Florida Institute of Technology

Department of Ocean Engineering

The Design of a 44 Foot Open Sports-Fishermen

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2.1 Introduction

This paper will present the design of a forty-four foot open fishermen. The design requires that the vessel cruise at 30 knots. Carry a load of six people, including crew, and a cargo load of 1000 lbs. The vessel must be able to run at cruise power for a minimum of 24 hours.

This paper will include a deck layout and longitudinal section of the vessel to be designed. The overall length, length on chine, maximum beam on chine and the speed length ratio. A first weight estimate and an expected planing performance analysis will be performed as pertaining to the dimensions of the vessel under development. Also included in this report will be a second weight estimate, hydrostatic and stability calculations, and LCG vs. LBC comparison. Accompanying this information will be a resistance calculation to ensure proper horsepower to propel the vessel at its design speed. A list of similar vessels will also be presented to be used as comparative reference.

This paper will also present the design of a propeller and rudder for use if it is deemed necessary to modify this vessel design for standard inboard power. This modification will only be necessary if the design requirements can not be made with a sterndrive propulsion system.
2.2 Vessel Layout and First Design

This vessel was designed to test the possibility of placing two sterndrive engines in a open-fisherman type vessel. The overall length(Loa), 44 ft, was chosen as this is a common midsize sportsfishing vessel and the market demand for such a craft should be high. As it was desired that this vessel behave in a seaway in a similar manner as other vessels which have come before it the other hull parameters will be representative of similar vessels. The design beam is currently 15 ft. The deadrise angle at the stern will be on the order of 4 degrees. This deadrise angle is considered very shallow by today’s standards. This shallow angle was necessary, however, to accommodate the sterndrive engine arrangement without having to drastically elevate the decksole. This shallow deadrise will also increase the planning performance of the vessel. It should be noted that by using this shallow angle this vessel may be more susceptible to “slamming” in a sea way than a vessel of comparable size. As for interior layout the vessel was designed to optimize space while still allowing comfort and convenience for the vessels operators and guest. Relating speed to length by the speed/length ratio a value of 4.522 was computed. When comparing this value to charts presented in class this vessel easily falls into the planing craft region.

This vessel is designed in the same manner as craft its size. However, it is hoped that a sterndrive power system may be implemented in this vessel. Offshore powerboats (racing boats) of this size use this power system, yet the recreational fishing boat uses mostly inboard power for a vessel this size. If sterndrive power may be utilized the living space below decks will be increased and the available fuel capacity should also be increased. It is hoped that this arrangement will be made possible by shifting generator, fuel load, water tanks, holding tank, and other equipment forward to counter the aft shift in the center of gravity resulting from shifting the engines aft. If it becomes apparent that this is not possible we reserve the right to use inboard power and remove the aft stateroom.

To see a deck profile and longitudinal profile please see figures 1 and 2 in appendix A, respectively.
2.3 First Weight Estimate and Planing Performance

As the first weight estimate is the determining factor for the planing performance these two topics will be included in the same section. They will however each be computed and discussed separately.

First Weight Estimate

The first weight estimate was performed using two separate methods, and then comparing the results with previous designs. The first method used was the Johansen method. This method relies on a simple equation given as:

\[ \text{Weight}(W) = 14500 - 1678.69 \times (L_{oa}) + 58.53 \times (L_{oa})^2 - 0.2667 \times (L_{oa})^3 \]

Using this equation a weight of 31223 lbs was determined. After determining the weight by the Johansen formula, the Calkin method was applied to the vessels design. This second method was used as a “check” on the first method. As these are theoretical estimates it is necessary to check there accuracy. The formula used was for cruisers and is given by:

\[ W = 2.933 \log(L_{oa}) - 0.13 \]

This formula produced a weight of 30708 lbs. This weight appears right on target when compared against similar vessels. As all of these methods appear to give approximately the same weight it can be assumed this is an accurate value for the first weight estimate.

Planing Performance Expectations

The expected planing performance was derived using chapter 77 (Preliminary Design of a Motorboat) page 829. This text states that for good planing performance the displacement(\(\Delta\))/length(L) quotient, given below, should not exceed 200.

\[ \Delta/ L \text{ quotient} = \Delta/(0.01L)^3 \]

Applying this formula, with displacement in long tons, a value of 293 was determined. It should be noted that this value is above the 200 limit for good planing performance. However, it has been observed that other vessels of similar form are still able to operate as planing craft. It is assumed that this vessel will also operate as a planing craft, yet its optimum performance may be hampered due to weight.
2.4 Second Weight Estimate

A second weight estimate was performed to try to more accurately predict the characteristics of the vessel. This second weight estimate simply added the weights of the major components of the vessel, such as the hull, fuel, engines, etc. The hull weight was calculated by summing the structural weight and the fittings weight. The weight of the fittings was given as 0.3 times the structural weight. The equation for the structural weight was given as:

\[ W_s = -0.0919(\text{Loa})^3 + 13.335(\text{Loa})^2 - 156.71(\text{Loa}) \]

This resulted in a structural weight of 11,093 lbs, and a fittings weight of 3,328 lbs. The other weights relied more on the mission design of the vessel. The weight of people on board was calculated assuming the average person weighs 170 lbs. multiplying this by six persons a weight of 1020 lbs. The engine weight was obtained from the manufactures relating to the amount of horsepower required for this vessel. The horsepower required is 882 brake horsepower. This number was increased to 1102 in order to accommodate extra weight. As such a fuel weight of 11,722 lbs was determined. Cargo weight was predetermined at 1000 lbs. And water weight was calculated at 1440 lbs.

When all of these weights are summed a total weight of the vessel is determined. For this report the current weight of the vessel is 34,658 lbs. This value is slightly higher than the first weight estimate. Yet the discrepancy is only approximately 400 lbs. This value certainly seems acceptable.

