Reduction of Ship Resistance through Induced Turbulent Boundary Layers

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ABSTRACT

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In this research, the resistance of dimpled hull forms and their potential relevance in reducing the resistance of modern conventional hull forms has been investigated. This research has been undertaken while trying to understand the flight dynamics of a dimpled golf ball. Non-streamlined bodies, like spheres, have significant pressure drag due to flow separation. Pressure drag can be reduced through separation delay. The flow around a sphere with a Reynolds number less than $3 \times 10^5$ has a laminar boundary layer, thus the flow would separate at an angle of $82^\circ$ relative to the stagnation point. If the Reynolds number is increased beyond $3 \times 10^5$ the boundary layer becomes turbulent and separation occurs at $125^\circ$. It is understood that dimples are placed on a golf ball to trip the boundary layer from laminar to turbulent in order to delay flow separation. It is believed that the same concept can be adapted for use on a ship to decrease viscous resistance. Two model boats were created, a control (an unmodified model) and the experiment hull (one modified with dimples). Based on the experimental investigations, it cannot be determined if the reduction of ship resistance through induced turbulent boundary layers was a success for this hull form; however, it can be concluded that it was not a failure. Stochastically speaking, based on the quantitative magnitude of the resistance measurements taken, $0.032$ kg (0.071 lb) to $2.713$ kg (4.787 lb), and the magnitude of error, $0.030$ kg (0.0660 lb), the recorded resistance values of each model could potentially be equal.
Table of Contents

Abstract ........................................................................................................................................... iii
List of Figures .................................................................................................................................. v
List of Tables ..................................................................................................................................... vi
List of Symbols ............................................................................................................................... vii
Acknowledgments .......................................................................................................................... viii
Introduction ...................................................................................................................................... 1
Literature Review ............................................................................................................................ 3
  Calm Water Resistance .................................................................................................................... 3
  Golf Balls ........................................................................................................................................ 5
Drag Reduction through Turbulent Boundary Layers ................................................................. 8
  Laminar Boundary Layers ................................................................................................................ 8
  Turbulent Boundary Layers ............................................................................................................. 11
  Separation and the Effects of Turbulence ....................................................................................... 13
Hypothesis ........................................................................................................................................ 20
Vessel Parameters ........................................................................................................................... 22
Results ............................................................................................................................................... 31
Discussion ......................................................................................................................................... 36
Conclusions ....................................................................................................................................... 41
References ........................................................................................................................................ 43
Appendix .......................................................................................................................................... 44
  MHL Error Analysis ....................................................................................................................... 44
  Raw Data ....................................................................................................................................... 45
  Sample Calculations ....................................................................................................................... 62
List of Figures

Figure 1: Featherie (Martin, 1968) .................................................................5
Figure 2: Guttie with raised ridges (Martin, 1968) ........................................6
Figure 3: Boundary layer over a flat plate (Pai, 1957) ...............................12
Figure 4: Drag free flow over a sphere (Anderson, 2005) .........................13
Figure 5: Adverse pressure gradient (White, 1991) ..................................14
Figure 6: Real flow over a sphere (Anderson, 2005) ................................15
Figure 7: Airflow comparison of smooth ball vs. golf ball (Scott, 2005) ......17
Figure 8: Boundary layer transition and the effects of roughness (Pai, 1957) ....18
Figure 9: Plan and profile view ..................................................................25
Figure 10: Body plan view .........................................................................26
Figure 11: Initial dimple pattern .................................................................29
Figure 12: Model dimpling (starboard view) .............................................29
Figure 13: Model dimpling (bottom view) ..................................................30
Figure 14: Resistance comparison – 4.27 m (14 ft) waterline ..................33
Figure 15: Resistance comparison – 5.18 m (17 ft) waterline ..................33
Figure 16: $C_{TS}$ vs. $F_n$ – 4.27 m (14 ft) waterline ..................................35
Figure 17: $C_{TS}$ vs. $F_n$ – 5.18 m (17 ft) waterline ..................................35
Figure 18: Vessel motions (Newman, 1978) .............................................38
Figure 19: Model A form factor determination ..........................................63
Figure 20: Model B form factor determination ..........................................64
List of Tables

Table 1: Taylor dimple criteria ................................................................. 7
Table 2: Dunlop full-shot test results ...................................................... 8
Table 3: Vessel parameters ..................................................................... 23
Table 4: Table of displacements and wetted areas (ship) ......................... 23
Table 5: Table of displacements and wetted areas (model) ....................... 23
Table 6: Table of offsets (ship) ............................................................... 24
Table 7: Table of offsets (model) ............................................................. 27
Table 8: Resistance comparison 4.27 m (14 ft) waterline ....................... 32
Table 9: Resistance comparison 5.18 m (17 ft) waterline ....................... 32
Table 10: Extrapolated resistance data – 4.27 m (14 ft) waterline .......... 34
Table 11: Extrapolated resistance data – 5.18 m (17 ft) waterline .......... 34
Table 12: Model-ship correlation calculations ......................................... 62
Table 13: Full scale extrapolation from model scale .............................. 65
## List of Symbols

- **B**: Beam, m (ft)
- **$C_A$**: Correlation allowance coefficient
- **$C_B$**: Block Coefficient
- **$C_F$**: ITTC 1957 model-ship correlation line
- **$C_R$**: Residuary-resistance coefficient
- **$C_T$**: Total-resistance coefficient
- **$C_W$**: Wave-resistance coefficient
- **$Fn$**: Froude number
- **$g$**: Acceleration due to gravity, m/s\(^2\) (ft/s\(^2\))
- **$h_{sa}$**: Admissible roughness, m (ft)
- **$I+k$**: Form factor
- **$L$**: Length, m (ft)
- **$LOA$**: Length overall, m (ft)
- **$LWL$**: Length at waterline, m (ft)
- **$p$**: Pressure, Pa (psi)
- **$Re$**: Reynolds number
- **$R_T$**: Total resistance, N (lb)
- **$S$**: Wetted-surface area, m\(^2\) (ft\(^2\))
- **$t$**: Time, s
- **$T$**: Draft, m (ft)
- **$u$**: Fluid velocity, m/s (ft/s)
- **$U$**: Free stream velocity, m/s (ft/s)
- **$V$**: Ship velocity, m/s (ft/s)
- **$\delta$**: Boundary layer displacement thickness, m (ft)
- **$\Delta$**: Displacement weight, tonnes (lb)
- **$\theta$**: Boundary layer momentum thickness, m (ft)
- **$\lambda$**: Scaling Factor
- **$\mu$**: Dynamic viscosity, kg/m-s (slug/ft-s)
- **$\nu$**: Kinematic viscosity, m\(^2\)/s (ft\(^2\)/s)
- **$\rho$**: Fluid density, kg/m\(^3\) (slug/ft\(^3\))
- **$\tau$**: Shear stress, N/ft\(^2\) (lb/ft\(^2\))
- **$V$**: Volumetric displacement, m\(^3\) (ft\(^3\))
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Chapter 1

Introduction

In an environmentally conscious society, fuel efficiency is more important than ever. “Green” seems to be the theme of every major marketing campaign. Car manufactures are competing to have the most fuel efficient vehicle. Supermarkets are giving away free reusable shopping bags. The United States government is providing tax refunds for homeowners willing to invest in more efficient air conditioners, solar panels, and cars.

Saving the environment means using less fossil fuel. Much advancement has been made in the Naval Architecture field to improve the efficiency of ships, including innovative streamlined designs, bulbous bows, and multi-hulled vessels. A vessel can be optimized for efficiency during the design phase, but what about ships that have already been built and are in service? How can their efficiency be improved?

