angle set at zero degrees, the tube was turned to represent apparent wind angles between -110 and 110 degrees. The change in the indicated airspeed, which clearly should be zero, is shown in figure 4.3 as a function of the apparent wind angle. To reduce the influence of the support tube, an extension was built which increased the distance between the tube and the cups from .5 inches to 2.0 inches. Although this reduced the worst case error from 13 percent to 7 percent, another method was chosen to get reasonable results before winter set in.

Figure 4.4 shows the indicated airspeed as a function of apparent wind angle and updraft angle, for wind angles from -50 to 50 degrees, and updraft angles from 0 to 30 degrees. A correction function was fit to these data, and is shown in figure 4.5. Although the allowable apparent wind angles will preclude any beam reaching tests this fall, this solution will allow us to get some data to work on
over the winter. During that time, some other type of wind speed transducer will be found or designed to solve this problem.

![Plot](image.png)

Figure 4.1: The relationship between the wind speed sensor pulse frequency and the true airspeed with an apparent wind angle of 0 degrees.

### 4.1.3 Clinometers

The pitch and heel sensors generate a voltage that is proportional to the inclined angle of the sensor. Like the wind angle sensor, the desired quantity (the angle) can be determined with a gain and an offset. The gain is found by inclining the sensors a known amount, and the offset is calculated so that the sensors read 0 when the boat is level.

### 4.2 Frame Calibration

The computations which allow us to determine the sail forces given the readings from the six load cells are done in two steps. First, we transform the load cell readings into the six forces acting on the frame with a 6x6 calibration matrix.
Figure 4.2: The effect of updraft angle on indicated airspeed with an apparent wind angle of 0 degrees.

Figure 4.3: The indicated airspeed as a function of apparent wind angle, with an updraft angle of 0 degrees.
Figure 4.4: The indicated airspeed as a function of apparent wind angle and updraft angle.

Figure 4.5: The calibration curve derived from the experimental data shown in figure 4.4.
Then, with the known weight of the frame and the heel and pitch readings, we remove the force due to the weight of the frame, rigging, and sails. Therefore, the calibration of the dynamometer requires the calculation of the calibration matrix and the calculation of the frame's weight and center of gravity.

4.2.1 6x6 Calibration Matrix

The process of determining the six forces acting on the frame from the six load cell readings is a slightly more complex version of determining the force acting on a load cell given the voltage generated by the Wheatstone bridge. In the case of the single load cell, the gain and the offset describe the relationship between the voltage and the load. If we exert a force on the load cell, then we will get just one reading.

However, the six component dynamometer has six degrees of freedom, so when we exert a force \( \mathbf{f} = [f_1, \ldots, f_6] \) on the frame we get six readings, one from each load cell. To derive one of the six forces \( f_i \) from these six readings, we must account for the contribution from each of the six load cells. Consider, for example, the determination of the side force. Referring to figure 1.3, it would seem that the side force is simply the sum of the readings from the two transverse load cells. If the frame were infinitely stiff and the rods were all perfectly aligned then this would be true. However, the rods cannot be perfectly transverse, and will therefore be inclined slightly forward or aft, and slightly up or down. Due to the fore and aft misalignment of the transverse rods, the longitudinal load cell will detect some of the side force. Due to the vertical misalignment of the rods, the vertical load cells will detect some of the side force. Thus, due to small
misalignments, the side force will be sensed by all six load cells. In other words,

\[ f_2 = \sum_{i=1}^{6} a_i l_i \]  \hspace{1cm} (4.1)

where \( f_2 \) is the side force, \( l_i \) are the load cell readings, and \( a_i \) are the calibration numbers analogous to the gain for the single load cell.

Let \( \vec{f} = [f_1, \ldots, f_6] \) be a six component vector which describes the forces acting on the frame, and \( \vec{l} = [l_1, \ldots, l_6] \) be a six component vector consisting of the six load cell readings. Then by writing equation 4.1 for all six forces we have

\[
\begin{bmatrix}
    f_1 \\
    f_2 \\
    f_3 \\
    f_4 \\
    f_5 \\
    f_6 \\
\end{bmatrix}
= \begin{bmatrix}
    a_{11} & a_{12} & a_{13} & a_{14} & a_{15} & a_{16} \\
    a_{21} & a_{22} & a_{23} & a_{24} & a_{25} & a_{26} \\
    a_{31} & a_{32} & a_{33} & a_{34} & a_{35} & a_{36} \\
    a_{41} & a_{42} & a_{43} & a_{44} & a_{45} & a_{46} \\
    a_{51} & a_{52} & a_{53} & a_{54} & a_{55} & a_{56} \\
    a_{61} & a_{62} & a_{63} & a_{64} & a_{65} & a_{66} \\
\end{bmatrix}
\begin{bmatrix}
    l_1 \\
    l_2 \\
    l_3 \\
    l_4 \\
    l_5 \\
    l_6 \\
\end{bmatrix}
\]  \hspace{1cm} (4.2)

or equivalently:

\[ \vec{f} = A \vec{l} \]  \hspace{1cm} (4.3)

where \( A \) is now the 6x6 calibration matrix. Like the determination of the calibration numbers for a single load cell with a physical calibration, the determination of the calibration matrix is done by applying known forces to the dynamometer.