To see the actual weight breakdown please refer to figure 2 in appendix A.
2.5 Determination of Water line Characteristics, Hydrostatic and Stability Calculations

To determine the waterline characteristics it is first necessary to determine the draft of the vessel at rest. With out this calculation it would not be possible to determine the exact waterline position. The draft for this vessel was determined by using the displacement and applying it to the hydrostatic properties graph in Appendix B. By using this graph it was possible to determine the exact draft of 2.4 ft for the current displacement of the vessel. Once the draft was known it was possible to determine the waterline length and where the waterline intersected the chine line. This was performed by sketching a waterline at the proper draft on the scale sketch of the vessel in appendix A. By this it was seen that the waterline length was 39.4 ft. It is also seen that the chine line intersects the waterline between sections three and four.

The projected area on chine and the location of the center of the chine area \(LCA_p\) was also determined in this section. Both of these parameters were determined using the Simpson rule. These calculations may be seen in appendix C. By these calculations the area on chine was given as 505 ft\(^2\). Likewise the \(LCA_p\) was given as 25 ft from the bow.

It should be noted that all of the hydrostatic and stability calculations were taken either directly or indirectly from the output of the GHS program (Appendix B). As the coordinates for the hull of this vessel were entered without the superstructure of the vessel it is important to realize that these results may not be accurate. As such the expected stability characteristics for this vessel may be faulty. This can only be remedied by starting over in GHS and adding the superstructure to the vessel. This variation should be minimal due to the limited superstructure, primarily a wind screen. As the GHS program has crashed this continued analysis will not be available at this time.
2.6 LCG vs. LCB

In order to make this comparison it was first necessary to determine both the longitudinal center of gravity (LCG), and the longitudinal position of the center of Buoyancy (LCB).

The LCB was simply determined using the hydrostatic properties graph in Appendix B. Using this graph a location of LCB was determined to be approximately six feet aft of amidships. The determination of the LCG was slightly more involved.

The LCG was determined by evaluating all the weights and determining how they reacted in relation to the stern of the vessel. As such LCG was given by:

\[
LCG = \frac{\sum w(x)}{\sum w}
\]

Where:
- \( w \) = the weight of any object in the vessel.
- \( x \) = the distance of that object from the stern of the vessel.

Using this method the LCG was determined to be 16 feet forward of the stern. In comparing these two values it is evident that the LCB and LCG act at exactly the same point on this vessel. It seems as though this position on the vessel is rather far aft for the LCB to be acting. However, as both LCB and LCG act at the same point it can only be assumed that the vessel is in level trim.

Number here? LCG? LCB?
2.7 Stability Calculation and Evaluation

Two types of stability will be covered in this section, static and dynamic. Static stability will be discussed using a GZ-\(\phi\) curve. Dynamic stability will be determined using the Blount-Codega formula.

Static Stability

Static stability would normally be calculated using the GHS program. Unfortunately time did not allow the use of this program for this section. As such the cross curves of stability were created using an assumed position of center of gravity acting on the keel. Using these cross curves of stability and a correction for the correct value of KG a GZ-\(\phi\) curve was created using excel. This plot may be seen below.

![Gz - phi curve](image)

It can be easily been seen from this chart that static stability can not be determined at this time. Unfortunately the curve in the chart never passes the zero mark on the righting arm. As such it appears that this vessel will never turn over. This is obviously not the case. It is believed that this occurred due to the lack of any type of superstructure on the vessel when it was entered into GHS. The program ran as though the vessel was a bare hull. As such there was no weight in the upper portion of the design to cause the vessel to want to over turn.

This problem will be corrected by the next report. It was not understood that not having a superstructure in the program would have this pronounced effect.

Dynamic Stability

The dynamic stability was evaluated by using the Blount-Codega formula. This formula is given as:

\[
\frac{A_p}{\sqrt{V}} < \frac{39A_p}{B_{wx}} + 4.52
\]
Using this relation it was determined that $7.6 < 5.69$. As this statement is mathematically incorrect it is apparent that some modification of the chine area is required so that proper dynamic stability is achieved.
2.8 Resistance Calculations

To better understand the horsepower necessary to propel this vessel at the desired 30 knots it was necessary to perform resistance calculations for the vessel. These calculations take into effect the resistance of the water on the hull, wind drag, and resistance due to wake effects. To perform resistance calculations it is necessary to determine several parameters. To see all of these parameters and the values associated with them please see Appendix D. This section will simply show the equations used and the output from these equations.

The first step in calculating the resistance of a vessel hull is to determine what is known as the bare hull resistance \( R_{BH} \). This takes into account the frictional and displacement effects on hull resistance. It should be noted that this step only takes into account the bare hull, i.e. no appendages attached to the hull. Calculating this bare hull resistance was a simple matter of applying the following equation:

\[
R_{BH} = \Delta \tan \tau + \frac{\rho \frac{V^2}{2} C_{L} A B_{nx}}{\cos \beta \cos \tau}
\]

where \( \tau \) is the angle of attack.

It should be noted that the first portion of this equation gives the residual resistance, and the second gives the frictional resistance.

After determining the residual resistance it is necessary to determine a resistance for appendages \( R_{app} \). This is assumed to be related to a the bare hull resistance and may be determined mathematically by:

\[
R_{app} = R_{BH} \left( \frac{1}{\eta_{A}} - 1 \right)
\]

Where \( \eta_{A} \) is given as a function of the volumetric Froude number.

Wind resistance was also calculated using a simple mathematical formula. This was accomplished using a relation that the wind resistance is equal to the area presented to the wind times the velocity of the wind squared times a raw value of 0.004. The frontal area for this vessel was calculated by examining the largest sectional area of the vessel. It was assumed that since this vessels only superstructure was the wind screen that the frontal area could be determined using the windsreen area and the largest section area of the vessel. As this calculation only requires the cross section this should be an accurate assumption. This section area was determined using the GHS data presented in Appendix B and some simple geometry.

