DESIGN OF A PROPELLER FOR A HUMAN-POWERED BOAT

by

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Abstract

Garry Hoyt required a propeller for his human-powered boat, the Mallard. He wanted one that would give a high efficiency over a range of flow coefficients, but remain less massive than a traditional propeller. The propeller that we designed is a two-bladed propeller that when operating at 250 rpm with an input power of 130 W should provide 49 N of thrust, enough to propel the Mallard at a speed of 5 mph. It should accelerate to this speed relatively quickly, possibly stalling briefly at very low speeds. Its acceleration should still be comparable to that of the traditional propeller, though. We do not know whether the propeller will perform as it is expected to since we have not tested it yet. We have designed and built a slipping clutch which we will use in conjunction with a motor to test the propeller. In our tests we will measure the average velocity of the craft over several intervals, and thereby get an estimate of the time it takes to accelerate to the design speed. From these tests three possible conclusions may result. First, we may conclude that the design is useful as it is, and nothing more will need to be done. Alternatively, we may be able to show that this type of propeller is inappropriate for use in applications such as ours. Finally, we may decide that another iteration of the design is necessary to verify whether this type of propeller is useful.

Thesis Supervisor: Professor David Gordon Wilson
Title: Professor of Mechanical Engineering

Dedication

My sincerest appreciation is expressed to Garry Hoyt for making this all possible by needing a propeller for his boat, and for his patience and assistance as the design progressed. I would also like to thank Dave Wilson for allowing me to undertake this project and his guidance along the way.

Many thanks as well go to Mark Drela for supplying his propeller design software and his advice, to Theo Korakianitis for letting me use his computer, and to Don Chan for designing a test rig. Without the help of these three gentlemen this project would never have gotten very far.

Finally, I would like to wish a very special thank you to my parents and to the brothers of Kappa Sigma for their moral support.

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Chapter 1

INTRODUCTION

Garry Hoyt, a prominent yacht designer, recently designed several advanced human-powered boats. One of these is known as the Mallard, and it is intended for use as both a pleasure craft and a utility boat. For instance, this craft would be useful as a sailboat dinghy to shuttle from a beach or dock to a larger craft moored in deeper water. What Mr. Hoyt needed was a new propeller for this boat.

He had a three-bladed propeller designed by Gene Larrabee, but it gave little thrust until almost up to its design speed. This type of performance is unsuitable for his application as the Mallard will not necessarily be operated close to the design speed most of the time. Instead of this propeller, Mr. Hoyt has been using a traditional type of propeller which he fashioned 'by eye'. This propeller works well, but since it was just made by approximation, we felt that a better one could be designed. The objective of this thesis was to design such a propeller.

This design, however, was not to be of a traditional type. Instead, it was to be more of a compromise between the two existing three-bladed designs - one that would give a high efficiency over a range of flow coefficients, but remain less massive than the traditional variety. We also opted to design a two-bladed, rather than a three-bladed propeller. The advantage of this is that a two-bladed propeller can be aligned vertically and locked into place such that the blades are mostly hidden behind the skeg. Thus the blades would produce less drag when the boat was being towed, as in the case of a sailboat dinghy.

Chapter 2

Definition of Design Parameters

Before undertaking the actual design of the propeller, there were a number of parameters which had to be determined. Many of these were simply dictated by the average performance of a human, and some basic characteristics of the boat itself. To begin with, Garry Hoyt informed us that the design speed should be about five miles per hour (mph). The shape of the boat would make it difficult to propel it significantly faster than this. In fact, it would be undesirable to go faster than this from a human standpoint since five mph is a comfortable cruising speed for a pleasure craft. By pedalling harder, the operator could make the boat go a little faster, but 5 mph was agreed upon as an appropriate design speed.

The maximum diameter of the propeller is limited by the location of the shaft on the boat and the amount of clearance this leaves with the hull. This dimension was given to me as 17 inches in diameter. This was rounded down very slightly to 0.4 meters to insure proper clearance. The diameter of the hub was set at 0.06 meters, which is the approximate dimension of the two existing propeller hubs.