Consider for a moment, the game of golf. It was noticed in the late 19th century that old battered golf balls performed better than smooth new ones. Advancements in the game coincided with advancement in science. In 1905 Ludwig Prandtl, hypothesized the existence of the boundary layer. Shortly after in 1906, William Taylor invented the modern dimpled golf ball. The relationship between the two is the drag reducing properties of the turbulent boundary layer.
It is the intent of this thesis to explore the relevance of turbulent boundary layers to reduce ship resistance. The theory is based primarily on the fluid mechanics of a golf ball; for this reason vessel selection plays an essential role in the success of the experiment. The streamlines over the unmodified hull must be subjected to the adverse pressure gradients that cause separation. If correct, dimples will create turbulent boundary layers that will delay the separation of the flow on the hull, reducing pressure drag. By reducing drag, the hull becomes more efficient, requiring less power for a given speed.
Chapter 2

Literature Review

Calm Water Resistance

Traditionally, resistance predictions of a vessel are performed via model testing. Here the resistance of a smaller but geometrically similar vessel, or a model, is measured and recorded. The total resistance generated by the model is then scaled up to the full size vessel. The scaling factor, $\lambda$, is the size relationship between the ship and model and is defined as,

$$\lambda = \frac{L_s}{L_m}. \quad [1]$$

The International Towing Tank Conference (ITTC) defines the resistance testing process in their 2008 “Recommended Procedures and Guidelines for Testing and Data Analysis Methods Resistance Test”. Let it be noted that subscripts “$M$” and “$S$” indicate values for the model and ship respectively. ITTC (2008) defines the total resistance coefficient as a function of the ITTC 1957 model-ship correlation line, $C_F$, the form factor, $(1+k)$, and the residuary resistance coefficient, $C_R$,

$$C_{T_M} = C_{FM} (1+k) + C_R. \quad [2]$$
Here, the total resistance coefficient, $C_T$, is the dimensionless form of the total resistance, $R_T$,

$$C_T = \frac{R_T}{\frac{1}{2} \rho SV^2}.$$  \[3\]

The 1957 ITTC model-ship correlation line is defined as,

$$C_F = \frac{0.075}{(\log_{10}(Re) - 2)^2}.$$  \[4\]

Here, $Re$ is the Reynolds number given by the expression,

$$Re = \frac{VL}{\nu},$$  \[5\]

where $V$ is velocity, $L$ is vessel length at the designed waterline, and $\nu$ is the kinematic viscosity of the fluid. The residuary resistance coefficient, $C_R$, from equation 2, is comprised mostly of wave making resistance. Since corresponding testing speeds between the model and ship are found using Froude’s law of comparison and the Froude number is based on wave patterns, the residuary resistance coefficient for the model is equal to the residuary resistance coefficient for the ship. The Froude number is,

$$Fn = \frac{V}{\sqrt{gL}}.$$  \[6\]

and Froude’s law of comparison is,

$$\frac{V}{\sqrt{gL_s}} = \frac{V_m}{\sqrt{gL_m}}.$$  \[7\]

The form factor, $(1+k)$, is unique to each vessel based on the ships geometry. The ITTC (2008) recommended procedure for determining the form factor is to use Prohaska’s method of 1966, where $C_{TM}/C_{FM}$ is plotted against $Fn^4/C_{FM}$. It is
assumed that in the low speed region, $Fn < 0.2$, the function $Fn^4$ will be a straight line and cross the y-axis, when $Fn=0$, at $(1+k)$. Since testing speeds close to zero don’t occur, the tangent must be extended to approximate the form factor.

After the model test is complete, the measured model resistance, $R_{TM}$, can be scaled to the ship resistance, $R_{TS}$, in the following seven steps. Let it be noted that the correlation allowance, $C_A$, in step 6, accounts for the roughness of the ship’s surface that is not present on the model.

1. Nondimensionalize measured $R_{TM}$ values into $C_{TM}$
2. Calculate $C_{FM}$ (*by use of model-ship correlation line*)
3. Determine form factor $(1+k)$
4. $C_{RM}=C_{TM}-C_{FM}(1+k)=C_{RS}$
5. Calculate $C_{FS}$
6. $C_{TS}=C_{FS}(1+k)+C_{RS}+C_A$
7. $R_{TS}=C_{TS}/2\rho SV^2$

**Golf Balls**

According to John Martin (1968), the first golf ball was sold in 1452 for ten shillings. As the game developed, so did the balls used to play the game. In 1618 James Melvill of St. Andrews Scotland invented the “featherie”, Figure 1.

![Figure 1: Featherie (Martin, 1968)](image-url)
This ball was made by stuffing a leather pouch with wet goose down. As the ball dried the leather shrank and down expanded. This created internal pressure that rendered a ball that could be hit long distances, bounce, and roll. Unfortunately, the ball became useless if wet.

By 1846 Rob Patterson invented a new golf ball, the “guttie”. The guttie was created from gutta percha, a rubber like sap from the Sapodilla tree found in Malaysia. When the rubber was heated, it could easily be molded into a perfectly round sphere with a nice, smooth surface. This ball was easily produced, and easily fixed if damaged. The immediate popularity of the guttie stemmed from its ability to resist moisture, meaning golfers could now play when it was wet or raining. However, golfers soon realized that it did not travel as far nor soar through the air as well as the featherie. Overtime, golfers discovered that the old battered gutties had better flight characteristics. John Martin (1968) states, “Then it began to be noticed that scarred-up balls ‘dooked and shied’ [ducked and darted] much less than smooth new ones did. So the gutties were given a few clips with a cleek or other iron club before being put on sale.”

By 1860, gutties were being manufactured using two part compression molds with built in “raised ridges” to provide new balls with surface imperfection needed for better flight, Figure 2.

Figure 2: Guttie with raised ridges (Martin, 1968)
It wasn’t until 1906 that William Taylor developed the modern dimpled golf ball. With a keen understanding of aerodynamics, Taylor defined a range of criteria that identifies the geometry of the dimple in order to optimize flight. This concept was later patented in 1908. Table 1 characterizes the dimple design.

<table>
<thead>
<tr>
<th>Ball Diameter, mm (in)</th>
<th>Percentage of Coverage</th>
<th>Dimple Diameter, mm (in) [% of ball diameter]</th>
<th>Dimple Depth, mm (in) [% of dimple diameter]</th>
</tr>
</thead>
<tbody>
<tr>
<td>41.15 (1.62)</td>
<td>25%-75%</td>
<td>2.29-3.81 (.09-.15) [5.56%-9.26%]</td>
<td>&gt;0.356 (.014”) [9.33%-15.56%]</td>
</tr>
<tr>
<td>42.67 (1.68)</td>
<td>25%-75%</td>
<td>2.29-3.81 (.09-.15) [5.36%-8.93%]</td>
<td>&gt;0.356 (.014”) [9.33%-15.56%]</td>
</tr>
</tbody>
</table>

In the 1930’s the sport became standardized. In 1931, American golf associations standardized ball sizes as 42.67mm (1.68 in) in diameter. In 1932, British and Canadian standard balls sizes became 41.15mm (1.62 in) in diameter. Regardless of manufacturer, golf balls were now all made with dimples. The average ball contained 332 to 338 dimples with dimple diameters between 3.18mm (.125 in) and 3.28mm (.129 in) and depths ranging from 0.305 mm (.012 in) to 0.343 mm (.0135 in). Notice these specifications are similar to Taylor’s dimple criteria, Table 1. The diameters, 3.18mm (.125 in) to 3.28mm (.129 in), fall exactly between Taylor’s optimized dimple diameters, 2.29 mm (.09 in) to 3.81 mm (.15 in). The dimple depths, 0.305 mm (.012 in) to 0.343 mm (.0135 in), are less than Taylor’s recommended minimum dimple depth of 0.356 mm (.014 in); however, the proportional relationship between dimple depth and dimple diameter is satisfied, 9.33%-15.56%.

John Martin (1968) references a study performed by Dunlop on a 41.15 mm (1.62 in) diameter golf ball. Here the carry, or the distance the ball travels in flight, is
compared among balls with several different dimple depths. Results are summarized in Table 2.

<table>
<thead>
<tr>
<th>Dimple Depth, mm (in)</th>
<th>Carry, m (ft)</th>
<th>Carry Plus Roll, m (ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.025 (.001)</td>
<td>107 (351)</td>
<td>134 (438)</td>
</tr>
<tr>
<td>0.102 (.004)</td>
<td>171 (561)</td>
<td>194 (636)</td>
</tr>
<tr>
<td>0.178 (.007)</td>
<td>194 (636)</td>
<td>212 (696)</td>
</tr>
<tr>
<td>0.254 (.010)</td>
<td>204 (669)</td>
<td>218 (714)</td>
</tr>
<tr>
<td>0.330 (.013)</td>
<td>218 (714)</td>
<td>239 (783)</td>
</tr>
<tr>
<td>0.406 (.016)</td>
<td>206 (675)</td>
<td>219 (720)</td>
</tr>
</tbody>
</table>

To understand why dimpling increases the travel of golf balls, one must understand a bit of fluid mechanics.