To calculate the final resistance for this vessel the bare hull resistance, appendage resistance, and wind resistance were simply summed for the respective velocities. It should be noted that for this final resistance it was assumed that the only velocity was due to the vessels speed. It should also be noted here that the resistance due to wave interaction was neglected in this calculation. It was felt that this resistance would vary so drastically with changing wave conditions that by using this resistance power calculations
would be almost impossible. Also the way in which a captain operates a vessel changes in rough water. As such the resistance calculations for this vessel are simply for smooth water. Table 1 below shows the three different resistance values as well as the total resistance for the vessel at several different design speeds.

Table 1: Velocity and Corresponding Resistance

<table>
<thead>
<tr>
<th>V (knots)</th>
<th>R_{th} (lb)</th>
<th>R_{app} (lb)</th>
<th>R_{wind} (lb)</th>
<th>R_{total} (lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>3219.342</td>
<td>177.8143</td>
<td>51.832</td>
<td>3448.988</td>
</tr>
<tr>
<td>13</td>
<td>3278.83</td>
<td>192.9393</td>
<td>87.59608</td>
<td>3559.365</td>
</tr>
<tr>
<td>16</td>
<td>4430.093</td>
<td>280.8535</td>
<td>132.6899</td>
<td>4843.637</td>
</tr>
<tr>
<td>19</td>
<td>4250.686</td>
<td>292.8362</td>
<td>187.1135</td>
<td>4730.636</td>
</tr>
<tr>
<td>22</td>
<td>5050.292</td>
<td>380.4296</td>
<td>250.8669</td>
<td>5681.588</td>
</tr>
<tr>
<td>25</td>
<td>4850.912</td>
<td>401.204</td>
<td>323.95</td>
<td>5576.066</td>
</tr>
<tr>
<td>28</td>
<td>4683.172</td>
<td>426.2977</td>
<td>406.3629</td>
<td>5515.832</td>
</tr>
<tr>
<td>31</td>
<td>4854.006</td>
<td>486.809</td>
<td>498.1055</td>
<td>5838.92</td>
</tr>
<tr>
<td>34</td>
<td>4600.334</td>
<td>508.3126</td>
<td>599.1779</td>
<td>5707.825</td>
</tr>
</tbody>
</table>

In calculating the resistance the running trim angle had to be calculated. This calculation for running trim reached as high as 20 degrees and then settled at about 10 degrees. This high angle of running trim is probably due to the aft location of the LCB. There was concern that this may be a problem with this design from the beginning. It is hoped that the addition of trim tabs will reduce the running angle. However, adding trim tabs will increase the resistance and thus the horsepower necessary to propel the vessel at the desired velocity.
2.9 Propeller Design

As this vessel is designed primarily for a stern drive configuration by Volvo it is probably not necessary to design a propeller. However, as it is still possible that this vessel may have to be retrofitted as an inboard vessel it is prudent to have this design already performed.

The propeller design in directly related to the horsepower needed to propel the vessel at its design speed. This horsepower is also directly related to the total resistance of the vessel as the horsepower must equal resistance to maintain a desired speed. As such the effective power is related by the resistance times the velocity divided by 550. It should be noted that 550 is a conversion factor for horsepower. This is the power that is needed to be delivered at the propeller.

This propeller design was based on a large part on design charts which are provided in Appendix E, as such only values for certain parameters will be given. It is important to realize that this vessel is designed for twin screws. As such each propeller will produce half the load necessary to propel the vessel. To begin this design it is first necessary to determine the thrust needed to be produced by each propeller. This was done using half the resistance and dividing by the wake fraction. Performing this gives a $T_{1/2} = 3102$ lb. After determining the thrust component then the blade area ratio (BAR) was calculated. In order to avoid cavitation the BAR was calculated using the following equation:

$$\text{BAR} = \frac{4T_{1/2}}{k(p_e - p_v)(1.067 - 0.229 \frac{P}{D} \pi D^2)}$$

This equation was then solved in terms of $k$ and $D$. By doing this it was possible to determine the propeller diameter that would allow for a certain amount of cavitation. This amount of cavitation was introduced by the $k$ value. The three values for $k$ used and there corresponding percentages of cavitation are given in table 2 below.

### Table 2: k and percent cavitation

<table>
<thead>
<tr>
<th>K</th>
<th>% cavitation</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.36</td>
<td>2</td>
</tr>
<tr>
<td>0.55</td>
<td>10</td>
</tr>
<tr>
<td>0.7</td>
<td>15</td>
</tr>
</tbody>
</table>

By using this method the propeller diameter was determined to be two feet. This in turn corresponded to 10% cavitation. Using these two parameters a BAR value of 1.11 was determined. Then finding a value for $K_T/J$ these two parameters could be applied to the design charts in Appendix E. From this chart a new $P/D$ value could be determined for optimum efficiency. This $P/D$ value was then placed in the BAR equation and the process repeated. This process is repeated until the values for $P/D$ agree. At this point it is necessary to determine the value for $\eta_m$. This value can be taken from the same charts as the $P/D$ values. Then using the effective power previously determined and dividing by
\( \eta_b, \eta_m, \) and \( \eta_R, \) the delivered power may be determined. It should be noted that for these calculations \( \eta_R \) was given as 1 and \( \eta_h \) was simply given as the wake fraction divided by the thrust fraction. Using this delivered power and dividing by the gearing efficiency the brake horsepower was determined. It should be noted that for this paper the gearing efficiency used was 90\%. This is due to the complex gearing shown in figure 4 in Appendix A. The brake horsepower determined was 989 horsepower per engine. As this brake horsepower is similar to the previously determined value of 1102 brake horsepower it is felt that this value is accurate and should be sufficient to achieve the design requirements of this project.

\[
\frac{D}{D} = 7
\]

\[
h = 7
\]
2.10 Rudder Design

As with the propeller design the rudder design is an iterative process dealing with
design charts. It should also be noted here that this rudder design should not have to be
applied to this vessel. As the vessel is designed for a sterndrive system the drives will turn
and act as the rudder for the vessel. As such this rudder design will be implemented only
if it becomes apparent that the sterndrive system will not work according to design. All of
the charts used may be seen in Appendix F. To begin this process the rudder area had to
be determined. This was performed using a chart and inputting the design waterline
length. From this chart a rudder area of 0.3 meters was determined. At this point in the
design it was also necessary to determine the rudders aspect ratio. For this an assumed
value of $AR = 2$ was used. Using the rudder area and the aspect ratio the chord and span
length could be determined. These values are chord length equal to 0.39 meters and span
equal to 0.78 meters. The thickness of the rudder could also be determined. As the only
charts available for this design were for a thickness of 15% of the chord length the
thickness of this rudder was determined to be 58.5 mm.