Two more parameters were determined from the human input standpoint. The rotational speed of the propeller was determined from a comfortable pedalling speed. Dave Wilson and I agreed that a comfortable pedalling speed for an average person would be abcut 60 revolutions per minute (rpm). The gear reduction in the boat's drive train is four to one, so the rotational speed of the propeller at the design condition would then be 240 rpm. We rounded this number up to 250 just so we could use rounder numbers. This rounding up has a small effect on the input values as it only raises the input speed to 62.5 rpm.

The power required to drive the boat is also constrained by human capabilities. According to Dave Wilson, a human can generally supply anywhere from 75 to 200 Watts for an extended period. The average person, however, would not be willing to pedal harder than about 150 Watts for much time, and this figure may be a little high. Thus we decided to aim for something which would require a little less than 150 Watts of input power.

Another parameter which needed defining was the lift coefficient at which the propeller blades stall. From their experience, Dave Wilson and Mark Drela estimated that this would be around a coefficient of 1.3. Later, however, Mark Drela indicated that the stall characteristics near the hub are not as important as near the tip. It is all right to have the hub stalling as long as the tip is not. He also said that there have been instances where people have measured lift coefficients near the hubs of similar types of propellers as high as 2.0 without them stalling. For the tip, though, we decided to aim for a stall limit of about 1.3.

This left only thrust to be defined. Thrust was a little more difficult to define, however. In ship design, the resistance of a ship at a given speed is the force required to tow the ship at that speed in smooth water.[1] Since thrust is the force required to push the ship at a certain speed, thrust and resistance have exactly the same value. It then remains to calculate the resistance of the Mallard. The two primary components of the total resistance are frictional resistance, due to the motion of the hull through a viscous fluid, and wave-making resistance, due to the energy that must be supplied continuously by the vessel to the wave system created on the surface of the water.[2]

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[1] Principles of Naval Architecture(New York, The Society of Naval Architects and Marine Engineers, 1967), p.228.

[2] Principles, op. cit., p.288.

The frictional resistance was calculated from the following equation found in D. W. Taylor's <u>The Speed and Power of Ships</u>;

$$R_F = f*S*V^n, \tag{2.1}$$

where R_F is the frictional resistance, S is the wetted surface area, V is the velocity of the vessel, and f and n are factors depending on the length and nature of the surface. Values for f and n were taken from tables in Taylor's book. I assumed that the surface of the Mallard is like that of a varnished surface since these are both smooth surfaces. From the tables I found f to be 0.0113, and n to be 1.825. I then calculated the wetted surface by taking measurements off of the lines drawing of the boat and taking an approximate integral of these values. This surface area turned out to be 40.8 square feet. From equation (2.1), then, the frictional resistance was calculated to be 6.72 pounds (lbs), or 29.9 Newtons (N).

The problem then was to somehow determine the wave-making resistance. Unfortunately, there is no way to calculate this. It can be found by making towing tests on the boat, but we had neither the time nor the facilities available to perform these tests. Taylor's book does have some general guidelines for what percentage of the total resistance is due to wave-making resistance for some types of boats. He indicates that a boat whose speed is very high for its length may have a wave-making portion exceeding 50 percent of the total. For slower vessels, it is generally more like 20 to 30 percent wave-making resistance. Since our design speed for the Mallard is relatively, but not extremely high, and in order to add some conservatism into our estimate, we approximated that the wave-making resistance accounted for about 40 percent of the total resistance. We were considering only wave-making and frictional resistance, so the total resistance was simply found by dividing the frictional resistance by 0.6 giving a value of 11.2 lbs. or 49 N. This gave us all of the parameters we needed to go on with the design.