**Drag Reduction through Turbulent Boundary Layers**

**Laminar Boundary Layers**

The fundamental equations of motion for a Newtonian fluid can be express through Navier-Stokes. According to Kundu (2008) Navier-Stokes may be expressed as shown in equation 8,

$$\rho \frac{Du_i}{Dt} = -\frac{\partial p}{\partial x_i} + \rho g_3 + \mu \frac{\partial^2 u_j}{\partial x_j \partial x_i}.$$  \[8\]

An expansion of equation 8 into component form yields,

$$\frac{\partial u_i}{\partial t} + u_1 \frac{\partial u_1}{\partial x_1} + u_2 \frac{\partial u_1}{\partial x_2} + u_3 \frac{\partial u_1}{\partial x_3} = -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \nu \left( \frac{\partial^2 u_i}{\partial x_1^2} + \frac{\partial^2 u_i}{\partial x_2^2} + \frac{\partial^2 u_i}{\partial x_3^2} \right).$$  \[9\]
\[ \frac{\partial u_2}{\partial t} + u_1 \frac{\partial u_3}{\partial x_1} + u_2 \frac{\partial u_2}{\partial x_2} + u_3 \frac{\partial u_3}{\partial x_3} = -\frac{1}{\rho} \frac{\partial p}{\partial x_2} + \nu \left( \frac{\partial^2 u_2}{\partial x_1^2} + \frac{\partial^2 u_2}{\partial x_2^2} + \frac{\partial^2 u_2}{\partial x_3^2} \right), \]  \tag{10}

\[ \frac{\partial u_3}{\partial t} + u_1 \frac{\partial u_3}{\partial x_1} + u_2 \frac{\partial u_3}{\partial x_2} + u_3 \frac{\partial u_3}{\partial x_3} = -\frac{1}{\rho} \frac{\partial p}{\partial x_3} + \nu \left( \frac{\partial^2 u_3}{\partial x_1^2} + \frac{\partial^2 u_3}{\partial x_2^2} + \frac{\partial^2 u_3}{\partial x_3^2} \right) + g_3. \]  \tag{11}

The nonlinearity of these equations makes them difficult to solve. This is due to their three dimensional nature and the lack of an explicit solution for pressure; however, key elements influencing the fluid flow for a certain set of circumstances can be identified by making assumptions and eliminating non-relevant terms. Assumptions that allow Navier-Stokes to be reduced include but are not limited to steady state flow, well developed flow, inviscid flow, two dimensional flow, and incompressibility. The simplified equations are applicable to circumstances where the base assumption can be applied.

“In 1905 Ludwig Prandtl...first hypothesized that, for small viscosity, the viscous forces are negligible everywhere except close to the solid boundaries where the no-slip condition must be satisfied” (Kundu, 2008). This hypothesis was the first explanation of the boundary layer, a thin, near wall layer of fluid that satisfied no-slip and accounted for viscous forces in an otherwise inviscid fluid. The behavior of the boundary layer is easiest to explain on an infinite flat plate. To determine the equations of motion within this boundary layer, the following assumptions can be made,

1. The flow is 2D: \( u_3 \) and \( x_3 \) are not considered.
2. The free stream flow is steady: \( \frac{\partial}{\partial t} = 0 \).
3. The flow is laminar: \( \nu \) is a molecular property of the fluid.
4. The fluid is incompressible: \( \frac{\partial u_1}{\partial x_1} + \frac{\partial u_2}{\partial x_2} = 0 \).
5. Variations across the boundary layer are much faster than variations along
the layer: \( \frac{\partial}{\partial x_1} \ll \frac{\partial}{\partial x_2} \) and \( \frac{\partial^2}{\partial x_1^2} \ll \frac{\partial^2}{\partial x_2^2} \).

6. Reynolds number is large: \( Re \gg 1 \).

7. The boundary layer is thin: \( \delta \ll x \).

8. Velocities along the boundary layer are much greater than those across the
boundary layer: \( u_2 \ll u_1 \).

Applying assumptions 1 though 8 eliminates non-applicable terms from equations
9, 10, and 11. For example, equation 9 can be immediately reduced to,

\[
u_1 \frac{\partial u_1}{\partial x_1} + u_2 \frac{\partial u_1}{\partial x_2} = -\frac{1}{\rho} \frac{\partial p}{\partial x_1} + \nu \left( \frac{\partial^2 u_1}{\partial x_1^2} + \frac{\partial^2 u_1}{\partial x_2^2} \right). \tag{12}
\]

Scaling, or non-dimensionalizing the remaining terms with respect to free stream
velocity and geometric length, reduces the equations of motion to the significant
terms based on order of magnitude. According to Kundu (2008) the approximate
equations of motion within the boundary layer in a steady state incompressible
laminar flow are:

\[
u_1 \frac{\partial u_1}{\partial x_1} + u_2 \frac{\partial u_1}{\partial x_2} = -\frac{1}{\rho} \frac{\partial p}{\partial x_1} + \nu \frac{\partial^2 u_1}{\partial x_1^2}, \tag{13}
\]

\[0 = -\frac{\partial p}{\partial x_2}, \tag{14}\]

\[rac{\partial u_1}{\partial x_1} + \frac{\partial u_2}{\partial x_2} = 0. \tag{15}\]

Hill (1992) defines equation 13 as the Momentum Equation. Using the 1921 von
Karman momentum integral method, Hill solves equation 13 in terms of
displacement thickness, \( \delta^* \), momentum thickness, \( \theta \), shear stress at the wall (skin
friction), \( \tau_0 \), and free stream velocity, \( U \). This approximation method finds a
solution that satisfies the integral across the boundary layer but may not satisfy the
integral at specific points. It is important because shear stress generated by the boundary layer can now be approximated in terms of two measurable quantities, free stream velocity and boundary layer thickness. The solution to equation 13 is the Karman momentum integral equation,

\[
\frac{d\theta}{dx} + \left(2 + \frac{\delta^*}{\theta}\right)\frac{\theta}{U} \frac{dU}{dx} = \frac{\tau_0}{\rho U^2},
\]

where the skin friction, \(\tau_0\), is equal to the shear stress at the wall and is expressed as,

\[
\tau_0 = \mu \frac{\partial u_1}{\partial x_2},
\]

the displacement thickness, \(\delta^*\), is the thickness of boundary layer that has the effect of making that thickness unavailable for free stream flow,

\[
\delta^* = \int_0^\infty \left(1 - \frac{u}{U}\right) dx_2,
\]

and the momentum thickness, \(\theta\), is the loss of momentum due to the presence of the boundary layer,

\[
\theta = \int_0^\infty \frac{u_1}{U} \left(1 - \frac{u_1}{U}\right) dx_2.
\]

Equation 16 is applicable to both laminar and turbulent boundary layers, although in turbulence the shear stress is not a function of molecular viscosity. Kundu (2008) states that \(\tau_0\) must be empirically defined in turbulent boundary layers.

**Turbulent Boundary Layers**

Considering an infinite flat hydraulically smooth plate, downstream flow becomes unstable as the local Reynolds number increases, \(Re_\infty\). If the local Reynolds number exceeds the critical Reynolds number then the flow transitions into turbulence.
According to Kundu (2008) the critical Reynolds number is a function of surface roughness, free stream instabilities, and the shape of the leading edge. For a flat plate $Re_{cr} \sim 10^6$.

Under these circumstances, transition to turbulence is not instantaneous. As the name implies, there exists a region of transition where the laminar flow becomes unstable before becoming turbulent. Figure 3 demonstrates the transition from laminar to turbulent flow.

![Figure 3: Boundary layer over a flat plate (Pai, 1957)](image)

To estimate the coefficient of friction in the turbulent boundary layer, Pai (1957) assumes that transition occurs all at one point. Pai (1957) identifies that the coefficient of friction is higher in a turbulent boundary layer than it is in the laminar boundary layer for a smooth flat plate. Kundu (2008) agrees that turbulent boundary layers generate more frictional resistance due to the greater macroscopic mixing in a turbulent flow.

Transition can be nearly instantaneous if the plate abruptly shifts from hydraulically smooth. Roughness elements can disturb the laminar flow causing
turbulent transition much further upstream. To keep viscous drag low, roughness is usually avoided.

**Separation and the Effects of Turbulence**

On a flat plate, turbulent boundary layers increase frictional resistance; but what about curved surfaces? Anderson (2005) defines the d’Alembert’s paradox as the conflict between theory and experiment when dealing with the subsonic flow over a sphere or infinitely long cylinder with the axis normal to the direction of flow. Theoretically speaking, if the flow were inviscid the streamlines would remain attached to the wall as seen in Figure 4. The pressure at the leading and trailing edge, $\theta=0^\circ$ and $\theta=180^\circ$ respectively, are equal. This means there is no drag.

![Figure 4: Drag free flow over a sphere (Anderson, 2005)](image_url)
Realistically, as the flow diverges about the centerline at the leading edge, the pressure gradient, $\frac{\delta p}{\delta x}$, is negative and the flow accelerates around the high side of the sphere, $\theta=90^\circ$, while the streamlines converge. As the flow progresses beyond $\theta=90^\circ$ to $\theta=180^\circ$ it converges back onto the centerline on the downstream side of the sphere. Here the streamlines diverge and the flow is forced to slow down because the pressure gradient, $\frac{\delta p}{\delta x}$, is positive and “pushes back” against the flow. According to White (1991) this causes points of inflection in the velocity profile of the boundary layer, $\frac{\delta u}{\delta x}$. If the pressure gradient decreases at the wall until it reaches zero, $\frac{\delta u}{\delta x}=0$, then the flow separates from the surface. Further downstream of the separation point $\frac{\delta u}{\delta x}$ is both negative and positive, meaning that fluid is back flowing near the wall. This can be seen in Figure 5.