Although the aspect ratio was assumed to be 2 this is not the actual effective
aspect ratio. Due to the proximity of the hull to the rudder the rudder acts as though it is
bigger than it actually is. As such the effective aspect ratio must be determined. Again
design charts are utilized for this determination. First the gap distance from the hull to the
rudder is determined. For this design this distance is given as 10 mm. This distance in
then divided by the chord distance. The number received is then applied to the charts to
obtain a value for k. This k value is then multiplied to the previous aspect ratio. This
gives the first iteration for the effective aspect ratio. From here the effective aspect ratio
must be determined from the stall angle. In this case an initial stall angle is assumed (25
degrees). This angle is applied to the chart and a respective k value is obtained. From
here the process is continued finding an angle and determining a k value to determine the
effective aspect ratio. This process is continued until the effective aspect ratio begins to
become the same. At this point the effective aspect ratio is determined. For this design
the effective aspect ratio is 2.8.

Using the effective aspect ratio and design charts it is then possible to determine
the coefficients of lift and drag for the rudder. From these the normal force acting on the
rudder and residual force may be determined. For this vessel these forces are; normal
force = 66.24 kN, residual force = 69.26 kN. Using these forces it is necessary to
determine the size of the rudder stock needed. Again using the design charts it is possible
to determine where the center of pressure acts on the rudder at zero degrees and at the
stall angle. From this observation the distance aft of the stock for the center of pressure at
stall is 0.0234 meters, and it acts at 0.38 meters down the rudder. These forces and
positions were then used to determine the moments and torque acting within the system.
Torque was simply the normal force times the distance of to the action point for pressure
at stall. And the moment was the distance down the rudder times the residual force.
Using these two values the effective moment was calculated using the following:

$$BM_E = \frac{M}{2} + \frac{1}{2} \sqrt{m^2 + T^2}$$
Using the results of this equation the diameter of the stock could be determined. The material used in the stock is mild steel with half the yield stress equal to $130 \times 10^6$. The equation used to determine the stock diameter is given as:

$$D = \sqrt[3]{\frac{BM_e(32)}{\pi\left(\frac{\sigma_{\text{yield}}}{2}\right)}}$$

Using this equation yield a stock diameter of 12.79 mm. This seems to be rather small for a rudder for a vessel of this size. However this is the value obtained through the analysis.
3.0 List of Similar Boats Used for Comparison

This list of comparable boats was compiled from many different sources. These sources being magazines and the Internet primarily. As this information may be obtained from many different sources it will be considered common knowledge and will not be included in the references section of this paper.

For ease of comparison these vessels have been placed in table format. This allows the user the ease of comparing one item easily without having to read about all aspects of each vessel.

<table>
<thead>
<tr>
<th>Manufacture</th>
<th>Model</th>
<th>L&lt;sub&gt;oa&lt;/sub&gt;</th>
<th>Beam</th>
<th>Draft</th>
<th>Approx. Δ(tlb.)</th>
<th>Fuel (gallons)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Luhrs</td>
<td>40 open</td>
<td>40'10&quot;</td>
<td>14'11&quot;</td>
<td>3'7&quot;</td>
<td>30,000</td>
<td>563</td>
</tr>
<tr>
<td>Viking</td>
<td>Open Sportfish/Express</td>
<td>43'</td>
<td>15'3&quot;</td>
<td>4'3&quot;</td>
<td>34,500</td>
<td>600</td>
</tr>
<tr>
<td>Pursuit</td>
<td>3400 Express Fisherman</td>
<td>36'4&quot;</td>
<td>12'9&quot;</td>
<td>2'8&quot;</td>
<td>14,000</td>
<td>350</td>
</tr>
<tr>
<td>Sea Ray</td>
<td>Sundancer 380</td>
<td>38'</td>
<td>13'</td>
<td>2'8&quot;</td>
<td>18,300</td>
<td>275</td>
</tr>
<tr>
<td>Sea Ray</td>
<td>42 Aft Cabin</td>
<td>45'3&quot;</td>
<td>14'3&quot;</td>
<td>37&quot;</td>
<td>27,000</td>
<td>350</td>
</tr>
<tr>
<td>Sea Ray</td>
<td>45 Express bridge</td>
<td>45'6&quot;</td>
<td>14'8&quot;</td>
<td>41&quot;</td>
<td>29,500</td>
<td>400</td>
</tr>
<tr>
<td>Silvertown</td>
<td>37 Convertible</td>
<td>41'3&quot;</td>
<td>14'</td>
<td>3'7&quot;</td>
<td>21,852</td>
<td>374</td>
</tr>
<tr>
<td>Cabo</td>
<td>45 Express</td>
<td>45'1&quot;</td>
<td>15'6&quot;</td>
<td>4'</td>
<td>34,800</td>
<td>850</td>
</tr>
<tr>
<td>Wellcraft</td>
<td>45 Excaliber</td>
<td>44'6&quot;</td>
<td>11'8&quot;</td>
<td>39&quot;</td>
<td>15,000</td>
<td>274</td>
</tr>
<tr>
<td>Wellcraft</td>
<td>40 Coastal</td>
<td>44'5&quot;</td>
<td>14'6&quot;</td>
<td>4'1&quot;</td>
<td>22,000</td>
<td>469</td>
</tr>
</tbody>
</table>