Chapter 3

Design

3.1 Design Procedure

With all of the design parameters specified, I could then use Mark Drela's 'ROTOR' software to design the propeller. To come up with a design, I simply had to input the number of blades, the diameter of the propeller, the rotational speed, the boat speed, the thrust, the tip lift coefficient, and the hub lift coefficient. Notice that thrust and power do not both have to be specified . When one is specified, the program calculates the other. Since thrust is our critical coefficient, as shown in chapter 2, I specified it and checked the value of power that the program gives with the limit specified by the design conditions. The main variables on this process are the lift coefficients. By changing these values, different designs resulted. The objective, then, was to find the values which gave a design that fit the design conditions best. At the design speed, the only necessary check was to assure that the design did not require too much power at that speed.

The next step was to check the design at slower speeds to insure that it does not stall when starting up. Fortunately, Mark Drela's software has a feature which enables the user to analyze the performance of a design at off-design points. For these tests, though, thrust was no longer the critical parameter. Equation (2.1) shows that the thrust is directly proportional to the velocity of the vessel, so the thrust required to move slower is lower. A certain amount of extra thrust is also desired in order to provide some acceleration. To check off design points, then, I did not specify thrust. Instead I specified the input torque. We decided that when someone is starting the boat s/he would be willing to supply about as much torque at the slower speeds as at the design speed. The software calculates the required torque at the design speed, so I simply specified that value when checking the off-design points.

When running the program to determine off-design operating points, the only parameters which needed to be prescribed were the boat speed and the critical parameter, torque. The software calculates all of the other parameters for those conditions. The returned values from the software were then compared with the design conditions to see that they fit. It turned out that the only values that needed to be checked were those for the lift coefficients. Both power and rotational speed decreased with decreasing velocity, so their values were okay as long as they met the requirements at the design speed. With constant torque, thrust increases with decreasing speed. Since the required thrust decreases with decreasing speed, the higher values given from the program insure that there is a margin of extra thrust to supply acceleration. Thus I had to check only the lift coefficients to see that they did not exceed the stall limit.

This whole process was repeated for different lift coefficient values until a design was found that met the requirements at the design speed and just did not stall at slower speeds.

3.2 Design Results

The design that I initially came up with had the operating characteristics shown in Table 3-I. When I showed this design to Mark Drela, however, he pointed out that the blades on this design were far too wide for our purposes, a factor that I had initially overlooked. He showed that this problem could be solved by pushing

Table 3-I: Initial Design Operating Characteristics

speed (m/s)	tip lift	hub lift	thrust (N)	eff (%)	power (W)	rpm	torque (N*m)
2.2	0.3	0.1	49.0	81	132.88	250	5.08
1.0	0.72	0.44	61.53	66	92.82	174	5.1
0.5	1.08	1.05	72.97	43	84.52	158	5.1
0.1	1.22	1.3	7.01	28	2.48	47	0.5

closer to the stall limits at slower speeds, actually allowing the blades to stall near the hub at low speeds. With this in mind, we determined that the lift coefficients could be pushed up to 0.4 at the design speed. The operating characteristics of this final design are shown in Table 3-II The dimensions of this final design are given in Table 3-III along with an untwisted view of the blade shape in Figure 3-1. The angle refered to in Table 3-III is the angle from the plane of rotation to the bottom of the blade.

3.3 Blade Profiles

From his previous experience, Mark Drela was able to recommend an approriate blade profile (or foil section). The section he selected is shown in Figure 3-2, and Table 3-IV gives the data for calculating the thickness at various points along the section. This shape is the generalized shape of a cross-section of the blade at any point along its length. This shape has an almost flat bottom which for simplicity of manufacturing can be made flat with negligible effect.

In Table 3-IV, T represents the length of the chord at that section. The X/T values are the stations along the length of the chord at which the thickness, Y is to be determined. The station X/T = 1.0 is at the trailing edge of the blade, and X/T = 0 is at the leading edge. To find the actual distance from the leading edge for a particular station, simply multiply the percentage X/T by the chord length, T.