![Figure 5: Adverse pressure gradient (White, 1991)](image)

In Figure 4 the positive, or adverse, pressure gradient between $90^\circ<\theta<270^\circ$ would cause the flow to separate. The pressure at the leading edge would be greater than the pressure at the trailing edge, $\theta=0^\circ$ and $\theta=180^\circ$ respectively, as seen in Figure 6.
This pressure gradient is known as pressure drag or form drag. For objects of curvature, the total resistance is a function of frictional resistance and form drag.

Kundu (2008) states the flow around a sphere with a Reynolds number less than $3 \times 10^5$ has a laminar boundary layer. This flow would separate at $\theta=82^\circ$. If the Reynolds number is increased beyond $3 \times 10^5$ the boundary layer becomes turbulent and separation occurs at $\theta=125^\circ$. In other words, a turbulent boundary layer has the ability to delay flow separation. Because of the sphere’s geometry, this would reduce the size of the downstream turbulent wake. In two dimensional terms, the area of separation is reduced. If the flow separates further downstream then the
pressure gradient from leading to trailing edge is smaller and form drag is reduced. Hill (1992) explains why a turbulent boundary layer delays separation,

…transition behavior takes place in the boundary layer, resulting in a significant reduction in its tendency to separate. The reason for this is that the turbulent fluctuations constitute a mechanism for transporting high momentum from the outer part of the layer to the region near the wall. The effect is as if there were an increased shear stress or…an increased coefficient of viscosity. This tends to raise the wall shear stress, but it also means that the turbulent boundary fluid can climb higher up the pressure hill than the fluid in the laminar layer…the fact that the turbulent layer tends to have much higher momentum (and shear stress) near the wall gives it an increased resistance to separation.

If the boundary layer is naturally laminar, turbulence can be forced to achieve separation delay. As mentioned earlier, alterations from a hydraulically smooth surface can cause instabilities and turbulence. This surface roughness can take the form of convexities or concavities. Figure 7 represents the effects of dimples on boundary layer transition and separation specifically in the case of a golf ball.
Figure 7 demonstrates how dimples are used to induce turbulent boundary layers. How can we be certain the dimples, or concavities, will create a turbulent boundary layer? According to Seebass (1990),

On the other hand, when the surface is concave, the fluid element moved to a greater radius is confined to a lower velocity region where the local radial pressure gradient is too low to force it back to a lower radius. The fluid element thus continues to move radially
outward until it meets the surface. This mechanism makes the flow over a concave surface inherently unstable.

In other words, flow over a concave surface is likely to become turbulent. On a small scale, dimples are concave surfaces.

Since a single dimple is relatively small compared to the surface area of the sphere, about 0.1%, the effects of a dimple can be estimated as a roughness element; therefore, a pattern of dimples can be estimated as surface roughness. The effect of roughness on transition is highly influenced by element shape, size, and spacing. Recognizing the work performed by Prandtl and Schlichting in 1934, Pai (1957) acknowledges that no surface is perfectly smooth. For this reason, a permissible amount of roughness should be determined so that smooth surface boundary layer approximations can be used in spite of the roughness. To determine the admissible roughness, $h_{sa}$, the following equation can be used.

$$\frac{U h_{sa}}{\nu} = 100.$$ \[20\]

This equation assumes the surface is covered with uniform sand roughness where $h_{sa}$ is the maximum allowable grain size before the roughness effects turbulent transition. The effects of a roughness element on turbulent transition can be seen in Figure 8.

![Figure 8: Boundary layer transition and the effects of roughness (Pai, 1957)](image)
Here, transition occurs more abruptly and further upstream compared to a hydraulically smooth plate. According to Dryden (1953), as the height of the roughness element, $h$, increases, the transition point, $x_t$, approaches the position of the roughness element, $x_h$. For engineering purposes, if the dimple dimensions are greater than the admissible roughness in equation 20, the dimples will influence turbulent transition. Designed correctly, dimples can almost instantly transition laminar boundary layers to turbulent boundary layers. This increases viscous resistance but delays separation, therefore reducing pressure drag. Given the correct circumstances dimples can be used to reduce the total drag on an object.
Chapter 3

Hypothesis

It is understood that dimples are placed on a golf ball to trip the boundary layer from laminar to turbulent in order to delay flow separation. This decreases the pressure gradient from leading to trailing edge and therefore decreases pressure drag. It is believed that the same concept can be adapted for use on a ship to decrease total resistance. This hypothesis is based on some fundamental assumptions:

1. The geometry of the vessel allows for flow separation.
2. The boundary layer of the vessel is laminar, may contain instabilities, but is not turbulent.

In order for the geometry of the vessel to accommodate flow separation the aft portions of the vessel must taper back towards the centerline. This would allow for the creation of the adverse pressure gradient that causes separation. If separation occurs then pressure drag is a contributor to the total resistance. Here the concept of separation control can be applied.

The vessel must also operate at relatively lower speeds to ensure the boundary layer is naturally laminar. Dimpling the hull, as done with a golf ball, will trip the boundary layer from laminar to turbulent, delay flow separation further
downstream, and decrease drag. If the boundary layer is naturally turbulent, then dimpling will have no affect on the mean flow.
Chapter 4

Vessel Parameters

Compared to a sphere or an infinite cylinder, a ship is relatively slender. For this reason, vessel selection plays a key role in testing this hypothesis. Planing hulls were eliminated as an option for this study for a number of reasons, primarily because the geometry of most planning vessels does not allow for flow separation. Very commonly, planning vessels have square transoms where the waterlines do not taper back towards the centerline. This is not conducive for separation delay. The fluid separates from the hull because the hull no longer exists downstream. In a sense, separation delay is like reattaching the flow to the object surface. If the object does not exist downstream, then the flow cannot be reattached.

Eliminating planning hulls also eliminated the complexity of the air/sea surface interface and narrowed the field of selection to displacement hulls. The geometry of a displacement hull remains submerged below the surface. This means the hull is exposed to one fluid. This is a simpler approach for a proof of concept. Additionally, displacement hulls commonly have waterlines with tapered aft portions and operate at relatively lower speeds. Displacement hulls are a far better hull form to test this hypothesis.
Vessel parameters for the vessel used in this study, at the tested displacements, are shown in Table 3. Vessel displacements and wetted surface areas are shown in Table 4 and Table 5.

### Table 3: Vessel parameters

<table>
<thead>
<tr>
<th></th>
<th>4.3 m (14 ft) Waterline</th>
<th>5.2 m (17 ft) Waterline</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Ship, m (ft)</td>
<td>Model, m (ft)</td>
</tr>
<tr>
<td>LWL</td>
<td>59.8 (196.3)</td>
<td>2.05 (6.73)</td>
</tr>
<tr>
<td>Breadth</td>
<td>9.9 (32.4)</td>
<td>0.34 (1.11)</td>
</tr>
<tr>
<td>Draft</td>
<td>4.3 (14)</td>
<td>0.15 (0.48)</td>
</tr>
<tr>
<td>$C_B$</td>
<td>0.626</td>
<td>0.626</td>
</tr>
</tbody>
</table>

### Table 4: Table of displacements and wetted areas (ship)

<table>
<thead>
<tr>
<th>Vessel</th>
<th>$T$, m (ft)</th>
<th>$\mathcal{V}$, $m^3$ (ft$^3$)</th>
<th>$\Delta$, tonnes (lb)</th>
<th>$S$, $m^2$ (ft$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1.83 (6)</td>
<td>488 (17243)</td>
<td>502 (1107023)</td>
<td>453 (4876)</td>
</tr>
<tr>
<td></td>
<td>2.44 (8)</td>
<td>723 (25530)</td>
<td>743 (1639023)</td>
<td>531 (5717)</td>
</tr>
<tr>
<td></td>
<td>3.05 (10)</td>
<td>974 (34408)</td>
<td>1002 (2209019)</td>
<td>608 (6546)</td>
</tr>
<tr>
<td></td>
<td>3.66 (12)</td>
<td>1239 (43772)</td>
<td>1275 (2810159)</td>
<td>686 (7384)</td>
</tr>
<tr>
<td></td>
<td>4.27 (14)</td>
<td>1516 (53551)</td>
<td>1559 (3437981)</td>
<td>764 (8224)</td>
</tr>
<tr>
<td></td>
<td>4.88 (16)</td>
<td>1804 (63704)</td>
<td>1855 (4089812)</td>
<td>842 (9065)</td>
</tr>
<tr>
<td></td>
<td>5.18 (17)</td>
<td>1951 (68894)</td>
<td>2006 (4422995)</td>
<td>881 (9487)</td>
</tr>
</tbody>
</table>