It should be noted that this list is representative of only a few vessels of common size. There are many more vessels that fall into this size range which are not listed here. As this project progresses it is certain that more vessels will be added to this list for comparison. It should also be noted that these vessels contain both sterndrive arrangements and conventional inboard propulsion. It was noted in the research of these similar vessels that the sterndrive application was used only in high speed cruiser type vessels. Most of the “fishing” vessels listed here were inboard.
4.0 Conclusion

This design shows one revolution around the design circle. This paper has simply portrayed the first steps in the design of the 44 foot open fisherman. From the calculations presented here it is evident that the planing performance may be questionable, yet through vessel comparison it appears that planing will still occur. It is felt that this poor performance may be due to the large fuel load necessary to complete the design mission. To accommodate this the deadrise angle of the vessel was greatly reduced to give the vessel planing characteristics of a flat plate. As this vessel is designed primarily as a pleasure craft it is not felt that it will be used in rough seas and thus will only be a problem in storm conditions. It also appears that the hydrostatic and stability calculations need adjustment. This can only occur after the vessel has been re-entered into GHS. Unfortunately the GHS program “crashed” before this recalculation could take place. As such the numbers previously calculated must be used at this time. The issue of dynamic stability must also be addressed. Although the LCB and LCG positions were at exactly the same point on the hull it is still possible that the sterndrive propulsion system will have negative effects on this design. As such this aspect of the design may be discarded for a more traditional approach. The sterndrive arrangement also appears to have a negative effect on the running angle of the vessel. It is hoped that this will be remedied with trim tabs. This could also be remedied by moving weight farther forward, yet this does not seem possible with this design. A rudder and propeller was designed in the instance that the sterndrive arrangement should prove ineffective. Implementation of this new inboard system would be labor intensive yet the design for this modification has already been prepared and should make the transition much simpler.
5.0 References


Chapter 77 “The Preliminary Hydrodynamic Design of a Motor Boat”

Calkin

Codega

Johansen


Appendix A
HULL Body Plan (1 component)
Scale = 1:30

Component 1: HULL.C
HULL Isometric Projection
Appendix B
### SECTION AREA CURVES
HULL.C Component of Part HULL

Trim: zero  Heel: zero

<table>
<thead>
<tr>
<th>Section Location</th>
<th>Reference Point Depth at 0.50</th>
<th>1.50</th>
<th>2.50</th>
<th>3.00</th>
<th>4.00</th>
</tr>
</thead>
<tbody>
<tr>
<td>22.00f</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>19.80f</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>19.24f</td>
<td></td>
<td></td>
<td></td>
<td></td>
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Distances in FEET. Areas in square FEET.-------

HULL Reference Point: Long. = 0.00  Trans. = 0.00  Vert. = 0.00
SECTION AREA CURVES
HULL.C Component of Part HULL

Depths (D) relative to HULL Reference Point
CROSS CURVES OF STABILITY
Showing righting arms in heel at VCG = 0.00

Trim: zero at zero heel (trim righting arm held at zero)

<table>
<thead>
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<th>Heel Angles in Degrees</th>
</tr>
</thead>
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<tr>
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<td>2.97s</td>
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<tr>
<td>12.78</td>
<td>2.38s</td>
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<tr>
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<td>1.73s</td>
</tr>
<tr>
<td>41.29</td>
<td>1.42s</td>
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<td>1.30s</td>
</tr>
<tr>
<td>72.42</td>
<td>1.27s</td>
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<th>70.00s</th>
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</thead>
<tbody>
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<td>5.74s</td>
<td>6.30s</td>
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<tr>
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<td>5.07s</td>
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<td>5.90s</td>
<td>5.50s</td>
<td>4.89s</td>
</tr>
<tr>
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<td>5.26s</td>
<td>4.78s</td>
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<tr>
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<td>5.02s</td>
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<td>4.80s</td>
<td>4.90s</td>
<td>4.83s</td>
<td>4.62s</td>
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Distances in FEET.--Specific Gravity = 1.025.--
Specific Gravity = 1.025  Assumed KG = 0.00 FT
"X" = Baseline
CURVES OF FORM
HULL.C Component of Part HULL

Trim: zero  Heel: zero

<table>
<thead>
<tr>
<th>Ref Pt</th>
<th>Volume (Cu Ft)</th>
<th>Block Coef</th>
<th>Displ/Length</th>
<th>WaterPl Coef</th>
<th>MaxSect Coef</th>
<th>Prismatic Coef</th>
<th>Long</th>
<th>Vert</th>
</tr>
</thead>
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<tr>
<td>0.50</td>
<td>3.6</td>
<td>0.136</td>
<td>6.0</td>
<td>0.597</td>
<td>0.500</td>
<td>0.354</td>
<td>0.228</td>
<td></td>
</tr>
<tr>
<td>1.50</td>
<td>226.1</td>
<td>0.297</td>
<td>114.3</td>
<td>0.800</td>
<td>0.587</td>
<td>0.709</td>
<td>0.370</td>
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<td>0.787</td>
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<td>3.00</td>
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<td>412.8</td>
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<td>0.824</td>
<td>0.842</td>
<td>0.622</td>
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</table>

Distances in FEET.----Length is true waterline.-----------------------

HULL Reference Point: Long.= 0.00  Trans.= 0.00  Vert.= 0.00
CROSS CURVES OF STABILITY
Showing righting arms in heel at VCG = 0.00

Trim: zero at zero heel (trim righting arm held at zero)

<table>
<thead>
<tr>
<th>Displacement</th>
<th>Heel Angles in Degrees</th>
</tr>
</thead>
<tbody>
<tr>
<td>LONG TONS</td>
<td>10.00s 20.00s 30.00s 40.00s 50.00s</td>
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</tbody>
</table>

| 0.10 |

Error: Not enough weight.