The thicknesses of the blade are measured from an imaginary line passing through the middle of the blade. This line, which can be seen in Figure 3-2 is at an angle of 1.56 degrees from the bottom of the blade. The two thicknesses are then measured above and below this line. YO/T is measured above the line, while YU/T is measured below the line. Again, the thickness is proportional to the length of the

Table 3-II: Final Design Operating Characteristics

speed (m/s)	tip lift	hub lift	thrust (N)	eff (%)	power (W)	rpm	torque (N*m)
2.2	0.4	0.4	49.0	83	129.75	250	4.95
1.0	0.95	1.36	65.83	65	100.0	192	5.0
0.5	1.29	1.5	63.62	34	91.19	174	5.0

Table 3-III: Blade Dimensions

station no.	radius (m)	chord (cm)	angle (deg)
1	0.04	8.34	62.04
2	0.06	8.94	52.55
3	0.08	9.10	44.71
4	0.10	8.74	38.33
5	0.12	8.00	33.18
6	0.14	6.98	28.98
7	0.16	5.69	25.53
8	0.18	4.04	22.67
9	0.19	2.93	21.42
10	0.20	1.00	20.25

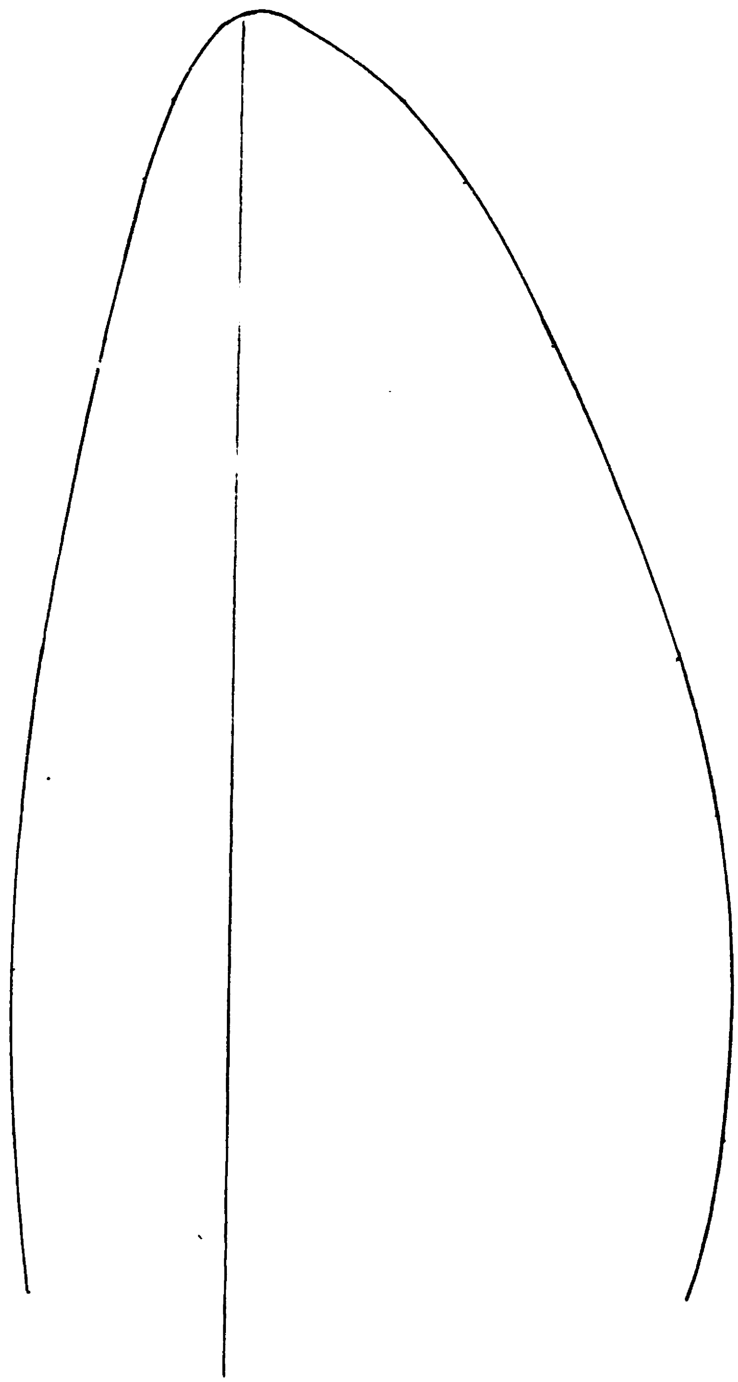

Figure 3-1: Untwisted View of Propeller Blade

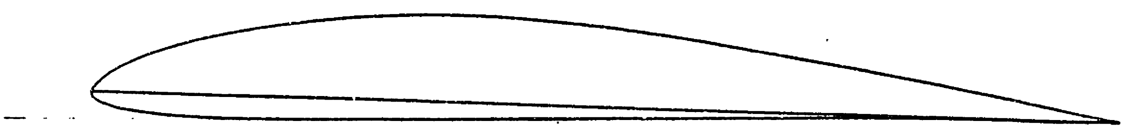

Figure 3-2: Foil Section

Table 3-IV: Foil Section Thickness Coefficients

NR	X/T	YU/T	YU/T
1	1.00000	.00000	.00000
2	.99893	.00016	.00004
3	.99572	.00065	.00016
4	.99039	.00153	.00027
5	.98296	.00288	.00033
6	.97347	.00471	.00030
7	.96194	.00697	.00019
8	.94844	.00958	.00002
9	.93301	.01247	-.00017
10	.91573	.01563	-.00036
11	.89668	.01895	-.00057
12	.87592	.02254	-.00081
13	.85355	.02635	-.00110
14	.82967	.03039	-.00148
15	.80438	.03462	-.00193
16	.77779	.03901	-.00247
17	.75000	.04355	-.00309
18	.72114	.04818	-.00391