### Table 5: Table of displacements and wetted areas (model)

<table>
<thead>
<tr>
<th>Model</th>
<th>$T$, m (ft)</th>
<th>$\mathcal{V}$, $m^3$ (ft$^3$)</th>
<th>$\Delta$, tonnes (lb)</th>
<th>$S$, $m^2$ (ft$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.063 (0.206)</td>
<td>0.020 (0.697)</td>
<td>0.020 (43.472)</td>
<td>0.533 (5.742)</td>
</tr>
<tr>
<td></td>
<td>0.084 (0.275)</td>
<td>0.029 (1.031)</td>
<td>0.029 (64.363)</td>
<td>0.625 (6.731)</td>
</tr>
<tr>
<td></td>
<td>0.105 (0.343)</td>
<td>0.039 (1.390)</td>
<td>0.039 (86.747)</td>
<td>0.716 (7.708)</td>
</tr>
<tr>
<td></td>
<td>0.126 (0.412)</td>
<td>0.050 (1.768)</td>
<td>0.050 (110.353)</td>
<td>0.808 (8.694)</td>
</tr>
<tr>
<td></td>
<td>0.146 (0.480)</td>
<td>0.061 (2.164)</td>
<td>0.061 (135.007)</td>
<td>0.900 (9.684)</td>
</tr>
<tr>
<td></td>
<td>0.167 (0.549)</td>
<td>0.073 (2.574)</td>
<td>0.073 (160.604)</td>
<td>0.922 (10.673)</td>
</tr>
<tr>
<td></td>
<td>0.178 (0.583)</td>
<td>0.079 (2.783)</td>
<td>0.079 (173.688)</td>
<td>1.038 (11.170)</td>
</tr>
</tbody>
</table>
Surface data for the ship was reversed engineering using the table of offset in Table 6. Ordinates in are spaced 6.1 m (20 ft) apart. The lines plan can be seen in Figure 9 and Figure 10.

Table 6: Table of offsets (ship)

<table>
<thead>
<tr>
<th>Ordinate</th>
<th>Waterline 0.61 m (2 ft)</th>
<th>Waterline 1.22 m (4 ft)</th>
<th>Waterline 1.83 m (6 ft)</th>
<th>Waterline 3.66 m (12 ft)</th>
<th>Waterline 5.49 m (18 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>n/a</td>
<td>n/a</td>
<td>n/a</td>
<td>0.20 (0.67)</td>
<td>2.04 (6.70)</td>
</tr>
<tr>
<td>0</td>
<td>n/a</td>
<td>n/a</td>
<td>n/a</td>
<td>1.12 (3.69)</td>
<td>2.66 (8.71)</td>
</tr>
<tr>
<td>0.5</td>
<td>0.20 (0.67)</td>
<td>0.31 (1.01)</td>
<td>0.41 (1.34)</td>
<td>2.45 (8.04)</td>
<td>3.78 (12.40)</td>
</tr>
<tr>
<td>1</td>
<td>0.71 (2.35)</td>
<td>1.12 (3.69)</td>
<td>1.74 (5.70)</td>
<td>3.47 (11.39)</td>
<td>4.29 (14.07)</td>
</tr>
<tr>
<td>1.5</td>
<td>1.43 (4.69)</td>
<td>2.04 (6.70)</td>
<td>2.66 (8.71)</td>
<td>4.09 (13.40)</td>
<td>4.29 (14.74)</td>
</tr>
<tr>
<td>2</td>
<td>2.35 (7.71)</td>
<td>3.06 (10.05)</td>
<td>3.57 (11.73)</td>
<td>4.45 (14.61)</td>
<td>4.70 (15.41)</td>
</tr>
<tr>
<td>3</td>
<td>4.09 (13.40)</td>
<td>4.49 (14.74)</td>
<td>4.70 (15.41)</td>
<td>4.80 (15.75)</td>
<td>4.90 (16.08)</td>
</tr>
<tr>
<td>4</td>
<td>4.60 (15.08)</td>
<td>4.90 (16.08)</td>
<td>4.90 (16.08)</td>
<td>4.90 (16.08)</td>
<td>5.00 (16.42)</td>
</tr>
<tr>
<td>5</td>
<td>4.70 (15.41)</td>
<td>5.00 (16.42)</td>
<td>5.00 (16.42)</td>
<td>5.00 (16.42)</td>
<td>5.00 (16.42)</td>
</tr>
<tr>
<td>6</td>
<td>4.60 (15.08)</td>
<td>4.90 (16.08)</td>
<td>4.90 (16.08)</td>
<td>5.00 (16.42)</td>
<td>4.90 (16.08)</td>
</tr>
<tr>
<td>7</td>
<td>3.98 (13.07)</td>
<td>4.39 (14.41)</td>
<td>4.56 (14.94)</td>
<td>4.80 (15.75)</td>
<td>4.90 (16.08)</td>
</tr>
<tr>
<td>8</td>
<td>2.70 (8.85)</td>
<td>3.27 (10.72)</td>
<td>3.47 (11.39)</td>
<td>4.09 (13.40)</td>
<td>4.49 (14.74)</td>
</tr>
<tr>
<td>8.5</td>
<td>1.84 (6.03)</td>
<td>2.35 (7.71)</td>
<td>2.66 (8.71)</td>
<td>3.27 (10.72)</td>
<td>3.88 (12.73)</td>
</tr>
<tr>
<td>9</td>
<td>1.02 (3.35)</td>
<td>1.43 (4.69)</td>
<td>1.63 (5.36)</td>
<td>2.14 (7.04)</td>
<td>3.06 (10.05)</td>
</tr>
<tr>
<td>9.5</td>
<td>0.25 (0.80)</td>
<td>0.47 (1.54)</td>
<td>0.61 (2.01)</td>
<td>1.02 (3.35)</td>
<td>1.84 (6.03)</td>
</tr>
<tr>
<td>10</td>
<td>n/a</td>
<td>n/a</td>
<td>n/a</td>
<td>n/a</td>
<td>0.41 (1.34)</td>
</tr>
</tbody>
</table>
Figure 9: Plan and profile view
Figure 10: Body plan view
To create the model, the 3D data was scaled using a scaling factor of 29.14. The table of offsets for the model is shown in Table 7.

Table 7: Table of offsets (model)

<table>
<thead>
<tr>
<th>Ordinate</th>
<th>Waterline</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>20.83 mm (0.82 in)</td>
</tr>
<tr>
<td>A</td>
<td>n/a</td>
</tr>
<tr>
<td>0</td>
<td>n/a</td>
</tr>
<tr>
<td>0.5</td>
<td>7.01 (0.28)</td>
</tr>
<tr>
<td>1</td>
<td>24.53 (0.97)</td>
</tr>
<tr>
<td>1.5</td>
<td>49.06 (1.93)</td>
</tr>
<tr>
<td>2</td>
<td>80.61 (3.17)</td>
</tr>
<tr>
<td>3</td>
<td>140.18 (5.52)</td>
</tr>
<tr>
<td>4</td>
<td>157.71 (6.21)</td>
</tr>
<tr>
<td>5</td>
<td>161.21 (6.35)</td>
</tr>
<tr>
<td>7</td>
<td>136.68 (5.38)</td>
</tr>
<tr>
<td>8</td>
<td>92.52 (3.64)</td>
</tr>
<tr>
<td>8.5</td>
<td>63.08 (2.48)</td>
</tr>
<tr>
<td>9</td>
<td>35.05 (1.38)</td>
</tr>
<tr>
<td>9.5</td>
<td>8.41 (0.33)</td>
</tr>
<tr>
<td>10</td>
<td>n/a</td>
</tr>
</tbody>
</table>
Tooling for the model was created in order to produce multiple identical hulls. This allowed for the creation of a control (an unmodified model) and the experiment hull (one modified with dimples). The control model was left unchanged while the conceptual model was modified with dimples. To mimic the design of the golf ball, design decisions were based on the relationships defined in Table 1. The diameter of the dimple was designed at 7.4% the diameter of the round bilge. Since the round bilge on the ship has a radius of approximately 2.5 m (8.2 ft), the diameter of the dimple is then 0.370 m (1.214 ft). The depth of the dimple would be 11% of its own diameter. This makes the depth of the dimple 0.041 m (.133 ft). Using the scaling factor, $\lambda=29.14$, the dimple dimensions on the model scale down to 12.7 mm (0.5 in) in diameter and 1.40 mm (0.055 in) in depth.

Initially it was decided that the model should be dimpled everywhere like a golf ball, Figure 11; however, final design only has dimples aft of the parallel middle body. It was decided that the wetted area at the parallel middle body and forward should maintain smoothness so as not to increase skin friction. Considering the direction of the flow, from bow to stern, the tapered portions of the hull aft of the parallel middle body provide the potential for flow instability and separation. Dimples were placed from this point aft, Figure 12 and Figure 13. Theoretically there should be localized increases in skin friction but overall an improvement of the mean flow and reduction of pressure drag.

It should be noted that due to available construction methods, the dimple depth varied from the theoretical 1.40 mm (0.055 in). The actual dimple depths are 1.905±0.245 mm (0.075±0.010 in).
Figure 11: Initial dimple pattern

Figure 12: Model dimpling (starboard view)
Figure 13: Model dimpling (bottom view)
Chapter 5

Results

Resistance tests were performed at the University of Michigan’s Marine Hydrodynamic Laboratory. All tests conformed to the ITTC Testing and Data Analysis Methods Resistance Test (2008). Let it be noted that model “A” refers to the control, or unmodified model, and model “B” refers to the concept, or dimpled model.