CROSS CURVES OF STABILITY
Showing righting arms in heel at VCG = 0.00

Trim: zero at zero heel (trim righting arm held at zero)

<table>
<thead>
<tr>
<th>Displacement</th>
<th>Heel Angles in Degrees</th>
</tr>
</thead>
<tbody>
<tr>
<td>LONG TONS</td>
<td>10.00s 20.00s 30.00s 40.00s 50.00s</td>
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</tbody>
</table>

| 0.10 |

Error: Not enough weight.

CROSS CURVES OF STABILITY
Showing righting arms in heel at VCG = 0.00

Trim: zero at zero heel (trim righting arm held at zero)

<table>
<thead>
<tr>
<th>Displacement</th>
<th>Heel Angles in Degrees</th>
</tr>
</thead>
<tbody>
<tr>
<td>LONG TONS</td>
<td>10.00s 20.00s 30.00s 40.00s 50.00s</td>
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</tbody>
</table>

| 0.10 |

Error: Not enough weight.

HYDROSTATIC PROPERTIES
No Trim, No Heel, VCG = 0.00

<table>
<thead>
<tr>
<th>LCF</th>
<th>Displacement</th>
<th>Buoyancy-Ctr.</th>
<th>Weight/ Moment/</th>
</tr>
</thead>
<tbody>
<tr>
<td>Draft---Weight(LT)---LCB-----VCB-----Inch-----LCF--Deg trim----KML-----KMT</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>0.500</td>
<td>0.10</td>
<td>8.24f</td>
<td>0.40</td>
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<tr>
<td>2.000</td>
<td>12.78</td>
<td>4.94a</td>
<td>1.44</td>
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<td>2.500</td>
<td>19.59</td>
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Distances in FEET.-------Specific Gravity = 1.025.--------Moment in Ft-LT.
Draft is from Baseline.
HYDROSTATIC PROPERTIES at LEVEL TRIM

1. Displacement 1 = 0.09 LT
2. LCB (use top scale)
3. UCB (KB) 1 = 0.01 FT
4. Immersion 1 = 0.003 LT/IN
5. LCF (use top scale)
6. Moment/Trim 1 = 0.08 FT-LT/Deg
7. KML 1 = 0.9 FT
8. KMT 1 = 0.04 FT

Specific Gravity = 1.025  Assumed KG = 0.00 FT
"K" = Baseline
Appendix C
### Area on Chine

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<th>Station #</th>
<th>y(l)</th>
<th>Cs</th>
<th>y(l)*Cs</th>
<th>r(l)</th>
<th>y(l)<em>Cs</em>r(l)</th>
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<td>9</td>
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<td>3</td>
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<td>10</td>
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Sum: 172.14 971.64

Area = $\frac{1}{3}h(172.14)$

$A \text{rea} = 252.472$ Square ft

Area = 504.944

LCA = 24.83569 ft from bow

### GZ - $\phi$ Curve Data

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<th>Sin $\phi$</th>
<th>Kg sin $\phi$</th>
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<th>Gz</th>
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<td>3.000000</td>
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Gz - phi curve
Appendix D
nomograph, Fig. 96. (This graph by Koelbel is valid when the propeller thrust, the resistance force and the resultant of the planing force all act through the CG). Hence, the mean wetted length, \( L_m \), and trim angle, \( \tau \), can be determined.

Savitsky also gives a formula to correct the mean wetted length ratio, \( \lambda \), to the keel wetted length ratio, \( \lambda_k \), if desired,

\[
\lambda_k = \lambda - 0.03 + \frac{1}{2} \left( 0.57 + \frac{\beta}{1000} \right) \left( \frac{\tan \beta}{2 \tan \tau} - \frac{\beta}{167} \right)
\]

where the value of \( \beta \) should be taken at the mid-chine-length position.

The value of \( \lambda_k \) should now be compared with the value of \( L_{WL}/b \). If \( \lambda_k \geq L_{WL}/b \) then the bow is not clear of the water and the craft is not fully planing. In that case the resistance may be calculated with the help of the formulas of Section 8.12. When \( \lambda_k < L_{WL}/b \), the bow is essentially clear of the water and the resistance can be predicted from the following equation:

\[
R_f = W \tan \tau + \frac{1}{2} \rho V^2 \lambda b^2 C_{P0} (\cos \tau \cos \beta)
\]

(73)

where \( C_{P0} \) is the friction coefficient according to the ITTC 1957 or the ATTC friction lines as a function of the Reynolds number, \( R_n = \frac{V_1 \lambda b}{v} \).

Here \( V_1 \) is the average bottom velocity, which is less than the forward planing velocity \( V \) owing to the fact that the planing bottom pressure is larger than the free-stream pressure. The following equation applies for zero deadrise:

\[
V_1 = V \left( 1 - \frac{0.0120 \tau^{1.1}}{\sqrt{\lambda \cos \tau}} \right)^{1/2}
\]

(74)

For other deadrise angles Savitsky (1964) gives con-

![Fig. 96 Nomogram for equilibrium conditions when all forces act through CG; \( \lambda \) and \( C_{P0} / \tau^{1.1} \) for given values of \( C_v \) and \( p/b \) (Koelbel)]
*Fig. 138 Chart showing areas for practical use of fully-cavitating propellers

\( \sigma = \frac{H}{V_a^2} = 22.57 \frac{H}{V_a^2} \) with \( H \) in feet and \( V_a \) in knots

(\( H \) = absolute pressure at shaft in feet of water and \( V_a \) = speed of advance in knots)

Zone 1: Fast region for fully-cavitating propellers
Zone 2: Marginal region with some cavitation on all propellers
Zone 3: Fast region for conventional propellers
Zone 4: Region of low efficiency for all propellers (\( \sigma < 0.64 \))
Appendix E
Diagram 11.14 Preliminary propeller design diagram

Fig. 11.14 Preliminary propeller design diagram

$K_A$ = 395.1 $\eta_f$ HP
$F = P^2V^2$
$J = \frac{nD}{nD}$

where:
- $T$ = thrust in tons
- $K_A = \eta_f$ HP
- $D$ and $F$ are in feet
- $V$ is in knots
- $n$ is in rpm

Diagram gives the maximum propeller efficiency, $\eta_m$, when any known value of $K_A$ corresponding values of efficient $J$ and face pitch.