19	.69134	.05285	-.00461
20	.66072	.05753	-.00549
21	.62941	.06215	-.00647
22	.59755	.06664	-.00752
23	.56526	.07090	-.00864
24	.53270	.07480	-.00983
25	.50000	.07821	-.01106
26	.46730	.08100	-.01233
27	.43474	.08310	-.01362
28	.40245	.08445	-.01491
29	.37059	.08500	-.01618
30	.33928	.08475	-.01740
31	.30866	.08369	-.01856
32	.27986	.08187	-.01962
33	.25000	.07938	-.02056
34	.22221	.07631	-.02135
35	.19562	.07270	-.02196
36	.17033	.06851	-.02236
37	.14645	.06408	-.02255
38	.12408	.05918	-.02246
39	.10332	.05393	-.02208
40	.08427	.04837	-.02145
41	.06699	.04264	-.02045
42	.05156	.03670	-.01901
43	.03806	.03039	-.01753
44	.02653	.02462	-.01569
45	.01704	.01871	-.01277
46	.00961	.01210	-.01024
47	.00428	.00666	-.00750
48	.00107	.00390	-.00442
49	.00000	.00000	.00000

chord at that place, so to get the actual thickness the given numbers must be multiplied by T. Since the chord starts to get smaller as it gets closer to the hub, the thickness of the blades would also start getting thinner. For stress purposes this would be undesirable. To solve this problem, Mark Drela said that inside 40 percent of the radius we could multiply the thickness coefficients by a constant value to make it some percentage thicker. I chose to select factors such that the thicknesses would continue to get larger as the sections approach the hub. Thus I determined that station number one, the closest station to the hub, should be made 40 percent thicker by multiplying the Y/T values by 1.4, and station number 2 should be made 20 percent thicker by multiplying the coefficients by 1.2.