Both models were tested at two displacements, 61.53 kg (135.65 lb) and 78.78 kg (173.69 lb). These displacements represent the fourteen and seventeen foot waterlines on the full size vessels. Table 8 and Table 9 are the comparative results of the resistance tests performed on the models. The values in Table 8 and Table 9 are the average resistance of the models while the carriage traveled at a constant speed. This neglects erroneous forces generated during acceleration and deceleration. Raw data for each run can be seen in the Appendix.
### Table 8: Resistance comparison 4.27 m (14 ft) waterline

<table>
<thead>
<tr>
<th>$F_n$</th>
<th>$R_{TM A}$ N (lb)</th>
<th>$R_{TM B}$ N (lb)</th>
<th>% difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.10</td>
<td>0.451 (0.101)</td>
<td>0.314 (0.071)</td>
<td>-30.39%</td>
</tr>
<tr>
<td>0.15</td>
<td>1.083 (0.243)</td>
<td>0.887 (0.199)</td>
<td>-18.11%</td>
</tr>
<tr>
<td>0.20</td>
<td>2.090 (0.470)</td>
<td>1.869 (0.420)</td>
<td>-10.58%</td>
</tr>
<tr>
<td>0.25</td>
<td>3.579 (0.804)</td>
<td>3.236 (0.727)</td>
<td>-9.59%</td>
</tr>
<tr>
<td>0.30</td>
<td>7.850 (1.764)</td>
<td>8.127 (1.826)</td>
<td>3.53%</td>
</tr>
<tr>
<td>0.35</td>
<td>12.268 (2.757)</td>
<td>12.770 (2.870)</td>
<td>4.09%</td>
</tr>
<tr>
<td>0.39</td>
<td>18.520 (4.162)</td>
<td>19.382 (4.356)</td>
<td>4.66%</td>
</tr>
<tr>
<td>0.40</td>
<td>19.986 (4.492)</td>
<td>21.191 (4.762)</td>
<td>6.03%</td>
</tr>
</tbody>
</table>

### Table 9: Resistance comparison 5.18 m (17 ft) waterline

<table>
<thead>
<tr>
<th>$F_n$</th>
<th>$R_{TM A}$ N (lb)</th>
<th>$R_{TM B}$ N (lb)</th>
<th>% difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.10</td>
<td>0.523 (0.118)</td>
<td>0.568 (0.128)</td>
<td>8.61%</td>
</tr>
<tr>
<td>0.15</td>
<td>1.247 (0.280)</td>
<td>1.271 (0.286)</td>
<td>1.95%</td>
</tr>
<tr>
<td>0.20</td>
<td>2.329 (0.523)</td>
<td>2.429 (0.546)</td>
<td>4.27%</td>
</tr>
<tr>
<td>0.25</td>
<td>4.188 (0.941)</td>
<td>4.231 (0.951)</td>
<td>1.03%</td>
</tr>
<tr>
<td>0.30</td>
<td>9.577 (2.152)</td>
<td>9.602 (2.158)</td>
<td>0.26%</td>
</tr>
</tbody>
</table>

Experimental error in the load cell is ± 0.294 N (0.066 lb). An error analysis for the Marine Hydrodynamic Laboratories was provided by the University of Michigan and is reported in the Appendix. Unfortunately, nearly all the results in Table 8 and Table 9 have overlapping error between model A and model B. If the results between model A and model B have overlapping error then the percent difference is not valid. The only results that did not have overlapping error were $F_n$=0.39 and $F_n$=0.40 in Table 8. This can be seen in Figure 14 and Figure 15.
Figure 14: Resistance comparison – 4.27 m (14 ft) waterline

Figure 15: Resistance comparison – 5.18 m (17 ft) waterline
The model resistance was then extrapolated into the resistance of the full size vessel. The correlation allowance, $C_A$, for the Marine Hydrodynamic Laboratories is $2.5 \times 10^{-4}$. Full size resistance data can be seen in Table 10, Table 11, Figure 16 and Figure 17.

**Table 10: Extrapolated resistance data – 4.27 m (14 ft) waterline**

<table>
<thead>
<tr>
<th>$Fn$</th>
<th>$R_{TS A}$ (N \text{ (lb)})</th>
<th>$R_{TS B}$ (N \text{ (lb)})</th>
<th>% difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.10</td>
<td>5.24E+03(\text{ (1.18E+03)})</td>
<td>3.82E+03(\text{ (8.59E+02)})</td>
<td>-27.08%</td>
</tr>
<tr>
<td>0.15</td>
<td>1.51E+04(\text{ (3.39E+03)})</td>
<td>1.43E+04(\text{ (3.20E+03)})</td>
<td>-5.39%</td>
</tr>
<tr>
<td>0.20</td>
<td>3.28E+04(\text{ (7.36E+03)})</td>
<td>3.40E+04(\text{ (7.64E+03)})</td>
<td>3.84%</td>
</tr>
<tr>
<td>0.25</td>
<td>6.12E+04(\text{ (1.37E+04)})</td>
<td>6.26E+04(\text{ (1.41E+04)})</td>
<td>2.27%</td>
</tr>
<tr>
<td>0.30</td>
<td>1.59E+05(\text{ (3.56E+04)})</td>
<td>1.79E+05(\text{ (4.03E+04)})</td>
<td>13.23%</td>
</tr>
<tr>
<td>0.35</td>
<td>2.59E+05(\text{ (5.81E+04)})</td>
<td>2.90E+05(\text{ (6.51E+04)})</td>
<td>11.92%</td>
</tr>
<tr>
<td>0.39</td>
<td>4.06E+05(\text{ (9.12E+04)})</td>
<td>4.50E+05(\text{ (1.01E+05)})</td>
<td>10.87%</td>
</tr>
<tr>
<td>0.40</td>
<td>4.41E+05(\text{ (9.90E+04)})</td>
<td>4.94E+05(\text{ (1.11E+05)})</td>
<td>12.17%</td>
</tr>
</tbody>
</table>

**Table 11: Extrapolated resistance data – 5.18 m (17 ft) waterline**

<table>
<thead>
<tr>
<th>$Fn$</th>
<th>$R_{TS A}$ (N \text{ (lb)})</th>
<th>$R_{TS B}$ (N \text{ (lb)})</th>
<th>% difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.10</td>
<td>6.09E+03(\text{ (1.37E+03)})</td>
<td>6.83E+03(\text{ (1.53E+03)})</td>
<td>12.07%</td>
</tr>
<tr>
<td>0.15</td>
<td>1.73E+04(\text{ (3.89E+03)})</td>
<td>1.70E+04(\text{ (3.83E+03)})</td>
<td>-1.60%</td>
</tr>
<tr>
<td>0.20</td>
<td>3.57E+04(\text{ (8.01E+03)})</td>
<td>3.67E+04(\text{ (8.25E+03)})</td>
<td>2.92%</td>
</tr>
<tr>
<td>0.25</td>
<td>7.20E+04(\text{ (1.62E+04)})</td>
<td>7.09E+04(\text{ (1.59E+04)})</td>
<td>-1.52%</td>
</tr>
<tr>
<td>0.30</td>
<td>1.96E+05(\text{ (4.40E+04)})</td>
<td>1.94E+05(\text{ (4.35E+04)})</td>
<td>-1.16%</td>
</tr>
</tbody>
</table>
Figure 16: $C_{TS}$ vs. $F_n$ – 4.27 m (14 ft) waterline

Figure 17: $C_{TS}$ vs. $F_n$ – 5.18 m (17 ft) waterline
Chapter 6

Discussion

The lack of difference between model A and model B is easily identifiable. Table 8 and Table 9 show that all results but two are non-valid results. Based on the size of the measurements taken and the magnitude of error, stochastically speaking, the recorded resistance values of each model could potentially be equal. In other words, there was effectively no difference in the resistance characteristics of model B compared to model A.

There are a number of reasons that could explain why the resistance of model B did not improve. One thought is that the dimples were incorrectly located, meaning they were placed too far forward or too far aft. If they were placed too far forward, assuming separation delay occurred, then there was an even balance between increased skin friction and decreased pressure drag. Remember that separation delay is like “reattaching” the flow. Since the flow is in contact with the surface longer, skin friction increases. In addition, dimpling the hull increases wetted surface area and makes the flow turbulent, also increasing viscous resistance. If the relation between increased skin friction and decreased pressure drag is 1:1, than the resistance characteristics of model B would be unchanged compared to model A. By placing dimples too far forward the mean flow is exposed to more dimples than
necessary, the flow is turbulent longer than necessary, and skin friction is increased as much as pressure drag is decreased.

If the dimples were placed too far aft, assuming separation occurred, than the dimples had no effect on the boundary layer. In other words, the flow separated from the hull before reaching the dimples. This is like not having dimples there at all. Under these circumstances the results of model B would be like testing model A twice. The lack of difference between the results would only prove the repeatability of the tooling that the models were created from.