Let a loading condition be the application of one six element force vector \( \vec{f} \) to the dynamometer. For each loading condition we get equation 4.3, where \( A \) is the unknown 36 element matrix, and both \( \vec{f} \) and \( \vec{l} \) are known 6 element vectors.
To determine the 36 components of \( A \) exactly we need exactly 36 equations, or 6 loading conditions.

To minimize the error in the calibration procedure, it is desirable to use more than 6 loading conditions, and then “fit” the matrix \( A \) to this data, much like a line is fit to many data points in the physical calibration of a single load cell. If \( N \) loading conditions are used for the calibration, then equation 4.2 becomes

\[
\begin{bmatrix}
  f_1^1 & f_1^N \\
  f_2^1 & f_2^N \\
  f_3^1 & f_3^N \\
  \ldots \\
  f_4^1 & f_4^N \\
  f_5^1 & f_5^N \\
  f_6^1 & f_6^N 
\end{bmatrix} =
\begin{bmatrix}
  a_{11} & a_{21} & a_{31} & a_{41} & a_{51} & a_{61} \\
  a_{12} & a_{22} & a_{32} & a_{42} & a_{52} & a_{62} \\
  a_{13} & a_{23} & a_{33} & a_{43} & a_{53} & a_{63} \\
  a_{14} & a_{24} & a_{34} & a_{44} & a_{54} & a_{64} \\
  a_{15} & a_{25} & a_{35} & a_{45} & a_{55} & a_{65} \\
  a_{16} & a_{26} & a_{36} & a_{46} & a_{56} & a_{66} 
\end{bmatrix} \begin{bmatrix}
  l_1^1 \\
  l_2^1 \\
  l_3^1 \\
  \ldots \\
  l_4^1 \\
  l_5^1 \\
  l_6^1 
\end{bmatrix}
\]

(4.4)

where the superscripts refer to the loading condition. If we let \( F \) and \( L \) represent the matrices containing the vectors \( \vec{f} \) and \( \vec{l} \) for all the loading conditions, then equation 4.4 can be rewritten as

\[
LTAT = FT
\]

(4.5)

where now we have a system of \( 6N \) equations, and the 36 unknowns in the calibration matrix. If \( N \) is greater than 6, the system is overdetermined, and we therefore use a least squares approximation to find \( A \). In other words, \( A \) is chosen to minimize the Euclidean norm of

\[
LTAT - FT
\]

Standard math packages have subroutines that solve this least squares problem.
\[
\begin{bmatrix}
0.0115 & 0.0111 & 0.0329 & 1.0520 & 0.0123 & 0.0141 \\
0.0164 & -0.0094 & 1.0398 & 0.0004 & 0.0316 & 1.1465 \\
-1.0073 & -0.9933 & 0.0022 & 0.0164 & -0.9924 & 0.0005 \\
-2.4420 & 2.4057 & 0.4700 & 0.0470 & -0.1374 & 0.5744 \\
2.0462 & 2.0647 & -0.0122 & -0.0753 & -9.2242 & -0.0084 \\
-0.1880 & 0.1551 & 2.0736 & 0.0018 & -0.2847 & -11.0819
\end{bmatrix}
\]

Table 4.1: The calibration matrix before the removal of the deck seals.

To perform the calibration, the sail test boat was first removed from the water and supported by jack stands. After carefully leveling the boat, six attachment points were made to facilitate the load application. A total of 16 loading conditions were then applied, resulting in the calibration matrix shown in table 4.1.

Several inconsistencies can be found in this calibration matrix. For example, the side force $f_2$ consists primarily of the forward transverse load cell reading (#3) times 1.0398 and the aft transverse load cell reading (#6) times 1.1465. Ideally these two coefficients should both be 1.00: a number greater than 1.00 indicates that not all of the side force is being transmitted to the dynamometer.

After some analysis, it was discovered that deck seal interference was the primary cause of the calibration errors. Contrary to our initial estimations, the stiffness of the seals was not insignificanct relative to the distortions of the frame in real world conditions. In other words, the frame distorts enough when sailing that the deck seals exert a significant force on the dynamometer.

After modifying the seals to eliminate any interaction between the frame and the deck the calibration was repeated, yielding the matrix in table 4.2.
\[
\begin{bmatrix}
0.00792 & 0.00748 & 0.01013 & 1.04264 & -0.00123 & 0.01015 \\
0.00240 & -0.00092 & 1.01727 & 0.00620 & 0.01005 & 1.04641 \\
-0.99484 & -1.00104 & 0.02592 & 0.01347 & -0.97848 & 0.00425 \\
-2.46982 & 2.45443 & 0.73152 & -0.01993 & -0.00871 & 0.72084 \\
2.10913 & 2.14408 & -0.01613 & 0.02024 & -9.36049 & -0.08571 \\
-0.11051 & 0.01723 & 1.93190 & -0.00419 & -0.09827 & -9.95528
\end{bmatrix}
\]

Table 4.2: The calibration matrix after the removal of the deck seals.