The last problem in defining the shape of the blades was to show how the profiles of the blade are oriented along the radius of the blade. Figure 3-3 shows a computer plot of a blade exactly like the one I designed, except this was for a three-bladed design. The only difference between this blade and the two-bladed version is that the two-bladed version is 1.5 times wider. The axis in this figure is simply a straight line from the center of the hub to the furthest tip of the propeller. By measuring along this length I found that thirty percent of the chord length lies toward the leading edge of this line. That is to say that the 31st stations on the profiles always line up along a radius from the center. It just so happens that this point also coincides with the thickest part of the blade.

All of this information together was enough to completely define the shape of the propeller blades.

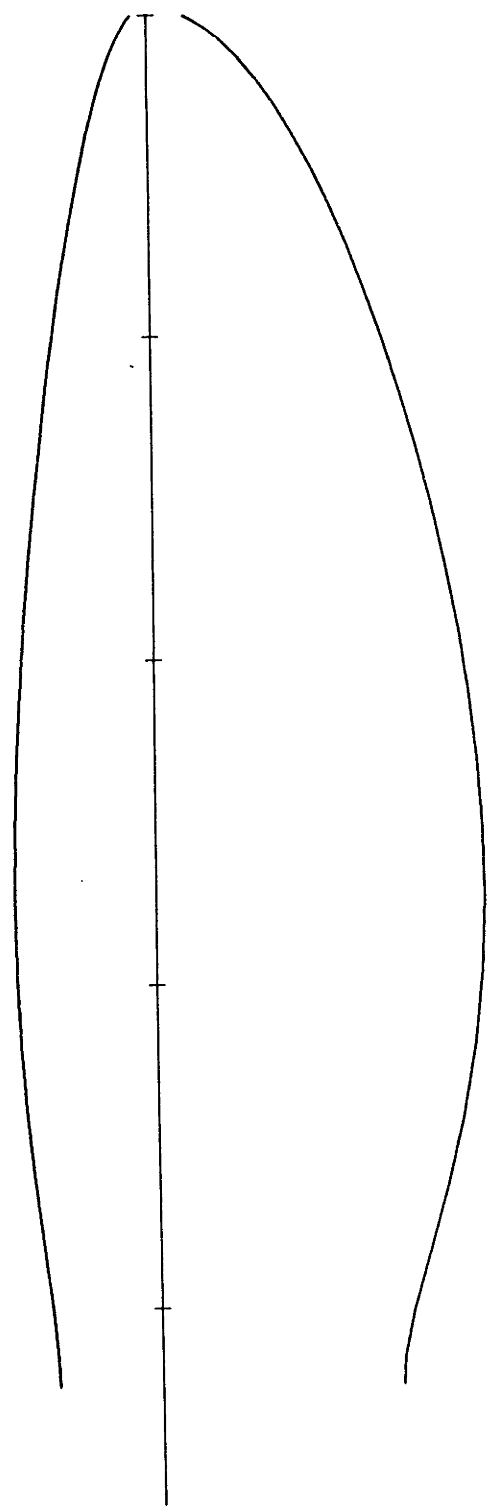

Figure 3-3: Untwisted View of Three-Bladed Propeller

Chapter 4

Construction and Testing

4.1 Construction

In order to have the propeller made, I submitted the information and descriptions in the previous chapter to Garry Hoyt, who, in turn, submitted the same information to a company he deals with that makes propellers. I also investigated the possibility of trying to construct it on the numerically controlled machines in the manufacturing labs here at MIT. Due to the time constraints and the complexity of the shape, however, this was found to be impossible.

4.2 Testing

The next aspect of this design was to come up with an appropriate testing procedure whereby we could test the performance of my design as well as the existing ones and compare them to see which is better. There was a wide variety of different ways we could have approached this problem. We could have tried to get some time in the propeller tunnels here at MIT and measured the flow characteristics of the water behind the propeller. This would have been much too difficult just to get the time in the tunnel, so we decided to look for other options. Garry Hoyt volunteered to let us use his Mallards, so we knew we could just attach the propellers to the boat and test them by pedalling them around in some way. Having a person pedal the boat around would be a very unscientific way of testing, though, because there is no way of making sure that the person was pedalling the same way every time. Thus we decided to try to come up with some way of attaching a motor to the drive train, assuring consistent input characteristics.