Table for $K_f$ and $J$ values for different $\eta_m$.
characteristics of the smaller diameter screws and higher frequency, make the contrarotating propellers attractive from the hydrodynamic point of view. Economically they would lead to a saving in weight and main machinery, but against this must be set the weight and complication of the gearing, coaxial running, and sealing problems of the contrarotating arrangement.

**Fully-Cavitating Propellers.** When the cavity on the back of a propeller blade has spread until it covers the whole of the back, which is then no longer wetted, the propeller is said to be operating in the fully-cavitating regime (Section 16.2). After the back of the section has been completely denuded of water, further increase in revolutions per minute cannot reduce pressure there any more, and so no additional lift can be generated by the blades. On the face, however, pressure continues to increase with higher revolutions and so does the total thrust, though at a slower rate than before cavitation began.

One advantage of such propellers is the absence of back erosion, because the cavitation bubbles no longer collapse on the back of the blades. Also, the unsteady forces resulting from intermittent cavitation will be much reduced so that less vibration may be expected [159].

Parsons observed propellers working in this regime in his small tunnel in connection with the Turbinia design, and they have long been used on high-speed racing motor boats, the characteristics being determined by extensive trial-and-error experiments. One of the pioneers in this research was V. L. Posdunine, of the Moscow Academy, who directed attention to the use of fully-cavitating propellers in a paper presented in London in 1944 [230]. He gave a theoretical model of the flow and deduced expressions for the thrust and ideal efficiency. It is of interest to note that he spoke at this time of "wedge-shaped blades," and in other papers showed sections quite similar to those used in such propellers today. He also stated that the action of such propellers
Rys. 306. Wykres $J = K_I, K_0, \eta_p$ dla śrub serii B.2.30 — Wageningen
**Speed vs. Resistance**

<table>
<thead>
<tr>
<th>$V$ (kts)</th>
<th>22</th>
<th>23</th>
<th>25</th>
<th>31</th>
<th>34</th>
</tr>
</thead>
<tbody>
<tr>
<td>$76%$</td>
<td>37.59</td>
<td>37.59</td>
<td>37.59</td>
<td>37.59</td>
<td>57.426</td>
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</table>

<table>
<thead>
<tr>
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<th>5418.447</th>
<th>5324.697</th>
<th>5595.352</th>
<th>6377.853</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_{res}$</td>
<td>5707.826</td>
<td>5707.826</td>
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**Prop Design**

Two props; thus one prop must generate $2715.5 \text{ lbf}$

\[
F_p = \frac{5707.826}{530} = 601.936 = 602 \text{ hp}
\]

From Charts Fig 6

\[
\eta = 0.97
\]

\[
\eta_w = 0.98
\]
\[ T_2 = \frac{R_{\frac{1}{2}}}{1 - t} = \frac{2953.91}{0.92} = 3102 \text{ lb} \]

\[ \text{BAR} = \frac{4(3102)}{0.45 - 0.05(2.44)(1.067 - 0.0229) \pi \varphi^2} \]

\[ \text{BAR} = \frac{6.17}{K(\pi \varphi^2)} \quad D = 1.2 \]

\[ \text{BAR} = \frac{12408}{1926.22 (0.233)(\pi \varphi^2)} = 12408 \]

\[ \text{BAR} = \frac{12408}{1506.75 kD^2} \]

\[ D = 1.8 \]

\[ \text{BAR} = \frac{2.26}{kD^2} \]

\[ D = 2.75 \]

\[ \text{L} = 70.26 (0.98) = 56.73 \text{ lbf} \]

\[ V = 56.73 \text{ lbf} \]

\[ K = \frac{3102}{1.99(56.73)^2} = 123 \]

\[ \frac{P}{D} = 1.65 \]

\[ 1.95 \]
$\text{BAR} = \frac{124.03}{k(1.25 + 0.0229(1.76 - 0.0229)(2.56))}$

$\text{BAR} = 0.5$

$\text{BAR} = 0.9$

$10\%$

$P_a = 1.79$

$P_a = 0.5$

$P_a = 1.79$

$P_a = 0.72$

$J = 1.4$

$1.4 = \frac{5.628}{n(2)}$

$n = 20.1 \text{ rps}$

$n = 20.1(60) = 1206 \text{ rpm}$

$\rho_w = \frac{1 - t}{1 - \text{air}} = 0.94$

$P_a = \frac{60.2}{0.72(1.94)} = 839.97 \text{ hp}$

$P_d = \frac{P_a}{\rho(0.9)} = 989 \text{ hp}$

Assume $\rho = 0.9$ due to gearing
approaches zero, it is naturally expected that the calculated load should approach the buoyant load. It is interesting to note from Fig. 12 that in the range $0.60 \leq C_r \leq 1.00$, the motion of the planing surface reduces the lift below the value which would be expected on a purely displacement basis. This effect is somewhat similar to the sinkage experienced by displacement vessels at low speeds. At $C_r = 1.0$, the total planing load is approximately equal to the hypothetical buoyant load. At $C_r > 1.0$ the positive dynamic reaction of the fluid on the planing bottom increases rapidly as the speed increases.