The dimples could also have been designed incorrectly. Assuming the dimples were located correctly and exposed to laminar flow, than the dimple diameter and dimple depth did not create turbulence and separation delay did not occur. Without separation delay, pressure drag remains unchanged. Here the dimples may be too small and/or too shallow.

Improper dimple design could also contribute to increased skin friction. Assuming the mean flow was not “over-exposed” to dimples; an improperly designed dimple could potentially create turbulence and delay separation, but increase skin friction such that in cancels out the decrease in pressure drag. Here, the dimples may too large and/or too deep.

Another explanation for the lack of differences may be vessel selection, meaning separation naturally does not occur on this particular vessel. Here the tapered portions of the stern do not curve fast enough to create an adverse pressure gradient and separation does not occur. Pressure drag is now negligible and therefore cannot be improved upon; however, this is a weak explanation because the presence of dimples in a flow that did not separate should have logically increased skin friction. This effect would have been reflected in the results.

Lastly, experimental error could have contributed to the experiments results. Sources of error in this experiment include but are not limited to yaw, sway,
location of tow points, and the size of the models. Yaw is the rotational movement of a vessel about the $z$ or $x_3$ axis. Figure 18 demonstrates vessel motions. If the vessel yaws during the resistance test, resistance would drastically increase. To prevent yaw, the University of Michigan’s Marine Hydrodynamic Laboratory (MHL) uses a “grass hopper.” This is a device that restrains the model from yawing while maintaining freedom of motion in heave and pitch. For this particular experiment, the vessel size did not accommodate the use of the grass hopper. The models were too small for the grass hopper to reach the bow. Instead, the clamps that hold the heave staff in place were tightened such that the frictional force of the clamps would restrain the model in yaw. To ensure proper alignment of the model, before each run the centerline of the model was aligned with the centerline of the carriage using a pendulum. To verify that the model had not yawed during the run, after each run the alignment was checked using a pendulum. The thoroughness of the procedure is such that yaw most likely did not contribute to experimental error; nevertheless, since the grass hopper was not used, it may exist and therefore needed to be discussed.

Figure 18: Vessel motions (Newman, 1978)
According to Timothy Peters, the assistant director at the MHL, the rails of the carriage are similar to a roller coaster. The wheels of the carriage roll on both the top and the sides of the rails so that it can travel along the rail yet not fall off. Due to tolerance stack ups, the carriage is capable of yawing from port to starboard up to ± 1 inch during a run. Systematic errors like this one affect the results of every run. The error is consistent and therefore even if the error was significant the results are usable for comparison studies. According to Timothy Peters, experimental error at the MHL is minimal and results can be considered accurate.

The tow points of model A and model B were installed at slightly different locations. Both tow points were installed along the centerline; both were 0.098 m (0.323 ft) above the baseline; however, the tow point of Model A was 1.099 m (3.604 ft) aft of the forward perpendicular while the tow point of model B was 1.032 m (3.385 ft) aft of the forward perpendicular. Different tow point locations means different moments were generated on the models during forward motion. Potentially, the models would squat and pitch differently during their runs, altering the comparative results between model A and model B.

Indisputably, the largest source of error came from the size of the models. A seven foot model is not large enough to test properly at the MHL. As discussed, the models were too small to use with the grass hopper. More importantly, a smaller model means smaller resistance measurements. Smaller resistance measurements mean that experimental error in the measurement equipment becomes a significant portion of the measurement itself. For the 4.27 m (14 ft) waterline, experimental error in the load cell was 2.75% to 187% of the measurements and averaged 35.51%. It was not until Fn exceeded 0.35 that error became 3% or less of the measurements being taken. At the 4.27 m (14 ft) waterline, only two valid differences were measured at Fn=0.39 and Fn=0.4. At the 5.18 m (17 ft) waterline
error ranged from 6% to 112% of the measurements and averaged 40%. With error being such a large portion of the measurement, results between model A and model B would need to vary greatly for differences to be valid. As seen in Table 8 and Table 9, that is not so.
Chapter 7

Conclusions

Based on these results, it cannot be determined if the reduction of ship resistance through induced turbulent boundary layers was a success for this hull form; however, it can be concluded that it was not a failure. The hypothesis was neither proved nor disproved providing motivation for further investigation. If this topic is to be reinvestigated, it is the suggestion of the author that a more systematic approach be taken.

The primary reason for the lack of valid experimental evidence is the small size of the models. Before retesting this hypothesis, larger models should be created. This would increase the magnitude of the resistance values making experimental error a smaller portion of the value measured. Here smaller percent differences between baseline and concept models would be considered valid. Increasing model size to 12 foot would triple the resistance values measured by the load cell. For future research, model size should be increased to a minimum of 3.66 m (12 ft).

Before reinvestigating this topic, flat plate testing should be performed to better understand the effects of dimples. A flat plate in the presence of an adverse pressure gradient would act as the baseline. By modifying the size, depth, and pattern of the dimples on the plate, it could be better understood how different...
dimples affect separation with respect to different pressure gradients. The information from these experiments would allow for educated decisions when it comes time to modify one of the model boats.

When a vessel is selected a wake survey should be performed on the control model to verify that the hull form is a candidate for separation delay. A wake survey would demonstrate that separation naturally occurs on the selected vessel and estimate the proportion of total drag due to form drag. This would act as a phase gate in the experimental process. Either the selected vessel would meet the separation criteria and move to the next step or it will reveal the hull to be unworthy of study and a new vessel can be selected.

Once a ship passes the wake survey, flow visualization should be used to determine separation points. Determining separations points allows for the proper location of dimples based on information gathered from flat plate testing. Flow visualization can be accomplished experimentally or be numerically simulated. This ensures the dimples are not placed too far forward or too far aft, optimizing the relationship between increased skin friction and decreased pressure drag.

The adverse pressure gradients that are causing separation should also be measured. The measured pressure gradients on the model can be correlated to the adverse pressure gradients in the flat plate testing. The size, depth, and pattern of the dimples can be designed from the information in the flat plate study that optimized separation delay.

Resistance tests on both models would then be compared as done in this thesis. In addition, after dimpling, the wake survey and flow visualization should be repeated on the concept model. By repeating the tests, it can be established that separation delay did or did not occur. Knowledge of this information would aid greatly when making conclusions.
References


Appendix

MHL Error Analysis

<table>
<thead>
<tr>
<th>Error Type</th>
<th>Error (% of FS)</th>
<th>Full Scale, N (lb)</th>
<th>Error, N (lb)</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hysteresis</td>
<td>0.08%</td>
<td>222.49 (50)</td>
<td>0.178 (0.04)</td>
<td></td>
</tr>
<tr>
<td>Linearity</td>
<td>0.10%</td>
<td></td>
<td>0.223 (0.05)</td>
<td></td>
</tr>
<tr>
<td>Non-Repeatability</td>
<td>0.03%</td>
<td></td>
<td>0.067 (0.015)</td>
<td></td>
</tr>
</tbody>
</table>

Load Cell Instrument Error = 0.293 (0.0658) N (lb)

Carriage Speed Sensor Error (no signal conditioning)

<table>
<thead>
<tr>
<th>Error Type</th>
<th>Error (% of FS voltage)</th>
<th>(FSV 0-10volts)</th>
<th>Error (volts)</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Linearity</td>
<td>0.10%</td>
<td>10</td>
<td>0.01</td>
<td>m/s (ft/s)</td>
</tr>
</tbody>
</table>

Carriage Speed Instrument Error = 0.008 (0.025) m/s (ft/s)

Calibration Errors

<table>
<thead>
<tr>
<th>Instrument</th>
<th>Calibration Method</th>
<th>Error (N (lb))</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load Cell</td>
<td>Calibrated using precision weights verified on precision digital scale</td>
<td>0.002 (0.0005)</td>
<td>N (lb)</td>
</tr>
<tr>
<td>Speed Sensor</td>
<td>Calibrated using Fluke Multimeter w/ Frequency accuracy 0.1% FS voltage</td>
<td>0.008 (0.025)</td>
<td>m/s (ft/s)</td>
</tr>
</tbody>
</table>

Amplifier Error

<table>
<thead>
<tr>
<th>Error Type</th>
<th>Error (% of FS voltage)</th>
<th>(FSV ±10volts)</th>
<th>Error (volts)</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Linearity</td>
<td>0.02%</td>
<td>20</td>
<td>0.001</td>
<td></td>
</tr>
</tbody>
</table>

Amplifier Error for Load Cell = 0.022 (0.0050) N (lb)

Data Acquisition Card Error - Quantizing Error

<table>
<thead>
<tr>
<th>Error Type</th>
<th>Error (bits)</th>
<th>(FSV ±10volts)</th>
<th>Error (volts)</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Linearity</td>
<td>3.052E-05</td>
<td>20</td>
<td>6.104E-04</td>
<td></td>
</tr>
<tr>
<td>Resolution</td>
<td>1.526E-05</td>
<td>20</td>
<td>3.052E-04</td>
<td></td>
</tr>
</tbody>
</table>