Don Chan and Dave Wilson worked on designing a rigging mechanism to hook a motor up to the propeller shaft. Their design, which is shown in Figure 4-1, consists basically of a slipping clutch and a gear reduction. The reduction serves the opposite purpose from the drive train existing in the boat. The existing drive train is a step up, increasing an input speed by four times. The reduction on the test rig, however, is a step down, reducing an input speed by a factor of about three. Note that the drive train must be removed from the boat in order to hook up the motor and the test rig. The slipping clutch keeps the input torque to the propeller shaft constant. As seen in Figure 4-1, there are two large disks with a frictional material between them like a brake pad. The frictional material, which we chose to be leather, is only attached to one of the disks. If the motor tries to apply too much torque, or if some back torque is introduced through the propeller, the leather will slip along the other disk, thus setting a maximum torque for the system. This whole device, then, can be attached to the existing shaft to provide a constant torque source for testing.

At publishing time, the actual tests had yet to be performed. We have lain out what the tests are going to be, though. We plan to take the test rig down to Newport, R.I. where Garry Hoyt keeps his boats and perform the tests there. The tests will involve first setting up several fixed markers along the shore. They will probably be set up at about 20-30 foot intervals. The first mark will be the starting line. We will line up the bow of the boat with the starting line and then start the motor. By clocking how much time it takes to pass each mark, we can determine the average velocity at which the boat was travelling for each segment of the test run. We will repeat this test several times for each of the three propellers, possibly varying the input torque for different tests. The data from these tests can be plotted to make curves of average velocity versus time, and these curves will