In this matrix we can see that the inconsistency noted above has been greatly reduced.

### 4.2.2 Frame Weight Correction

To determine the sail forces we must be able to distinguish between the forces on the dynamometer due to the weight of the frame and rigging, and those due to the propulsion of the sails. When the boat is at rest, the load cells are zeroed so the computer indicates that no loads are acting on the frame. If the boat heels over, with no sails up, the weight of the frame begins to exert a force on the transverse load cells and reduces the force acting on the vertical load cells. Thus, the computer will indicate that there is both a vertical and a horizontal force acting on the frame, caused entirely by the weight of the frame, sails and rigging.

With the pitch and heel angle information it is possible to correct the forces calculated from the calibration matrix so that they are not affected by the changing gravitational force. Then, with no sails up, the boat could be inclined in both pitch and heel, and the computer would continuously indicate that no sail forces were present.

Figure 4.6 shows geometrically how the gravitational force is sensed by the heel and pitch clinometers. Let \( \vec{g} = [g_1, g_2, g_3] \) be the gravitational acceleration

46
Figure 4.6: The relationship between the gravitational acceleration vector and the measured heel and pitch angles.

vector of magnitude $g$. Let $\phi$ be the heel angle and $\theta$ the pitch angle, so that by inspection

$$g_1^2 + g_2^2 + g_3^2 = g^2 \quad (4.6)$$

$$\frac{g_1}{g_3} = -\tan \theta \quad (4.7)$$

$$\frac{g_2}{g_3} = \tan \phi \quad (4.8)$$

Solving for $\vec{g}$ in terms of $\theta$ and $\phi$ leads to

$$g_1(\theta, \phi) = \frac{g \tan \theta}{\sqrt{1 + \tan^2 \theta + \tan^2 \phi}} \quad (4.9)$$
\[
\begin{align*}
g_2(\theta, \phi) &= \frac{-g \tan \phi}{\sqrt{1 + \tan^2 \theta + \tan^2 \phi}} \quad (4.10) \\
g_3(\theta, \phi) &= \frac{-g}{\sqrt{1 + \tan^2 \theta + \tan^2 \phi}} \quad (4.11)
\end{align*}
\]

Now let \( \vec{g}_0 \) be the gravity vector when the boat is at rest, so that \( \vec{g}_0 = g(\vec{0}, 0) \) which is clearly \([0, 0, -g]\). The change in the gravitational force acting on the frame is then

\[
\Delta \vec{g} = \vec{g} - \vec{g}_0
\]

If \( \vec{f} = [f_1, \ldots, f_6] \) are the forces acting on the dynamometer, and \( \vec{f}^* = [f_1^*, \ldots, f_6^*] \) are just the forces generated by the sails. Then,

\[
\begin{bmatrix}
f_1^* & f_2^* & f_3^*
\end{bmatrix} = \begin{bmatrix}
f_1 & f_2 & f_3
\end{bmatrix} - m \Delta \vec{g} \quad (4.12)
\]

where \( m \) is the weight of the frame and rig. Additionally, if \( r_{cg}^- = [x_{cg}, y_{cg}, z_{cg}] \) is the center of mass of the frame and rig relative to the origin of our coordinate system, then

\[
\begin{bmatrix}
f_4^* & f_5^* & f_6^*
\end{bmatrix} = \begin{bmatrix}
f_4 & f_5 & f_6
\end{bmatrix} - r_{cg}^- \times [m \Delta \vec{g}] \quad (4.13)
\]

To determine the total correction needed to remove the weight forces from the dynamometer reading we apply equations 4.12 and 4.13 twice: once to remove the loads due to components fixed relative to the boat (such as the mast), and once to remove the loads that can change position (like the boom and sails).

To determine the mass \( m \) of the stationary components the frame was temporarily supported by the deck while the vertical load cells were disconnected.
and zeroed. When the load cells were reconnected to the frame, the new vertical load cell readings represented the weight of the frame and rig. The vertical load is given approximately by

\[ mg = 0.9948l_1 + 1.001l_2 + 0.9785l_5 \]  \hspace{1cm} (4.14)

(see the calibration matrix in table 4.2), which amounted to roughly 778 lbs. Omitting the other three load cell readings has a negligible effect on this calculation since the three vertical load cells contribute 99.98 percent of the vertical load in this loading condition.

In addition to the total frame weight, the weighing experiment gives us the longitudinal center of gravity. Assuming that the lateral center of gravity is zero, the last task in the calibration was the determination of the vertical center of gravity, which was found by inclining the sailboat. For each heel angle, we recorded the heeling moment acting on the dynamomter and then found the moment arm that best fit the data.

To make the correction for the components that move, we weigh each component and approximate its location for a given wind angle. For example, the position of the boom when sailing upwind is roughly 20 degrees inboard from the apparent wind. Also, the spinnaker pole is usually perpindicular to the apparent wind when sailing downwind.

In summary, the process of calculating the sail forces begins by reading the load cells. Then, the forces acting on the frame are calculated with the calibration matrix. Finally, the known weight of the both the fixed and movable rig components are subtracted from the measured dynamometer reading with
Bibliography