DAQ Card Error for Load Cell = 0.015 (0.0034) N (lb)

Propagating Error Totals

<table>
<thead>
<tr>
<th>Method</th>
<th>Error (N (lb))</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Root-Sum-Squares (RSS) Method</td>
<td>0.294 (0.0660)</td>
<td>± N (lb)</td>
</tr>
<tr>
<td>Assume Errors are Independent</td>
<td>0.011 (0.0354)</td>
<td>± m/s (ft/s)</td>
</tr>
</tbody>
</table>
Raw Data

Resistance Data- Model A - 4.27 m Waterline

Run 0001

Resistance Data- Model A - 4.27 m Waterline

Run 0002
Resistance Data - Model A - 4.27 m Waterline

Run 0005

Time (s)

Speed (m/s)  Drag (N)

Run 0006

Time (s)

Speed (m/s)  Drag (N)
Resistance Data - Model A - 4.27 m Waterline

Run 0007

Run 0008
Resistance Data - Model A - 4.27 m Waterline

Run 0009

Run 0010

Resistance Data - Model A - 4.27 m Waterline
Resistance Data - Model A - 4.27 m Waterline

Run 0011

Run 0012
Resistance - Model B - 4.27 m Waterline

Run 0001

Run 0002
### Sample Calculations

Table 12: Model-ship correlation calculations

<table>
<thead>
<tr>
<th>$V_{\text{carriage}}, \text{m/s (ft/s)}$</th>
<th>$R_{\text{TM}A}, \text{N (lb)}$</th>
<th>$Fn$</th>
<th>$C_{\text{TM}}$</th>
<th>$Re_M$</th>
<th>$C_{\text{FM}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.448 (1.471)</td>
<td>0.523 (0.118)</td>
<td>0.10</td>
<td>5.017E-03</td>
<td>8.64E+05</td>
<td>4.840E-03</td>
</tr>
<tr>
<td>0.673 (2.207)</td>
<td>1.247 (0.280)</td>
<td>0.15</td>
<td>5.316E-03</td>
<td>1.30E+06</td>
<td>4.434E-03</td>
</tr>
<tr>
<td>0.897 (2.944)</td>
<td>2.329 (0.523)</td>
<td>0.20</td>
<td>5.577E-03</td>
<td>1.73E+06</td>
<td>4.176E-03</td>
</tr>
<tr>
<td>1.120 (3.675)</td>
<td>4.188 (0.941)</td>
<td>0.25</td>
<td>6.437E-03</td>
<td>2.16E+06</td>
<td>3.992E-03</td>
</tr>
<tr>
<td>1.347 (4.420)</td>
<td>9.577 (2.152)</td>
<td>0.30</td>
<td>1.018E-02</td>
<td>2.60E+06</td>
<td>3.849E-03</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$V_{\text{carriage}}, \text{m/s (ft/s)}$</th>
<th>$R_{\text{TM}B}, \text{N (lb)}$</th>
<th>$Fn$</th>
<th>$C_{\text{TM}}$</th>
<th>$Re_M$</th>
<th>$C_{\text{FM}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.477 (1.468)</td>
<td>0.568 (0.128)</td>
<td>0.10</td>
<td>5.470E-03</td>
<td>8.63E+05</td>
<td>4.842E-03</td>
</tr>
<tr>
<td>0.673 (2.208)</td>
<td>1.271 (0.286)</td>
<td>0.15</td>
<td>5.410E-03</td>
<td>1.30E+06</td>
<td>4.433E-03</td>
</tr>
<tr>
<td>0.898 (2.948)</td>
<td>2.429 (0.546)</td>
<td>0.20</td>
<td>5.803E-03</td>
<td>1.73E+06</td>
<td>4.175E-03</td>
</tr>
<tr>
<td>1.121 (3.679)</td>
<td>4.231 (0.951)</td>
<td>0.25</td>
<td>6.488E-03</td>
<td>2.16E+06</td>
<td>3.991E-03</td>
</tr>
<tr>
<td>1.347 (4.419)</td>
<td>9.602 (2.158)</td>
<td>0.30</td>
<td>1.021E-02</td>
<td>2.60E+06</td>
<td>3.849E-03</td>
</tr>
</tbody>
</table>
Figure 19: Model A form factor determination

\[ y = 0.7601x + 1.0363 \]
Figure 20: Model B form factor determination

Model B - 5.18 m (17 ft) Waterline

\[ y = 0.7318x + 1.0934 \]
### Table 13: Full scale extrapolation from model scale

#### Model A – 5.18 m (17 ft) Waterline

<table>
<thead>
<tr>
<th>$C_{TM}$</th>
<th>$C_{FM}$</th>
<th>$(1+k)$</th>
<th>$C_R$</th>
<th>$V_{ship}$ (ft/s)</th>
<th>$Re_S$</th>
<th>$C_{FS}$</th>
<th>$C_{TS}$</th>
<th>$R_{TS}$, $N$ (lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.017E-03</td>
<td>4.840E-03</td>
<td>1.0363</td>
<td>1.48E-06</td>
<td>2.42 (7.94)</td>
<td>1.43E+08</td>
<td>1.980E-03</td>
<td>2.30E-03</td>
<td>6.09E+03 (1.37E+03)</td>
</tr>
<tr>
<td>5.316E-03</td>
<td>4.434E-03</td>
<td>1.0363</td>
<td>7.21E-04</td>
<td>3.63 (11.91)</td>
<td>2.14E+08</td>
<td>1.871E-03</td>
<td>2.91E-03</td>
<td>1.73E+04 (3.89E+03)</td>
</tr>
<tr>
<td>5.577E-03</td>
<td>4.176E-03</td>
<td>1.0363</td>
<td>1.25E-03</td>
<td>4.84 (15.89)</td>
<td>2.86E+08</td>
<td>1.799E-03</td>
<td>3.36E-03</td>
<td>3.57E+04 (8.01E+03)</td>
</tr>
<tr>
<td>6.437E-03</td>
<td>3.992E-03</td>
<td>1.0363</td>
<td>2.30E-03</td>
<td>6.05 (19.84)</td>
<td>3.57E+08</td>
<td>1.747E-03</td>
<td>4.36E-03</td>
<td>7.20E+04 (1.62E+04)</td>
</tr>
<tr>
<td>1.018E-02</td>
<td>3.849E-03</td>
<td>1.0363</td>
<td>6.19E-03</td>
<td>7.27 (23.86)</td>
<td>4.29E+08</td>
<td>1.705E-03</td>
<td>8.20E-03</td>
<td>1.96E+05 (4.40E+04)</td>
</tr>
</tbody>
</table>

#### Model B – 5.18 m (17 ft) Waterline

<table>
<thead>
<tr>
<th>$C_{TM}$</th>
<th>$C_{FM}$</th>
<th>$(1+k)$</th>
<th>$C_R$</th>
<th>$V_{ship}$ (ft/s)</th>
<th>$Re_S$</th>
<th>$C_{FS}$</th>
<th>$C_{TS}$</th>
<th>$R_{TS}$ (lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.470E-03</td>
<td>4.842E-03</td>
<td>1.0934</td>
<td>1.76E-04</td>
<td>2.42 (7.93)</td>
<td>1.42E+08</td>
<td>1.981E-03</td>
<td>2.59E-03</td>
<td>6.83E+03 (1.53E+03)</td>
</tr>
<tr>
<td>5.410E-03</td>
<td>4.433E-03</td>
<td>1.0934</td>
<td>5.63E-04</td>
<td>3.63 (11.92)</td>
<td>2.14E+08</td>
<td>1.871E-03</td>
<td>2.86E-03</td>
<td>1.70E+04 (3.83E+03)</td>
</tr>
<tr>
<td>5.803E-03</td>
<td>4.175E-03</td>
<td>1.0934</td>
<td>1.24E-03</td>
<td>4.85 (15.91)</td>
<td>2.86E+08</td>
<td>1.799E-03</td>
<td>3.46E-03</td>
<td>3.67E+04 (8.25E+03)</td>
</tr>
<tr>
<td>6.488E-03</td>
<td>3.991E-03</td>
<td>1.0934</td>
<td>2.12E-03</td>
<td>6.05 (19.86)</td>
<td>3.57E+08</td>
<td>1.747E-03</td>
<td>4.28E-03</td>
<td>7.09E+04 (1.59E+04)</td>
</tr>
<tr>
<td>1.021E-02</td>
<td>3.849E-03</td>
<td>1.0934</td>
<td>6.00E-03</td>
<td>7.27 (23.86)</td>
<td>4.29E+08</td>
<td>1.705E-03</td>
<td>8.11E-03</td>
<td>1.94E+05 (4.35E+04)</td>
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</table>