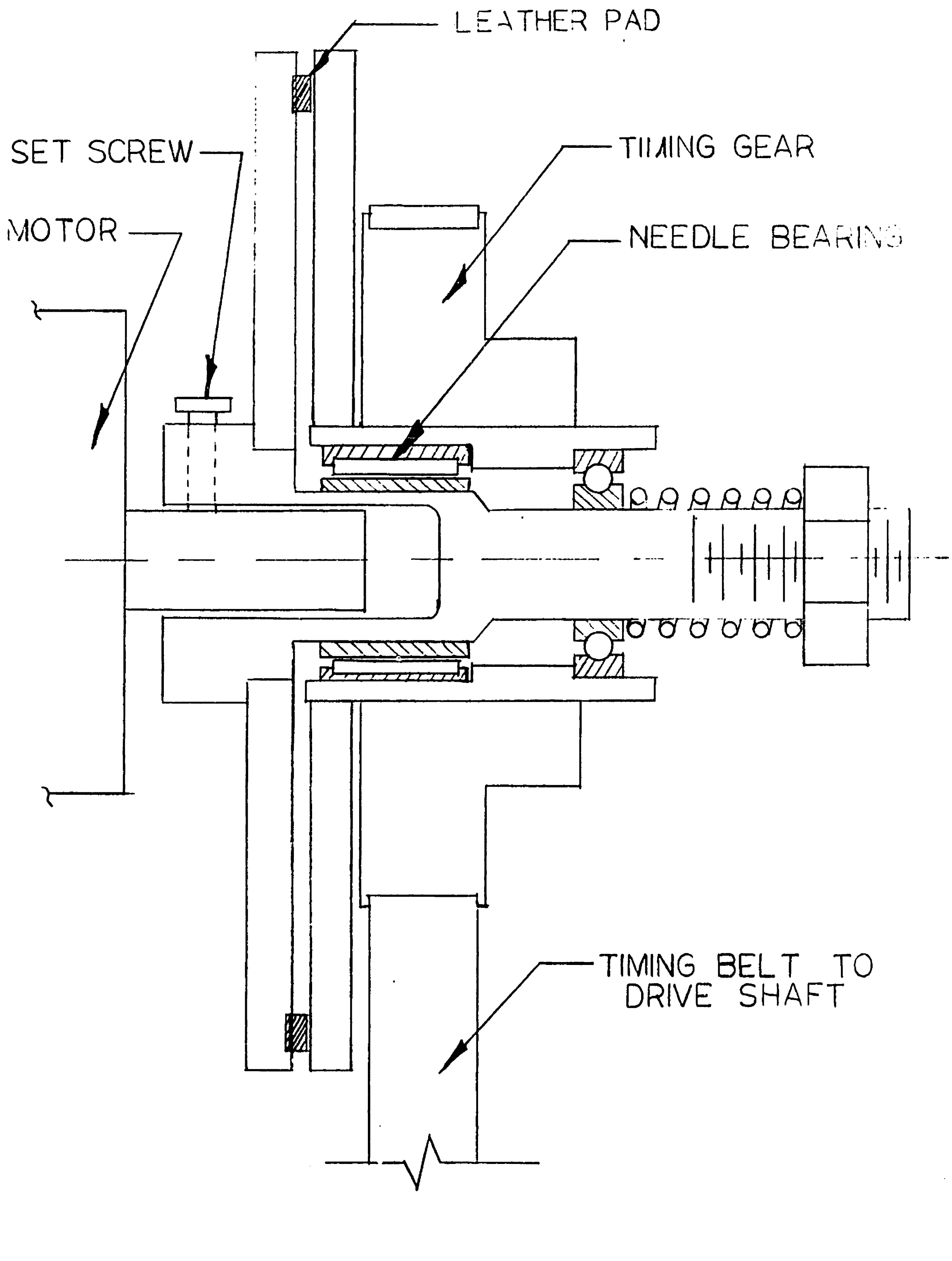

Figure 4-1: Test Rig

indicate which propeller accelerates faster with a constant input torque. Ideally, we would like for my design to accelerate at least as fast, if not faster than, the traditional style propeller, but we will not know if this is so until we actually run the tests.

Chapter 5

Conclusions and Recommendations

The result of this design is a propeller whose performance should be at least as good as that of a traditional style propeller while causing less drag when not in use. Unfortunately, the tests have not yet been run, so we do not know exactly how the propeller is going to perform. Theoretically, what should happen when someone tries to use the boat with this propeller is that s/he will start off by applying a constant torque equal to the torque to be applied at the cruising speed. The propeller should accelerate the boat up to speed relatively quickly, without stalling at any point. Realistically, I expect that the propeller may stall at very low speeds, but once a little speed is gained it should not stall anymore. The time it takes to accelerate needs to be at least as fast as the time for a traditional type of propeller which is currently being used. If the acceleration time is significantly slower for my design, it would not be reasonable to recommend using it instead of the old design. We know that my design will induce less drag just by examining the area of the blades. My design has much narrower blades, so it will surely induce less drag.

One major problem that could occur is that the propeller could stall too much, causing an increase in the acceleration time. If this occurs, I would recommend trying another iteration of the design. We would have to go back to the ROTOR software and design another propeller, this time with lower lift coefficients at the design speed. We would also have to make sure that we lowered the value of what we were considering to be the lift coefficient at which stall occurs. To make this second iteration easier, it would be best to find some way of measuring the lift coefficient at which the first design was stalling. I do not know how this is done, but I am confident that Mark Drela probably knows how if we ever really need to know.

Another possibility is that the propeller does not stall excessively, but still has what we judge to be too long of an acceleration time. In this case we will have proven that this type of propeller is inappropriate for use on a boat. There is also some possibility that the propeller works just as we had planned it to, so nothing more would need to be done. Hopefully the tests, which we will be running soon, will yield some useful data so that we can draw some concrete conclusions.

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