**Lift of Deadrise Planing Surfaces**

For a given trim and mean wetted length-beam ratio, the effect of increasing the deadrise angle is to reduce the planing lift. This lift reduction is caused primarily from a reduction in the stagnation pressure at the leading edge of the wetted area. It will be recalled from the discussion of wetted areas that the angle between the stagnation line and keel is given by the equation $\gamma = \tan^{-1}(\tan \alpha / 2 \tan \beta)$. When $\beta = 0$ the stagnation line is normal to the keel and normal to the free-stream velocity so that full stagnation pressure $1/2 \rho U^2$ is developed. For increasing values of $\beta$, the angle $\gamma$ de-

---

*Fig. 11  Lift coefficient of a deadrise planing surface*
\[ A = \text{Rudder Area} \times \text{sum of rudder areas if more than one rudder fitted} \]

\[ A_L = \text{Immersed Lateral Area (based on static condition)} \]

**SAILING YACHTS and MOTOR SAILERS**

\[ A = 0.07 A_L \text{ to } 0.11 A_L \]

\[ A = 0.05 A_L \text{ (for } A_L > 40 \text{ m}^2 \text{) to } 0.105 A_L \text{ (for } A_L > 85 \text{ m}^2 \text{)} \quad \text{---Ref. 23} \]

**LOW SPEED POWER CRAFT**

\[ A = 0.03 A_L \text{ to } 0.05 A_L \]

**SEMI-DISPLACEMENT and PLANNING CRAFT**

\[ A = 0.03 A_L \text{ to } 0.04 A_L \quad (A_L \text{ in static condition}) \]

Following mean curve is based on data from Refs. 3 and 24 for satisfactory existing craft (rudders) operating in slipstream of propellers.

---

**Fig. 16** TYPICAL RUDDER AREAS FOR SMALL CRAFT
for Free Stream Rudder

Fig 6a LIFT COEFFICIENT: ALL-MOVABLE Rudders

Drawn for:
- $Re = 2 \times 10^4$
- TAPER RATIO $T/R = 0.80$
- THICKNESS RATIO $t/c = 0.10$
- SQUARE TIPS

See Figs 4, 5, and 6a for corrections due to changes in $Re$, $T/R$, and $t/c$

Fig 6b DRAG COEFFICIENT and LIFT/DRAG RATIO
ALL-MOVABLE Rudders

Drawn for:
- $Re = 2 \times 10^4$, $t/c = 0.10$

See Fig 6b for corrections to $C_D$ due to changes in $Re$ and $t/c$.
WL = 42.5 ft

Fig 16

Rudder Area = 0.3 m²

\[ V_R = 1.2 \times a = 1.2 \times (56.28) = 67.536 \text{ ft/s} \]

Assume AR = 2

\[ C = \sqrt{\frac{0.3}{2}} \quad C = 0.39 \text{ m} \]

\[ S = 0.39(2)^{0.78} \text{ m} \]

\[ \omega_t = 0.3(0.6) - 0.26 = 0.22 \]

\[ V_R = 1.15 \times (67.536) \times (1.22) \]

\[ V_R = 15.69 \text{ m/s} \]

\[ Re = \frac{15.69 \times 0.39}{1.19 \times 10^{-6}} = 5.14 \times 10^6 \]

\[ \xi = 0.15 \times (0.39)^2 = 53.5 \text{ mm} \]
Gap effect

\( \text{Gap} = 10 \text{ mm} \quad \text{C} = 390 \text{ mm} \)

\( \frac{10}{390} = 0.026 \)

\( k = 1.7 \)

\( AR_E = 1.7 (z) = 3.4 \)

1) \( AR_E = 1.37 (z) = 2.74 \)

2) \( AR_E = 1.4 (z) = 2.8 \)

\( \alpha = 27.5^\circ \)

3) \( AR_E = 1.43 (z) = 2.86 \)

\( \alpha = 26.8^\circ \)

4) \( AR_E = 1.4 (z) = 2.8 \)
\[ C_L = 1.2 \]
\[ C_D = 0.33 \]
\[ C_N = 1.8 \cos 27^\circ + 0.33 \sin 27^\circ \]
\[ = 1.75 \]
\[ W = 1.75 \times 0.5 \times 1025 \times 0.3 \times (15.69)^2 \]
\[ N = 66.24 \text{ kN} \]
\[ C_R = \sqrt{1.8^2 + 0.33^2} \]
\[ = 1.83 \]
\[ R = \frac{1}{2} \rho V^2 A_R C_f = 0.5 \times 1.33 \times 1025 \times 0.3 \times (15.69)^2 \]
\[ R = 69.26 \text{ kN} \]
\[ C_{\text{pit}} = 0.18(\cos) \times 0.0702 \text{ m} \]
\[ C_{\text{pit}} = 0.24(\cos) \times 0.93 \times 0.00234 \text{ m} \]
\[ 0.0936 \times 0.0702 = 0.00634 \text{ m} \]
\[ \text{Top hull ratio} = 0.8 \]
\[ C_{\text{pit}} = 0.495 (\cos) \times 0.78 (m) = 0.3861 \text{ m} \]
Torque at stall

\[ T = 66.24 \text{ kn} \times (0.0234) = 1.55 \text{ kn/m} \]

Bending Moment

\[ M = R \times (C_{ps}) = 69.26 \times (0.3361) = 26.74 \]

\[ BM_e = \frac{M}{2} + \frac{1}{2} \sqrt{M^2 + T^2} \]

\[ = \frac{26.74}{2} + \frac{1}{2} \sqrt{26.74^2 + 1.55^2} \]

\[ BM_e = 13.37 + .5 \times (26.73) \]

\[ BM_e = 26.76 \]

Stock Diameter:

\[ D = \sqrt[3]{\frac{BM_e (32)}{\pi (10^4)}} \]

\[ D = \sqrt[3]{\frac{26.76(32)}{\pi (130 \times 10^6)}} \]

\[ D = 0.0128 \text{ m} = 12.79 \text{ Mm} \]

Dimensions of\ Mild Steel